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Conformable fixture systems with flexure pins for improved workpiece holding

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Abstract

The design of fixture systems for low stiffness freeform components is challenging. The tendency of thin components to deform under clamping, and vibrate during machining, are long-standing and persistent industrial problems. Excessive clamping-induced workpiece deformation can lead to poor dimensional accuracy, and weakening of the finished component due to residual stresses. Machining-induced vibration can affect the accuracy and surface finish of components, and also cause increased wear on the cutting tool, or even damage to the spindle bearings of the machine tool. Satisfactory fixturing of low stiffness components with high geometric complexity using conventional methods can be difficult and costly, due to the large numbers of clamps and/or dampers required to fully support and stabilise the workpiece. Fixturing of small components is particularly challenging, due to the lack of space for clamping and damping features. Furthermore, typical fixture systems cannot easily accommodate any manufacturing variability of blanks. These issues are particularly relevant to the machining of legacy components, where it is typically necessary to develop appropriate fixture solutions at low cost, and with short lead times.

This work presents an innovative concept of additive manufactured fixture, based on tuneable deformable supporting structures known as flexures, to improve the holding of delicate workpiece geometries during machining. The concept takes full advantage of *design for additive manufacture* to enable near full-face contact between the workpiece and fixture, which provides a conformable supporting surface for freeform components that improves fixture performance compared to existing designs. Arrays of deformable flexures occupy the space between locating features, and engage with the component when clamping is applied. They are shown to have minimal effect on the frequency response of the workpiece-fixture system relative to conventional fixtures, meaning an existing fixture can be replaced with a flexure-based solution without the need to alter any tooling or machining parameters in the manufacturing process.

Initially, the dynamic properties of flexures are studied to investigate the effect of flexure geometry on frequency response, with finite element models developed to predict flexure eigenfrequencies. Following this, a simplified clamping setup, comprising a row of flexures to simulate a thin section through a fixture, is used to investigate the ability of fixture conformability to reduce workpiece deformation, with comparisons made against a conventional fixture with a rigid clamping surface. Finally, the findings from these investigations are used to develop additively manufactured machining fixture blocks, which are used to hold a simple cantilever workpiece during end milling, and to investigate the ability of flexures to damp workpiece vibrations during machining.

Finite element models have been developed and refined, and shown to be able to consistently predict the eigenfrequencies of flexures, both individually and in large arrays within a fixture, with an error below 5%. When studying fixture conformability, the flexure configuration was found to reduce the clamping force required to hold aerofoil-type workpieces with geometric errors by up to 60% compared to a rigid clamping surface. This was observed in simulations and experiments to be due to a mechanism whereby flexures deform to accommodate geometric errors in the workpiece, while a rigid clamping surface will deform a distorted workpiece until it adopts the profile of the fixture, which therefore reduces clamping-induced deformation by more than 40% in some configurations. In machining, it is found that flexure resonance can be utilised to reduce workpiece vibration, by selecting milling parameters that excite the workpiece at the first eigenfrequency of the flexures. In good agreement with models, this frequency-specific damping effect is found to reduce vibration of the workpiece by 25% compared to n-2-1 fixtures, and 34% compared to rigid vice-type fixtures.

This thesis provides an understanding of the behaviour of flexures, and shows how this knowledge can be exploited in the design of fixtures to reduce the deformation and vibration experienced by delicate workpiece geometries.

Publications originating from the thesis

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Nomenclature

AM	additive manufacture
CAD	computer-aided design
CAFD	computer-aided fixture design
CAM	computer-aided manufacture
CMM	coordinate measuring machine
DIC	digital image correlation
DOF	degree of freedom
FDM	fused deposition modelling
FE	finite element
FFT	fast Fourier transform
FRF	frequency response function
RMS	root mean square
SLA	stereolithography
SLM	selective laser melting
SLS	selective laser sintering
UTM	universal testing machine

1 Introduction

1.1 Background

In 2017, air passenger numbers exceeded four billion for the first time. The number of annual passenger journeys has doubled since 2005, and is projected to grow at a rate of 3.6% annually over the next 20 years [1]. Recent increases in the price of jet fuel mean that aircraft fuel efficiency is becoming an increasingly important factor in airline running costs. The average age of a commercial airliner at retirement from service is currently 27 years, however there is an industry trend towards shorter aircraft working lifetimes due to the latest models offering considerably reduced running costs compared to the oldest aircraft still in service [2].

Another factor in the tendency for younger fleets is the shift away from outright ownership of aircraft in favour of operating leases, in which airlines pay manufacturers predetermined ongoing costs for each flight they operate, instead of the one-off upfront cost of the aircraft. This mimics the model that has existed for Rolls-Royce engines since the 1960s, in which commercial airliner engines are purchased on a *power-by-the-hour* basis. This provides an engine, complete with maintenance and replacement service, on a fixed cost per flying hour basis, thereby eliminating the financial risk associated with unscheduled repairs [3].

The vast range of engines supported by Rolls-Royce, each containing thousands of unique components, necessitates an extensive catalogue of spare parts, along with their associated tooling. In addition to ubiquitous modern engines, support remains in place for legacy engines, enabling maintenance and repair of older aircraft. This requires fixtures to be held for a large library of components, many of which are seldom required. By operating a global network of maintenance centres, it becomes increasingly impractical to store ever-increasing stocks of legacy tooling in each location. It is time consuming to manufacture a bespoke fixture by conventional methods, and works out to be cost-ineffective if the

fixture is only used a few times; therefore a potential alternative is a shift to additively manufactured fixtures produced on demand.

A switch to additive manufacture of legacy tooling would enable fast-response production of a comprehensive range of bespoke tooling and fixtures, without sacrificing on the accuracy required for precision components. It also presents an opportunity for the benefits of additive techniques to be exploited to improve upon existing fixture performance. Ceasing to produce and store large inventories of tooling is likely to provide significant cost savings, since it removes the need to pre-emptively manufacture multiple sets of tooling and ship them to facilities around the world, and the use of additive technologies means the speed at which new fixtures can be prepared is likely to improve.

A typical turbine or compressor blade features a thin delicate aerofoil geometry, such as the example in Figure 1-1, and undergoes many stages of manufacture (with considerable value added) during production in order to achieve the levels of accuracy, repeatability, and performance required from aerospace components. The cost of reworking or scrapping components is therefore significant, and innovations are continuously sought in order to increase the proportion of components manufactured ‘right first time’. A typical blade will have its near net shape realised by casting or drop forging, requiring subsequent machined finishing processes in order to reach the final geometry, such as grinding of *fir tree* mounting profiles, and milling of critical surfaces and features. Producing the necessary finished geometries requires sophisticated tooling and fixturing solutions, and additive manufacture of fixtures has the potential to improve component yields through improved workpiece support and damping of vibrations.

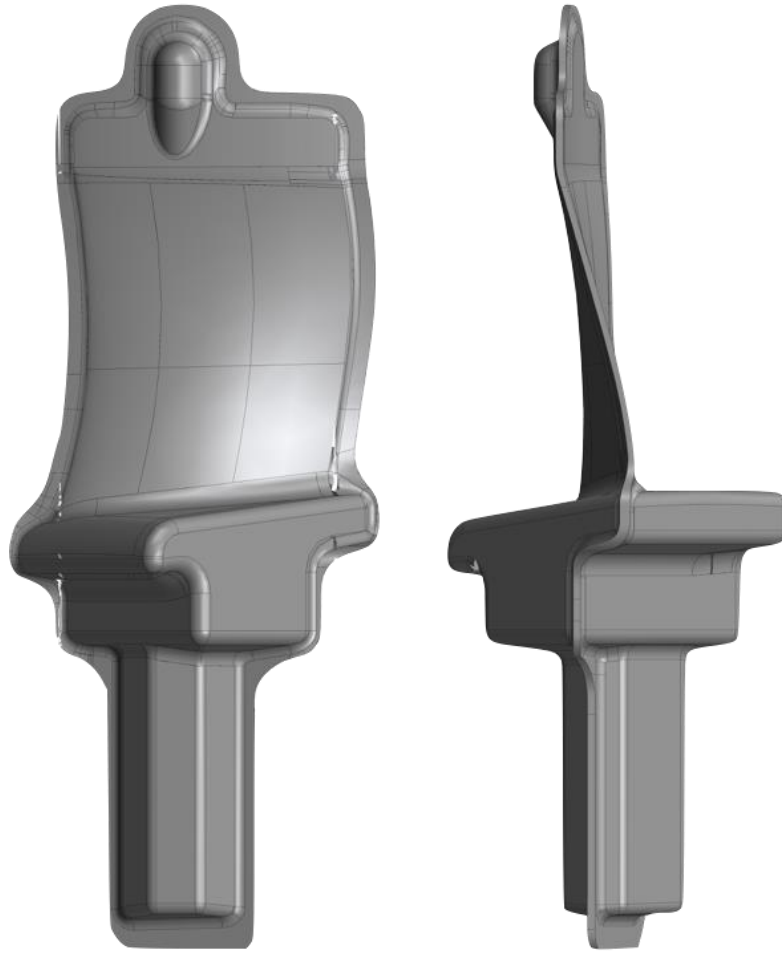


Figure 1-1. Example blade before machining, showing a thin freeform aerofoil geometry.

Issues may arise when it is necessary for components to be held on complex geometry and/or low stiffness surfaces. Conventional fixtures based on domed locators that are currently employed for machining of aerofoils can lead to inaccurate workpiece orientation due to the very small contact area between the workpiece and fixture. Additionally, the transfer of large holding forces through the reduced surfaces of locator pins increases the likelihood of surface damage in thin components.

The principles of the n-2-1 fixturing methodology are based extensively upon the assumption that components to be fixtured are rigid, i.e. that location of a component surface using point or line contacts is sufficient to hold that component rigidly. However, it is clear that this assumption does not hold true for thin components that are prone to deformation under clamping loads. Despite this, n-2-1 fixturing is widely employed in the machining of aerofoil

components. The application of high machining loads can therefore lead to workpiece vibration if thin components are inadequately supported between locators. In such cases, careful reduction of cutting parameters is necessary to ensure stable machining and achieve accurate surfaces with good surface finish. This restriction of machining speed is a sub-optimal approach to achieving the required features, since it slows down production, and does not address the issue of clamping-induced component deformation. Mounting an external damping device to a vibration-susceptible workpiece has the potential to allow faster processing, however this may introduce further issues, e.g. restriction of tool access.

Rigid vice-style fixtures, which provide enhanced support to workpiece surfaces through full-face contact, may appear to offer a solution to vibration caused by aggressive machining. However, any benefit to the finished component brought about through reduced vibration could potentially be offset by errors introduced due to workpiece deformation if there is any mismatch between the surface profiles of the workpiece and fixture. This is because any deviation in the workpiece from its expected geometry may necessitate excessive deformation in order to achieve holding in a rigid fixture, or even prevent accurate workpiece location entirely.

Fixturing using discrete locating elements, as opposed to continuous contact surfaces, therefore remains preferable for components subject to significant variability, since it reduces the potential for component-wide deformation, despite the greater susceptibility to localised deformation at clamping points. However, there exists a clear opportunity for a fixture that is capable of supporting the workpiece in the style of a rigid fixture, but is also able to accommodate workpieces with geometric errors and variabilities.

1.2 Problem definition

Fixturing thin and delicate components presents numerous challenges to industry. Components of low stiffness can be difficult to hold due to their inherent susceptibility to deformation in response to clamping loads, and

processing a workpiece in a deformed state leads to spring-back when it is released from the fixture, resulting in geometric errors in the finished component. Low stiffness geometries are prone to vibration during machining, which results in poor surface finish.

All manufacturing processes are inherently subject to some degree of variability, meaning that every component produced will exhibit small deviations from the nominal geometry, however certain near net shape processes (e.g. casting and composite layup) are susceptible to large geometric variations, especially for thin component geometries. This presents a challenge when a machining blank that has been manufactured using a high-variability process must be fixtured for subsequent stages of manufacture. Conventional fixture systems do not compensate for any variations from the nominal workpiece geometry, and therefore components with errors are at risk of being deformed during clamping so that they adopt the profile of the fixture.

Additionally, if a workpiece vibrates during machining, it may not be possible in all circumstances to alleviate vibration through alteration of machine parameters, e.g. due to the constraints of the machine, which means other methods of reducing vibration must be sought. A common, but rudimentary, technique for controlling workpiece vibration is to increase clamping loads in order to physically inhibit displacement at the antinodes of vibration, however this has the potential to induce unacceptable levels of deformation in low stiffness components. Therefore, external damping solutions are commonly employed in order to attenuate vibration, typically in the form of energy-harvesting devices based on springs or magnets, or tuned mass dampers. These damping devices are usually ‘off the shelf’ solutions, and therefore are not bespoke to the specific workpiece geometry. For some workpiece geometries, e.g. smaller components, there may be a very limited selection of suitable damping solutions available, due to the difficulty of mounting a device in proximity to affected regions of the component, or the positioning of a damping device restricting tool access. Additionally, a commercial broadband damping device may not be capable of providing sufficient damping at the desired frequency. In either case, a bespoke damping solution may be required.

Additive manufacture shows promise for the quick and cost-effective manufacture of fixture systems for complex component geometries, however very few examples of additive fixtures exist in the literature. While there is clear potential for fixture systems to benefit from the inclusion of features that allow accommodation of workpiece geometric variations, and the integration of workpiece-specific damping, the feasibility and efficacy of such a fixture system has never been proven.

This research is directed to understand and explain the scientific issues in the development of additively manufactured fixtures for thin, low stiffness components that are subject to geometric variability, in order to provide improved holding capability, and reduce workpiece vibration in machining.

Initially, research was undertaken to understand the how the geometry of deformable supporting elements affects their frequency and buckling responses. This knowledge was subsequently applied in a novel concept fixture featuring deformable supports that conform to workpiece geometries in order to accommodate geometric variations in fixtured components. Finally, the performance of an additively manufactured variant of this fixturing concept was investigated for machining applications, holding a low-stiffness cantilever workpiece during milling in order to verify their suitability for machining applications, and to evaluate the effectiveness of damping via the deformable supports. This thesis presents the development of novel machining fixtures that take full advantage of the capabilities of additive manufacture, detailing a rigorous methodology for the design and performance evaluation of the fixture concept, in order to provide informed recommendations for the production of fixtures that offer improved capability for processing low stiffness components.

1.3 Aims and objectives

1.3.1 Aim of the research

The aim of this work is to model, design, and demonstrate a novel concept of conformable fixturing system that features arrays of tuneable deformable

supports, hereafter referred to as flexures. Flexures are simple cantilever-like geometries whose resonant frequencies and buckling properties are altered by varying their cross section and height. They offer improved support to complex geometry workpieces by occupying the voids between conventional locating features, and couple with the workpiece to provide a large contact area when clamping is applied.

By fully exploiting the capabilities of additive manufacture, the presented fixture concept aims to improve upon the performance of existing fixtures manufactured by conventional processes, through reduction of clamping deformation and machining vibration in the workpiece. The proposed concept takes advantage of computer integrated manufacture techniques for the rapid design and manufacture of fixtures that perform optimally for the required application, thus addressing key challenges associated with processing delicate workpiece geometries.

1.3.2 Objectives of the research

1. Conceptual design of novel fixture systems that feature numerous supporting flexures that conform to the complex surfaces of freeform components, to improve holding of low-stiffness workpieces by achieving near full-face contact between the fixture and workpiece. The flexures feature tuneable geometries, allowing their eigenfrequencies to be shifted so they act as tuned dampers to dissipate vibration away from the workpiece, without adversely affecting the dynamics of the machining process.
2. Understand the variables that influence the frequency response of a structure, to facilitate the design of flexures that are tuned to specific frequencies for use in fixtures. Investigate the dynamic properties of various flexure geometries, individually and in arrays, when subjected to compressive loading, using a testing platform with removable flexure pegs in conjunction with finite element (FE) models to predict flexure response.
3. Develop an experiment to compare the clamping behaviour of a freeform workpiece in conventional and conformable fixtures, evaluating the ability of deformable flexures to reduce clamping-induced deformation in a workpiece with pre-existing deviations from its nominal geometry. Create

FE simulations of clamping, and validate the model experimentally using a simplified fixture prototype, to measure deformation experienced by a workpiece during clamping, and compare the performance of conformable fixtures with a conventional rigid fixture.

4. Design a machining experiment that involves end milling a low-stiffness cantilever workpiece held in an additively manufactured fixture, comparing the level of vibration experienced by the workpiece in a flexure-based fixture with values from conventional fixtures. Develop an FE model to simulate the machining response of the workpiece and fixture. Validate the model experimentally to determine the relative performance of flexure and conventional fixtures, and prove the viability of additive fixtures for machining applications.

1.4 Structure of the thesis

Following this introductory chapter, the research in this thesis begins with **Chapter 2**, a detailed literature review covering fields relevant to this work, including: basic fixturing requirements and elements of fixture systems; machining vibrations, chatter, and process stability; passive damping, its mechanisms and applications; simulation and finite element modelling of dynamic systems, with a focus on machining processes; and additive manufacture and its applicability to damping and fixturing.

Chapter 3 details the methodologies and equipment required for the experimental aspects of the thesis. Included is the hardware and software used to study frequency response, conformability, and machining performance of the fixture concept. It details experimental setups, including sensors, data acquisition, and signal processing, along with their working principles. Also included is the methodology for finite element models, which were used concurrently to aid the experimental work throughout this research.

Chapter 4 presents the novel fixture concept investigated this research. It addresses the limitations of conventional fixture systems for challenging workpiece geometries, introducing the case for additively manufactured fixtures to improve holding and machining performance. Detailed is the process for

controlling and tuning properties of the flexure geometry, and how this can be used to improve fixture performance during clamping and machining. Discussed are natural frequencies, how they are defined by component geometry, and how they are measured experimentally. Also described is a testing platform with removable flexure pegs that is used to develop and refine an FE model for prediction of flexure frequency response.

Chapter 5 investigates the capability of the fixture concept to accommodate workpieces with errors from the nominal geometry. With the aid of FE models, simplified 2D fixtures are designed for holding a sample blade-like component, allowing comparisons between a conventional rigid fixture, and a conformable prototype. The clamping force required to hold components is measured, along with the deformation they experience at fixed clamping loads.

Chapter 6 describes a machining proof of concept to test the suitability of an additive fixture for milling applications, in which the vibration damping capability of flexures is also investigated. Machining stability analyses and FE models are used to identify optimal tooling, and the machining parameters of significance to the fixture system. The level of vibration experienced by the workpiece during machining is measured in three different fixture combinations (one novel flexure-based, and two conventional fixtures) at a selection of machining speeds, in order to investigate the frequency-specific damping effect.

Chapter 7 summarises the benefits of flexures for conformability and machining performance, and discusses additional capabilities of the fixture concept. The conclusions of the thesis are presented, including the novel contributions and key findings of this research, and potential opportunities for further work are identified.

1.5 Highlights and significant contributions of the thesis

This thesis reveals many key findings that facilitate the manufacture of fixtures with improved performance in workpiece holding and machining. From the results obtained, the following highlights have been identified:

1. The frequency response of a deformable support structure (flexure) can be tuned by altering its geometry, so that it resonates at a desired frequency when clamped against a workpiece in the fixture. Using FE models, it was shown that the frequency response of flexures in an additively manufactured fixture can be predicted with a high degree of confidence.
2. Due to deformable flexure supports, the clamping load required to achieve full holding of a workpiece is found to be lower with the novel conformable fixture than in a conventional rigid fixture. This is a significant finding for the holding of delicate components that are unable to withstand large clamping loads.
3. When there is a mismatch between the surfaces of a fixture and the workpiece to be fixtured (due to errors in the workpiece geometry), workpiece deformation during clamping is greatly reduced with the novel conformable fixture compared to a conventional rigid fixture. It is observed that the conformable fixture is capable of adopting the shape of the fixtured workpiece, rather than deforming the workpiece until it adopts the shape of the fixture. This finding has ramifications for the fixturing of components which are inherently susceptible to large geometric variability.
4. Additive manufacture has been demonstrated to be suitable for the manufacture of machining fixtures, with printed fixtures shown to successfully withstand machining loads, maintaining holding and location, throughout milling of slots in a low stiffness cantilever workpiece. This is a major finding in the field of fixturing, since it allows the unique capabilities of additive technologies, along with their cost and speed benefits, to be exploited in fixtures for complex applications.
5. Integration of tuned flexures within the fixture has been shown to reduce workpiece vibration during machining when flexures are ‘activated’ by matching cutting excitation to the major flexure resonant frequency. This method of integrating damping directly into a fixture is a significant finding, which may enable machining of components that are not compatible with conventional dampers, and therefore cannot currently be machined due to excessive vibration.

2 Literature review

2.1 Introduction

Machining a thin-walled component of high quality, with good dimensional accuracy, surface finish, and repeatability requires consideration and optimisation of the entire manufacturing process. This starts with a *design for manufacture* approach to determining the features and geometry of the component. Next, it requires fixturing that is suitable for the nature of the component and process, granting access to the surfaces to be machined, while providing accurate location and secure holding. In machining, the tool and parameters must be selected so that the process is dynamically stable, taking into account factors such as depth of cut, and limitations of the material – this is of particular importance in milling, where cutter interactions are discrete and periodic, and therefore more prone to causing chatter. Figure 2-1 illustrates the factors that must be considered in the process of developing a fixture that meets all of its requirements.

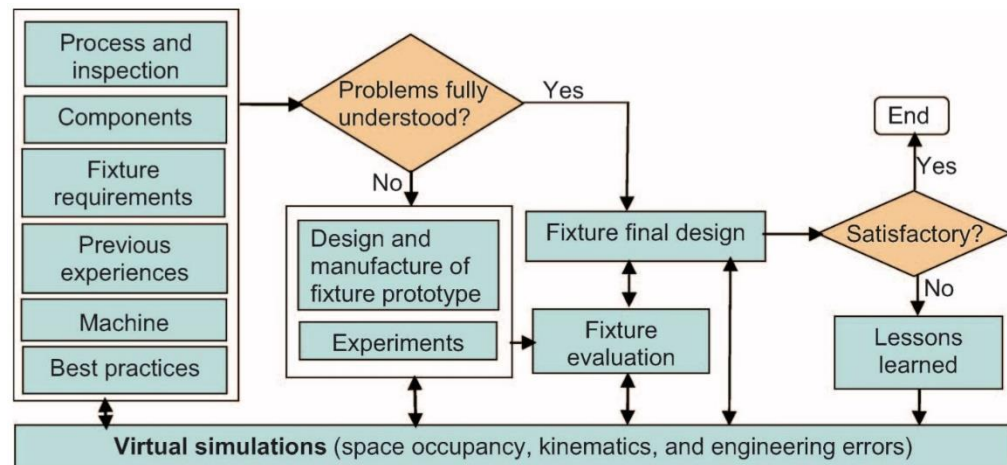


Figure 2-1. Flow diagram of fixture development process [4].

This chapter contains a literature review that includes the state of the art in fixturing, vibration, milling, additive manufacturing, and damping. It covers fields relevant to the design of a machining fixture solution for improved holding of low stiffness components.

2.2 Fixturing for thin-walled and low stiffness workpieces

This section presents an overview of recent developments in general fixturing capability and performance, especially those enabled or improved by modelling and other computer-aided manufacture (CAM) techniques, with an emphasis on thin-walled and flexible components. It then examines fixturing solutions to accommodate workpieces that are subject to geometric variations or residual stresses due to previous stages in manufacturing, and finally presents a selection of fixture systems and techniques with notable innovations for specific applications.

2.2.1 Optimisation of fixture performance, and computer-aided fixture design

Good fixturing is essential to the quality and accuracy of a machined component, and an inadequate fixture can serve as the weak link in a production system, introducing errors that limit the accuracy of an otherwise well-designed machining operation. As a result, there has been considerable research interest in improving fixture quality, concerning optimal position of locating and clamping elements, and their influence on the static and dynamic behaviour of the workpiece. Reliable prediction of the performance of a fixture without the need for extensive testing and prototyping saves time and cost, therefore modelling of fixture performance, and integration with computer-aided fixture design (CAFD) methods that automate the design of fixtures to various extents, are of great interest to industry.

Li and Melkote [5] studied the optimisation of fixture element layout to minimise location errors due to localised elastic deformation at fixturing points, which caused rigid body displacements of the workpiece. A MATLAB-based program optimised the location of clamps and locators in a simple component with a 4-2-1 fixturing layout, shown in Figure 2-2, applying best practices such as spreading out locators for greater locational stability, and positioning clamps so that they were centrally aligned with opposing locators for improved

clamping stability. An FE analysis of the component comparing the pre- and post-optimisation fixture layout revealed significantly more uniform force distribution on the locators, and improved the flatness of the top surface, reducing the deflection gradient by 80%.

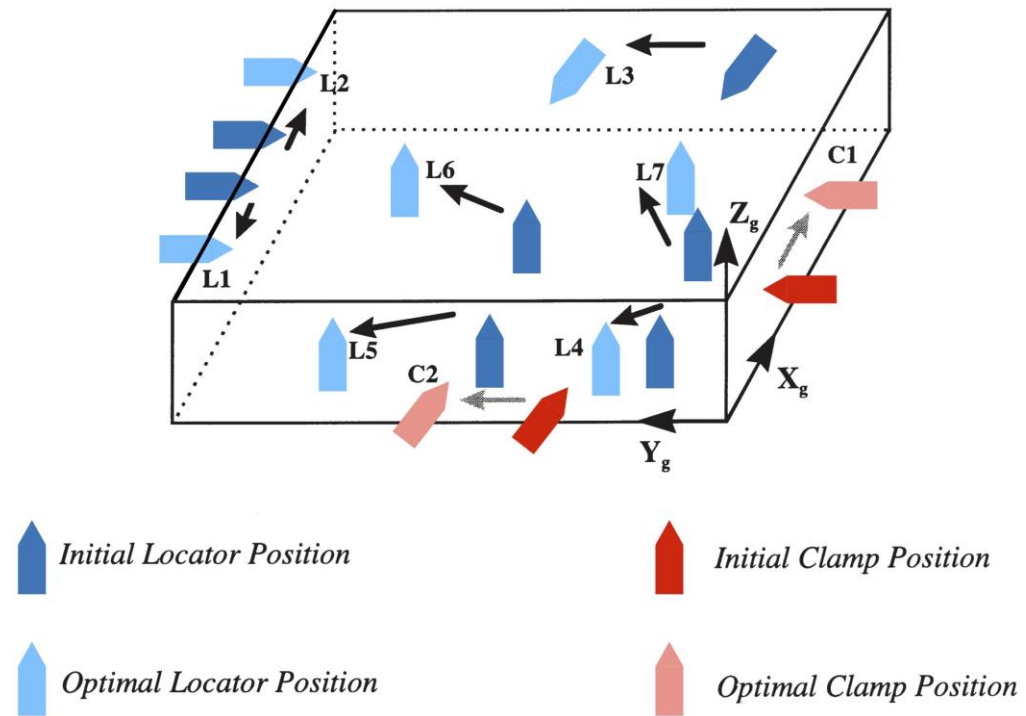


Figure 2-2. Initial and optimised position of locators and clamps for a simple workpiece [5].

Siebenaler and Melkote [6] studied small localised workpiece deformations at clamp locations. They studied the impact of parameters such as mesh density and contact interactions in an FE model on the accuracy of predictions. They predicted local deformations as a result of clamping on a box section geometry (which had a magnitude of 20-40 μ m) with error below 5%, and also determined that 98% of all compliance in the workpiece-fixture system can be predicted by modelling only the workpiece and clamping tips. Kumar and Paulraj [7] also looked at minimising workpiece deflections, using FE integrated with a genetic algorithm to rapidly optimise clamping positions and forces for a drilling operation. The model was able to predict errors between nominal and actual dimensions of the machined regions.

Liu et al [8] examined a simple thin walled component used in a peripheral milling operation. Using a FE model based on cutting force data, machining deformations were predicted. From this displacement data, a heuristic algorithm was able to determine optimal locator positions from an arbitrary start point. Maximum surface error was reduced by up to 25%, shown in Figure 2-3.

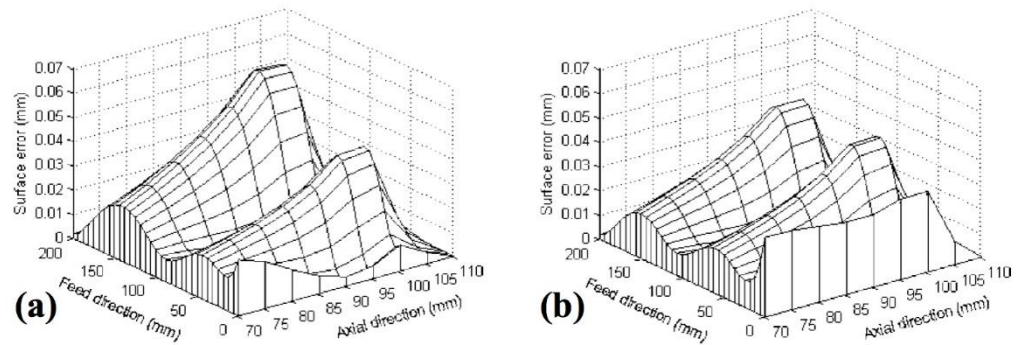


Figure 2-3. Comparison of surface errors using initial (a) and optimised (b) locator placement [8].

Raghu and Melkote [9] built upon their earlier work by examining the impact of clamping sequence on errors in of the fixtured workpiece. For a simple 3-2-1 fixture in clamping of a solid flat workpiece, shown in Figure 2-4, they investigated the effect of applying the two clamps simultaneously compared with applying each clamp before the other, with reaction forces measured by a load cell in one of the locators, and location error determined by measuring the displacement at a fixed point on the workpiece. Location error was consistently small regardless of clamping sequence, at 6-7 μ m. Predictions were within 2 μ m of the measured value, however this corresponds to a high percentage of error in the prediction. Force prediction was found to be fairly inaccurate, due to the assumption that clamp forces at a locator opposite one clamp do not change as clamps on other surfaces are subsequently applied. Despite the low accuracy of force and displacement predictions, the system successfully predicted the relative performance of different clamping sequences, and therefore can be useful for qualitative prediction of optimal clamping sequence – especially given that the analysis takes seconds to run, compared to hours for a comparable FE model.

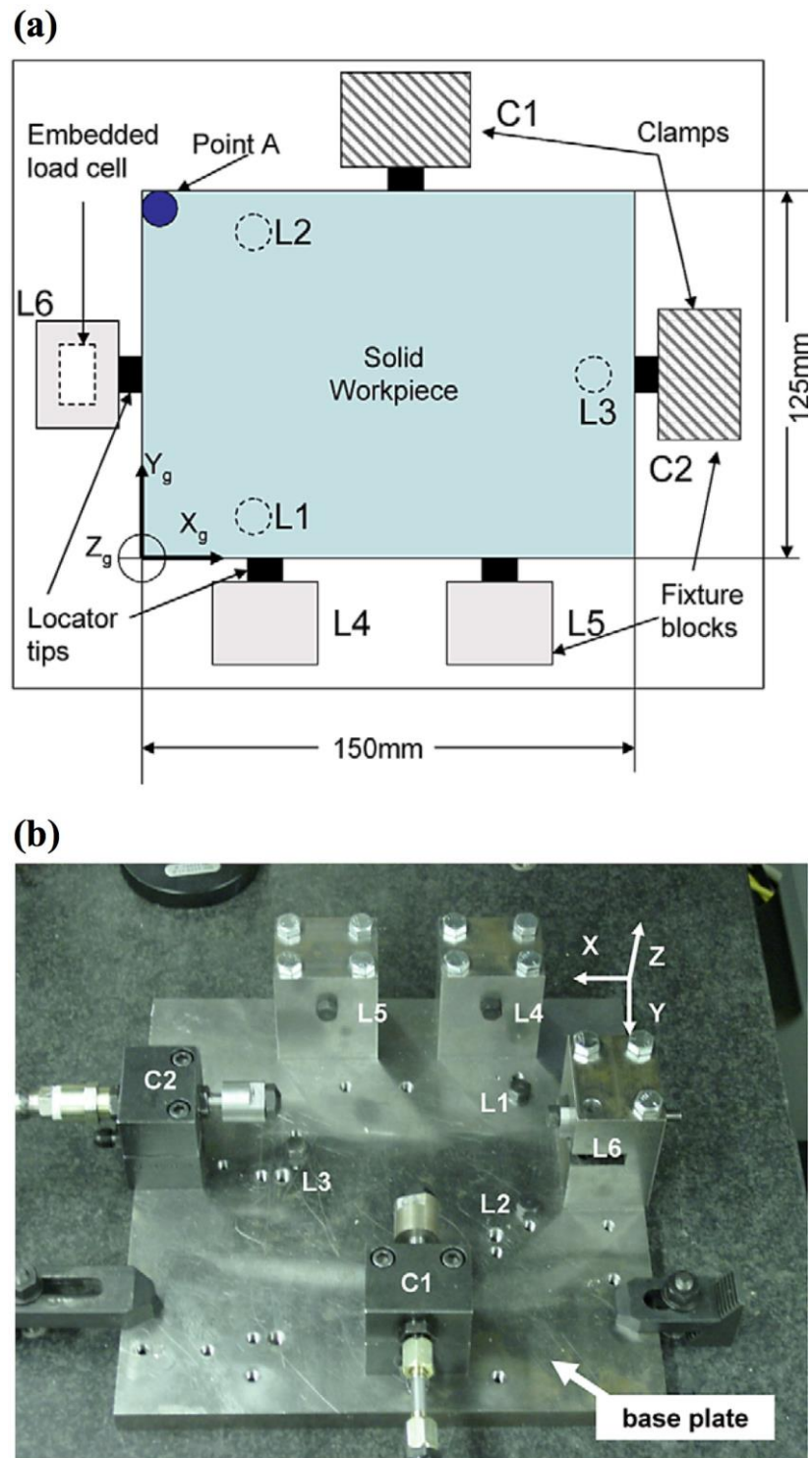


Figure 2-4. Schematic (a) and photograph (b) of a typical 3-2-1 fixture, used for optimisation of clamping sequence [9].

Due to the many mechanisms that influence the workpiece and fixture during machining, it may not be obvious which locating elements are responsible for an out-of-tolerance feature on the workpiece. Marin and Ferreira [10] reported on a methodology developed in conjunction with Caterpillar Inc. to link locator

errors in the fixture to deviations from geometric tolerances in surface profiles and holes. They were able to predict compliance of a manufactured feature to tolerances based only on the errors present in the locator and, in reverse, to quantify permissible errors in the location based on the tolerances required of a feature. Wang et al. [11] developed a method to decompose sources of surface errors in a machining fixture into those caused by locating and machining, shown in Figure 2-5. They presented a case study with a ground turbine blade, combining FE and CMM data, and were able to attribute out-of-tolerance features to workpiece movement and twist. These methods allow the identification of fixture elements that cause issues in production, meaning only troublesome elements are rectified, avoiding the unnecessary modification of a fixture that is otherwise fit for purpose.

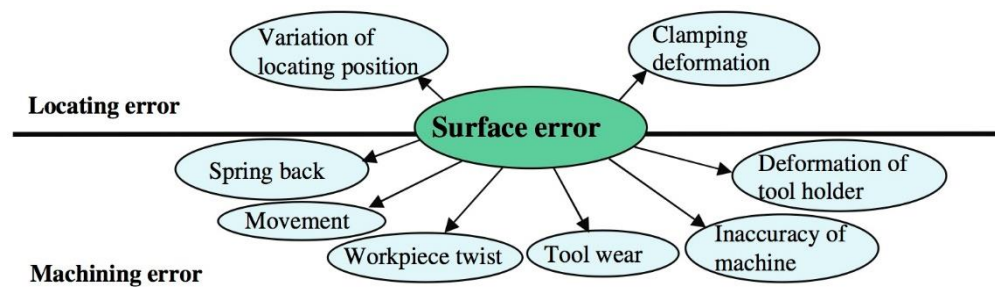


Figure 2-5. Sources of surface error in a machined component, grouped into those caused by location and machining [11].

Li and Melkote [12] developed a model that used a multi-objective optimisation to determine optimal clamping force for secure workpiece holding with minimal deformation when subjected to quasi-static machining loads. Deng and Melkote [13] built upon this to determine the minimum clamping forces needed for dynamically stable fixturing, to avoid slipping and lift-off phenomena at workpiece-fixture interfaces, while preventing excessive deformation due to over-clamping. Initial clamp conditions, vibration of the workpiece and fixture under machining loads, and changes in the system due to material removal are considered, with a particle swarm optimisation method employed to generate clamp force values. A key finding is that minimum required clamping force reduces significantly as material is removed (by up to 60% in the component shown in Figure 2-6). This suggests that better machining results could be

achieved due to reduced deformation of the workpiece if a dynamically varying clamp system were employed, since the initial clamping forces mean the workpiece is over-clamped in its finished state.

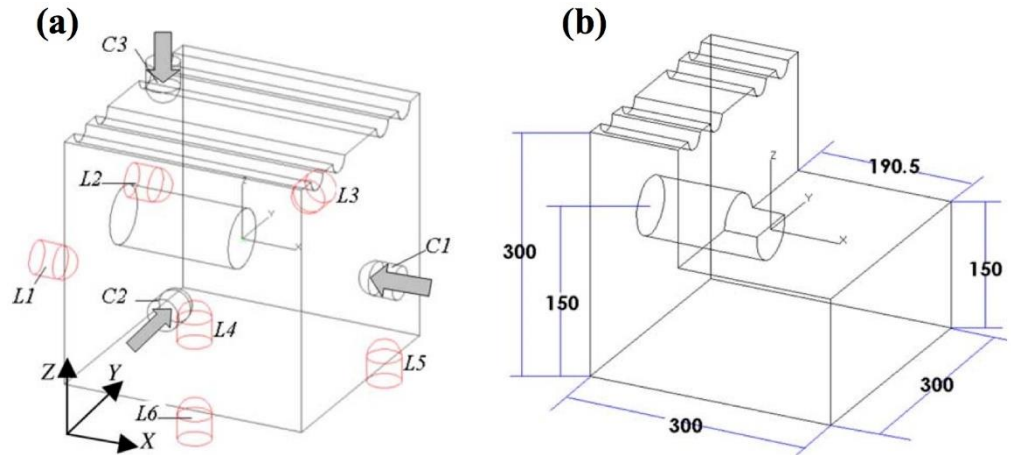


Figure 2-6. Sample component before machining (a), where the clamping required reduces as material is removed on the way to its machined state (b) [13].

Deiab and Elbestawi [14] developed a model that considers dynamics of the workpiece-fixture system for improved fixture design in a face milling operation for flexible components, with experimental friction data used as the basis for an FE model of friction interactions between the workpiece and fixture. Deiab [15] improved upon this model, using a flexible casing as a workpiece to study how stability and flatness change with clamp location. Properties of the system including workpiece and fixture geometry, and cutting conditions, were used to determine natural frequencies and deflections during machining. Based on the frequency response in its initial fixtured state, clamps were repositioned so that regions experiencing large displacement due to vibration were held down by the clamps. After several iterations of this process, resonant displacement of the casing was minimised, and the first eigenfrequency had almost doubled due to elimination of a bending mode along the length of the structure. This permitted machining using the desired parameters without causing resonance of the workpiece, demonstrating how fixture optimisation techniques can create a concurrent engineering environment for fixture design that reduces trial and error.

Tadic et al. [16] investigated workpiece-fixture interface compliance in machining conditions, and built a large-scale dedicated test rig for use with a vertical machine tool for measuring forces and displacements of the fixture during machining. A pivoting mechanism, seen in Figure 2-7, used weights to apply clamping to a component such that it could bounce away from the cutter if cutting reaction forces exceeded the clamping load. Spherical locating tips with diameters of 10mm and 60mm replicate domed locators at the component interface. Machining excitation was introduced by drilling a specimen at a range of feed rates. The weights determined the magnitude of the clamping force, and acted to maintain contact between the sample and workpiece, with a load cell configured to measure tangential forces. While the tangential force transmitted to both spherical tips was similar, the larger locating tip generally experienced slightly lower lateral displacement, with the difference becoming more pronounced at aggressive material removal rates with low clamping force. The system is unusual in that it considers a single workpiece-locator interface, which cannot normally be studied in isolation when it is part of a complex workpiece-fixture system. It is therefore ideally suited to the experimental verification of simulated clamp interface behaviour.

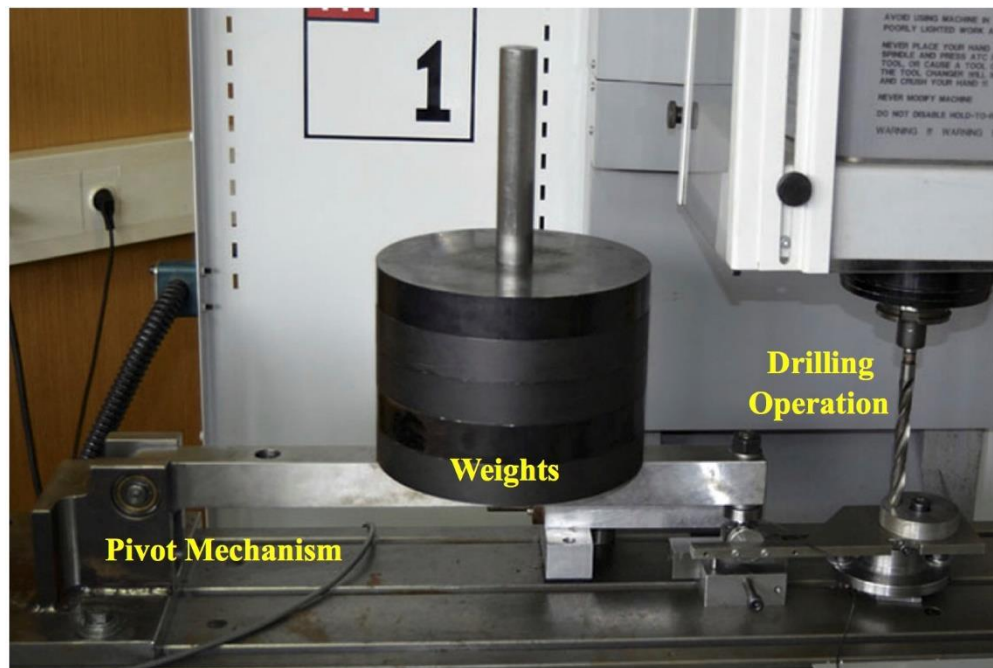


Figure 2-7. Rig for studying workpiece-fixture contact during machining [16].

2.2.2 Conforming and adapting fixture systems, and fixturing of workpieces with pre-existing residual stresses and distortions

A great deal of research has aimed to enhance the quality of manufacturing processes, in order to improve the accuracy and repeatability of components. This is generally desirable, so long as costs associated with process improvements do not outweigh the benefits, since an upstream manufacturing operation that produces high quality blanks will mean components arriving in subsequent stages of processing will be more consistent. However, some processes are inherently subject to variability, such as welding, forging, casting, and composite lay-up, meaning it is not always possible to achieve high consistency of incoming blanks for machining. In this case, a fixture that is able to accommodate variations in incoming workpiece geometry can compensate for this inconsistency, enabling the production of final components that conform to the tolerances required of them.

Hurtado and Melkote [17] used commercially available air-actuated pin arrays in a flexible fixture that conforms to component surfaces. They used opposing arrays in a vice configuration so the component was supported on two sides by pins. Keeping 2 pins in contact with the component on one side of the vice, they tested three configurations on the other side with 4, 8, and 12 pins in contact. The pins were actuated so they adopted the profile of the workpiece, and then were locked in place so they retained its profile when removed from the workpiece. The vice was clamped with a force of 800N, with a 50N lateral perturbation force applied to measure deformation in the component. Displacement of the workpiece relative to the fixture was found to increase as conformability increased, suggesting that increasing the number of pins in this setup resulted in over-constraining of the workpiece, without the advantage of the additional support provided by full-face contact. The same authors [18] went on to create an algorithm to determine optimal number and position of pins for the array system, plus optimal clamping forces for a given workpiece and machining load.

Al-Habaibeh et al. [19] used a pin array based on low melt alloy to fixture components with generic aerofoil geometries in aluminium and steel for machining, shown in Figure 2-8. The pin fixture, which was in contact with the blade across its entire aerofoil surface, was compared to a dedicated fixture for machining performance, and found to offer improved surface R_a on the machined aerofoil surface of a steel workpiece; with the dedicated fixture performing better for machining grooves into the root surface of an aluminium workpiece. While it was verified that the pin system was capable of holding the workpiece for machining, deformation of the blades was not measured.

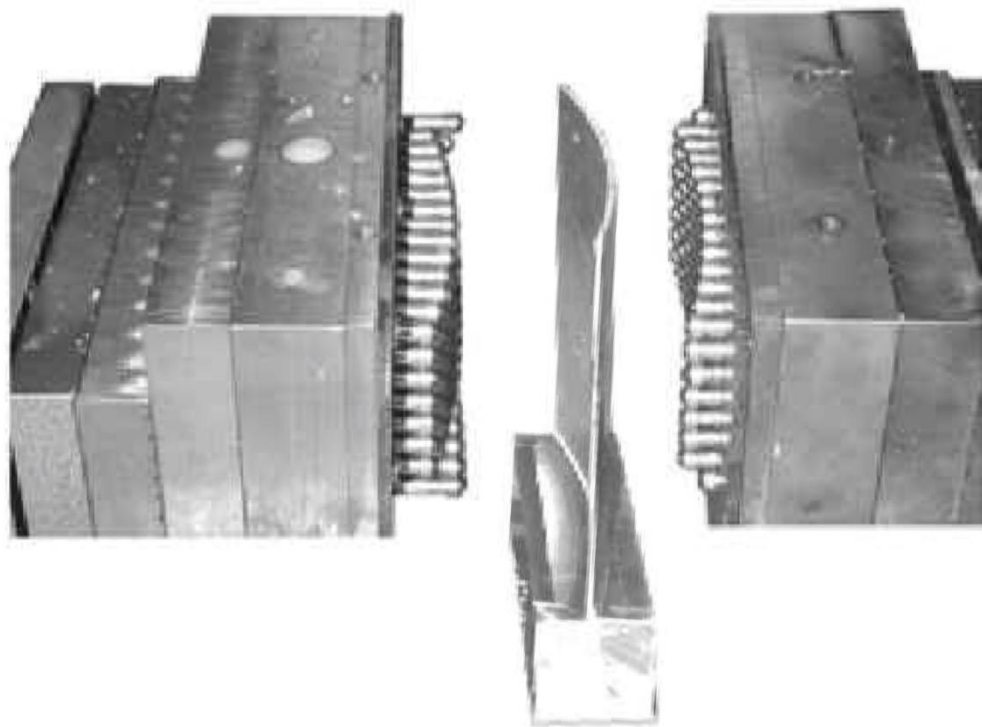


Figure 2-8. Reconfigurable pin fixture for machining an aerofoil component [19].

Aoyama and Kakinuma [20] also experimented with conforming pin fixtures based on a low-melt alloy. Rather than clamping via the movable pins, they used them only to support to the face of a thin deformable workpiece, with clamping applied independently via separate clamps and locators, as shown in Figure 2-9. When the chamber of low-melt alloy was heated, and the alloy reached its liquid state, it caused the pins to rise and make contact with the workpiece due to buoyancy, therefore resulting in a very low contact force. The initial design was shown to result in unacceptably high thermal expansion for a precision fixturing

application, however an improved design, which suspended the heated chamber so it was no longer in contact with the fixture base and reduced the volume of alloy from 50ml to 5ml, was found to greatly reduce thermal effects. Displacement of a supporting pin due to thermal contraction upon cooling was improved from $42\mu\text{m}$ to below $3\mu\text{m}$. The supporting effect of the pins was shown to greatly improve stiffness of a 1mm thick fixtured plate component, reducing deformation when a 100g mass was placed at the centre of the plate from $50\mu\text{m}$ when unsupported, to $1.5\mu\text{m}$ with the pins in place. Contact with the pins was found to damp vibration in the workpiece, with hammer testing revealing a threefold increase in the damping ratio, along with total suppression of the first eigenmode of the workpiece, and a 66% reduction in vibration at the second.

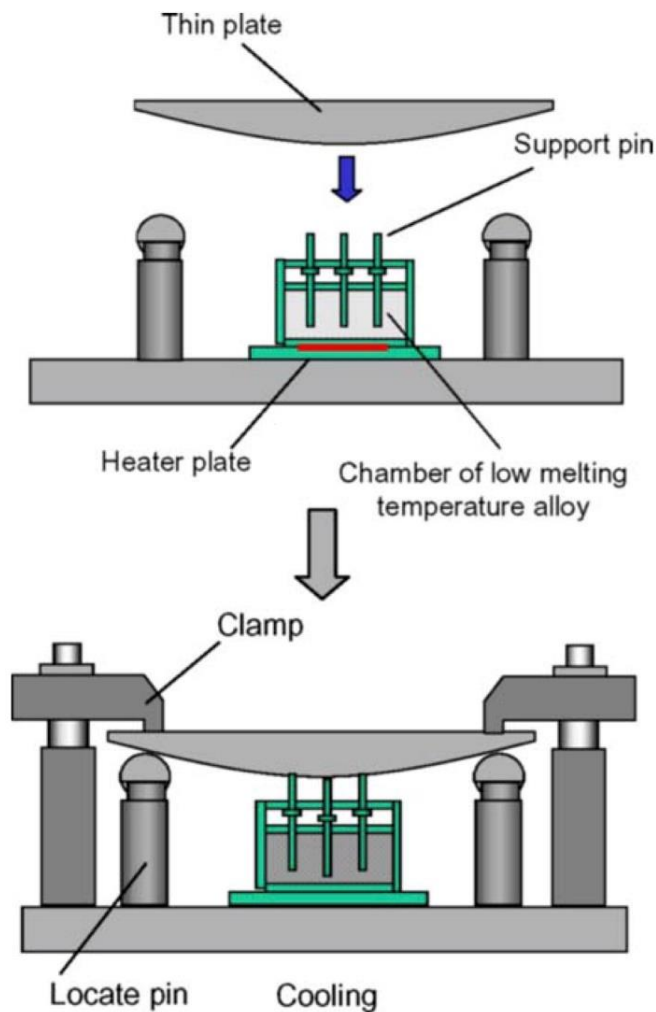


Figure 2-9. Pin-support fixture based on low-melt alloy [20].

Yang et al. [21] machined example monolithic components for aerospace applications in aluminium and titanium. They examined residual stresses induced in the workpieces before machining, to represent blanks that had been previously subjected to a forging or rolling process. For aluminium components, they found the primary source of distortions in the component after machining to be residual stresses that were present in the blank prior to machining. For titanium components, residual stresses had little effect post-machining, with the most important factor for minimisation of errors found to be the selection of appropriate machining parameters in order to minimise machining loads. Following this, Cerutti and Mocellin [22] considered residual stresses going into the milling process in order to study post-machining distortion of a component cut from rolled aluminium stock, arising due to re-distribution of stress. The machined component, shown in Figure 2-10, exhibited a noticeable bowing effect since material was only removed from the top side of the component, leaving the rolled bottom surface unmachined, resulting in the ends of the workpiece curling up by 0.5mm over its 500mm length. Post-machining distortion due to pre-existing stresses was predicted accurately by their FE model, meaning analytical techniques can feasibly reveal methods of improving distortion in machined components.

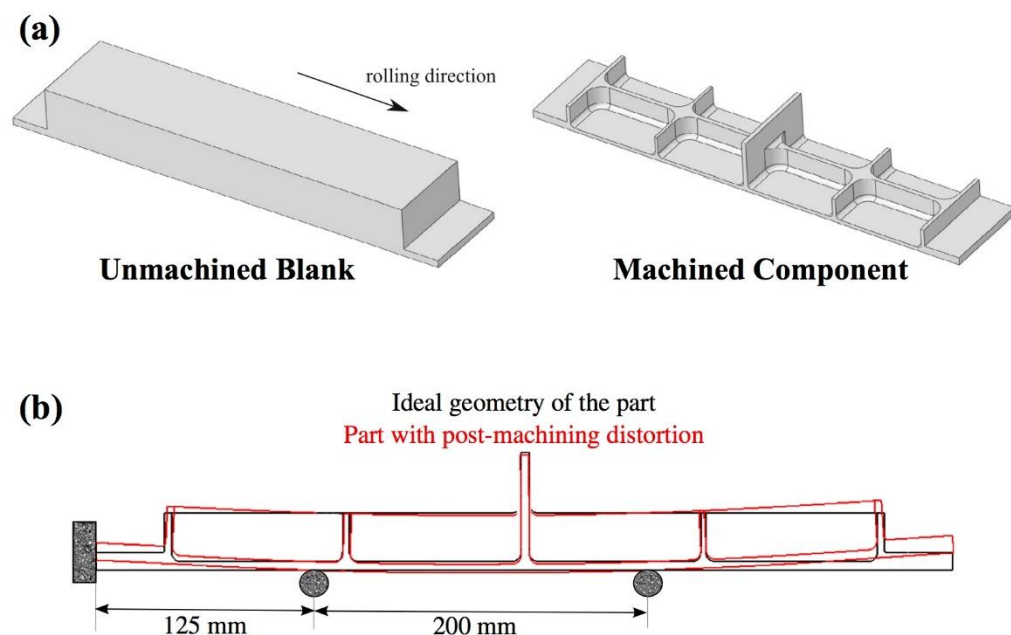


Figure 2-10. Component in its pre- and post-machining state (a), with visualisation of post-machining distortion (b) [22].

Wang et al. [23] demonstrated a method of fixturing using a low-melt alloy, which has applications for machining thin and delicate components. Instead of using the alloy to actuate and hold pins in contact with the workpiece (as in [20]), they used it to support the workpiece directly by fully encapsulating it in alloy within the fixture. From the fixtured blank, they were able to machine the complex features shown in Figure 2-11 using a 5 axis machine. The low-melt fixture was shown to produce better accuracy and flatness in thin-walled regions compared to a conventional fixture. Despite the 12 minute changeover time required to re-heat the alloy, and the 4% loss of alloy during each machining cycle, the process is still viable due to its numerous advantages for fixturing of challenging or delicate geometries. Since the alloy conforms to the shape of the workpiece, it can accommodate and securely hold a component regardless of variations in its geometry, and with negligible force applied to the workpiece. This results in considerably reduced distortion of components compared to a conventional fixture with clamps, which tend to apply highly localised clamping force to deform the workpiece until it is in contact with locators. Thin wall regions are highly susceptible to deformation due to clamping and cutting forces, but when encapsulated in low-melt alloy, they are rigidly supported across their full surface without any clamping force, which prevents deformation and also greatly inhibits vibration.

Low melt alloys have been shown to exhibit low solidification bias (expansion or contraction of encapsulation material upon solidification) compared to waxes, for which shrinkage of approximately 5% upon cooling is common. Low melting temperatures for encapsulating alloys are desirable, since some alloys melt at temperatures higher than the melting point of common non-metallic workpiece materials such as ABS, and thermal cycling of thin components during encapsulation and release from the fixture has been shown to cause unacceptable warping [24].

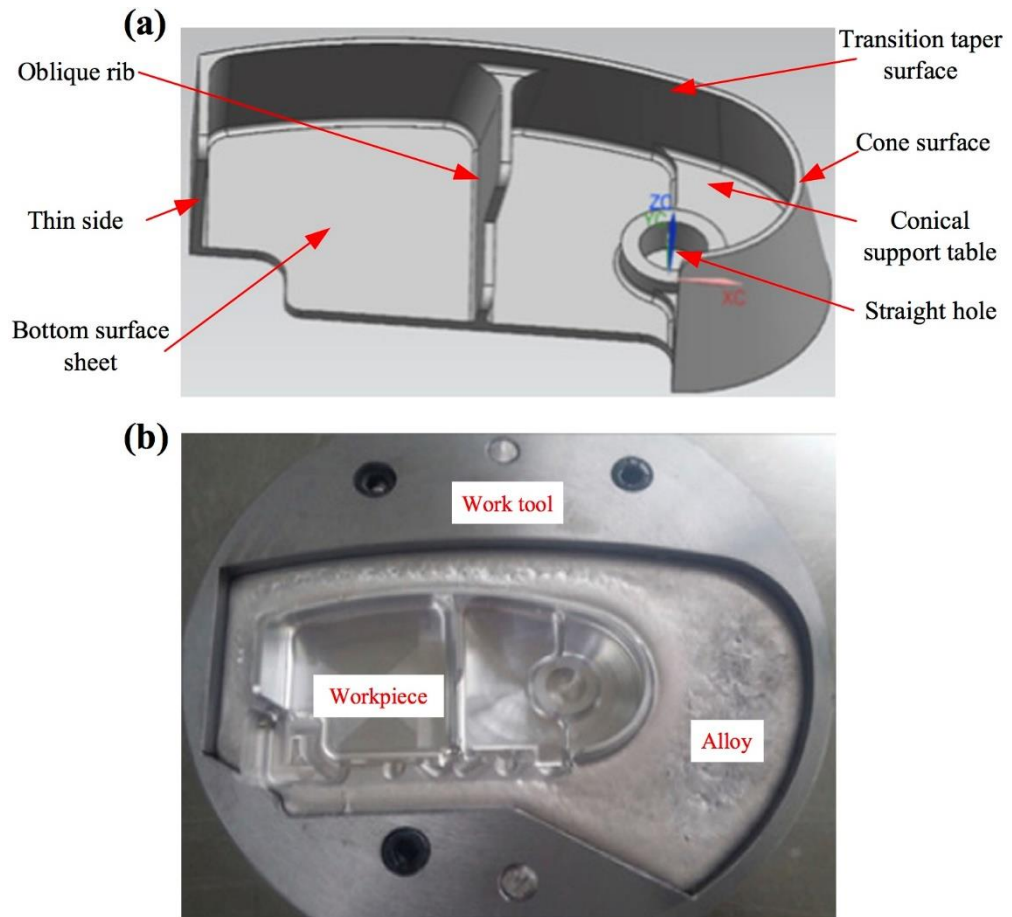


Figure 2-11. Complex thin-walled component for machining (a), shown encapsulated in a low-melt alloy fixture (b) [23].

2.2.3 Innovations to improve fixturing performance and capability

In this section, a diverse selection of various notable fixturing innovations are presented, with applicability to a range of workpiece types and manufacturing systems.

Yeh and Liou [25] identified that clamping faults can be detected in a fixture as a result of the change in its dynamic behaviour. In a mass production environment, where all workpieces are the same, and clamping locations and forces are consistent, then the frequency response of the assembled workpiece-fixture system should be identical. If a fixture is incorrectly assembled, or the workpiece is inadequately clamped due to wear or damage of the fixture, there will be a sizeable shift in the natural frequencies of the system. Variations between individual workpieces, and due to different operators, will have a very

small effect on frequency response relative to that of a clamping defect, meaning issues can be detected easily. They found that the first natural frequency in their example shifted by 300Hz between its unclamped and fully clamped states.

Zeng et al. [26] developed a novel fixture with a simple method of vibration suppression for a thin walled box geometry. As can be seen in Figure 2-12, screws were positioned to support the walls of the box. This inhibits displacement at the most flexible region of the component, the antinode of the first bending mode of the wall, which resulted in significant vibration damping, and led to improved surface finish and accuracy in the machined regions.

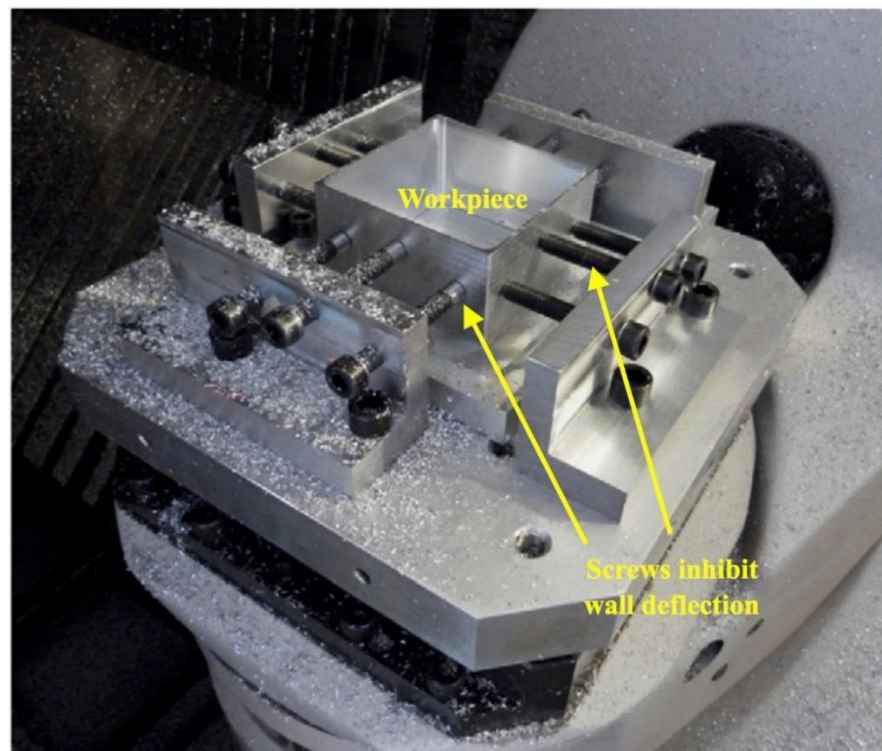


Figure 2-12. Simple workpiece supports to reinforce a thin wall region and reduce vibrations [26].

He et al. [27] investigated pre-tensioning of workpieces in a fixture prior to machining by introducing a small initial deforming force to stiffen the component, as shown in Figure 2-13, which was found to reduce deflections in face milling of a simply supported thin beam.

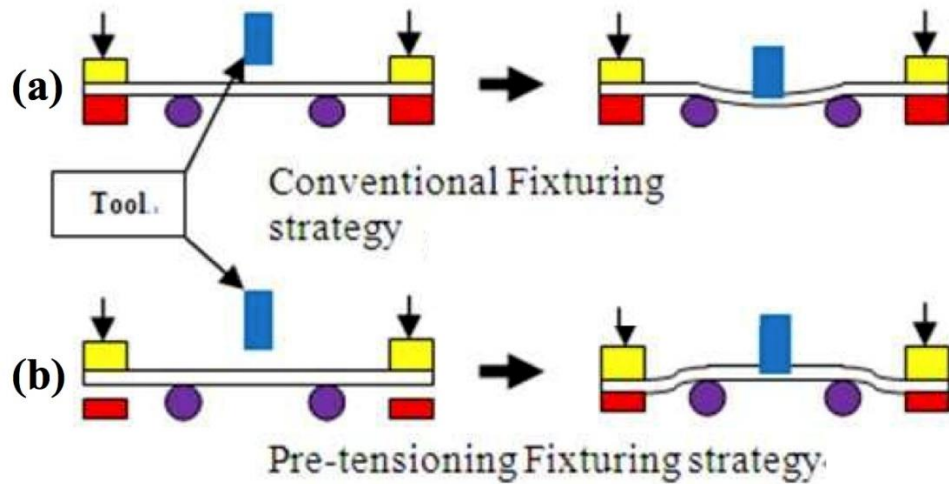


Figure 2-13. Deflections experienced by a thin workpiece in face milling without (a) and with (b) pre-tensioning [27].

