

Active suspension structure for micro-scale probing systems

Khalid Alblalaih, BSc MSc

Thesis submitted to the University of Nottingham for the degree of
Doctor of Philosophy

January 2016

Abstract

The continued trends of product miniaturisation and increased part complexity have led to a requirement for highly accurate coordinate metrology, suitable for small parts. Coordinate metrology of such parts is performed on a micro-coordinate measurement machine (micro-CMMs), for which a specialised micro-scale probing system is required. These probing systems consist of a probe onto which a stylus is mounted. The probe provides significant flexibility allowing the stylus tip to be easily deflected during contact with a surface. Achieving an optimum stiffness for the probe represents a significant design challenge, and often leads to undesirable compromises. For example, as stylus tip size reduces the contact pressure for a given load increases, requiring the probe stiffness to be kept as low as possible to prevent damage to the part surface; however, for a more robust probing system the stiffness should be increased.

This thesis presents an improved tactile micro-probing system that makes use of an active suspension structure that can be tuned to have either low or high stiffness as required for each phase of a measurement. Development of the probe includes analytical and numerical modelling for a range of solutions as well as empirical investigations into the manufacture of a smart suspension structure for a prototype probing system. Modelling results demonstrate significant stiffness reduction is possible by using the concept of adjusting the internal strain of suspension beam elements. In principle stiffness may be reduced down to zero at the point of beam buckling. It is also shown that such a probing system can provide isotropic stiffness for a range of different styli lengths.

A prototype of the suspension structure was fabricated using a chemical etching process and 6.6 mm long stylus. The stiffness of the structure was assessed by measuring the modal frequencies of the suspension structure that correspond to vertical and lateral probe motion. Using this method, results show it is possible to reduce the frequency of the vertical mode and the torsional mode by 70 % and 33 %, respectively. Using finite element analysis it is shown that this equates to a reduction in vertical and lateral stiffness to 12 % and 46 % of their initial value, respectively,

representing a ratio of the vertical to lateral stiffness of 1.7, which is close to the isotropic stiffness.

A novel control system is presented that monitors and controls stiffness, allowing easy switching between “stiff” and “flexible” modes. During switching, the stylus tip undergoes a displacement of approximately 18 μm , however, the control system is able to ensure a consistent flexible mode tip deflection to within 12 nm in the vertical axis. Combining stage errors with the probing system linearity error, the stylus tip zero offset position error and the probing system measurement repeatability, gives an estimate of a combined uncertainty for the probing system of 58 nm (coverage factor, $k = 2$), which demonstrates the potential of this innovative variable stiffness micro-scale probe system.

Publications

Alblalaih, K., Kinnell, P, Ratchev, S (2014) An evaluation of stiffness modulating techniques for a micro-CMM tactile probe. In , Conference Proceedings - 14th International Conference of the European Society for Precision Engineering and Nanotechnology, EUSPEN 2014, pp.

Alblalaih, K., Kirk, T, Lawes, S, Kinnell, P (2014) Fabrication of a smart suspension structure of micro tactile probing. In , Procedia Engineering, pp. 1164-1167.

Kinnell, PK, Bail, R, **Alblalaih, K.**, Segal, J, Ratchev, S (2012) Rapid manufacturing of micro devices with integral electrical tracks. In , Procedia Engineering, pp.734-737.

Alblalaih, K., Kinnell, P, Lawes, S (2015) Fabrication and characterisation of a novel smart suspension for micro-CMM probes, *Sensor and Actuator A-Physical*, 232, pp.368-375.

Alblalaih, K., Lawes, S, Kinnell, P (2016) Variable stiffness probing systems for micro-coordinate measuring machines, *Precision Engineering*, 43, pp.262-269.

Alblalaih, K., Kinnell, P., Lawes, S., Desgaches, D., Leach, R. Performance Assessment of a New Variable Stiffness Probing System for Micro-CMMs. *Sensors* **2016**, 16, 492.

Acknowledgements

First of all I would like to express my sincere thanks to my supervisors Dr Peter Kinnell (Loughborough University) and Dr Simon Lawes for providing me excellent advice and sharing their experience and support throughout my work. I would also like to thank everyone who has assisted me in carrying out this project.

I wish to thank King Abdulaziz City for Science and Technology (KACST) and the Saudi Arabian Cultural Bureau in London for the scholarship provided to allow this work.

Also, I would like to thank Mr Jagpal Singh and Dr Jon Petzing at Loughborough University for providing metrology facilities to complete the experiments. I would also like to thank Dr James Claverley (National Physical Laboratory) for providing access to the nano-positioning stage.

Finally, I would like to give special thanks you to my mother *Hailah*, my father *Saleh*, and all my family.

Table of contents

Abstract.....	i
Publications.....	iii
Acknowledgements.....	iv
Table of contents	v
List of Figures.....	x
List of Tables	xxii
Nomenclature.....	xxiv
Chapter 1 Introduction	1
1.1 Background.....	1
1.2 Aim and objectives	2
1.3 Thesis outline.....	2
Chapter 2 Literature review.....	5
2.1 Background and motivation.....	5
2.1.1 Coordinate metrology.....	5
2.1.2 Anatomy of CMMs	8
2.1.3 Challenge of measuring the micro-parts	9
2.2 Summary of current state-of-the-art in micro-CMMs	11
2.3 Current state-of-the-art in micro-CMMs probes.....	17
2.3.1 Miniaturized 3D probing system.....	17
2.3.1.1 METAS probing system.....	18
2.3.1.2 NPL probing system.....	18
2.3.1.3 Triskelion probe from IBS Precision Engineering.....	20
2.3.1.4 Probing system by Hefei University of Technology.....	20
2.3.2 MEMS-based probing system	22
2.3.2.1 Micro-CMM probe by PTB	23
2.3.2.2 Micro-CMM probe by TUE	24
2.3.3 Opto-tactile sensor.....	26

2.3.3.1	Opto-tactile sensor by PTB	26
2.3.3.2	Opto-tactile sensor by NIST and HIT	28
2.3.4	Vibrating micro-scale probing system	30
2.3.4.1	Vibrating probe by Mitutoyo	31
2.3.4.2	Three-axis vibrating micro-CMM probe by NPL	32
2.3.4.3	Three-axis vibrating micro-CMM probe by Ilmenau University of Technology.....	33
2.3.5	Summary of current state-of-the-art in micro-CMMs probes	35
2.4	Methods for variable stiffness suspension structure	38
2.4.1	Tuning stiffness element by using shape memory alloy elements	38
2.4.2	Tuning stiffness element by modifying the effective geometry of the structure	38
2.4.3	Tuning stiffness element by adjusting the internal strain.....	39
2.4.3.1	Joule heating effect	39
2.4.3.2	Effects of compressive axial preload	41
2.4.4	Tuning stiffness by applying additional force to either augment or reduce the action of external loads	42
2.4.5	Summary of stiffness modulating techniques	44
2.5	Literature review summary and identification of knowledge gaps	46
2.5.1	There are no examples of variable stiffness probing systems	47
2.5.2	There is no easy way to minimise contact-force anisotropy of the probing system without being limited by stylus length.....	47
2.5.3	There are no established methods for operating an active suspension micro-probing system	48
Chapter 3	Research Methodology.....	49
3.1	Research approach	49
3.1.1	Develop a realistic concept for applying variable stiffness.....	51
3.1.2	Investigate into suitable stiffness modification strategies that may be employed in such a probe	51
3.1.3	Investigate the ability to achieve isotropic probing force without being limited by stylus length.....	52
3.1.4	Investigate a method of manufacturing for a prototype probe system....	53
3.1.5	Compare a numerical solution with experimental result in order to validate the simulation, and understand the limits of the manufacturing.	53

3.1.6 Develop a strategy that allows the control system switch between stiff and flexible stiffness	54
3.2 Chapter summary.....	54
Chapter 4 Methods for varying probe stiffness.....	56
4.1 Evaluation of a range of methods to manipulate the vertical stiffness of the probing system.....	57
4.1.1 Methodology	57
4.1.2 Thermal actuator	60
4.1.3 Piezoelectric actuator	63
4.1.4 Electrostatic actuator	65
4.1.5 Electromagnetic actuator.....	68
4.2 Discussion.....	71
4.3 Chapter summary.....	73
Chapter 5 Design of a variable stiffness suspension structure.....	74
5.1 Suspension structure performance considerations.....	74
5.2 Analytical performance model	76
5.2.1 Vertical and Lateral Stiffness of the Tri-beam design	76
5.2.1.1 Tri-beams vertical stiffness	77
5.2.1.2 Tri-beams lateral stiffness	78
5.2.2 Vertical and Lateral Stiffness of the Delta-beams design	84
5.2.2.1 Delta-beams vertical stiffness	84
5.2.2.2 Delta-beams lateral stiffness	84
5.3 Numerical modelling of the suspension structure	86
5.4 Performance analysis	90
5.4.1 Stiffness parametric study on suspension structure for tri-beam and delta-beam design	91
5.4.2 Comparison between the analytical and numerical models	95
5.4.3 Minimising the anisotropy of the probing system.....	98
5.5 Measurement principle of the suspension structure deformation	102
5.5.1 Displacement of the stylus tip in the vertical direction.....	103
5.5.2 Displacement of the stylus tip in the lateral direction.....	104
5.6 Chapter summary.....	106

Chapter 6 Embodiment, fabrication and experimental validation of a variable stiffness suspension	107
6.1 Design considerations for integrating a micro-actuator with the suspension structure	107
6.2 Manufacturing of the smart suspension structure	110
6.3 Suspension structure, evaluating variable stiffness (spring steel, AISI 1095)	115
6.3.1 Assembly of the suspension structure on the test rig	115
6.3.2 Experimental methodology for measuring stiffness of the suspension structure	117
6.3.3 Measuring vertical and torsional frequency of the suspension structure using a laser vibrometer	118
6.3.4 Measuring vertical and torsional frequency of the suspension structure using a designed test rig	119
6.3.5 Finite element simulation of suspension structure performance	121
6.3.6 Analysis of results and discussion	126
6.4 Probing system, evaluating variable stiffness (titanium, Grade 5)	128
6.5 Probe performance over time	136
6.6 Chapter summary	138
Chapter 7 Operational strategies and Performance assessment of developed micro-CMMs probe	140
7.1 The variable stiffness micro-scale probing system	140
7.1.1 Operating sequence for the variable stiffness probing system	141
7.1.2 Control strategy for the variable stiffness probing system	142
7.2 Experimental setup for the probing system characterisation	143
7.3 Suspension structure performance evaluation	146
7.3.1 Characterisation of the probing system	146
7.3.1.1 Environmental long-term stability	146
7.3.1.2 Performance in stiff mode	147
7.3.1.3 Repeatability in the stiff mode	150
7.3.1.4 Performance in flexible mode	151
7.3.1.5 Repeatability of flexible mode probing	160
7.4 Chapter summary	160
Chapter 8 Conclusion and future work	163

8.1 Key findings.....	164
8.2 Future work.....	168
References	171

List of Figures

Figure 2-1: Several complex geometry were measured using contact system, (a) a high aspect ratio microhole [10], (b) profile measurement of LED lens [10] and (c) micro-holes of an injector nozzle [26].	7
Figure 2-2: Photograph of the conventional CMM [31].	8
Figure 2-3: Schematic diagram of the probing system.	9
Figure 2-4: Photograph of the small CMM (SCMM) that was developed in the NPL [42].	12
Figure 2-5: Photograph of Isara-400 ultra-precision CMM manufactured by IBS Precision Engineering [23].	13
Figure 2-6: Schematic showing the effect of the abbe error on the measurement devices [51].	13
Figure 2-7: Photograph of the F25 micro-CMM provided Zeiss [53].	14
Figure 2-8: Photograph of nano-CMM (NMM-1) manufactured by SIOS [57].	15
Figure 2-9: (a) Photograph of the TriNano N100 Nano-CMM provided by Xpresspe [14], (b) schematic diagram that shows the working principle of the developed stage [59].	16
Figure 2-10: Image of METAS probing system [64].	18
Figure 2-11: (a) Photograph of the NPL probe [42], (b) schematic showing the main components of the NPL probe [42, 43].	19
Figure 2-12: Photograph of IBS precision engineering Triskelion probe [65].	20
Figure 2-13: Schematic diagram shows the main parts of the probing system that was developed by Hefei University of Technology [32, 66].	21

Figure 2-14: (a) Image of the probing system has been developed by Hefei University of Technology, (b) schematic diagram shows the main parts of the probing system [67, 68].	22
Figure 2-15: Photograph of the silicon based probing system that has been developed by PTB [16].	24
Figure 2-16: Image of the probing system that has been developed by Prill [74, 75].	25
Figure 2-17: Image of Gannen-XP which has been developed by Boss (TUE) [77].	26
Figure 2-18: (a) Image of 2D opto-mechanical probing system developed by PTB [81], (b) schematic diagram shows the working principle of the 3D opto-mechanical probing system [78].	27
Figure 2-19: Schematic diagram shows the working principle of the detecting system at (a) lateral direction (deflection mode) [82, 83] and (b) vertical direction (buckling mode) [84].	29
Figure 2-20: Schematic diagram shows the working principle of the detecting system of the fibre probe based on two orthogonal micro focal length collimation [3].	30
Figure 2-21: (a) Image of UMAP Vision System with exploded view inset for one dimensional vibrating probe [88], (b) schematic shows the main parts of one dimensional vibrating probe [89].	32
Figure 2-22: a) Image of the 3D vibrating microprobe by NPL [36], b) schematic diagram shows the main parts of vibrating micro-CMM probe [91].	33
Figure 2-23: (a) Photograph of triaxial microprobe, (b) schematic diagram shows working principle of the triaxial microprobe [87].	35
Figure 2-24: Schematic of modulating the stiffness by modifying the effective geometry of the structure. Applying static force (F) leads the two rigid links (L) to cause lateral deflection at the pivot of the flexural stiffness, where the distance (b) can be either increased or decreased [95].	39