Boonsuk and Frank [28] developed a ‘sacrificial fixture’ system in place of conventional clamps and locators, that took inspiration from support structures in 3D printing, for a component machined from round steel stock material. The billet was fixtured at each end in three jaw chucks on a rotary indexer with a single axis of rotation, and progressively machined until the final component geometry was realised, but was still joined to the stock material by a small support section. The final stage involved cutting away these support sections so the component could be removed. The machining sequence is shown in Figure 2-14.

Violante et al. [29] combined off-the-shelf fixturing components with additively manufactured (AM) custom inserts printed in alumide to enhance fixturing of a freeform sheet metal component. The component was laser scanned to obtain its geometry, from which the contact inserts were designed to conform exactly to its features, meaning each contact point between the component and fixture was a bespoke insert. By using modular fixturing components to comprise the majority of the fixture, with only small printed inserts, as shown in Figure 2-15, the cost was much lower than a fully bespoke additive fixture, and it is also likely to be stiffer and more durable than an all-printed fixture. Use of a coordinate measurement machine (CMM) determined that the tolerances of the sheet component were lower than those of a sintered AM insert, meaning the

inserts were sufficiently accurate for this application in their as-printed state without the need for post-processing.

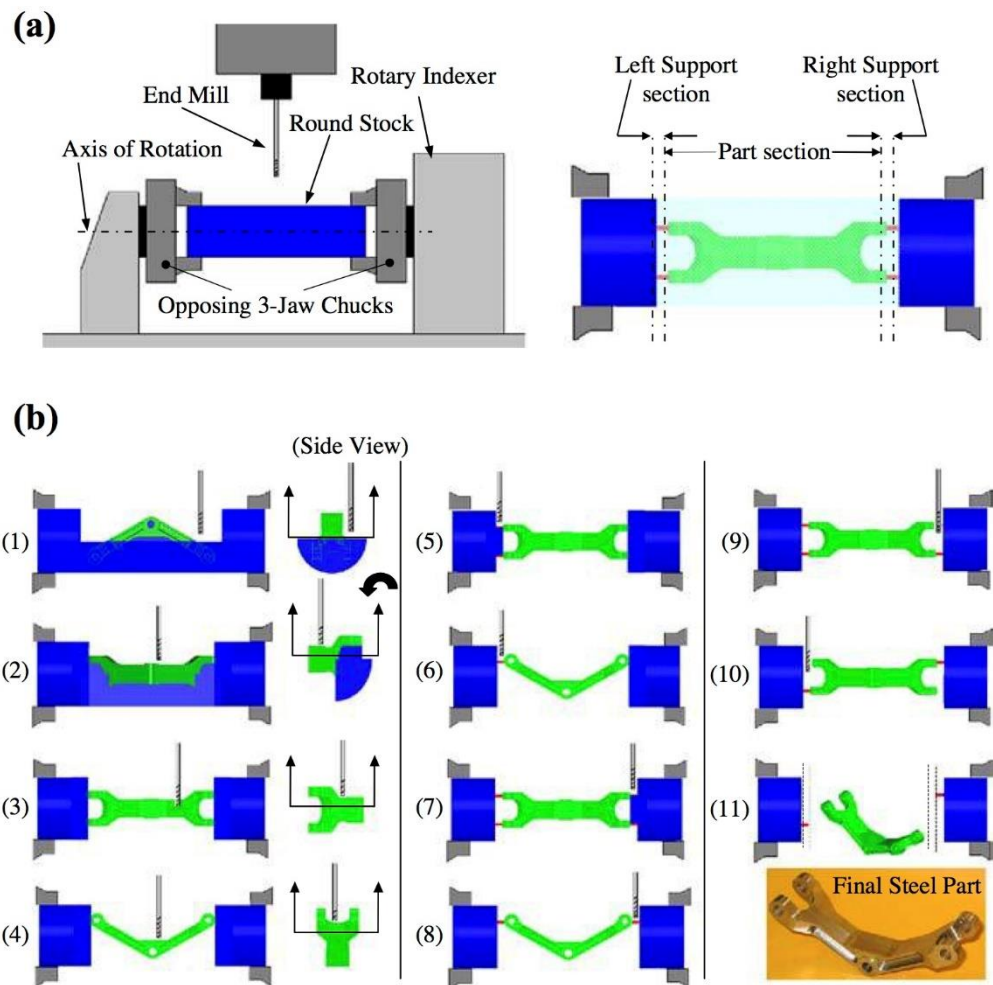


Figure 2-14. Machining configuration for sacrificial fixture (a), and stages of the machining process (b) [28].

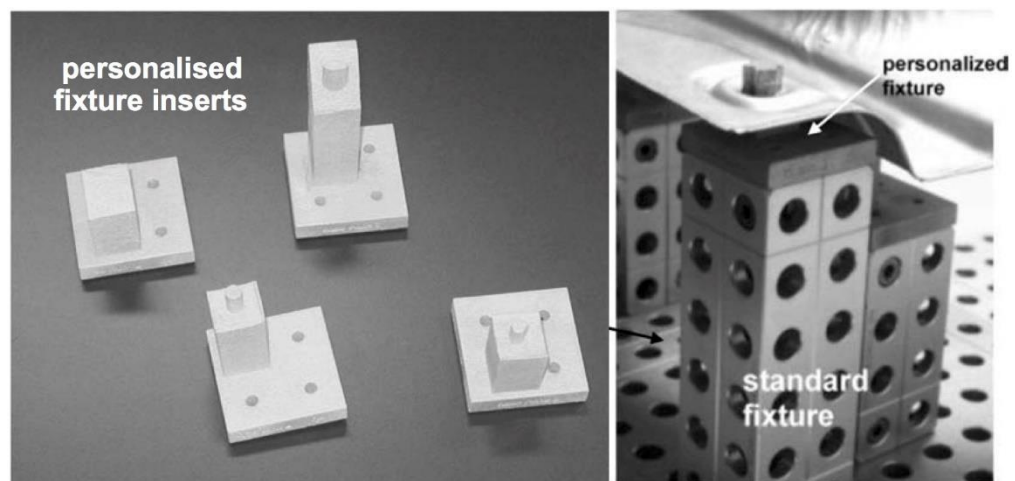


Figure 2-15. Bespoke inserts for a partially-AM fixture [29].

Möhring et al. [30] developed an intelligent fixture system with piezoelectric patch transducers to detect vibrations during the machining of a blisk, which provided counter-excitation via a piezo actuator to destructively interfere with vibration. They tested the concept in end milling a thin wall cantilever with a single degree of freedom (1-DOF) active damper, shown in Figure 2-16, which was found to improve surface R_a from $3.35\mu\text{m}$ to $0.68\mu\text{m}$ when the system was engaged. A subsequent setup with a mock-up blisk using 4 flat blades, where counter-excitation was rotational, was found to yield similar reductions in vibration, with corresponding surface finish improvements. An FFT showing the frequencies present in machining revealed that the tooth engagement frequency was detected when the actuator was not active, however it was completely removed when the system was counter-excited, which accounts for the isolation of machining vibrations.

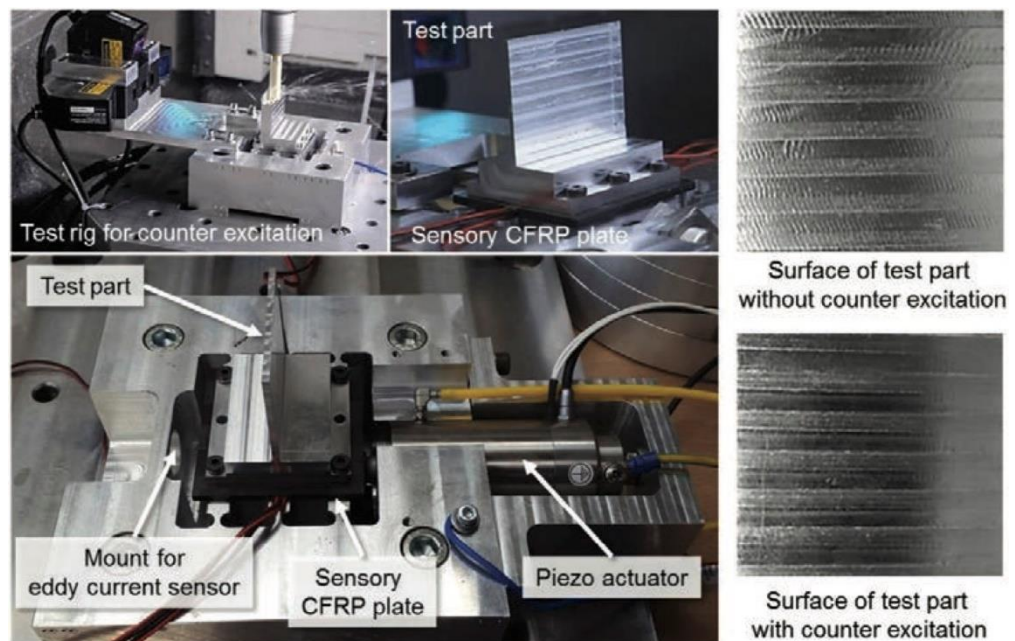


Figure 2-16. Active fixture for reduction of machining vibrations by counter excitation [30].

2.2.4 Summary

A fixture provides the critical link between the workpiece and machine tool, meaning it has a significant impact on the quality of the finished machined component. Fixture design is particularly important for holding compliant workpieces, which are susceptible to deformation when restrained in a fixture.

CAFD techniques, including optimisation of clamp and locator placement, and FE to predict performance, allow the production of high quality fixtures, eliminating the need for lengthy prototyping and design revision stages.

In [21], residual stresses in aluminium components from a preceding forging or rolling process were shown to be the predominant source of distortions in the final machined component. However, such residual stresses can also be introduced by an ill-fitting fixture that plastically deforms the workpiece during clamping. A fixture that is capable of reducing clamping stress will therefore reduce the likelihood of workpiece plastic deformation, which has the potential to improve the accuracy of finished components.

Conformable pin fixtures, which adopt the shape of the workpiece before clamping is applied, have been used to reduce deformation in low stiffness components. In [19], a close-packed pin array fixture is used to achieve near full-face contact with a thin turbine blade workpiece, and shown to be capable of holding aluminium and steel components for machining. In [20], a sparse array of conforming pins, which support only a small region of the workpiece and are not used for location or to apply clamping, was shown to have a significant damping effect on the workpiece.

A conforming pin array fixture therefore has the potential to reduce stresses in the workpiece, while offering improved support that inhibits movement of the workpiece, thereby improving accuracy of the finished component. However, there are no examples reported in the literature of a machining fixture investigated primarily for its ability to accommodate variations in blank geometries, or investigations into the ability of pin fixtures to attenuate vibration.

2.3 Vibrations, chatter, and machining stability

A sound understanding of the dynamic behaviour of a workpiece-fixture system before and during milling allows high quality components to be manufactured. This requires an awareness of the behaviour of structures in free and forced

vibration. Vibration of the tool or workpiece leads to poor accuracy and surface finish of components, premature wear of tools, and even damage to the machine. Therefore knowledge of the dynamic behaviour of the tool, workpiece, and fixture allows their resonance to be prevented during milling, and knowledge of machining dynamics allows stable milling at high material removal rates. This section looks at work undertaken to characterise the dynamic behaviour of elements in a workpiece-fixture system, as well as stability analysis of the milling process.

2.3.1 Natural frequencies and free vibration of structures

All structures have inherent dynamic behaviour, which is the product of numerous factors including geometry and material. In addition, they are affected by external factors such as fixturing/constraining, forces acting on the system, and temperature. These determine the natural frequencies of the system, each of which corresponds to a vibrational mode shape. Whether a structure is a tool, workpiece, or fixture, understanding its frequency response is imperative to the prevention of undesired vibration. The resonance of a system is not always problematic, however, and is often exploited for damping or energy harvesting. Therefore, the ability to understand the parameters that influence free vibration is essential in order to control when and where a structure will resonate.

Emam and Nayfeh [31] looked at the free vibration behaviour of composite beams under different constraining conditions in their pre- and post-buckled states. The orientation of the plies (the layup) was found to be the biggest influencing factor in determining mid-span deflection of the beam under loading, with a unidirectional layup found to support the greatest axial loads before the onset of buckling. In the pre-buckling domain, the natural frequency of the beam decreases under loading until it is zero at the onset of buckling. In the post-buckling domain, the frequency starts to increase as the load increases. As is shown in Figure 2-17, the post-buckling frequency response is significantly affected by the layup pattern in the laminate, meaning the layup can be manipulated to control the static (buckling) behaviour, and dynamic

(post-buckling) response, allowing the design of beams that are optimised for their specific application.

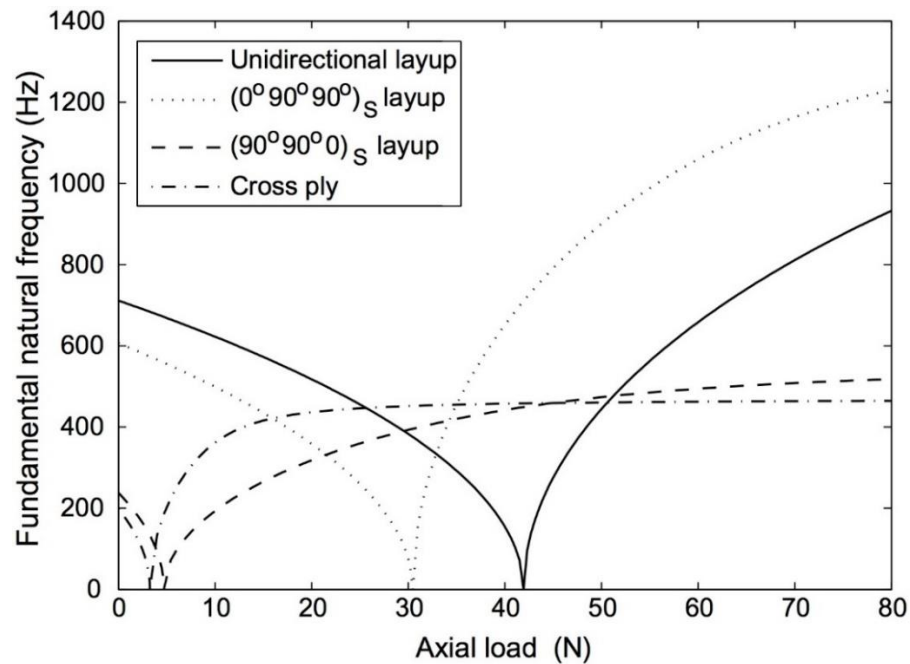


Figure 2-17. Shift of natural frequency in pre- and post-buckled fixed-hinged composite beams with various layups [31].

Piana et al. [32] expanded upon previous work examining the buckling properties of thin geometries with low warping stiffness, by manufacturing a 0.9m long aluminium structure with a double I-beam cross section, shown in Figure 2-18, that resulted in a non-negligible warping stiffness. The effects of axial compressive loading on the natural frequencies of the beam were examined in a universal testing machine (UTM), with two schemes of fixturing permitting free and partially restrained warping. For free warping, only the webs of the I-beams were gripped, and for partial restraint, the webs and flanges of the beams were gripped. Additionally, a defect in the form of a slot cut into the flanges of the I-beam was introduced to reduce torsional rigidity in some of the samples. In all cases, the natural frequency was shown to decrease as the loading increased. The samples with defects exhibited little difference in their dynamic responses compared to the undamaged samples, with the only difference being a reduced failure strength. Restraining the beams was shown to stiffen them, increasing the natural frequency and compressive strength.

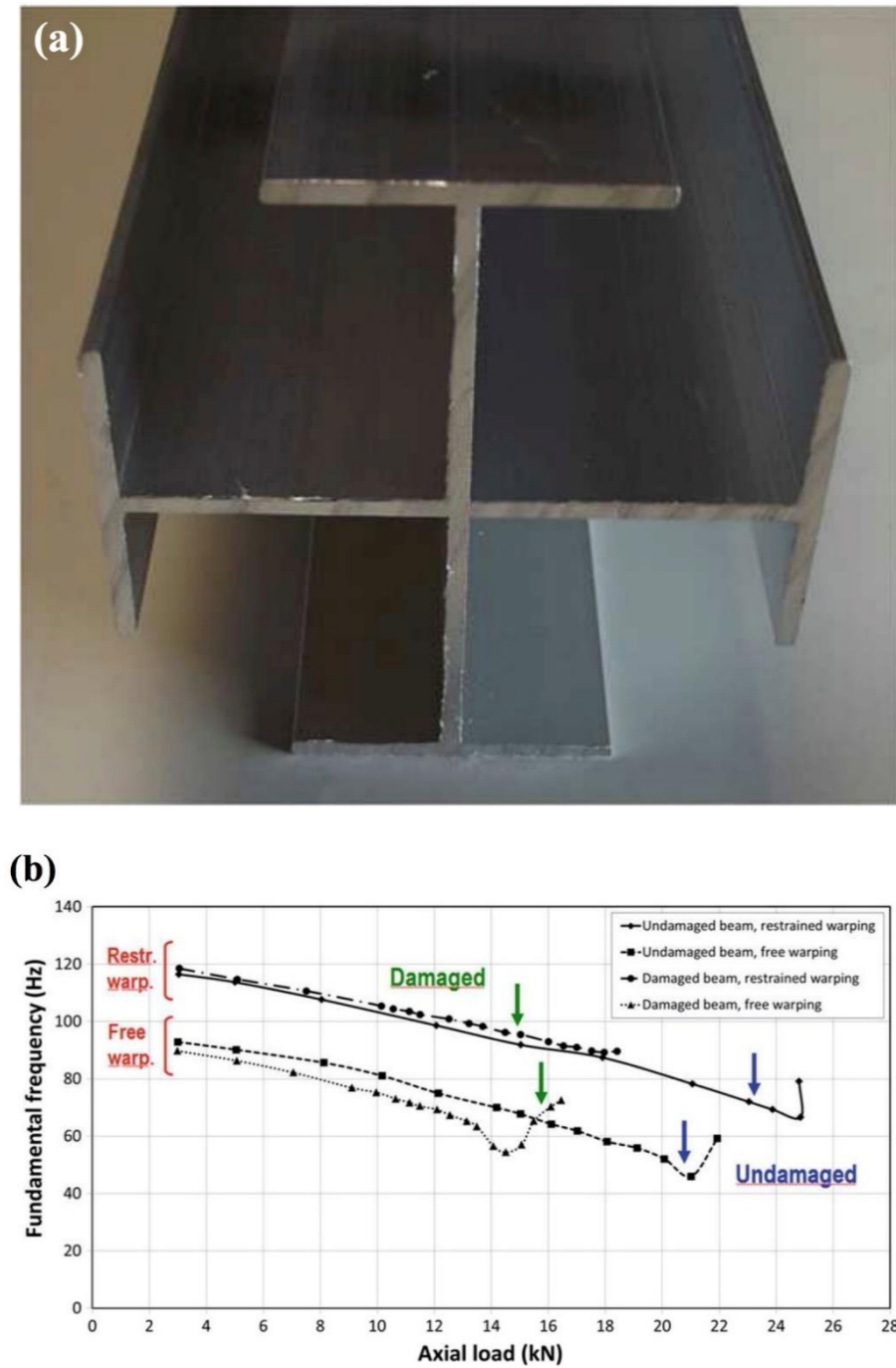


Figure 2-18. Double I-beam section with sides of length 50mm (a), and the shift in first natural frequency due to loading for the 4 experimental cases (b) [32].

Gawande and More [33] similarly investigated the effect of defects on natural frequency by cutting notches into a steel cantilever beam. Thin notches were wire cut to depths of 2, 4, and 6mm into the long side of the 10×25mm profile of the beam, effectively reducing thickness to as little as 4mm at the location of the notch, therefore significantly affecting its bending response. The notches were cut either 100mm or 200mm from the free end of the cantilever, which had

a total length of 500mm. The natural frequency of the notch-free beam was determined by hammer testing, and compared to the six notched configurations. It was found that the effect on dynamic response increased as the notch became deeper, and closer to the fixed end of the cantilever – the frequency of the first bending mode was lowered by 58% compared to the undamaged beam for a 6mm notch positioned 200mm from the free end. This experimental data was compared with predictions from analytical and FE models, which showed that the impact of the notch on natural frequencies is significantly underestimated by the models; in some cases the measured frequency was less than half that predicted by the MATLAB or ANSYS models. This indicates that relatively small defects can have a large effect on the dynamics of a flexible geometry, and therefore prediction of their influence requires improvement.

2.3.2 Machining stability

In milling, chatter occurs when the spindle speed causes the cutter or workpiece to vibrate. Because the tool is dynamically coupled to the workpiece during cutting interactions, this leads to time-dependent chip thickness, which manifests itself as a wavy surface on the machined component. Chatter can arise due to vibration of the tool, tool holder, or spindle (tool chatter); or when a workpiece begins to vibrate in response to forced excitation from the cutter (workpiece chatter). Thin-walled structures are particularly susceptible to chatter, and their machining is considered in greater depth in section 2.4. This section examines developments for the reduction of chatter, and improvement of material removal rates.

While chatter has been observed and studied since the 1950s using various numerical and time-domain analytical methods, Budak and Altintas [34] and [35] were the first to formulate a comprehensive methodology for prediction of chatter stability. By considering both the workpiece and cutter each as a 2-DOF system, shown in Figure 2-19a, they were able to accurately predict stable spindle speeds, with corresponding depths of cut. The resulting stability lobe diagrams, such as the example in Figure 2-19b, show the maximum stable depth

of cut for a given machining speed, enabling quick identification of optimal machining parameters.

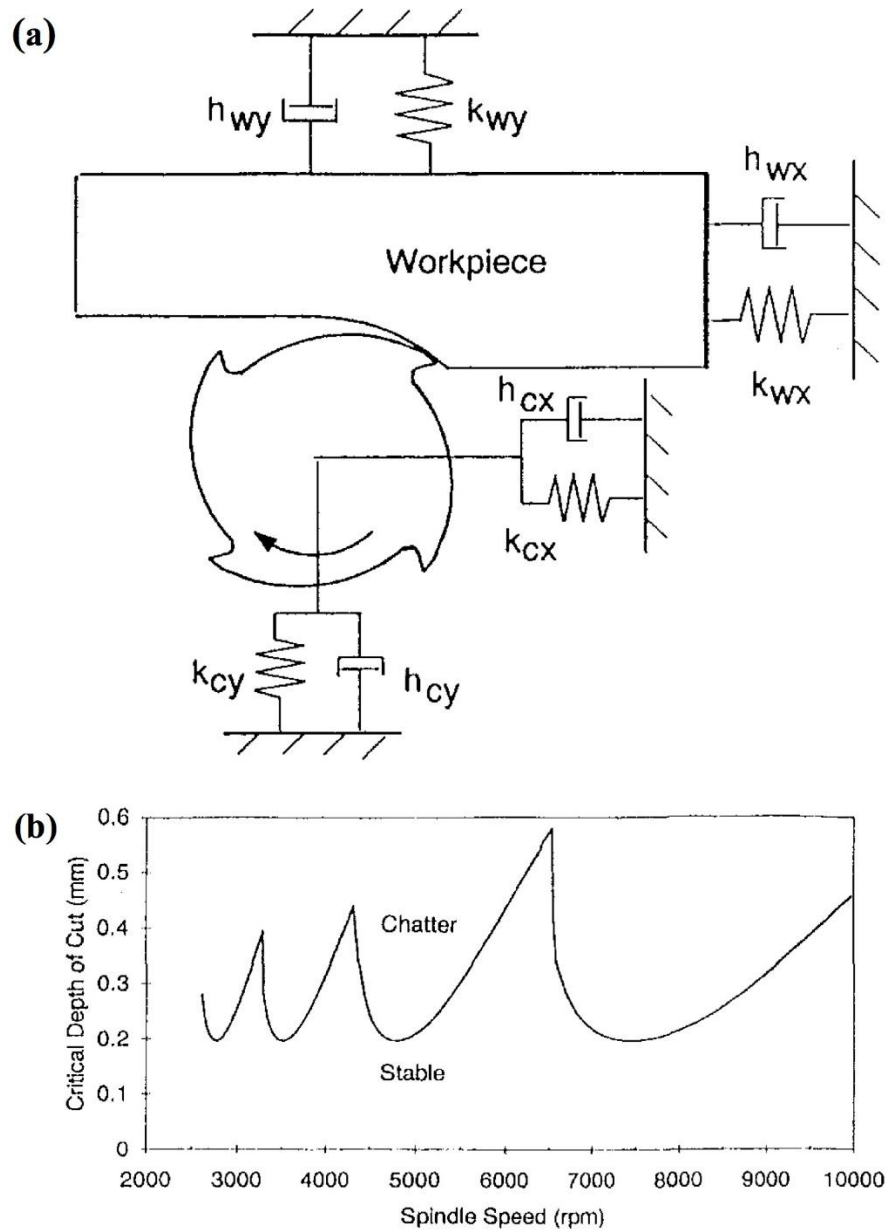


Figure 2-19. Multiple DOF system model for cutter and workpiece (a), for generation of a chatter stability lobe diagram (b) [35].

This frequency domain approach allows for chatter to be predicted with greatly reduced computational expense compared to previous methods. Since lower material removal rates are commonly employed to avoid chatter, this method allows the potential for greatly improved productivity. Commercial software,

including MetalMAX MilSim from MLI, and CUTPRO from MAL, generate stability lobe diagrams based on this algorithm.

Building upon his work on stability, Budak [36, 37] studied the effect of cutters with a variable pitch angle between the flutes on machining stability. Non-uniform spacing of teeth means the time between tooth interactions is not constant, thereby disturbing the regeneration mechanism observed in chatter, due to the introduction of multiple diminished cutter excitation frequencies in place of a single prominent frequency. He developed a model to determine optimal pitch depending on chatter frequency, with theoretical stability found to improve progressively as the number of teeth increases. This was validated with a variable pitch cutter (6 teeth with pitches of 48°, 52.8°, 57.6°, 62.4°, 67.2°, and 72°), and found to eliminate chatter in roughing (100-125mm depth of cut) of a titanium component. Peak force was reduced by up to 40% compared to a constant pitch cutter, enabling higher feed rates to be used. Furthermore, surface quality was so dramatically improved that ‘semi-roughing’ processes were no longer necessary with variable pitch cutters, due to the improved surface finish and dimensional control.

Another feature of the cutter that can affect stability is its helix angle. Zatarain et al. [38] studied the influence of helix angle on stability, investigating a cause of unstable milling known as *period doubling*, or *flip*, which occurs when excitation occurs at exactly twice the chatter frequency of a tool-workpiece system. The effect of the helix is an angular shift between the planes of the cutter at different heights, meaning higher planes come into contact with the workpiece after the lower ones at the cutting tip. When axial immersion of the cutter is equal to half the pitch of the helix (or 1.5 times, 2.5, etc.), it becomes susceptible to flip, due to repetitive impact-driven chatter at twice the frequency of the cutting interaction. This gives rise to *islands* of instability inside the stable regions of the stability plot, an example of which is shown in Figure 2-20. Therefore, chatter arising in an apparently stable region, but where axial depth of cut is not close to an integer multiple of helix pitch, may be alleviated by changing to a cutter with a different helix angle.

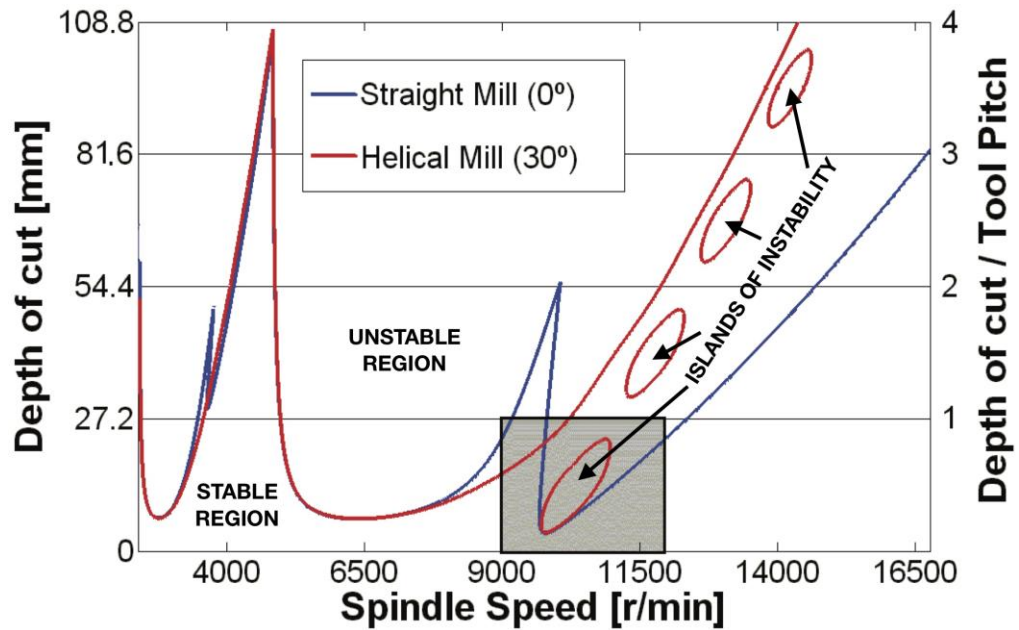


Figure 2-20. Islands of instability (due to cutter helix angle effects at large axial depths of cut) within the stable region of a stability plot [38].

Patel et al. [39] experimentally proved the existence of unstable islands due to helix angle, and investigated the impact of radial immersion of the tool on their presence. They found that true islands exist only for small radial depths of cut, as they tend to become subsumed into the conventional lobes for larger immersions, due to the dynamic influence of multiple cutter teeth in simultaneous contact with the workpiece dominating the mechanism of period doubling.

The ability to produce an accurate stability plot depends on accurate input of the parameters needed to determine stability, which include full specifications of the cutter, dynamic properties of the cutter in situ, and cutting force coefficients of the material to be machined. Cutting force coefficients relate cutting forces to feed per tooth, and are vital to the stability analysis because they constitute the workpiece/material definition in the calculation of stable cutting depth. The two coefficients most commonly used in stability prediction are the tangential cutting force coefficient K_t , (typically in N/mm^2), and the relative coefficient K_r , which gives the ratio of radial to tangential cutting forces. Campatelli and Scippa [40] developed a model to predict forces for different cutting conditions, proving a linear relation between cutting speed and

shearing coefficients, enabling reasonable prediction of cutting force coefficients when process parameters were altered. For small ranges of spindle speed, cutting force coefficients can be assumed to be constant, however Grossi et al. [41] used a machine with a high maximum spindle speed of 42000rpm to demonstrate the speed dependence of cutting force coefficients over larger ranges, shown in Figure 2-21, from which they generated revised stability plots compensating for these variations.

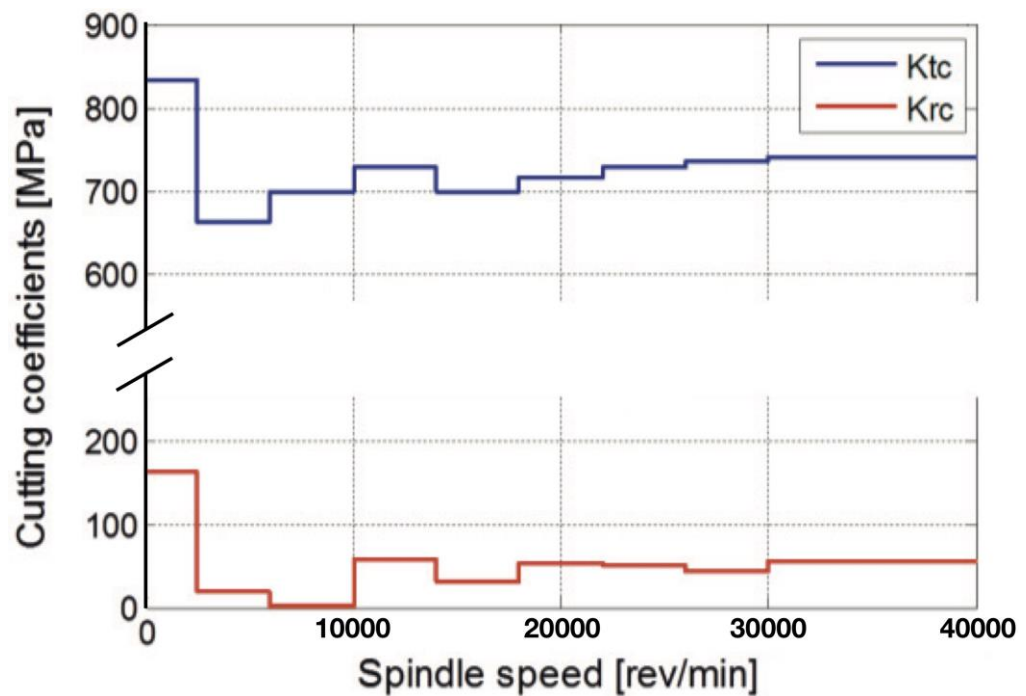


Figure 2-21. Plot showing spindle speed dependence of the tangential and relative cutting force coefficients used for calculation of machining stability [41].

Conventional stability plots are based on the natural frequencies of the tool when it is stationary, however a number of factors can influence the tool response when the spindle is rotating, and therefore there are potential advantages of characterising the machine in its operating state. Li et al. [42] used operational modal analysis with six accelerometers around the machine, and a seventh on the workpiece, in order to determine natural frequencies of the machine. White noise excitation was achieved using a spindle speed pattern that corresponded to uniform excitation between 300 and 700rpm, and this was compared with narrowband excitation with the spindle at constant speeds 400rpm and 600rpm, and also hammer testing with the machine in its stationary

state. While they did not attempt to match eigenfrequencies with mode shapes of the machine, a variation in detected frequency of 1-2% was typical between the different excitation methods, with certain modes proving almost undetectable depending on the excitation, thus indicating that the frequency response of the machine is sensitive to the speed at which it is operating.

Ozsahin et al. [43] were able to produce a frequency response function (FRF) for a rotating tool by effectively reversing the normal procedure used to generate a stability plot. Instead of starting with the natural frequencies of the tool and then using this to determine stable machining speeds and depths of cut, they experimentally determined chatter frequencies and their corresponding stable depths of cut, and used this to determine the frequencies of the tool under cutting conditions. From this they generated a full stability plot based on properties of the rotating tool. This was achieved for various combinations of tool and holder, with the final plots verified experimentally, and found to be good predictors of chatter stability.

Liu et al. [44] proposed a method of chatter identification involving in-process monitoring of cutting torque. Compared to conventional chatter detection methods based on a fixed threshold between stability and chatter, adopting a time-varying approach was found to result in earlier detection of chatter, while also reducing the number of false alarms (where a change in the tool path or workpiece geometry are erroneously identified as chatter initiation). For on-line process monitoring systems that detect chatter and dynamically alter cutting parameters to avoid it, this marks a potential improvement to productivity, due to improved prevention of chatter, and prevention of unnecessary reductions in material removal rate due to false identification of chatter.

Compared to machining of structures with low aspect ratios, where dynamics of the workpiece can often be assumed to be time-invariant, milling a thin-walled component usually results in measurable change to its dynamic response as cutting progresses, which has ramifications for its machining stability. This is due to proportionally higher material removal in thin workpiece geometries, and also the location of the cutter relative to regions of high and low stiffness in the

workpiece, which affect the frequency response of the workpiece-tool system. Song et al. [45] focused on this second phenomenon, and developed a time discretisation method to show how stability evolved between different cutter positions along the workpiece during milling of thin walls. By determining the FRF of the tool-cutter system at increments along the tool path, they were able to calculate numerous snapshots of stability as the cut progressed. These individual stability plots were combined to give a 3D plot showing the evolution of stability throughout the duration of the cut, as shown in Figure 2-22.

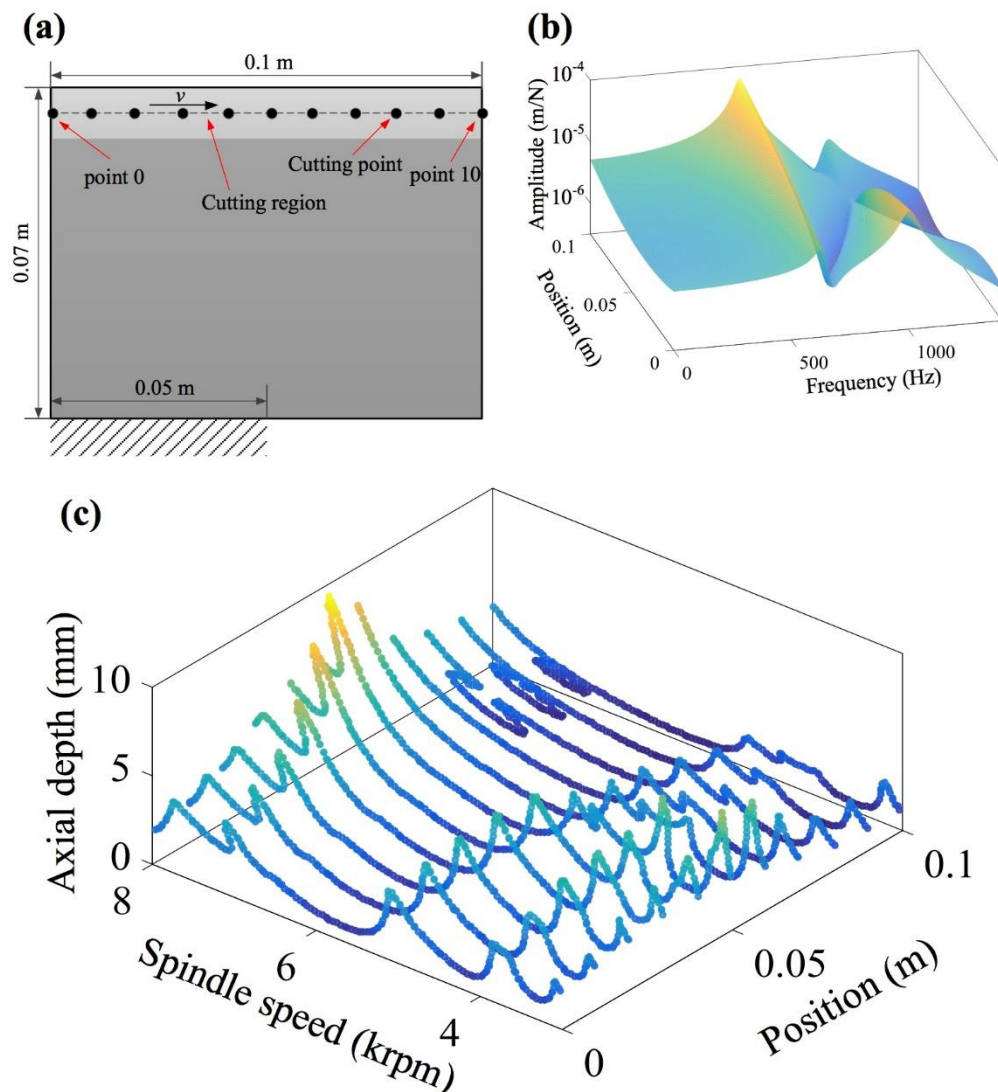


Figure 2-22. Geometry of a workpiece showing cutter positions at which frequency response was measured (a), with its corresponding 3D FRF (b), and a 3D stability plot demonstrating evolution of stable depth during cutting (c) [45].

Process damping is a phenomenon that occurs at low cutting speeds, where interactions between the cutter flank and workpiece improve the stability of machining, resulting in greater stable depth of cut than is predicted by a conventional stability model. Ahmadi and Ismail [46] compared a conventional multi-frequency solution method of determining stability with a semi-discretisation method that takes into account process damping for improved calculation of low-speed stability. While both methods agreed on the stability of high-speed machining, there was a considerable difference at low speed. Experimental validation confirmed that semi-discretisation provides accurate stability for low speeds, meaning failure to include process damping in the multi-frequency solution results in poor stability predictions for low cutting speeds. While the multi-frequency method usually relies only on the zero order of a Fourier expansion of time-varying coefficients governing regenerative chatter, also including higher order harmonics was found to greatly improve the accuracy of low-speed stability predictions, but at the expense of significantly longer computation time. Thus, the semi-discretisation method was found to be a good compromise between computation time and accuracy. Li et al. [47] verified a process damping-based method of stability prediction for machining of Ti-6Al-4V, which is typically machined at low speeds due to its poor machinability, highlighting the importance of accurate low-speed stability prediction. As can be seen in Figure 2-23, process damping has a clear effect on the stability of the machining process, as a number of samples are machined stably at spindle speeds <1500rpm, where axial depth of cut exceeds the stable baseline depth predicted by the model without process damping.

Wang et al. [48] investigated cutter posture with a ball nose cutter in five-axis machining for the relief of chatter. When milling a freeform geometry, depending on the tool path, the cutter may not always be normal to the workpiece, therefore it is valuable to understand the impact of machining with the cutter at various angles relative to the component. They explored how changing the lead angle (in the direction of feed) and tilt angle (perpendicular to feed) affected feasible depth of cut. They produced a ‘posture stability graph’ seen in Figure 2-24, a variant on a conventional stability lobe diagram that

shows stable depth of cut for a given feed rate and spindle speed as contours, with tilt and lead angles as variables.

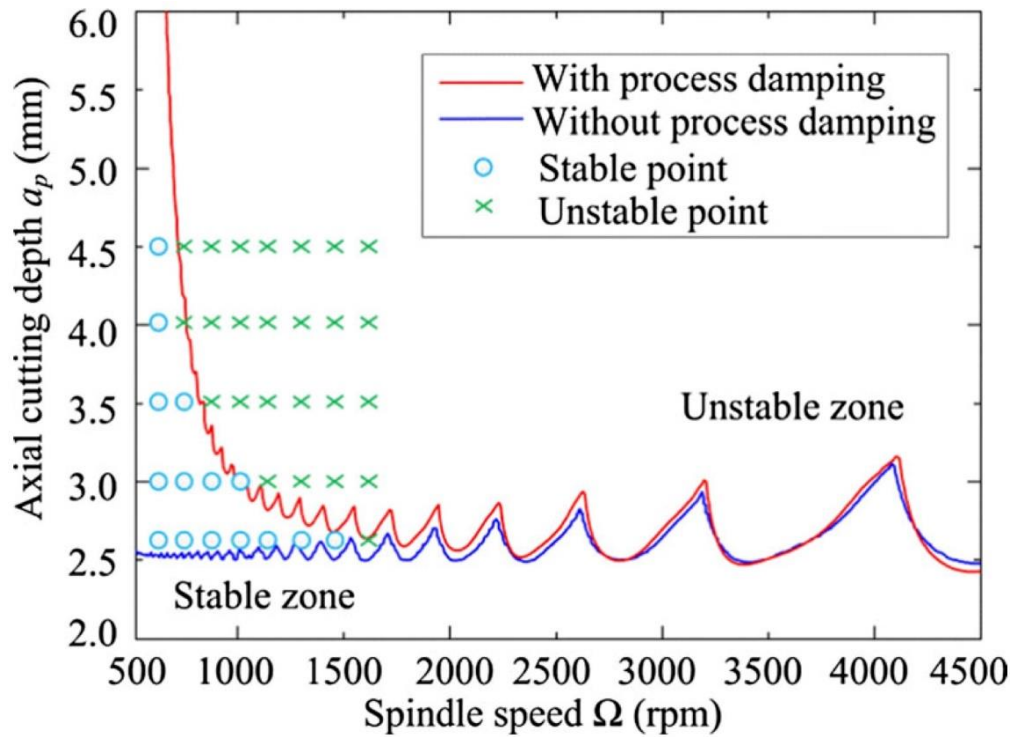


Figure 2-23. Comparison of stability lobes with and without consideration of process damping [47].

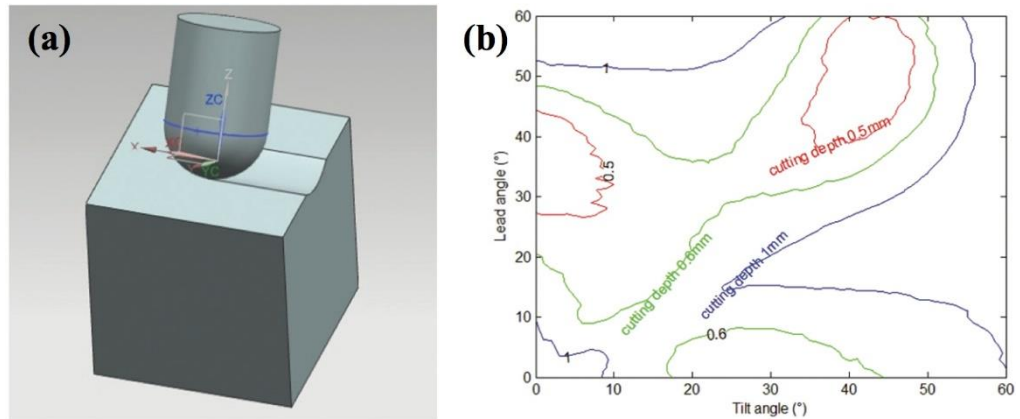


Figure 2-24. Illustration of non-rectilinear cutter posture (a), and plot of cutter posture-dependent stable depth of cut (b) [48].

Olvera et al. [49] investigated the mass effect of an accelerometer on a thin component (100×50×1mm, mass 15g), and therefore the error this can introduce when taking workpiece frequencies into account during calculation of machining stability. Using an accelerometer and a laser Doppler vibrometer,

they were able to compare the frequency response of thin walled components with and without the mass of the accelerometer. They found that even a miniature accelerometer (0.4g mass of the unit, 0.58g effective mass with hanging cable) noticeably shifts the natural frequency of the system compared to the non-contact measurement from the vibrometer, as can be seen from the FRF in Figure 2-25. The frequency shift caused by accelerometer mass effects was verified by measuring component frequency response using the vibrometer, while the accelerometer was still mounted to the workpiece. They showed how these errors in frequency response affect the stability plot, but demonstrated the ability to eliminate the effect of the accelerometer in order to provide true (accelerometer-free) frequencies of the system for their stability calculations by compensating for the added mass, and the damping provided by the accelerometer cable.

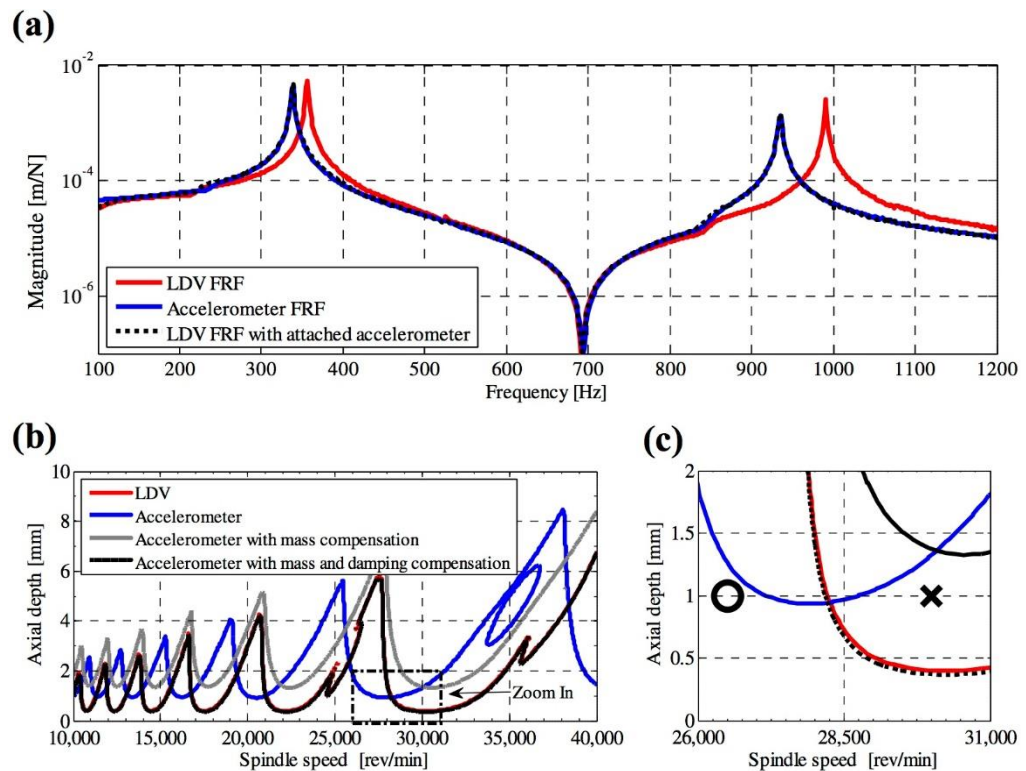


Figure 2-25. FRF (a) and stability lobe diagram (b) showing the effect of an accelerometer on the frequency response, and therefore machining stability, of a thin wall cantilever geometry, with a close-up of the stability plot over a small frequency range (c) [49].

Li et al. [50] developed a method for fast generation of a stability plot using only the FRF of the tool, without the need for long computations. The method is based on approximations, and involves a series of calculations that give the

critical points on each stability lobe (the spindle speeds and cutting depths where stability is at its minimum and maximum values), which can then be plotted on a graph and joined with parabolic curves to create a stability lobe diagram. As can be seen from Figure 2-26, the result is a very good approximation of the conventional analytical curve, and could in many cases prove sufficiently accurate when a quick indication of optimal spindle speed or baseline depth of cut is required.

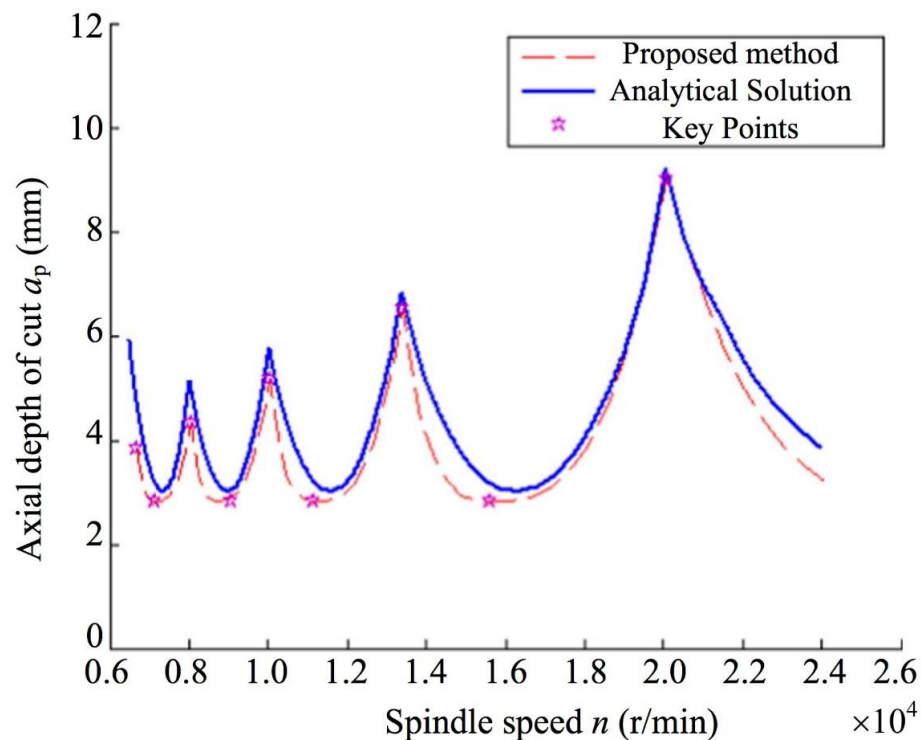


Figure 2-26. Comparison of fast approximation method, and conventional analytical method of stability calculation [50].

2.3.3 Summary

The dynamic responses of the workpiece, fixture, and tool can all influence chatter, and therefore determine whether a machining operation, such as milling, will be stable. Improving methods and computational power enable rapid generation of stability lobe diagrams based on the applicable input parameters, and mean that stability can be predicted with little effort. It is more common to focus on increasing the critical depth of cut, rather than altering the frequencies at which the system is stable and unstable. Stable cutting depth can be increased using a number of methods in order to allow more aggressive

processing, including varying the tooth pitch [36], helix angle [38], and posture [48] of the cutter. Application-specific techniques to improve accuracy of stability predictions include characterising the frequency response of a spinning tool [42], and incorporating process damping for cutting at low speeds [46]. In situations where conventional stability calculations are not possible, stability can be determined experimentally over a small speed range and extrapolated to obtain the necessary parameters for a full stability prediction [43], and stability can even be approximated with good accuracy by hand calculations [50].

While stability analyses use the properties of the tool as the main input, another factor not fully considered in a stability lobe diagram is the frequency response of the workpiece. If a workpiece is excited near to one of its eigenfrequencies, it will start to resonate, giving rise to workpiece chatter. Changes to the frequency response of a structure as a result of removing DOFs and subjecting it to external forces, for example a workpiece clamped in a fixture, has received relatively little attention in the literature, despite workpiece resonance making up one of the two possible causes of chatter. Understanding how the frequency response of a structure changes as it is constrained and loaded allows selection of machining parameters that do not cause resonance of the workpiece, and potentially facilitates the design of fixture systems that incorporate features to further minimise workpiece vibration during machining.

2.4 Milling of low-stiffness components

Building upon the previous section on vibrations and stability, this section focuses purely on developments relating to the machining of structures with low stiffness, particularly for thin wall geometries, including modelling of the machining process. Definitions of thin-walled structures vary, however any workpiece with a height-to-thickness ratio in excess of 5:1 is considered in this thesis to fall under the thin wall classification.

2.4.1 Innovations for passive improvement of machined component quality

Building upon the dedicated fixturing solutions presented in section 2.2, this section details auxiliary systems for improving the quality and accuracy of machined components. These tend to be solutions that can be incorporated into an existing workpiece-fixture system, usually to reduce vibrations of the workpiece.

Kolluru et al. [51] improved the machinability of a thin-walled nickel superalloy casing by damping its surface through the addition of neoprene sheet and tuned mass dampers affixed to its walls using viscoelastic tape, shown in Figure 2-27. In addition to providing damping, the system also improved mass properties (due to inertia) and stiffness characteristics (caused by stretching of the neoprene) of the casing. This is shown to effectively damp out the targeted frequency range corresponding to resonance of the casing wall, as well as neighbouring frequencies, which results in a reduction in machining vibrations of over 75%. Given the compact and unobtrusive nature of the neoprene and damper system, it is suitable for retrofitting in a milling operation that experiences unacceptable levels of vibration, over an entire complex workpiece, or a single troublesome region.

Later, Kolluru and Axinte [52] improved upon their design with a 3D printed self-actuating device based on pre-tensioned torsion springs that exert a radial force on the case to provide inertia, shown in Figure 2-28. One disadvantage they discussed about their previous solution was the time-consuming removal of viscoelastic tape from the casing after machining, however this solution replaces the tape with the spring mechanism to hold the neoprene in place.

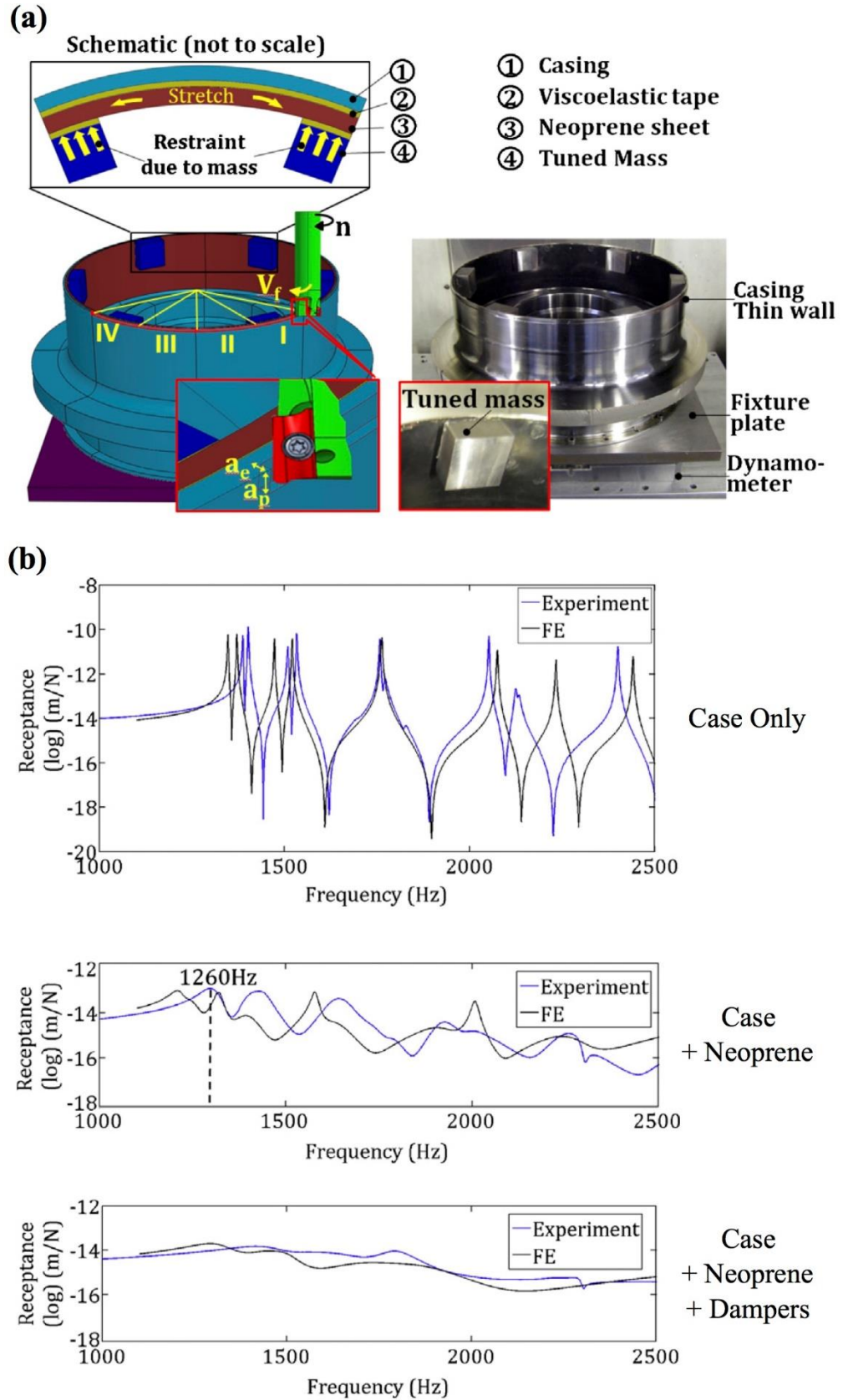


Figure 2-27. Addition of neoprene sheet and tuned mass dampers to a casing for reduction of machining vibrations (a), and FRFs showing suppression of mode shapes as damping was increased (b) [51].

The viscoelastic nature of the neoprene continues to damp the structure, and the spring mechanism provides stiffness during vibration, meaning mass, stiffness, and damping are all imparted to the casing. Dynamic response tests indicated a 65-fold increase in damping of the first mode, a 30% improvement over their previous tuned mass damper concept shown on the previous page. Machining results were similarly impressive, with an 8 times reduction in RMS vibration, due to the effect of stretching the neoprene on increasing stiffness of the workpiece. Like the previous solution, it does not restrict access to the casing, and can be adapted for any size and shape of hollow prismatic component by adding or removing segments to the six used in their example.

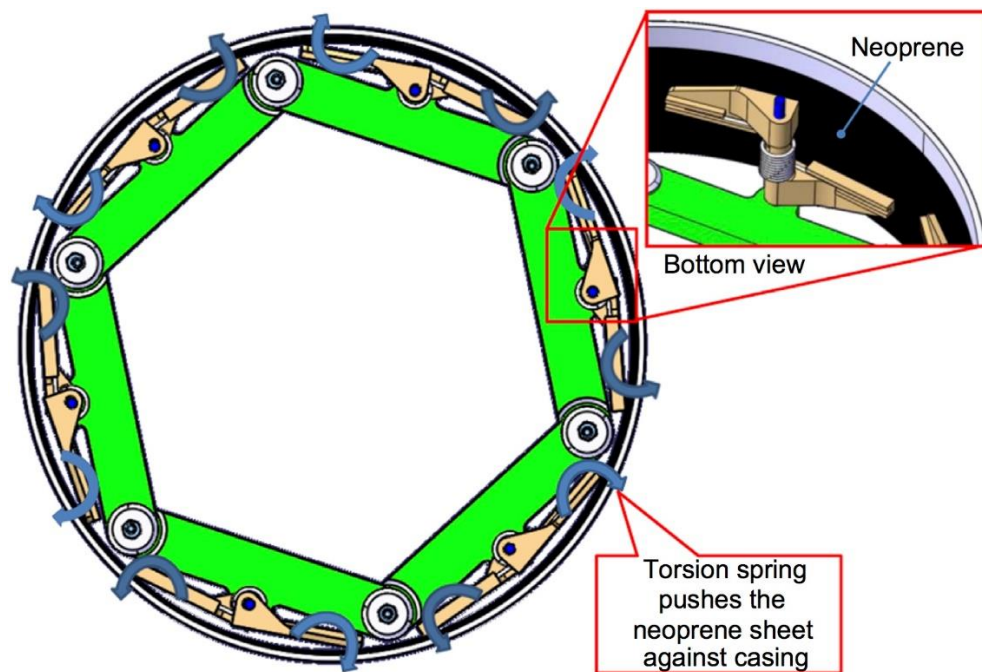


Figure 2-28. 3D printed ancillary device to improve the machinability of a casing [52].

Yang et al. [53] designed a lightweight damping device based on eddy current damping that can be mounted using adhesive to a component in a vibration-sensitive area in order to reduce vibrations. The damper, shown in Figure 2-29, uses a permanent magnet suspended between two springs so that it is able to slide up and down a rod. A conductive casing exploits the Lenz effect so that a relative motion against the magnet is generated when the damper is subjected to vibration, in order to produce a damping effect. They studied an aluminium box with wall thickness of 5mm, with hammer testing yielding a reduction in

magnitude of the first three natural frequencies by 67-91% when the damper was added. In milling, the damper was found to dramatically improve surface finish by reducing vibration in the workpiece, with the effect becoming more pronounced as feed increased. With all other parameters remaining constant, surface R_a at 400mm/min improved from 0.49 μ m without the damper to 0.15 μ m when it was added, and at 1200mm/min it was improved from 4.76 μ m to 0.53 μ m. Addition of the damper was shown by means of a machining stability analysis to increase baseline critical depth of cut from 3.5mm to 11.1mm. When other parameters were maintained but spindle speed was varied, the damper was found to reduce vibration by 34 to 51% in stable milling conditions.