Figure 2-25: Schematic illustration of integrating the thermal actuator to provide compressive force for stiffness reduction. F_y is the lateral force that is generated by the comb-drive actuator to produce a tensile stress for increasing the stiffness element.	40
Figure 2-26: Maximum potential energy (E_δ) versus stable position (δ) at different situations: (a) the compressive force is equal to zero and there is one stable position; (b) the barrier energy is less than the maximum potential energy; and (c) the barrier energy is larger than the maximum potential energy [96].	40
Figure 2-27: Experiment set up which shows the differential micrometer placed at one end of the beam to generate a compressive force [97].	41
Figure 2-28: Schematic diagram of tuning the lateral stiffness of cantilever beam using axial compressive force [98].	42
Figure 2-29: Schematic of harvesting device that used attractive electromagnetic force to decrease the stiffness at the free end of cantilever beam. The repulsive force was applied at the bottom to increase the stiffness [99].	42
Figure 2-30: Schematic of integrating the active probe membrane with atomic force microscopy. The incident laser beam was used to measure the displacement of the active membrane [100].	43
Figure 2-31: Schematic of integrating the electrostatic actuator with nanoforce sensor [101].	44
Figure 3-1: Presentation of research methodology which describes and links the main chapters.	50
Figure 4-1: Illustration of the application of a variable stiffness probing system.	57
Figure 4-2: Schematic diagram of the cross-section of a beam to demonstrate the symmetry along the z -direction.	59
Figure 4-3: (a) cross-beam and (b) tri-beam designs which consist of four and three suspension beams respectively, joined to the intermediate body. (c) rectangular-beam	

and (d) delta-beam designs consisting of eight and six suspension beams respectively connected to a larger intermediate body.	59
Figure 4-4: Schematic illustration of integrating a thermal actuator with the suspension system, where the stiffness of each beam can be modulated individually by grounding the intermediate body.	60
Figure 4-5: Modulated vertical stiffness using thermal actuator versus applied voltage.	62
Figure 4-6: Illustrates the schematic of the integration the piezoelectric actuator with suspension system.	63
Figure 4-7: Effective stiffness vs. applied voltage for integrating piezoelectric actuator with the micro probing system.	64
Figure 4-8: Shows the schematic of the integration of the electrostatic actuator with the probing system.	65
Figure 4-9: (a) Shows the schematic of the one-fourth of the model integration of the electrostatic actuator with probing system and the boundary conditions. (b) The high aspect ratio mapped mesh employed.	66
Figure 4-10: Effective stiffness and electrostatic force against applied voltage for electrostatic actuator.	67
Figure 4-11: Scanning force and effective stiffness vs. applied voltage where the over-travel distance is 7 μm	68
Figure 4-12: Shows the schematic of the integrating an electromagnetic actuator with the probing system.	69
Figure 4-13: Electromagnetic force ($F_{e,z}$) at the centre axis of the Archimedean coil versus the distance (D_o) between the coil and the permanent magnet where the applied current is 1 A.	71
Figure 5-1: (a) Tri-beam design which consists of three suspension beams, joined to the intermediate body, (b) delta-beam design consisting of six suspension beams	

connected to a larger intermediate body. Guided roller constraints are shown at the ends of each of the beams for illustration only to indicate a degree of freedom allowing axial load to be applied, with the arrows used to indicate applied load vectors.	75
Figure 5-2: Shows the boundary conditions for estimating the stiffness in the z direction that is subjected to a compressive force and vertical force at the probe tip.	77
Figure 5-3: Free body diagram for the intermediate body and the spring beams under a lateral force F_y applied at the probe tip. The free body diagram for the spring beams corresponds to the local coordinate for each beam. The shear force (f_z), bending (M_ϕ), torsional moments (M_η) and axial compressive force (P_o) on the intermediate body are analysed corresponding to the global coordinate.	79
Figure 5-4: (a) Top view of the intermediate body illustrating the distance between centre of rotation (o) about x -axis and the end of spring beams. (b) Illustrating the intermediate body is pinned at the centre where a lateral in-plane displacement of the central intermediate body is zero due the symmetry about the x and y directions.	80
Figure 5-5: Shows the boundary conditions of loading the probe tip at y -direction and free body diagram for the spring 1. This spring is subjected to shear force (f_z) (as it is aligned with the lateral deflection) and bending moment (M_ϕ) about the x -axis. The analysis is considered to find the shear force (f_z) and the bending moment (M_ϕ) at the end of the beam as function of the axial compressive force (P_o).	81
Figure 5-6: Converting the torsional stiffness to the lateral stiffness of the micro-probe CMM.	83
Figure 5-7: Half-model of the delta-beams design showing the guided boundary conditions at the end of each beam, the compressive loads applied to the beams, and all the resulting forces and moments. The free body diagram for the spring beams corresponds to the local coordinate for each beam. The shear force (f_z), bending moment (M_ϕ), torsional moment (M_η) and axial compressive force (P_o) on the intermediate body are analysed corresponding to the global coordinate.	85

Figure 5-8: The main geometric parameters of the two considered design variants. (a) The tri-beam design and (b) The delta-beam design.....	87
Figure 5-9: Schematic diagram of the cross-section of a beam to demonstrate the symmetry along the z -direction.	87
Figure 5-10: Shows finite element mesh that is generated for the tri-beam design and boundary conditions at loading point where all 3 rotational degrees of freedom are considered.	89
Figure 5-11: Finite element mesh that is generated for the delta-beam design and boundary conditions at loading point where all 3 rotational degrees of freedom are considered. Each segment has two springs beam that share the same local coordinate system.....	90
Figure 5-12: Stiffness versus axial displacement determined by analytical model. (a), (b) and (c) varying thickness (t), width (w_b) and length (l_b), respectively for the tri-beam design, and (d), (e) and (f) thickness (t), width (w_b) and length (l_b), respectively, for the delta-beam design.....	93
Figure 5-13: Stiffness versus axial displacement determined by analytical model (a) A plot showing the effect of changing the radius of the intermediate body (R) for the tri-beam design. (b) and (c) show change in the distance between the centre of the intermediate body and the spring beam (O) and change in the half width of each arm (r) for the delta-beam design, respectively.	95
Figure 5-14: A plot of the vertical to lateral stiffness as a function of the axial displacement (d_o) for the tri-beam designs, with 3.8 mm stylus length, as calculated in FEA analysis (COMSOL), analytical and approximate solution.....	97
Figure 5-15: A plot of the vertical to lateral stiffness as a function of the axial displacement (d_o) for the delta-beam design, with 3.8 mm stylus length, as calculated in FEA analysis (COMSOL), analytical and approximate solution.....	98
Figure 5-16: Stiffness versus axial displacement determined by FEA for (a) the tri-beam and (b) delta-beam design. Different effective stylus lengths ($l_s = 3.8$ mm, 6.6 mm and 8 mm) have been considered.....	100

Figure 5-17: A plot of the ratio of the vertical to lateral stiffness as a function of the axial displacement (d_o) for both delta- and tri-beam designs, with 3.8 mm stylus length.....	101
Figure 5-18: Schematic view showing the position of the displacement transducer for (a) the tri-beam design and (b) the delta-beam design.....	103
Figure 5-19: Layout of the three displacement sensors used to measure the displacement of loading the probe tip at vertical direction.....	104
Figure 5-20: Layout of the three displacement sensors used to measure the displacement of loading the probe tip at (a) x -direction and (b) y -direction.....	105
Figure 6-1: Position of the three piezoelectric actuators used to apply load to the spring beams via compliant mechanisms.....	108
Figure 6-2: Schematic diagram showing the compliant mechanism which consists of two parallel beams.....	110
Figure 6-3: (a) Shows the micro-milling and erosion process to fabricate the suspension structure for the micro-tactile probe. (b) Initial prototypes of the suspension structure.	111
Figure 6-4: (a) Contour plot of measuring the thickness of the suspension structure that is manufactured based on conventional micro-milling and micro-electro-discharge-machining. (b) Shows the profile plot of the step change between the frame (A) and the intermediate body (B).....	111
Figure 6-5: Beam thickness was reduced to 100 μm using laser-ablation process..	112
Figure 6-6: (a) Photograph of the middle layer of the suspension structure which made the spring beam. The thin sections of spring beams and paddles appear to be darker in colour due to the chemical etching process. (b) The spring steel sheet has been etched to produce the top and bottom layers.	113
Figure 6-7: (a) Outline illustration of the three spring steel layers defined using chemical etching that form the smart suspension structure. (b) The assembled prototype of the smart suspension structure for 3D-probing system.	114

Figure 6-8: (a) Contour plot of measuring the form of the suspension structure that is manufactured based on chemical etching. (b) Shows the profile plot of the step change between the intermediate body (A) and the end of sensor paddle (B).....	115
Figure 6-9: Schematic drawing of the assembly of the variable stiffness micro-probing system, with exploded view inset.	116
Figure 6-10: Assembled probe fixture, with three mounted piezoelectric actuators for compressive force application, and three capacitive sensors.	117
Figure 6-11: Shows the schematic diagram of the setup used in the initial experiment to find the natural frequency.	118
Figure 6-12: Sweep the frequency experimentally from the 2200 Hz to 3400 Hz to find beam resonant frequencies using a laser vibrometer where the vertical and torsional natural frequencies are 2469 Hz and 3270 Hz, respectively.....	119
Figure 6-13: A prototype of the active suspension structure where the suspension structure was made from spring steel (AISI 1095).	120
Figure 6-14: Configuration and experimental setup.	120
Figure 6-15: A plot of the vertical and the torsional frequencies of the suspension structure against the applied voltage (DC voltage), as calculated experimentally. .	121
Figure 6-16: The main geometric parameters of the tri-beam design.....	123
Figure 6-17: Shows results for zero applied compressive load: (a) Vertical displacement resulting from an applied vertical load of 1 mN at the centre of the structure, (b) Torsional stiffness analysis of the suspension structure resulting from 6.6 $\mu\text{N m}$ at the centre of the structure.....	124
Figure 6-18: Shows mode shapes for zero applied compressive load: (a) First mode, where the suspension structure vibrates vertically, and (b) Second mode where the centre of the structure has a torsional motion.	124

Figure 6-19: Graph of modal frequency against axial displacement of the suspension structure, as calculated for symmetric and asymmetric beam in FAE analysis (COMSOL).	125
Figure 6-20: Schematic diagram of the cross-section of a beam etched using two different strategies to demonstrate non-symmetrical nature of a beam manufactured with a single-sided etch as compared to a double-sided etch.....	126
Figure 6-21: Graph of modal frequency of the suspension structure against the normalized scale at the observed minimum frequency for the vertical direction, as calculated for the asymmetric beam in experimental result and FAE analysis.....	127
Figure 6-22: The prototype of the smart suspension structure, with main structural elements labelled where the middle layer (suspension structure) is made of titanium.	129
Figure 6-23: (a) Contour plot of measuring the thickness of the suspension structure (titanium part) that was sandwiched between two layers (spring steel) where the total thickness is 400 μm . (b) Shows the profile plot of the step change between the intermediate body (A) and the frame (B).	130
Figure 6-24: (a) Schematic cross-sectional diagram of the probing system. (b) The main geometric parameters of the modified stylus system.	130
Figure 6-25: Assembled probing system, with three mounted piezoelectric actuators for compressive force application, and beneath the suspension structure three capacitive sensors.	131
Figure 6-26: Shows mode shapes for zero applied compressive load: (a) The first mode ($f = 1114 \text{ Hz}$), where the full micro-probe vibrates vertically, and (b) The second mode ($f = 1423 \text{ Hz}$), where the centre of the micro-probe has a rotational motion.	132
Figure 6-27: (a) A plot of the vertical and the torsional frequency of the full micro-probe against the applied static voltage determined experimentally for 6.6 mm stylus length. (b) The modulated frequencies of the full probe as estimated by FEA analysis plotted as a function of axial displacement (d_o).	133

Figure 6-28: Modulated frequencies of the full micro-probe, estimated by FEA and plotted as a function of axial displacement (d_o). Three conditions are considered: manufactured geometry, manufactured geometry with gravity and a structure with optimised geometry to support a balanced load.	135
Figure 6-29: Modulated stiffness of the full micro-probe, estimated by FEA and plotted as a function of axial displacement (d_o). Two conditions are considered: manufactured geometry and a structure with optimised geometry to support a balanced load.	135
Figure 6-30: Output displacement from capacitive sensors against the displacement of the stylus tip in z direction.	136
Figure 6-31: (a) Schematic cross-sectional diagram of the probing system with stylus length 5.6 mm and 300 μm ruby tip, (b) a modified probing system, with three mounted piezoelectric actuators for compressive force application, and beneath the suspension structure three capacitive sensors.	137
Figure 6-32: Output displacement from three capacitive sensors against the displacement of the stylus tip in z direction for modified probing system.	138
Figure 7-1: Simulated stiffness and frequency against the axial displacement showing the stiff and the flexible mode of the modified probing system, as calculated FEA analysis.	141
Figure 7-2: Block diagram of tuned stiffness for two modes (stiff and flexible). Note that DC_c and DC_a are the controlled voltage and amplified voltage, respectively. ...	143
Figure 7-3: Schema of the setup for characterisation of the micro-probing system in z direction.	144
Figure 7-4: (a) An image of the experimental setup used for testing the micro-probe in the vertical direction, (b) actuator configured for lateral displacements, (c) actuator configured for vertical displacements.	145
Figure 7-5: Temperature within the isolation box containing equipment setup 80 hours.	147