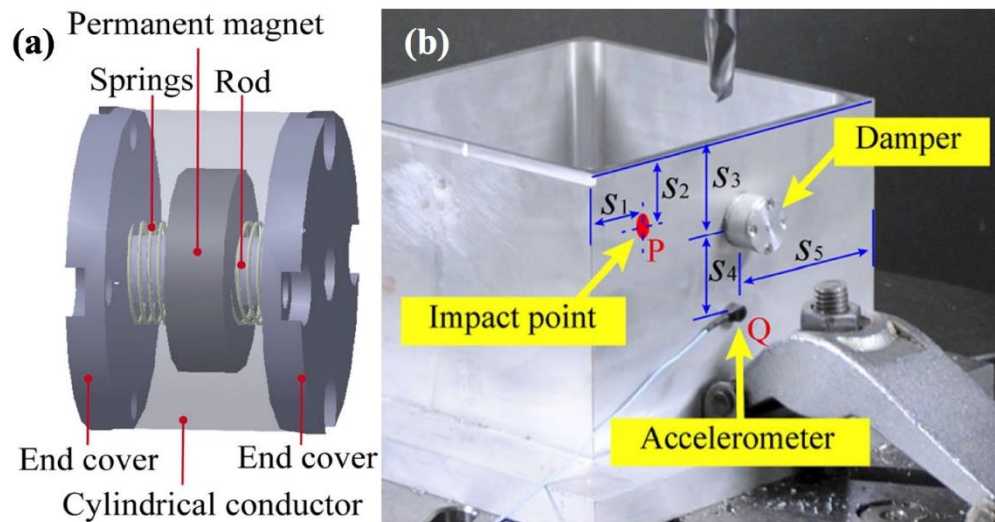


Figure 2-29. Schematic (a) of compact eddy current damper, shown in situ on a thin wall workpiece (b) [53].

Yang et al. [54] later created a similarly compact damping system for mounting on a thin wall based on a tuneable low-stiffness mass. Shown in Figure 2-30, the damper features a ring with an internal mass suspended from low stiffness supports, and can be rotated to tune the dynamic effect on the surface to which it is mounted. The angle of the damper mass block relative to the workpiece determines its relative stiffness, which controls the frequency at which the mass resonates. Adding the damper to a thin wall is shown to have a large effect on its frequency response, resulting in the first mode of the undamped structure (680Hz) being split into two modes, one with a higher frequency and one lower, with an antiresonance at the undamped first mode frequency. Depending on the

orientation of the mass damper, magnitudes of damped vibrations range from 9.9-1.3% of the undamped levels. In machining experiments with the damper at the optimal orientation, vibration was reduced by more than 75%, with an 80% increase in stable depth of cut, and surface finish improvements of more than 80% for both up and down milling.

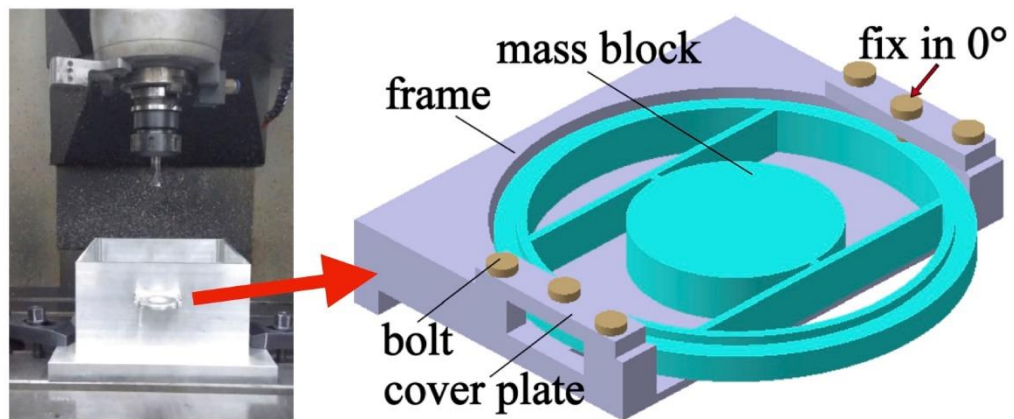


Figure 2-30. Damper with rotating element for tuneable stiffness [54].

Matsubara et al. [55] developed a novel method of supporting a thin cantilevered component by touching the free end of the workpiece to a pivoting mechanism, shown in Figure 2-31. The pivot could be fixed in position or allowed to rotate freely, and was pushed into contact with the cantilever with forces of 1-50N. The pivoting system was shown to offer superior vibration suppression, and reduced chatter susceptibility, with lower contact forces resulting in improved damping behaviour.

Kalinski et al. [56] developed a system for tuning the workpiece natural frequency by altering the stiffness of workpiece holding, thereby allowing resonance to be avoided. Workpiece holding is generally designed to be stiff, however by designing a holding mechanism as a 1-DOF system with tuneable stiffness, shown in Figure 2-32, it is possible to alter workpiece frequency response, permitting machining at a greater range of speeds, through suitable adjustment of fixture properties. Increasing the stiffness of the fixture for an example component changes its natural frequency from 86Hz to 139Hz, which permits optimal machining at speeds between 10000rpm and 17000rpm – easily allowing processing at a peak in stable milling depth of cut for a typical tool.

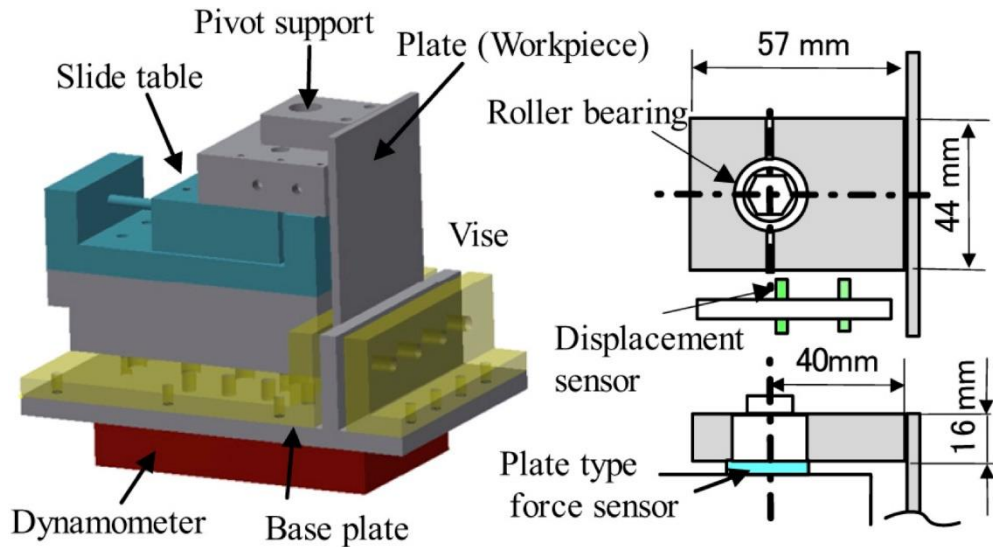


Figure 2-31. Pivoting support for milling a cantilever workpiece [55].

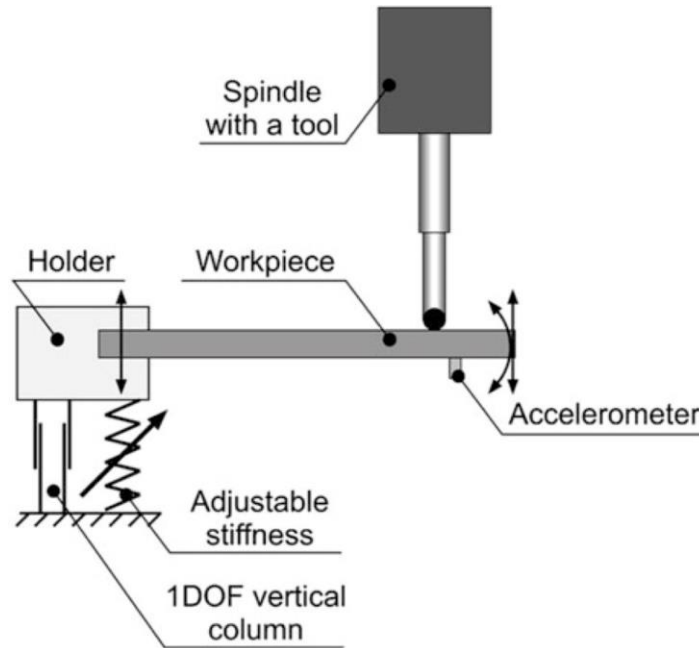


Figure 2-32. Variable stiffness workpiece holder for alteration of frequency response [56].

Ma et al. [57] attempted to suppress vibrations by submerging a thin workpiece in magnetorheological fluid, which acts to oppose all movement when the system is powered, creating a damping effect. The strength of the magnetic field was regulated via the current supplied to it, with a 2A current generating a 0.5T field that is shown to reduce the amplitude of the first mode, and machining vibrations, by approximately 50% compared to the deactivated fluid, which provides negligible damping in the absence of a magnetic field.

2.4.2 Simulation of machining operations

Finite element modelling allows the machining process to be modelled in great detail, with the main constraint being computational expense. It enables the prediction and study of simple cutting interactions, changing conditions along the tool path due to material removal, and also stress and deformation in the workpiece.

Budak and Altintas [58] produced an early FE model of milling thin-wall plates to determine form errors. They milled cantilevered titanium structures with wall thicknesses of 2.45mm and 6.3mm with a 35mm axial depth of cut. Two methodologies were used: a rigid model that neglected the effect of tool and workpiece deflections on cutting force; and a flexible model that took deflections into account. Despite the primitive nature of the model, cutting forces were simulated accurately using the flexible model. Surface errors were also predicted effectively throughout the cut, enabling scheduling of feeds along the tool path in order to keep form errors within acceptable limits.

Ratchev et al. [59] developed a modelling approach for thin components that adjusted for workpiece deflections resulting in tool path optimisation to minimise errors. A schematic is shown in Figure 2-33. By modelling flexible cutting forces that accounted for material properties of the tool and workpiece, static deflections of the workpiece could be predicted, allowing compensation in the tool path to avoid under- or over-removal of material. This was verified using a dynamic force prediction simulation of milling interactions at discrete points along the tool path, and updated as necessary. Cutting force predictions were shown to have error of approximately 8% based on experimental validation in cutting of aluminium cantilevers with a 5mm wall thickness.

Building upon this, Rai and Xirouchakis [60] designed a machining simulation environment that integrates with APT files from CAM systems for automated process planning. It can aid with fixture selection, and carries out transient thermomechanical milling simulations taking residual internal stresses into

account, allowing for complex milling operations involving multiple tool passes to be modelled.

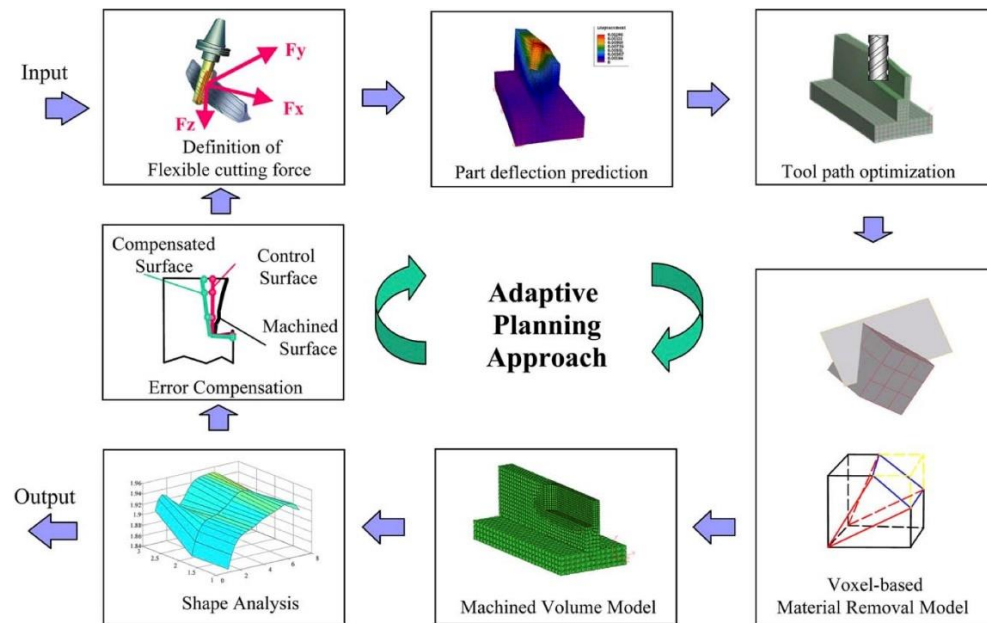


Figure 2-33. Process diagram for minimisation of errors in milling of flexible components [59].

Fixtures are often assumed to be rigid, and in FE modelling it is common practice to replace fixturing elements with immobile boundary conditions to reduce the complexity of the analysis. Wan et al. [61] identified deformation of locating elements as a significant source of error in the machining of high stiffness components, and the main factor influencing machined accuracy. They created a system that used a simulation of the fixture-workpiece system to adjust the tool path to compensate for errors, as shown in Figure 2-34. A key advantage of their method is that it uses the outputs of FE models to automatically modify the tool path sent to the machine, with no need for processing of data by the user, or manual modification of cutter location.

Kersting and Biermann [62] examined techniques for modelling displacement of a turbine blade during milling. They investigated the conditions that cause the regenerative vibrations that give rise to chatter, and used an FE model to visualise the machined surface of the blade in stable and unstable conditions, as seen in Figure 2-35. While FE was able to accurately predict the presence of

chatter, and its corresponding effect on surface finish, the process was computationally inefficient, due to the complexity of mesh modification and re-meshing. It took a long time to model a small region of the blade, therefore making it unsuitable for the modelling of a roughing process where detail is required over the entire blade surface. They therefore proposed use of an ‘oscillator-based’ simulation, which is based on the modal properties of the blade, requiring FE-generated FRFs at different locations and time steps on the blade. Although a large amount of input data is required, this model is much more efficient, resulting in a quick, detailed simulation of a finishing process over a large area of the workpiece. It also enabled accurate simulation of a roughing process that was impractical for a purely FE-based model, due to excessively long computation times.

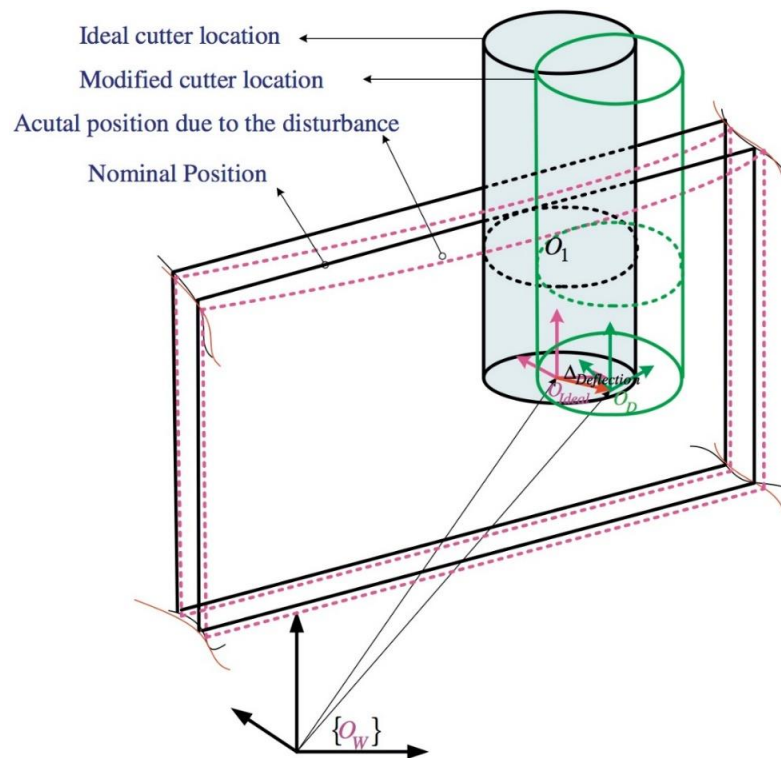


Figure 2-34. Compensation of tool location for minimisation of machining errors [61].

Yang et al. [63] modelled the in-process dynamic evolution of thin wall geometries with curved surfaces, and developed a methodology for quick calculation of the stability based solely on the initial dynamic response of the workpiece, requiring only one modal analysis of the system. This is achieved

by ‘softening’ the mesh after the cutter has passed that location instead of removing material, which would require the part to be re-meshed, demonstrated in Figure 2-36. The regions which are to be removed by the cutter simply have their mass and stiffness properties reduced after the tool has passed by, creating the same effective conditions as material removal, but without the need for re-meshing of the model. The tool path is discretised, saving computational expense by allowing interpolation over regions where little change occurs, and enabling near-continuous steps as small as one element where the geometry is more complex. Stability lobe predictions were verified experimentally, and found to accurately predict stable depth of cut, despite considerable savings in computation time. Results showed how a dominant eigenfrequency shifts from 1900Hz to 2500Hz throughout the cutting process, therefore demonstrating the importance of considering the time-dependent nature of structural dynamic response.

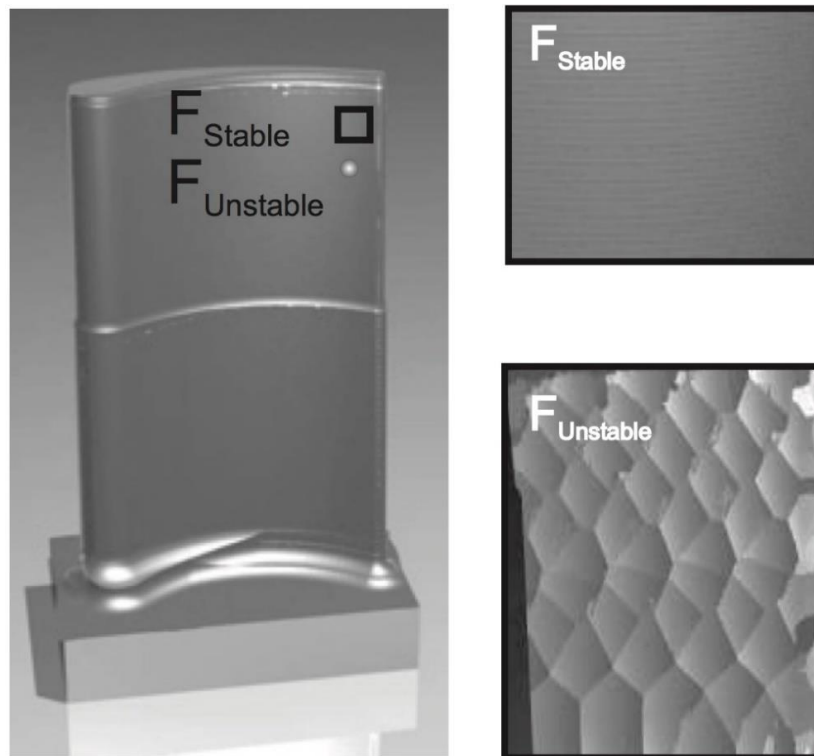


Figure 2-35. Simulated stable and unstable machined surfaces of a blade [62].

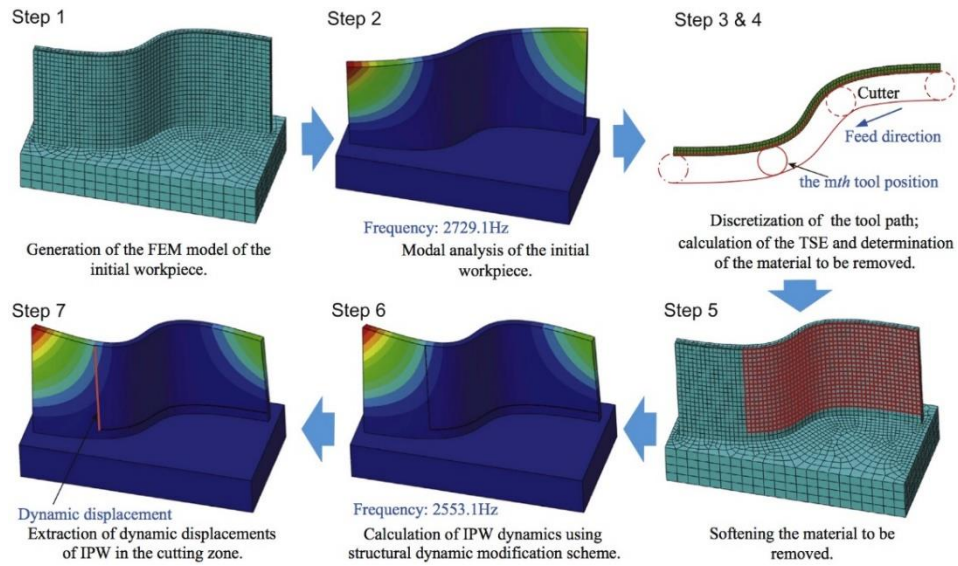


Figure 2-36. Evolution of the dynamic behaviour of a curved thin wall during end milling [63].

Dong et al. [64] created an FE model for milling where locators were modelled as spring-damper systems, in order to improve the prediction of errors arising from workpiece-fixture deformations. Figure 2-37 shows how both static deformation of the workpiece arising due to clamping forces at locators, and dynamic deformation due to spring-back caused by cutting forces, can result in removal of more or less material than intended.

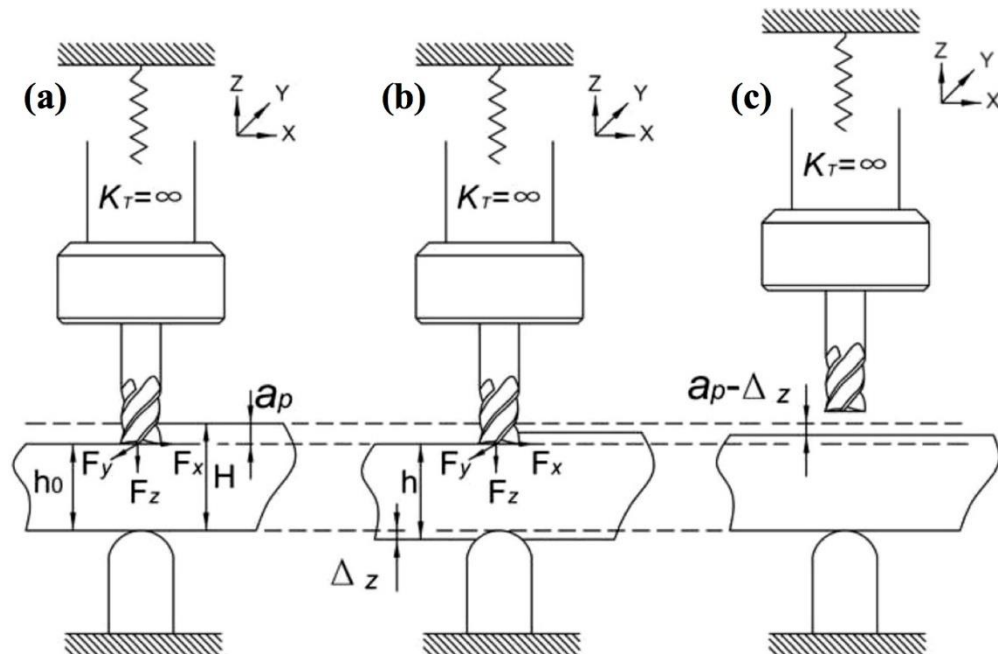


Figure 2-37. Influence of workpiece deformation at locators on machined surface errors: ideal cutting (a), under-cutting due to deformation at locator (b), and variable cutting due to cutter spring-back (c) [64].

2.4.3 Summary

Geometries of low stiffness are some of the most challenging to machine, due to their susceptibility to deformation and vibration under machining loads. However, modelling the machining operation using FE and analytical/numerical methods allows the design of processes which produce accurate and consistent components that are free from significant errors. Simulations have been used to account for deflections of the tool and workpiece [58], deformation of locators in the fixture [61], and to automatically adapt tool paths accordingly [59]. It has been possible not only to predict the onset of chatter, but also to visualise machined surface to determine the surface finish for given machining parameters [62]. The perpetually increasing power of computer hardware, along with innovative techniques to reduce computation time, for example dynamic altering of mesh properties during cutting to eliminate the need for re-meshing, and the discretisation of machining stages to combine high resolution outputs at critical stages with interpolation between sparse points at non-critical stages [63], mean that it is becoming increasingly feasible to simulate every stage of even the most complex machining processes.

Innovative systems and devices have been developed to improve the quality of machined thin walls, usually by providing damping to the structure, with examples including tuned masses [51] and magnetic eddy current dampers [53]. Although these systems are designed to be compact, their ability to be reduced in size is often limited by the need to achieve a certain mass (for tuned mass damping), or resonant frequency (for energy dissipating vibrating structures). Common to all of these systems is that they are external devices, necessitating their installation and removal as an additional stage of production, and also potentially restricting tool access. No systems presented in the literature incorporate these damping measures directly into the fixtures, however the potential advantages of their integration include simplification of the manufacturing process, leading to faster production and greater consistency of machined components.

2.5 Additive manufacture

Since this work aims to produce bespoke fixtures with numerous complex features and freeform surfaces, the best-suited technique is additive manufacture (AM), commonly also referred to as *rapid prototyping* or *3D printing*. This section undertakes a review of the fundamental capabilities of AM processes, the influence of printed component orientation on its performance and cost, and select innovations that have enhanced the capabilities of AM by augmenting printed components.

AM is a recent innovation, in contrast with subtractive (grinding, milling) and formative (moulding, casting) manufacturing processes which have evolved into their modern forms over thousands of years. Additive processes can create components in metals, polymers and minerals, with numerous systems available commercially. In 2014, AM made up an estimated \$667m out of \$11.8tn in global manufacturing value added [65], but while it constitutes only a tiny fraction of all manufacturing processes today, it generates considerable research interest, and revenues are expected to increase tenfold by 2030. This is due to ever-improving process capabilities, along with falling costs that will continue to decrease as adoption becomes more widespread, resulting in greater economies of scale.

2.5.1 Process capabilities

AM is unique among manufacturing techniques because it allows components to be produced without consideration of the principles of *design for manufacture*, as the designer can create features and geometries with no need to allow for workpiece holding, tool access, or even assembly. Vayre et al. [66] provide a useful example of how the design of a simple metallic component can change when CNC (computer numerically controlled) manufacturing is replaced with an additive process. While an additive process is capable of manufacturing the conventional bracket shown in Figure 2-38, by considering only the features needed and loads to which it is subjected, the design with topology optimised for AM looks dramatically different.

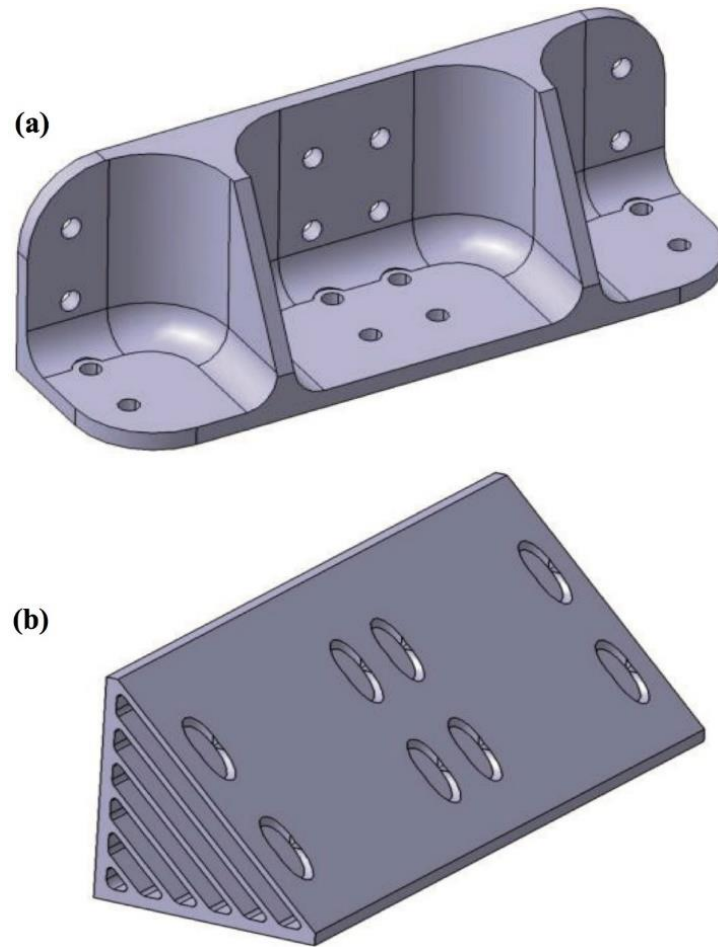


Figure 2-38. Comparison of conventional bracket designed for manufacture by CNC machining (a), and alternative design optimised for additive manufacture (b) [66].

Topology optimisation can be used to reduce the geometry of a component to use the minimum possible quantity of material in order for it to perform a particular function. Combined with AM, this exact geometry can be printed directly, as demonstrated by Aremu et al. [67], who optimised an example component for a particular loading case. The printed geometry in Figure 2-39 differs radically in appearance from the conventional design, despite being able to carry out the same function.

The flexibility of AM techniques means they are ideally suited to the creation of intricate geometries such as lattices and matrices that include many elements with small variations in their properties, such as the examples in Figure 2-40. Yang and Zhao [68] reviewed AM-enabled design theory, discussing the implications of AM for manufacturing, assembly, and performance of

components in research, identifying a divide between true design for AM, and conventional design methodologies which are modified for AM. They suggest that combining functional requirements and process knowledge simultaneously in the design process will lead to improved components that exploit the full capabilities of AM, by reducing reliance on design principles that are irrelevant to AM, such as manufacturing and assembly considerations.

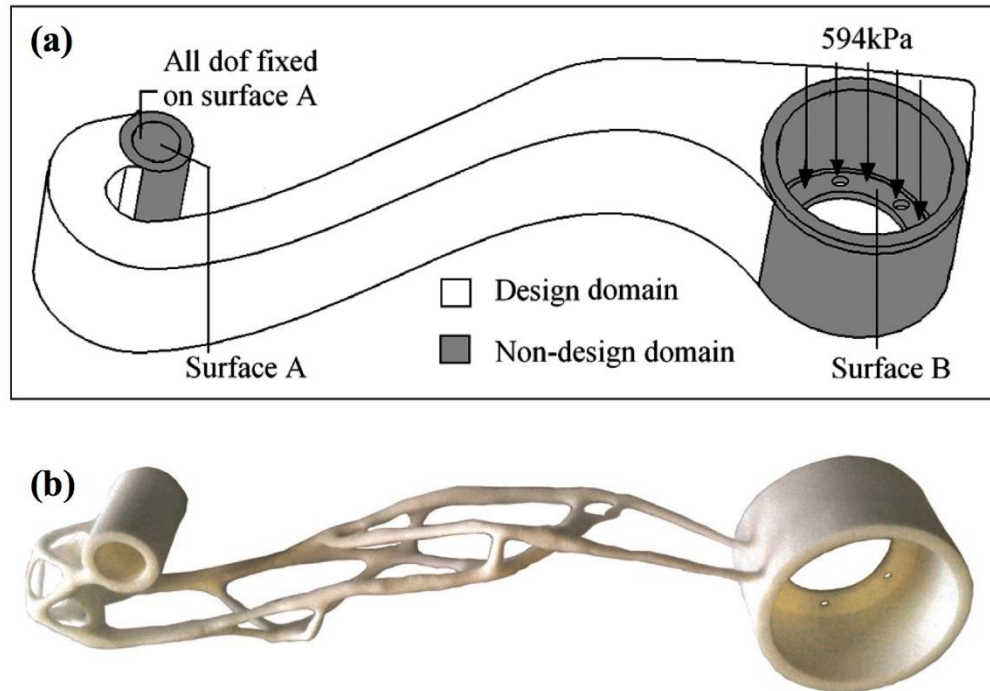


Figure 2-39. Initial component design with key functional requirements (a), and topologically optimised component produced using SLS (b) [67].

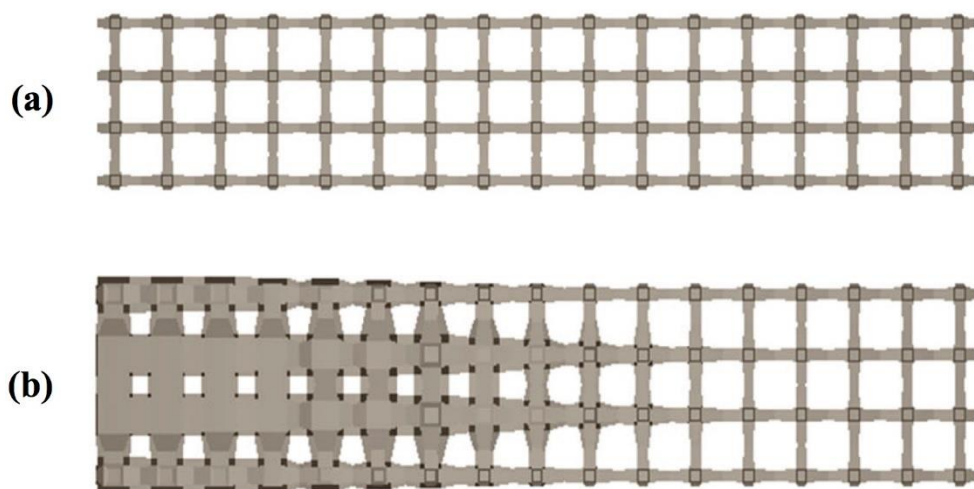


Figure 2-40. Lattice components for AM with homogeneous (a), and graded heterogeneous (b) structures [68].

While machining is ideally suited to the production of components with high accuracy and modest complexity, with AM there is no cost directly associated with additional complexity, since the cost and time required to produce a component depends only on its physical dimensions (volume or z height, depending on the process), rather than its intricacy. However, this flexibility comes at the expense of accuracy. In addition to fundamental post-processing, such as the dissolution or cutting away of supporting features, depending on the application it may be necessary to machine surfaces flat or parallel, or to drill out holes to ensure roundness, in cases where an AM process is unable to achieve the tolerances required of a particular component. To reduce the need for avoidable reworking or post-processing, methodologies of *design for additive manufacture*, such as those produced by Guido and Zimmer [69] are being published with increasing frequency. They highlight the difficulty in three common AM processes – selective laser sintering for various polymers, selective laser melting for various metals, and fused deposition modelling (FDM) for ABS plastic – of realising certain geometries, such as sharp corners/points, overhangs, narrow gaps, and transitions where features meet. They propose guidelines on how challenges in realising these features can be overcome without affecting the overall properties of the component, for example by increasing the surface area of regions which, due to the orientation and slicing of the model, are initially printed as two distinct features, but become joined at a later stage in the build process as the model reaches its full height.

The practice of post-process machining of a component produced by AM is a typical example of hybrid manufacturing, in which multiple operations or machines are used on a single component. This allows the advantages of two or more techniques to be combined, while minimising their respective disadvantages, for example milling of locating surfaces on a complex AM component to enable accurate assembly. Newman et al. [70] devised a hybrid process planning system that combines additive and subtractive manufacturing processes with inspection techniques for repurposing existing components, of which a schematic is given in Figure 2-41. The methodology involved direct deposition of net shape material onto an existing component, which could subsequently be machined away in order to produce a component with different

specifications. Components with out-of-tolerance features, or a previous design from the desired final component could therefore be remanufactured into the required specification, thereby preventing the scrapping of the original component.

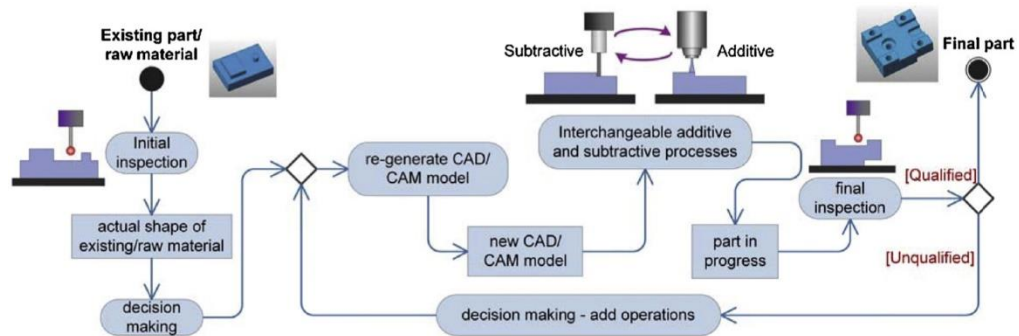


Figure 2-41. Flow diagram of a hybrid manufacturing system suitable for the repair, revision, or recycling of components [70].

2.5.2 Influence of build orientation and its optimisation

Since there is no need to consider workpiece holding or tool access for the majority of AM components, except in cases where manual post-processing is required, it is generally possible to print them in any orientation. As is shown in this section, build time and cost, as well as strength, accuracy and surface finish of the component are all influenced by their orientation. Determining the optimal build orientation usually involves finding a compromise between all of these factors, and solutions have been proposed to generate optimal orientations based on the priorities of the user. Competing AM techniques have different advantages and disadvantages, meaning the optimal orientation for a particular process may prove unsuitable for other additive processes.

Cheng et al. [71] were the first to consider optimisation of component orientation in AM to reduce errors. They considered errors inherent to any AM process, e.g. loss of accuracy due to approximation of component features with a triangular mesh in the STL file, and the loss of detail and surface integrity from the ‘staircase’ effect when inclined surfaces are sliced into horizontal layers for the build process; as well as those specific to the stereolithography process, e.g. ‘overcuring’ where resin beneath an overhanging feature is

unintentionally cured due to penetration by the laser beyond the desired depth, and the ‘container effect’ where uncured resin is trapped inside internal geometries of the component, creating surface tension which can lead to distortion. From this they created guidelines, including to maximise the number of horizontal surfaces, and maximise the number of holes whose axes were in the slicing direction, and weighted each factor so that contradicting guidelines could be resolved into an optimal orientation. To automate the process of determining best build orientation, Masood et al. [72] used a genetic algorithm to determine volumetric errors in the component at different rotations about a chosen axis, enabling errors to be quantified for a given layer thickness, aiding selection of the optimal build orientation. They verified their algorithm experimentally by testing components printed using FDM.

Processes such as FDM are increasingly able to manufacture functional components, meaning the mechanical properties of printed components must be considered alongside geometric accuracy when selecting a build orientation. The strength of printed components is often therefore the most important factor in determining build orientation, due to the anisotropy of AM processes. Keles et al. [73] printed a number of flat tensile testing specimens in ABS using FDM, comparing the fracture strength of three unique build orientations, shown in Figure 2-42. The first orientation printed the two-dimensional profile of the tensile specimen in the xy plane, meaning each printed layer was identical, with the number of layers printed in the z axis determining the specimen thickness. The second orientation was a rotation of the first by 45° about the z axis of the system, so that the two samples had raster angles relative to the loading gauge of $\pm 45^\circ$ and $0^\circ/90^\circ$ respectively. The third orientation rotated the first sample by 90° about the x axis, meaning the 2D profile of the system was built up in the z direction so that each printed layer was different. Some of the samples included an intentional defect in the form of a hole printed in the gauge length of the specimen, and the scanning electron microscope images in Figure 2-42 show the effect that orientation can have on the characteristics of the hole.

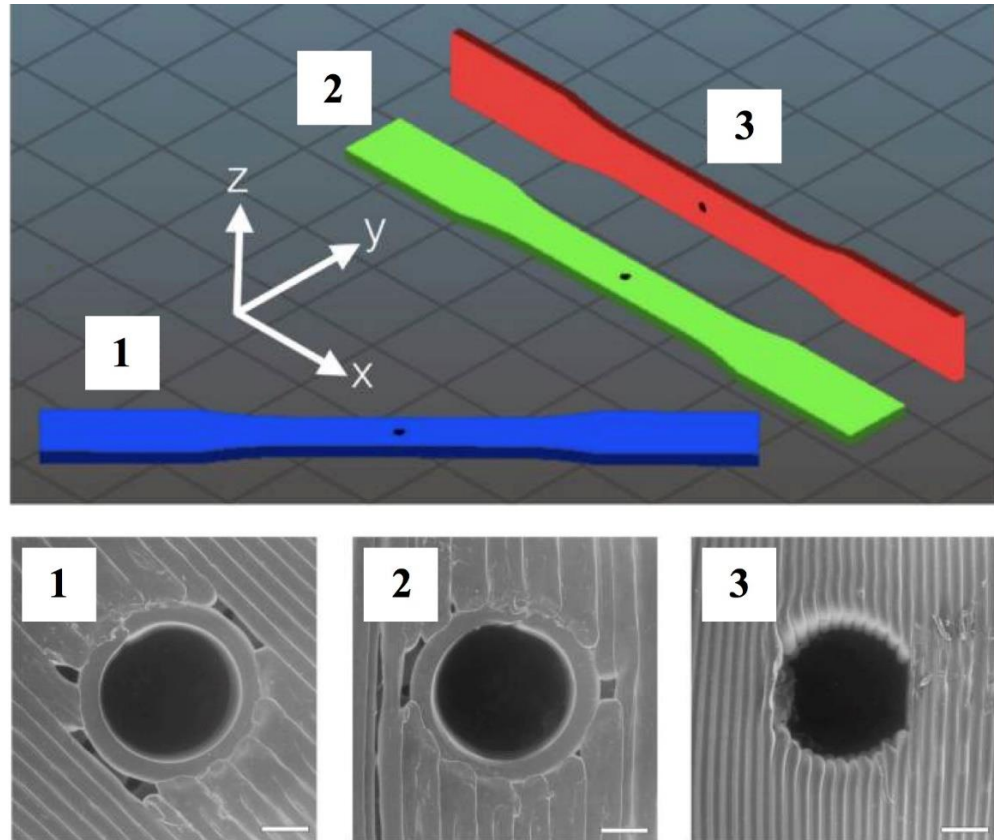


Figure 2-42. Comparison of the same hole feature realised using three different build orientations in FDM [73].

The third orientation exhibited the highest fracture toughness and strain to failure, because the printed layers are parallel to loading with a higher ratio of printed outer skin to internal raster material in the sample, which gives the specimen greater longitudinal tensile strength. While this is shown to be the optimal orientation for tensile strength, it is significantly worse than the other orientations for geometric accuracy, due to the staircase effect present on rounded surfaces including the circular hole, and the taper from the grip region to the gauge region. This exemplifies the challenge involved in selecting the orientation of a printed component, as there is no single optimal orientation, and trade-offs between strength and accuracy are therefore necessary.

Di Angelo et al. [74] studied geometric errors in FDM components due to the staircase effect (illustrated in Figure 2-43), with a view to optimising component orientation for improved surface finish. Since R_a intrinsically filters the long wavelength component of printed ‘steps’ of material in favour of more localised

defects, they proposed P_a as an improved index for measuring printed component surface quality, and used a model to accurately predict P_a for any angle of deposition.

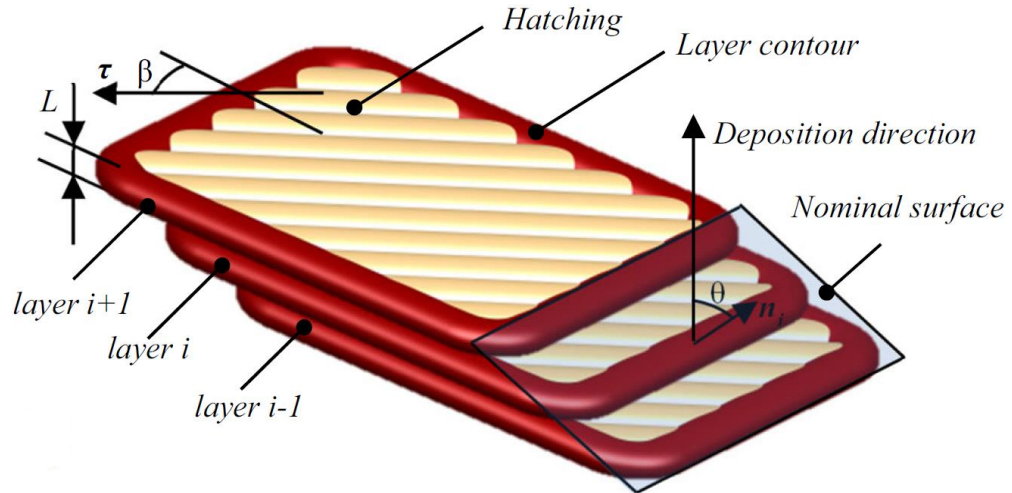


Figure 2-43. Staircase effect on an oblique printed surface in FDM, highlighting the nominal surface profile [74].

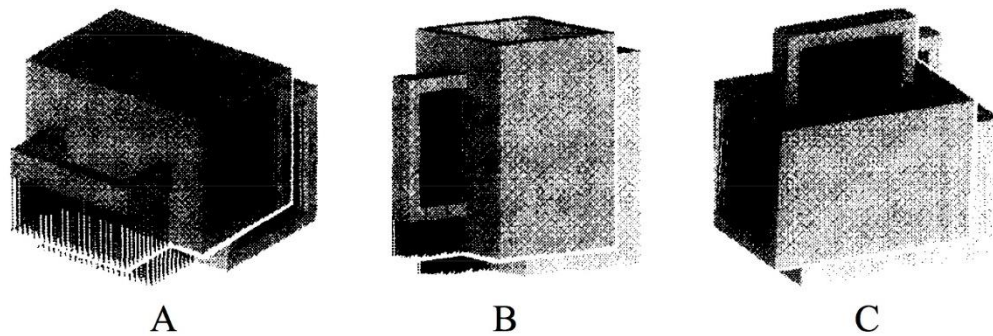
An established method of improving surface quality in FDM components is to chemically treat them using dimethyl ketone (acetone). Exposing components to acetone vapour or submerging them in a dilute acetone solution softens the surface of the ABS, allowing small flow between layers, which fills cavities and smooths ridges in the geometry. This technique visibly improves the surface finish, and is commonly employed in artistic or ‘novelty’ applications of AM. Garg et al. [75] experimented with the effect of acetone exposure on the strength of printed ABS, by measuring the tensile and flexural strength of treated components printed in various orientations. They showed acetone treatment to improve surface roughness by up to 98.6%, with the R_a value of a component built with a raster pattern at 30° to the flat surfaces (such that it exhibited significant staircasing) decreasing from $15.39\mu\text{m}$ before exposure to $0.220\mu\text{m}$ after. Acetone was found to have a slight negative impact on flexural strength, due to softening of the outer layer of ABS, which reduced the stiffness of the upper and lower surfaces. Tensile strength was shown to improve under certain conditions, however. The z direction of printed components (normal to build layers) generally has low tensile strength compared to other directions, due to the relatively poor adhesion between layers, however the application of acetone

was demonstrated to improve this adhesion, resulting in an improvement in tensile strength of approximately 75%. This meant a component oriented with its gauge length in the z direction exhibited tensile strength indistinguishable from components whose gauge length was parallel to x and y. This finding has significant ramifications for build orientation in FDM, as the use of acetone can potentially reduce the compromise required: a weak orientation with good dimensional accuracy can be strengthened, and an orientation with poor appearance but high strength can have its surface finish improved.

Alexander et al. [76] were among the first to consider selection of build orientation to minimise cost, and compared the costs of printing the same component using two competing AM processes: stereolithography and FDM. A simplified cup component, shown in Figure 2-44, was printed in three orientations. It was designed to be rectilinear, so that all surfaces of the component aligned with the axes of the machines, meaning the orientation chosen would have minimal impact on component accuracy. The results highlight the different factors that affect cost between the two processes: in both cases orientation C was the most expensive to produce, due its large z height (SLA only) and significant volume of internal supports due to overhanging geometries (SLA and FDM). For stereolithography, orientation A was cheapest due to its shorter height, despite using a greater volume of printed material than B. This is because orientation B has a higher material cost, as a greater quantity of (non-reusable) resin is required to realise the full height of the component. For FDM, orientation B was cheapest, simply due to the reduced overhanging geometry compared to the other orientations, which required less support material. Unlike resin bath processes, build height does not directly influence material cost in FDM, as only the required material is printed, with no waste of resin to consider.

In [75], the impact of component orientation on build time and material consumption was also studied, revealing a significant difference in build time between orientations, shown in Figure 2-45. More surprisingly, there is also a difference between orientations in the volume of material used of more than 10%, even though the input model was the same in each case. It is expected that

some orientations would require more support material than others, however this variation in model material consumption (which is independent of support material) indicates that orientation can determine the fraction of voids within the component, and also suggests that orienting a build with planar surfaces that do not align with the axes of the machine can result in a low-resolution realisation of the geometry. This reinforces just how significant orientation selection can be for the accuracy and strength of the printed component.



	Orientations					
	A		B		C	
	FDM	SLA	FDM	SLA	FDM	SLA
Total prebuild cost (\$)	10	10	10	10	10	10
Total build cost (\$)	55	58	46	81	60	84
Total post processing cost (\$)	28	34	16	32	43	35
Total cost (\$)	93	102	72	123	113	129

Figure 2-44. Comparison of 3D printing cost using two processes for the same component, printed in three orientations [76].

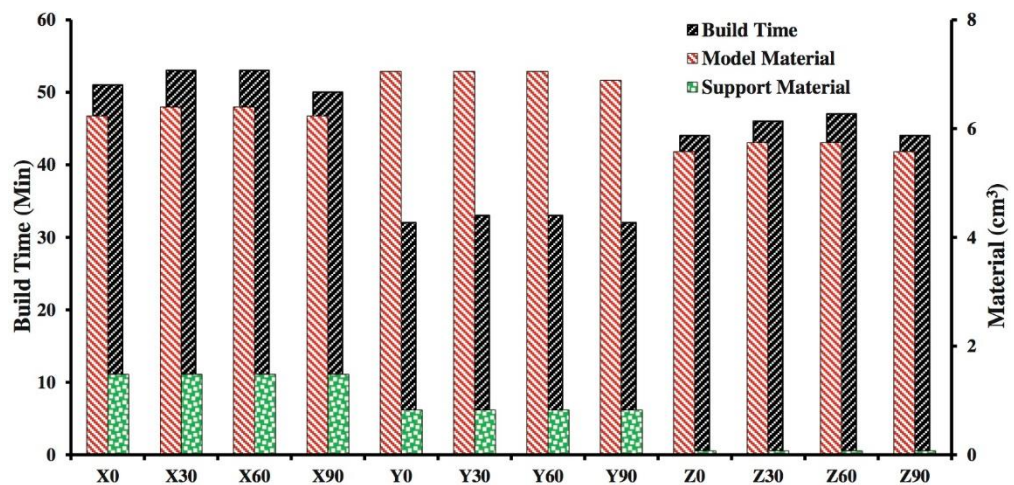


Figure 2-45. Comparison of build time and material consumption for various build orientations in FDM [75].

While commercial FDM machines use a separate soluble material for support structures, it is common in some AM processes, particularly powder bed or resin-based processes, to use the same material for both the component and its supports. This means support structures must be removed manually, and therefore minimising the presence of support structures in the build can significantly reduce the time (and therefore cost) required to post-process the component. Since the cost of metallic powders for AM is an order of magnitude higher than polymer powders and filaments, reducing the volume of support structures in a component also has a greater impact on its total cost. Therefore decreasing the need for supports is likely to be a higher priority when selecting the orientation for a metallic component. Morgan et al. [77] generated an optimisation algorithm using MATLAB to determine the optimum orientation to minimise support material used, which was consistently able to reduce material consumption compared to the orientation suggested by pre-processing software in a Renishaw selective laser melting (SLM) system, therefore providing a useful cost-saving tool for selection of build orientation.

2.5.3 Innovative applications of AM, and printed component enhancement

This section includes examples of novel structures and solutions realised using additive techniques, including the augmentation of AM components with foreign materials, and treatments for synergetic improvement of printed components.

Kantaros et al [78] produced periodic cellular scaffold structures using FDM, with geometry typical of orthopaedic implants used to encourage fusion with bone. Three open cell designs were used, shown in Figure 2-46, with identical volume fractions, based on body-centred and face-centred cubes, as well as a novel design based on rings. The compressive strength and shear stress data obtained experimentally showed good agreement with an FE model, with the body-centred cubic scaffold exhibiting superior strength.

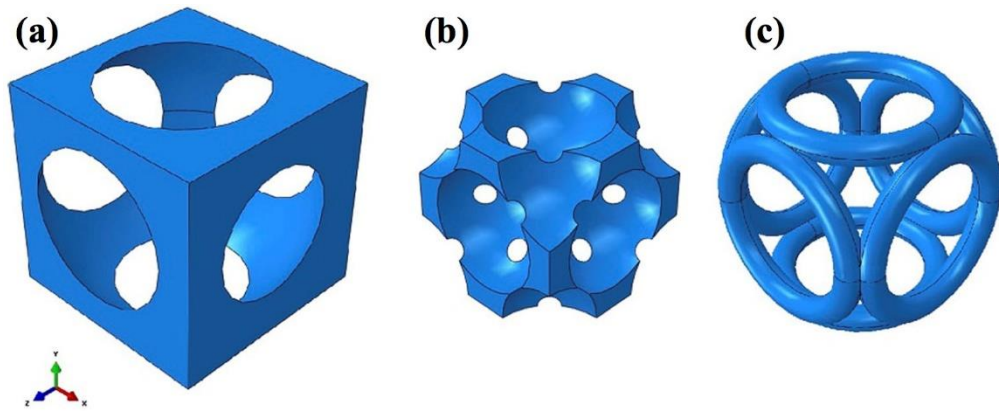


Figure 2-46. Unit cell geometries for compressive testing of cellular printed structures: body-centred cubic (a), face-centred cubic (b), and ring (c) [78].

Krznar et al. [79] used a custom 3D printed fixture for use with a 3D scanning system for reverse engineering. The system is advantageous for mounting delicate geometries, particularly those which cannot support their own weight, on a turntable. Figure 2-47 shows a custom printed fixture for a drill housing.

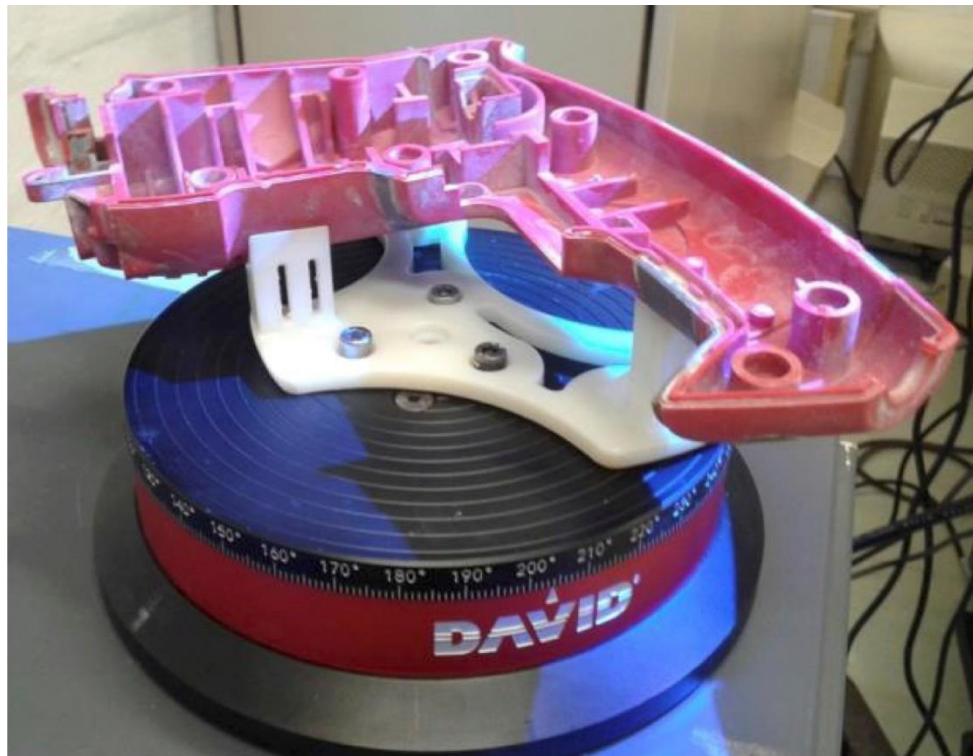


Figure 2-47. Custom 3D printed fixture for 3D scanning [79].

Gameros et al. [80] devised a partially-AM fixture concept based on a hydraulically-actuated diaphragm, shown in Figure 2-48. Produced by SLM in

titanium, it was found to be capable of generating clamping forces in excess of 1000N, and demonstrated to be suitable for holding a component during a grinding operation. Despite the suitability for fixturing workpieces with a range of complex geometries, the system is of high cost and complexity, due to the need for extensive post-processing of the printed components in order to accommodate the hydraulic elements.

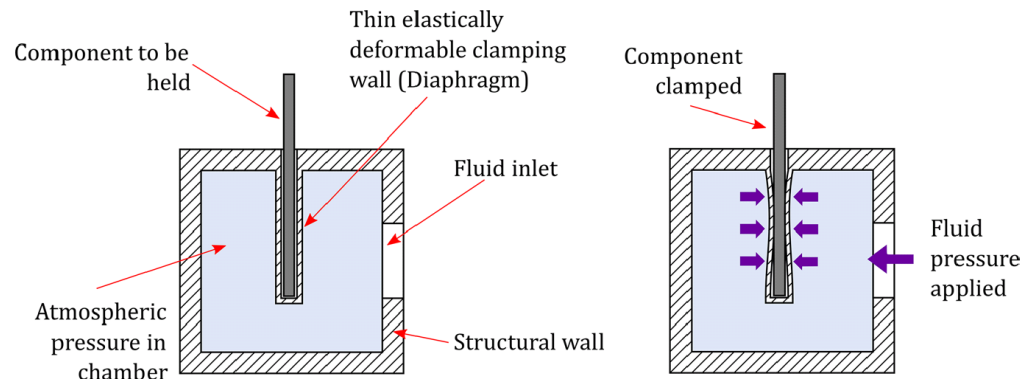


Figure 2-48. Partially-AM machining fixture based on hydraulically-actuated diaphragm [80].

Esfahani et al. [81] developed a novel method of reinforcing SLM titanium scaffold structures, based on those used for orthopaedics, to independently control stiffness and toughness. Matching the stiffness of implants to the bones they support is key to their success, therefore the stiffness of printed scaffolds is typically controlled by increasing or decreasing the porosity of the printed structure. However, this may lead to unacceptably low toughness if a low stiffness is required. They found it was possible to improve toughness in a high-porosity scaffold by 30%, with a 9% increase in stiffness, by filling the pores of the scaffold with thermoplastic PET as shown in Figure 2-49, to create a titanium-polymer composite structure. This allows the durability of a structure to be improved where functional constraints may prevent changes to its stiffness or external dimensions.

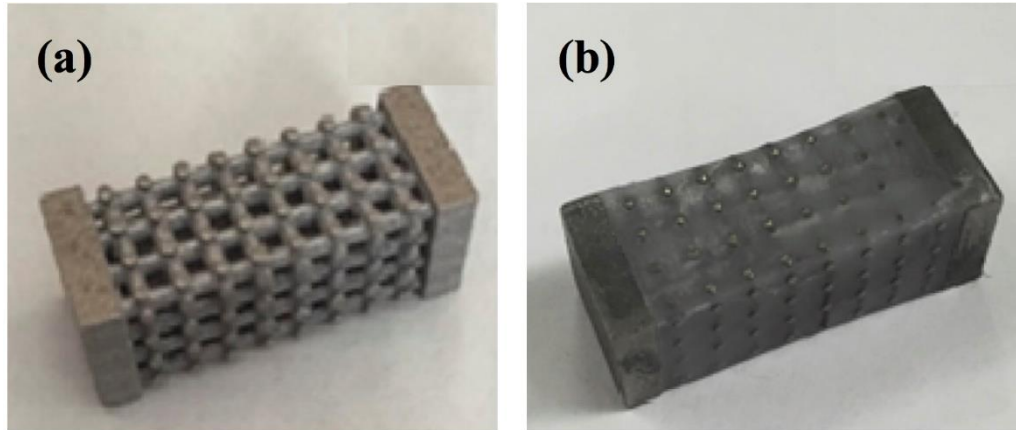


Figure 2-49. 3D printed titanium scaffold before (a) and after (b) filling with polymer [81].

Nakagawa et al. [82] modified the print head in an FDM process so that the printed filament was infiltrated with carbon fibres, shown in Figure 2-50, effectively allowing the direct 3D printing of CFRP components. It was found that additional heat treatment was necessary to ensure adequate thermal bonding of carbon fibres to the ABS matrix, but this led to a 50% increase in the tensile strength of components. Fibre pull-out was observed in all failed specimens to varying degrees, including those that were heat treated, suggesting that tensile strength could be improved further if refinements were made to the process in order to enhance fibre-matrix bonding.

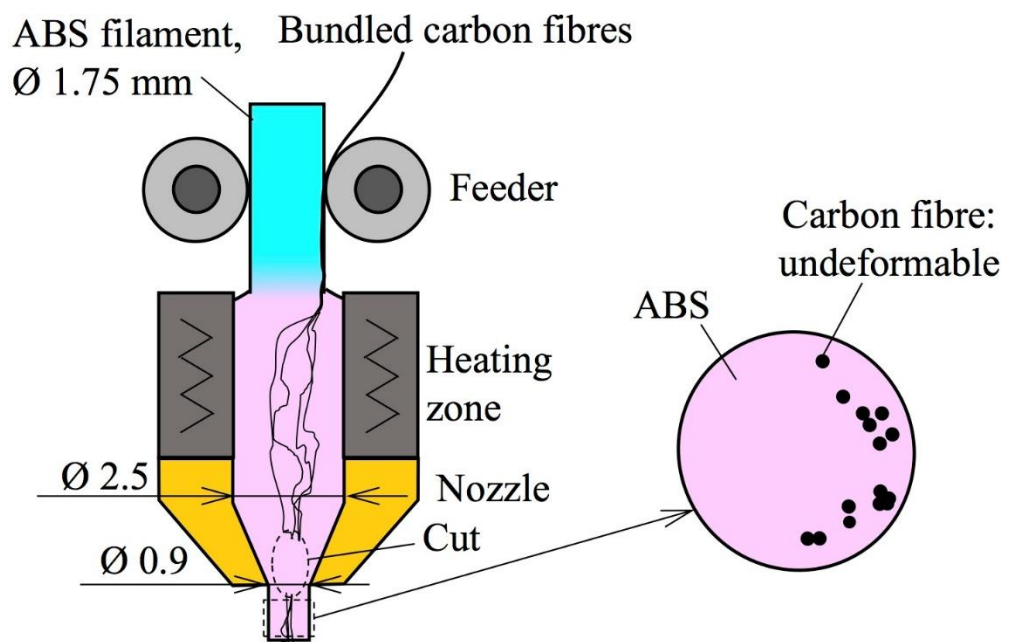


Figure 2-50. Addition of carbon fibres to the FDM process for reinforcing of printed components [82].