Figure 7-6: A plot of the magnitude of the stylus tip drift and relative humidity within the isolation enclosure over an eighty hour period. The steepest gradient area is indicated with an overlaid circle; in this region the rate of change of stylus tip position reaches a maximum of 6 nm per minute.	147
Figure 7-7: Schema showing (a) a front view of the platform that was used to calibrate the probing system in z axis, (b) a top view of the platform that indicate the position of three capacitive sensors and the calibration directions of the probing system in x and y axes.	148
Figure 7-8: Output voltage from three capacitive sensors against the displacement of the probe tip in each of the four displacement ranges.	149
Figure 7-9: Bar charts showing repeatability in stiff mode for three nominal displacements (4 μm , 8 μm and 14 μm). (a) Shows results for the z direction, (b) shows results for the y direction.	151
Figure 7-10: Frequency plot illustrating how actuator drift reduces following a series of actuator load cycles.	152
Figure 7-11: Frequency plot showing the magnitude of the stylus tip drift as a result of the impact of actuator creep at the flexible mode (600 Hz). This drift at the stylus tip and the tuned frequency in Figure 7-10 were measured at the same time.	153
Figure 7-12: Output voltage from three capacitive sensors against the displacement of the probe tip in (a) z direction and (b) y direction, which are equal to 1 μm to generate probe calibration in vertical and lateral direction respectively.	154
Figure 7-13: Plot shows the residual against fitted value, where the maximum nonlinearity is (a) 23 nm in z direction and (b) 4 nm in y direction for full range (6 μm).	155
Figure 7-14: Plot showing the sensitivity of vertical frequency with respect to the displacement at the probe tip in the z direction.	157

Figure 7-15: Plot showing control of the position of the probe tip in three dimensions between stiff and flexible mode using resonant frequency (the frequency of the stiff and flexible mode are 1057 Hz and 600 Hz respectively).	159
---	-----

List of Tables

Table 2-1: Comparison between tactile and optical CMM measurements.	6
Table 2-2: Summary of current state-of-the-art in micro-CMMs probes. (*) Indicates the measurement range of the probing system, (+) Indicates the resolution of the sensor. (---) Indicates the value is not published anywhere	37
Table 2-3: Summary of stiffness modulating methods.	46
Table 4-1: Structure main parameters and material properties of cross-beam design used to evaluate a range of micro-actuator.	60
Table 4-2: Thermal and electrical properties of titanium (Grade 5) [107].	62
Table 4-3: Dimensions and properties of piezoelectric actuator.....	63
Table 4-4: Dimensions and properties of electrostatic actuator.	66
Table 4-5: Dimensions and properties of electromagnetic actuator.....	70
Table 4-6: Micro-actuators performance for compensating probing force of 1 mN..	73
Table 5-1: Geometric relationship of the linear and the angular displacements.....	82
Table 5-2: Structure main parameters of the tri-beam and the delta-beam design (define all terms).	88
Table 5-3: Material properties of the two configuration designs (titanium, Grade 5) [119].	88
Table 5-4: Comparison between theoretical and finite element analysis (COMSOL) results for the tri-beams and delta-beam design.....	97
Table 5-5: Finite element analysis results of isotropic stiffness for the tri-beams and delta-beam design.....	102
Table 6-1: Parameters that was set of using LaserTec 60 HSC.	112

Table 6-2: Structure main parameters of the tri-beam design.....	123
Table 6-3: Material properties of the tri-beam design (spring steel, AISI 1095)....	123
Table 6-4: Comparison of the measured and the finite element predicted frequencies for the suspension structure. (*) Indicates the percentage reduction in frequency. .	128
Table 6-5: Main parameters of the modified micro-probe.....	131
Table 6-6: Material properties of the micro-probe (Renishaw, A-5000-7800) [21].	131
Table 7-1: Specification of the control system used to tune the frequency in both modes	143
Table 7-2: Sensitivity coefficients of the three capacitive sensors at the probe tip for displacement in $z(\delta_1)$ and $y(\delta_2)$ directions, before and after switching between flexible and stiff operating mode.	156
Table 7-3: Stylus tip positional repeatability following repeated stiffness mode switching, calculated based on a series of eight repeated switches from flexible mode to stiff mode and then back to flexible mode, as illustrated in Figure 7-15.....	160
Table 7-4: Repeatability of the flexible mode for the x and y directions. The measurement was repeated ten times for each displacement (nominal displacement is $4\text{ }\mu\text{m}$).	160

Nomenclature

l_b	Length of spring element (m)
w_b	Beam width of spring element (m)
t_b	Beam thickness of spring element (m)
A_b	cross-section area of spring beam (m ²)
$l_I \times w_I$	Surface area of the intermediate body for cross-beam design (m ²)
E	Young's modulus of titanium (GPa)
$\tilde{E}$	The effective Young's modulus for wide beam (GPa)
ρ	Density of titanium (kg m ⁻³)
ν	Poisson's ratio
I_x	Second moment of area of the beam with respect to x axis (m ⁴)
k_{iz}	Initial vertical stiffness (N m ⁻¹)
k_z	Modulated vertical stiffness (N m ⁻¹)
k_{iy}	Initial lateral stiffness (N m ⁻¹)
k_y	Modulated lateral stiffness (N m ⁻¹)
T	Ambient temperature (°C)
α	Coefficient of thermal expansion (K ⁻¹)
ρ_e	Electrical resistivity of titanium (Ω m)
K	Thermal conductivity of titanium (W m ⁻¹ K ⁻¹)
V	Applied voltage (V)
P_o	Axial compressive force (N)
d_o	Axial compressive displacement (m)
$w_a \times t_a$	Piezoelectric actuator cross-sectional area (m ²)
l_a	Piezoelectric actuator length (m)
n	Number of piezoelectric layers
E_3	Young's modulus of PZT-5H in axial direction (GPa)
d_{33}	Strain coefficient of PZT-5H in axial direction (m V ⁻¹)
ϵ_0	Permittivity of air (C ² N ⁻¹ m ⁻²)
D	Displacement of the suspension system due to the actuator (μ m)
D_o	Initial gap between the electrostatic actuator and substrate (μ m)
$X_{standstill}$	Over-travel distance (μ m)
$F_{e,z}$	Electromagnetic force in the z axis (N)

B_r	Remanence of the magnetic material (T)
H_z	Magnetic field along the central axis z axis ($A\ m^{-1}$)
a_o	Inner mean radius of Archimedean coil (mm)
b_o	Outer mean radius of Archimedean coil (mm)
g_o	Gap of spiral coil (mm)
A_c	Cross sectional area of the planar coil (mm^2)
N	Number of turns
ρ_{coil}	Resistivity of the coil, copper ($\Omega\ m$)
A_{PM}	Surface area of the permanent magnet (mm^2)
H	Height of the permanent magnet (mm)
F_z	Probing force at vertical direction (N)
F_y	Probing force at lateral direction (N)
P_o	Axial compressive force (N)
d_o	Axial displacement (μm)
β_o	Dimensional form of the axial force
δ_x	Displacement of the probe tip at x axis (μm)
δ_y	Displacement of the probe tip at y axis (μm)
δ_z	Displacement of the probe tip at z axis (μm)
f_z	Shear force on each beam (N)
M_θ	Applied resultant moments (N m)
M_ϕ	Bending moments (N m)
M_η	Torsional moments (N m)
ϕ	Angular displacement about x axis
η	Angular displacement about y axis
S_{po}	Sensitivity of the lateral stiffness (m^{-1})
l_s	Stylus length (mm)
l_p	Length of sensor paddles for tri-beam design (mm)
t_p	Thickness of sensor paddles for tri-beam design (mm)
f_p	Resonant frequency of the cantilevered paddles (Hz)
ΔS	Sensor sensitivity ($m\ V^{-1}$)
l_c	Distance between the centre of the rotation and the capacitive sensors (m)
V_o	Voltage reading at the initial gap distance (V)
C_y	Stiffness of the compliant mechanism in the y direction ($N\ m^{-1}$)

C_z	Stiffness of the compliant mechanism in the z direction (N m^{-1})
t_c	Beam thickness of the compliant mechanism (m)
w_c	Beam width of the compliant mechanism (m)
l_{cc}	Beam length of the compliant mechanism (m)
f_v	Vertical frequency of the probing system (Hz)
f_t	Torsional frequency of the probing system (Hz)
f_o	Initial frequency of the probing system (Hz)
f_{min}	Frequency of the probing system at turning point (Hz)
D_p	Diameter of ruby ball (μm)
D_b	Diameter of stem holder (mm)
AC	Sinusoidal load (V)
DC	Static load (V)
DC _c	Controlled voltage (V)
DC _a	Amplified voltage (V)
m_{eff}	Effective mass of the probing system (kg)
σ_k	Standard deviation of the vertical stiffness (N m^{-1})
σ_f	Standard deviation of the vertical frequency (Hz)

Chapter 1 Introduction

1.1 Background

This thesis concerns improvement in the practice of coordinate metrology. Coordinate metrology describes the measurement of a components surface form by probing along a set of known axes, most commonly the 3 Cartesian axes. Typically the coordinate metrology is performed on a coordinate measurement machine (CMM) [1]. The three dimensional measurements can then be used to control and assure the effective manufacture to tolerance of components [2]. To ensure that measurements are useful, it is necessary to optimise the measurement processes for the highest possible certainty, but this must be done in balance with a reasonable measurement cycle time.

Metrology has had to evolve with the miniaturization of measurement systems and their growing complexity. Such systems are hard to manufacture, as they incorporate complex assemblies of micro-scale 3D components; these may be formed of many sub-components manufactured using different technologies that may not be collocated. Due to increasing demands of small feature size of these micro-assemblies and the great accuracy required for their creation, the field of micro and nano-dimensional metrology has been developed [3, 4]. In recent years contact based systems for the measurement of micro-components such as micro-coordinate measurement machines (micro-CMMs) have been developed and have become commercially available, as will be described in chapter 2. At the heart of contact-based measurement systems is a mechanical contact probe used for triggering or mapping contact with a surface. The probing system is a critical part of the metrology loop, and the overall accuracy of a micro-CMM is heavily dependent on the performance of the probing system [5]. However, there are several conflicting factors limiting the quality of measurements obtainable using existing tactile micro-probe systems. When designing micro-scale tactile probes, a design trade-off must be made between the stiffness and flexibility of the probing element. Flexibility is needed to ensure low forces during contact with a sensitive part, and avoid causing damage but, the probe must also be stiff enough to overcome attractive surface forces and to ensure it is not fragile, easily damaged or sensitive to inertial loads.

1.2 Aim and objectives

To address the need for a tactile probe that is both flexible and stiff, a solution is required that can manage the compromise between the need for a flexible probe (low probing force) and a stiff probe (robust, insensitive to surface forces and inertial loads). Therefore, **the aim of this project is to develop a tactile micro-probing system that makes use of an active suspension structure to modulate the probe stiffness as required to ensure optimal stiffness conditions for each phase of a measurement process.**

The decision has been made not to consider optical CMMs within the scope of this project. Optical CMMs offer many advantages over tactile CMM, but they are limited by an inability to measure certain geometries such as deep trenches, step sidewalls and high aspect ratio holes. The differences between tactile and optical CMM measurements will be explained in more detail in chapter 2.

To achieve the aim of this research, specific objectives have been constructed and summarised as follows:

- To develop a realistic concept for applying variable stiffness.
- To investigate practical strategies that may be employed to produce a variable stiffness suspension.
- To investigate the ability to achieve isotropic probing force without being limited by stylus length through the aid of analytical modelling and computer simulation.
- To investigate manufacturing techniques that may be used to create prototype variable stiffness probing systems, and employ these in the production of test probes.
- To compare a numerical solution with experimental result in order to validate the simulation and understand the limits of the manufacturing.
- To develop a strategy that allows the control system to switch between stiff and flexible stiffness.

1.3 Thesis outline

The thesis has been constructed to have eight chapters which cover all specified objectives. Each chapter has an introduction which describes the contents, and a

chapter summary which provides the reader with the key findings of that chapter. The thesis has started with introductory chapter and organised as follows:

Chapter 2 introduces background information about coordinate metrology at beginning to provide a reader the importance of the tactile micro-CMMs probes compared to non-contact micro-CMMs. Under the background information, the challenges of measuring micro-parts are highlighted. The current state of the art in micro-CMMs and probes is reviewed and the limitation of current micro-CMM probes is presented. Published methods of achieving a variable stiffness suspension structure are presented and key research gaps are identified. The limitations of the micro-CMM probes have identified several gaps based on current understanding of the operation of contact micro-probes which are described at the end of this chapter.

Chapter 3 presents an overview of the research approach that describes activities including mathematical modelling and experimental work.

At the beginning of chapter 4, several techniques that can possibly modulate the stiffness of the suspension structure have been evaluated based on the literature review. A range of micro-actuation methods which include thermal, piezoelectric, electrostatic and electromagnetic, have been considered to compare and evaluate the feasibility of integrating these micro-actuators into the probing system.

To further develop the concept aimed in the project of modulating the stiffness of the micro-probing system, it is required to go beyond modulating the vertical stiffness as presented in chapter 4. In chapter 5, a probing system that can be modulated to have either low or high stiffness in 3D as required using axial compressive force is presented. Two examples of suspension structure are presented for comparison. Also, a range of stylus lengths is considered to achieve isotropic stiffness with the same geometric parameters. The investigation considers the performance and mechanical behaviour of the proposed sensors using analytical methods and finite element analysis using COMSOL4.3a. The chapter also covers a design and the consideration of a sensor unit to detect and measure a movement of the suspension structure during operation of the probe.

Chapter 6 presents an overview of fabrication process of the proposed variable stiffness suspension structure that may be used as part of future micro tactile probe sensors.