2.5.4 Summary

AM shows clear potential for the production of functional bespoke fixtures with a high degree of complexity. Given the ability to print virtually any geometry, it becomes easier to support the freeform surfaces of complex components in a fixture. Cellular and other repeating structures could be used to allow localised tuning of different regions of the fixture, in order to impart unique stiffness and even damping properties to the workpiece-fixture system, which could improve the dynamics of the machining process.

Despite this potential, there are very few examples in the literature of the use of AM fixtures, and these are mainly limited to metrology applications rather than machining. While Violante et al. [29] and Krznar et al. [79] both employ AM components to conform to freeform components, their applications are based on conventional fixturing methodologies, and use printed components that could have been manufactured by conventional means, albeit at a greater cost than AM. They do not achieve full-face contact with the workpiece, and therefore their designs offer no benefit over a conventional fixture in terms of workpiece support. For machining, an AM fixture has the potential to offer a considerable benefit, as the fixture can support the workpiece over entire surfaces to prevent deformation and chatter, however no examples exist in the literature that exploit this capability.

2.6 Systems and structures for damping of vibrations

For applications ranging from acoustics to architecture, structures that are capable of dissipating forced vibrations, or have unusual stiffness properties, are of interest to academia and industry. Reduction of vibrations can be achieved by shifting resonance of a system away from a critical excitation frequency (vibration isolation), or by harvesting kinetic energy from excitation and converting it to heat (vibration damping).

Damping systems can either be active, where sensors provide feedback to a powered control system that causes actuators to provide destructive interference

to vibrations; or passive, where energy is dissipated through the structure due to features inherent to its material or geometry that are tuned towards a particular frequency range. This thesis is concerned with passive systems, due to their greater flexibility for the small fixtures investigated in this work.

Damping phenomena are exploited to reduce vibrations in systems of all sizes, from bridges to microelectromechanical systems devices. The most common type of damper for large systems is the tuned mass damper, such as the one used to suppress seismic vibrations in the Taipei 101 skyscraper. The building has been shown to have a first bending frequency of approximately 0.15Hz, and damping is achieved by means of a 660 tonne mass suspended from the 91st floor, which is designed to withstand the forces of a 2500 year return period seismic event [83].

On a smaller scale, simple mass-spring devices exploit viscoelastic damping to attenuate vibrations, and can be very effective. A vibration damping table for mounting a precision interferometer, with legs comprising three steel cylinders resting on springs, was subjected to vibration at 100Hz, and was shown to transmit just 10^{-7} of the excitation energy laterally, and 3×10^{-6} vertically to the table surface [84].

This section covers general approaches to attenuation of vibration, and presents a number of novel application-specific damping systems.

2.6.1 Negative stiffness phenomena

The stiffness of a structure has a significant effect on its frequency response, with stiffer structures resonating at high frequencies. As clamping forces acting on a structure increase, its stiffness decreases, causing a shift in its natural frequencies towards zero. Negative stiffness implies that displacement of a system occurs in the opposite direction to an applied force, which gives rise to unusual dynamic properties that can be exploited to oppose vibrations. It is generally associated with buckled structures, where the geometry is able to snap between bistable states, and has been shown to cause damping effects.

Lakes [85] discovered the negative stiffness properties of buckled silicone tubes, which exhibit displacement in the opposite direction to an applied force, as shown in Figure 2-51. The buckled tubes were fabricated into unit cells, and assembled into a composite comprising multiple cells, which was shown to display damping orders of magnitude greater than the bulk silicone material. Lakes et al. [86] later demonstrated that negative stiffness inclusions of vanadium dioxide can be stabilised in a positive stiffness matrix of pure tin. Due to temperature-sensitive ferroelasticity, changes in the crystal structure of the vanadium dioxide at certain temperatures allow the structure to exhibit stiffness and damping greater than a composite with inclusions of infinite positive stiffness.

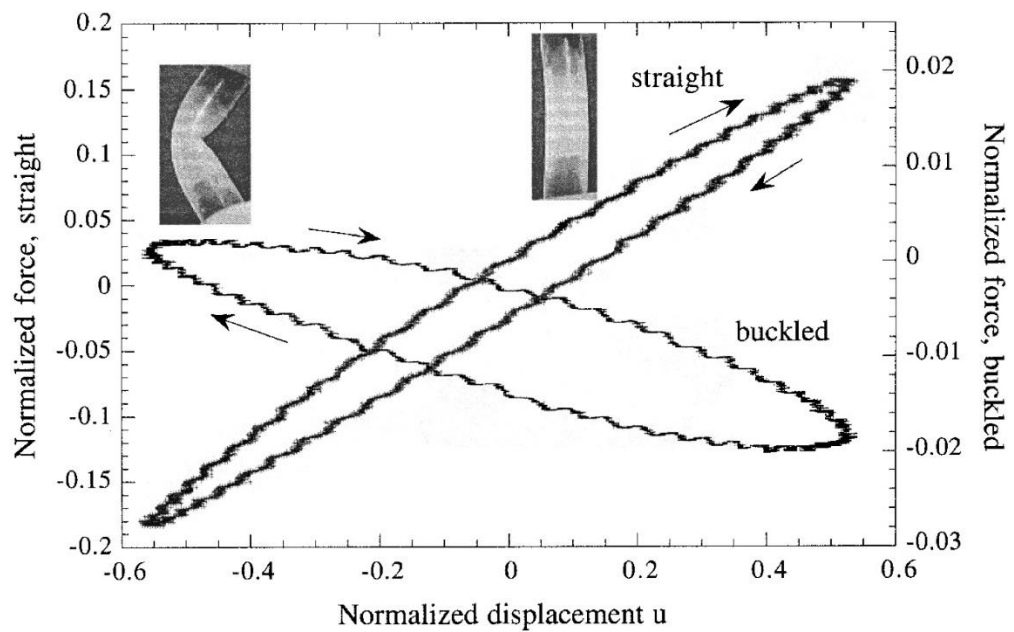


Figure 2-51. Force-displacement plot comparing the behaviour of positive (straight) and negative (buckled) stiffness structures [85].

Sun et al. [87] developed numerical and FE models for vibration isolating and absorbing metamaterial beams, explaining the mechanism behind damping in metastructures subjected to harmonic excitation. By considering the 2-DOF dynamic system of an absorber inside an isotropic beam as a 1-DOF system (as if the presence of the absorber is unknown to an observer), phase differences between the beam and internal absorber can create negative effective mass and

negative effective stiffness of the beam-absorber metastructure. If the frequency of excitation falls within the bandgap of the absorber, it vibrates creating shear forces and bending moments that stop propagation of vibrations through the wider structure.

Kashdan et al. [88] designed a system exploiting the negative stiffness characteristics of bistable structures for damping applications. They 3D printed an assembly, shown in Figure 2-52, which used a spring to constrain a compressed beam in a metastable state, preventing it from adopting a stable shape. They excited the structure using a shaker, demonstrating that small compression of the beam (0.4% strain) could shift the first natural frequency of the beam sufficiently far from its uncompressed natural frequency that the system displayed negative transmissibility, meaning amplitude of beam displacement was less than the input to the system.

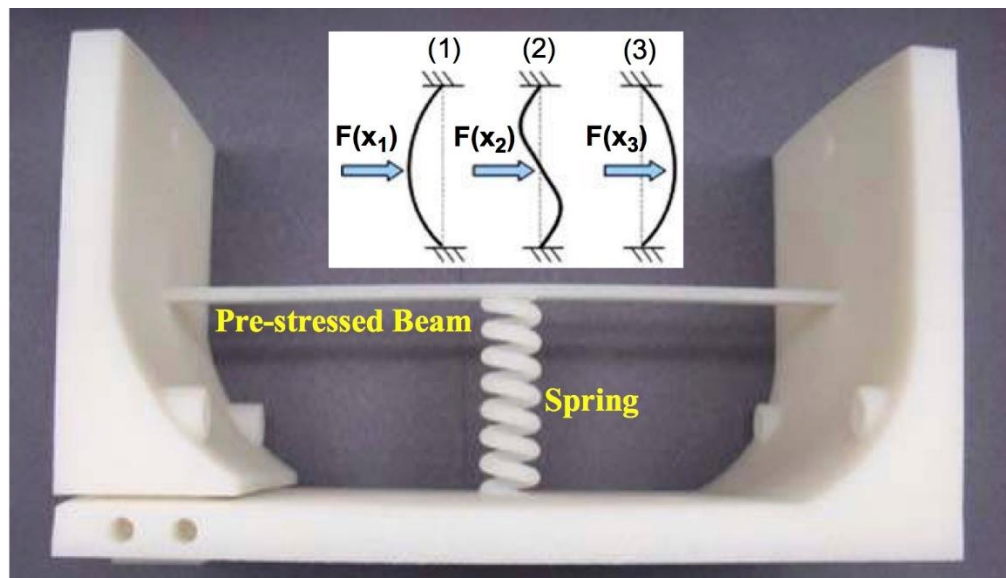


Figure 2-52. 3D printed damper system with a compressed beam, where the spring holds the beam in a metastable state (2), preventing it from obtaining a stable bending shape (1 or 3) [88].

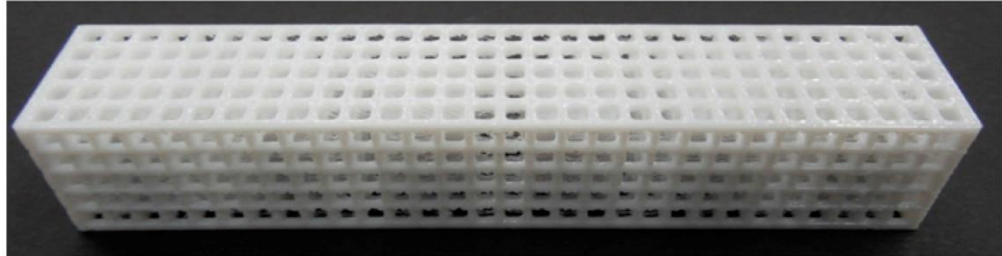
2.6.2 Lattices and cellular structures

Structures with repeating geometries have applications for damping, due to the ability to locally alter properties of cells/repeating elements, with negligible effect on the overall structure. This allows metamaterial regions of low stiffness,

or with particular resonant properties, to be incorporated into an otherwise rigid structure, such that they create a bandgap that causes vibrations in a particular frequency range to be absorbed or redistributed.

Matlack et al. [89] designed metastructures for vibration absorption along this principle, using steel cubes encapsulated into a polycarbonate lattice by 3D printing an open-cell grid around them. By varying the level of lateral support for the encapsulated blocks in a single row of blocks, as can be seen in Figure 2-53, their stiffness and motion can be controlled. This allows creation of wide bandgaps where no resonance of the structure occurs. The bandgaps of the structure are a superposition of its flexural, torsional, and longitudinal mode shapes, and therefore altering the constituent modes by increasing the size of blocks, or the space between them, or changing the thickness of elements in the lattice is shown to shift the central frequency of the bandgaps, and allow its width, to be increased to cover a wider frequency range.

(a)



(b)

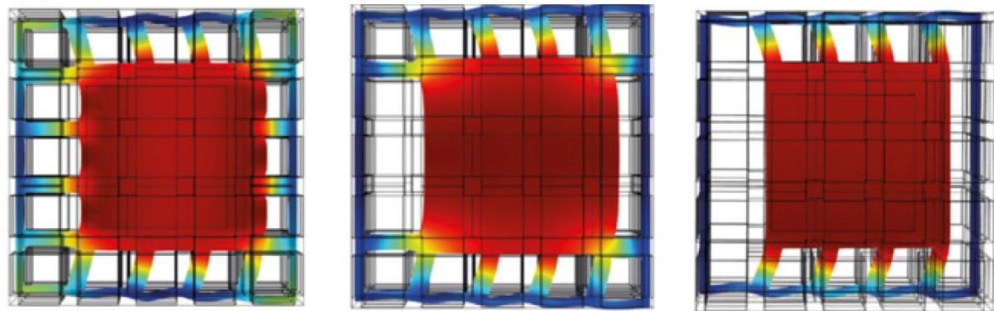


Figure 2-53. Metastructure comprising a one-dimensional chain of six steel blocks encapsulated in a polycarbonate lattice (a), and FE models visualising how lateral support to blocks, shown in red, can be varied by removing links in the lattice (b) [89].

Baravelli and Ruzzene [90] developed chiral lattices cut using waterjet from elastomeric material, with steel inserts to act as distributed vibration absorbers.

Shown in Figure 2-54, they were designed to resonate internally in order to damp vibrations when placed inside an aluminium frame. A periodic lattice was used for uniform properties across the structure, alongside a graded lattice with varying insert masses, so that inertia distribution along the beam could be controlled. Both systems damp impact-induced oscillations significantly faster than the frame alone. Installing the periodic structure reduces the response of the system at its first mode frequency by approximately 20dB, with a single additional mode visible in the FRF of the structure, due to the internal resonance of the lattice. The graded structure achieved significantly greater damping of approximately 40dB at the original natural frequency, due to energy confinement in the band gap; however, there were numerous additional modes detected due to multiple resonances of the different masses in the lattice.

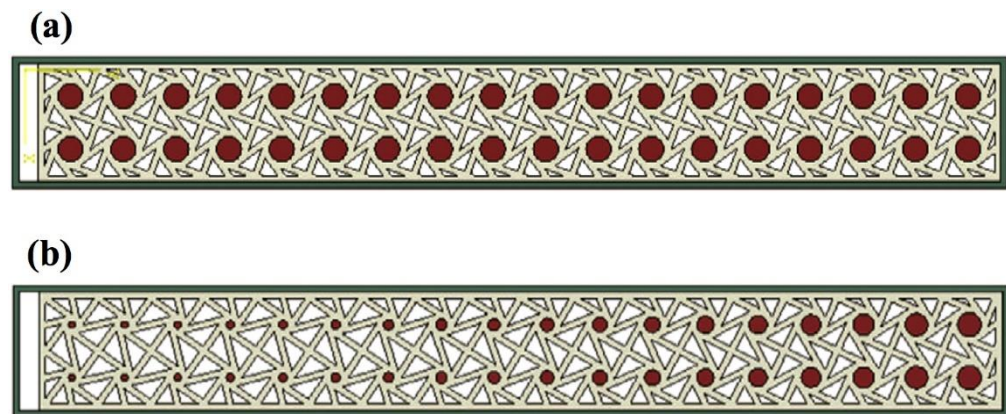


Figure 2-54. Periodic (a) and graded (b) chiral lattices for vibration damping, with the frame structure indicated by the dark border [90].

Essink and Inman [91] sought to expand upon the work of Baravelli and Ruzzene, producing very similar lattices with optimised geometries using 3D printing. They used two frame geometries with wall thicknesses of 5mm and 10mm, and printed lattices in the highest and lowest stiffness materials available for their AM system, enabling the testing of two usage cases for chiral lattice dampers: a low stiffness frame with a stiff lattice; and a stiff frame with a low stiffness lattice. Hammer modal testing was used on the frames with and without the lattices installed, enabling direct comparison of their effect on the dynamics. The high stiffness lattice was the quicker of the two to damp vibration following the hammer impact, and it shifted vibrations to lower frequencies than the

lattice-free structure, appearing to completely damp out the higher-frequency modes detected without the lattice. The low stiffness lattice shifted the natural frequencies of the frame-lattice system higher, and achieved attenuation at the first mode frequency of approximately 30dB. While the magnitude of damping was less than that obtained by Baravelli and Ruzzene [90], it was spread over a larger bandgap width, meaning the structure was damped against vibrations over a frequency range that was three times larger.

2.6.3 Magnetic phenomena

Magnets arise frequently in the damping of structures, due to their unique qualities among passive damping systems. While dampers based solely on magnetic interactions have been proposed, such as the system developed by Lowth and Nagy [92], which exploits the Lenz effect to resist the movement of magnets inside a copper tube, conventional mass-spring systems augmented with magnets are much more common, and have been studied extensively in the literature.

Bonisolì and Vigliani [93] created a typical magnetic damper setup, based on permanent rare-earth magnets, comparing the performance of conventional elastic suspension (springs only) with elastomagnetic and pure magnetic systems. Magnet-based damping was shown to reduce the relative displacement between the static base and suspended platform, and increase the natural frequency of the system, with faster decay of oscillations than the elastic-only system. While magnetic dampers are typically based on the repulsion of magnets to stiffen a system, Carrella et al. [94] designed a system using magnetic attraction. They suspended a magnet vertically between two identical springs, with top and bottom magnets at the end of the springs connected to a rigid frame. All three magnets were oriented in the same direction so that they were attracting each other, held apart by the springs, meaning the magnets generated negative stiffness to counteract the positive stiffness provided by the springs in order to create a *quasi-zero-stiffness*, or *high static, low dynamic stiffness* (HSLDS) mechanism. The HSLDS system is shown to reduce the

transmissibility of input displacement compared to a simple mass-spring system, with superior load-bearing capacity.

Zhou and Liu [95] developed a highly tuneable HSLDS isolator, shown in Figure 2-55, with a steel beam operating as a mechanical spring, with attached permanent magnets allowing the structure to operate passively, or semi-actively by means of electromagnets positioned parallel to the beam. The gap between the electromagnets was used to tune the system when it is operating passively, and the current supplied to the electromagnets used as the tuning parameter when the magnets were engaged. For the passive system, increasing the electromagnet gap from 70mm to 120mm increased the natural frequency of the beam from 5Hz to 12 Hz, with the effect of increasing the gap rapidly tapering to zero as the gap increased beyond 120mm. For the active system, the natural frequency of the beam decreased from approximately 10Hz to 2Hz as current to the electromagnets was increased from 0 to 2A, for an electromagnet separation of 80mm. Adjustment of these two parameters allows the damper to function over a broad range of frequencies, and replacement of the beam with one of higher or lower stiffness would enable further adaptability.

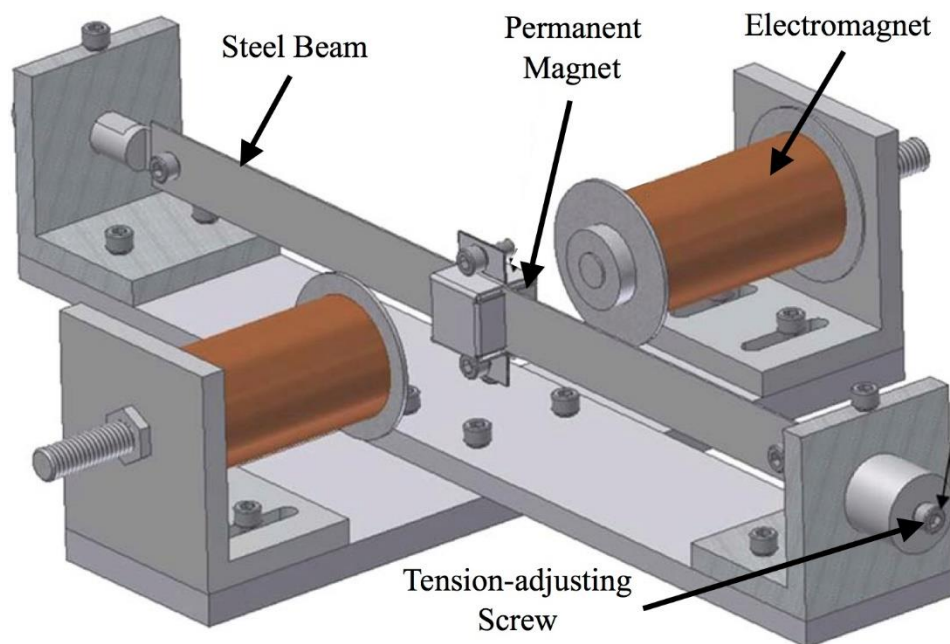


Figure 2-55. High static, low dynamic stiffness (HSLDS) vibration isolator with electromagnets for passive or semi-active operation [95].

2.6.4 Hybrid and miscellaneous applications of damping

This section includes various applications of damping, which do not fall into the previous categories, but nonetheless demonstrate innovative applications of vibration suppression.

Constantinou and Roy [96] demonstrated the capability of low stiffness 3D printed geometries to flex when excited, and exploited this to harvest energy by mounting magnets to the free end of a spring geometry so that the magnet would pass through a coil when the structure resonated, generating a small power output.

Cottone et al. [97] exploited buckled piezoelectric beams to harvest vibration energy. A PZT film was applied by screen printing to a steel shim, which was clamped at each end and excited by a shaker, with the setup shown in Figure 2-56. One of the clamps was advanced on a moving stage to provide compression, and electrodes in the piezoelectric film were used to measure the voltage generated in response to displacement. The RMS voltage generated was shown to be an order of magnitude higher for the samples with compression of 0.05mm and 0.07mm, compared to the un-buckled beam, indicating the energy harvesting/dissipating potential for compressed deformable beams subjected to vibrations.

Mayer et al. [98] manufactured ‘active struts’ by encapsulating piezoelectric shunt dampers based on piezoceramic multilayer actuators within 3D printed structures. Using SLM, they paused printing of a steel housing to allow installation of dampers, and then resumed the process so the dampers were encapsulated. The concept was demonstrated in a truss structure built with members comprising these dampers, but there exists a broader potential for achieving damping by encapsulating such dampers within any printed structure.

Altan et al. [99] manufactured composite sandwich panels for damping, using plastic injection forming to shape the skins and core of the panel in a single process for improved panel strength. The panels had aluminium skins and cores

based on polypropylene (PP), in three configurations (pure PP, PP with 30% styrene block copolymer, and PP with 5% titanium dioxide). In addition to a bending test, the panels were subjected to a force transmissibility test, using a shaker to obtain the damping performance of each panel. The styrene block copolymer configuration produced the greatest damping, with a damping ratio of 0.18; however it exhibited approximately 20% lower bending strength than the other two panel types. Pure PP had a damping ratio of 0.13, and the TiO₂ configuration provided less than this (0.06), possibly due to lower quality of bonding between the skins and core.

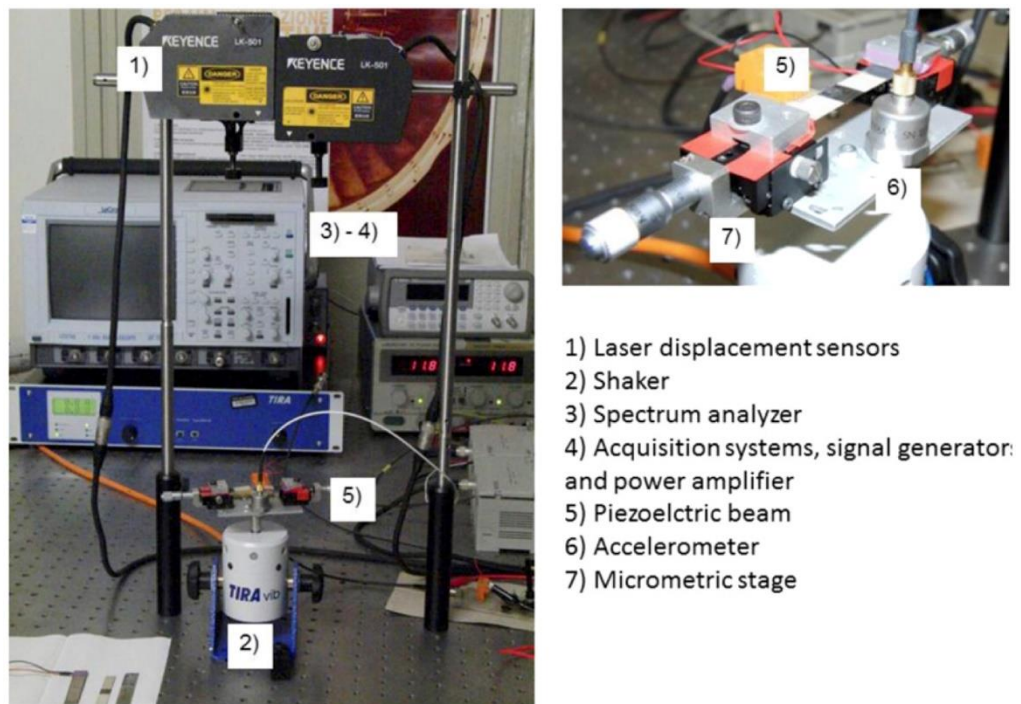


Figure 2-56. Setup for harvesting vibration energy using a piezoelectric beam [97].

Chiba and Furukawa [100] used viscoelastic polyimide tape for damping of vibrations in aluminium beams. Beams of different thicknesses were bolted at each end to a jig, shown in Figure 2-57, so that the middle of the beam was free to vibrate, and the jig was excited by a shaker. Tape was added where the beam made contact with the jig, and despite a small contact area (19×20mm at each end of the beam) and thickness (60µm), this had a significant effect on the dynamics of the system. Adding a single layer of tape reduced acceleration measured on the beam by 88% for the thinnest, and by approximately 75% for

the two thicker beams. Simultaneously, the resonant frequency of each beam was shifted 300-400Hz lower. Full results are included in Figure 2-57c. While the addition of subsequent layers of tape continued to shift the frequency lower, it had an unusual effect on the level of vibration. After the initial large reduction in vibration with application of the first layer of tape, the addition of a second layer had a negligible effect beyond the first, and with a third layer, the level of vibration started to increase back towards the tape-free values. This is attributed to frequency dependent shearing properties typical of viscoelastic tape. For such a minor modification, the tape elicited a dramatic damping effect, highlighting the capability of viscoelastic tape for reduction of vibration.

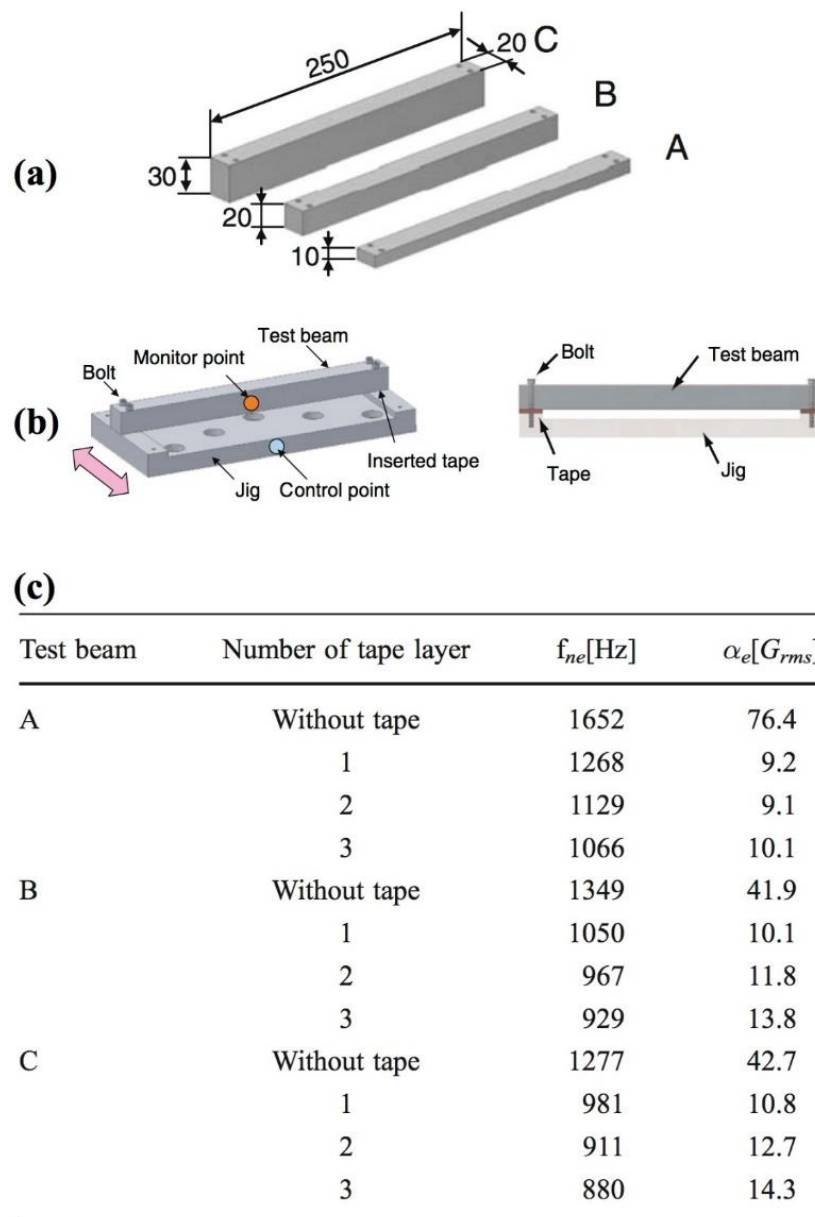


Figure 2-57. Beam geometries (a) and configuration for damping with viscoelastic tape (b), with corresponding resonant frequency and RMS acceleration data (c) [100].

2.6.5 Summary

Damping phenomena are widely exploited in numerous industrial and research contexts, and can be applied to eliminate troublesome vibrations or harvest energy. The ability of compressed or buckled structures to exhibit negative stiffness has been linked to considerable reductions in impulse transmissibility through a structure [85], and holding a buckled structure in a metastable state is shown to result in a significant shift in its eigenfrequencies [88]. Cellular structures and lattices can be used to impart localised damping effects in different regions of a structure [89], or to reinforce a structure by changing its frequency response and stiffness [91].

Magnets are prevalent in damping systems due to their ability to resist motion through attraction and repulsion. Vibration can be reduced using the Lenz effect [92], or by using magnets to augment or replace the springs in a spring damper [93]. Electromagnets can be used to tune the response of a magnetic damper by altering the strength of the magnetic field acting on the moving elements [95].

Vibrations have also been used for energy harvesting, by using displacement of a resonating structure to move a magnet relative to a coil [96], and using piezoelectric coatings on resonating structures [97]. Structural components have been augmented with damping functionality through the addition of concealed shunt dampers [98], and inclusions of polymers with unique vibration properties [99].

These examples demonstrate ingenious exploitation of damping phenomena for the manipulation of vibration. There is the potential in machining for many of these damping innovations to be incorporated into workpieces, fixture locators, and tool beds, however, as discussed in section 2.4, the examples present in the literature tend to be auxiliary systems that are retrofitted to machining configurations that experience problematic vibration, as opposed to tooling that is designed from the outset to control vibrations.

2.7 Summary and gaps in the research

In this chapter, a review of the current research has been presented on fixture design and optimisation, and prevention of vibration in machining, relevant to the design of an improved machining fixture for a thin-walled component. This section summarises these findings, and identifies gaps in the research which could lead to improvements over existing fixture designs.

2.7.1 Summarised findings from the literature

In fixture design, CAFD techniques allow the simulation and optimisation of fixtures so that the workpiece is located and held in the best possible way to reduce deformation and vibration. Residual stresses, and deformation of the workpiece in the fixture are a major source of errors in the final machined component. Fixtures incorporating pin arrays show promise for improved workpiece holding during machining, due to their ability to support components without deforming them, and also to reduce vibrations.

Free and machining vibration behaviour of a workpiece is instrumental in obtaining chatter-free machining conditions. Numerous methods have been proposed to increase stable depth of cut, and also to improve the speed and accuracy of stability calculation. Most attention is paid to predicting stability of the tool, with little consideration given to resonance of the workpiece. Workpiece frequency response varies as material is removed, but it is initially determined by the clamping and constraining applied by the fixture. Understanding the effect of external conditions on frequency response has the potential to lead to fixture designs that improve machining stability.

Milling thin-walled geometries is challenging due to vibration and deformation of the workpiece. Simulation of machining processes has proved useful because it allows workpiece vibrations to be predicted and avoided, along with adjustments to the toolpath to ensure accuracy of the finished geometry. Various solutions to improve machinability and reduce vibrations in thin walls have been proposed, by exploiting stiffening or damping phenomena; however, these are generally external devices, which add bulk and may restrict tool access. The

speed and repeatability of the machining process could be improved if such systems were integrated into fixtures, thus eliminating the need for auxiliary devices.

Additive manufacturing continues to redefine established methods for the design and production of complex one-off components. Freeform surfaces and intricate geometries can be printed directly from CAD files, with selection of a suitable process and orientation serving as the only significant design consideration. In spite of its flexibility and the potential cost savings, the existing AM applications of fixtures are extremely limited in scope, and do not exploit the benefits of AM over conventional manufacturing processes. AM has not yet been proven for machining fixture applications, however it offers the clear potential to provide greatly improved workpiece support through full-face conformability.

Damping phenomena have been exploited in a wide range of applications to reduce vibration or generate power. Systems demonstrated in the literature have utilised negative stiffness behaviour of buckled structures, metastructural properties of lattices, and magnetic effects including electromagnetism. Many of these examples have applicability to the machining of vibration-prone structures, where they can be used to control vibrations, and to improve the accuracy and surface quality of machined components.

2.7.2 Gaps in the research

Based on the findings of the literature review, the following gaps in existing research have been identified:

Academic Challenge 1: Little investigation has been conducted into conformable fixture systems that are capable of accommodating workpiece blanks with deviations from their expected nominal geometries. Much attention is paid to reducing workpiece deformation through optimised placement of clamps and locators in the fixture, but not to compensating for workpieces that are already distorted following a previous stage of manufacture. Pin array

fixtures have been used to accommodate freeform geometries, offering improved workpiece support by means of a larger contact interface, but they have not been investigated for their ability to conform to varying workpiece geometries, in order to reduce deformation of a fixtured workpiece.

Industrial Benefit 1: Holding and supporting of low stiffness workpiece geometries is improved through full-face contact. Deformation of the workpiece in the fixture is reduced, thereby reducing errors present in the finished component, because the workpiece is no longer deformed so that it adopts the shape of the fixture. Components with minor distortions that are within tolerance for their intended application, but are unable to fit inside a rigid fixture system, will no longer need to be scrapped, since the fixture is now able to accommodate them.

Academic Challenge 2: The frequency response of a structure, and how this changes in response to loading and constraining, has generated little research interest. Simple cantilever geometries, such as locators or supports in a fixture, have natural frequencies that can be tuned by altering their length and thickness, causing the structure to resonate at a desired frequency. The effect of constraining and clamping on the frequency responses of single and multiple structures has not previously been studied in detail.

Industrial Benefit 2: Eliciting the resonance of one or many supporting elements that are in contact with the workpiece in a fixture has the potential to dissipate vibration energy away from the workpiece during machining. Reduction of machining vibrations leads to greater manufacturing yields, and higher quality components with improved dimensional accuracy and surface finish.

Academic Challenge 3: There are various examples in the literature of auxiliary damping solutions that improve the machinability of thin walls, however these are generally bulky external devices, such as tuned mass dampers, which are retrofitted to a workpiece when stable machining cannot be obtained. Some workpieces are too small to be augmented with external damping, as they lack any suitable surfaces onto which a damper can be

mounted. In cases where it is possible to mount a damping device, the damper may restrict tool access due to its large size relative to the workpiece. No previous attempt has been made to integrate a damping device directly into the fixture, to improve upon the established practice of mounting devices externally to the workpiece.

Industrial Benefit 3: Improved machinability of thin-walled workpieces, which are some of the most challenging geometries to machine. Fixtures often have relatively large voids between locating features, and these regions are potentially large enough to accommodate supporting elements that provide damping to the workpiece, eliminating the need for external devices. A holistic approach to the reduction of machining vibrations, where damping is added during the fixture design stage, will enable the addition of damping to workpieces which otherwise could not accommodate external damping.

Academic Challenge 4: Bespoke fixture systems are expensive and time-consuming to manufacture by conventional means, due to their complexity. Despite the growth of 3D printing for the production of increasingly high-performance components, limited examples of machining fixtures produced by AM exist in the literature, and the viability of a fully-AM fixture for a milling operation has not been established.

Industrial Benefit 4: The cost of producing AM fixtures is independent of their complexity, making it ideally suited to the manufacture of bespoke machining fixtures with numerous complex features. AM is the best technique for rapid and cost-effective production of fixtures that conform to component surfaces in order to better support fixtured workpieces, and accommodate variations in their geometries, while featuring integrated frequency-tuned damping structures that do not require external mounting or impede tool access.

These gaps are broad in scope, however they represent challenges for which a solution could provide a meaningful advance in fixturing technology. This work aims to address all four gaps simultaneously in a single fixture solution.

3 Experimental setups and methodologies

3.1 Introduction

The gaps in research identified in the previous chapter reveal the benefits that can be achieved through the use of a fixture system that conforms to workpiece geometries to compensate for geometric variability, and integrates frequency-specific damping. This chapter presents the methodologies used to investigate the objectives of this research, including experimental setups, hardware such as sensors and tooling, and software used for modelling and signal processing.

The experimental component of this research is comprised of four major sections. Firstly, measurement of frequency response, which reveals the natural frequencies of a structure, and is used throughout this thesis to facilitate understanding of the vibration behaviour exhibited by workpieces and fixtures. Second and third are the methodologies for measuring clamping-induced workpiece distortion, and machining-induced workpiece vibration, and their associated experimental setups for comparing the novel concept fixture with conventional fixture designs. Described finally is the method for finite element modelling, which was used throughout this research to develop workpiece and fixture geometries, and to determine optimal parameters such as clamping loads and machining excitation frequencies, in order to predict the behaviour of the concepts presented.

3.2 Experimental setup for obtaining frequency response

Modal testing was conducted at numerous stages throughout this research to obtain the frequency response of various structures, to aid with fixture design and to verify models. In this section the main equipment, setups, and methods for modal testing are presented.

All structures have dynamic properties inherent to their geometry and material, which can be affected by factors including external forces, and the removal of degrees of freedom; both of which are applicable to a workpiece in a fixture. Of

greatest relevance to this research are eigenfrequencies, also known as natural frequencies or modes, which are the frequencies at which a structure will tend to vibrate if it receives external excitation, and their corresponding mode shapes. These frequencies are significant in machining, because excitation of a workpiece at one of its eigenfrequencies will cause it to resonate, which results in poor surface finish and geometric accuracy of the workpiece.

In this work, modal testing was undertaken using a hammer for excitation, with an accelerometer to capture frequency response. Other methods of obtaining frequency response include the use of a shaker to provide excitation, and while they provide uniform excitation in the frequency domain, shaker units are generally large and therefore difficult to mount for testing of small components. Laser Doppler Vibrometers can be used in place of an accelerometer to allow non-contact measurement of vibration, however they are better suited to measuring larger systems, and are difficult to set up in confined spaces. Multiple input/multiple output (MIMO) modal testing, in which the test structure is excited from two or more locations with vibration response captured by two or more sensors, is commonly employed in industry for testing of complex geometries, however single input/single output (SISO) testing was deemed sufficient for this research, since the structures to be investigated have simple vibration responses. The hammer/accelerometer setup was selected for its versatility in obtaining frequency response from a range of small workpieces and fixtures. Selection of a suitable hammer tip material means that excitation is uniform across the desired frequency range, and accelerometers are available with masses below 1 gram to minimise mass effects on lightweight structures.

Figure 3-1 shows a schematic of the hammer testing setup for obtaining the frequency response of a component, comprising an accelerometer affixed to the component, and a hammer to deliver a single impulse.

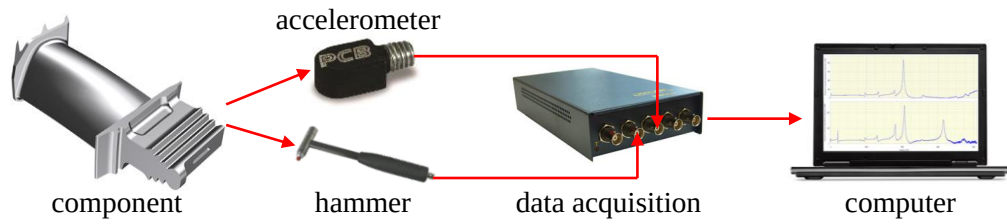


Figure 3-1. Measurement chain schematic for the hammer modal testing process.

The hammer and accelerometer are connected to data acquisition hardware, from which the signals of an impact can be processed to generate the Frequency Response Function (FRF), which is a plot of inertance (structural vibration response normalised by hammer excitation impulse) against frequency, from which eigenfrequencies of the structure can be identified from peaks in inertance. An example FRF from a hammer test is given in Figure 3-2, which shows peaks in inertance corresponding to eigenfrequencies at approximately 700Hz, 1300Hz, 1900Hz and 3700Hz. In this example, the low peak height of the first two eigenfrequencies is due to proximity of the accelerometer to a node of these vibration modes.

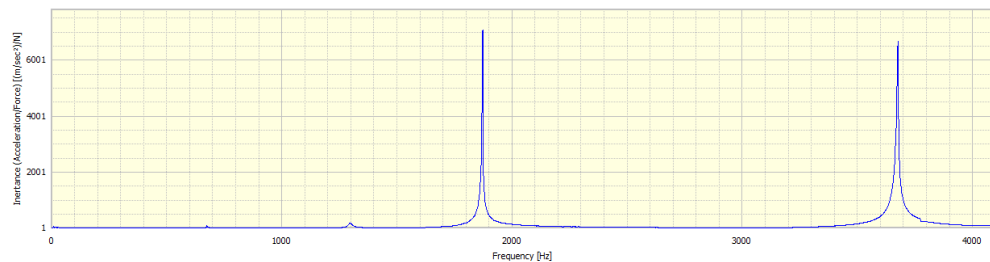


Figure 3-2. Example Frequency Response Function (FRF) of a structure from hammer modal testing.

Conducting a hammer modal test involves the following steps:

- 1. Constrain the test structure:** The component to be modal tested must be constrained in order to create the desired testing conditions. Depending on the geometry of the test structure, the nature of holding and clamping (if any) can have a significant impact on its frequency response, due to inhibition of displacement or pre-loading of the structure. Depending on the data required, the component can be suspended *free-free* using elastic, clamped to replicate a particular

configuration such as a machining setup, or tested in situ on a workbench or similar surface.

2. Determine optimal accelerometer placement and impact site:

Accelerometer placement is important for the detection of eigenfrequencies, since mounting an accelerometer on the node of a particular mode shape will render that mode virtually undetectable. Similarly, positioning the accelerometer far away from readily vibrating (e.g. low stiffness) regions of the component may result in failure to detect vibration in those locations. Repeating modal tests with the accelerometer in multiple locations will ensure detection of all eigenfrequencies. Drive point measurement, in which hammer excitation occurs at the same location and in the same orientation as the accelerometer, is the preferred method of detecting all eigenfrequencies within a structure. Cross point measurements, where excitation occurs at a different location on the structure to the accelerometer, can be used to verify reciprocity (when swapping the locations of hammer impacts and the accelerometer with each other produces two identical FRFs), which is an established indicator of a repeatable testing setup.

3. Excite the structure: Carry out a series of hammer impacts on the test structure. Use of an appropriate hammer tip material ensures uniform excitation over a suitable frequency range. Impacts must be *clean* in order to minimise noise and enable accurate calculation of frequency response. Taking the average response from several repeated (identical) impacts improves accuracy, and allows plotting of the coherence function, which enables verification of test quality. Signals from the hammer and accelerometer are captured by a data acquisition system.

4. Generate the frequency response function: The time domain signals from the hammer and accelerometer are processed and converted to the frequency domain. The resulting FRF plot shows the relative response of the test structure across the specified frequency range, thus revealing its eigenfrequencies.

3.2.1 Impact hammer

Since the geometries considered for modal testing in this research are relatively compact (generally less than 100mm in any dimension) and of low mass (typically below 0.5kg) a compact instrumented impulse hammer was chosen to allow accurate localised excitation of small components. The PCB Piezotronics 086C01 instrumented hammer, whose specifications are given in Table 3-1, has a measurement range of $\pm 444\text{N}$, and was used for all modal testing undertaken in this thesis. It featured removable impact tips in four materials to enable uniform frequency excitation up to 16kHz.

Table 3-1. Specifications of the PCB 086C01 Impact Hammer [101].

Model	 PCB 086C01
Sensitivity	11.2mV/N
Measurement Range	$\pm 444\text{N}$ (peak)
Mass	0.100kg
Overall Length	216mm
Head Diameter	15.7mm
Tip Diameter	0.63mm

3.2.2 Accelerometers

When selecting an accelerometer for modal testing, an important consideration is the size and mass of the accelerometer relative to the test subject, since adding a large accelerometer will have a noticeable effect on the mass and stiffness of the component, which will undesirably alter its dynamic response. The PCB Piezotronics 352C23 single-axis accelerometer has a mass of 0.2g and compact

dimensions, among the smallest commercially available, and was chosen due to its high sensitivity and ease of mounting on small component surfaces.

While the 352C23 is ideal for modal testing, with minimal mass loading effect on the test component, it is not suitable for measuring machining vibration. The PCB 352B01 was instead used for measuring workpiece vibration during machining, discussed later, due to its ability to withstand higher peak acceleration, at the expense of larger size and mass. It was also used for hammer testing of machining samples, since using the same accelerometer for modal testing and machining allows the eigenfrequencies of the workpiece to be known exactly without the need to adjust for accelerometer mass. In all cases, optimal accelerometer placement for modal testing was determined through multiple preliminary tests with the accelerometer mounted to the sample at various locations using petro wax. Due to the viscoelastic properties of wax, which can lead to inaccurate stiffness measurement of eigenfrequencies in low stiffness structures, the final hammer test was conducted with the accelerometer fixed in place at the preferred location using cyanoacrylate adhesive. Table 3-2 contains the specifications of both accelerometer models.

Both accelerometer models contain a plate of piezoceramic material which is cut in an orientation that makes it sensitive to shear loads. When the accelerometer vibrates, the shearing it experiences produces an electrostatic charge in the ceramic plate due to the piezoelectric effect. The accelerometers are *low impedance* sensors, containing an integrated microelectronic amplifier that converts high impedance charge signals into low impedance voltage signals proportional to the vibration experienced, meaning their output is suitable for direct recording, without the need for a charge amplifier. Power is supplied to the accelerometer by the data acquisition system that captures its output vibration signals, thus eliminating the need for an external coupler to supply power.

Table 3-2. Specifications of the accelerometers used for hammer testing and machining [102, 103].

		
Model	PCB 352C23	PCB 352B01
Sensitivity	0.5mV/(m/s ²)	0.1mV/(m/s ²)
Measurement Range	±9810m/s ²	±49050m/s ²
Frequency Range	2.0-10000Hz (±5%) 1.5-15000Hz (±10%)	2.0-10000Hz (±5%) 1.0-20000Hz (±10%)
Resonant Frequency	≥70kHz	≥65kHz
Mass	0.2g	0.7g
Dimensions	8.6×4.1×2.8mm	8.1×ø6.1mm
Casing Material	Anodised Aluminium	Titanium

3.2.3 Data acquisition and processing

Each hammer test involves striking the test structure with an instrumented hammer, which contains a load cell. The impact delivers an impulse that, assuming suitable selection of the material for the hammer tip, uniformly excites at all frequencies across the frequency range of interest. As is illustrated in Figure 3-1, the accelerometer then captures the response of the component. Both the hammer and accelerometer are connected to a data acquisition system, which is connected to a computer running software that analyses the signals. It interprets the time domain force inputs and vibration outputs, and uses Welch's method [104] to express their relationship in the frequency domain in the form of a FRF, a plot which shows the magnitude of structural response normalised by excitation force (known as inertance), against frequency. From the FRF, numerous dynamic properties of the structure can be identified, including eigenfrequencies of the test structure (indicated by frequencies at which the

inertance peaks), damping and modal mass of each eigenfrequency (determined from the width of its corresponding peak), and even a visualisation of mode shapes (using multiple FRFs comparing tests with the accelerometer placed in various locations on the test structure). In order to compensate for any nonlinearities in the test structure and ensure accuracy of frequency response data, the FRFs were generated from five averaged hammer impacts, using the H1 estimator methodology, which assumes the (hammer) input is noise-free, with any noise present in the system existing only on the (accelerometer) response. To ensure repeatability, the quality of each FRF was checked by inspecting its coherence, to ensure a high signal to noise ratio of the vibration response captured by the accelerometer. This process is summarised in Figure 3-3.

A Prosig P8004 4-channel 24-bit data acquisition system, specifications given in Table 3-3, was selected for its versatility in capturing modal testing signals (along with other signals discussed in the following subsections), and because it is capable of powering low impedance sensors directly without the need for a constant current signal conditioner. Its accompanying DATS software package is a powerful tool for acquiring signals, with features such as live multi-signal preview, recording in engineering units for each sensor, and automatic acquisition triggering. It features an integrated module for hammer testing, which automatically determines optimal sensor gain levels and signal windowing, and warns if adverse testing conditions are detected, such as a double hammer impact, or if the test structure is not at rest prior to impact. Once data has been acquired, the software can perform a comprehensive range of common signal analyses, in addition to numerous application-specific analyses.

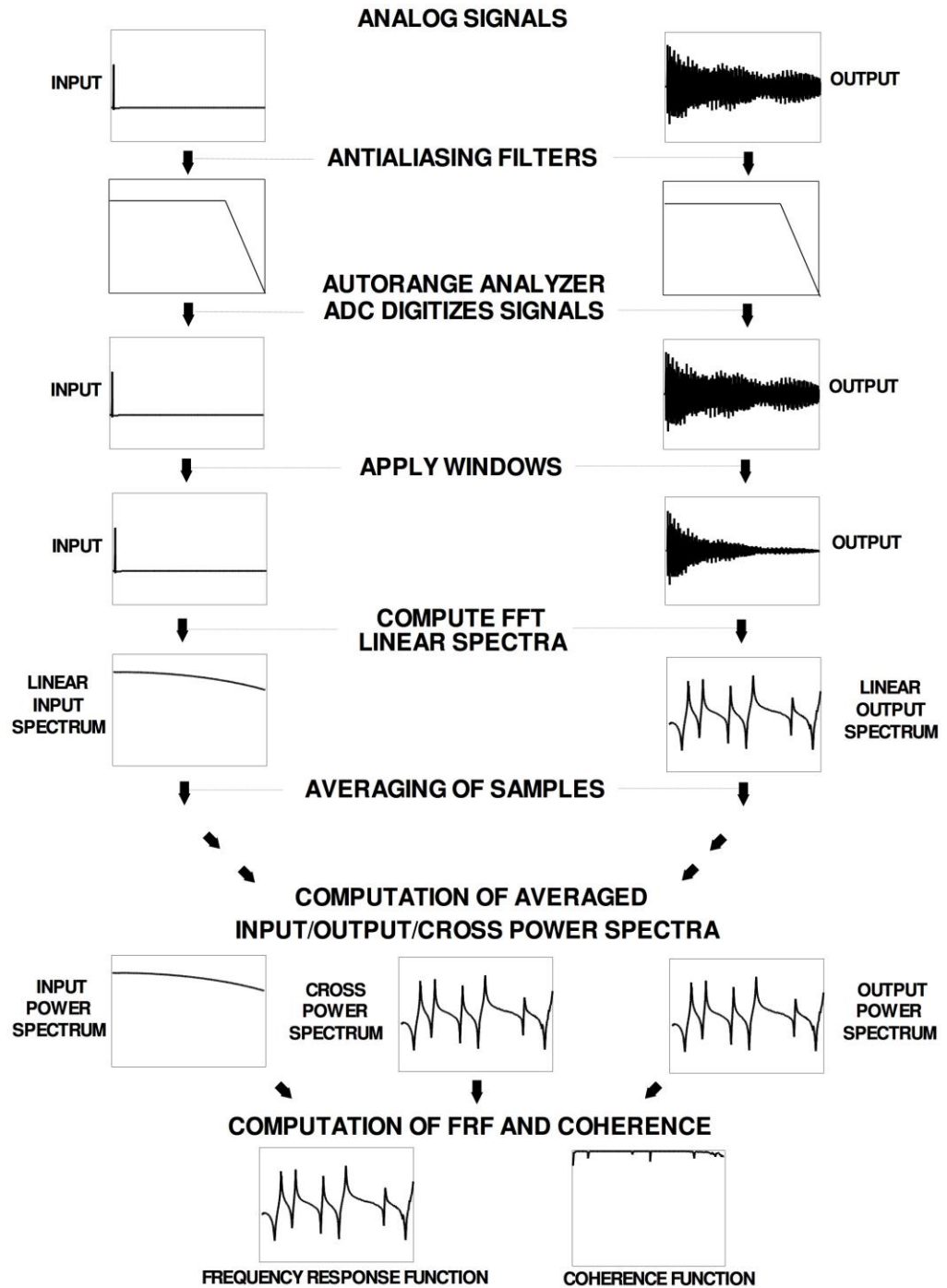


Figure 3-3. Stages of signal processing in modal analysis, showing how the input (hammer) and output (accelerometer) signals are used to produce the FRF [105].

Table 3-3. Specifications of the Prosig P8004 Data Acquisition System [106].

Model	 Prosig P8004
Analogue Inputs	4 Channels plus Tachometer
Input Voltage Range	$\pm 10\text{mV}$ to $\pm 10\text{V}$ (Over-voltage protection $\pm 24\text{V}$)
Maximum Sample Rate	24 bit: $100,000\text{s}^{-1}$ 16 bit: $400,000\text{s}^{-1}$
Sensor Power Supply	24-27V DC, 2-20mA constant current supply
Computer Interface	USB

3.3 Experimental setup for investigation into flexure conformability on workpiece distortion

The first proof of the fixture concept involved measuring the deformation of workpieces at various clamping loads when held in a fixture, and this section presents the setups and methods involved.

A test setup was devised in which a rigid fixture design was compared against a conformable fixture prototype, to investigate whether the novel fixture design was capable of reducing deformation in workpieces with pre-existing distortions from their nominal geometry. Shown in Figure 3-4, the configuration involved a thin, low-stiffness workpiece held between an upper and lower fixture block, replicating a 2D section through a fixture for a 3D component.

Clamping was applied to the fixture by advancing a moving stage to compress the assembly, with force measurement obtained from a load cell.

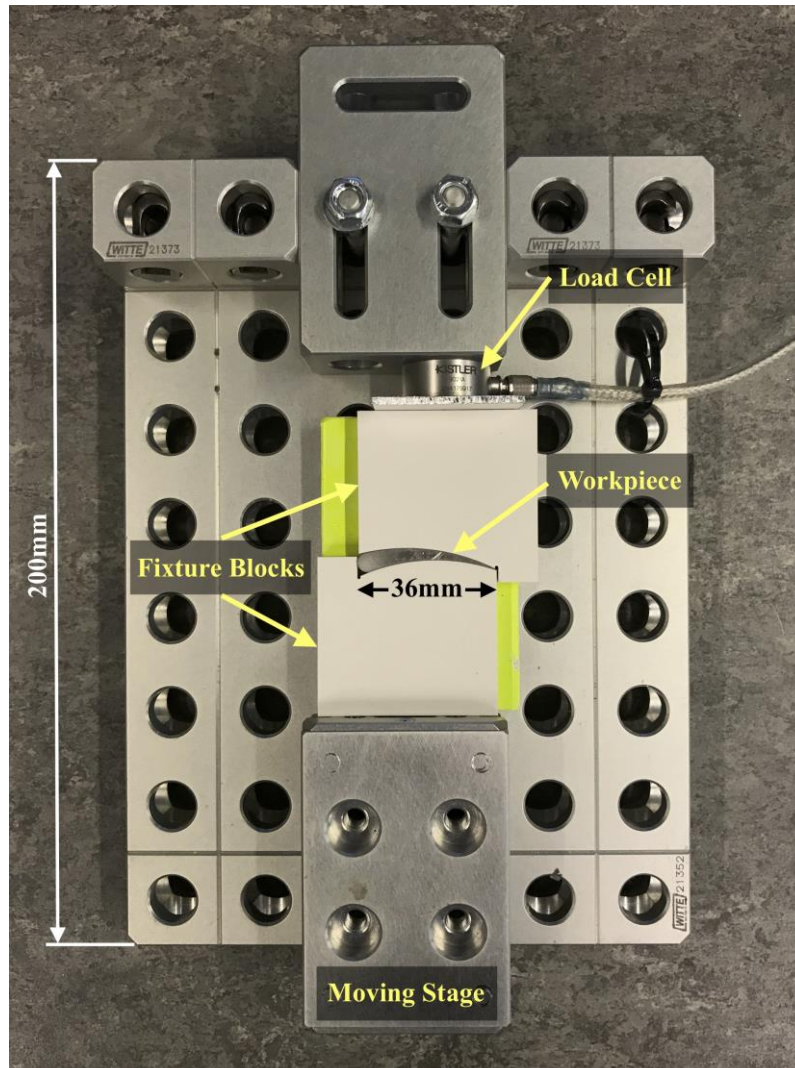


Figure 3-4. Experimental configuration for measurement of workpiece deformation under clamping in the rigid fixture system.

With this setup, it was possible to measure the clamp force required in order to achieve full holding of the workpiece, along with the deformation it experiences at different levels of clamping. The following steps were undertaken to investigate fixture conformability:

- 1. Assemble the workpiece and fixture blocks:** Blade-like workpieces, including a nominally sized component alongside components with varying severity of artificially-induced geometric errors to represent manufacturing variabilities, were clamped between pairs of fixture blocks: a conventional rigid fixture, and a novel conformable prototype

fixture. The chosen workpiece and fixture combinations were assembled on a planar substrate, so that the workpiece-fixture assembly could be compressed to replicate clamping of a workpiece in a fixture.

2. **Apply clamping to the assembly:** One end of the assembled fixture blocks sits against a rigid stop, while the other is in contact with a moving stage, which is advanced to apply compressive loading. A load cell is placed in-line with the assembly, and connected to a data acquisition system in order to measure the applied clamping load. Deformation of the workpiece and fixture in response to clamping is captured using a Digital Image Correlation (DIC) system.
3. **Process clamping response data:** The clamping force applied at key stages of loading, such as full holding of the component and the onset of elastic deformation, is revealed. In addition, the deformation of each workpiece-fixture combination at fixed clamping loads was quantified. From this data, direct comparisons can be made between the efficacy of the two fixture systems for the holding of components with initial distortions.

3.3.1 Manufacture of 2D components

The workpiece and fixture parts shown in Figure 3-4 were cut using a microwaterjet F4 machine from WATERjet AG. The F4 is capable of rapidly cutting parts with very high accuracy, with inbuilt control software that determines optimal cutting parameters for the chosen material and section thickness, and its specification machine is given in Table 3-4.

The fixtures were designed to accommodate the nominal component geometry, with contact surfaces that were the inverse of those on the nominally sized blade. A total of eight further blade-like components (shown in section 5.3) were manufactured with deviations from the nominal blade geometry in order to simulate workpieces with geometric errors with of increasing severity, and allow assessment of the relative performance of the two fixture systems in holding distorted components.

Table 3-4. Specifications of the microwaterjet F4 machine [107].

	
Model	WATERjet microwaterjet F4
X/Y/Z Axis	1000×600×140mm
Cutting Accuracy	±0.01mm
Positioning Accuracy	±0.003mm
Cut Surface Quality	Up to N6 (R _a 0.8μm)

In an ideal case, in which no error exists between the profiles of the fixture and nominally sized workpiece, the workpiece experiences zero deformation as clamping is applied. The use of microwaterjet to cut the components ensured that the contact surfaces between the fixture blocks and the nominal workpiece geometry were accurate conjugates, meaning the nominal component fit in the fixtures with negligible mismatch between the profiles of the contact surfaces. The accuracy of the cut profiles minimises deformation of the nominally sized component during clamping, meaning any deformation observed in the components with geometric errors can be attributed solely to their deviation from the nominal profile, with negligible influence from other sources of error. Data on the accuracy of components cut by microwaterjet is presented in Chapter 5.

3.3.2 Clamp force measurement

Clamping force applied to the workpiece was measured using a Kistler 9021A load washer, with specifications given in Table 3-5. It was selected for its

compact dimensions and high sensitivity, which enabled accurate measurement of clamp forces from the fixture. The measurement chain for clamp force measurement is illustrated in Figure 3-5.

Table 3-5. Specifications of the load washer used for clamping force measurement [108].

	
Model	Kistler 9021A
Dimensions	Outer Diameter: 22.5mm Hole Diameter: 10.5mm Height: 10mm
Axes of Measurement	Single (axial direction)
Measurement Range	F_z : $\pm 35\text{kN}$
Sensitivity	4.3pC/N

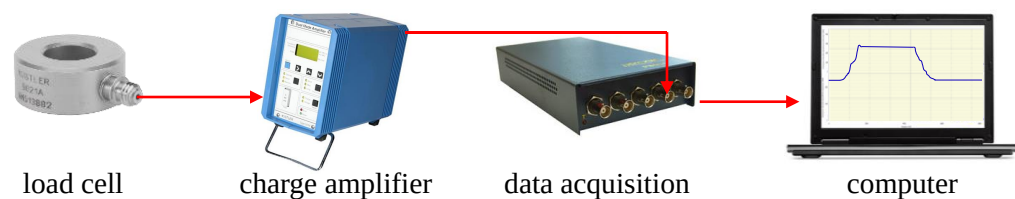


Figure 3-5. Measurement chain schematic for determining clamping force.

Operating on a similar principle to the accelerometers described previously, the Kistler 9021A load cell exploits the piezoelectric effect to measure force using a sensing element comprised of a stack of pre-loaded quartz crystals. When the cell is subjected to clamping, the crystals are compressed, thus generating a charge proportional to the clamping force, which is collected by electrodes sandwiched between the crystals. Since piezoelectric crystals only produce an

electrical output when they experience a change in load, the sensor is incapable of truly static measurements due to leakage of electrostatic charge, however the time constant of the 9021A makes it suitable for quasi-static force measurements used to determine clamping force.

As a *high impedance* force sensor, the charge is routed to a charge amplifier through a cable with high resistance insulation (shielding against triboelectric noise, a charge that is generated when a cable is moved or flexed, which is indistinguishable from electrostatic sensor output). A charge amplifier contains a high-gain inverting voltage amplifier, with insulated MOSFET solid state circuitry to minimise drift (undesired change in output signal that is not a function of the measured variable), that converts sensor charge into a usable low impedance voltage signal. A Kistler 5010B single channel charge amplifier, detailed in Table 3-6, was used to generate a continuous voltage output signal, which was acquired by the Prosing P8004.

Table 3-6. Specifications of the charge amplifier used for clamping force measurement [109].

	
Model	Kistler 5010B
Measurement Range	$\pm 10\text{-}999000\text{pC}$
Scale Settings	$2 \times 10^{-4} - 1 \times 10^7 \text{MU/V}$ where MU = mechanical unit, e.g. N
Sensor Sensitivity	0.01 to 9990pC/MU 0.01 to 9990mV/MU
Output	$\pm 10\text{V}$

3.3.3 Digital Image Correlation (DIC)

Deformation of the workpiece during clamping in the fixture was measured using a Dantec Q-400 Digital Image Correlation (DIC) system, for which full specifications are given in Table 3-7. DIC facilitates three-dimensional, non-contact measurement of displacement and strain, permitting detailed visualisation and analysis of local and global deformation effects arising due to clamping in the fixture. It is highly accurate, generating more data than is possible to obtain using manual measurement techniques, and operates by comparing images of a test sample at various stages of deformation, tracking blocks of pixels to measure surface displacement, generating full field 2D and 3D deformation vector fields and strain maps. A stochastic speckle pattern is painted onto each sample, and the configuration is photographed before and during application of loading by two identical cameras positioned at different angles relative to the specimen. The correlation of images from multiple cameras allows subpixel detection of deflection, achieving resolutions of the order of one part per million of the field of view (resolution in engineering units therefore depends on the dimensions of the test subject) [110]. Calibration is automatic using specialised crosshair and checkerboard targets. A typical unprocessed image from DIC is shown in Figure 3-6.

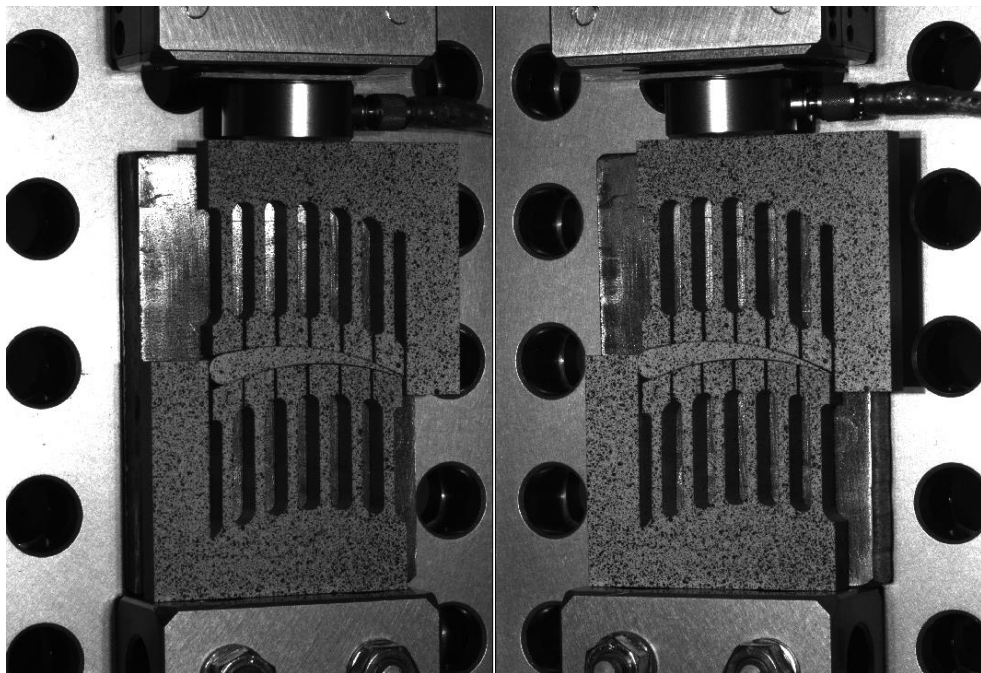
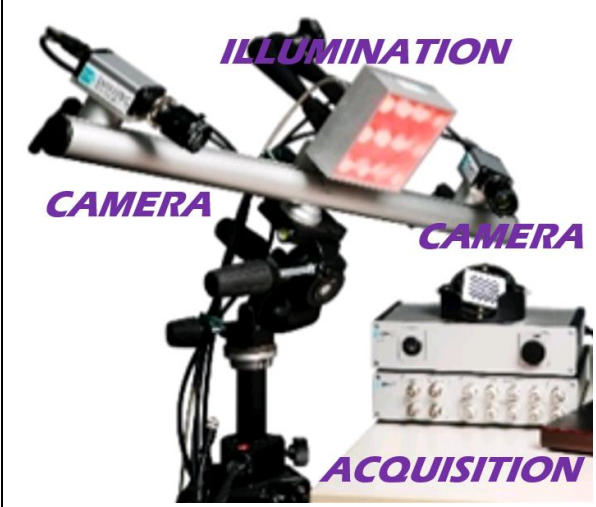


Figure 3-6. Example image captured by the DIC system, showing the perspective from each camera, and the speckle pattern on the component and fixture.

Table 3-7. Specifications of the Digital Image Correlation system [111].

	
Model	Dantec Q-400
Camera Resolution	5 Megapixel Up to 0.01 pixel displacement resolution
Measured Object Dimensions	<1mm to >1m
Illumination source	High-intensity, monochromatic, homogeneous
Correlation speed	80ms (live real-time preview) Full-resolution, full-field capture at 4fps
Analogue inputs	8 channels for synchronous sensor acquisition with images

The Dantec Istra 4D software is used in conjunction with the Q-400 to capture imagery for correlations. It has the capability to overlay preview correlations on live images of the test sample, as seen in Figure 3-7. It also stores measurements from analogue inputs, such as the load cell used in this configuration, within the data file for each image. Once image capture and correlation is complete, clamping-induced strain and displacement of the test sample can be shown in any direction (or as a magnitude), and plotted relative to the initial state, or any intermediate stage of clamping that was captured by the system. The sensor interface means that visualisation of displacement in the test sample is displayed alongside the outputs from any sensors connected, e.g. to show the clamping load at which a certain level of displacement occurs.

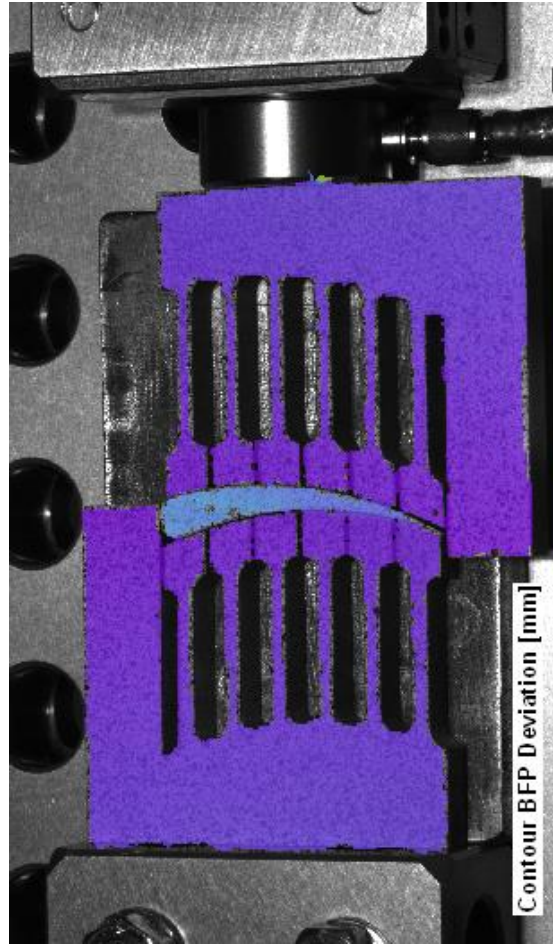


Figure 3-7. Example live correlation preview correlation from Istra 4D overlaid on an image of the test sample.

3.4 Experimental setup for validation of fixture concept in machining

Following investigations into the ability of conformable fixtures to accommodate blade-like workpieces with distortions, a validation of the concept for machining applications was necessary. Detailed in this section are the equipment and methodologies used to investigate workpiece machining response.

Since the fixtures developed in this research are to be produced by additive manufacture (AM), a proof of concept was devised in which printed fixtures were used to hold a workpiece during milling in order to verify the suitability of an AM fixture for machining applications.

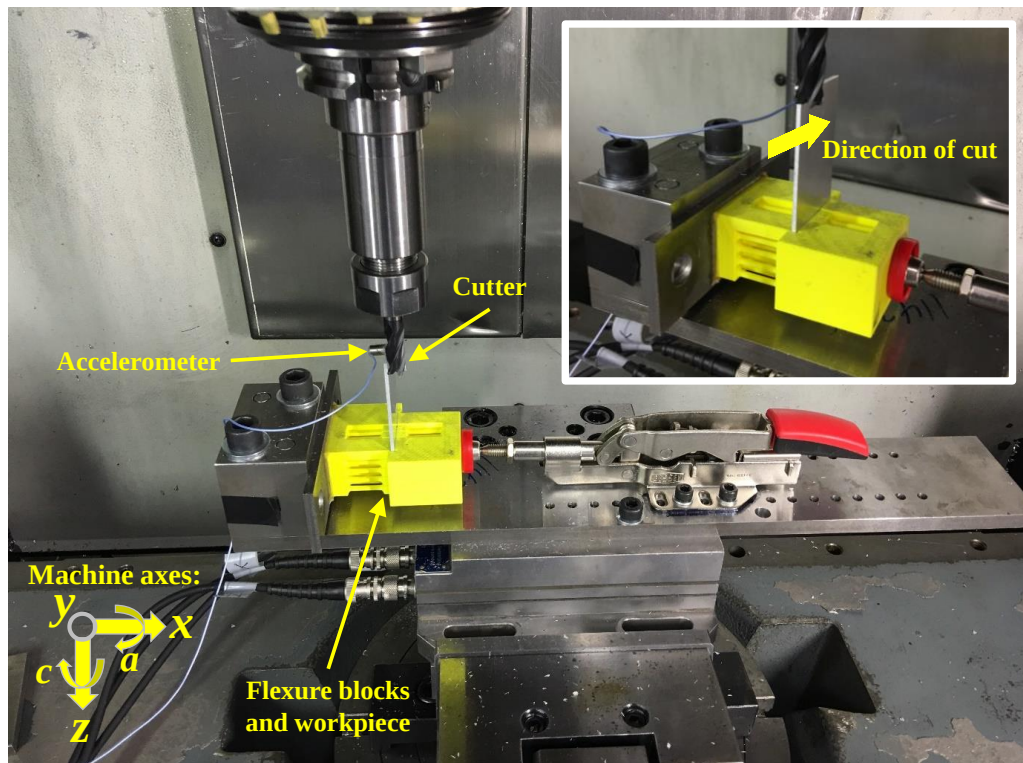
For this, an aluminium cantilever workpiece representing a simplified aerospace component was held in an AM fixture and end milled while vibration data was captured from an accelerometer, as shown in Figure 3-8. Milling was selected due to its ubiquity in industry, and the clean periodic excitation provided by a rotating milling cutter; a key consideration when the dynamic behaviour of the workpiece and fixture is to be investigated.

A flat cantilever workpiece was chosen for machining trials, due to its simple geometry compared to the blade-like workpieces used to study conformability. The blade geometry is more complex, introducing greater potential for errors in machining, because of its non-uniform stiffness across the aerofoil profile. It requires a complex tool path in order to maintain a constant rate of material removal in end milling, while it is simple to achieve time-invariant material removal with a cantilever using a straight tool path. The cantilever geometry also exhibits a simple free vibration response, dominated by a first bending mode of the free region, with a low density of eigenfrequencies across the range of excitable machining speeds with this setup. In all machining configurations the dimensions of the workpiece were kept constant, at 37×79×2mm. This cantilever length was selected for its frequency response, with a fundamental frequency that was distinct from the chosen machining excitation frequencies, and resonant frequency of the flexures; the 37mm width was used so that the fixed end of the workpiece made secure contact with the locating side walls of the fixture blocks; and a thickness of 2mm was chosen to ensure that workpiece stiffness was sufficiently low, such that the cantilever would be able to vibrate readily during machining.

To hold the workpiece during machining, three fixture variants were manufactured additively: a novel concept prototype, and two control fixture designs to allow comparison with conventional fixturing designs. All three fixtures featured the same external dimensions and locating features, but varied in the level and nature of support afforded to the workpiece.



(a)



(b)

Figure 3-8. Overview of Hermle C 800 U with ancillary testing equipment (a), and close-up of machining setup (b).

Appropriate tooling was selected by means of a machining stability simulation, and kept the same for all machining trials. The milled profile had depths of cut of 2mm axially, and 1mm radially, meaning no more than one cutting tooth made contact with the workpiece at any given moment using the chosen cutting tool (tool details are given later in this section). Cutting interaction from a single cutter tooth ensured that the milling action created clean periodic excitation. This is significant when investigating the effect of milling at different tool excitation frequencies, since simultaneous workpiece contact with multiple cutting teeth can diminish the periodic effect of excitation, introducing noise which makes the excitation frequency less prominent. Samples were machined at three spindle speeds, each of which corresponded to excitation frequencies of significance to the workpiece-fixture system, with feed rate increased in proportion to speed in order to maintain constant feed per tooth.

The following process was followed for each milling trial:

1. **Select fixture type and machining speed:** The cantilever workpiece is held in either the novel flexure-based fixture, or one of the control fixture designs. Machining occurs at one of three speeds, none of which cause resonance of the workpiece, however the middle speed causes resonance of the supporting flexure pins in the novel fixture. The remaining two speeds serve as control speeds in order to permit like-for-like comparisons between fixture designs in normal usage.
2. **Machine workpiece, acquiring vibration response:** A slot is milled along the surface of the fixtured workpiece, and workpiece vibration is measured by the accelerometer mounted near the slot at the free end of the cantilever. Accelerometer signals are recorded by the data acquisition system described previously.
3. **Process signals to quantify workpiece vibration:** Stable machining response (after any tool entry effects have subsided) is sampled, and the RMS of vibration is calculated. RMS vibration is compared for each fixture and machining speed combination, in order to assess performance of the fixture concept.

3.4.1 Additive manufacture

Additive manufacture allows the production of components directly from CAD models, with far fewer constraints on manufacturability than conventional processes, often enabling the manufacture of components that could not be realised conventionally. Modern AM processes enable the rapid and cost-effective manufacture of parts of great complexity, and in functional materials which mean they can be used for engineering applications.

The fixture prototypes presented are manufactured using the Fused Deposition Modelling (FDM) technique in ABS. A Stratasys Dimension SST 1200es 3D printer was chosen due to its ability to produce small features with a layer thickness of 0.254mm. Printing takes place inside a hermetic temperature-controlled chamber to ensure high repeatability of printed components. The machine can also print soluble acrylic copolymer support material, allowing overhanging geometries to be created without the need for manual post-processing, as the supports simply dissolve when placed in a bath of alkaline detergent solution, without affecting the main printed structure. Samples were printed in solid ABS for maximum strength, and oriented in order to give mechanical properties that were as close as possible to an isotropic structure. Specifications of the system are given in Table 3-8.

Three pairs of identical fixture blocks were produced by FDM (shown in section 6.2), with optimal geometries determined from models. The models used material properties for the AM material derived experimentally via mechanical testing from printed ABS test samples. All fixture blocks were printed in the same orientation, and using an identical raster pattern, to ensure identical geometry and mechanical properties. Each fixture block was machined to ensure flatness of key workpiece contact surfaces, and this was carried out before the printed support material had been dissolved in order to protect the blocks, by processing them while the entire geometry was encased by support material for maximum stiffness.

Table 3-8. Specifications of the Stratasys Dimension SST 1200es FDM 3D Printer [112].

	
Model	Stratasys Dimension SST 1200es
Printed Material	Model: Stratasys ABSplus-P430 Support: Stratasys Soluble SCA-1200
Maximum Build Size (X/Y/Z)	254×254×305mm
Printed Layer Thickness	0.254mm (model and support)
Printed Component Accuracy	±0.200mm, or ±0.005mm/mm (whichever is greater)

3.4.2 Machine tool

A Hermle C 800 U 5-axis machine tool, shown in Figure 3-8a, was used for the proof of concept milling trials. It has a maximum spindle speed of 15,000rpm (250 revolutions per second), and the feed rate can be altered in proportion with the chosen spindle speed in order to maintain consistent chip length, regardless of the frequency at which the workpiece is excited. Table 3-9 shows the key specifications of the machine.

Table 3-9. Specifications of the Hermle C 800 U Machine Tool [113].

Model	 Hermle C 800 U
Control	Heidenhain TNC 430
Spindle	Vertical, 10.5kW, ISO40, Maximum speed 15000rpm
X/Y/Z Axis	800×600×500mm
A Axis	-20° to +90°
C Axis	360°

3.4.3 Machining stability

Tool chatter occurs when the selected machining parameters are unstable for the chosen cutting tool, causing it to vibrate, which results in poor quality of machined surfaces in the finished component. It is possible to predict stable machining conditions for a given machining setup using data from the cutting

tool, machine spindle, and workpiece material, in order to generate a stability lobe diagram (also known as a stability plot, peak-to-peak, or PTP diagram), such as the example in Figure 2-19b. The stability lobe diagram shows how the maximum axial depth of cut that can be achieved before the onset of chatter varies with spindle speed of the machine, when other parameters such as radial depth of cut, choice of tool, and workpiece material remain constant.

MLI MilSim software was used to generate stability plots for numerous combinations of cutter diameter and overhang, in order to aid with selection of the tooling used throughout the milling validation presented in this thesis. MilSim is part of the MetalMAX package of machine tool dynamic characterisation software, which can be used to predict maximum stable depths of cut, and other parameters including optimal machining speeds and feed rates, based on measurements taken from a machine tool, spindle and cutter [114]. Full details of tooling are given in section 3.4.4.

Machining stability is highly dependent on the cutting tool, tool overhang, and machine spindle. A key stage in generating a stability plot was to determine the frequency response of the tool. Hammer modal testing was conducted with the cutter in situ in the machine, with drive point impacts on the flank of the cutter. The test was repeated with the accelerometer repositioned 90° around the cutter, in order to acquire the frequency response in the directions of the x and y axes of the machine spindle. The FRFs were interpreted to determine the eigenfrequencies of the tool, along with their corresponding stiffness, damping ratio, and modal mass, and this data was input to the MilSim simulation. Other details of the tool required include the diameter, number and spacing of cutting teeth, helix angle, and the chosen overhang in the tool holder.

This cutter data, along with the cutting stiffness coefficients of the workpiece material, and the chosen values for feed per tooth and radial depth of cut, are used by the software to define properties of the tool. A stability lobe diagram can then be generated over the desired range of machine speed and axial depth of cut. Figure 3-9 is a screen capture from MilSim showing the inputs used to

generate a stability lobe diagram for the machining configuration presented in this thesis.

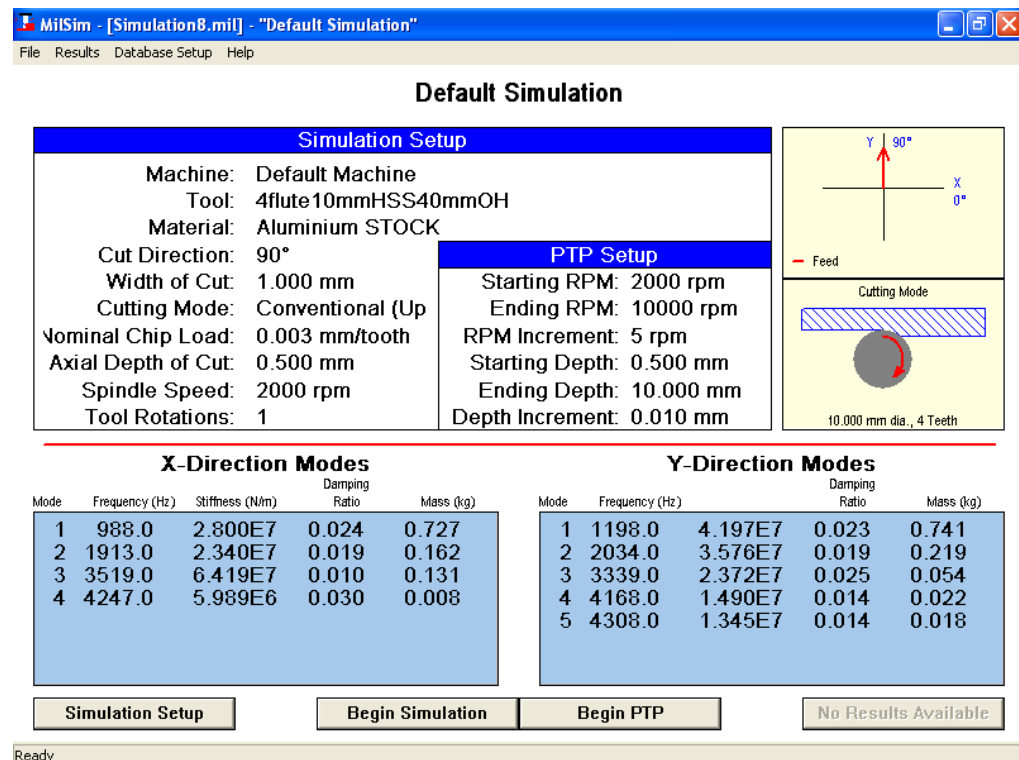


Figure 3-9. Example inputs to generate a stability (PTP) plot in MilSim.

3.4.4 Cutting tool

The cutter selected for machining was a 10mm diameter TiAlN-coated high speed steel 4-flute end mill, model Swiss Tech SWT-165-9010A. It allows the workpiece to be excited at a frequency of up to 1000Hz at the maximum speed of the Hermle C 800 U, providing a broad range of excitable frequencies. It is rated for large rates of material removal in aluminium (>50μm chip length). Its specifications can be found in Table 3-10.

It was chosen for its relatively large diameter which imparts high stiffness, therefore reducing its susceptibility to chatter. In end milling it is suitable for exciting the workpiece with a single cutting tooth for the chosen radial depth of cut, without multiple cutting teeth will making simultaneous contact with the workpiece (see section 6.3.1 for further detail). A conservative feed per tooth of

3 μ m was used for machining to ensure controllable levels of vibration were sustained within the unsupported region of the cantilever workpiece.

Table 3-10. Specifications of the Swiss Tech SWT-165-9010A end mill.

Model	 Swiss Tech SWT-165-9010A
Diameter	10mm – h6 (shank), k10 (cutter)
Overall Length	72mm
Flute Length	22mm
Flute Count	4
Helix Angle	30°
Material	TiAlN-coated high speed steel

The dynamic properties of the cutter make it suitable for this machining setup, permitting stable cutting of slots in aluminium cantilevers. With the chosen radial depth of cut, the stability analysis revealed that stable machining was possible at almost 200% of the chosen axial depth of cut across the entire machining speed range, with stable peaks permitting significantly greater material removal.

3.4.5 Vibration data acquisition

The PCB 352B01 accelerometer (specifications given in Table 3-2) was used to capture workpiece vibration during machining, and was mounted to the end of the cantilever workpiece as shown in Figure 3-8b. Its low sensitivity made it ideal for machining, as it is capable of withstanding large peaks of acceleration without damage. Data was captured using the same Prosig P8004 system that was used for other signal acquisition, at a sample rate of 10,000s⁻¹, which was high enough to prevent aliasing at any of the frequencies of interest during

machining in accordance with the Nyquist-Shannon sampling theorem. The measurement chain is shown in Figure 3-10.



Figure 3-10. Measurement chain schematic for workpiece vibration in machining.

3.5 Method for Finite Element (FE) analysis

At each of the previous stages of experimental work, the configurations tested were modelled in order to predict their behaviour, and verify experimental results, using Finite Element (FE) models generated in Abaqus FEA. The modal testing, conformability, and machining elements each required different analysis steps, and the methodologies used are detailed in the following subsections. The use of simulation proved invaluable throughout this research for predicting the geometric properties, and responses to external conditions, of numerous structures and assemblies, therefore eliminating guesswork in determining geometries of interest and suitable testing parameters.

3.5.1 General FE approach

The finite element method generates solutions to problems involving complex physical systems by dividing them into a series of smaller systems. These individual discretised problems, or finite elements, numbering in the thousands in complex analyses, allow a large problem to be broken down into a series of matrix equations. Solving these equations for each problem, and at each time increment, provides a solution to the overall problem, and assignment of physical properties to the elements, such as material definitions, allows numerical values for natural frequency [115], stress and displacement [116] etc. to be generated. The system to be simulated is partitioned and meshed (split into discrete elements) in accordance with the requirements of the model. Results can be visualised to show displacement or stresses in a structure, and the output from a simulation can be used for subsequent FE analysis.

The matrix equations which describe the behaviour of finite element systems are defined by Hamilton's Principle [117], whose fundamental statement can be expressed as:

$$\delta \int_{t_0}^{t_1} (T + W) dt = 0$$

Where T is the kinetic energy of the system, W is work performed by internal and external forces, t is time, and δ is the variational operator. According to the principle of *virtual work*, Hamilton's Principle can be reduced to:

$$\delta W = 0 \quad (if \ T = 0)$$

In the case of modelling the response of a mechanical system that receives some excitation, the displacement experienced by the system in response to forces are determined by the stiffness relationship in each element (i) at each time interval. The equation in matrix form is given by:

$$\{f\}_i = [k]_i \{d\}_i$$

Where f is the force acting on that element, k is the stiffness, and d is the resulting displacement. Applying continuity equations to define the relation between neighbouring elements, in order to associate loads and displacements across multiple elements, allows linking of all equations to produce a single matrix equation for all elements (for each time interval):

$$\{F\} = [K]\{D\}$$

Where F , K , and D are force, stiffness, and displacement of the whole system. When boundary conditions are defined, in order to specify which points in the system have known displacements, the equation is modified:

$$\{F_{BC}\} = [K_{BC}]\{D_{BC}\}$$

Such that the global stiffness matrix can be inverted and the equation solved to reveal the vector of unknown displacements:

$$\{D_{BC}\} = [K_{BC}]^{-1}\{F_{BC}\}$$

This process is repeated for each time step, meaning transient dynamic analyses involve extremely large amounts of computation. However, modern computer hardware allows even relatively complex simulations to be run in a matter of hours. Post-processing of the dataset produced by the solver allows the requested metrics for each element to be quantified. Readers are referred to

Becker [118] for a comprehensive theoretical introduction to finite element analysis.

An overview of the entire modelling process is shown in Figure 3-11. In each case, the structures to be modelled were designed in CAD software, with their geometries exported as STEP files for analysis. The imported geometries were partitioned until the entire structure had been simplified into a series of constituent prismatic geometries. This enabled an effective meshing strategy with the inbuilt Abaqus meshing engine. Each structure was meshed with elements appropriate to the intended analysis, taking care to ensure that mesh quality was high (minimisation of element skew, parallel deviation, and elongation). Mesh density (the number and size of elements) was refined to the point of convergence, meaning that simulation outputs no longer changed as mesh became finer, thus providing maximum accuracy without excessive computation times.

After mesh assignment, each structure was assigned mechanical properties, and all parts (if more than one) were arranged in an assembly. Next boundary conditions were used to constrain the parts to realistically recreate the conditions of the structure to be tested (e.g. clamped, suspended). Where surfaces in the model will make contact, e.g. a workpiece clamped in a fixture, contact interactions must be created to define the friction properties of those surfaces. Once the model is constrained, the analysis can be defined. This can include external conditions to be resolved such as forces, displacements, and periodic excitation. Once the entire job definition has been created, the model can be run.

Upon completion of the simulation, the output can be viewed in the results module of the software. This can be used to visualise displacements, deformations, stresses, and numerous other parameters specified in the model. From this, data can be exported for further analysis, or the simulated state can be used as a starting point for a subsequent stage of analysis.

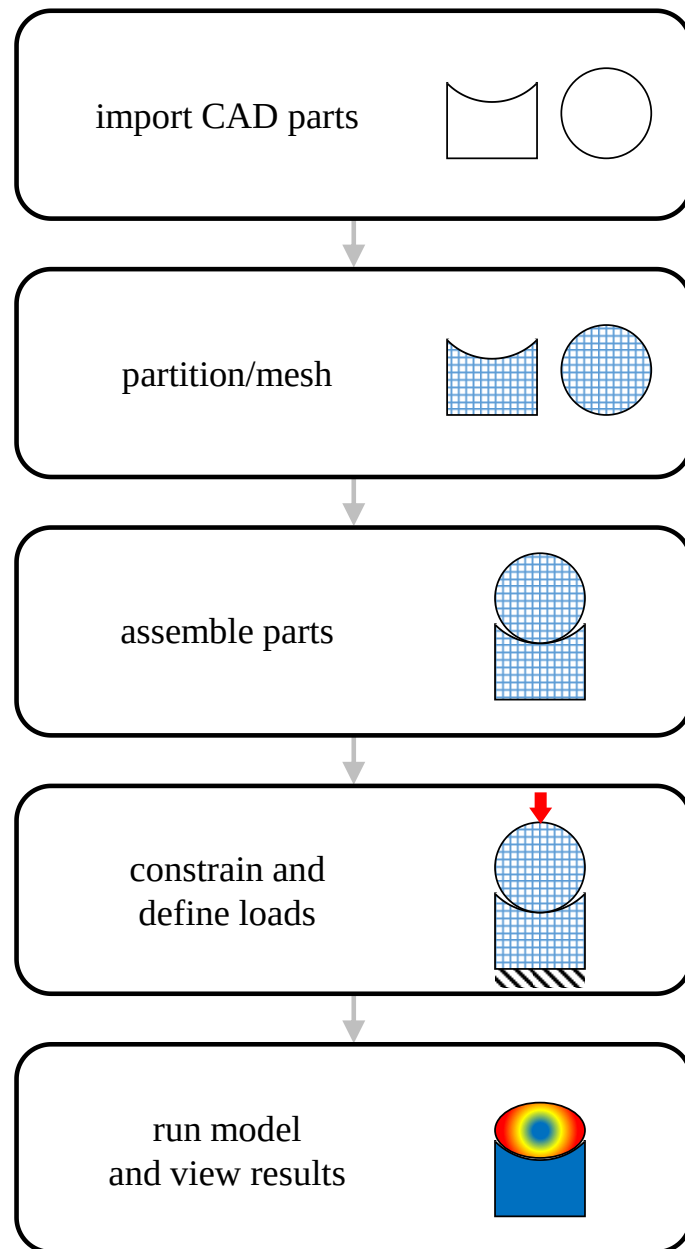


Figure 3-11. Flow chart describing the major stages of an FE analysis.

3.5.2 Modal analysis

One of the simplest varieties of FE model conducted in this research is modal analysis, in which the eigenfrequencies of a structure can be determined, along with a visualisation of their corresponding mode shapes. The modal analysis step of a model is relatively quick to compute, as no loading or excitation is defined; the only input is the geometry of the model, from which unitless visualisations of displacement are produced, such as the example in Figure 3-12b.

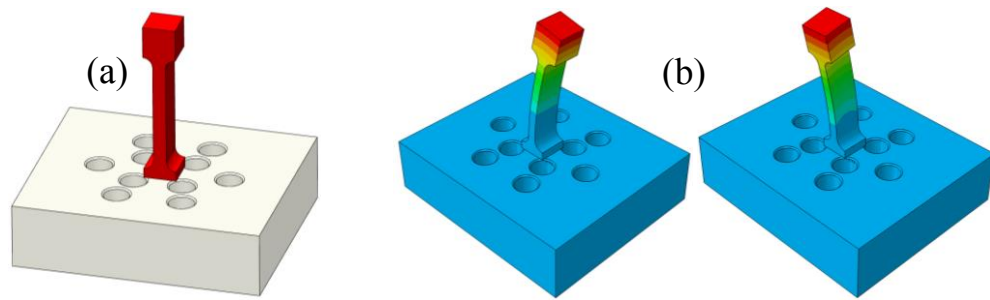


Figure 3-12 – FE modal analysis showing (a) input geometry from CAD, and (b) visualisations of the first two mode shapes of the structure.

The input geometry can be modelled free-free, where it is not constrained in any degree of freedom, or with boundary conditions applied to simulate ‘real-world’ configurations such as clamping or mounting. Modal analyses are useful for extracting the frequency response from individual components, but also for large assemblies containing multiple components. This can be especially useful for predicting the behaviour of multiple real-world components when they are assembled together.

While modelling any anisotropic structure increases the complexity of the analysis, simulating AM structures produced by FDM presents a particular source of uncertainty that must be investigated. In addition to the different mechanical properties in each axis of the printer (orthotropy), the printed geometry departs noticeably from the CAD model (and therefore the structure modelled in FE) at the sub-mm scale (see Figure 2-43), due to the nature of printed filament layers and raster patterns. There is little published work on the modelling of such printed structures, however it was determined that the use of optimal printing parameters enabled accurate modal analysis simulations with isotropic material definition. Setting the FDM machine to print solid structures (no material-saving hollow internal structures), at the highest quality setting (improved accuracy at the expense of build speed), along with optimal component orientation (flexure supports normal to printed layers), results in prediction of eigenfrequencies which were shown experimentally to have small error.

Modal analyses were validated against the hammer modal tests described previously, with the models refined to achieve optimum accuracy. The good predictive capability of FE modal simulations allowed the frequency responses of assemblies (e.g. a fixtured workpiece), and even additively manufactured components, to be predicted with error well below 5%.

3.5.3 Conformability assembly

For the study of fixture conformability on workpiece deformation, several aspects of the fixture were simulated. Before modelling workpiece-fixture assemblies themselves, FE was used to optimise the design of fixture blocks for modal response and compressive strength. This required *frequency* (modal) and *buckle* linear perturbation analyses.

To simulate the behaviour of a workpiece when clamped in a fixture, an assembly was created with a workpiece between two fixture blocks, recreating the setup shown in Figure 3-4, with loading applied via a *static, general* analysis step over a single time interval. Since all components in the assembly were of constant section, the model was created as a 2D system in order to reduce computation time. More details of both analyses are given in Chapter 5.

3.5.4 Machining dynamic response

The final and most sophisticated model involved simulating the machining response of a fixtured workpiece. The fixture blocks used to hold the cantilever workpiece during machining (Figure 3-8b) were produced by additive manufacture, with their geometries determined using FE. Modelling the individual fixture blocks ensured that the locating and supporting elements were of adequate strength to sustain clamping and machining loads, and also allowed refinement of the flexures to ensure they exhibited suitable stiffness and frequency response properties.

Based on this fixture geometry, the dimensions of the cantilever workpiece were determined so that it would be sufficiently stiff to withstand machining, but also sufficiently flexible that machining with suboptimal parameters would cause noticeable vibration.

While the models discussed previously have only simulated the free vibration responses of structures to obtain their eigenfrequencies, simulating the machining response of a workpiece-fixture system involves modelling forced vibration. The machining simulation built upon the previous models used to investigate conformability, with the first stage of analysis involving clamping of the workpiece in the fixture. This clamped state was used for a modal analysis to determine the eigenfrequencies of the system. In order to identify frequencies of significance to the structure, to aid with the selection of suitable machining parameters, a harmonic analysis was carried out on the structure. Following this, a transient *modal dynamics* analysis step was undertaken, in which machining excitation was replicated via a force applied to the workpiece with a periodic amplitude that mimicked milling forces at various spindle speeds, allowing the transient response of the assembly to be studied. Modelling the dynamic behaviour of the various fixture systems, in conjunction with machining stability analyses, was used to determine excitation frequencies of interest to the system, which were subsequently validated in the machining experiment described previously. The model is outlined in greater detail in Chapter 6.

3.6 Summary

The methodologies presented in chapter cover the major research components of this thesis, providing equipment specifications, and overviews of experiments and simulations. Sensors, tooling, and software are detailed, along with descriptions of the processes followed in order to enable thorough performance evaluation of the novel fixture concept.

While each of the four experimental components of this research have been described separately, there is considerable overlap between sections, since modal analysis was used at all stages throughout the project, and all of the physical systems tested were modelled using FE. In the following chapters of this thesis, the data these methodologies produce are discussed in detail (modal testing – Chapter 4; fixture conformability – Chapter 5; machining evaluation – Chapter 6). Each chapter contains a commentary on how this work realises its

objectives by contributing to knowledge in frequency response testing, holding of delicate components, and reducing workpiece vibration in machining, in order to demonstrate an innovative fixture that offers improved capability over existing systems.

4 Novel fixture concept

4.1 Introduction

Presented in this chapter is the novel fixture concept based on tuneable supporting structures, hereafter referred to as flexures. The key features, advantages and disadvantages of conventional fixture systems are discussed, identifying potential opportunities for flexures to improve fixture performance. The design of flexures is explained, including how their geometries are tuned. Finally, a series of flexure pegs in a reconfigurable testing platform were subjected to modal testing, and used to develop and refine a FE model to predict flexure behaviour.

4.2 Concept overview

Fixtures are essential for the machining, assembly, and inspection of components, providing critical location and holding. Location accuracy is extremely important for the production of components with accurate and repeatable geometries, since a poor quality fixture has the potential to introduce significant errors, thus limiting the accuracy of an otherwise precise manufacturing process. A workpiece is located by removing degrees of freedom until it is entirely constrained, and this can be achieved creating contact across any combination of points (e.g. domed locators), lines (e.g. V-blocks), and planes (e.g. vices). Holding is achieved via the application of clamping forces, which resist any process forces (e.g. cutting forces) during manufacture in order to maintain location of the workpiece.

For the manufacture of thin complex geometries, such as aerofoil components, two approaches are typically employed: location and clamping based on discrete locators (e.g. modular fixtures), or planar clamping surfaces (e.g. vices). The working principle of these fixture schemes, along with the novel concept, are illustrated for a typical low stiffness freeform workpiece in Figure 4-1.

Figure 4-1a shows the most commonly employed fixturing approach for freeform components, in which the workpiece is placed on discrete locators

following an n-2-1 fixturing scheme, with an opposite force applied to ensure workpiece-locator contact. This approach provides limited support to the surface of the workpiece in regions where there is no contact with locators, meaning the component is able to vibrate freely if machining excitations cause it to resonate. With the exception of highly localised deformation where clamping forces are concentrated about the locators, there is also limited potential for deformation of the entire workpiece, since the fixture will not deform the surfaces of the component if there are small errors in its geometry.

Another existing fixturing system, and common alternative to the above approach, is shown in Figure 4-1b. The fixture follows the contours of a freeform component with a continuous rigid clamping surface. This full-face contact provides enhanced support to the workpiece, and reduces stress at the workpiece-locator interface by distributing clamping forces over the entire surface of the component. It can also potentially help to prevent workpiece vibration during machining, since displacement of the component is inhibited. However, since location and holding is achieved by mating the conjugate surfaces of the fixture and workpiece, there is significant potential for the introduction of undesired workpiece deformations if the workpiece has any geometric errors/deviations from its nominal geometry. This is because a low-stiffness component will be deformed during clamping so that it fully adopts the profile of the fixture, until any gaps between the workpiece and fixture have been closed.

Shown in Figure 4-1c is the innovative fixturing concept developed to improve upon the holding capability of existing fixtures, by alleviating concerns regarding workpiece vibration and deformation in the manufacture of thin, complex geometries. The concept is based on the first fixture with discrete locators, but enhances it by filling the voids between locating features with an array of deformable inter-locator flexures. The flexures conform to the workpiece by following its surfaces to achieve near full-face contact between the fixture and the workpiece.

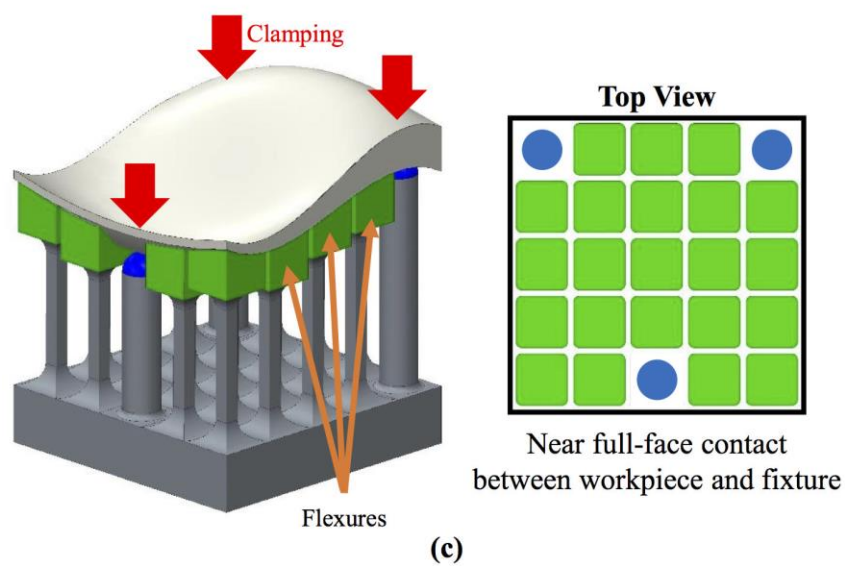
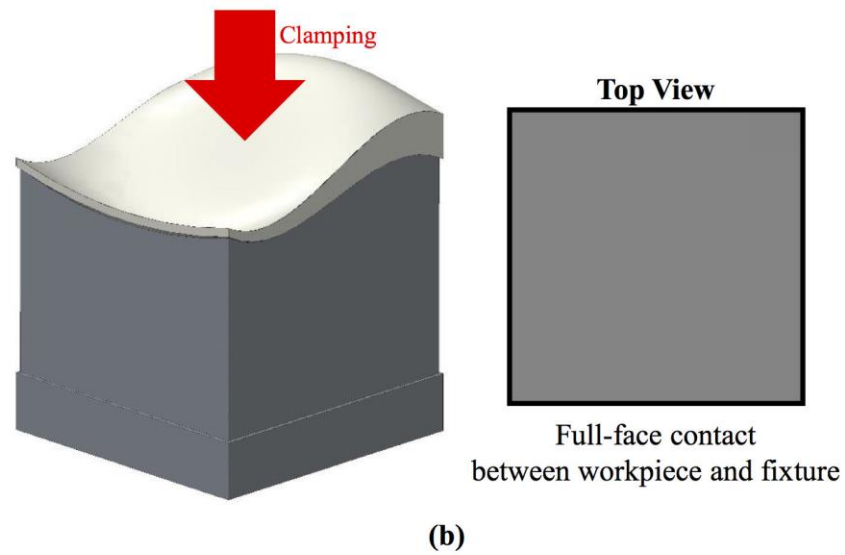
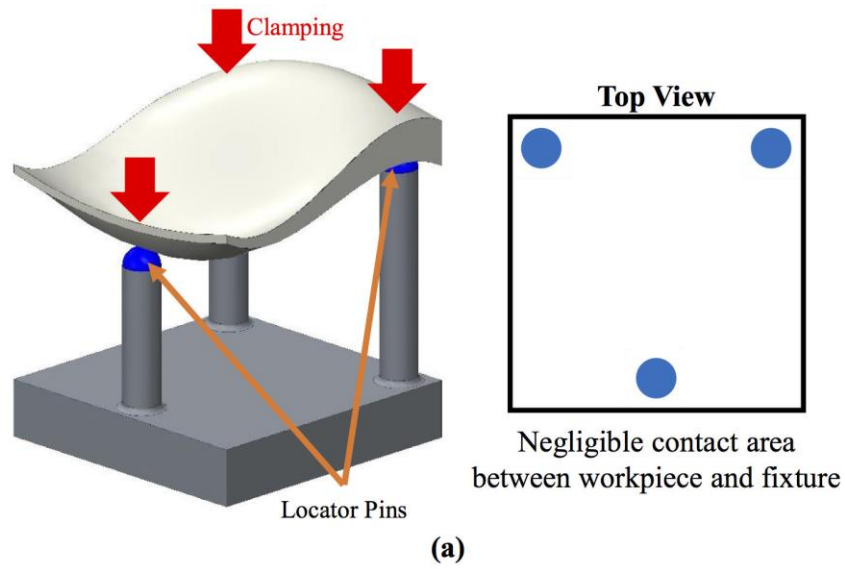


Figure 4-1. Conventional fixture for a freeform component with domed locators only (a); conventional fixture with a rigid conjugate surface (b); and novel conformable fixture with flexures to support the workpiece (c).

Flexures feature a large workpiece contact surface at the end of a deformable cantilever of tuned geometry with a known frequency and/or buckle response. By incorporating elements from both conventional fixture systems, the flexure concept aims to produce a hybrid fixture system that combines the benefits of fixturing using discrete locators (reduced workpiece deformation) and rigid planes (improved workpiece support). The inclusion of flexures provides support across the entire workpiece surface, thus incorporating the displacement-inhibiting benefit of a continuous rigid clamping surface, but without introducing the associated potential for deformation of the component if mismatches exist between the profiles of the workpiece and fixture. This is due to the deformable nature of flexures, which are designed to exhibit lower stiffness than the components they are supporting, meaning they are able to support component surfaces without introducing significant deformation.

The novel fixture concept presents an important development for the fixturing of blanks produced by manufacturing processes which are inherently subject to substantial variability, e.g. welding, casting, forging, and composite layup, where the exact geometry of blanks is inconsistent and difficult to predict. The conformable nature of the flexure concept means that fixtures are able to accommodate components with variations from their nominal geometry, without simply deforming them until they adopt the shape of the nominal component. This method of achieving full-face support is likely to offer a significant advantage in the fixturing of thin components, by providing inhibition of vibration-induced displacements, while introducing negligible deformation compared to a rigid fixture system. The ability of flexures to accommodate workpiece geometric variability is investigated in Chapter 5.

By increasing the contact area between the workpiece and fixture from discrete points to large surfaces, holding loads are distributed over a significantly larger area, which eliminates points of concentrated stress in the workpiece. This lower stress helps to reduce clamping-induced elastic deformation, and prevent damage due to plastic deformation. Aside from the implications for component accuracy, minimisation of clamping deformation during machining is also

desirable since material removal from a workpiece while it is in a deformed state will increase residual stresses, which weaken the finished component.

Another advantage of the flexure system is in the machining of thin, vibration-prone workpiece geometries. By tuning the geometries of flexures, it is possible to shift their frequency response so that they resonate at almost any chosen eigenfrequency. Exciting flexures to elicit resonance offers the potential for damping capability, since flexures vibrate readily due to their low stiffness, and this can be exploited to assist with the dissipation of energy. When flexures are coupled with a workpiece in a fixture, they have the potential to offer a path of least resistance for vibration in the system, meaning a resonating flexure can help to convert kinetic energy in the system to thermal and other losses, thus creating a damping effect that reduces vibration in the workpiece. This is investigated in Chapter 6.

A key feature of flexures is their low stiffness and small size relative to the workpiece-fixture assembly as a whole, which gives them a low modal mass compared to other eigenfrequencies present in the system, e.g. modes of the workpiece. This means the addition of flexures to an existing fixture will result in negligible alteration to the frequency response of the workpiece-fixture system. This is a major advantage because it means flexures can be introduced to a problematic fixture setup without necessitating the modification of existing clamping methods, tooling, or cutting parameters in a machining operation.

Given the complexity of the flexure fixture concept, with its array of tightly packed flexures and large freeform surface, it lends itself ideally to additive manufacture. It would be time-consuming, and therefore costly, to manufacture a fixture block such as the example presented in Figure 4-1c by machining, and it may prove impossible to manufacture as a single piece by conventional methods, due to closely packed internal features that cannot be reached with milling tools. With additive techniques there is no need to consider *design for manufacture* when creating the fixture, since printed support structures allow the realisation of virtually any geometry, including overhanging regions and pre-assembled elements. Unlike with conventional manufacturing techniques,

there is no inherent cost associated with the addition of complexity, meaning it is advantageous to maximise the quantity and complexity of features where this will benefit fixture performance. Therefore additive manufacture is an integral aspect of the concept, because it allows the greatest possible flexibility of fixture geometries to be produced, and is the fastest and most cost-effective way of realising the necessary features.

To enable development of the flexure concept into a viable fixture system, the attributes that govern the behaviour of flexures must be understood so that they can be tuned in order to exploit the conformability and damping phenomena. Flexure responses must be accurately predicted, so that printed fixtures exhibit the desired properties, in order for additive manufacture to compare favourably to conventional processes in terms of cost and speed. The following stages are therefore necessary for the design of flexures leading to the realisation of an additive flexure-based fixture that improves upon existing designs:

1. Start with a simple prismatic geometry (e.g. a square pillar), and transform it into a flexure by adding features such as an enlarged workpiece contact surface, and a mounting region that constrains it to a base substrate as a cantilever. The cantilever, workpiece contact, and mounting features must be blended (e.g. with a chamfer or radius) to avoid sharp transitions between regions, which would otherwise reduce the strength and stability of the flexures when they are deformed during clamping.
2. Analyse the flexure for suitable properties, such as stiffness and frequency response, e.g. by buckling or modal testing. If needed, tune flexure properties by altering its geometry (flexure height, and cross section of the prismatic region) until the desired properties are obtained.
3. Simulate the response of the flexure using an FE model to ensure that its properties can be predicted accurately. Once the model has been refined and is proven to be reliable, it can be used to predict the response of flexures such that entire printed fixtures exhibit the desired properties to improve holding and machining performance.

These steps are necessary to understand the features that govern the compressive and frequency response of flexures, which enables the design of innovative

flexures with improved performance. Exploiting the features of the novel fixture concept will therefore facilitate the manufacture of components of greater accuracy, due to the absence of problematic clamping-induced deformation, and reduction of vibration during machining. The process followed for the design, testing, and modelling of flexures is detailed in the following subsections, with validations of flexure fixture conformability and machining performance presented in subsequent chapters.

4.3 Flexure design

As can be seen in Figure 4-2, the geometry of a flexure is defined by a thin, deformable column, with a simple prismatic structure, of known response in compressive loading and/or free vibration. A flexure is created by mounting the base of the column to the fixture substrate to constrain it as a cantilever, and thickening the column at its free end into a large contact surface in order to accommodate and support a freeform workpiece, thus creating a quasi-continuous fixturing surface.

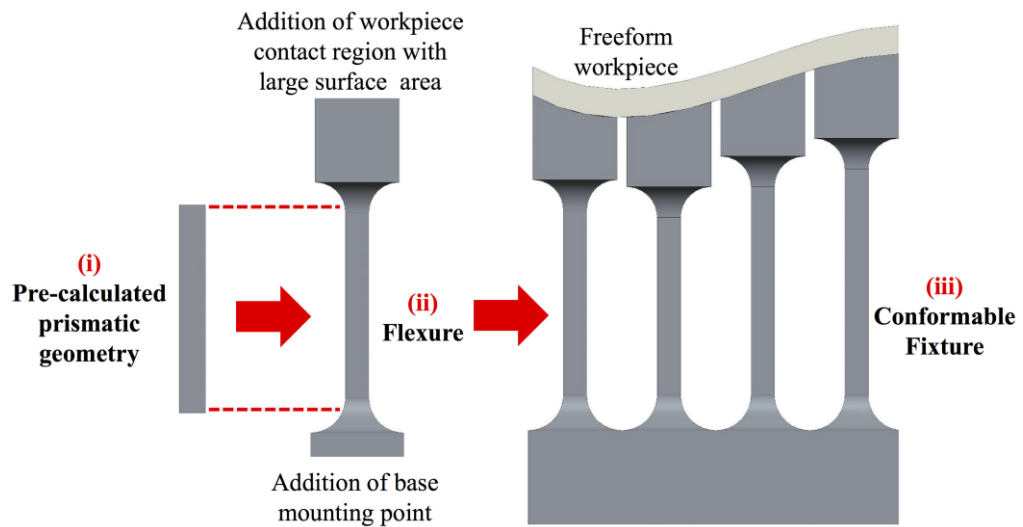


Figure 4-2. Example flexure-based fixture, showing how its features are derived from an initial known flexure geometry.

The first experimental component of this research was to investigate the effect of different flexure geometries on frequency response. A mounting plate was designed alongside a number of prototype flexure test pegs with varying height and thickness of the flexure region. Two different profiles of flexure peg were

examined: a square pillar design, in which the flexure region was symmetrical in depth and thickness; and a two-dimensional I-beam, in which the flexure region had a rectangular cross section. It was necessary for pegs to be securely and rigidly coupled with the base in a reconfigurable way, as is shown in Figure 4-3, however it would not be possible to achieve rigid mounting of additive components printed in ABS without gluing them in place, which would prevent easy reconfiguration of the pegs, and therefore the parts were manufactured conventionally in aluminium. However, since additive manufacture allows the production of solid components with near-isotropic properties, the response of an aluminium flexure is therefore directly analogous to a printed one.

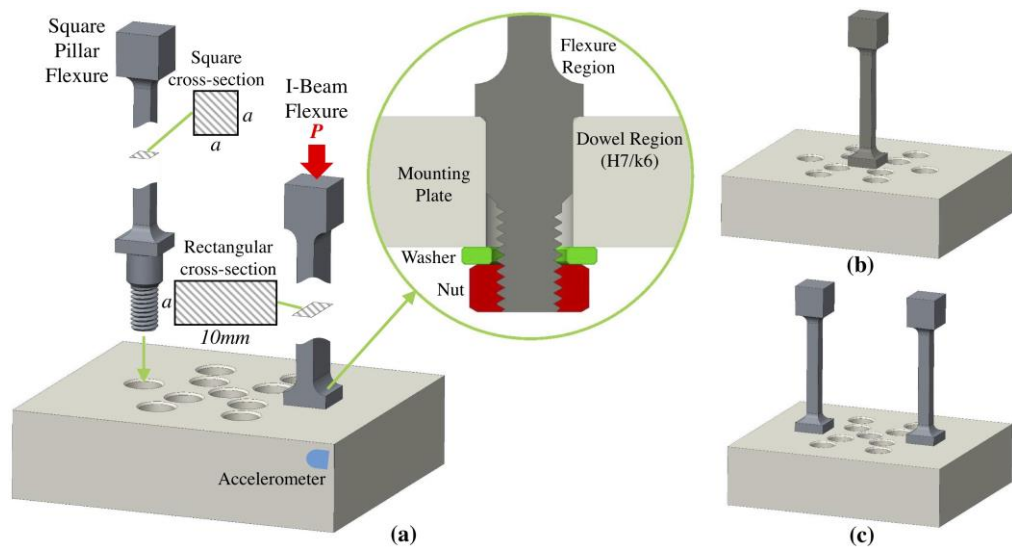


Figure 4-3. Flexure peg variants and mounting plate, showing the mounting mechanism (a), with example single peg (b) and multiple peg (c) testing configurations.

The flexure pegs were scaled up in size compared to those that would be found in a typical fixture, with the aim of facilitating more accurate measurement and analysis of flexure response to clamping loads and dynamic excitation. All flexure pegs had the same external dimensions, so that they were completely interchangeable: 50mm overall length of the unsupported region, and a cube with sides of 10mm at the free end to give the flexures a flat upper contact surface (for the dimensions of each flexure variant, see Table 4-3). Each peg incorporated a dowel-based mounting system at its root, which allowed pegs to be securely fixed to the mounting plate in a variety of configurations (individually, in rows, and in square/hexagonal arrays). The pegs were tightened

in place to a uniform torque of 4Nm, to ensure dynamic coupling between the flexures and the base. The PCB 352C23 accelerometer was attached to the mounting plate as shown for modal testing.

Flexures behave as cantilevers with a simple, predictable vibration response. In a fixture, the flexures will be in compression when clamping is applied and they become engaged with the component. Thus, the sensitivity of the frequency response to clamping loads must be considered if a chosen eigenfrequency needs to be maintained during clamping for the purpose of damping workpiece vibration. A preliminary experiment was therefore undertaken to investigate how the eigenfrequencies of a flexure-like geometry are affected by clamping. A length of ¼” square aluminium bar was clamped longitudinally in a vice, with clamping forces measured by a load cell. Hammer modal testing was conducted at increasing compressive loads to measure the shift of mode frequencies in response to the applied clamping. As can be seen in Figure 4-4, the first eigenfrequency shifted negligibly for modest loads (less than 1% shift at 500N), with the mode frequency only starting to decrease appreciably when clamping approaches 2kN (corresponding to a compressive stress of 49.6 MPa).

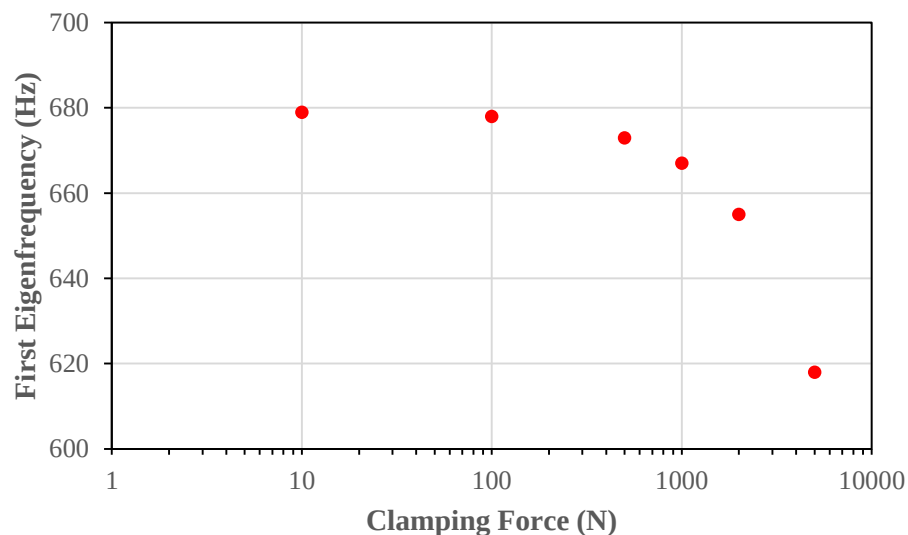


Figure 4-4. Shift of first eigenfrequency under increasing compressive loading.

These stresses are considerably higher than would be found in a typical fixture for a low stiffness workpiece, meaning the shift of critical eigenfrequencies can

therefore be assumed to be negligible as long as flexure compression remains elastic, and purely longitudinal (meaning no stiffening effects are present due to lateral bending of the flexure). From this it was concluded that the frequency response of flexures is not affected by the addition of clamping loads of the magnitude required to hold a workpiece in a fixture. This is a significant finding because it means flexures behave predictably when incorporated into a fixture, and therefore the dynamic response of a fixture incorporating flexures is not clamping force dependent.

The frequency response of each flexure peg geometry was captured by hammer modal testing. In addition to individual flexure pegs, multiple flexures were also tested, including pairs of identical flexures and rows of three dissimilar pegs. The mounting plate was held along its short edges in a custom jig by a toggle clamp, shown in Figure 4-5. The plate was raised so that its underside was not in contact with the base of the jig, in order to minimise the influence of mass effects from the jig and workbench. Hammer testing results, along with the models that were produced to predict flexure response (and later the response of entire fixtures) are presented in the following subsection.

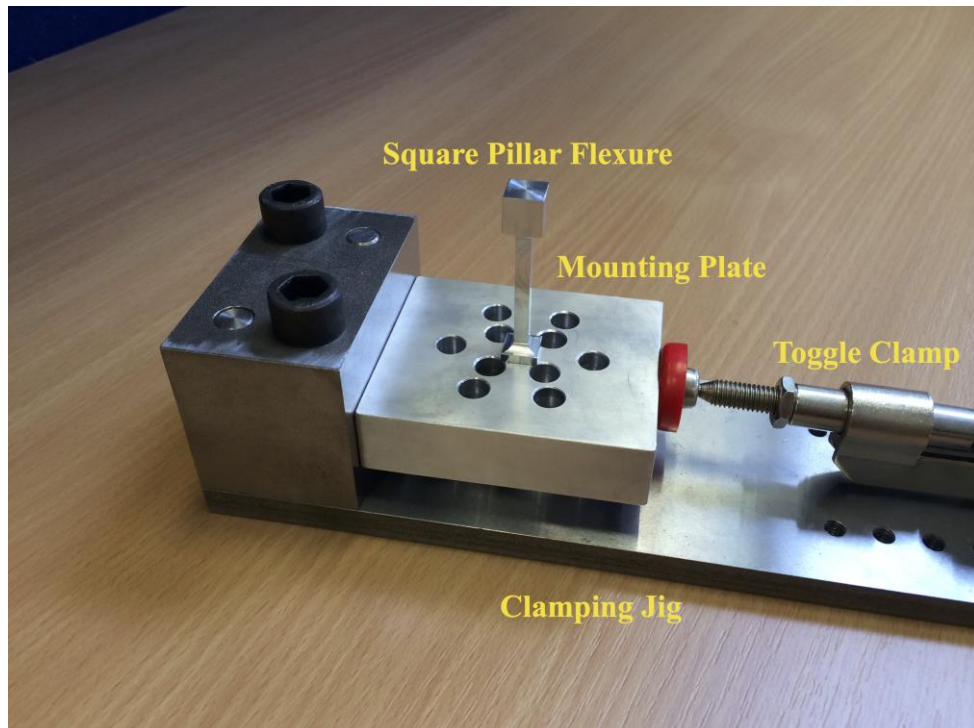


Figure 4-5. Jig holding the mounting plate with a single flexure peg for measurement of frequency response.

4.4 Models to predict flexure frequency response

An FE model was created to predict free vibration response of the flexure pegs in the mounting plate, allowing comparisons between simulated and experimental modal testing data. It is essential that the models are accurate, because the inclusion of flexures must not interfere with the existing frequency response of the workpiece-fixture system, in order for any benefits brought about by flexures not to be offset by undesired alteration of the dynamics of the machining process, e.g. by causing resonance of the workpiece. Flexures must resonate at the chosen frequency within a fixture in order for any frequency-specific damping effect to materialise, and therefore the precise prediction of flexure response before a fixture is printed is a vital component of the novel fixture concept.

The model, shown in Figure 4-6, was constructed using Abaqus, with geometries imported directly from CAD models. Material definitions for the aluminium parts are given in Table 4-1. The mounting system for flexure pegs was de-featured by simplifying the geometry of the thread to a simple cylinder with the same external dimensions. Mounting of the peg to the plate was replicated by defining a contact interaction for the dowel region and plate contact surface using the inbuilt *rough* friction formulation in Abaqus. This means no slip can occur once surfaces are in contact, and since the components are pre-assembled in the model, this interaction property is therefore analogous to a tie constraint. Complex features of the geometries were partitioned extensively to allow for optimal mesh seeding, with parts meshed primarily using C3D8R elements (8-node linear hexahedral brick elements with reduced integration), and C3D6 wedge elements used to mesh circular features. Linear elements were used because they were found to enable faster simulation of frequency response compared to 20-node quadratic elements, without affecting accuracy. This is because mode shape visualisations are dimensionless, and therefore the reduced accuracy of linear elements for predicting deformation and stress had no adverse effect on the model, since these outputs were not requested. Particular attention was paid to mesh quality, with an aspect ratio of

less than 1.01 (where lower is better and 1 is ideal) achieved for all elements within the deformable region of the flexure.

Table 4-1. Material definitions for modal analysis of flexure pegs.