Following the manufacture of the suspension structure, the assembly of this part with three piezoelectric micro actuators, capacitive sensors and stylus is considered in this chapter. The concept of modulating the stiffness of the micro-probing system and achieving the isotropy of the probing force is validated in this chapter.

Chapter 7 presents the operation strategy of the developed micro-CMMs probe. A novel control system is presented that monitors and controls stiffness as a function of the resonant frequency of the probe suspension. In addition, the probing system is calibrated in both stiff and flexible modes and a number of significant probing system errors are evaluated, including linearity, repeatability, and long-term stability.

Chapter 8 highlights the key findings and entire results in the research work. Also, the suggested areas of future work are presented at the end of this chapter.

Chapter 2 Literature review

The purpose of this chapter is to provide the reader with background information about coordinate metrology and highlight the challenge of measuring micro-parts. Moreover, current state-of-the-art in micro-CMMs is reviewed in this chapter. The basic working principle of the probing system and the current generation of micro-CMM probes use a range of construction methods to produce low stiffness, low mass probe systems are presented. Following this is a summary of current state-of-the-art in micro-CMM probes defines the problem needs to be investigated.

When developing a realistic concept that enables the modulation of probing force of a micro-probing system, it is essential to investigate suitable stiffness modification strategies that may be employed in such a micro-probe. Therefore, various designs for variable stiffness have been reviewed in this chapter.

The gaps in the literature are identified based on factors limiting the quality of measurements obtainable using existing contact micro-probe systems. These gaps found in the literature will allow the aim of this research work to be defined and highlighted.

2.1 Background and motivation

A brief introduction of coordinate metrology is given in this section. This introduction highlights the importance of tactile metrology, anatomy of CMM and challenge of measuring the micro-parts using small stylus tip.

2.1.1 Coordinate metrology

The production line, increased part complexity, and the advancing capability of high-precision manufacturing technology, have all resulted in the development of highly accurate measuring device such as coordinate measuring machines (CMMs).

The CMM can be defined according to ISO 10360 as “a measuring system with the means to move a probing system and capability to determine spatial coordinates on a workpiece surface” [6]. The measurement approach of the CMM can be either contact or non-contact method.

Non-contact methods use either a scanning approach or an areal optical sensor approach to produce an image of the component or measure the height in z direction as function of x and y axis [7]. Because optical CMMs capture an aerial image and do not require contact with the part, so they can move quickly and take the measurement with high speed [8]. However, there are some factors limiting the quality of measurements obtainable using existing non-contact systems as illustrated in Table 2-1. These limits include transparent materials, or geometries with surfaces not accessible by light; for example, optical measuring devices face difficulties when measuring complex geometries such as three dimensional measurements of high aspect ratio features, micro-lenses, sphere diameter and edge radius [9, 10].

Table 2-1: Comparison between tactile and optical CMM measurements.

Features	Measurement approach	
	Non-contact measurement based on 3D optical profilometer	Tactile system
Maximum permissible error (PME)	$\approx 2\text{-}3$ micrometers [11, 12]	$\approx 0.1\text{-}0.25$ micrometers [13, 14] and more reliable than optical system [15]
Contact force	No exerted force on the workpiece; suitable for soft material [8]	Typical less than 1 mN [16]
Contamination	No contamination	Evidence that contamination can cause significant measurement error [17]
Measure high aspect ratios and surface with high slope	The maximum slope angle limited by numerical aperture (NA) [18] Typically imaging artefacts will be not observed for highly slope $> 30^\circ$ [19], or for narrow holes [20]	High aspect ratio stylus can access intricate features [21] e.g micro-hole
Measurement speed	Dense point cloud over an imaging area [8]. Imaging rate up to 1000 Hz [22] Areal sensor: image area for a typical arrangement 0.5 inch sensor and 20 X objective $\approx (400 \times 500) \mu\text{m}$	Scanning rates up to 1 mm s^{-1} Travel speed in three axes up to 10 mm s^{-1} [14, 23]
Imaging surface that are specular or transparent	The quality of measurements can be influenced	Unaffected
Level of operator skill required	Does not required a highly skilled worker [24]	Risk of damage to probe and objective requires a trained and experienced operator [25]

Even though tactile micro-CMMs are limited by fragility and contamination, they provide reliable measurements for a wide range of surfaces [10]. For example, Figure 2-1 (a) shows micro-CMM probe which has $80\text{ }\mu\text{m}$ diameter, is able to measure a deep hole. Also, measuring a surface with high slope is not easy task for optical metrology where it can be measured using contact system as can be seen in Figure 2-1(b). Another example which presents difficulty for the non-contact metrology is measuring micro nozzle injector as shown in Figure 2-1(c). This nozzle can be measured using tactile technology with probe tip of $74\text{ }\mu\text{m}$ diameter. Therefore, the contact micro-CMMs will be considered in this project.

To better understand the requirements of micro-CMM probe, it is necessary to consider the anatomy and operations of a CMM as described in next section.

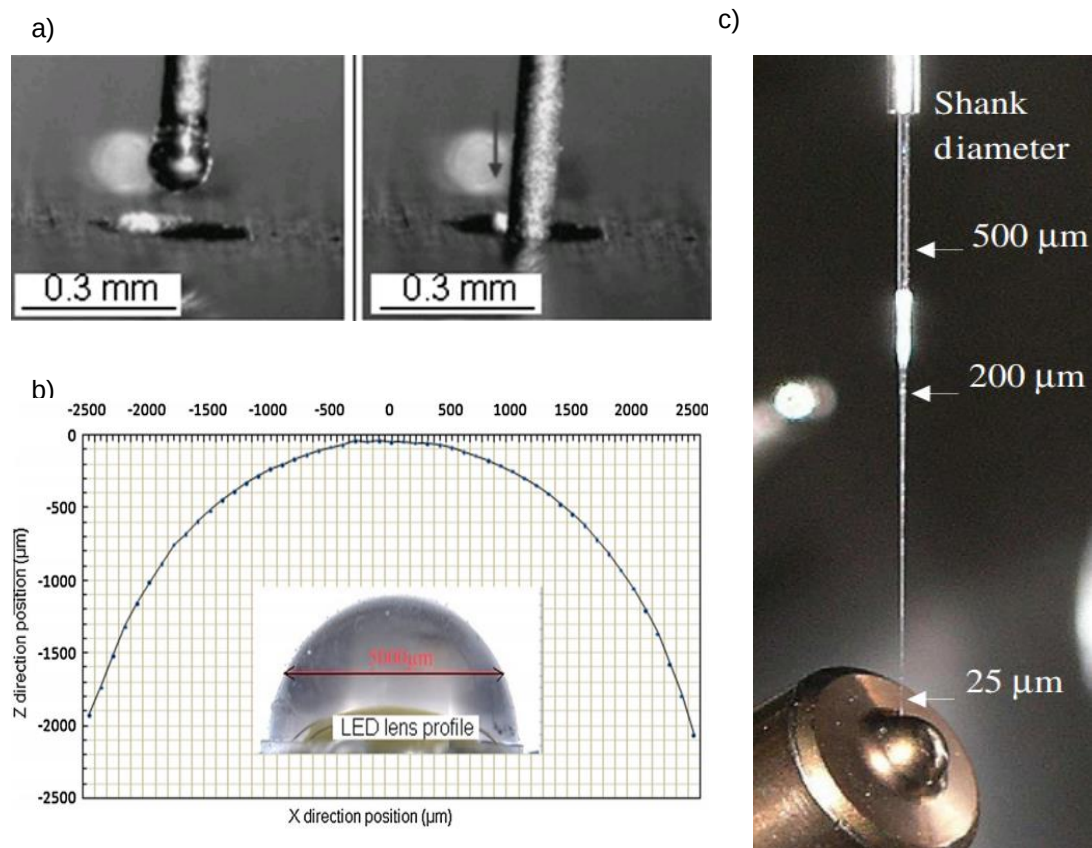


Figure 2-1: Several complex geometry were measured using contact system, (a) a high aspect ratio microhole [10], (b) profile measurement of LED lens [10] and (c) micro-holes of an injector nozzle [26].

2.1.2 Anatomy of CMMs

Typical tactile CMMs consist of four main components as shown in Figure 2-2 [27]. The first component is the drive unit which brings the probing system or the workpiece to make the contact between them to take the measurement. The measuring volume can be limited by range of measuring length along three axes. The control panel is the second part that used by the operator to take the measurement manually through approaching and retracting the CMM's probe from the workpiece. The third part is the measuring software including software correction which is designed to compensate the geometric error [28]. For example, error can be produced from drive unit in three axes which a result of imperfect geometry [29, 30].

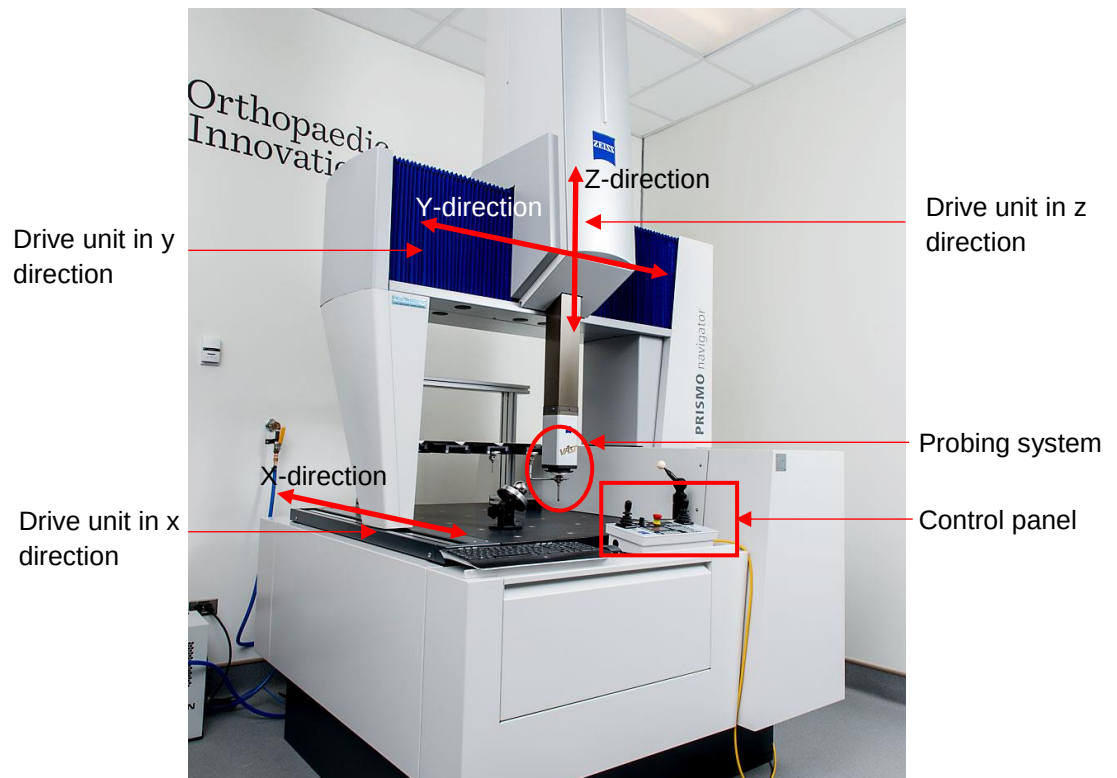


Figure 2-2: Photograph of the conventional CMM [31].

Fourthly, at the heart of both macro-CMMs and micro-CMMs is a specialised probing system that allows tactile measurement of small, delicate and intricate parts. The probing system is a critical part of the metrology loop, and the overall accuracy of a micro-CMM is heavily dependent on the performance of the probing system [5]. It is this probing system that will be the focus of this work.

Conventional micro-scale probing systems typically consist of a stylus system, suspension structure and displacement transducer [32]. A schematic representation of a conventional micro-scale probing system is shown in Figure 2-3, with the key elements annotated. During a measurement, when the stylus tip makes contact with a surface it is deflected from an equilibrium position where the contact force is transmitted to the suspension structure. The resulting suspension structure deflection is detected using displacement transducers. There are two type of probing system. The first type is a touch-trigger probe. A probe can be considered to be a touch-trigger type if the position of the stylus tip is only recorded when the probe first makes contact with workpiece [33]. The second type is known as scanning probe. By continuously monitoring a probes sensor outputs and applying an appropriate transfer function the position of the stylus tip can be known continuously while the stylus tip is in contact, therefore the probe can be dragged along the sample surface to measure the topography of the surface continuously without retraction [16]. This scanning operation allows for fast measurements, acquiring more data points in an area in a given amount of time.

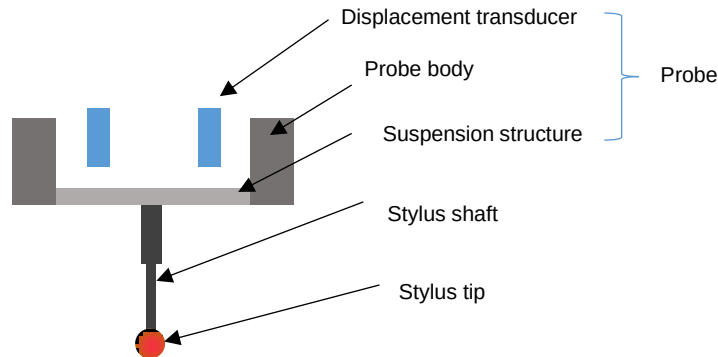


Figure 2-3: Schematic diagram of the probing system.

Now that the basic operations and elements of a CMM have been described, it is necessary to consider the specific challenges faced when measuring micro-parts.

2.1.3 Challenge of measuring the micro-parts

While the measurement principle is similar to conventional-scale probing systems, there are a number of special features and design requirements that are specific to micro-scale probing systems.