2014 Aluminium Alloy	
Density (kg/m³)	2780
Young's Modulus (GPa)	74
Poisson's Ratio	0.33

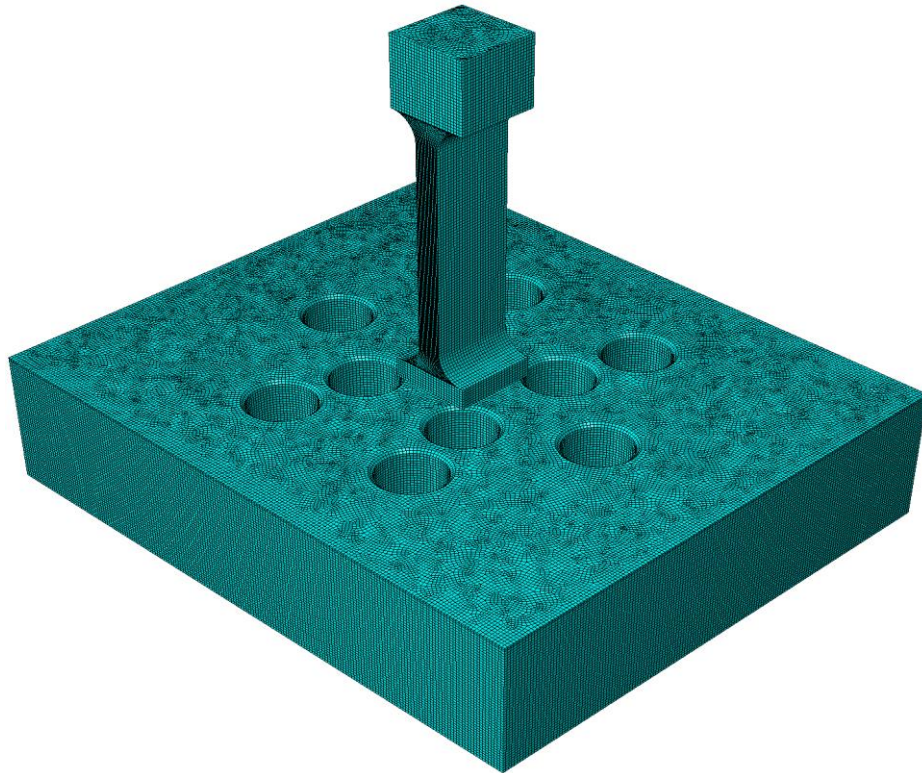


Figure 4-6. Meshed parts for modal analysis simulation of flexure pegs in the mounting plate.

The plate was constrained along its sides to recreate the clamping conditions used experimentally for modal testing. The modal analysis step (Linear Perturbation: Frequency) used the Lanczos eigensolver to extract the natural frequencies of the flexures. 10 different configurations were modelled, each with between one and three flexure pegs, following an identical process to create every model to ensure consistency between configurations.

A mesh convergence study was undertaken to determine optimal mesh refinement for the flexures, so that models could achieve greatest possible accuracy in the minimum computation time. The first three eigenfrequencies of a peg in the mounting plate were calculated, repeating the analysis with various densities of the mesh. The mesh density of the entire model was varied so that the number of elements over the deformable region of the flexure (30mm in height) ranged from 30 to 240, with the full dataset given in Table 4-2. It can be seen that convergence to the nearest 1Hz (the resolution of the hammer testing kit) occurs when the flexure height is comprised of 180 elements, with further increases to mesh density having a negligible impact on mode frequencies. This density, which corresponds to nominal element dimensions of a cube with sides of 0.167mm, was therefore used for each subsequent flexure model.

Table 4-2. Mesh sensitivity study for modal analysis, showing shift of mode frequencies as mesh density is increased.

Elements Along Flexure Length	Mode 1 (Hz)	Mode 2 (Hz)	Mode 3 (Hz)
30	1099.3	2445.2	7056.4
60	1126.0	2454.6	7219.9
90	1130.9	2456.3	7249.4
120	1132.6	2456.8	7259.9
150	1133.4	2457.1	7264.5
180	1133.8	2457.3	7266.8
240	1134.1	2457.4	7267.4

Experimental modal testing was conducted with the accelerometer attached to the mounting plate, using cyanoacrylate adhesive to provide rigid coupling, and prevent accelerometer movement between hammer impacts. Drive point impacts (exciting the structure at the location of the accelerometer) were used to ensure simultaneous uniform excitation of all flexure pegs, and to eliminate any potential mass effects that placement of the accelerometer directly on a flexure may have on its frequency response. These measures help to reduce

errors arising due to the modal testing setup, and also minimise any dissimilarities between the experimental and simulated configurations. By ensuring the mesh was of high quality and sufficient density, and that the parts were accurately constrained to create a true representation of the experimental hammer testing setup, the predicted eigenfrequencies show excellent agreement with experimentally-derived values. The full dataset is given in Table 4-3 and Table 4-4.

The error between simulated and experimental values was below 5% in all cases, thus proving the model to be highly accurate. This is a significant finding because it demonstrates that FE models can be used to reliably predict the frequency response of flexures with any dimensions. The presence of multiple pegs in close proximity on the mounting plate is shown to have a small effect on the frequency responses of individual flexures, with shift of the first eigenfrequency shown to be typically no more than 2%; however this phenomenon is accurately predicted by the models, and therefore can be accounted for when simulating fixture geometries. This ability of a flexure to behave independently of neighbouring flexures, mounting surfaces, locating features, etc. is due to their relatively low stiffness, which causes them to vibrate readily, and typically at much lower frequencies than their surroundings. This has useful implications for the design of fixture systems incorporating flexures, since the tuned resonant properties of a flexure in isolation will still be present when that flexure geometry is placed alongside locators and other flexures. The demonstrated predictive capability will therefore will allow the manufacture of flexures, including by additive technologies, whose properties are known and understood, and which exhibit the expected resonant properties with very low error.

Table 4-3. Examples of mode shapes of individual flexures mounted to a substrate, comparing simulated (FE) and experimentally validated (hammer testing) eigenfrequencies.

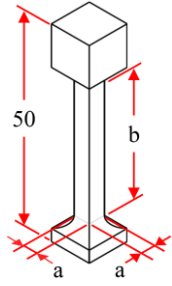
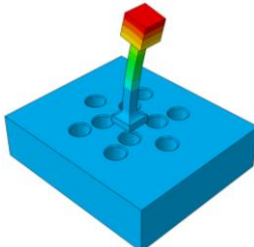
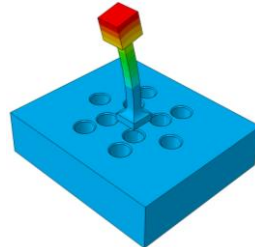
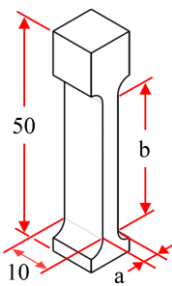
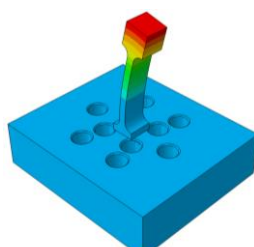
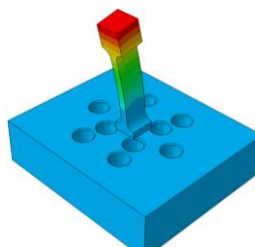
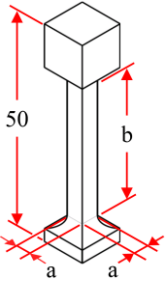
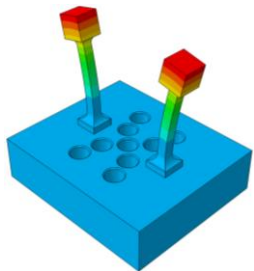
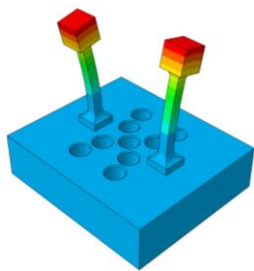
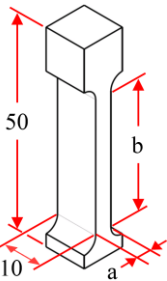
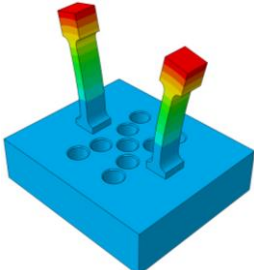
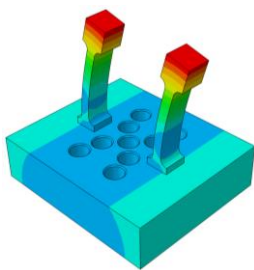
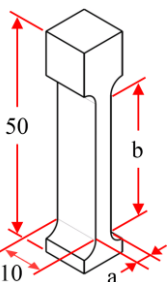
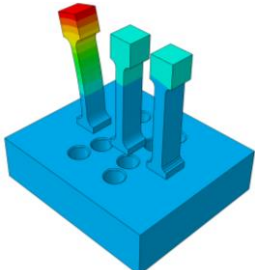
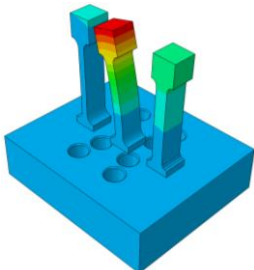
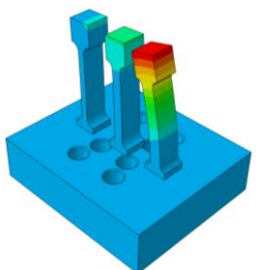
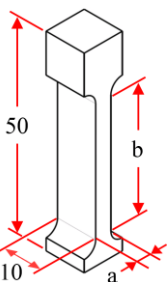
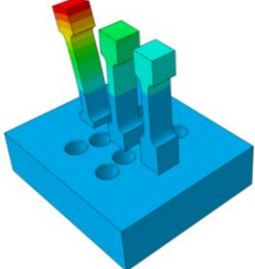
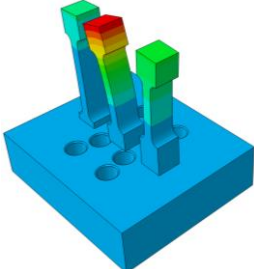
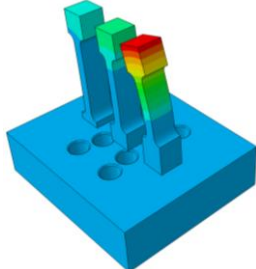
Flexure Arrangement		Natural Frequencies	
Single Square Pillar Flexure  $a = 4\text{mm}$ $b = 30\text{mm}$		 Simulated: 773Hz Validated: 739Hz Error: 4.60%	 Simulated: 783Hz Validated: 756Hz Error: 3.57%
Single I-Beam Flexure (Various Geometries) 		 Simulated: 1134Hz Validated: 1094Hz Error: 3.66%	 Simulated: 2457Hz Validated: 2359Hz Error: 4.15%
$a = 3\text{mm}$ $b = 30\text{mm}$		Simulated: 784Hz Validated: 753Hz Error: 4.12%	Simulated: 2250Hz Validated: 2149Hz Error: 4.70%
$a = 5\text{mm}$ $b = 30\text{mm}$		Simulated: 1499Hz Validated: 1429Hz Error: 4.89%	Simulated: 2614Hz Validated: 2496Hz Error: 4.72%
$a = 4\text{mm}$ $b = 25\text{mm}$		Simulated: 1362Hz Validated: 1301Hz Error: 4.69%	Simulated: 2666Hz Validated: 2559Hz Error: 4.18%
$a = 4\text{mm}$ $b = 20\text{mm}$		Simulated: 1647Hz Validated: 1583Hz Error: 4.04%	Simulated: 2867Hz Validated: 2753Hz Error: 4.14%

Table 4-4. Examples of mode shapes of multiple flexures, comparing simulated (FE) and experimentally validated (hammer testing) eigenfrequencies.

Flexure Arrangement	Natural Frequencies	
Twin Square Pillar Flexures  $a = 4\text{mm}$ $b = 30\text{mm}$	 Simulated: 735Hz Validated: 709Hz Error: 3.67%	 Simulated: 741Hz Validated: 715Hz Error: 3.64%
Twin I-Beam Flexures  $a = 4\text{mm}$ $b = 30\text{mm}$	 Simulated: 1070Hz Validated: 1036Hz Error: 3.28%	 Simulated: 1199Hz Validated: 1164Hz Error: 3.01%
I-Beam Flexures of Varying Thickness  $a =$ i) 3mm ii) 4mm iii) 5mm $b = 30\text{mm}$	 Simulated: 768Hz Validated: 741Hz Error: 3.64%	 Simulated: 1119Hz Validated: 1088Hz Error: 2.85%
	 Simulated: 1514Hz Validated: 1452Hz Error: 4.27%	
I-Beam Flexures of Varying Height  $a = 4\text{mm}$ $b =$ i) 30mm ii) 25mm iii) 20mm	 Simulated: 1108Hz Validated: 1081Hz Error: 2.50%	 Simulated: 1340Hz Validated: 1286Hz Error: 4.20%
	 Simulated: 1677Hz Validated: 1645Hz Error: 1.95%	

4.5 Fixturing applications of flexures

The successful modelling of small numbers of tuned flexures proves that the concept is scalable, and therefore the properties of an individual flexure will persist when they are used in a fixture. It is therefore possible to develop entire fixtures, containing many flexures, in order to exploit the benefits to which flexures give rise. In this section, two key features of flexure-based fixtures are presented in detail.

Figure 4-7 shows a section taken through a fixture, and illustrates the ability of fixture conformability to reduce the deformation experienced by a low stiffness workpiece with geometric variability during clamping. In this example, it can be seen that the workpiece (shown in green) deviates from the profile of the fixture due to pre-existing geometric errors, because there is a visible gap between the workpiece and fixture in its unclamped state. When clamping is applied to the rigid fixture system, the workpiece is deformed until it adopts the profile of the fixture, thus introducing significant deformation. In the flexure fixture system, the flexures grip and conform to the workpiece when clamping is applied, thus allowing the component to be fixtured with significantly lower deformation.

The situation with the rigid fixture, in which the workpiece is deformed by clamping forces, is highly undesirable, since this is likely to introduce errors into any subsequent processes for which the fixtured component is used. For inspection processes, this would lead to inaccurate measurements of the component geometry, and for machining it would lead to distortion/elongation of machined features. An investigation into this ability of flexures to reduce clamping-induced deformation is undertaken in Chapter 5.

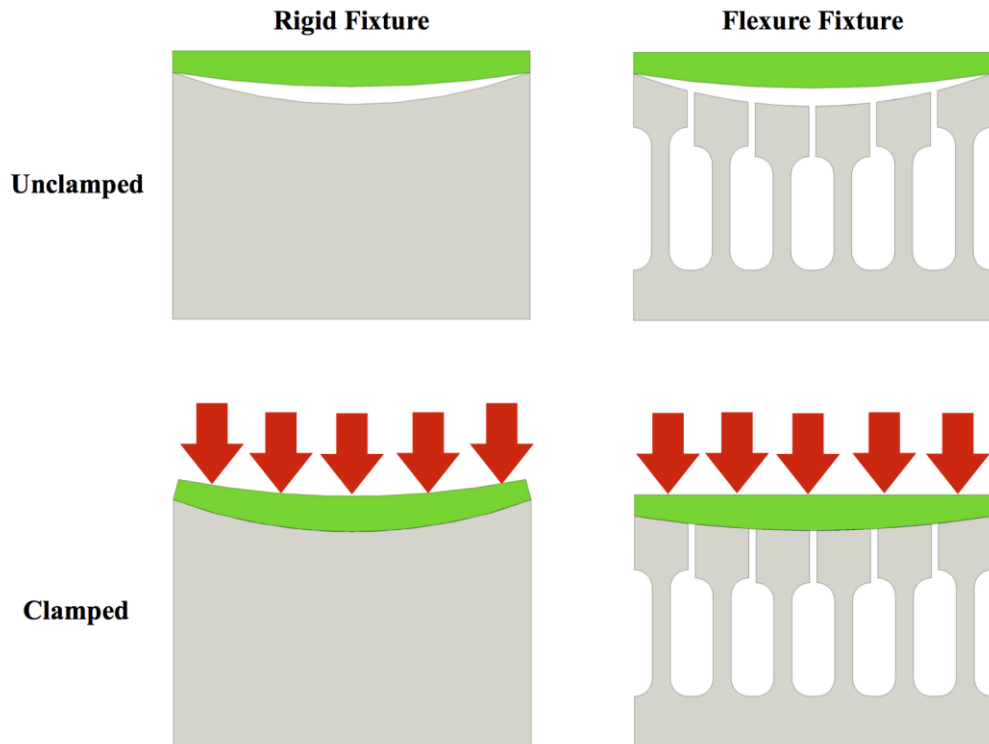


Figure 4-7. Illustration of fixture conformability, showing how flexures reduce deformation compared to a rigid fixture when clamping a workpiece with geometric errors.

A further capability of the flexure concept is demonstrated in Figure 4-8, which shows another section through a fixture (in this case in a region where locators are present). It illustrates how thin components are susceptible to vibration (shown in orange) during machining when the workpiece is unsupported between locating features. In this example, the flexures are tuned so that machining excitation causes them to resonate. Flexure resonance helps to dissipate energy present in the workpiece-fixture system, such that vibration of the workpiece is reduced, without affecting workpiece location.

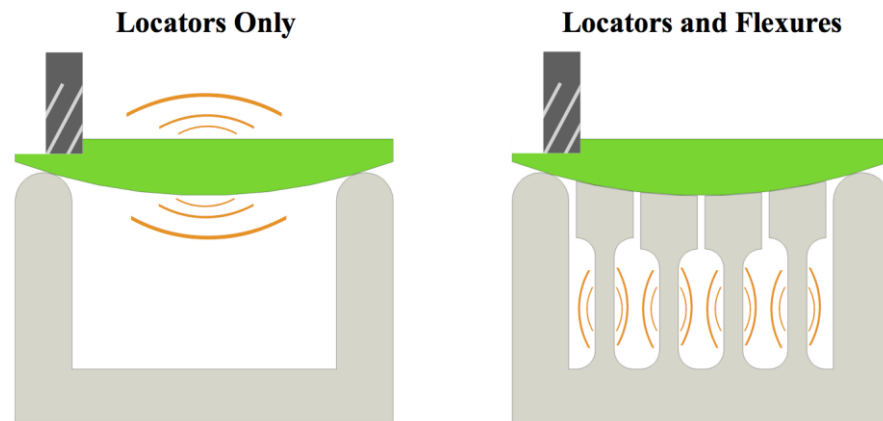


Figure 4-8. Illustration of flexure damping capability in milling, showing how flexures can resonate to help dissipate energy, thus reducing vibration of the workpiece.

Vibration of the workpiece during machining tends to cause harmful effects including poor quality of machined surfaces, and inaccuracy of machined features. Reduction of this vibration to within acceptable levels using the flexure concept therefore has the potential to improve component yields. A proof of concept with an additively manufactured machining fixture used to damp workpiece vibration is presented in Chapter 6.

Since flexures are able to offer a level of support to the workpiece that is comparable to a continuous surface, while minimising the associated potential for deformation, and also damp workpiece vibration, the novel fixture concept presents a clear opportunity to improve fixturing technology for low stiffness components.

4.6 Summary

In this chapter the novel fixture concept is presented, highlighting the features and shortfalls of existing fixture designs, and establishing the potential benefits that flexures can introduce for workpiece support and vibration damping. The need is established for additive manufacture to fully realise the concept, due to its unique ability to quickly and cheaply produce bespoke fixtures with complex workpiece supporting features.

The key features of a flexure are outlined, along with the derivation of its geometry based on the required stiffness and resonant properties. Test pegs are

used to study the dynamic response of flexures, with the significant discovery that flexure resonant behaviour is unaffected by clamping loads of the magnitudes used to fixture thin components.

A finite element model was used to predict flexure dynamic response, and shown to be highly accurate. Since the behaviour of a flexure is not significantly influenced by its surroundings, individual flexures can therefore be scaled up into large arrays of flexures in a fixture, whose behaviour can be predicted with a high degree of confidence.

Following the categorisation of flexure behaviour, the novel capabilities of the fixture concept are explained in detail, including flexure conformability to reduce clamping deformation of components with errors, and frequency tuning of flexures to damp machining vibrations of the workpiece.

The combination of these developments will therefore permit the creation of novel fixtures that offer considerable improvement upon existing technologies for the manufacture of low stiffness components.

5 Flexure conformability for improved workpiece holding

5.1 Introduction

Presented in this chapter is an investigation into the first key capability of the novel concept: the ability of fixture systems incorporating flexures to improve workpiece holding. This is achieved through conformability, which is the ability of flexures to deform in order to better accommodate the workpiece geometry, thus reducing clamping-induced workpiece deformation, without affecting location. FE models are developed to determine the optimal technique for preserving dynamic properties and response under buckling conditions of flexures so that they can vary in geometry to accommodate a freeform workpiece with minimal shifting of their properties. Experiments were devised to compare performance between flexure and conventional rigid fixtures in the holding of components with geometric errors of varying severity, in order to study the effect of fixture conformability on the clamping forces required to hold a workpiece, and the deformation it experiences during clamping.

5.2 Preservation of flexure properties for fixturing freeform workpiece surfaces

In order to support the surfaces of a component with curved surfaces, such as the example in Figure 4-1c, the fixture must follow the contours of the workpiece such that they share conjugate contact surface geometries to create full-face contact. To allow this, each individual flexure must have a unique geometry in order to support the region of the workpiece with which it makes contact. If the flexures originate from a substrate that is a flat plane, such as the example in Figure 5-1a, then each flexure will differ in height, therefore giving rise to significant variation in frequency and buckling response between flexures. Variation of properties between flexures can be addressed, however if the substrate also follows the profile of the workpiece, shown in Figure 5-1b, such that all flexures are of uniform length.

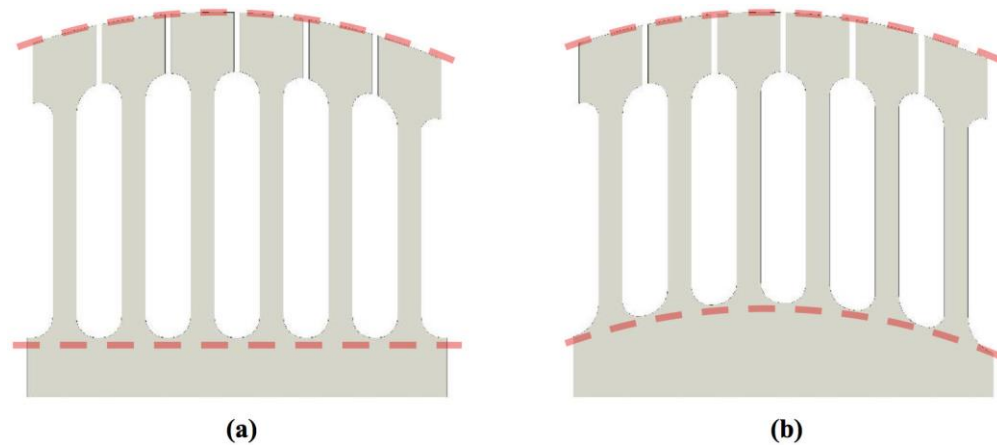


Figure 5-1. Single row of flexures in a section of a conformable fixture, showing examples where the flexures vary in height, giving rise to separate eigenfrequencies for each flexure (a); and where flexures are of uniform height, and therefore exhibit uniform eigenfrequency (b).

Although fixtures with dissimilar flexures are simple to design, since there is no need to tailor the geometry of each individual flexure, this would not allow full realisation of the fixture concept. The beneficial characteristics of flexures can only be exploited to maximum effect if they all have identical properties, due to the combined influence of multiple similar structures acting simultaneously on the workpiece. Therefore it was decided that only the fixture type with flexures of constant eigenfrequency (Figure 5-1b) would be pursued in this research.

Finite element analysis was used to determine the optimal design in order to impart uniform properties to all flexures. Adapting the model used to simulate flexure response in Chapter 4, a row of six flexures was modelled, with the geometry imported from CAD as a STEP file. Since the flexure prototypes would be manufactured in ABS, the model included the material definition presented in Table 5-1, and a *linear perturbation, buckle* analysis step was added to enable prediction of buckling response as well as frequency.

Table 5-1. ABS material properties used to model flexure frequency and buckle response.

ABS	
Density (kg/m³)	995
Young's Modulus (GPa)	2.4
Poisson's Ratio	0.35

When a conformable fixture follows the contours of a freeform component, each flexure must by definition have a unique geometry in order to support unique regions of the workpiece, however it is nonetheless possible to tune these unique structures so that they all exhibit near-identical behaviour. When flexures originate from a curved substrate that follows a workpiece, and are all of uniform external dimensions and cross section, there are two methodologies for creating the geometry in order to preserve dynamic properties between flexures, illustrated in Figure 5-2. Both approaches concern how the deformable region of the flexure is blended to the substrate and workpiece contact regions. The first method (Figure 5-2a) involves using identical blending radii at each interface, which results in small variations of the height of the deformable region, while the second (Figure 5-2b) involves using a constant deformable height, varying the blending radius in order to give a horizontal interface between the different regions of the flexure.

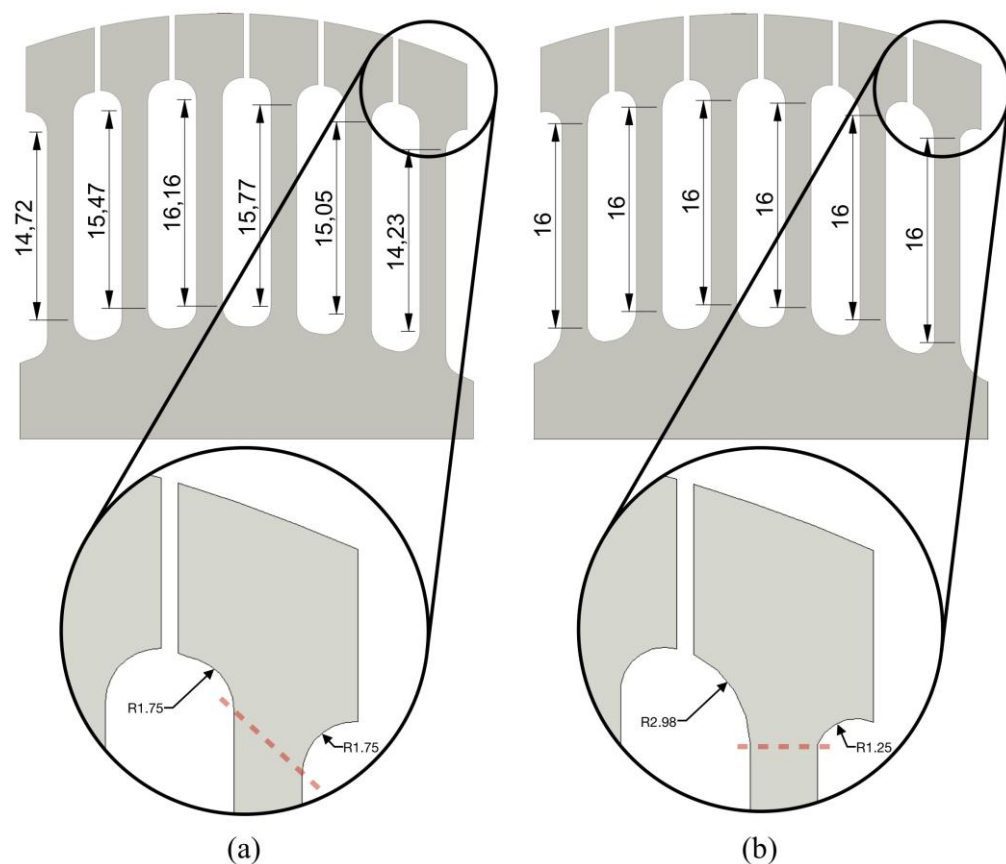


Figure 5-2. Competing approaches to fixture design for preservation of uniform frequency response across multiple conformable flexures: uniform blending radii (a), in which the height of each flexure deformable region varies; and uniform deformable height (b), which necessitates dissimilar radii. The interface between the deformable and workpiece contact regions is indicated with a red line.

Both methods were shown by the models to be capable of imparting near-identical frequency response to each flexure, creating similar mass and damping properties for each flexure. However, the simulations showed that the first approach, in which the deformable ‘column’ of each flexure differed in height, resulted in greater variability of buckling strength across the different flexures. Based on these findings, the method of uniform deformable length was identified as the superior method, since it permits dissimilar flexures to have near-identical frequency and buckle response. This is illustrated in Figure 5-3, along with a simple fixture in which the flexure substrate does not follow the component, to demonstrate the extent to which it is possible to preserve frequency and buckle properties when conforming to a freeform workpiece.

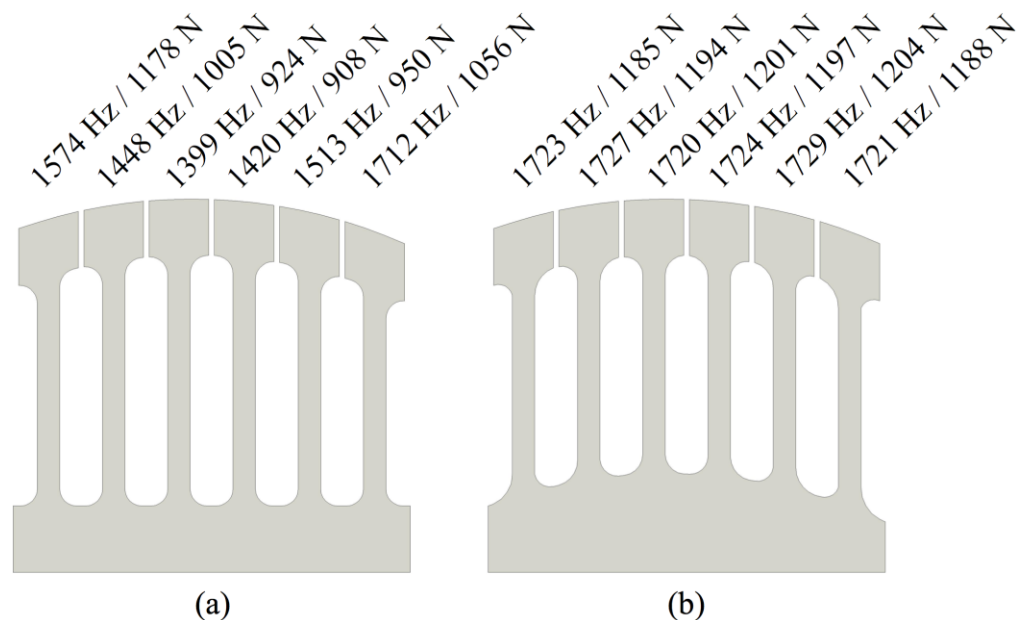


Figure 5-3. First eigenfrequency and buckling load of flexures from FE models, comparing dissimilar fixtures from a flat substrate (a), to those which have been optimised to give identical frequency and buckle response (b).

The simulated data shows the optimised technique for generating flexure geometry (Figure 5-3b) to be highly effective in preserving their properties, with very little variation in frequency and buckle response between neighbouring flexures. For the optimised flexure geometry system, standard deviation was just 0.2% of the mean for the first eigenfrequency, and 0.6% for its buckling strength. This is in sharp contrast to the unoptimised system with dissimilar flexures (Figure 5-3a), for which the standard deviation of the mean was 7.8%

for frequency, and 10.1% for buckling. This example highlights the need to ensure that flexure geometry is consistent in order to elicit any collective beneficial effect from the tuning of multiple flexures, since failure to account for curved workpiece surfaces can result in variation in critical eigenfrequencies by hundreds of Hz between flexures.

5.3 Conformable fixture proof of concept

Conformability describes the ability of a fixture to accommodate variations in workpiece geometry, thus minimising the deformation experienced by a component with deviations from its nominal geometry when it is clamped. The low stiffness, deformable nature of flexures means they can be used to conform to component surfaces, achieving holding at low clamping forces and without introducing significant deformation, while a conventional rigid fixture has the potential to simply deform any component until it adopts the shape of a fixture. Conformability has significant implications for fixturing components with large geometric variability, since the ability to hold a workpiece with dimensional errors without introducing significant deformation will improve component accuracy and yield rates.

The beneficial effects of fixture conformability were investigated by comparing a conformable flexure-based fixture against a conventional fixture with rigid full-face supporting surfaces. A simple fixture was designed for a representative aerospace component to evaluate the effects of fixture conformability on workpiece deformation during clamping. A simplified quasi-freeform blade-like component, designed to fit in the fixture in Figure 5-3b, was produced in aluminium, alongside two pairs of fixture blocks in ABS. The components, shown in Figure 5-4, had a section thickness of 6mm, and were designed to recreate a thin section through a fixture and workpiece, in order to enable analysis of clamping behaviour. The flexure variant comprised six flexures, representing a single row out of the entire array of flexures that would be present in a complete fixture.

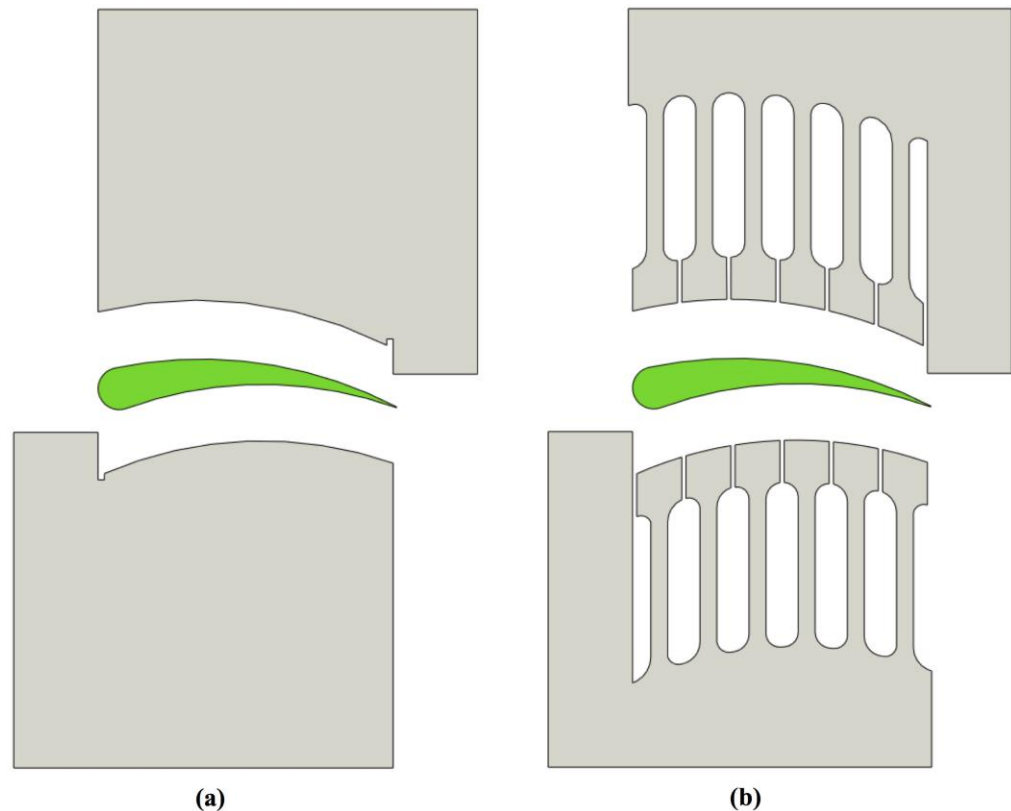


Figure 5-4. Rigid (a) and flexure (b) fixture systems, with the blade-like component shown in green.

The first set of fixture blocks (Figure 5-4a) utilised the row of six flexures studied in the previous section, with the addition of an end locating wall to provide deterministic workpiece location. A second block with a further six flexures comprised the other half of the fixture, allowing the blade to be held by applying a clamping force to the fixture assembly. The other pair of fixture blocks (Figure 5-4b) featured the same external dimensions and locating features, but the region containing flexblocks were manufactured by cutting their profiles from 6mm plate in their respective materials using the microwaterjet F4 machine (specifications given in Table 3-4).

The geometry of the blade-like component, shown in Figure 5-5a, consists entirely of curved surfaces. The internal diameter of the lower contact surface of the blade defines its chord height or sagitta, S . The nominal value for the blade sagitta, S_0 , was specified as 3mm. In addition to this nominal blade geometry ($S = S_0$), eight variants of the blade geometry were designed with errors deliberately introduced into their geometries (four blades where $S > S_0$,

and four with $S < S_0$), to replicate components with increasing severity of geometric variation from the nominal component profile. The sagitta error, ε_s , ranged from -1mm (flattened blade) to 1mm (arched blade), and was designed so that four of the blades would have relatively minor errors in their geometries, while the other four would have significant distortion. The sagitta of each component is defined by $S = S_0 + \varepsilon_s$, with $\varepsilon_s = 0$ indicating a nominally sized (error-free) component. The geometry of the distorted blades is illustrated in Figure 5-5b).

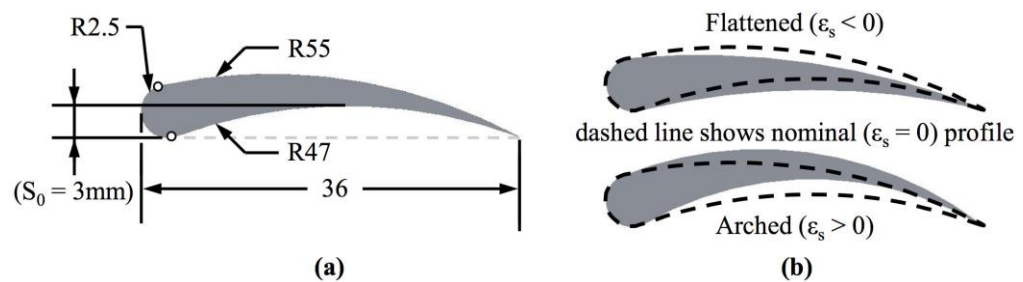


Figure 5-5. Nominal component geometry highlighting sagitta of the lower contact surface (a), and flattened/arched variants to represent distorted component variants (b).

All nine blade workpieces (the nominally sized component, along with each of the four flattened and four arched variants with distortions) and the fixture blocks were inspected using a Mitutoyo Apex Euro-C-A12 BRT-1210 coordinate measurement machine (CMM), shown in Figure 5-6, to determine the accuracy of the components.

A 1mm diameter touch probe was selected, and used to take discrete measurements at increments of 0.1mm around the entire perimeter of the blade. The true sagitta value for each blade was determined using the inbuilt Mitutoyo software on the CMM computer, with sagitta values determined $\pm 0.06\text{mm}$ (95% confidence). Line form tolerances for the upper and lower contact surfaces were also calculated using Mitutoyo software, in accordance with ISO 1101:2017 [119], following the procedure for a line profile specification related to a datum system. Component data for each blade variant is presented in Table 5-2.

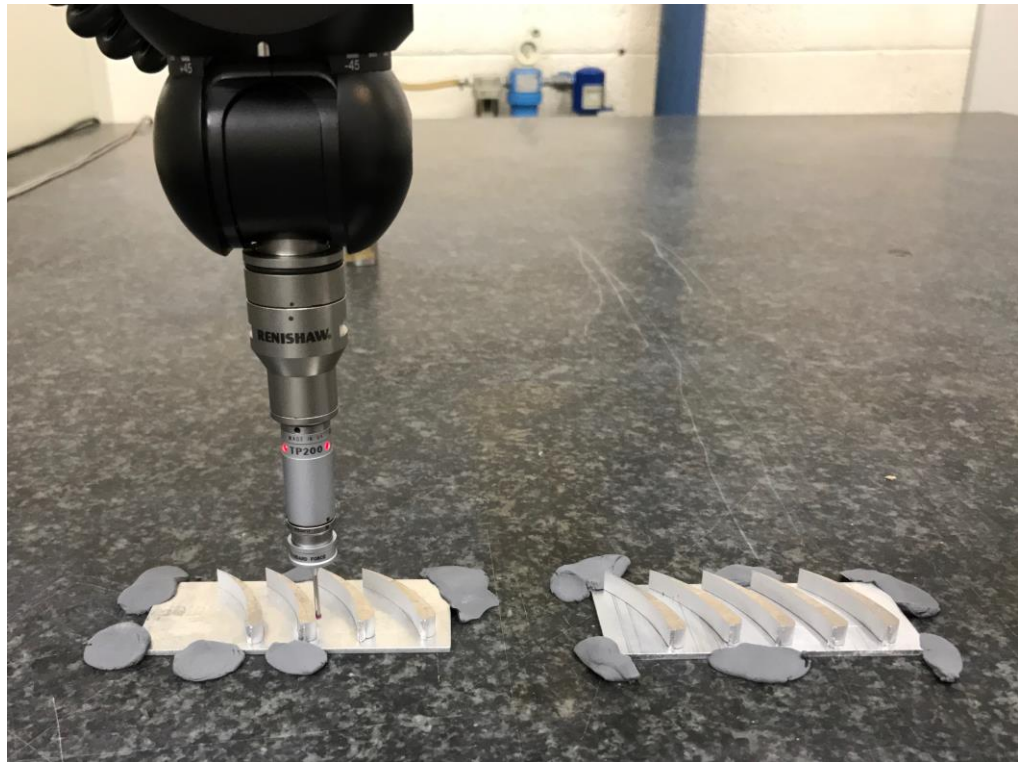


Figure 5-6. Measurement of blade geometries with the CMM.

For each blade, the sagitta was measured to be slightly smaller than the expected value, on average by 0.086mm. However, this error was also consistent across the fixture blocks, whose contact surface profiles were found to be an accurate inverse for the nominally sized ($\epsilon_s = 0$) blade. The error was therefore not deemed to be significant, since the principle requirement of this system is that the nominal sized blade fits in the fixtures without appreciable deformation, meaning that any deformation experienced by the distorted blades could thus only be attributed to their deviation from the nominal blade profile. The data also shows that the upper and lower contact surfaces of all blade variants had a line form tolerance of 0.2mm or better from their respective measured diameters, indicating high accuracy of surfaces cut by microwaterjet.

Table 5-2. Comparison of nominal blade sagitta to actual values measured using the CMM, with form tolerances of the upper and lower contact surfaces.

Design Sagitta (mm)	Actual Sagitta (mm)	Form Tolerance (mm) of Contact Surfaces
2.000 ($\epsilon_s = -1\text{mm}$)	1.92	Upper: 0.07 Lower: 0.07
2.500 ($\epsilon_s = -0.5\text{mm}$)	2.40	Upper: 0.10 Lower: 0.18
2.750 ($\epsilon_s = -0.25\text{mm}$)	2.66	Upper: 0.08 Lower: 0.14
2.875 ($\epsilon_s = -0.125\text{mm}$)	2.77	Upper: 0.13 Lower: 0.15
3.000 ($\epsilon_s = 0\text{mm}$)	2.89	Upper: 0.05 Lower: 0.06
3.125 ($\epsilon_s = 0.125\text{mm}$)	3.04	Upper: 0.18 Lower: 0.10
3.250 ($\epsilon_s = 0.25\text{mm}$)	3.19	Upper: 0.09 Lower: 0.08
3.500 ($\epsilon_s = 0.5\text{mm}$)	3.42	Upper: 0.11 Lower: 0.16
4.000 ($\epsilon_s = 1\text{mm}$)	3.94	Upper: 0.10 Lower: 0.20

A potential source of geometric inaccuracy in waterjet components, which was identified as a possible cause for the discrepancy between specified and manufactured geometries, is kerf taper. This is where width of the cut varies depending on the depth of jet penetration into the workpiece, illustrated in Figure 5-7. This means the profile cut by the waterjet becomes wider as it travels further from the nozzle, due to divergence of the jet, potentially giving rise to non-perpendicular cut surfaces, which would otherwise be vertical under ideal

cutting conditions. A sample of cut surfaces were inspected using the CMM, and found to have a modest reverse kerf taper, with taper angles of up to 0.4° . Over the 6mm thickness of the blade components, this means the bottom of the cut is up to 0.084mm wider than at the top, which corresponds to a maximum possible error due to perpendicularity of 0.042mm, as is shown in Figure 5-7b. Perpendicularity was therefore deemed not to be a significant source of error, due to the small impact of non-parallel surfaces over the relatively thin section thickness of these blade geometries.

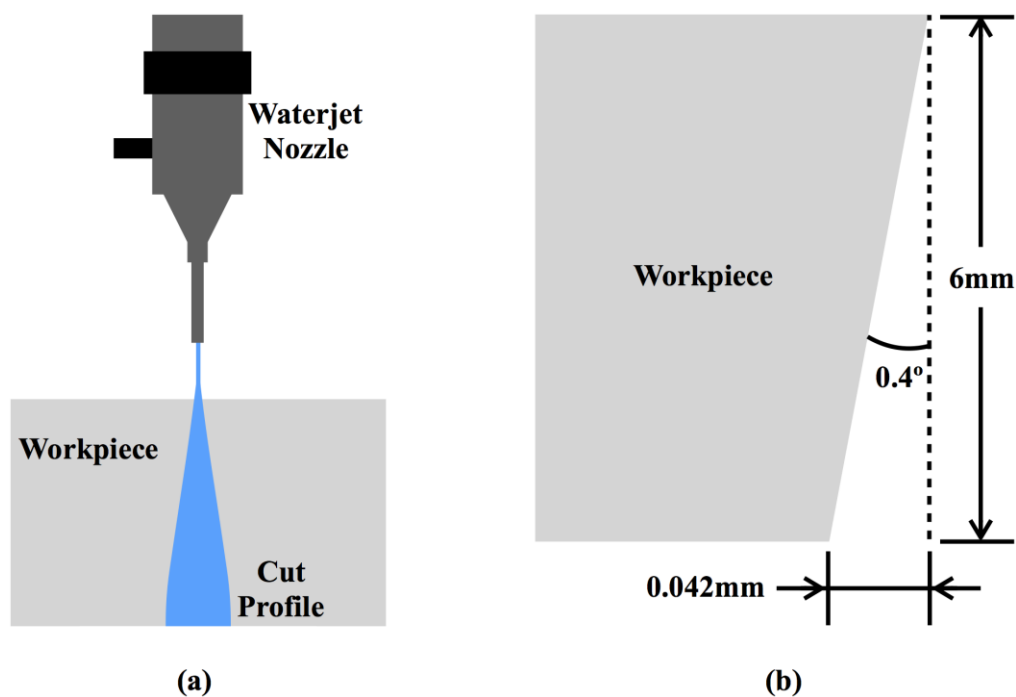


Figure 5-7. Schematic of waterjet cutting, showing tapered kerf due to divergence of the jet (a), and the effect of a 0.4° reverse taper on the blade workpiece geometry (b).

From the CMM data, it was therefore determined that the cutting accuracy of the microwaterjet machine was high. An excellent fit between the nominal component and the fixture blocks, and good form tolerance across all cut profiles is attributed to the use of pre-programmed optimal cutting parameters with the microwaterjet machine controller, as well as the relatively low hardness of the two materials, and the small section thickness chosen for the components.

In order to verify accuracy of the models used to predict flexure properties in the previous section, the flexure blocks were subjected to hammer modal testing. The blocks were suspended free-free using elastic, and subjected to drive point impacts with the accelerometer attached to the substrate region. For both blocks, the frequency response functions showed a prominent peak in inertance at a single frequency for the first eigenfrequency of the system, coinciding with simultaneous overlapping resonance of all six flexures. As is shown in Figure 5-8, the peaks corresponding to flexure resonance were centred around 1674Hz and 1680Hz respectively for the top and bottom flexure blocks, compared to the mean simulated frequency of 1723Hz; an error of below 3%, thus indicating very good predictive capability of the models used to design the flexures.

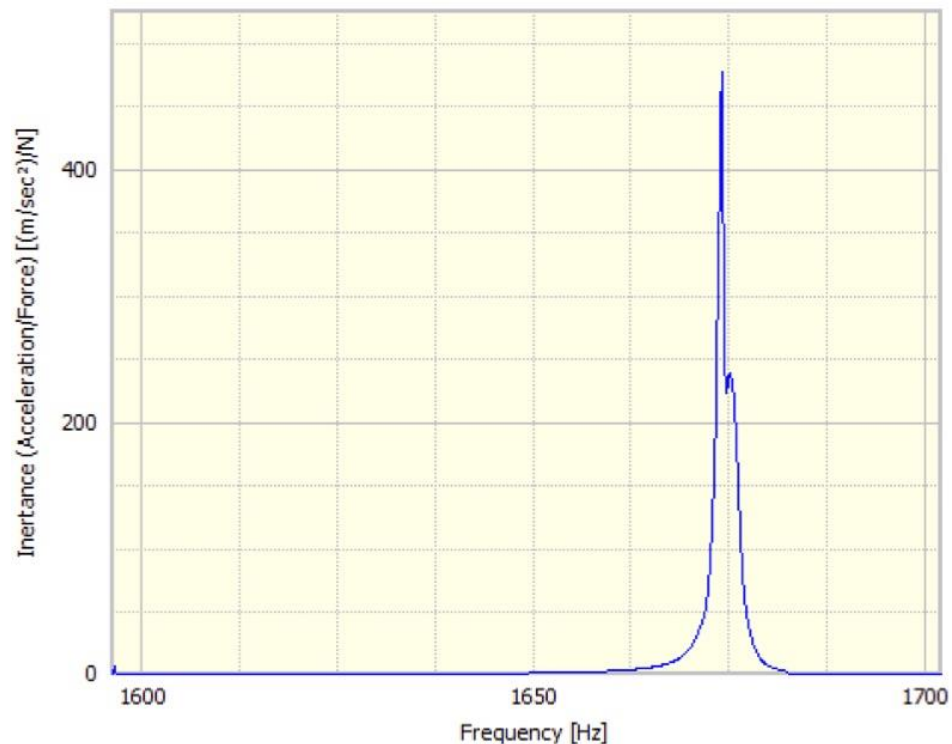


Figure 5-8. Frequency response function for the upper flexure block, showing simultaneous resonance of all six flexures centred about a central peak at 1674Hz.

5.4 Investigations into clamping force and deformation for workpiece geometries with geometric errors

Following design and inspection of the fixture blocks and components, a series of experiments were devised to investigate the effect of fixture conformability on the workpieces with deviations from the nominal blade geometry. Since the

fixture blocks were designed for the nominally sized blade component, the distorted blades would therefore be subjected to deformation during clamping in order for them to be held securely in the fixtures. These distorted blade geometries were used to compare the relative behaviour of the two fixture systems, and investigate the ability of flexures to accommodate components with errors of increasing severity.

A test rig was assembled from modular fixturing apparatus, shown in Figure 5-9a, which allowed compression of the workpiece-fixture assemblies, and measurement of clamping force. The experimental setup comprised a flat surface onto which the component and fixture blocks were placed, with clamping applied by advancing a precision moving stage to compress the assembly against a rigid stop, so that it acted like a vice. In-line with the assembly was a load cell connected to a data acquisition system, to measure the clamping force applied to the system. Since the upper and lower workpiece contact surfaces of the fixture blocks are designed to match the nominally sized component, gaps are visible between the distorted blades and the fixture in its unclamped state, as shown in Figure 5-9b.

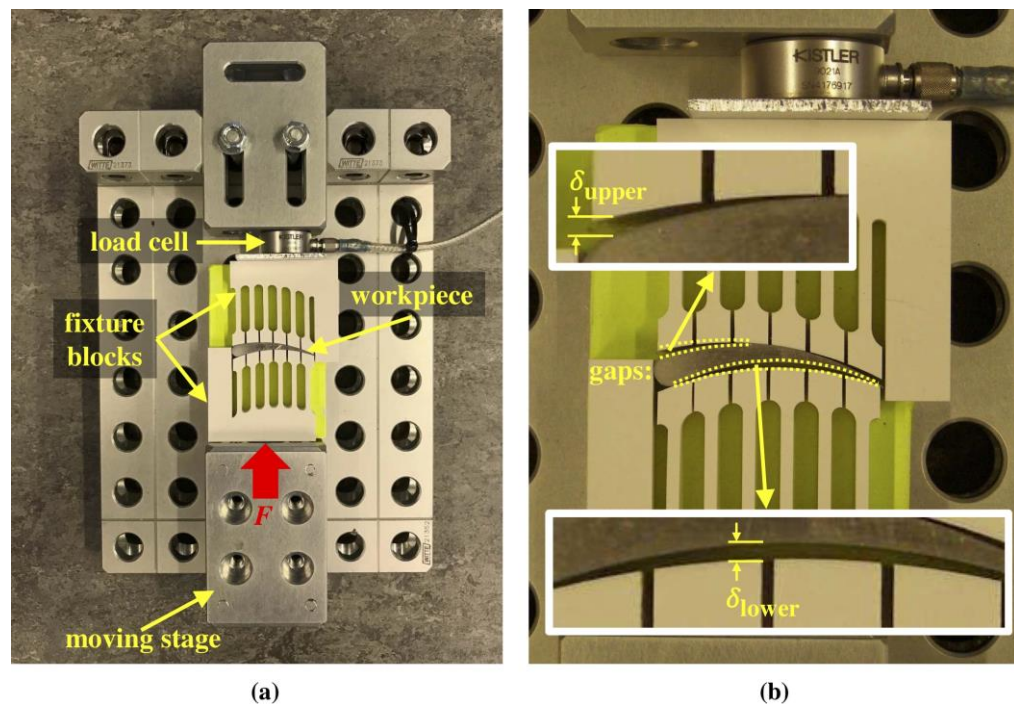


Figure 5-9. Overview of test rig for clamping blade-like components in fixture blocks (a), with a close-up of a distorted component prior to clamping, exhibiting visible gaps (δ_{upper} and δ_{lower}) between the contact surfaces of the component and the fixture (b).

This configuration was used to measure the level of clamping force required to securely hold each workpiece in both of the fixtures, and also to investigate the level of deformation experienced by the systems during clamping.

5.4.1 Clamping force experiment

The first experiment to compare the rigid and conformable fixture systems involved clamping all of the blade variants until they were securely held by each fixture, and measuring the force required to achieve holding.

The testing procedure was consistent for each combination of blade and fixture. The fixture blocks and workpiece were placed in the test rig, initially assembled so that the workpiece was in contact with the fixture, but with zero clamping force applied. Clamping was gradually increased by advancing the moving stage at a rate of approximately 4mm per minute, with continuous force data acquired during clamping. Full holding of the workpiece was defined as the point when it was in firm contact with the fixture across its entire width on its top and bottom surfaces, meaning any gaps, such as those visible in Figure 5-9b, had been closed (i.e. $\delta_{\text{upper}} = \delta_{\text{lower}} = 0$). The test was repeated three times for each test configuration to ensure accuracy and repeatability of the data.

The results show a consistent and significant reduction in the clamping required to hold components with distortions from the nominal geometry in the flexure fixture system. Conformability is shown to reduce the clamping force required to achieve full holding of components by up to 60% compared to the conventional rigid fixture system, with an average reduction of 49% across all configurations. The data from the clamping force study is given in Figure 5-10.

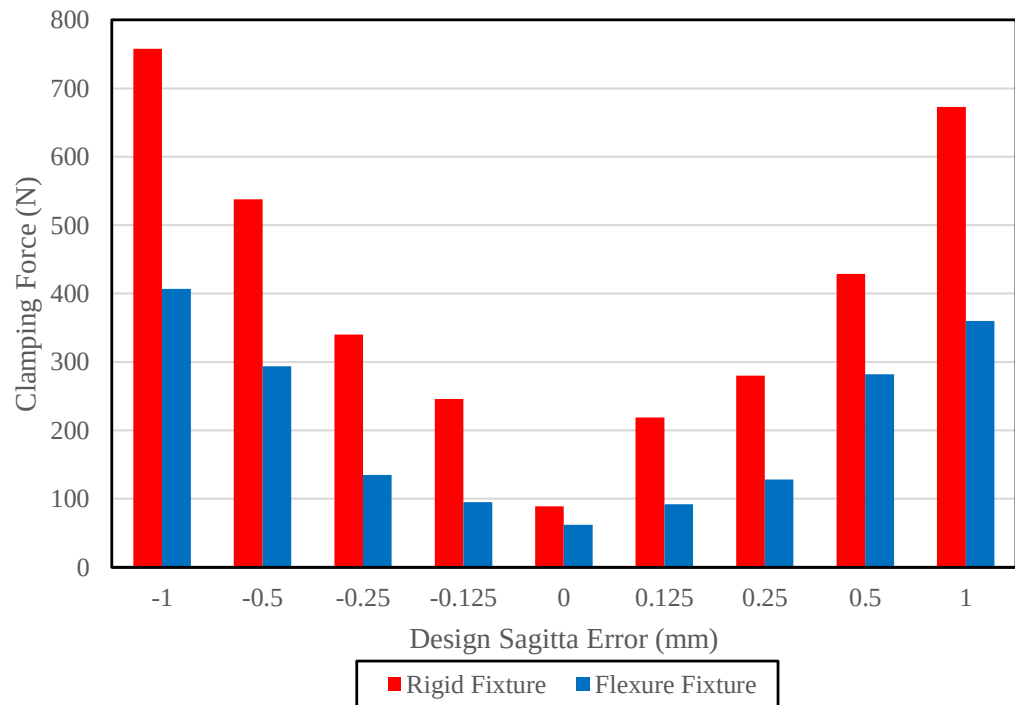


Figure 5-10. Clamping force required to achieve full holding against initial distortion in the workpiece, comparing the rigid and flexure fixture systems (average from three repeated measurements yielding 4% average repeatability)

In addition to reducing clamp force, flexures were found to be capable of holding all nine blade geometries without causing damage. In the conventional fixture system, three of the blades with the greatest deviations from the nominal geometry were plastically deformed at their trailing edges, with the blade ‘crimped’ approximately 1mm from its end (shown in Figure 5-11), due to concentrated forces in this region from the rigid clamping surface.



Figure 5-11. Blade exhibiting slight crimping on its trailing edge.

The ability of the flexure fixture system to achieve secure workpiece holding at considerably reduced clamping loads is a considerable advantage for delicate, easily damaged components. Reduced clamp forces mean that stress in the component is lower, and it is therefore less likely to be deformed during holding. While a reduction in clamping forces shows promise for conformable flexures, the use of large clamping forces alone is not intrinsically problematic, and only presents an issue when it leads to excessive workpiece deformation. It was therefore determined that a further investigation into workpiece deformation during clamping was necessary, in order to fully understand the capability of the fixture concept.

5.4.2 Clamping deformation experiment

Workpiece deformation within a fixture can be highly deleterious. Even when a component is not plastically deformed, it is desirable to minimise elastic deformations, since any feature machined on a deformed workpiece will have errors in its geometry and/or placement once the component is released from the fixture and is able to spring back to its undeformed shape. Any material removal from a workpiece while it is in a deformed state also has the potential to weaken the finished component through the introduction of residual stresses. In order to investigate the ability of the fixture concept to reduce deformation through conformability, the deformation of the components was studied at various fixed clamping loads, F , of 100N, 200N, and 500N. This permits direct comparisons of the deformation experienced by the workpiece in the two fixture systems for a given clamping force.

The test rig used to investigate clamping forces was created in a FE model, shown in Figure 5-12, to enable analysis of the deformation experienced by the workpiece and fixture in response to loading. Due to all components having constant section thickness, the parts were imported from CAD as flat surfaces, with the model created as a 2D system with constant thickness of 6mm using the plane stress assumption, in order to reduce computation time compared to a 3D model. Material definitions for the ABS and aluminium parts is given in Table 5-3. The entire assembly was meshed using quadrilateral CPS4R (4-node bilinear plane stress quadrilateral) elements with reduced integration. Boundary

conditions replicated the test rig, with the assembly constrained at one end and loaded from the other. Since the model involved the interaction of multiple parts, contact relationships were defined for the workpiece and fixture surfaces, with penalty friction interactions specified using the standard isotropic Coulomb friction model in Abaqus. Loads were defined to simulate clamping, and a static analysis (*static, general*) was carried out for each configuration. A mesh sensitivity study of the blade components revealed that a nominal element size of 0.125mm (corresponding to approximately 24 elements through the blade at its midpoint) yielded convergence of blade deformation to the nearest 1µm with both fixture types, thus providing optimal accuracy for the minimum computation time. The flexure blocks were meshed in accordance with the convergence study undertaken in Chapter 4, with the conventional fixture blocks seeded with a proportionate mesh density.

Table 5-3. Material properties used to model compression of the workpiece-fixture assembly.

	2014 Aluminium	ABS
Density (kg/m³)	2780	995
Young's Modulus (GPa)	74	2.4
Poisson's Ratio	0.33	0.35
Yield Stress (MPa)	108	31

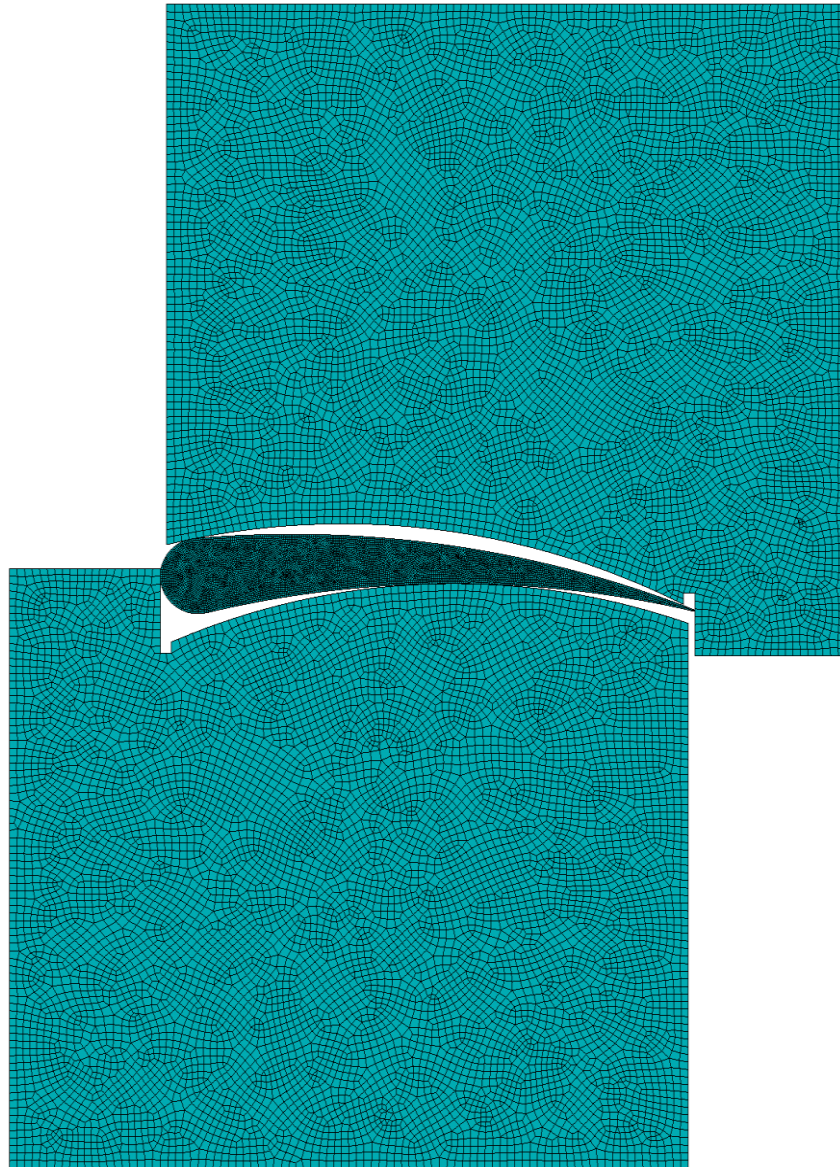


Figure 5-12. FE model used to analyse workpiece deformation during clamping, showing the most flattened ($\epsilon_s = -1\text{mm}$) blade in the rigid fixture before loading.