Micro-scale probing systems must impart low contact forces, typically less than 1 mN [16]. To allow intricate features to be measured, micro-scale probing systems are equipped with a slender stylus and small stylus tip. The stylus tip is typically around 100 μm in diameter; however, smaller tips are advantageous as they allow better access to intricate features [21]. A significant problem with small stylus tips is that they can result in large contact pressure. Even with contact forces of a few millinewtons, evidence of surface damage has been observed [34]. With the trend to further reduce the stylus tip diameters, this issue becomes increasingly problematic, as contact pressure is inversely proportional to the cube root of stylus tip radius squared [34, 35].

To eliminate contact-induced surface damage, two options exist. The first option is to increase probe sensitivity, and reduce the probe tip velocity. This results in lower deflections of the stylus tip during a measurement, and consequently a lower contact force is developed. The second option is to decrease the probe stiffness, leading to less contact force for a given displacement [36]. In practice, reducing probe tip velocity and, therefore, increasing measurement time is rarely an attractive scenario; hence, the second option is favoured, and stiffness is reduced as much as possible.

While low stiffness is beneficial as it reduces contact force, there are a number of negative effects that must be considered and overcome. Firstly, low stiffness probes may be sensitive to small inertial loads generated as the probe is moved by the micro-CMM. These inertial loads must not result in stylus tip displacements that are mistaken for real surface contacts. Probes that exhibit this problem require the operating speed of the micro-CMM to be reduced and, as a result, there is an unwanted increase in measurement time [36]. Secondly, as the stylus tip makes contact with a surface, relatively strong surface attraction forces occur at the contact interface [37, 38]. These surface forces may result in the probe becoming stuck to the surface which can lead to catastrophic failure of the probing system, and contamination of the workpiece. Thirdly, low stiffness structures are difficult to handle and manufacture, and can be easily damaged by shock loading.

For a given stylus tip deflection, achieving isotropic contact force requires the probe stiffness to be isotropic [39]. Isotropic stiffness means that the probing system generates the same reaction, at the stylus tip, force for all equivalent magnitudes of

displacement applied to the stylus tip [39]. Isotropic stiffness is beneficial as it reduces systematic errors during 3D measurements [37]. Also, when the ratio between the vertical and lateral stiffness is high, the effect of slip-stick can be occurred during scanning the stylus tip over the measurement surface. The stylus tip stick to the measurement surface while the scanning is running until the static friction between the stylus tip and workpiece becomes smaller than the kinetic friction. This can cause a rapid movement during the scanning which called slip-stick phenomenon [37]. Achieving isotropic stiffness is difficult due to the relatively long stylus that acts as a lever, causing stiffness in a direction coaxial to the stylus to be significantly higher than the stiffness in directions perpendicular to the stylus.

A solution is required that can manage the compromise between the need for flexible (low probing force) and stiff probes (robust, insensitive to surface forces and inertial loads). To support the needs of micro-CMMs a wide range of probing systems has been reported to understand the state-of-the-art and how those technologies handle this compromise and the broader challenge of anisotropy.

2.2 Summary of current state-of-the-art in micro-CMMs

This section of the thesis presents a review of micro-coordinate measuring systems. This section will first consider the emergence of the micro-CMM as a research instrument, and establish the current cutting edge for measurement uncertainty, then go on to consider the commercially available micro-CMM devices, and the implications that their designs might have for the development of a suitable microprobe. The first nano-CMM was developed by the University of Tokyo in 1996 [40]. The optical grating scales used on the stage give a 10 nm resolution. The straightness error of the stage was evaluated in both x and y axes and found to be around 500 nm and 600 nm. Also, the stage has a repeatability of approximately 20 nm (standard deviation). The measurement volume of the nano-CMM is 100 mm³ [41]. Following this development, the ‘NPL small CMM’ was developed in the National Physical Laboratory (NPL) in 1999 (see Figure 2-4). This CMM was designed to measure maximum length of 50 mm in three directions with 200 nanometres volumetric uncertainty (coverage factor, $k = 2$) [42, 43]. The position of the probe carrier with respect to the metrology frame was measured during the operation using multi axes interferometer [44]. This was the first micro-CMM

designed in United Kingdom [45]. Since then there have been many other research institutes that have developed micro-CMM devices, and in recent years a few commercialized instruments have come to market. The Institute of Advanced Industrial Science and Technology (AIST) in Japan has developed a system with a targeted 50 nm-measurement uncertainty and measuring range $(30 \times 30 \times 10)$ mm [46]. Also, a micro-CMM with enhanced structural accuracy has also been developed by the cooperation between Hefei University of Technology and National Taiwan University where the aimed repeatability is 30 nm and measuring range is $(25 \times 25 \times 10)$ mm [47].

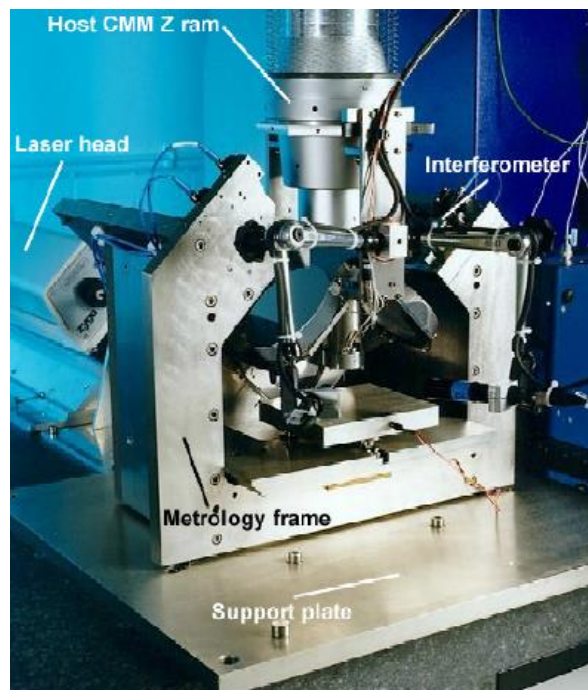


Figure 2-4: Photograph of the small CMM (SCMM) that was developed in the NPL [42].

Micro-CMM devices are now commercially available and could be a suitable platform for a variable stiffness probe. For example, the Isara-400 ultra-precision CMM was developed by IBS Precision Engineering in Netherlands – Eindhoven (see Figure 2-5). The three laser interferometers which are aligned along three linear axes and intersected at the probe tip have been utilised in this micro-CMM to overcome Abbe error [48, 49]. Figure 2-6 shows the effect of the abbe error on the measurement where that the point of interest is shifted as a result of the angular error. This angular error occurs when linear motion of a moving table or drive unit is deviated as result of imperfections in the structure.

The measuring volume is $(400 \times 400 \times 100)$ mm. The resolution of the laser interferometer in three axes is 1.6 nm and the 3D uncertainty including the probing system is 109 nm (coverage factor, $k = 2$) [23]. The probing system is modular and multiple probes can be used, and the specifics of the probing system will be described in section 2.3 [23, 50].



Figure 2-5: Photograph of Isara-400 ultra-precision CMM manufactured by IBS Precision Engineering [23].

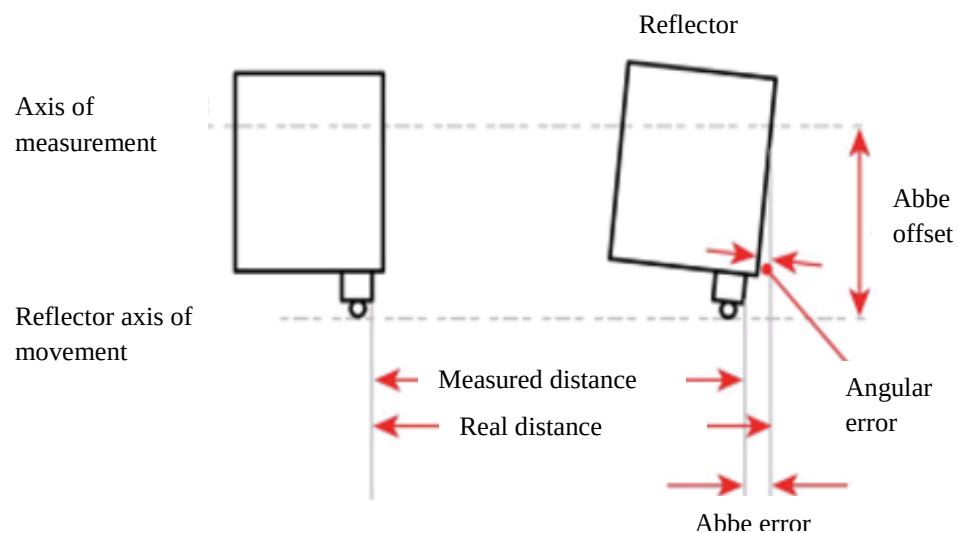


Figure 2-6: Schematic showing the effect of the abbe error on the measurement devices [51].

Another micro-CMM developed for commercial purposes, though no longer sold, is the F25 which was developed in Eindhoven University of Technology and manufactured by Carl Zeiss GmbH. This machine is capable of measuring volume of $(100 \times 100 \times 100)$ mm with the resolution in all machine axes of 7.8 nm [52, 53]. The maximum permissible error for length measurement is quoted of 250 nm [13]. The probing system for the F25 is silicon-based and has a flexible cross structure.

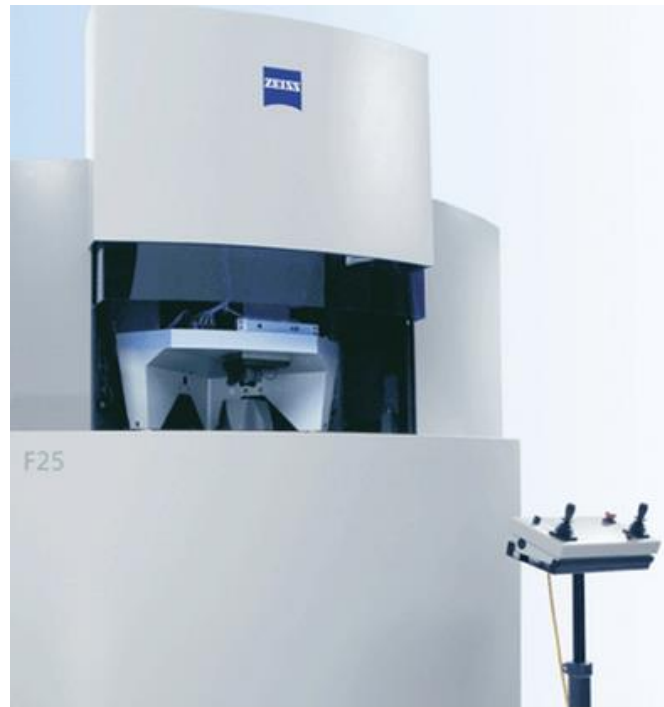


Figure 2-7: Photograph of the F25 micro-CMM provided Zeiss [53].

Further development of the CMM was achieved by Ilmenau University of Technology. This nano-CMM was manufactured by SIOS GmbH (see Figure 2-8) and has a measuring range of $(25 \times 25 \times 5)$ mm with resolution of 0.1 nm [54, 55]. It has been designed to integrate different probe system such as laser focus sensor, atomic force sensor, white light sensor and 3D-micro probe. The control looped system has been implemented to compensate the angular deviation of a guided system [56].

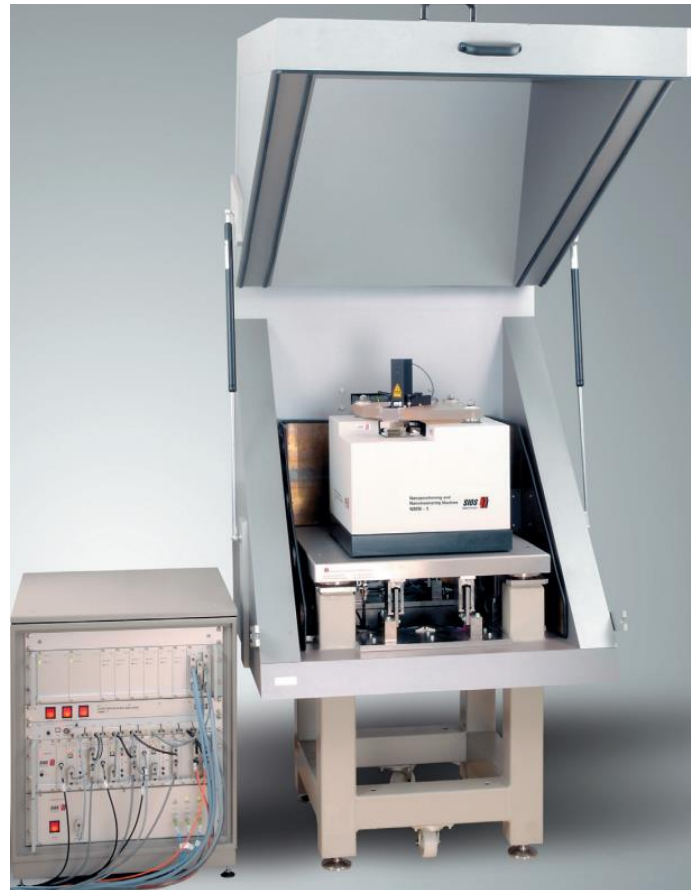


Figure 2-8: Photograph of nano-CMM (NMM-1) manufactured by SIOS [57].

Eindhoven University of Technology (TUE) in Netherlands has recently developed a CMM system that they claim is a true nano-CMM. The nano-CMM system has been commercialised with name of TriNano N100, seen in Figure 2-9(a). The main novelty of this system is that the abbe error was minimised in the three directions. Figure 2-9 (b) shows the working principle of the stage where the measurement axis of three encoders intersect into the centre of the stylus tip to satisfy the abbe offset over the measurement volume. The maximum permissible error for length measurement is quoted of 140 nm and volumetric measuring range is 200 cm³ [14, 58, 59].

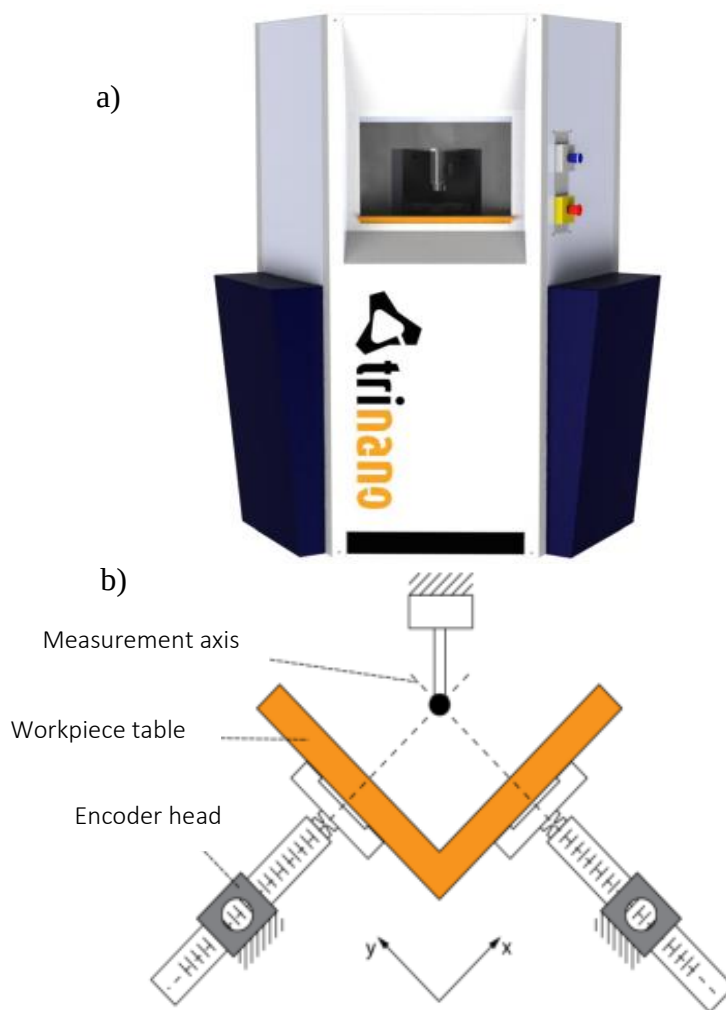


Figure 2-9: (a) Photograph of the TriNano N100 Nano-CMM provided by Xpresspe [14], (b) schematic diagram that shows the working principle of the developed stage [59].

A CMM or CMM probe is only useful if it can be calibrated to a known standard. At these length of uncertainty the calibration methods can be very involved. Calibration methods are not discussed in detail in this thesis, but the required information can be found in ISO 10360-2 series [6]. Also, more information about calibration artefacts for micro-CMMs can be found in [25].

It can be seen that there is no one golden standard instrument or approach for performing coordinate metrology at the microscale. The envelope of performance of these leading micro-CMMs systems is similar but the exact solutions used vary

significantly. The problem of achieving the best tactile probing conditions is expanded further when considering probe design. It is at the probe level that properties such as contact force, accessibility and measurement rate are defined. The next section will take a detailed look at the current state-of-the-art in micro-CMM probes.

2.3 Current state-of-the-art in micro-CMMs probes

A wide variety of tactile probing systems have been reported using different technologies to improve the performance of the micro-CMMs in terms of accuracy and low uncertainty. This section will review the current state of the art, with a particular focus on the lessons that can be learned about achieving low contact force, while maintaining robust and low uncertainty operation. The current generation of commercial micro-CMM probes use a range of construction methods to produce low stiffness, low mass probe systems. These tactile probing systems have been classified in different categories by Flack, Claverley & Leach (2014) [39]. The first category is the miniaturized probing system where the size of suspension structure was reduced to provide low probing force. The 3D tactile probing system which is based on micro electro mechanical system (MEMS) technology is considered as a second category. The third category is the opto-mechanical probing systems. The vibrating probe system, which used a change of vibrating amplitude and phase angle to detect the measurement surface, is the fourth type of the probing system. The specification and key challenges for each type of probing system will be addressed in this section.

2.3.1 Miniaturized 3D probing system

The working principle of miniaturized probing system is the same as the conventional CMM probes which was described section 2.1.2. To reduce probing force and get access to small features are the main goal behind reducing the size of the CMMs probe [35]. Various displacement transducers were used with a metallic suspension structure to measure a displacement at the stylus tip such as inductive, capacitive and optical sensor. Different designs of the miniaturized CMM probes which have been developed will be reported in this section as follow:

2.3.1.1 METAS probing system

The Federal Institute of Metrology in Switzerland (METAS) has developed a tactile probing system in cooperation with Mecartex [60]. The suspension structure was based on flexure hinges and made from aluminium as can be seen in Figure 2-10. The flexure hinge was manufactured using EDM and has thickness of 60 μm . The displacement at the stylus tip is measured using inductive sensor [61].

This system has an isotropic stiffness which is around 20 N m^{-1} and it has been designed to have three degrees of freedom at stylus tip (x , y and z). The effective mass of the probing system is around 7000 mg [60] and due to the low stiffness and high mass, a permanent magnet has been integrated with probing system to compensate the gravity effect. It has been reported that the working range, resolution and repeatability are $\pm 100 \text{ }\mu\text{m}$, less than 1 nm and 5 nm, respectively. The stylus system of METAS probe is exchangeable with probe tip diameter from 100 μm to 1000 μm [34, 62, 63].

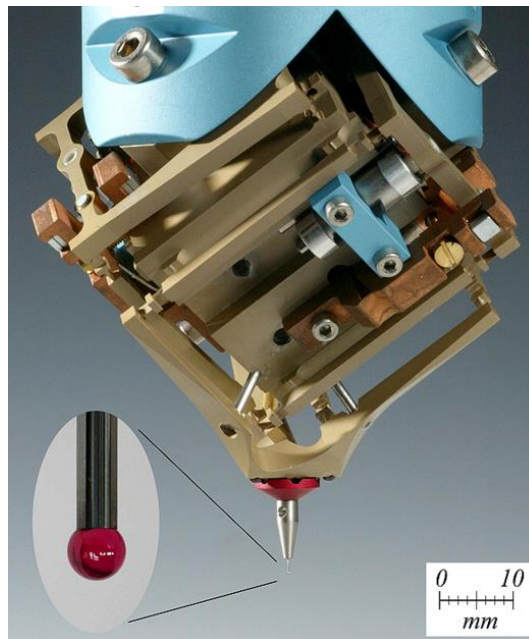


Figure 2-10: Image of METAS probing system [64].

2.3.1.2 NPL probing system

The National Physical Laboratory (NPL) in United Kingdom developed a probing system in 1999 which is based on the capacitive sensor as can be seen in Figure 2-11 (a) [44]. The three capacitive sensors were designed to measure the movement of the

suspension structure when the stylus tip made contact with workpiece. The suspension structure consists of central disc, three tungsten carbide tubes and three flexure structures made from beryllium copper sheet which have 50 μm thickness. The stylus was mounted on middle of the central disc. The main function of the three tungsten carbide tubes is to transmit the displacement from the central disc to the flexures as can be seen in Figure 2-11(a).

The overall stiffness of the probing system is approximately 10 N m^{-1} where the probing force at the stylus tip is equal in three directions. It has been reported that the effective mass of probing system is around 350 mg and the diameter of the stylus tip is 300 μm . The measuring range is based on the gap distance between the suspension structure and capacitive sensor which is about $\pm 20 \text{ }\mu\text{m}$ and has resolution of 3 nm. This probe has been integrated with small CMM and it has been reported that uniaxial standard uncertainty is around 100 nm (coverage factor, $k = 2$) [16, 42].

a)



b)

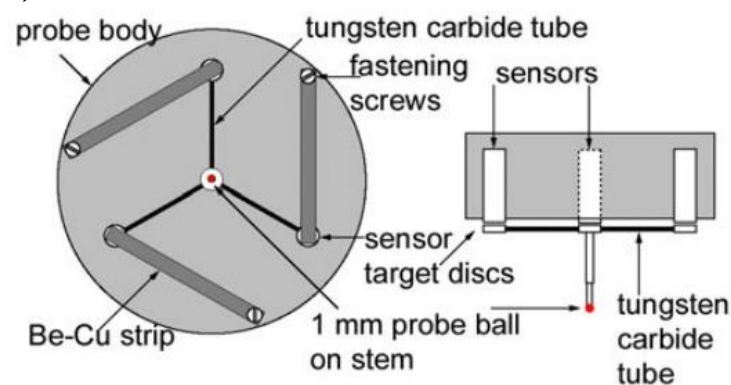


Figure 2-11: (a) Photograph of the NPL probe [42], (b) schematic showing the main components of the NPL probe [42, 43].

2.3.1.3 *Triskelion probe from IBS Precision Engineering*

The working principle of this probe is based on that of the NPL probe. However, the suspension structure is made of Invar to enhance thermal stability where spring element of the NPL probe was made of beryllium copper. IBS Triskelion probe has been integrated into the Isara-400 ultra-precision CMM [48, 49]. This probing system can work either as scanning probe or as trigger probe.

Three type of IBS Triskelion probe are available commercially with different specifications. Standard probe is the first type which has stylus length of 8.5 mm, effective mass of 160 mg, stylus tip diameter of 500 μm and isotropic stiffness (70 N m^{-1}). The measurement range and the resolution are $\pm 10 \mu\text{m}$ and 2 nm, respectively. The 3D measurement uncertainty is reported to be less than 15 nm (coverage factor, $k = 2$). The stiffness of type two and three which have stylus tip diameter of 70 μm and 1 mm, respectively are not isotropic [65]. The second was designed to measure and access small components with 6 mm stylus length. To measure a high aspect ratio features, the third type was used with 13 mm stylus length.



Figure 2-12: Photograph of IBS precision engineering Triskelion probe [65].

2.3.1.4 *Probing system by Hefei University of Technology*

Hefei University of Technology in China has developed several tactile probing systems based on optical sensor. The first probing system consists of a floating

suspension plate which has 2.5 g and suspended by four long wires to reduce the probing force; these wires have 180 μm diameter and 29 mm length, see Figure 2-13. The stylus has length of 5 mm and diameter of stylus tip is 313 μm . The probing system has anisotropic stiffness which in vertical and lateral direction is around 11 N m^{-1} and 118 N m^{-1} respectively. A laser focus probe has been used as sensor to detect the motion of the float plate and has measurement range of 5 μm . The contact force is low and found to be around 109 μN . The pre-travel distance that the probing system needs to be displaced at the stylus tip before the sensor trigger the surface was 2.5 nm. The measurement repeatability of the probing system was estimated to be around 9 nm (standard deviation) [66].

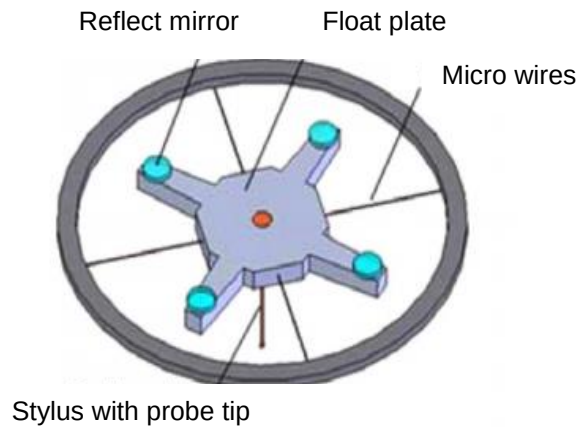


Figure 2-13: Schematic diagram shows the main parts of the probing system that was developed by Hefei University of Technology [32, 66].

A further development of the previous designed [32, 66] was achieved by Rui-Jun. The probing system consists of leaf spring, floating plate, plane mirror and stylus as shown in Figure 2-14 (a). The main function of the floating plate is to transmit the probing force from the stylus tip to the leaf spring. The deflection of the suspension structure can be measured using a laser interferometer that monitors the plane mirror. The previous design has a short measurement range (5 μm) which is limited by the DVD pick-up head sensor used to measure position. To extend the measurement range to $\pm 20 \mu\text{m}$, the DVD pick-up head has been replaced by developed miniature Michelson interferometer to detect the deformation of floating plate as result of displacing the stylus tip. Also, it has been found that the tension of suspended wire in

the previous design was difficult to control and make it uniform during the assembly. Therefore, the suspended wire has been replaced by leaf spring made from beryllium-copper alloy as can be seen in Figure 2-14 (b) [66, 67].

The stiffness at the stylus tip is isotropic and equals to 500 N m^{-1} and has total mass of 630 mg. The length and tip diameter of the stylus system, which was mounted on floating plate, are 11 mm and 1 mm, respectively. It has been reported that the repeatability and resolution of the probing system are around 30 nm and 1 nm, respectively [68].