The response of blade and fixture combination was simulated, providing deformation data for each configuration at the three clamping loads relative to the unclamped state. A methodology for defining workpiece deformation was devised, and is illustrated in Figure 5-13. On each blade, two points were defined: P1, which is half way through the thickness of the blade at the midpoint of the lower contact surface; and P2, which is the centre point of the circle that defines the leading edge of the blade. The relative positions of P1 and P2 before and after clamping were used to describe workpiece deformation, by measuring the distance between the points in the direction of clamping, h . Clamping-

induced deformation was quantified as the difference between the initial (h_1) and clamped (h_2) distances, with a difference of zero ($h_1 = h_2$) indicating no relative movement between the two regions of the blade, i.e. deformation-free clamping.

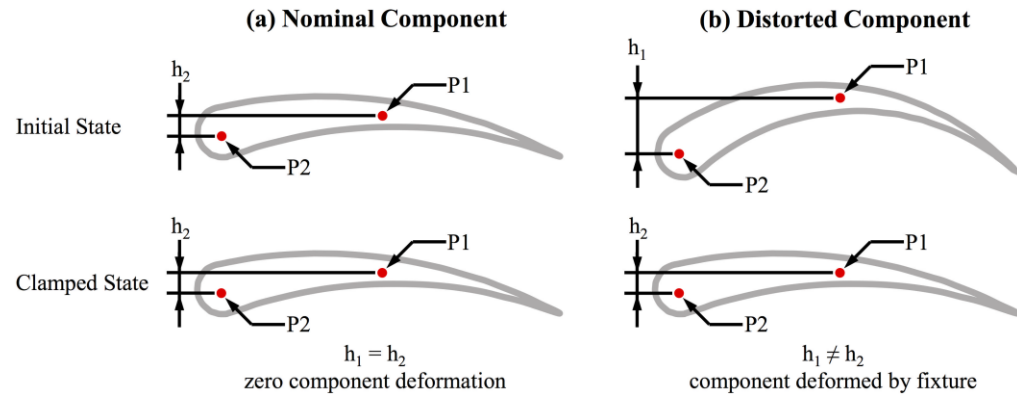


Figure 5-13. Methodology for quantifying component deformation, illustrating the ideal case when the component matches the nominal profile of the fixture, and is not deformed during clamping (a); and the case when the component is distorted, and therefore is deformed until it adopts the shape of the fixture (b).

The simulations predict that workpiece deformation is lower using the flexure fixture with all blades at every level of clamping. Across all configurations, deformation (as defined by $h_1 - h_2$, the change in distance between P1 and P2) is 22% lower with the flexure system. In the configuration with the most severely distorted flattened blade (shown in Figure 5-12) at 500N clamping, the model predicts that flexures reduce workpiece deformation by 40% compared to the rigid block.

In order to experimentally validate these models, the testing with fixed clamping loads was undertaken using the test rig with the fixture and blade parts. The Dantec Q-400 digital image correlation (DIC) system presented in Chapter 3 was used to capture displacement of the system during clamping. Facet size was set to 13 pixels, with a grid spacing of 8 pixels, to produce a strain map of the entire workpiece-fixture assembly at each stage of loading. The full field displacement data produced by DIC allows application of the same methodology used to quantify workpiece deformation in FE models, thus

enabling direct comparison of simulated and experimental results. The setup for DIC is shown in [Figure 5-14](#).

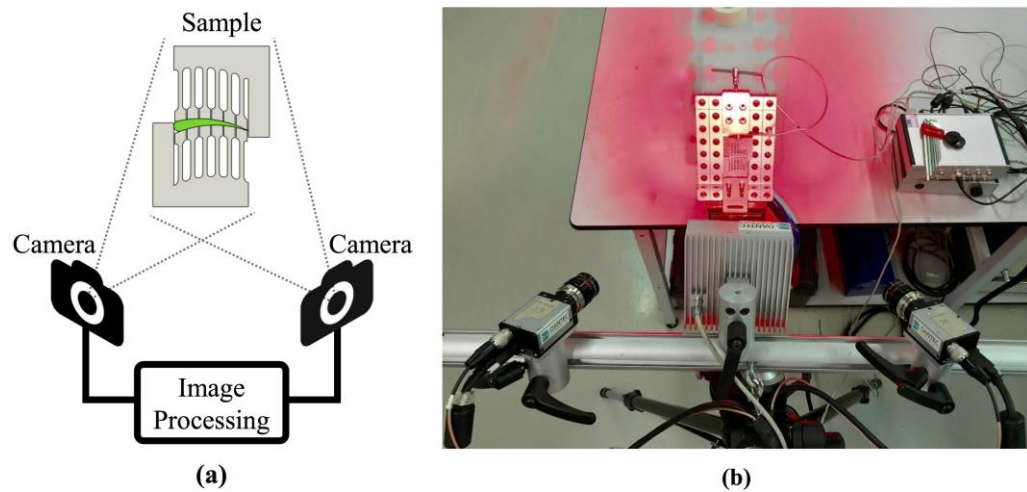


Figure 5-14. Schematic (a) and photograph (b) of the digital image correlation system configuration for analysing component deformation during clamping.

Correlations of images taken of the workpiece-fixture assemblies, comparing the initial unloaded state to the three levels of clamping, reveal that the workpieces remained adequately located throughout loading of the fixture. In all cases, the blades did not significantly shift perpendicular to the direction of clamping, with a maximum lateral displacement measured to be $60\mu\text{m}$. Out-of-plane displacement (lifting of the blade off the test rig surface) was negligible, with maximum displacement observed as $18\mu\text{m}$. In both cases this level of shifting of the blade did not adversely affect clamping performance of the system. In addition, undesired blade movements are likely to be even lower in a complete fixture with a considerably deeper section thickness, compared to the thin section through a fixture presented in this proof of concept. Although the out-of-plane displacement was experimentally shown to be small, it was not considered by the simulations, which were approximated in 2D, and therefore this has the potential to introduce a minor source of error to the models.

With the nominally-sized (distortion-free) blade geometry, predicted and observed blade deformation was very small in both fixture systems. The models, which assumed a perfect error-free fit between the fixture and the workpiece, predicted displacements at the highest clamping load of $3\mu\text{m}$ in the flexure

fixture, and $4\mu\text{m}$ in the rigid fixture. DIC revealed actual deformation to be only slightly higher, at $10\mu\text{m}$ and $13\mu\text{m}$ respectively. This consistent with the measurements obtained from the CMM, which showed mismatch between the profiles of the fixture blocks and the nominal blade to be minimal. Low deformation of the nominally sized blade in the fixtures therefore indicates high quality of the experimental setup, meaning there is very little error introduced during clamping. Subsequently, any deformation of components with geometric errors can be attributed solely to their deviation from the nominal geometry, and not any other sources of error present in the system.

For the blades with pre-existing errors, the experimental data obtained from DIC generally follows the predictions from FE models. The flexure system consistently outperformed the rigid fixture for all blades and clamp forces, producing an average reduction in workpiece deformation of 23% across all configurations, with the advantages of conformability becoming more pronounced as clamping was increased. The data is presented in Figure 5-15 to Figure 5-18 for the varying degrees of blade distortion. The deformation data from DIC, which is averaged from two repeated measurements, is marked on the plots for each load (with error bars to show the standard deviation for each configuration), and the data from FE is represented by lines, since the models predict that workpiece deformation is linear over the clamping force range studied.

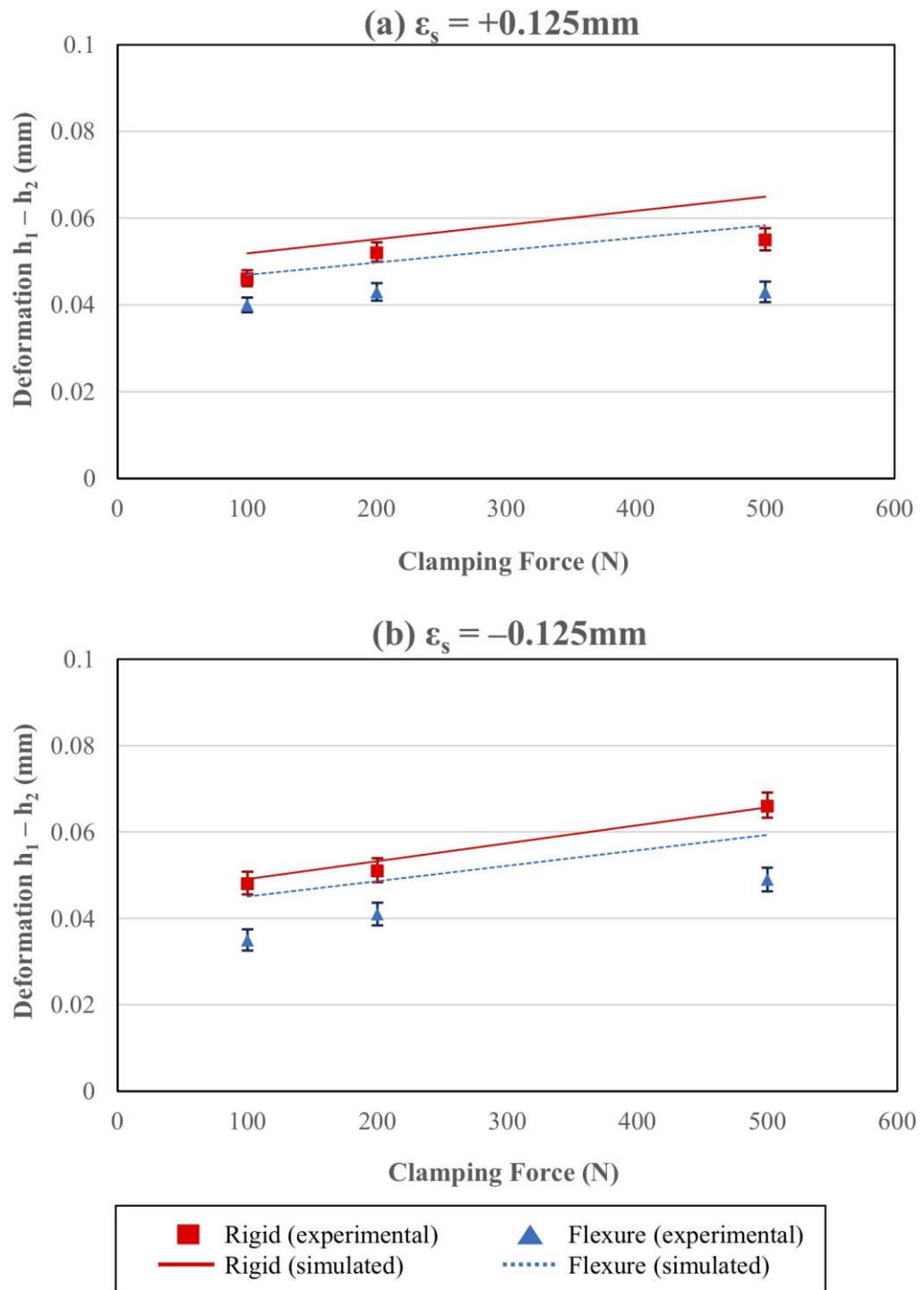


Figure 5-15. Plots of workpiece deformation against clamping force for the $\varepsilon_s = \pm 0.125\text{mm}$ blades, comparing simulated and experimental values for each fixture system.

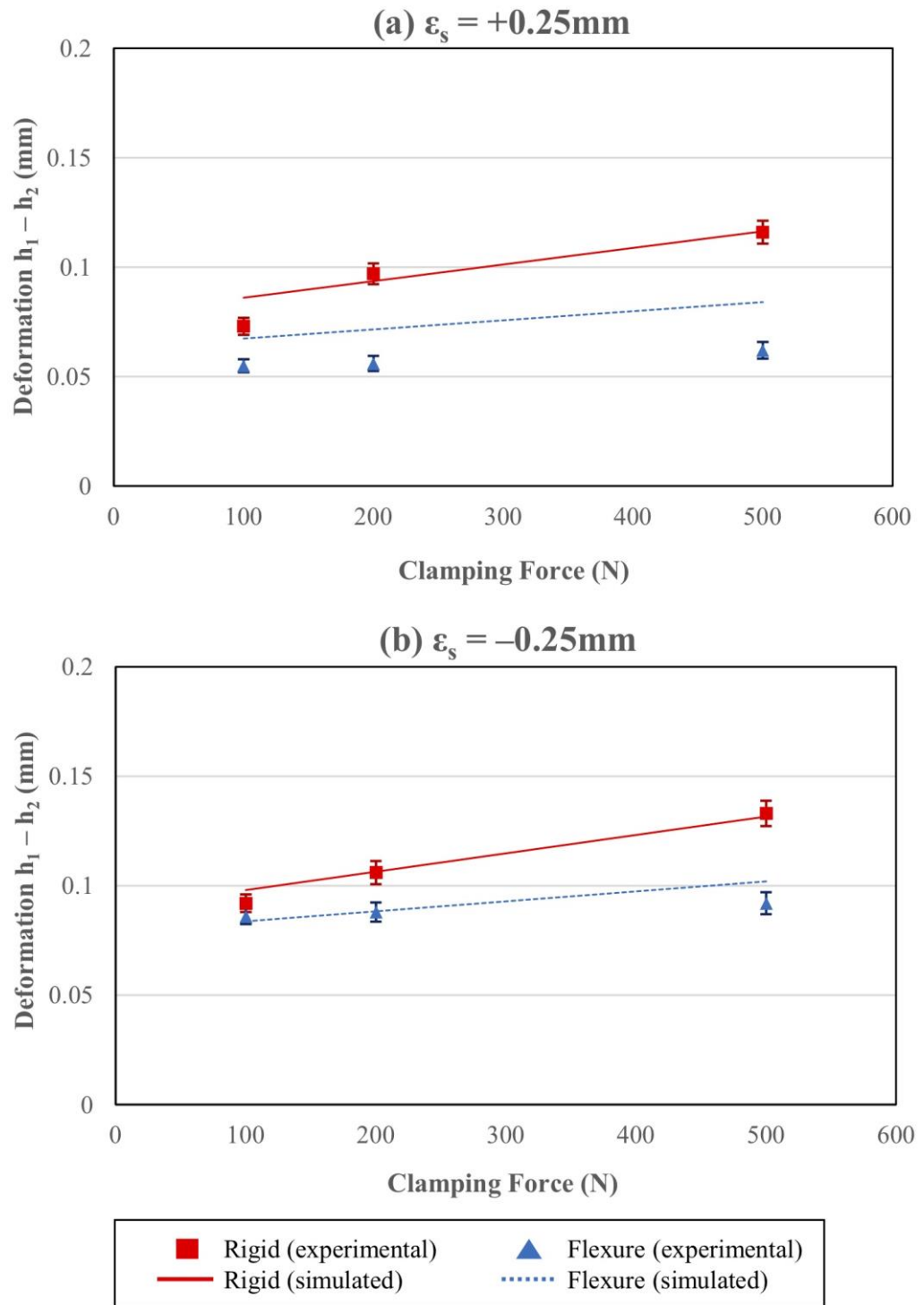


Figure 5-16. Plots of workpiece deformation against clamping force for the $\epsilon_s = \pm 0.25\text{mm}$ blades, comparing simulated and experimental values for each fixture system.

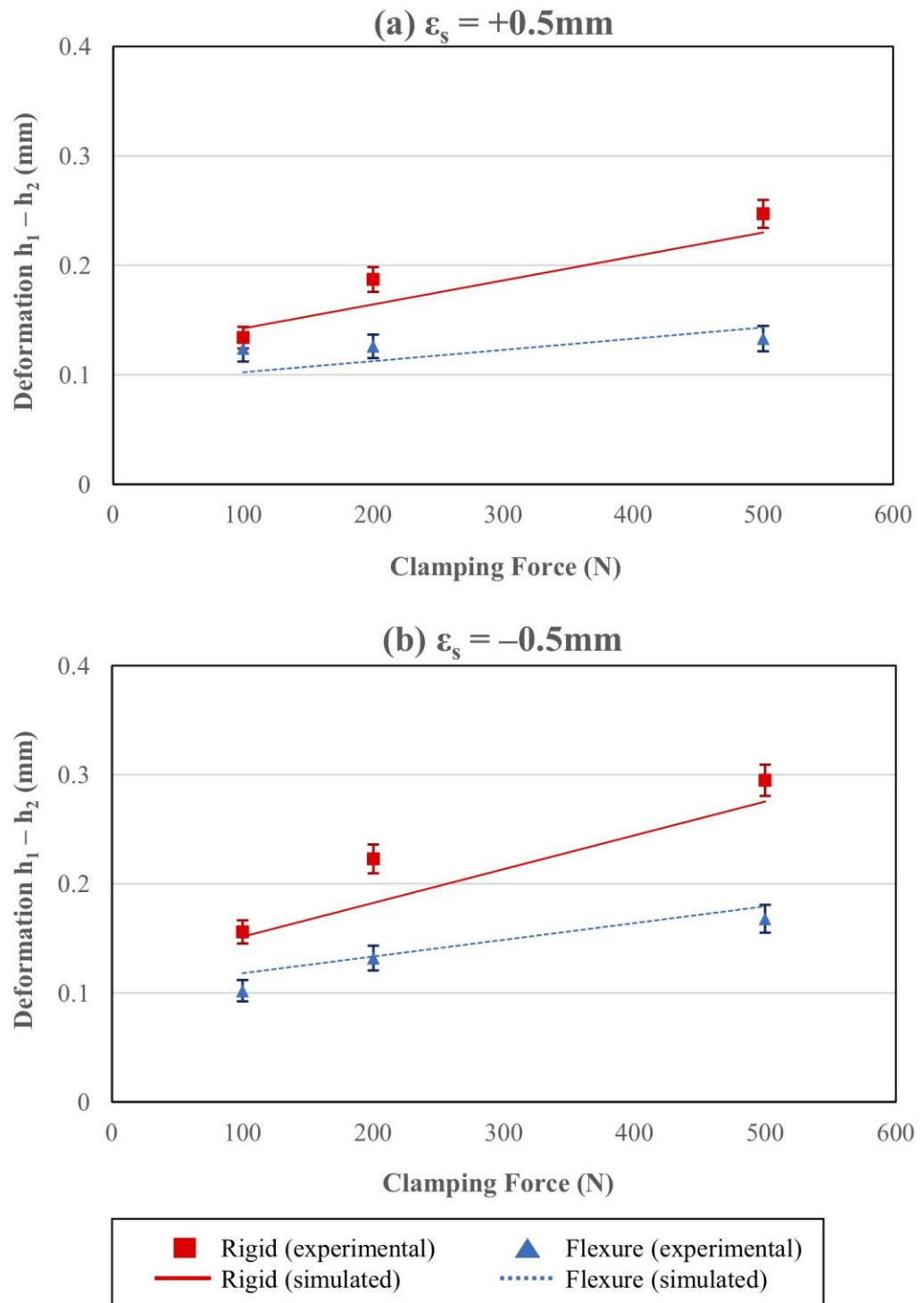


Figure 5-17. Plots of workpiece deformation against clamping force for the $\epsilon_s = \pm 0.5\text{mm}$ blades, comparing simulated and experimental values for each fixture system.

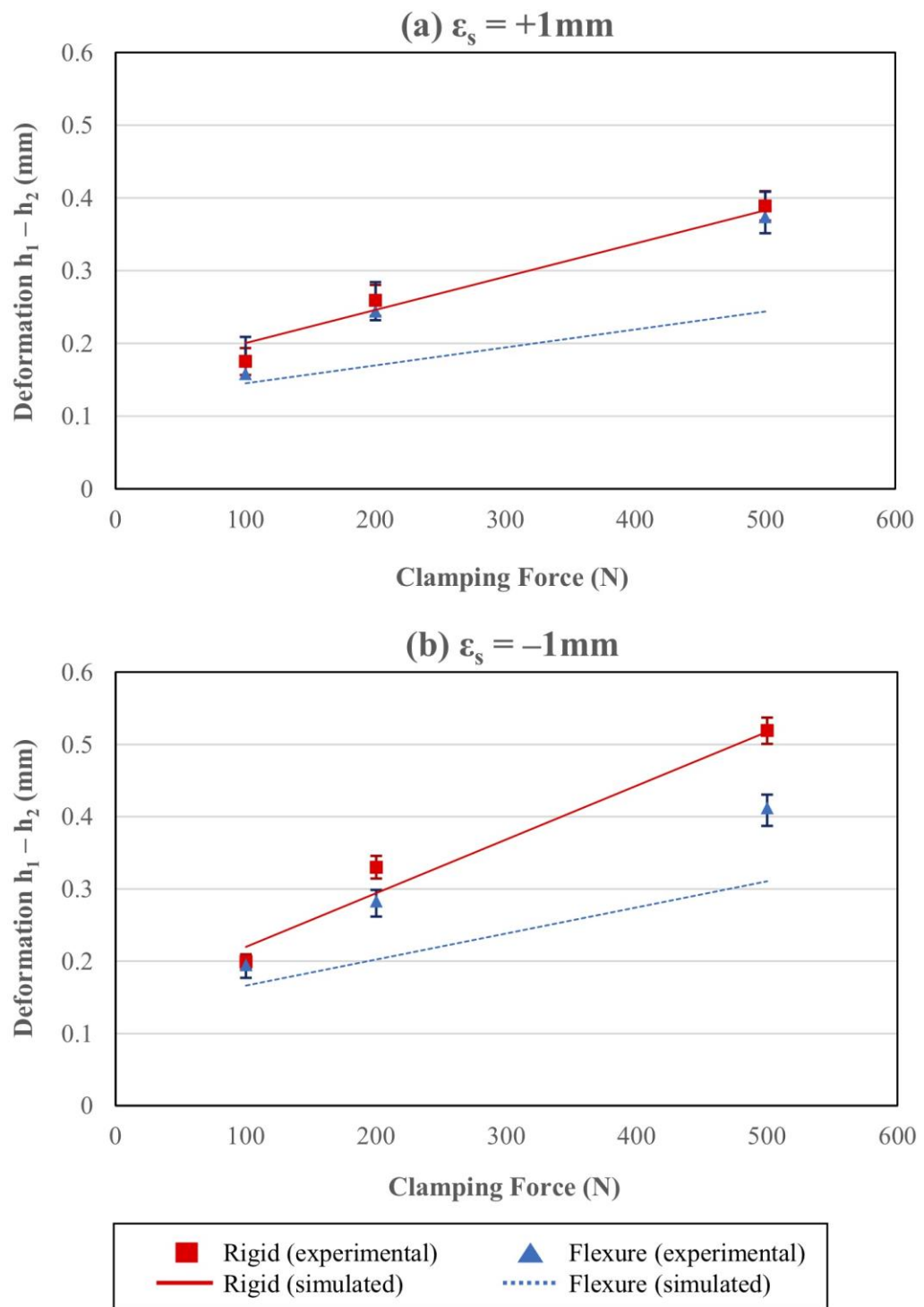


Figure 5-18. Plots of workpiece deformation against clamping force for the $\varepsilon_s = \pm 1\text{mm}$ blades, comparing simulated and experimental values for each fixture system.

For the blades with the smallest sagitta errors (Figure 5-15), the models predict little divergence in performance of the two fixture systems as loading increases, suggesting that small blade distortions can be accommodated reasonably well in the rigid fixture system. The DIC data shows that blade deformation is consistently smaller than predicted by the models for both fixtures, but particularly for the flexure fixture. The experimental data shows that deformation with the flexure fixture is on average 21% lower than the rigid system, however, due to the small initial distortion of these blade geometries, deformation is small in all cases: maximum deformation was measured at 0.066mm, observed for the $\epsilon_s = -0.125\text{mm}$ blade at 500N clamping in the rigid fixture.

Increasing the pre-existing distortion to 0.25mm (Figure 5-16) shows better agreement between simulated and experimental values for the rigid fixture system. Experimental data shows that the flexure system continued to reduce deformation compared to the rigid system by a greater margin than was predicted by the models. Flexures reduce workpiece deformation by an average of 28% across the range of loads, and up to 47% at maximum clamping force (Figure 5-16b). For errors of 0.5mm (Figure 5-17), the models prove more accurate in their prediction of flexure behaviour, with flexures reducing deformation compared to the rigid fixture by an average of 34%, up to a maximum of 46% (500N, Figure 5-17a). With distortions of 0.25mm and 0.5mm, the flattened ($\epsilon_s < 0$) blades show better agreement with models, exhibiting a more linear response as loading increases (Figure 5-16b and Figure 5-17b).

A phenomenon with the flexure fixture that is evident across the entire range of blades, but is more apparent for the arched ($\epsilon_s > 0$) blade geometries, is that workpiece deformation appears to increase in response to clamping force at a slower rate than is predicted by the models. This implies that the conformable fixture system does not significantly deform the arched component beyond the level needed to achieve full holding. Deformation appears to taper off after holding is achieved, with only modest increases in deformation as clamping increases beyond this point. This is a significant benefit for conformability,

because it means these blade geometries are less sensitive to over-clamping in a flexure fixture, while a rigid fixture would simply continue to deform the component in proportion to the applied clamping force, regardless of whether full holding had already been achieved.

As can be seen in Figure 5-18, the two components with the greatest initial distortion exhibit significant distortion under both fixture systems. While the general observations on fixture behaviour continue to apply to the blades with 1mm initial distortion, the improvement in workpiece deformation for the flexure system is significantly diminished. This appears to be due to components with the most extreme distortions approaching the limitations of the system, meaning it is unable to fully compensate for this severity of dimensional variation. While the flexures still reduce clamping deformation compared to the rigid fixture, it is by a significantly smaller margin than the models predict, with an average improvement of only 9%. This reduced effectiveness of the conformable fixture is due to some of the flexures struggling to grip onto workpiece surfaces which, due to severe distortion of the blade geometry, are at an oblique angle to flexure contact surfaces.

Across the experimental dataset, there is a clear trend that workpiece deformation in the rigid fixture system is relatively small at 100N clamping, with deformation rapidly increasing between 100N and 200N. This is because 100N is insufficient clamping force to achieve full holding with any of the distorted blades in the rigid fixture system, however as clamping reaches 200N, the components have started to deform more significantly. For configurations with modest distortions ($\epsilon_s \leq \pm 0.5\text{mm}$), where the flexures are at their most effective, workpiece deformation in the flexure fixture is reduced by an average of 19% at 100N, 28% at 200N, and 36% at 500N compared to the solid fixture.

The data obtained from DIC shows reasonable agreement with models, clearly confirming the predicted trend that the conformable fixture reduces deformation compared to the rigid fixture system, despite failing to predict the limitations of the system for the components with the greatest initial distortion. Although there is some divergence between simulated and experimentally-derived magnitudes

of clamping-induced deformation, FE predictions for the relative performance of the two systems are therefore verified. Analysis of experimental data shows that the conformable fixture consistently reduces clamping-induced deformation in workpieces whose geometric errors mean they cannot be held without the introduction of deformation. DIC results show that deformation experienced by the workpiece is proportional to the applied clamping force, with a clear trend that the performance of the two systems diverges as clamping increases, meaning flexures elicit the greatest benefit at higher clamping forces.

In addition to the behaviour of the workpiece, it is also necessary to examine the behaviour of the fixture blocks during clamping in order to fully explain the effects of conformability. This was achieved by studying the correlation data for the entire data field, and comparing the compressive behaviour to FE models. An example of this is presented in Figure 5-19, which compares the outputs from DIC and FE for both fixture types at 500N clamping for the configuration where $\varepsilon_s = 0.5\text{mm}$.

With this workpiece geometry, the flexure system was shown to significantly reduce deformation. For both FE and DIC, vertical displacement (in the direction of the applied clamping) of the workpiece-fixture assembly is visualised. In the rigid fixture (Figure 5-19a and b), it is evident from the large contour across the component that it is significantly deformed; deformation was predicted to be 0.223mm by the model, and measured as 0.247mm from DIC. While the component is deformed, there is very little displacement of the rigid fixture blocks, indicated by their near-uniform colouring. In contrast, for the flexure fixture (Figure 5-19c and d), there is much smaller variation in vertical displacement across the workpiece, as the blades have only a slight contour across their width. Deformation was predicted to be 0.142mm, with experimental data showing 0.133mm. Significantly, deformation of the flexures is observed, with localised regions of displacement visible where the flexures meet the workpiece, and the emergence of a noticeable contour visible across the fixture.

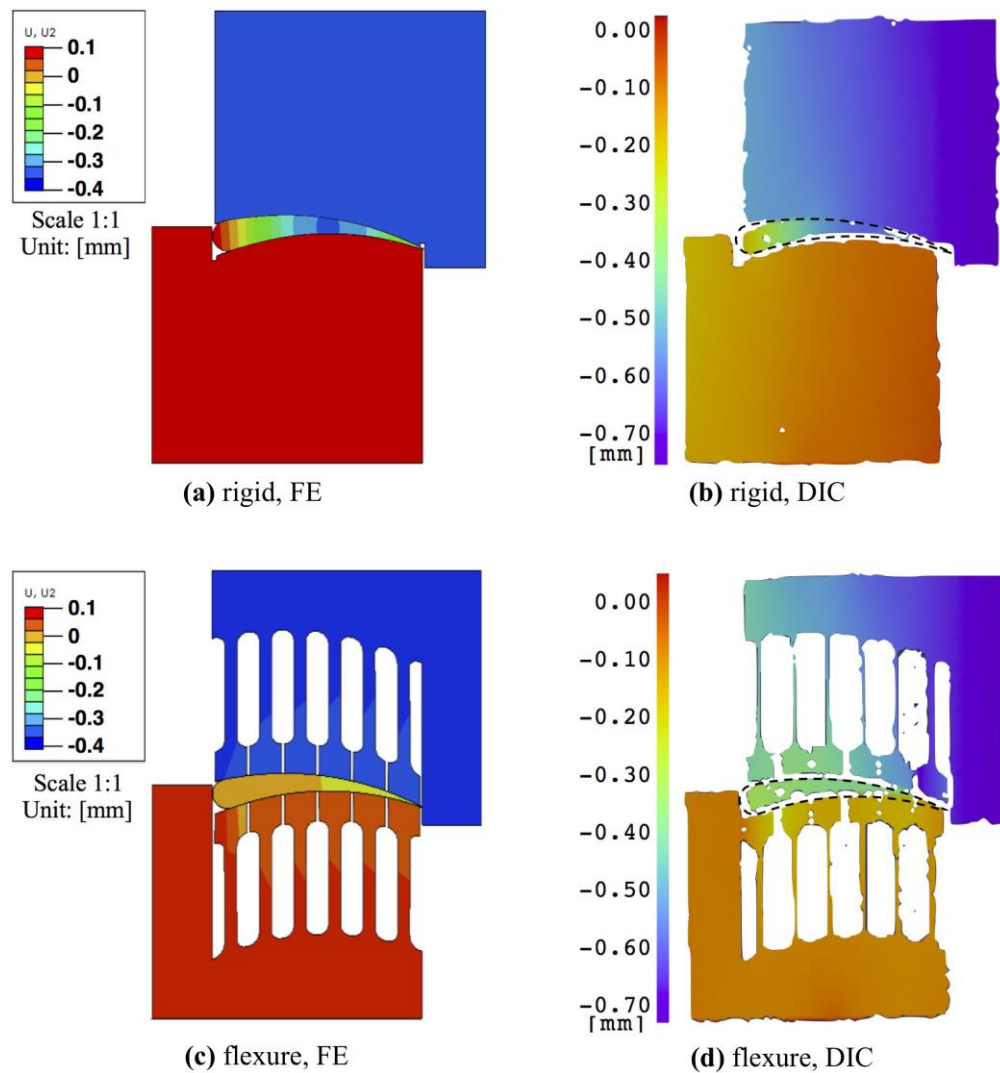


Figure 5-19. Example of visualised deformation of the $\varepsilon_s = 0.5\text{mm}$ component at $F = 500\text{N}$ in rigid and flexure blocks, comparing FE simulated and DIC experimental data for the two fixture systems.

These visualisations therefore reveal that the rigid fixture system introduces greater deformation to the workpiece during clamping because deformation of the fixture blocks is minimal. The result is that workpieces with geometric errors are simply deformed until they take on the shape of the fixture. This explains how the conformable fixture system is able to significantly reduce blade deformation, since the flexures deform in order to accommodate the workpiece without significantly deforming the workpiece itself. This represents a major finding, since flexures therefore have the potential to significantly reduce the deformation experienced by workpieces of variable geometry when they are held in a fixture.

The limitations of the flexure system were mentioned for the $\varepsilon_s = \pm 1\text{mm}$ components, where initial workpiece distortion was too extreme for conformability to significantly reduce deformation of the blades. This is caused by failure of the flexures to grip the component during clamping when pre-existing distortions create large gaps between the workpiece and fixture prior to clamping, such that workpiece surfaces are no longer approximately tangential to fixture surfaces. This has been observed to cause slipping of the flexures along the workpiece surface, as can be seen in Figure 5-20, which results in reduced effectiveness of fixture conformability.

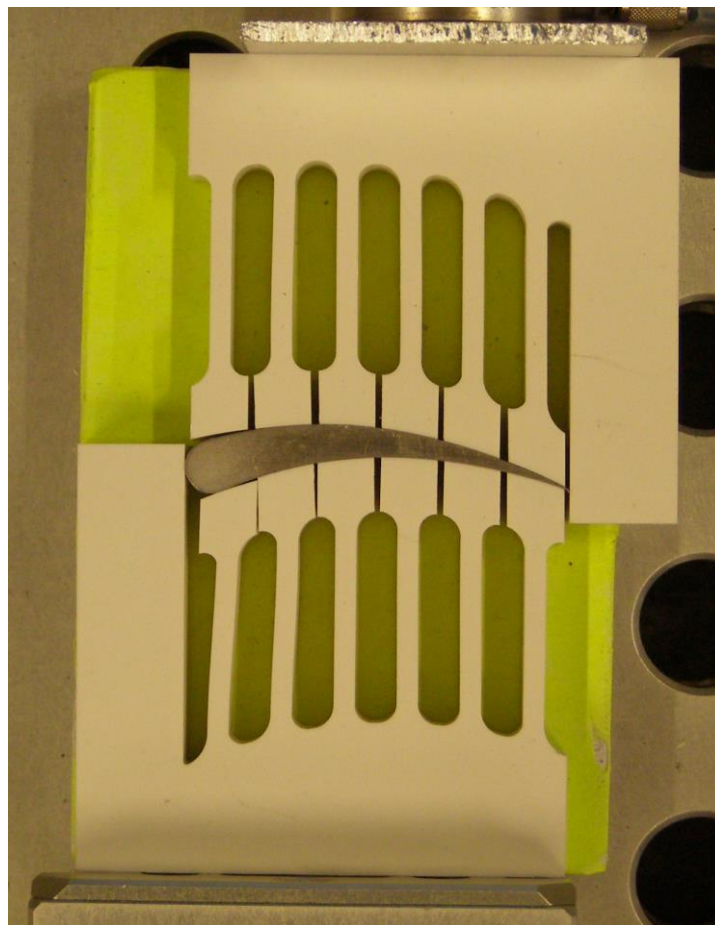


Figure 5-20. Conformable fixture holding the $\varepsilon_s = 1\text{mm}$ component at 500N clamping, exhibiting noticeable slipping of the lower left flexures.

When a flexure fails to grip the workpiece, and instead slips along the contact surface as clamping increases, that flexure is prevented from bearing the full clamping load that would be sustained if full contact was made with the workpiece. This means that clamping loads are distributed over a smaller

footprint across the remaining flexures that remain in firm contact with the workpiece. This is undesirable, since the concentration of clamping forces within the fixture means that the workpiece is much more likely to experience large deformation. Therefore ensuring (where possible) that all flexures in a conformable fixture are engaged with the workpiece during clamping will help to further minimise deformation, by distributing clamping over the greatest possible area.

While flexure slipping was found to adversely affect the ability of flexure-based fixtures accommodate the most severely distorted workpieces, this could potentially be addressed in future revision to the flexure concept. In the current prototype, all flexures are oriented parallel to the clamping force, however orienting the flexures normal to workpiece surfaces could potentially reduce slipping. While a switch to normal-oriented flexures would not alter the angle of incidence between flexure contact surfaces and the surface of a distorted workpiece, the flexures would nonetheless be less susceptible to slipping. This is because clamping reaction forces between the workpiece and flexures would be transferred more longitudinally through each flexure, thus reducing the lateral forces acting on the flexures, and therefore decreasing their tendency to slip.

The investigation into the effect of fixture conformability on clamping deformation demonstrates that flexures offer a clear advantage over non-conforming rigid surfaces. Flexures are shown to be capable of maintaining acceptable levels of clamping-induced deformation (meaning finished components that are within tolerance can be produced, and are free from damage such as crimping or other plastic deformation) in workpieces with far greater dimensional variations than is possible in a conventional rigid fixture system. This is because a proportion of the deformation that is necessary to achieve full workpiece holding is borne by the fixture rather than the component.

5.5 Summary

This chapter presents an investigation into fixture conformability with flexures, and its effect on clamping-induced workpiece deformation. Flexures are shown

to significantly improve holding of thin, delicate workpiece geometries that are subject to dimensional variability, by greatly reducing the deformation to which they are subjected during clamping.

The optimal method of generating flexure geometry for freeform workpieces was determined using FE, and shown to impart consistent buckling and frequency properties (variation between flexures well below 1%). Using this methodology, a simplified fixturing setup was developed based on thin fixture blocks, which replicated a thin section with a single row of flexures through a larger fixture for a freeform aerospace component.

An experiment was devised to compare conformable flexure-based fixtures against a conventional fixture with a rigid clamping surface. Components with deliberate geometric errors of increasing severity were manufactured and used to investigate the ability of flexure fixtures to accommodate workpiece variability. Initial investigations into the clamping force required to fixture the blade workpieces revealed that the flexure fixture achieves secure holding at up to 60% lower clamping forces, with an average reduction of 49%.

Following this, an investigation into workpiece clamping deformation in the two fixture systems once again revealed considerably improved performance of the flexure fixture system. In reasonable agreement between models and experimental data, flexure conformability was shown to reduce deformation of workpiece geometries with errors by an average of 23%, with reductions over 40% achieved in many configurations.

This deformation-reducing phenomenon was shown from DIC and FE data to be caused by the ability of flexures themselves to deform, rather than imposing their own profile upon the component. This makes flexures highly effective in accommodating geometric variations, since the rigid system simply deforms the workpiece until it adopts the nominal fixture profile.

The limitations of the flexure system for accommodating severely distorted geometries was revealed to occur when workpiece sagitta deviated from its nominal value by more than 30%. At this point, the flexures begin to lose their effectiveness in accommodating the distorted geometry, because they become susceptible to slipping along workpiece surfaces instead of gripping them, meaning that flexures are unable to bear full clamping loads.

Based on these findings, conformable flexures are able to offer significant improvements in workpiece holding. Reduction of workpiece deformation through conformability therefore has the potential to enable the fixturing of extremely delicate and variation-susceptible geometries, while providing improved support, thus leading to greater accuracy of the finished component than is possible with existing fixture technologies.

6 Machining with flexures for reduced workpiece vibration

6.1 Introduction

This chapter investigates the second key capability of the fixture concept: the ability of machining fixtures incorporating flexures to damp workpiece vibration of thin components during milling. Presented is a proof of concept to assess the viability of additively manufactured fixtures for machining operations, and to allow comparisons between the novel fixture concept and conventional fixture designs for levels of vibration experienced by the workpiece. Fixture blocks are designed with the aid of FE models, alongside a simple cantilever workpiece. A machining stability analysis is used to determine suitable tooling and optimal machining parameters, with FE used to determine machining frequencies of interest. The machining setup is modelled to predict relative performance of conventional and flexure-based fixtures, with simulations verified experimentally.

6.2 Additively manufactured machining fixtures incorporating flexures

If fixtures containing flexures are to prove useful for machining applications, they must be able to withstand machining loads, while maintaining location and holding of the workpiece. For an entire fixture system to benefit from the advantages of flexures demonstrated in the previous chapter, additive manufacture is the only viable method for quick and cost-effective production. Therefore an important stage in the validation of the concept is to test the suitability of additive fixtures for machining applications.

For the novel fixture concept to be exploited in a machining fixture, any potential advantages in holding capability introduced by flexures must not be negated by any detrimental effects on workpiece machinability compared to a conventional fixture. This could be due to problematic changes in the dynamic properties of the workpiece-fixture system, or simply the inability to maintain adequate holding of the workpiece during machining. Therefore experiments

were undertaken to assess the viability of an additive flexure-based machining fixture for milling a low stiffness workpiece. Also investigated was the exploitation of flexure resonance to achieve a frequency-specific workpiece vibration damping effect.

Complex workpiece geometries, such as the blade-like components used to investigate conformability, pose three challenges for the evaluation of the flexure fixture concept in machining applications. Firstly, complex components typically vary in stiffness across their geometry, which results in time-dependent machining response across different regions of the workpiece. Secondly, freeform geometries tend to have a more complex vibration response, with a greater number of eigenfrequencies over any given frequency range, meaning they are more susceptible to resonance. And finally, the curved profile of a freeform workpiece necessitates a complex tool path in order to maintain a constant rate of material removal. Therefore, at this stage of the research, in order to reduce the number of potential variables and sources of error, a flat workpiece geometry with uniform stiffness was selected for milling experiments, in order to enable constant, resonance-free machining response with a simple linear tool path. The use of a flat workpiece means that each flexure supports an identical region of the workpiece, since there are no curved surfaces to accommodate, meaning all flexures therefore have identical geometry.

The fixtures used to investigate machining performance were manufactured additively in ABS using the fused deposition modelling (FDM) process. Printed ABS filament is likely to vary in properties from bulk material, and therefore to enable accurate modelling of flexures in a printed fixture, tensile testing was undertaken on example printed ABS flexures in order to obtain mechanical properties of the materials. The flexures, which had a gauge length of 20mm, were tested using an Instron universal testing machine (UTM), with a stochastic speckle pattern applied to enable motion tracking for measurement of strain.

The cross section of the gauge region was a square with sides of 2.03mm, which corresponds to the thickness of exactly eight printed layers of ABS filament in

the z axis of the machine. This is four times the minimum wall/feature thickness recommended for the Stratasys Dimension 1200es printer, and was chosen because it offers a reasonable compromise in terms of strength and compressibility, allowing tuning of the flexure geometry over a large eigenfrequency range. The flexures were printed in the xz orientation, to allow direct comparison between experimental values and those from the supplier material datasheet. Three identical flexures were tested to failure, with tensile displacement applied at a quasi-static rate of 1mm per minute (in accordance with ASTM D638 [120]), and the mechanical properties taken as the average from the three samples. The testing proved highly consistent, with the failure load ranging from 135 to 137N. Shown in Table 6-1, the measured values for yield stress proved to be very close to those from the data sheet, with the modulus of the material found to be slightly lower.

Table 6-1. Printed material properties for ABSplus-P430, comparing the manufacturer's specification with values obtained experimentally from printed samples.

	Data Sheet	Tensile Test
Yield Strength (MPa)	31	31.13
Young's Modulus (GPa)	2.200	2.064

These values were used to model printed fixtures so that the frequency response of flexures could be predicted accurately. Since additive components are orthotropic, with very low tensile strength between printed layers, the additive fixture blocks were designed and oriented for printing so that no tensile forces would be acting in the z axis (between the layers) during usage. This allowed the fixture blocks to be modelled as an isotropic system.

To enable validation of the flexure concept for machining applications, three pairs of fixture blocks were designed. They were developed from the fixtures used to investigate conformability, based around rows of six flexures, using the same external dimensions. In contrast to the testing configuration studied previously, which used a single row of flexures to simulate a section through a fixture, these fixture blocks comprise entire fixtures and therefore have a depth

of five rows of flexures. Two of these rows contain locating features in order to provide workpiece location in the direction of applied clamping. The fixture blocks are shown in Figure 6-1. In addition to the fixture with flexures, two fixture block designs were produced without flexures to allow comparison between the flexure system and conventional fixturing methods. Common to all three block types is the inclusion of three solid locators on the main workpiece clamping surface, and the locating end wall. These features ensure that workpiece location is deterministically defined, and can be maintained throughout machining.

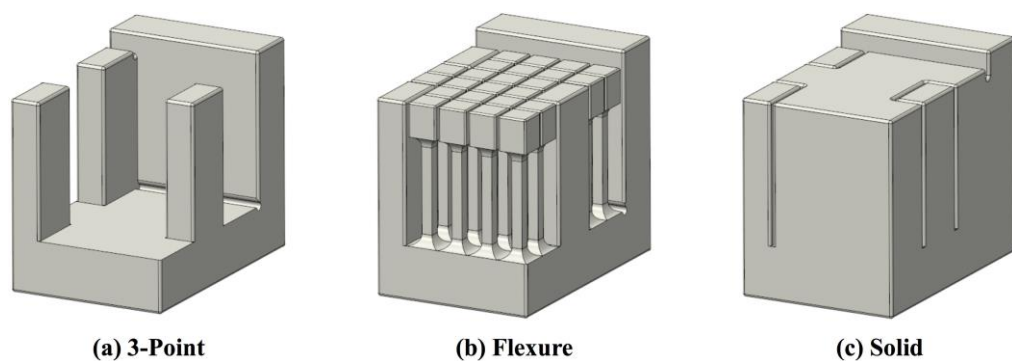


Figure 6-1. The three additively manufactured machining fixture block designs.

The first fixture block design (Figure 6-1a) contains only the three locators, which means the workpiece is completely unsupported between the locators in the style of a pin-based fixture. It is hereafter referred to as the 3-point fixture system. The second block (Figure 6-1b) builds on this with the addition of 24 identical flexures occupying the space between the locators, and is therefore referred to as the flexure block. The final fixture block (Figure 6-1c) entirely fills the space between the locators to create a rigid contact surface, making it comparable to a vice, and is referred to as the solid block. To accommodate the gaps between flexures necessary for them to be able to move independently under compression, the flexure fixture has a workpiece contact region that has 95% of the area of the solid fixture contact surface. The 3-point fixture has a contact area that is only 16% of the solid fixture, meaning the solid and flexure fixture systems effectively provide full-face workpiece support, while the 3-point system provides minimal support to the workpiece.

The square pillar cross section of the flexures is desirable for inducing resonance since they are symmetrical, and therefore do not deform preferentially in any direction. With an asymmetrical flexure, the first bending modes in the two orthogonal directions will occur at distinct frequencies, however the use of a symmetrical flexure geometry means that the first eigenfrequency is caused by simultaneous resonance in both directions. This is known as a repeated root, where two orthogonal vibration modes occur at the same eigenfrequency, causing the flexures to resonate readily and prominently at that frequency, and is therefore advantageous if resonant behaviour is to be exploited.

The frequency response of the flexure block was analysed by adapting the existing model that was used to accurately predict resonant behaviour of the flexure pegs in the testing platform, and the conformable fixtures. The flexures designed for the flexure fixture blocks were based on the sample printed flexures used for tensile testing, but with the deformable length shortened from 20mm to 16mm, which was predicted to give a first flexure eigenfrequency of 400Hz. The final printed flexure fixture blocks exhibited resonance at 410Hz, an error of 2.5%, thus reinforcing the strong predictive capability of the model, even for additive components.

The two conventional fixture types permit useful comparisons against the flexure fixture, since they represent real fixturing applications. Significantly, the flexure fixture system sits between the 3-point and solid fixture designs in terms of mass, stiffness, and contact surface area with the workpiece. As a result, it should be expected under normal conditions that the machining performance of the flexure fixture would fall between the two conventional fixtures for any metric that is dependent on the physical characteristics of the fixture blocks. Therefore, any advantage arising through the use of flexures cannot simply be attributed to greater workpiece contact compared to a 3-point fixture, or lower rigidity of holding compared to a solid fixture.

To ensure good engagement between the flexure fixture and the workpiece, all flexures needed to be taller than the locators they surrounded, shown in Figure

6-2, so that they were lightly compressed when the workpiece was clamped in place. Since the entire clamping surface (flexures and locators) was milled across to ensure that it was even and flat, this was achieved by simply milling the locators to be slightly shorter than the flexures around them.

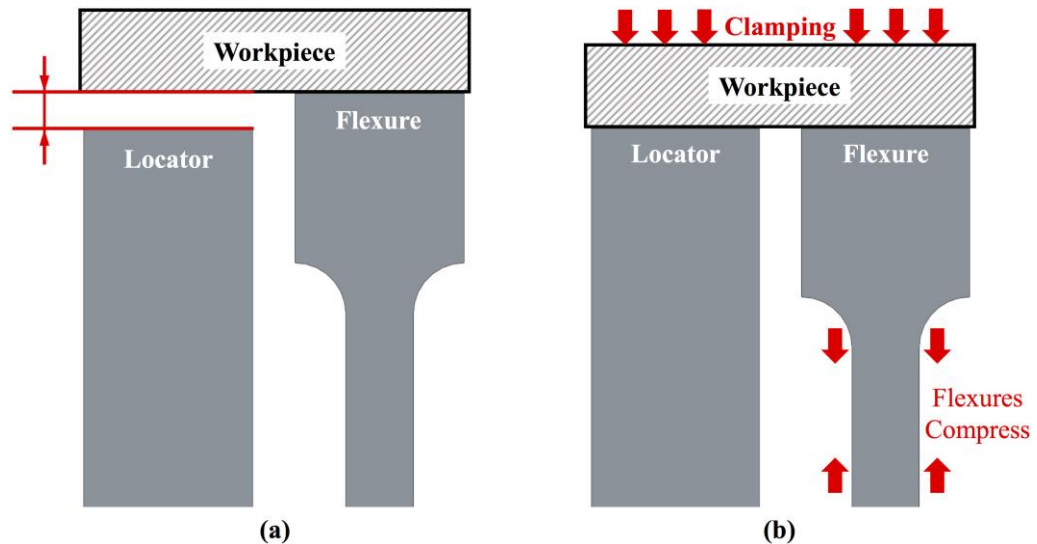


Figure 6-2. Illustration of unclamped (a) and clamped (b) fixture, showing how over-height flexures are compressed as clamping pushes the workpiece against the locators.

The optimal flexure height difference was investigated using Fujifilm Prescale LLW pressure measurement film. It was found that a gap of $150\mu\text{m}$ (corresponding to less than 1% compressive strain) provided consistent flexure contact, while minimising deformation, as shown in Figure 6-3. Dynamic coupling of flexures was verified using hammer modal testing, which revealed flexure resonance to be clearly detectable, with negligible shift in frequency response compared to when no height difference was present (no gap). Since the locators are rigid compared to the flexures, they serve to limit maximum flexure deformation, preventing them from becoming over-compressed such that their frequency response is affected. Smaller height differences between the flexures and locators below $100\mu\text{m}$ were shown to provide unreliable flexure engagement, while gaps in excess of $250\mu\text{m}$ were susceptible to flexure compression that was not purely longitudinal, which therefore altered the frequency response.

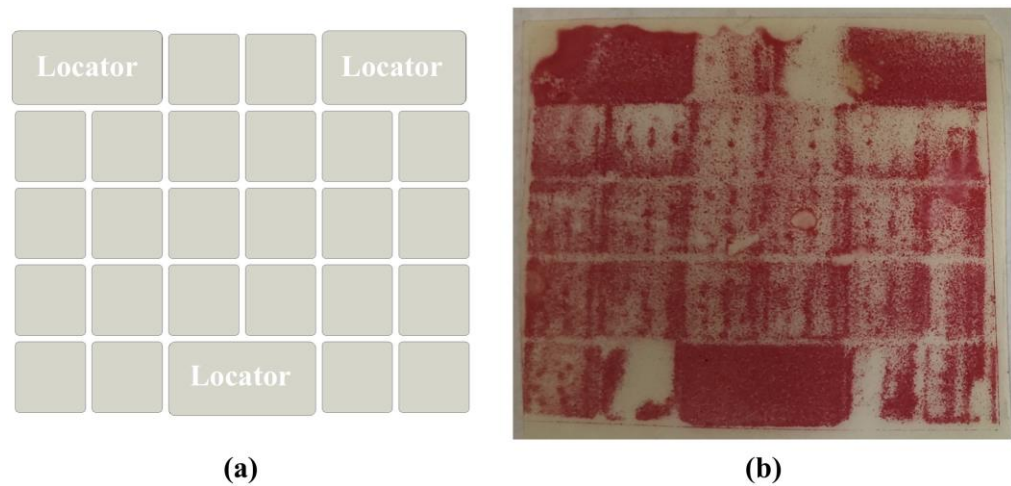


Figure 6-3. Clamping surface profile of the flexure fixture block with locators highlighted (a), with pressure measurement film showing good flexure-workpiece engagement (b).

6.3 Machining a low stiffness workpiece

An experimental configuration was devised to enable comparison between the three types of flexure block in terms of the level of workpiece vibration experienced by a simple workpiece during machining. The Hermle C 800 U 5-axis vertical machining centre was used to mill cantilevers cut from 2mm aluminium plate, shown in Figure 6-4. Matching pairs of fixture blocks were used to hold the cantilevers, with a toggle clamp in a custom jig used to apply a constant clamping load of 860N. Small slots were end milled in the top of the workpiece, with the PCB 352B01 single axis accelerometer used to measure workpiece vibration, and a Kistler 9257 dynamometer used to measure cutting forces. Varying the spindle speed of the machine allowed alteration of the frequency at which the workpiece-fixture system was excited by the cutter teeth, with feed rate adjusted proportionally to provide constant feed per tooth at all spindle speeds.

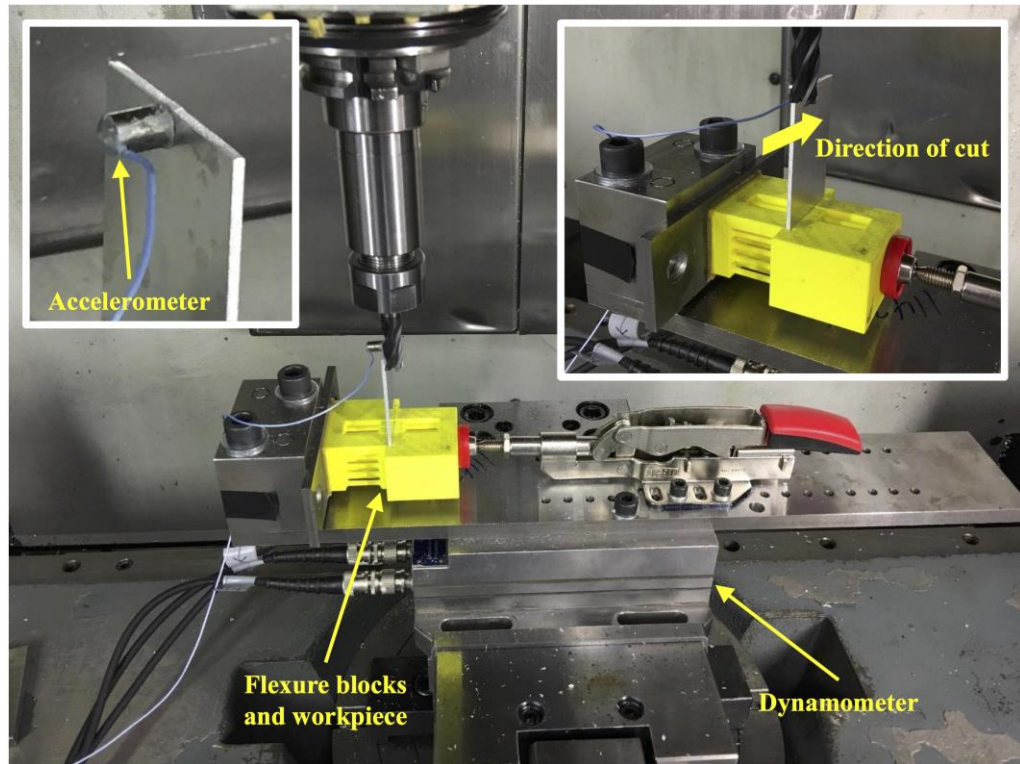


Figure 6-4. Machining setup for evaluating the performance of the flexure-based fixture.

The cantilever workpiece geometry is ideally suited to the measurement of machining-induced vibration, thus facilitating evaluation of the relative performance of the different fixture systems. This is because its thin wall geometry makes it inherently susceptible to vibration at its unsupported end, but it has relatively few eigenfrequencies to cause resonance, and its straight geometry allows use of a simple tool path. The nature of a cantilever means that its frequency response is defined primarily by its length, and the geometry chosen for machining had a total length of 79mm (free region length 49mm), with a width of 37mm (meaning it rested against the locating walls of the upper and lower fixture blocks), which gave a nominal first eigenfrequency at 500Hz in the flexure fixture. This length was selected because it was found during preliminary trials to be stiff enough to prevent uncontrollable vibration during machining, and because its 500Hz first bending frequency is sufficiently distinct from the first eigenfrequency of the flexures (410Hz), thus permitting machining excitation between these two frequencies without invoking resonance of either system.

Hammer modal testing was also undertaken on the cantilever workpiece in the two conventional fixture systems, to measure the effect of the different fixture types on the frequency response of the workpiece. Compared to the flexure system, the results showed that there is negligible shift of the first eigenfrequency of the workpiece with the 3-point fixture (a reduction of less than 0.2%). This is because the low stiffness of the flexures means they did not significantly alter the behaviour of the workpiece-fixture system. For the solid fixture system, the first natural frequency of the cantilever was shifted 3.2% higher, due to the stiffening effect of the rigid support on the cantilever, however this change was deemed to be acceptably small. In addition to the eigenfrequencies, key parameters that can be derived from the frequency response function (FRF) include the damping ratio, stiffness, and mass of each mode. These are potentially significant because, while shifting of natural frequencies between fixture types indicates some difference in the nature of holding, frequency shifts alone do not reveal whether the stiffness or damping of the structure has changed, or whether the first eigenfrequency becomes more or less prominent compared to other modes. Therefore further processing of the modal testing data was undertaken to analyse the frequency responses of the workpiece in the three fixture systems.

A standard FRF is presented as the modulus of frequency response, plotting the inertance (acceleration per unit force from the hammer impact) value for each frequency over the specified range, as discussed in Chapter 3. Using the analysis tools in DATS Analysis, the complex (real and imaginary) components were extracted from the FRF for each fixture system, to enable extraction of the four key parameters necessary to calculate damping, stiffness and mass data for each eigenfrequency. From the imaginary signal, the first peak in inertance was used to determine the frequency of the first mode. Next, an omega arithmetic integration function was used on the imaginary component to convert accelerations into displacements, in order to change the units from inertance to receptance (displacement per unit force, with units of m/N), to reveal the value of the receptance at the first mode. Finally, from the real component of the FRF, the frequencies which corresponded to the minimum and maximum values of inertance for the first eigenfrequency were noted. Using the eigenfrequencies,

their corresponding receptance, and the frequencies at peak inertance, it was possible to fully characterise the frequency response for each fixture system: damping ratio was calculated from the width of the frequency peak in the real component of the FRF; stiffness was calculated from the damping ratio and the imaginary component receptance of the first mode; and modal mass was calculated from the mode frequency and stiffness. Complete frequency response data is presented in Table 6-2.

Table 6-2. Detailed frequency response, comparing properties of the first workpiece eigenfrequency for each fixture type.

Fixture Type	Eigenfrequency (Hz)	Damping Ratio	Stiffness (N/m)	Modal Mass (kg)
3-Point	501	0.00699	3.438×10^4	0.00347
Flexure	502	0.00598	3.658×10^4	0.00368
Solid	518	0.00579	5.767×10^4	0.00545

The frequency data shows that there is relatively little difference in the resonant behaviour of the three fixture systems, meaning the workpiece frequency response is therefore not significantly affected by fixture type. This means that switching between these fixtures will not introduce any adverse effects on the dynamics of a machining process. The 3-point and flexure systems have a difference in their first eigenfrequency of 1Hz, with correspondingly similar stiffness. The solid system has somewhat distinct properties from the other two fixtures, with a higher workpiece eigenfrequency and greater stiffness. The damping ratio of the cantilever workpiece is lowest in the solid fixture, which is due to the reduced potential for free workpiece vibration to be dissipated via the fixture. Due to its large workpiece contact area, the flexure system has a damping ratio that is similar to the solid system, however the low stiffness of flexures means that its stiffness is much closer to the 3-point system. For each of the derived parameters, the flexure fixture falls between the two conventional fixture systems, thus reinforcing the expectation that it should be the middle performing of the three fixtures under typical machining conditions.

6.3.1 Machining stability analysis

Selection of appropriate tooling is essential in machining, since unsuitable machining conditions will create unstable conditions that lead to tool chatter. The regenerative mechanism of chatter means that it can cause large vibration of the workpiece, even when the machining parameters would otherwise give rise to satisfactory cutting conditions. To determine the conditions necessary to permit stable machining, a stability analysis was conducted. This allowed selection of suitable tooling and machining parameters, and resulted in generation of a stability lobe diagram to determine the maximum axial depth of cut that permits stable milling conditions for a given spindle speed range.

In order to produce a stability plot, the software requires data from the chosen cutting tool and workpiece material, as well as the proposed machining parameters. By specifying the chosen radial depth of cut and feed per tooth values, the simulation is able to predict the maximum axial depth of cut that can be achieved before the onset of tool chatter over a desired range of machining speeds. A 4-flute cutter was deemed to provide a sufficient range of excitable frequencies in machining, with excitation at 1000Hz at possible at the maximum speed of the machine. End mills with diameters of 6mm and 10mm were investigated, with the 10mm cutter chosen due to its higher ratio of diameter to radial depth of cut, which means that cutting excitation is provided by a single cutting tooth, explained in Figure 6-5. The use of a smaller cutter would result in more than one tooth being in simultaneous contact with the workpiece, which has the potential to diminish the clean periodic nature of excitation, due to the cutter being in constant contact with the workpiece, instead of momentarily moving off the workpiece surface between cutting interactions. Use of a larger cutter is also advantageous due to its higher stiffness, which makes it less susceptible to chatter.