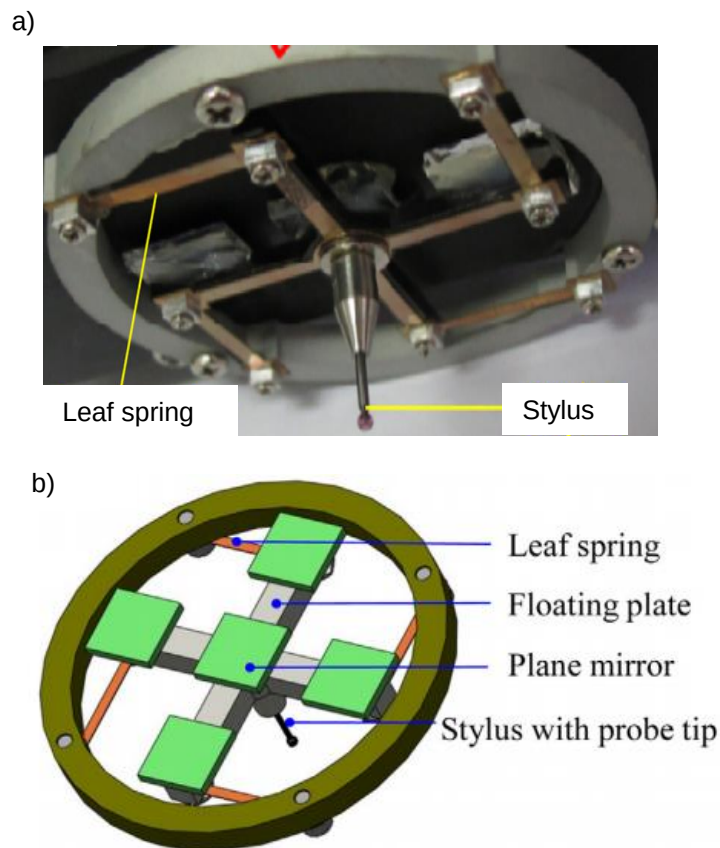


Figure 2-14: (a) Image of the probing system has been developed by Hefei University of Technology, (b) schematic diagram shows the main parts of the probing system [67, 68].

2.3.2 MEMS-based probing system

The working principle for MEMS-based CMM probe is similar to the miniaturized 3D probing system, but taken further and utilising silicon production methods for manufacturing the suspension structure. These systems utilise the integration of the

ultra-precision micro-engineering and integrated circuit to reduce the probing force, the effective mass and achieve highly sensitive detection [38].

The deflection of the stylus tip can be measured by various displacement transducers such as piezo-resistive which use deposition technique on the flexure [69]. Also, capacitive and optical sensors have been used on MEMS-based probing system where the sensor can be located above the suspension structure to measure the deflection. Different designs have been developed will be presented in this section as follow:

2.3.2.1 *Micro-CMM probe by PTB*

Physikalisch-Technische Bundesanstalt (PTB) in Germany has developed fabricated micro-CMM probes based on silicon technology. This probe system consists of membrane, boss, stylus and wafer or frame. The stylus is mounted on the boss and contacted to the frame through the flexible membrane which has thickness of 20 μm . The working principle of this probe is to detect the displacement at the stylus tip through the deformation of the membrane. The piezo-resistive is etched on the membrane to make a sensor sensitive to the motion of the stylus tip [70]. The stiffness at the stylus tip is quite high and not isotropic. The stiffness in the vertical and lateral direction is around 55000 N m^{-1} and 2500 N m^{-1} , respectively. The effective mass is not published anywhere.

The maximum working range in the vertical direction is 10 μm . The resolution of the probing system is 5 nm in vertical direction and 3 nm in lateral direction. The repeatability of the probing system in vertical direction was estimated to be 20 nm [71].

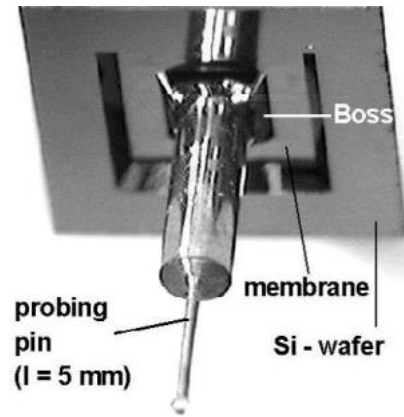


Figure 2-15: Photograph of the silicon based probing system that has been developed by PTB [16].

To reduce the stiffness ratio between the vertical to the lateral stiffness, a further investigation at PTB has been conducted. The membrane design was replaced by twin design attached in parallel. The single design has four beams where the stylus is mounted in the middle. The ratio between vertical and lateral stiffness was reduced 0.75. The stiffness at the stylus tip was measured as 5530 N m^{-1} and 7400 N m^{-1} in vertical and lateral direction, respectively [72, 73].

2.3.2.2 *Micro-CMM probe by TUE*

Eindhoven University of Technology (TUE) in Netherlands has developed several micro-CMM probes. The first micro-CMM probe has been developed by Prill [74].

The stylus system is glued on the intermediate body that is made of aluminium. Three slender rods connect to the intermediate body to the silicon chip as can be seen in Figure 2-16. The main function of the flexible three slender rods is to allow the deflection at stylus tip that is transmitted through the intermediate body. Piezo-resistive strain gauges have been deposited on three rods to measure the displacement at the stylus tip and to reduce the hysteresis [74, 75]. The stiffness at stylus tip is 18 N m^{-1} and 100 N m^{-1} in lateral and vertical direction, respectively. The effective mass is around 20 mg. The measurement range resolution, repeatability are around 20 μm , 1.2 nm and 4 nm [74].

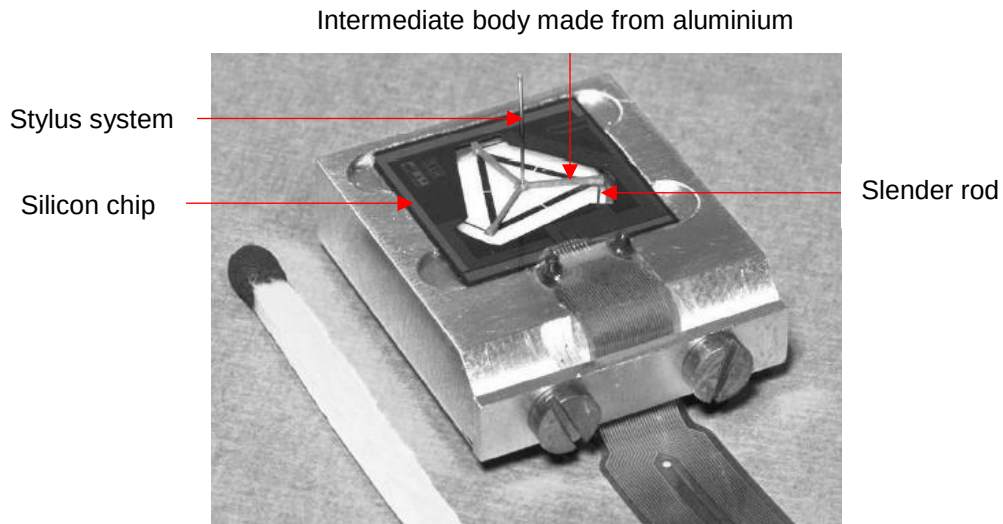


Figure 2-16: Image of the probing system that has been developed by Prill [74, 75].

Prill's micro-CMM probe has been developed by Boss [76]. This probing system is available commercially and called Gannen-XP as shown in Figure 2-17. The enhancement is that the whole suspension structure is made of silicon instead of using aluminium as intermediate body. The probing system has an isotropic stiffness which is around 450 N m^{-1} and low effective mass in both vertical (25 mg) and lateral direction (8 mg). It has been reported that the working range and the repeatability are of 5 μm and 2 nm, respectively. The resolution of the probing system is 1 nm and the hysteresis is less than 1 nm for the full working range. The combined uncertainty is reported to be 10 nm (coverage factor, $k = 2$) [76].

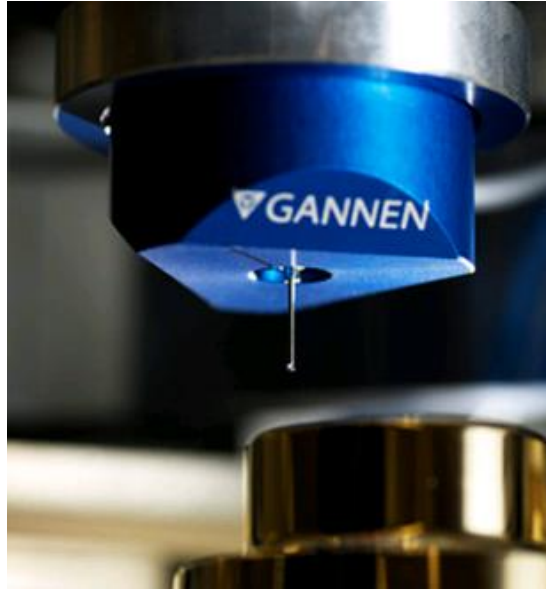


Figure 2-17: Image of Gannex-XP which has been developed by Boss (TUE) [77].

2.3.3 Opto-tactile sensor

The main advantage of opto-mechanical micro-CMM probe is that there is no suspension structure. The stylus is deflected when the probing element makes contact with work piece. This can be considered the main difference between the mechanical and opto-mechanical micro-CMM probe. In the following section the different sensor systems were designed to detect deflection of the stylus when the probing element made contact will be reported.

2.3.3.1 *Opto-tactile sensor by PTB*

PTB developed an opto-mechanical micro-CMM probe as shown in Figure 2-18 (a). The stylus is made of optical fibre to allow the light to be transmitted to illuminate the sphere of the stylus tip. To detect the deflection of the stylus tip in x-y plane, the backlight of illuminated probing sphere is monitored by camera system. The deflection the stylus tip causes a shift in the backlight position which can be measured by image processing. The sub-pixel resolution that can be achieved in lateral direction is 50 nm [78]. The diameter of the fibre and probing ball are 15 μm and 25 μm , respectively. This probing system has the ability to have contact force within range of micronewton and the probing uncertainty is fall between 0.2 μm to 0.3 μm [35, 71].

To detect the deflection in the z direction, a second camera has been integrated into the system. The arrangement of the mirror allows second camera to measure the deflection of the second target or sphere in z direction [79] as can be seen in Figure 2-18 (b). This system has been commercialised by Werth Messtechnik GmbH [80]. The stiffness data is not published anywhere, but the design appears highly anisotropic.

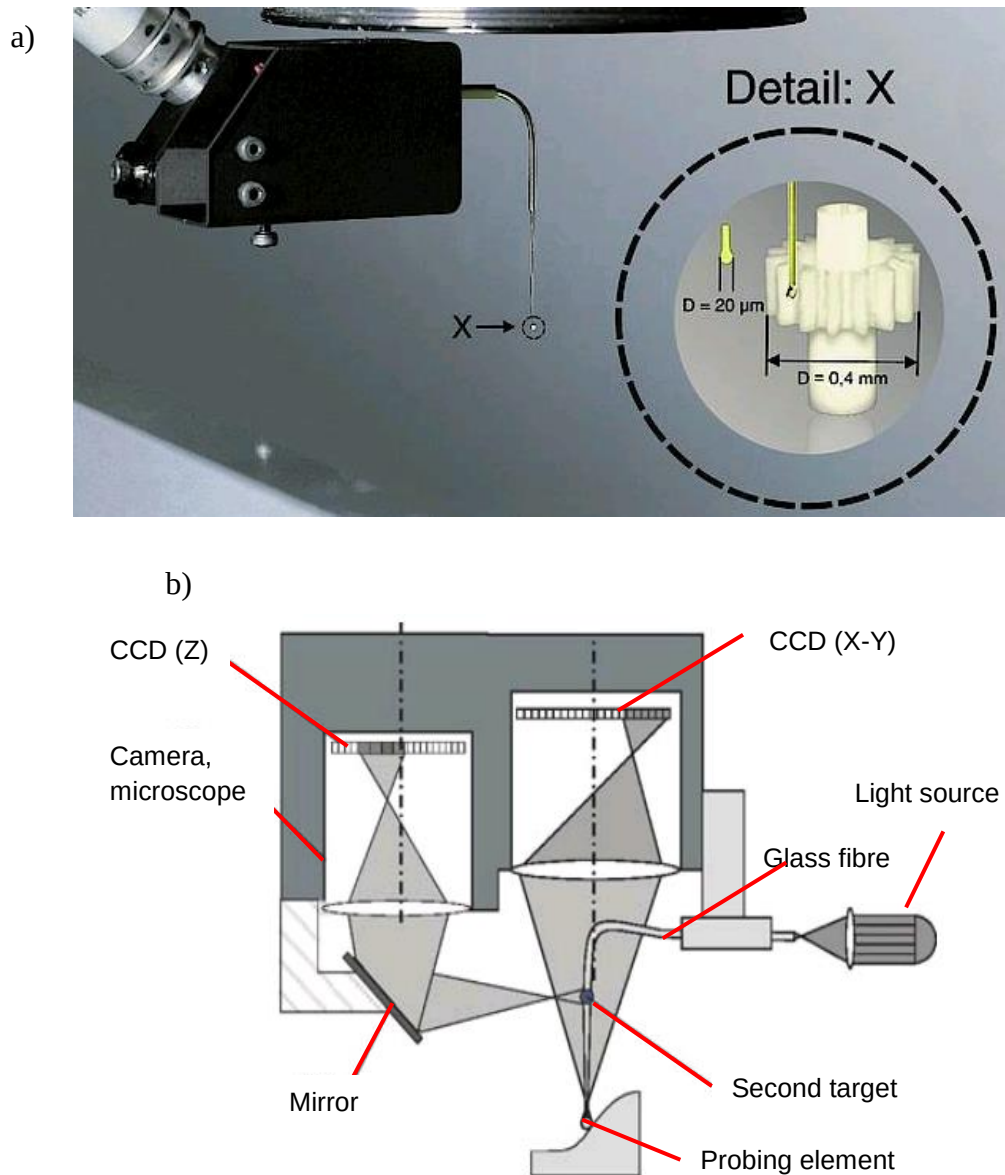


Figure 2-18: (a) Image of 2D opto-mechanical probing system developed by PTB [81], (b) schematic diagram shows the working principle of the 3D opto-mechanical probing system [78].