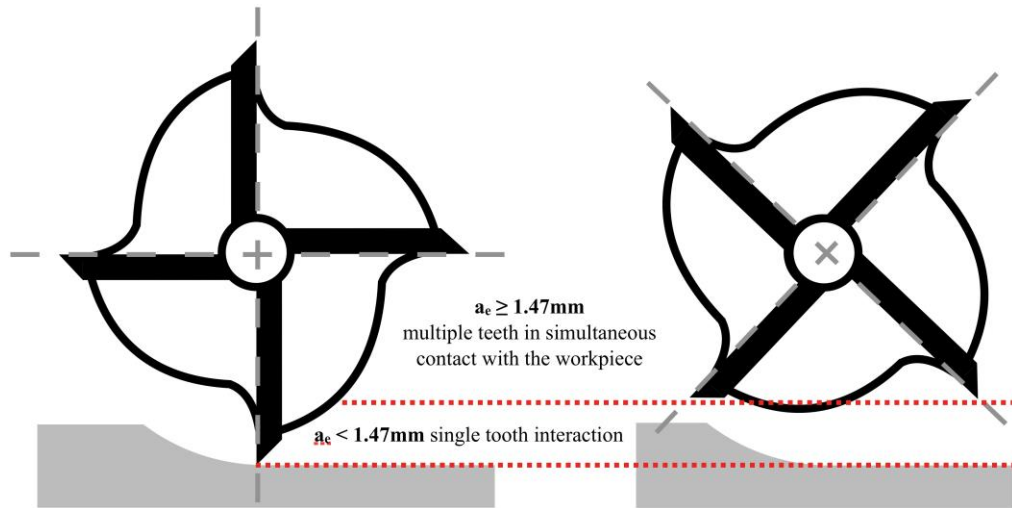


Figure 6-5. Schematic (to scale) of a Ø10mm end mill cutting a 2mm thick cantilever workpiece, showing how radial depth of cut (a_e) affects the maximum number of teeth in contact with the workpiece. When $a_e < 1.47\text{mm}$, a maximum of one cutting tooth will be in contact with the workpiece at any given moment.

Hammer modal testing was then undertaken with the cutter in situ in the machine (mounted in a collet chuck tool holder, which was inserted into the machine spindle). The tool overhang was selected as 40mm, since this was short enough for the tool to maintain high stiffness, but also long enough for the tool holder to clear the fixture assembly during machining. The modal test was conducted twice, with the accelerometer repositioned 90° around the cutter between tests, to capture the frequency response of the tool in the spindle in its x and y axes. The frequency response functions were interpreted to determine the additional data from each eigenfrequency needed for the stability analysis, in order to determine the relative influence of each mode on the cutter, with the full dataset given in Table 6-3 and Table 6-4. In addition to this frequency data, the helix angle and tooth spacing was entered into the simulation to define the tool.

Table 6-3. Tool frequency response in the x direction of the machine.

X-Axis Eigenfrequencies				
Mode	Frequency (Hz)	Damping Ratio	Stiffness (N/m)	Modal Mass (kg)
1	988	0.024	2.800×10^7	0.727
2	1913	0.019	2.340×10^7	0.162
3	3519	0.010	6.419×10^7	0.131
4	4247	0.030	5.989×10^6	0.008

Table 6-4. Tool frequency response in the y direction of the machine.

Y-Axis Eigenfrequencies				
Mode	Frequency (Hz)	Damping Ratio	Stiffness (N/m)	Modal Mass (kg)
1	1198	0.023	4.197×10^7	0.741
2	2034	0.019	3.576×10^7	0.219
3	3339	0.025	2.372×10^7	0.054
4	4168	0.014	1.490×10^7	0.022
5	4308	0.014	1.345×10^7	0.018

In order to define the workpiece material in the model, it was necessary to specify its cutting force coefficient, commonly also known as specific cutting force, or cutting stiffness, in the direction of cutting. The software requires only a single coefficient value, k , corresponding to the shearing component of the cutting interaction in the direction of cutting. This simplifies the process of obtaining the coefficients, since there is no need to resolve x , y and z direction cutting forces into their tangential, radial, and axial components, or to consider ploughing force interaction during cutting. The coefficient was obtained experimentally through preliminary machining trials. Replicating the milling operation to be used on the cantilever workpiece, the selected tool was used to shoulder mill a small cut (2mm axial and 1mm radial depth of cut) into thick

blocks of 2014 aluminium (the same alloy as the cantilever workpieces), which were held in a machine vice mounted to a dynamometer. A constant spindle speed of 3000rpm was used, with cutting undertaken at four different feed rates: 30, 60, 90, and 120mm/min, which corresponded to feed per tooth of between 2.5 μ m and 10 μ m. Machining was undertaken with and without coolant, which demonstrated coolant to be unnecessary for this combination of material and parameters; therefore dry machining was used for all subsequent experiments. For these parameters, machining was empirically observed to be stable (free from chatter) in all cases, since cutting force data showed a clean signal free from sudden spikes in cutting force, and the noise from cutting was quiet and continuous. For each feed rate, the RMS force in the direction of cutting was calculated, with cutting force per unit chip width (radial depth of cut) plotted against chip length (feed per tooth). The gradient of the line of best fit between these points corresponds to the cutting force coefficient required for the simulation, and the data is presented in Figure 6-6.

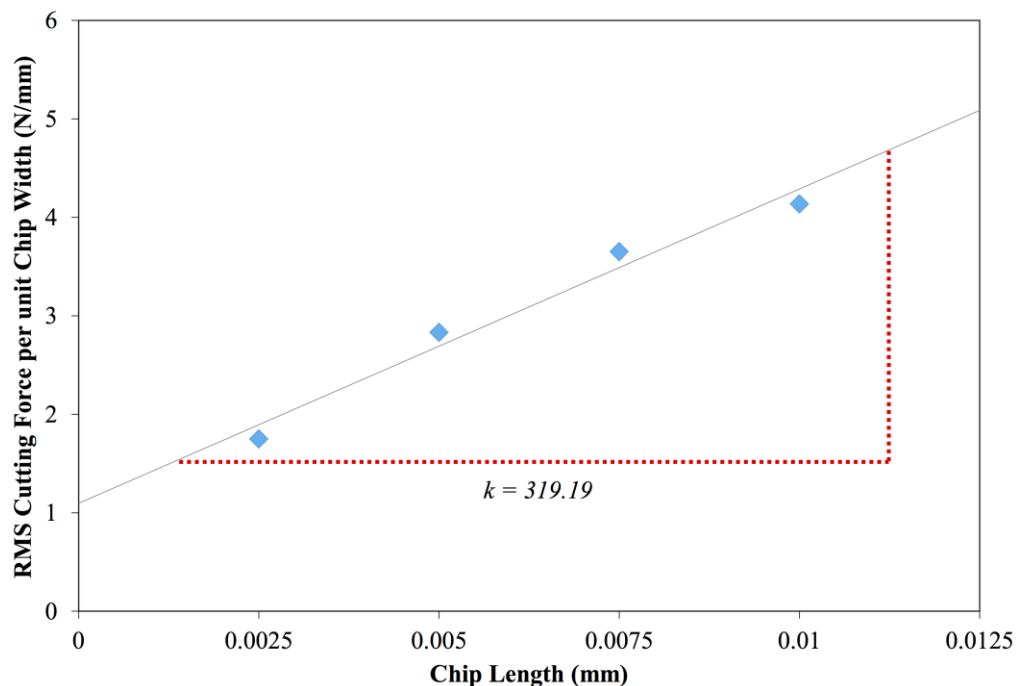


Figure 6-6. Plot of cutting forces against feed per tooth to determine the cutting force coefficient, k .

From the plot it can be seen that the cutting force coefficient, k , is equal to 319.2N/mm², and this value was used for the stability analysis simulations. This

value falls within the typical range of coefficients for non-heat treated low-silicon aluminium alloys [121]. Once properties of the cutting tool and workpiece material had been determined, only the machining parameters, plus the range of machining speed and axial depth of cut values to be simulated, remained to be defined.

For the machining stability simulation, it was determined that 1mm would make for a suitable radial depth of cut, so that the milled profile in the cantilevers would have similar dimensions to the cuts used to acquire cutting coefficients for the material. In order to reduce the likelihood of machining instability affecting the milling of the cantilevers, a conservative feed per tooth of $3\mu\text{m}$ was selected. The machining stability simulation was run for spindle speeds covering the range 2000-8000rpm, for axial depths of cut ranging from 0.5-10mm. The stability plot output is given in Figure 6-7.

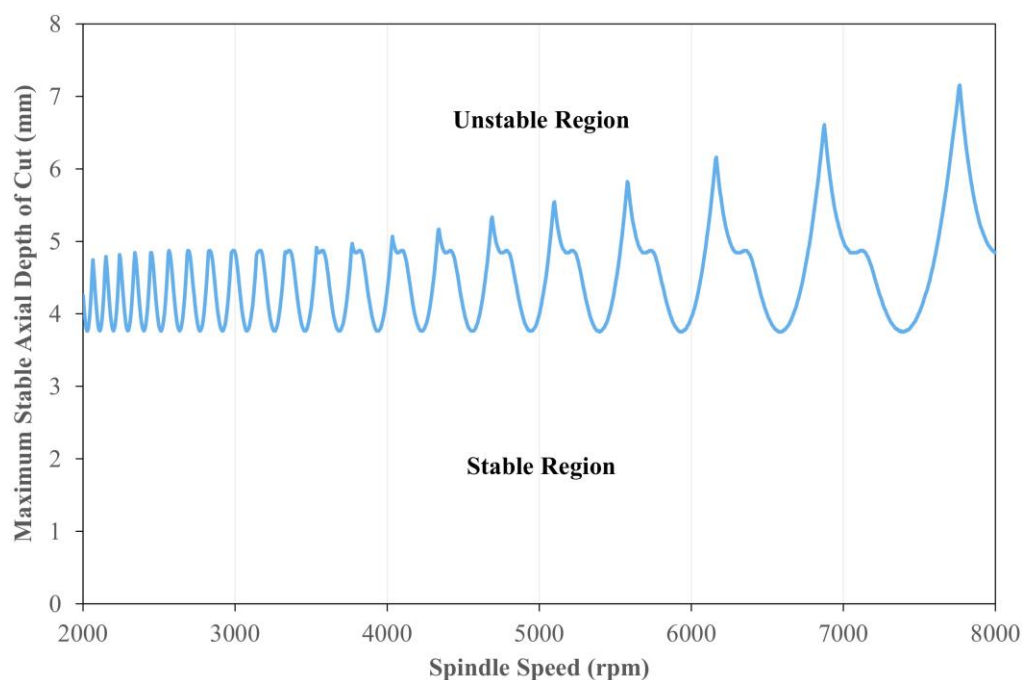


Figure 6-7. Stability lobe diagram for cutting 2014 aluminium with a 10mm 4-flute end mill.

The stability analysis shows that, for this combination of material and cutter, stable machining can easily be achieved over the simulated range of parameters. The appearance of the stability lobes towards the middle and higher end of the

speed range, where neighbouring lobes appear to cut into each other, indicates the influence of at least two mode frequencies on the stability of the tool. The stable baseline axial depth of cut is just below 4mm, meaning that cutting will be stable (chatter-free) at all spindle speeds as long as this cutting depth is not exceeded. From this baseline, it is clear that the cut profile that was used to evaluate cutting coefficients (1mm radial depth of cut, and 2mm axial depth of cut) will also enable chatter-free milling of the cantilever workpieces, since it is significantly below the threshold that causes chatter. The data reveals a peak in cutting stability at 6170rpm, which is very near to the resonant frequency of the flexures (6150rpm, which corresponds to excitation at 410Hz), meaning the tool will be close to optimal stability when machining at the resonant frequency of the flexures.

The stability analysis was experimentally verified by preliminary machining trials, in which 2mm thick cantilevers were milled at various combinations of spindle speed and depth of cut, in order to check whether the analysis correctly predicted stability of machining conditions. Using 1mm radial depth of cut, 12 configurations were machined: spindle speeds of 6150rpm, 6300rpm, 6600rpm, and 6900rpm, at axial depths of cut of 3.5mm, 4.5mm, and 5.5mm. These parameters were chosen because they represent a mix of stable and unstable conditions, as shown in Figure 6-8, with the three configurations shown in red expected to exhibit chatter.

The blue line in Figure 6-8 indicates the maximum depth of cut that can be achieved before the onset of chatter, sometimes referred to as the *critical depth of cut*. Therefore markers corresponding to machining configurations where depth of cut is beyond the stable threshold (above the line) indicate configurations where chatter is expected. Below the threshold, machining should prove stable. This was found to be true in all cases, with the plot correctly predicting the stability of all 12 configurations, thus verifying the accuracy of the stability analysis. Machining at 6150rpm and 6900rpm was found to be chatter-free at all three depths of cut, since those spindle speeds coincide with peaks of stable cutting depth. For the two middle speeds, the greatest depth of

cut proved unstable in both cases, with the intermediate cutting depth also found to cause chatter at 6600rpm, since these three configurations were above the stability threshold. Machining at the lowest depth of cut (3.5mm) was found to be stable in all cases, since this falls below the stable baseline of 3.8mm.

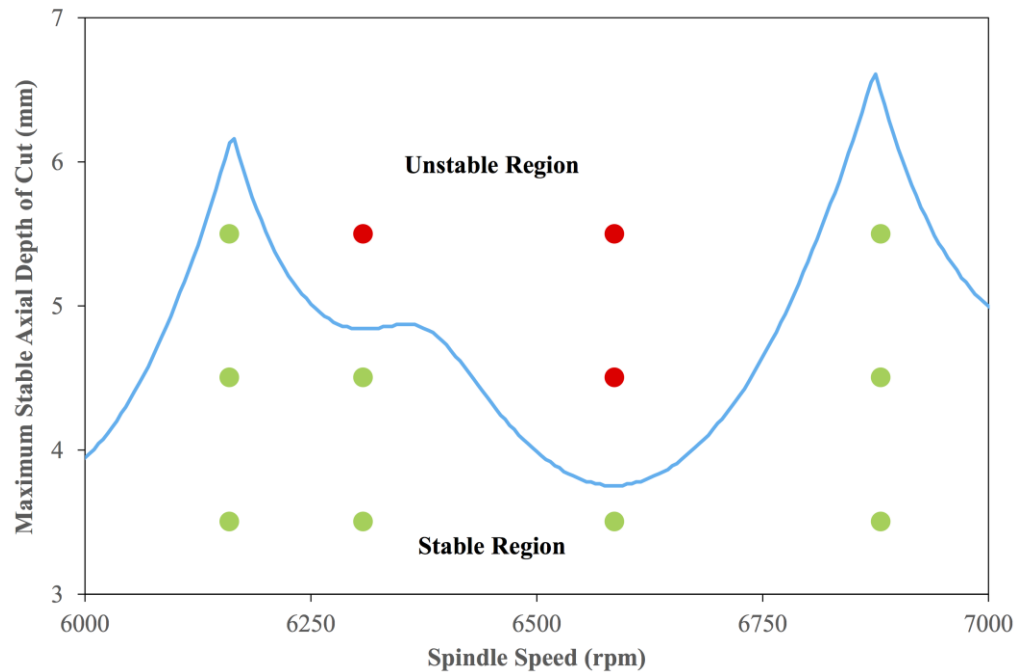


Figure 6-8. Magnified region of stability lobe diagram, showing the configurations (as dots) machined in order to verify the stability prediction. Red markers indicate unstable configurations where tool chatter was present, with green markers indicating stable machining conditions.

This machining stability analysis shows that chatter should not be an issue at any spindle speed for axial depths of cut below 3.8mm. Therefore the cantilever workpieces can be machined in the additive fixtures at any speed without tool instability, so long as the simulated stable cutting parameters are maintained.

6.3.2 FE models

In order to determine the spindle speeds of greatest interest for machining, and to predict the relative response of the three fixture systems, the milling setup was replicated in FE. The fixture blocks had been modelled previously, and simulations of entire machining process were built by developing upon the existing model. The cantilever workpiece was created using the Abaqus part sketcher, and meshed following a convergence study with the same C3D8R

brick elements used for the fixture blocks. Material property definitions for the components are given in Table 6-5. The first (preliminary) stage of the model involved applying the static clamping load to the fixture assembly (*static, general* analysis step), and defining contact interactions between the workpiece and fixture surfaces. This clamping recreated the conditions of the fixture blocks and workpiece clamped in the jig inside the machine and, in the case of the flexure blocks, compressed the flexures so that the workpiece was resting against the locators. Models of the machining assembly, which comprised two identical fixture blocks and a cantilever workpiece, are shown in Figure 6-9. To preserve the accurate predictive capability of the previous flexure models, mesh density and quality in the deformable region of the flexures was very high (16 elements across the flexure width, with an aspect ratio of below 1.01).

Table 6-5. Material properties for the printed fixture blocks and cantilever workpiece.

	2014 Aluminium	ABS
Density (kg/m³)	2780	1040
Young's Modulus (GPa)	74	2.064
Poisson's Ratio	0.33	0.35
Yield Stress (MPa)	108	31.13

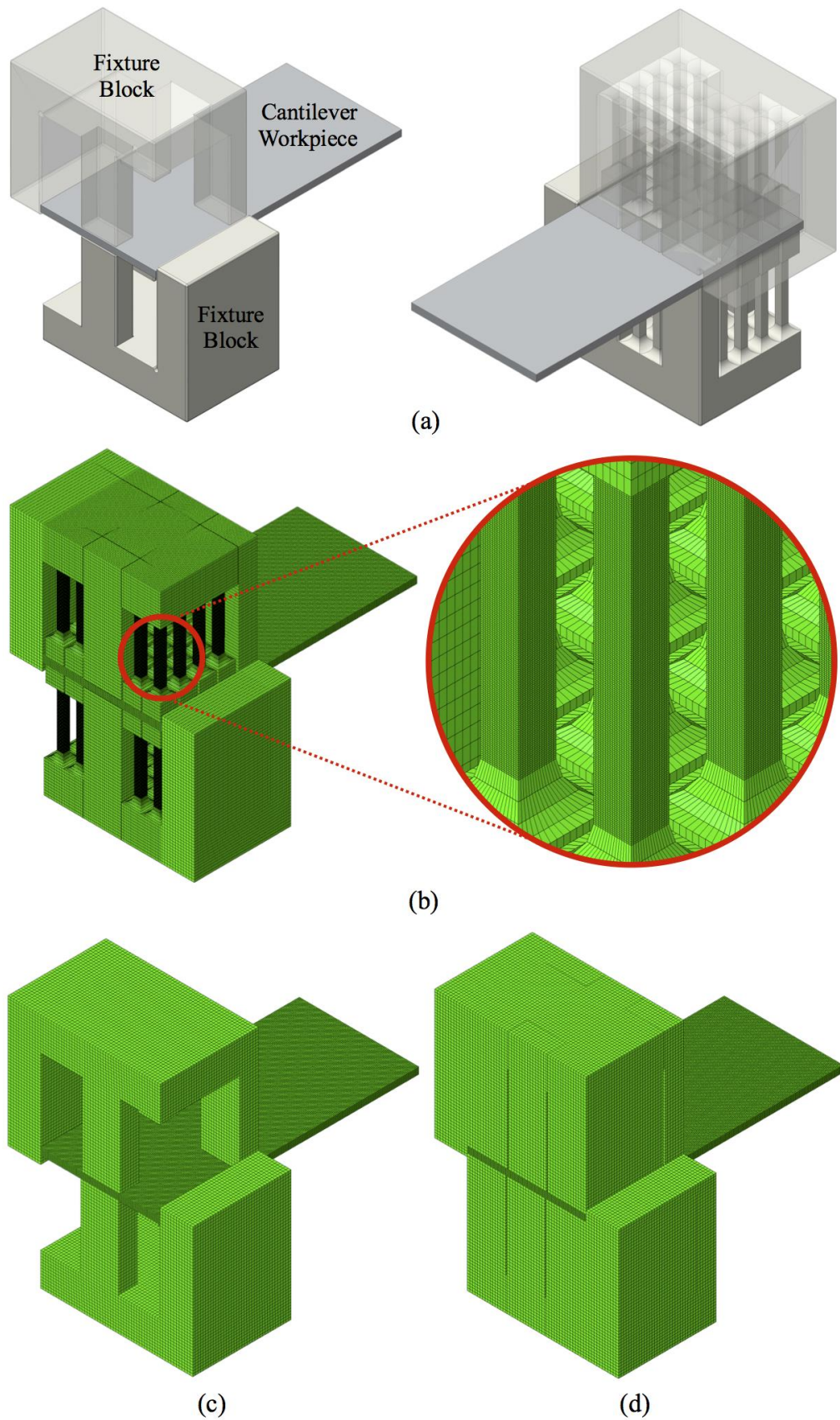


Figure 6-9. Assemblies of the machining configuration for a cantilever workpiece held by additive fixtures: illustration from CAD showing how the component is held in the fixture blocks (a); FE model of the flexure fixture assembly showing detail of the flexure mesh (b); and FE models of the workpiece in the 3-point (c) and solid (d) machining fixtures.

The clamped state of the assembly was used as the starting point for subsequent stages of dynamic analysis. Firstly, a *linear perturbation, frequency* (modal analysis) step was completed to extract the first 30 eigenfrequencies, so that all dominant modes with any significant influence on the behaviour of the system were included in subsequent stages of analysis. In good agreement with the preliminary hammer modal tests conducted on the machining assembly, the natural frequency of the cantilever was shown to increase progressively from the 3-point fixture, to the flexure and solid systems. Flexures were shown to have minimal impact on workpiece frequency response, indicated by the small difference in workpiece frequency response between the flexure fixture and the two control systems.

Table 6-6. Simulated and experimental values for the first eigenfrequency of the cantilever workpiece in the three flexure systems. Values in parentheses indicate eigenfrequency shift relative to the flexure fixture system.

First Workpiece Eigenfrequency			
	3-Point	Flexure	Solid
Simulated	493Hz (-1.0%)	498Hz	511Hz (+2.6%)
Experimental	501Hz (-0.2%)	502Hz	518Hz (+3.2%)

After the modes of the system had been modelled, a harmonic analysis (*linear perturbation: steady-state dynamics, modal*) was undertaken for the three fixture systems. This analysis simulates the steady-state response of the systems due to harmonic excitation across a frequency sweep, i.e. predicts the relative response of the cantilever workpiece to different cutting excitation frequencies. Therefore, this enables identification of the excitation frequencies of greatest interest to the system, by predicting frequencies where the system is affected by resonance. Excitation was simulated at cutting excitation frequencies from 150Hz (2250rpm) to 550Hz (8250rpm), increasing in increments of 10Hz, with a point load applied to a single node at the free end of the cantilever to mimic a simplified cutting force.

From the results of the harmonic analysis it was determined, as expected, that excitation of the cantilevers at 500Hz would cause them to resonate, leading to excessive workpiece vibration. Since it is advantageous for quality of the final component to minimise workpiece vibration, it was decided that machining at this speed would be avoided. Simulation at 410Hz was observed to cause resonance of the flexures, since this excitation frequency coincided with their first eigenfrequency. Accordingly, this was identified as a significant excitation frequency for investigating the potential for resonance of flexures to elicit a damping effect on the workpiece. To enable comparison of performance between the fixture systems with and without resonance of the flexures, at least one control machining speed, which did not coincide with resonance of the flexures or workpiece, was also needed. Since the 6150rpm spindle speed, corresponding to resonance of the flexures at 410Hz, was very close to a stable peak in the stability lobe diagram, it was determined that any control speed should therefore also coincide with a stable peak in order to allow directly comparable machining conditions. Using the FE outputs and the stability prediction, it was determined that cutting at 3000rpm (200Hz excitation) and 6900rpm (460Hz excitation) would make suitable lower and upper control machining speeds. These speeds were chosen because they had both been used on cantilevers in preliminary machining trials, and found to provide stable machining conditions: 3000rpm was used to determine the cutting force coefficients of the workpiece material, and 6900rpm was used to verify the stability lobe diagram. The relative stability of the flexure resonant speed and two control speeds is illustrated in Figure 6-10. Due to increased spacing between stable lobes at higher cutting speeds, 6900rpm was also the only appropriate upper control spindle speed, since it was the sole stable peak frequency in the simulated range that was higher than the flexure resonant frequency, and did not also excite the workpiece near its resonant frequency.

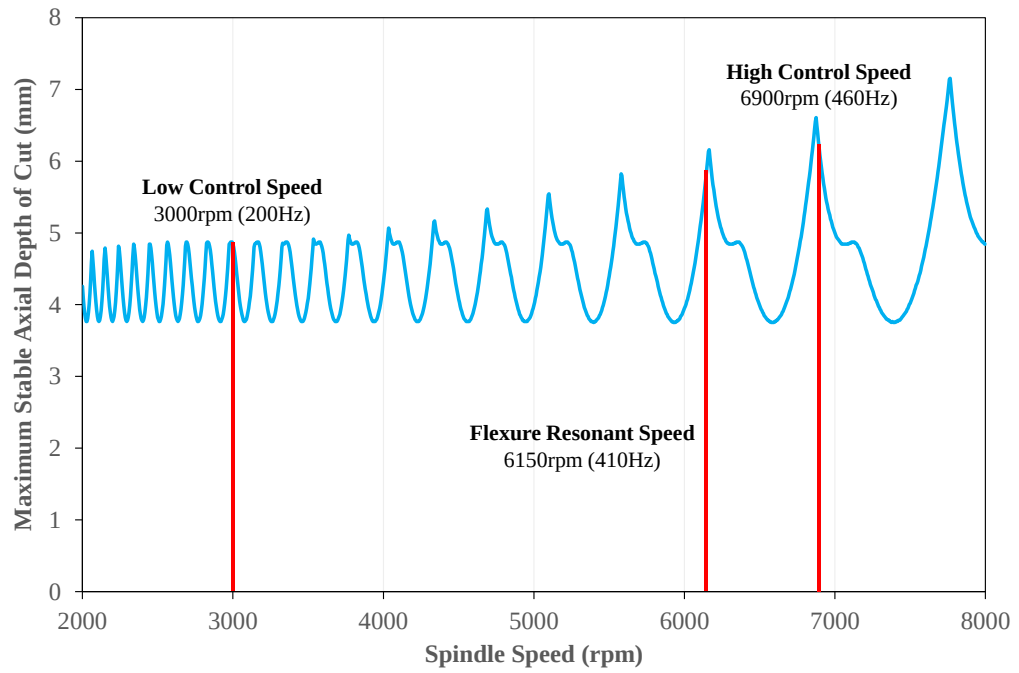


Figure 6-10. The selected machining speeds with their corresponding machining stability.

Following this study into suitable machining speeds and conditions, the model was used to simulate the milling of cantilevers in the three fixture blocks, at the three chosen speeds. This involved a transient modal dynamics simulation, by adding a *modal dynamics* analysis step to the model, to accurately predict the behaviour of the workpiece-fixture assemblies in response to time-dependent loading that recreated machining forces. To improve the accuracy of simulation, the damping ratios from Table 6-2 were included in the definition of the analysis. The displacement and acceleration outputs were requested for each time increment, but only for nodes on external surfaces of the model. Comprehensive stress, strain, displacement, velocity, acceleration and force data was requested for every node, but only for every 100th increment.

Force data from preliminary trials had revealed peak cutting force to be constant regardless of spindle speed, at approximately 6N for each cutting interaction, as long as feed per tooth was kept constant at 3 μ m. Therefore a point load, $M(t)$, was defined to act on the free end of each cantilever workpiece at the location of the start of the cut. A magnitude of 6N was applied with an amplitude defined in Abaqus to match cutting excitation at 200Hz, 410Hz, and 460Hz. The time increment was set to 10⁻⁵ seconds, to provide a high resolution simulation of

milling response. Machining-induced oscillations of the cantilever were found to stabilise within 4 cutter interactions, and therefore only the first 10 interactions for each cut were simulated. Acceleration data was exported from the location where the accelerometer was placed on the workpiece, and analysed in MATLAB. Workpiece vibration was quantified as the RMS of acceleration during stable cantilever oscillation in response to the continuous machining excitation, which therefore permitted direct comparison between the different fixture types and machining speeds.

The simulation data showed that, under control machining conditions (3000rpm and 6900rpm) where excitation does not cause the flexures to resonate, workpiece vibration was lowest in the 3-point fixture system, with the greatest vibration yielded by the solid fixture system. Performance of the flexure fixture at these machining speeds was commensurate with a conventional fixture of middling stiffness and damping properties, with workpiece vibration levels falling between those of the 3-point and solid fixture systems. However, when the workpiece was machined at the speed that was identified to cause flexures to resonate (6150rpm), the flexure fixture produced the lowest workpiece vibration of the three fixture systems. This was followed by the 3-point system, with the solid fixture still causing greatest vibration of the workpiece. This frequency-specific improvement in flexure fixture performance is attributed to resonance of the flexures, which assists with the dissipation of energy away from the workpiece. This was evident in the model, as can be seen in Figure 6-11, which shows that flexures resonate with a peak displacement, U , of approximately $5\mu\text{m}$ during machining. The location and holding capability of the fixture was shown to be unaffected by flexure resonance, since no other regions of the fixture blocks resonated, which meant the workpiece remained firmly located throughout the simulation.

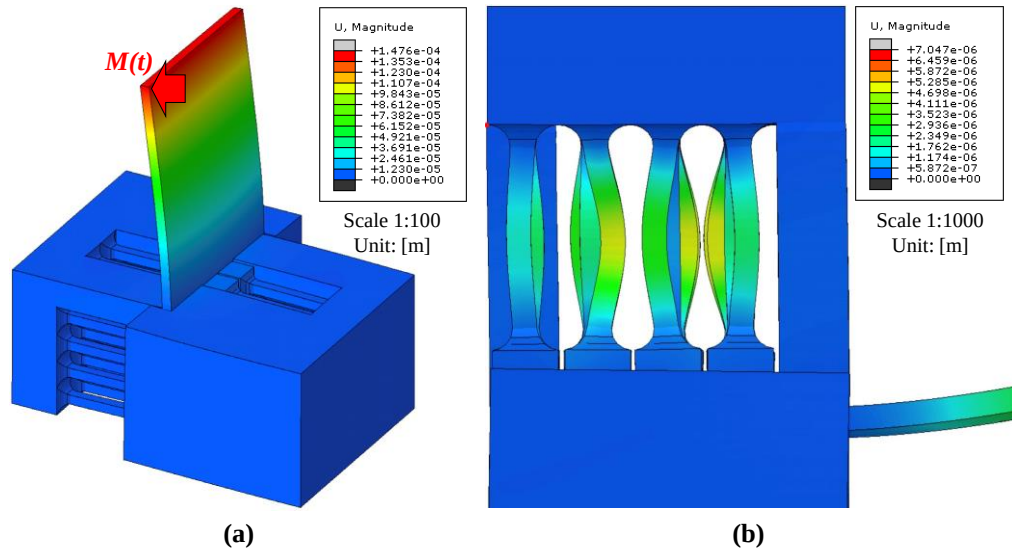


Figure 6-11. Simulated machining response of the cantilever workpiece at 6150rpm, which causes resonance of the flexures, showing an overview of the setup at peak cutting-induced displacement (a), and a close-up of the flexures during resonance (b).

The data from the control cutting conditions showed that RMS workpiece vibration in the flexure fixture consistently sat approximately half way between the levels of vibration for the two conventional fixture types. However, when cutting excitation caused resonance of the flexures, workpiece vibration in the flexure fixture was predicted to be 18% lower than in the 3-point fixture, and 25% lower than in a solid fixture. This suggests that flexures can offer a significant improvement in machining performance through damping of workpiece vibration, via a mechanism of resonance-induced energy dissipation. These predictions are discussed alongside experimental data in the following section.

6.3.3 Experimental verification

In order to verify the models, and also to assess the viability of additive fixtures for machining applications, the milling setup, which had been refined using the stability analysis and extensive modelling, was experimentally validated. A total of 9 configurations were tested: the 3-point, flexure and solid fixture systems, each at 3000rpm, 6150rpm and 6900rpm. The geometry of the workpiece, cutting feed per tooth, and dimensions of the cut were constant for each sample in order to mimic the conditions of the model as accurately as possible.

Milling was undertaken with the tool in the same tool holder, and with the same 40mm overhang used for the stability analysis, to ensure that cutting stability remained as predicted. The machine feed rate was set to 36mm/min for 3000rpm, 73.8mm/min for 6150rpm, and 82.8mm/min for 6900rpm in order to maintain 3μm feed per tooth for each cutting speed. Full machining parameters are given in Table 6-7. To achieve the intended cut depths (2mm axial and 1mm radial), and maintain them throughout cutting, each test sample workpiece was inspected prior to machining using a lever indicator in the machine spindle to ensure alignment of the workpiece with the axes of the machine. Tool positioning was reset for each sample by touching the cutter to the workpiece in each axis, to ensure an accurate and repeatable cut profile. For each test, the vibration signal from the accelerometer was sampled and windowed so that a fast Fourier transform (FFT) could be carried out to verify that the spindle was rotating at the specified speed, and therefore ensure that cutting excitation was occurring at the expected frequency. To quantify vibration levels, the signals were sampled from the middle of the cut, ignoring the first and last few seconds in order to eliminate tool entry and exit effects, with RMS vibration calculated to allow direct comparison with simulated values. For each configuration, three repeat tests were undertaken so that average vibration values could be used to ensure accuracy of the data.

Table 6-7. Machining parameters for the three machining speeds. Values are based on 2mm axial depth of cut and 1mm radial depth of cut, at 3μm feed per tooth, using a 4 tooth cutter with a diameter of 10mm.

Excitation Frequency (Hz)	Spindle Speed (rpm)	Feed Speed (mm/min)	Material Removal Rate (mm³/min)	Cutting Speed (m/min)
200 (control)	3000	36.0	72.0	94.2
410 (resonant)	6150	73.8	147.6	193.2
460 (control)	6900	82.8	165.6	216.8

Machining was undertaken successfully for all configurations, thus proving that the additive fixture concept is capable of holding a workpiece and maintaining location during machining. The printed fixture blocks showed no sign of deterioration or damage after milling of numerous samples, and prolonged periods subjected to clamping forces, therefore proving additive manufacture to be viable for the machining of these components in low-volume production.

Figure 6-12 presents RMS vibration experienced by the workpiece in each configuration, comparing simulated and experimental values for every machining speed and fixture type. The experimental data shows good agreement with the models, with the performance of the three fixtures predicted consistently. Echoing the data from FE, a clear trend emerges at control machining speeds that workpiece vibration in the flexure fixture is broadly comparable to vibration levels of the 3-point and solid fixture systems. At the flexure resonant speed, workpiece vibration is reduced in the flexure fixture, in agreement with the simulations, with the experimental results showing the damping effect to be more pronounced than the models predict.

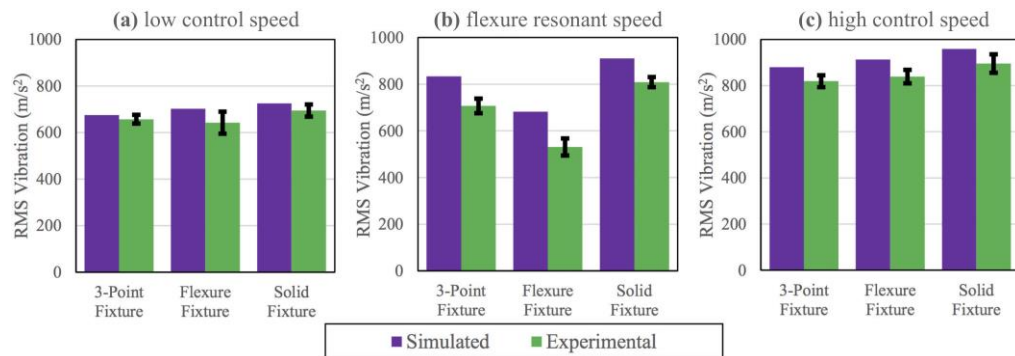


Figure 6-12. Plots of simulated and experimental workpiece vibration under the three machining fixture systems at cutter excitation frequencies of 200Hz (a); 410Hz, resulting in flexure resonance (b); and 460Hz (c).

When the flexures resonate, workpiece vibration is reduced significantly compared to the conventional fixture systems: by 25% compared to the 3-point fixture, and by 34% compared to the solid fixture. These reductions in vibration with the flexure fixture system are greater than the improvements predicted by models (by a margin of over 35%). This means the frequency-specific damping

effect of flexures is able to reduce workpiece vibration to an even greater extent than is suggested by FE.

It might be expected that the solid fixture system would prove effective in preventing workpiece vibration, due to its rigid clamping surfaces inhibiting displacement of the workpiece. This does hold true, however, for the cantilever workpiece studied in this thesis, since the solid fixture system has been found to consistently yield the highest workpiece vibration of the three fixture systems across all machining configurations. This is because the majority of the workpiece geometry is unsupported by the fixture blocks, meaning the fixture is unable to inhibit displacement at the free end of the cantilever. Instead, the low damping ratio and high stiffness of the workpiece in the solid fixture means that the workpiece-fixture system struggles to dissipate energy effectively. By contrast, the 3-point fixture system has a high damping ratio and low stiffness, which means energy is dissipated relatively easily, and explains why this system produces the lowest workpiece vibration at control machining speeds. The behaviour of the workpiece in the flexure fixture system during control conditions is in accordance with its stiffness and damping ratio: vibration levels are lower than the solid fixture, but higher than the flexure fixture. However, this behaviour changes at the spindle speed where the excitation frequency from the cutter teeth coincides with the first eigenfrequency of the flexures, thus causing them to resonate. When the flexures become activated through resonance, they prove significantly more capable than the 3-point fixture system in dissipating energy. This is because excitation energy from cutting is transferred through the workpiece to the flexures, which are dynamically coupled with the workpiece. When flexures resonate, their displacement during vibration is effective in dispersing kinetic energy into the surroundings as heat, meaning their behaviour is comparable to an external damping device, and is therefore highly effective in reducing vibration of the workpiece. When the flexures are activated by the machining excitation, they are the only region of the fixture block to resonate. The locating features in the fixture remain unaffected, which means accuracy of workpiece location is not affected by flexure resonance.

The experimental data proves that designing flexures to resonate at the frequency which corresponds to cutting excitation in a machining operation can reduce vibration in thin components by at least 25% compared to 3-point and solid fixture designs. This means the flexure fixture system, when operating at the resonant frequency of the flexures, combines the favourable workpiece holding attributes of conventional fixture types, while reducing workpiece vibration to a level that cannot be achieved in any conventional fixture system.

6.4 Summary

This chapter contains a proof of concept for an additively manufactured machining fixture based around low stiffness flexures, which is shown to be viable for holding a workpiece during an end milling operation, and capable of significantly reducing workpiece vibration compared to conventional fixture designs.

Pairs of fixture blocks, derived from the geometry shown to benefit workpiece holding in Chapter 5, were developed into complete machining fixtures. Three variants were produced, including a novel flexure-based fixture, as well as two conventional fixtures to enable relative performance in machining to be studied. Following mechanical testing of printed material, it was shown that the models used to determine optimal flexure geometry were highly accurate in predicting flexure frequency response in the additively manufactured fixture block.

A machining configuration was developed based on these fixture designs, in order to verify the suitability of additive fixtures for machining applications, and to study the vibration experienced by a low stiffness aluminium cantilever workpiece during end milling of a small slot. Modal testing of the three fixture variants while holding the workpiece was used to reveal the damping and stiffness properties of each fixture, proving the difference in frequency response between fixture variants to be small.

Prior to milling, a machining stability analysis was used to determine optimal tooling and machining parameters, to ensure that machining trials would not be

affected by tool chatter, and to allow milling to take place at optimal cutting stability. The stability lobe diagram was experimentally verified and proven to be accurate. Next a finite element model was created for the full machining assembly, and used to determine the machining speeds of significance to the system. Transient milling response was then simulated for the selected machining parameters, in order to predict the relative behaviour of the different fixture types.

Under normal cutting conditions, the models predicted that vibration experienced by the workpiece in each fixture was proportional to its stiffness and damping characteristics, with the flexure fixture producing vibration that was between the two conventional fixture types. When cutting excitation caused the flexures to resonate, however, a mechanism of frequency-specific energy dissipation was observed in the flexure fixture, which allowed the resonating flexures to act as dampers, in order to reduce workpiece vibration compared to the other fixture types.

The models were then verified with machine trials, which showed good agreement between simulated and experimental data. Experimentally, flexures were shown to reduce vibration by an even greater margin than predicted, with the flexure fixture producing 34% lower workpiece vibration than the solid fixture. Additive manufacture was therefore shown to be viable for the production of machining fixtures, and to be capable of significantly reducing workpiece vibration, making the concept ideally suited to low volume production where high accuracy is essential, such as the manufacture of legacy aerospace tooling.

Based on these findings, the novel flexure concept shows significant promise for the manufacture of vibration-sensitive workpiece geometries. Therefore the significant reductions in workpiece vibration elicited by the flexure fixture have the potential to enable the machining of geometries that cannot be milled in existing fixtures, due to excessive vibration affecting the accuracy and surface finish of the final component. The introduction of flexures is shown to have a negligible impact on dynamic properties compared to a conventional fixture,

including the frequency response of the workpiece, meaning the novel concept can therefore be introduced into a problematic manufacturing system without necessitating the alteration of existing machining parameters.

7 Discussion, final conclusions, and future work

7.1 Summary of the flexure fixture concept

The flexure fixture concept has been demonstrated to provide a significant improvement over conventional fixturing solutions in two metrics that can affect workpieces with thin, low stiffness geometries: deformation during clamping, and vibration during machining. This has the potential to significantly improve the quality of finished components, by improving their dimensional accuracy, surface finish, and strength. These advances will increase the number of components manufactured right first time, therefore reducing the costs associated with the scrapping or re-working of high value-added aerospace components.

Building upon previous work involving the support of thin geometries between locators [20], flexures are shown to be capable of holding components at significantly lower clamping forces than conventional fixtures, therefore reducing deformation of the workpiece. This is a major development in fixture technology, since flexures will potentially enable the fixturing of extremely delicate components that cannot be held in existing fixtures due to unacceptably large workpiece deformation. Expanding beyond the capabilities of reconfigurable pin fixtures [19], but without the need to control and actuate pins, the conformability demonstrated by the flexure concept also dramatically improves the ability of a fixture to accommodate workpiece geometric variability. This may help to prevent the scrapping of blanks that cannot fit in a conventional fixture, but are nonetheless capable of producing a finished component that is within tolerance. Furthermore, the damping effect yielded by eliciting resonance of flexures causes a pronounced reduction in workpiece vibration. This has the potential to enable the machining of extremely vibration-sensitive geometries, which cannot be machined conventionally due to excessive vibration.

Flexures offer a considerable advantage over other methods of damping, since attenuation of workpiece vibration is achieved passively, and without the need

for any external damping devices outside the fixture. While effective, the use of such dampers [53, 54] is not always possible, due to a lack of suitable surfaces or features on which a device can be mounted. Because the damping effect in the flexure concept is intrinsic to the geometry and material of the fixture, and specifically frequency-tuned to match the workpiece and process, it means the damper can be fully optimised for each application. This eliminates the need for external energy sources, and also the periodic maintenance required for external dampers, therefore reducing running costs of the fixture. By replacing discrete clamps with a quasi-continuous workpiece contact surface, the fixture is greatly simplified, and is able to stabilise the workpiece without introducing localised concentrations of stress.

It is anticipated that the flexure concept will prove particularly beneficial for smaller components, where space for traditional clamping and damping elements is at a premium, making any type of fixturing difficult to implement. Flexures particularly have the potential to displace highly specialised non-conventional fixturing solutions for delicate and vibration-prone workpiece geometries, such as encapsulation in low-melt alloys [23], and the suspension of the workpiece in magnetorheological fluid [57]. By supporting the workpiece with optimised supports, clamping deformation of the component is low, with resonant properties of the flexures tuned to specifically target workpiece vibration. This can be achieved while maintaining tool access, since the entire area in contact with the fixture provides a damping effect, thus eliminating the need for any additional damping.

The complexity of flexure-based fixtures, which incorporate features including freeform workpiece contact surfaces, and arrays of intricate, closely-packed flexures, means the concept lends itself ideally to additive manufacture. It would prove time-consuming and costly to manufacture a flexure fixture system by conventional means, requiring many stages of manufacture and the assembly of multiple parts. Despite a paucity of current examples, which have been limited to niche machining applications [80], additive manufacture allows the cost-effective production of entire pre-assembled fixtures in a single build, with very short lead times. Finite element models enable prediction and refinement of

fixture geometry, meaning flexures are imparted with consistent buckle response, and the desired modal properties. Use of simulation in conjunction with additive techniques means that fixtures can be printed directly from models, thus eliminating the need for multiple stages of prototyping. The future incorporation of fixture layout optimisation processes [5] represents a natural extension of the computer integrated manufacture element of the flexure concept.

A further distinct advantage of additive manufacture is ease of customisation, which encourages the use of bespoke fixturing that is fully optimised for each application. With additive techniques, each fixture can be unique without impacting on manufacturing time, while the manufacture of individual bespoke fixtures by conventional methods adds considerably to the cost of manufacturing and lead time compared to producing runs of identical fixtures. The use of additive fixtures printed on demand for legacy components has the potential to change approaches to legacy tooling. In many cases it would prove cheaper to print fixtures when needed, as opposed to storing large inventories in factories and repair facilities, and the use of additive manufacture would allow fixturing to be manufactured much more quickly than by conventional processes.

The flexure fixture concept therefore represents an extension to n-2-1 fixturing principles. It builds upon established techniques, and increases their capability for processing some of the most delicate and intricate workpiece geometries, by maximising use of the previously unoccupied space between locating features. Its potential is greatly increased through the integral use of additive technologies, and it is ideally positioned to satisfy industrial demands for increasingly complex and precise components.

7.2 Recommendations for the flexure fixture concept

The flexure concept has been demonstrated to provide significant benefits for thin and delicate workpieces in milling, however it is also potentially applicable to a wider range of workpiece geometries, and manufacturing processes. While

the greatest improvements to component quality can be achieved by combining the benefits of flexure conformability and damping, each phenomenon in isolation is nonetheless capable of improving performance compared to conventional fixtures, and therefore many processes can benefit from flexures.

The ability of the flexure fixture to conform to dimensional variations is beneficial not only for machining, but for any fixturing application where accurate location of the workpiece is required, and deformation of the workpiece has the potential to cause issues. Examples include metrology and inspection, where it is necessary to use a fixture to datum the component, but where any deformation will affect the accuracy of measurements. The high degree of workpiece support provided by flexures, and their ability to provide holding with minimal deformation, also makes them ideally suited to fixturing for welding of sheet metals, where components generally have very low stiffness, often deforming significantly under their own weight, and blanks are highly susceptible to dimensional variability. Similarly, there are numerous applications outside of machining where the frequency-specific damping effect of flexures would be highly advantageous. Passive damping is required in many fields, and frequency-tuned damper systems are especially useful due to their ability to target problematic vibration at a specific frequency, with small influence on other frequencies. Building upon these cases, the following examples and recommendations for the fixture concept are made, in order for it to be applied with maximum ease and effectiveness:

In addition to the thin workpieces presented in this work, flexures are also able to benefit components with low aspect ratio geometries. While the high aspect ratio geometries studied could only be supported by flexures from two sides due to their small section thicknesses, it is possible to support thicker components with flexures on many faces. Since freeform fixturing surfaces with flexures are easy to realise, components can even be fully encapsulated by flexures such that the only exposed regions are those where tool access is required. The geometric freedom afforded by additive manufacture enables the use of innovative fixturing approaches in order to improve machining performance, e.g. by

providing internal support to an aeroengine casing in order to stiffen its structure and provide frequency-specific damping during machining.

In its current form the workflow for the concept, which includes the design, simulation, and refinement of the flexure-based fixtures, is a manual and labour-intensive process. However a significant proportion of this process could be automated with relative ease, in order to facilitate the wider adoption and commercialisation of the concept. Code could be developed to take the workpiece geometry and requirements of the machining process, as defined by the user, and generate fixture geometries and ideal flexure properties, taking advantage of parameterised design tables in CAD, and subroutines in FE. It would also be possible to integrate established CAFD methods of determining optimal locator placement, thus eliminating trial and error in fixture design, and providing a truly optimised fixture with minimal expertise required of the end user. There is even the potential to link the fixture design process with machining stability analyses, in order to provide the user with a ‘complete package’ of optimal fixture design, tool selection, and machining parameters for each application.

The machining fixture demonstrated in this thesis comprises flexures that are only tuned to damp machining at a single frequency. However there are circumstances in which multiple excitation frequencies (cutting speeds) are employed during machining, e.g. for multiple passes at different speeds, or for machining of separate regions of the workpiece with different parameters. To fully exploit their damping capability, flexures in the fixture therefore need to be able to resonate at every frequency that affects the system. This could be achieved using *square pillar* flexures similar to those used in the printed machining fixture block, but with alternating eigenfrequencies between neighbouring flexures. Alternatively, asymmetrical *I-beam* flexures could be used, where the bending eigenfrequency in each of the orthogonal directions corresponds to one of the machining frequencies. Another possibility is if damping is required over an entire frequency range, rather than a small number of discrete frequencies. In this case a sufficient number of flexures, with

eigenfrequencies distributed across the desired range, would collectively provide a broadband damping effect.

Fused deposition modelling enabled the additive manufacture of machining fixtures in ABS plastic that were shown to be sufficiently strong, rigid, and durable for a fairly conservative milling process. It may be necessary, however, for some workpiece materials and machining processes to use more aggressive parameters, which will subject the fixture to greater loads. In such cases, fixtures may need to be of higher strength than can be achieved in plastics, and therefore metallic processes such as selective laser melting may be required to ensure accuracy and consistency of fixturing. Alternatively, if a fixture with flexures is to undergo an extended service life, used for more than just a small batch of components, then metallic processes will provide superior durability. Therefore the viability of additive processes for metallic components merits investigation, in order to allow flexures to augment the widest possible range of applications.

Although the fixture concept presented in this thesis demonstrates a considerable improvement over existing fixtures, it is nonetheless an enhancement built upon conventional fixturing techniques, as opposed to a fully reimagined approach to fixturing. As additive techniques improve, it may prove advantageous to print fixture elements directly around the component, thereby semi-permanently joining the workpiece to the fixture, and therefore blurring the distinction between the workpiece and fixture. Pre-attaching fixturing elements, such as the flexure fixtures demonstrated in this thesis, to the surfaces of a machining blank presents many potential advantages, including removal of the need for clamping forces in order to achieve location, and circumvention of re-fixturing errors during multi-stage manufacturing processes. The possibilities would become even greater if additive techniques develop the capability to produce components whose mechanical properties are indistinguishable from those manufactured by casting or forging, thus enabling direct printing of high-performance aerospace components. As with those conventional processes, components would likely still undergo a series of finishing machining operations in order to ensure consistency, dimensional

accuracy and surface integrity. To facilitate this, locating and supporting features would be printed as part of the component, which could then be easily removed after machining. As with the printing of fixtures onto a conventionally manufactured workpiece, a major advantage of direct attachment of fixtures to the surfaces of the component is that each fixture becomes truly bespoke, and therefore eliminates issues relating to poor workpiece-fixture fit.

Combining the conformability and machining proofs of concept presented in this work, and implementing the capabilities discussed in this subsection, therefore has the potential to greatly improve fixture technology. The application of additive techniques in conjunction with computer integrated manufacture enables improvements to the fixturing of virtually any component geometry. This has the potential to enable holding and machining of workpiece geometries that would be impossible to process in conventional fixtures, due to any combination of dimensional variability, low stiffness and strength, and susceptibility to vibration. Current work on the flexure concept, along with the future steps required to develop it into a viable multipurpose fixturing solution, are presented in Figure 7-1.

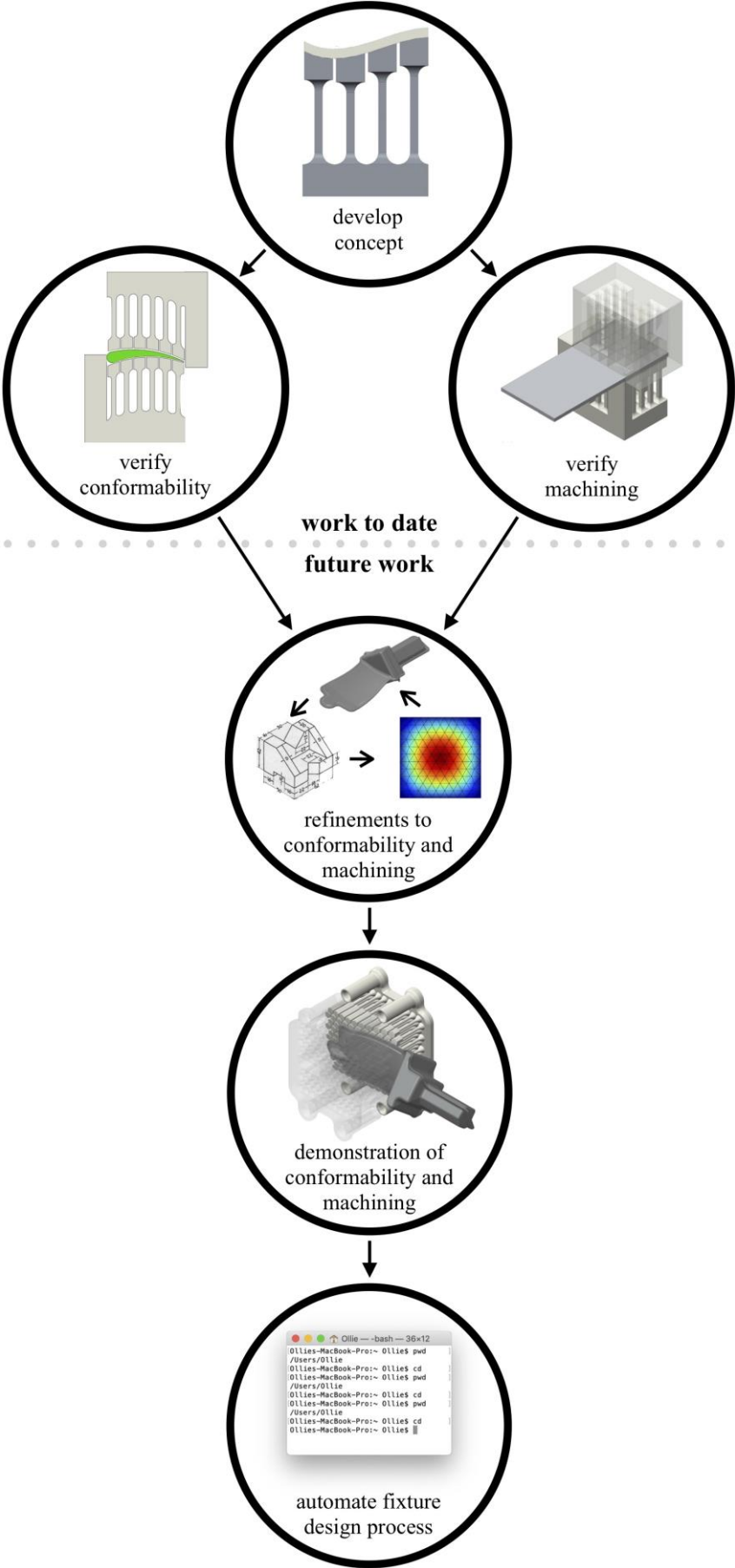


Figure 7-1. Flow chart showing proposed future development of the fixture concept.

7.3 Conclusions and key findings of the thesis

In this research, novel flexure geometries were designed, tuned, and modelled, for the design of machining fixtures that improve holding for low stiffness workpieces. The work was driven by the need to improve the accuracy of delicate aerospace components by reducing the deformation and vibration they experience during manufacture. Existing fixtures must compromise between prevention of deformation and vibration, and do not compensate for geometric variability of the workpiece blanks. The full motivation for this research is detailed in Chapter 1.

Chapter 2 contains an extensive and detailed literature review, covering fixturing techniques, vibration phenomena, machining of low stiffness components, additive manufacture, and passive damping systems. While extensive research has been undertaken to reduce workpiece deformation through optimisation of fixture layout, little attention is paid to the fixturing of components with pre-existing distortions. Many examples of workpiece damping solutions exist in the literature, however these typically take the form of external devices, and are therefore not integrated within the fixture. Additionally, there are few works on the tuning of structural frequency response, and very limited examples of fixtures produced by additive manufacture.

It is followed by Chapter 3, which presents the experimental methods, equipment, and setups used to undertake the work presented in this thesis. The research performed in this work starts in Chapter 4, where the flexure concept is introduced, alongside preliminary experiments and models examining the effects of tuning flexure geometry in a fixture. Following this, two attributes of flexure-based fixtures were investigated: conformability, to reduce clamping deformation; and frequency-specific damping, to reduce workpiece vibration.

Conformability, which takes advantage of the low stiffness and deformability of flexures to accommodate workpieces with geometric variations, is presented

in Chapter 5. Following an initial investigation using FE to determine optimal fixture geometry for uniform stiffness and modal properties between flexures, a simplified flexure-based fixture was designed and manufactured in ABS. Clamping was applied to hold aluminium workpieces with varying levels of initial distortion, with deformation experienced by the component compared between flexure and conventional fixtures.

The frequency-specific damping effect of flexures during machining is detailed in Chapter 6. Following extensive modelling, a machining experiment was designed for end milling of a simple aluminium cantilever workpiece, held using additive fixture blocks. A machining stability analysis was used to determine suitable cutting parameters, with machining undertaken using three fixture types: the novel flexure-based fixture, plus two conventional fixture systems to allow comparison of performance.

The key findings and novel contributions from each section are summarised in the following subsections.

7.3.1 Flexure design and simulation (Chapter 4)

- The effect of clamping loads on the frequency response of flexures was shown to be small, as long as deformation remains elastic and purely longitudinal. For an example flexure geometry, shift of the first eigenfrequency was 0.9% at 500N clamping, and only 4% at 2kN. This means that flexure frequency response is not clamping dependent for holding forces of the magnitude expected within a fixture.
- The frequency response of flexures was found not to be significantly influenced by the presence of neighbouring flexures. Using flexure pegs in the reconfigurable testing platform, the first eigenfrequency of individual pegs varied by no more than 2% for all peg arrangements. This means the behaviour of individual flexures remains essentially unchanged, regardless of whether a single flexure or a large array is being investigated.
- The FE models used to predict the frequency response of flexures were found to be highly accurate. Error between simulated and experimental

eigenfrequency values for the flexure pegs was below 5% in all configurations, which included individual and multiple flexure pegs, meaning the frequency response of flexure-based fixtures can be predicted with a high degree of confidence.

7.3.2 Deformation reduction in clamping (Chapter 5)

- When each flexure within a fixture requires a unique geometry in order to support distinct regions of a freeform workpiece, optimal design of flexure geometry enables all of the flexures to exhibit near-identical properties. FE models show that adjusting the blended regions of the flexure to preserve its deformable length results in less than 1% variation in buckling strength and frequency response.
- Conformability of flexures was shown to dramatically reduce the clamping forces required to fixture a complex workpiece. The flexure fixture was found to achieve holding at up to 60% lower forces compared to a conventional rigid fixture, with an average reduction of 49% across all configurations tested. Flexures were also shown to be able to hold all of the workpiece geometries without damage, while the rigid fixture system introduced plastic deformation to three of the most distorted components.
- Workpieces with geometric errors were shown to experience greatly reduced deformation during clamping with flexures. Clamping-induced deformation was up to 47% lower in the flexure fixture than in the rigid fixture system, with an average improvement of 36% observed for moderately distorted components at 500N clamping force.
- The flexure fixture system was found to be unable to fully compensate for extreme pre-existing distortion of the workpiece. While flexures were the better performing fixture system in every configuration, the improvement is diminished for the components with the most severe geometric errors. For the blades where $\varepsilon_s = \pm 1\text{mm}$, deformation in the flexure system was only 9% lower than in the rigid fixture, compared to an average reduction of 23% across the entire dataset. This is due to flexures failing to grip the surfaces of a severely distorted workpiece.

- Improvements to clamping-induced workpiece deformation in the flexure fixture system are due to the ability of the flexures to deform in order accommodate the workpiece, without significantly deforming the workpiece itself. Meanwhile the rigid fixture deforms negligibly during clamping of distorted components, meaning that deformation is borne predominantly by the workpiece.

7.3.3 Vibration damping in machining (Chapter 6)

- The FE models for prediction of flexure frequency response were found to be highly accurate for additive manufactured components. The printed flexure machining fixture block exhibited a first eigenfrequency that was within 2.5% of the design value. This error is very small for an additive component, therefore reinforcing the excellent predictive capability of flexure behaviour with the models.
- The introduction of flexures was shown to have a minimal impact on the frequency response of the workpiece. The first eigenfrequency of the cantilever workpiece in the flexure fixture was 0.2% higher than the 3-point fixture, and 3% lower than the solid fixture, with damping ratio and stiffness values between the two conventional fixtures. This means flexures can be introduced to replace an existing fixture, with negligible impact on the dynamics of a machining process.
- Additive manufacture was demonstrated to be capable of producing viable machining fixtures. The printed fixture blocks exhibited sufficient strength and durability to withstand milling of multiple cantilever workpieces, and maintained accurate workpiece location and holding throughout machining.
- Resonance of the flexures was found to create a frequency-specific damping effect that significantly reduces workpiece vibration. Machining the workpiece using parameters that excite flexures at their first eigenfrequency was shown to elicit a reduction in workpiece vibration of between 25 and 34% compared to the conventional fixtures. At control speeds, where FE models show that flexures do not resonate, the flexure system is the middle performing of the three fixtures in terms of workpiece vibration.

7.4 Future Work

In addition to the enhancements of the fixture concept suggested earlier in this chapter, the following avenues for potential future work have been identified from this research, in order to improve the quality of models, and allow comprehensive analysis of the effects of flexure-based fixtures on workpiece quality:

- 1. Combine deformation- and vibration-reducing features into a single fixture.** The two beneficial capabilities of the flexure fixture concept are investigated separately in this thesis. To realise the maximum potential of flexure-based fixtures for improved quality of thin components, both phenomena should be exploited simultaneously within the fixture. The compounded effect of reduced deformation and vibration on the quality of components with geometric errors therefore merits investigation.
- 2. Refine contact and friction definitions for more accurate simulation of fixture conformability for components with large distortions.** The models used to simulate clamping proved reasonably accurate for moderate geometric errors in the workpiece, however they struggled to fully predict behaviour in components with the greatest initial distortions. This could potentially be addressed by investigating custom contact definitions, with friction properties that are contact pressure and/or slip rate dependent, in order to improve the accuracy of conformability models.
- 3. Re-orient flexures to reduce slipping phenomena during clamping.** The conformability study revealed a tendency for flexures to slip laterally across workpiece surfaces, instead of gripping them, when the flexure contact surface is at an oblique angle to the flexure column. The phenomenon is compounded for distorted workpiece geometries, where workpiece contact surfaces are not tangential to the fixture. This could be alleviated by orienting each flexure so that it is normal to the workpiece, thus reducing the likelihood of flexure slipping.
- 4. Further inspection and testing of machined components.** Workpiece vibration directly affects surface finish and dimensional accuracy, and is therefore a useful indicator of component quality. However, further study of

machined components may provide greater insights into the difference between flexure and conventional fixture systems. Metrics that could be investigated include dimensional accuracy and surface integrity of machined features, plus the strength and fatigue life of the finished component.

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