2.3.3.2 *Opto-tactile sensor by NIST and HIT*

The National Institute for Standards and Technology (NIST) in United States of America was developed the sensing system of the opto-tactile probe in 2004 [82]. The opto-tactile probe that was developed by PTB was limited by reflecting the light from the sidewall that is radiated by the probing sphere. This leads to reduce the ability of the photodetector to obtain enough light [3]. To solve this problem, the deflection of the stylus is measured when the probe tip makes contact at the lateral direction by focusing at the stylus itself which used as focus cylindrical lens (see Figure 2-19 (a)). To measure the position of the stylus tip in the vertical direction, the buckling modes can be utilised where the slender rod is buckled when the axial compressive force applied at the stylus tip (see Figure 2-19 (b)). The stylus has 20 mm length and the diameter of the fibre and probing ball are 50 μm and 75 μm , respectively. The contact force was estimated to be around 0.2 μN when the stylus tip is displaced by around 20 μm . The probing system was calibrated over a 15 μm and 0.3 μm range in the lateral and vertical direction, respectively. The resolution that can be achieved in lateral direction is ± 10 nm. The measurement repeatability of the probing system were reported to be around 35 nm. This probing system was examined to measure 100 μm diameter and the expanded uncertainty was estimated to be 110 nm (coverage factor, $k = 2$) [82, 83].

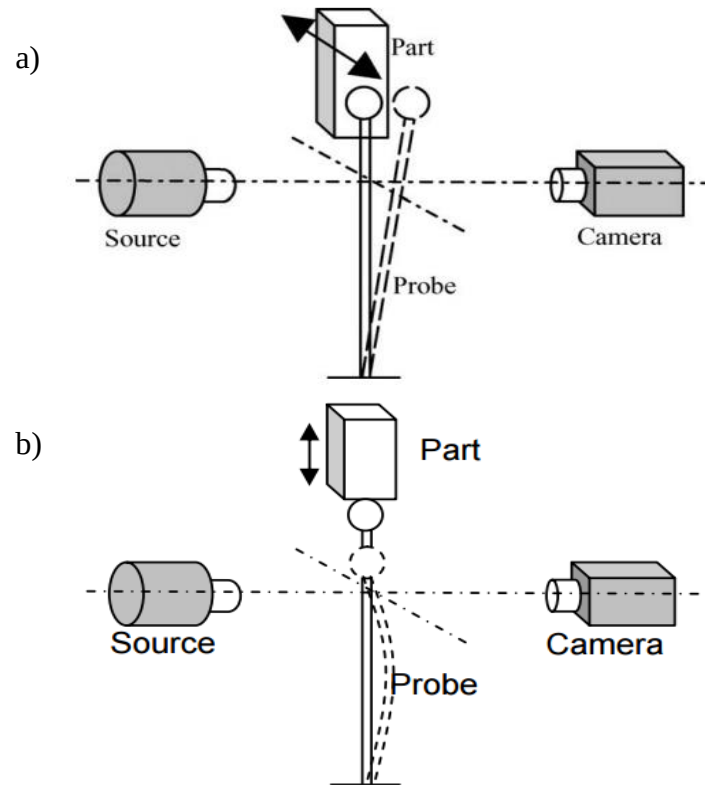


Figure 2-19: Schematic diagram shows the working principle of the detecting system at (a) lateral direction (deflection mode) [82, 83] and (b) vertical direction (buckling mode) [84].

A Further development of the sensor system for the NIST's probe was achieved by Harbin Institute of Technology (HIT). The first development was to use the fibre probe as a collimating cylindrical lens to image the light source to CCD camera, this technique helped to reduce the resolution down to 3 nm [85]. However, there is a coupling signal between the x and y axis; and the sensor system is not able to detect the displacement at vertical direction [3]. To overcome these problems, two orthogonal micro focal length collimators were used to detect the deflection of fibre stylus in three directions as illustrated in Figure 2-20. The stylus has 15 mm length and the diameter of the stylus tip is 131.52 μm . The contact force at the lateral direction or deflection mode was estimated to be less than 10 μN . The probing system was calibrated over a 5 μm and 1.5 μm range in the lateral and vertical direction, respectively. The system in the vertical direction is not linear as a result of nonlinearity buckling mode. The resolutions that can be achieved in lateral and vertical direction are 3 nm and 5 nm, respectively. This probing system was

examined to measure diameter of 638 μm and the measurement repeatability (standard deviation) was estimated to be 50 nm.

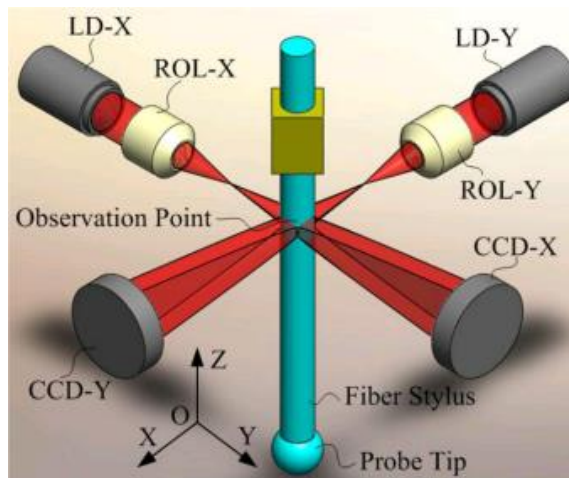


Figure 2-20: Schematic diagram shows the working principle of the detecting system of the fibre probe based on two orthogonal micro focal length collimation [3].

2.3.4 Vibrating micro-scale probing system

The most promising recent work that aims to overcome surface force problems has come from vibrating micro-scale probing systems [36, 86, 87]. The probing system is excited at its resonant frequency to vibrate the stylus tip where the interaction between the probe tip and surface force of workpiece leads to change dynamic response and therefore detecting the measurement surface without having to make full contact.

The optimal performance of the vibrating probe can be achieved when the stylus vibrates in the direction that is perpendicular to measurement surface [86]. Several single axis vibration probe systems have been developed and developing vibrating multi-axial microprobes to be sensitive in all directions is under research now. For vibrating probes the stiffness of the suspension is no longer critical for making low force and high sensitivity, as they rely on only partial contact for triggering. Instead isotropic performance requires the probe to vibrate with same frequency and amplitude in three directions [45].

The operation of these probing systems requires a complex control strategy, and the exact nature of the interaction between the vibrating stylus tip and the surface has yet to be fully described.

2.3.4.1 *Vibrating probe by Mitutoyo*

Mitutoyo has developed and commercialised a one dimensional vibrating probe in 2002 as can be seen in Figure 2-21(a) [88]. It has been designed to measure small and deep holes such as micro-holes of an injector nozzle.

The stylus has 3 mm length and 20 μm diameter. The stylus tip was made of glass and has a 30 μm diameter where the aspect ratio is about 100:1. This stylus was mounted on bulk of piezoelectric actuator. This bulk of piezoelectric actuator has been divided to driving and sensing electrode as can be seen in Figure 2-21 (b). The stylus tip is vibrated at the resonant frequency of 349 kHz. A change in the vibrational amplitude and phase of sensing electrode can be used to detect a contact measurement [89].

The repeatability of the probing system is in micrometre range and the detected system can measure down to 10 nm [87]. The main advantage is that the probing force is reduced down to micronewton [90].

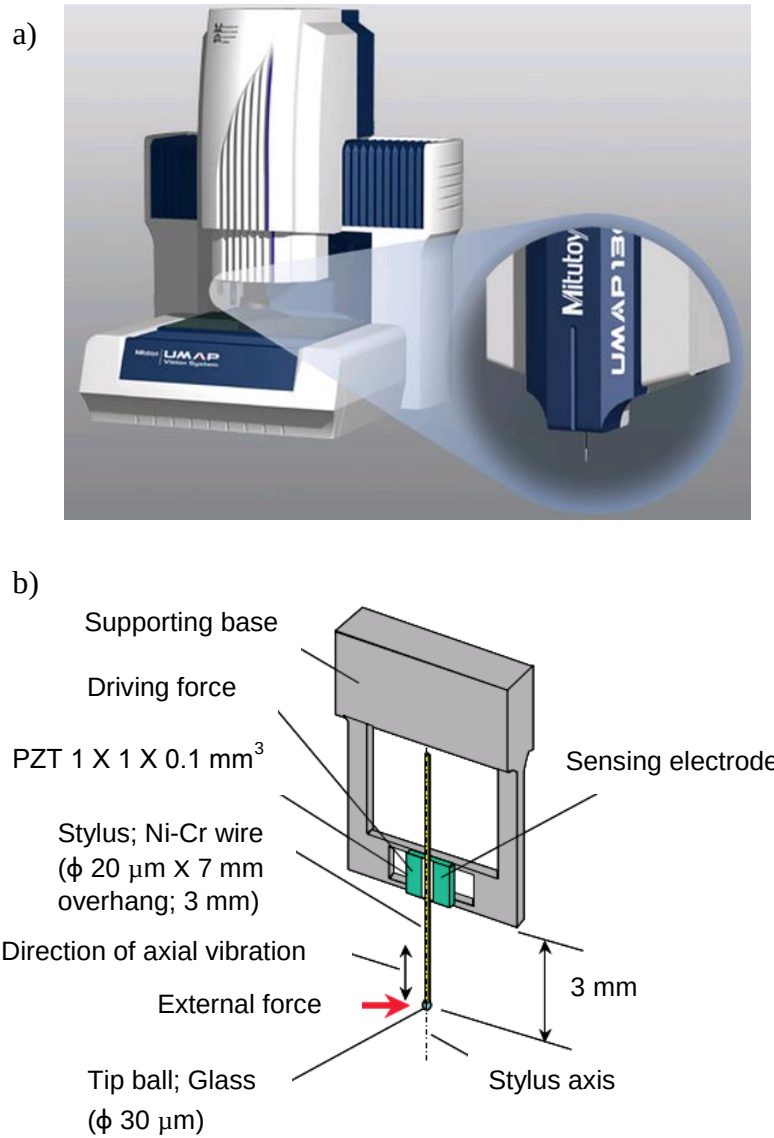


Figure 2-21: (a) Image of UMAP Vision System with exploded view inset for one dimensional vibrating probe [88], (b) schematic shows the main parts of one dimensional vibrating probe [89].

2.3.4.2 Three-axis vibrating micro-CMM probe by NPL

The National Physical Laboratory (NPL) developed a design of three dimensional vibrating probe as shown in Figure 2-22 (a). The probing system consists of triskelion device, stylus, six piezoelectric actuators and six sensors. The flexure of triskelion device has 2.5 mm length, 200 μm width and 10 μm thickness [45]. The stylus is mounted on the middle of the suspension structure and has 70 μm (radius of stylus tip) and 2.5 mm (length) [36].

This probing system was designed to vibrate the probe normal to the surface with same frequency and amplitude in all three directions, thereby providing an isotropic response. The stylus tip is forced to vibrate at certain amplitude and frequency (around 1.5 kHz) using two piezoelectric actuators deposited in each flexure. The interaction between the stylus tip and work piece leads to a change in the amplitude and the phase and can be measured by two piezoelectric sensors located at the flexure. The repeatability of the vibrating probing system was estimated to be 9 nm and the linearity error is around 25 nm [45].

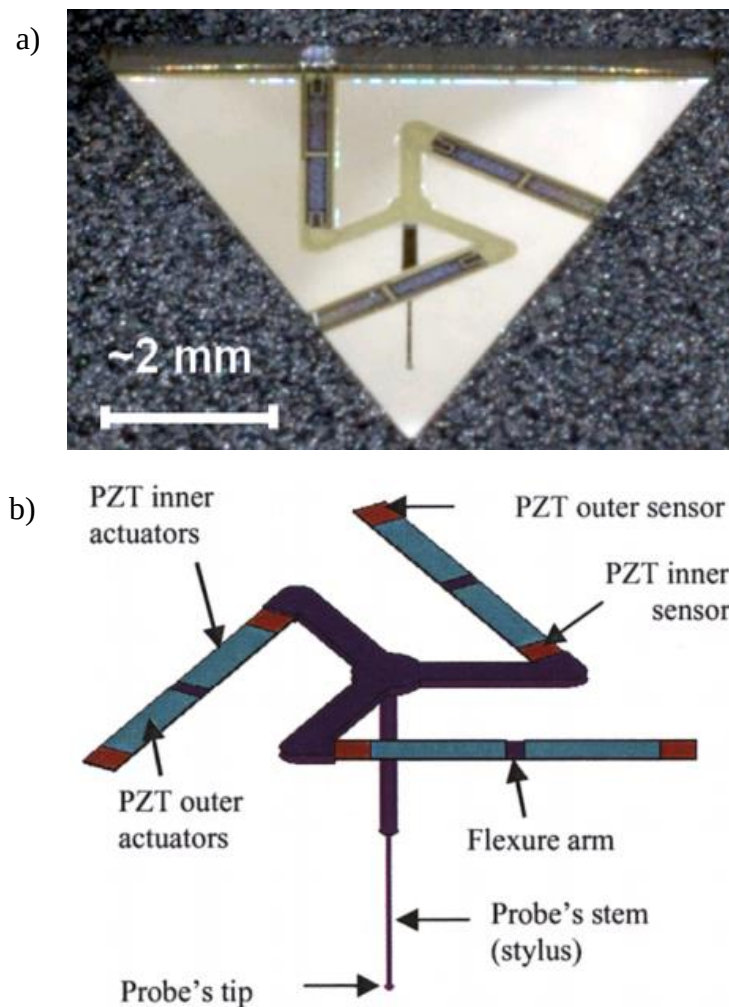


Figure 2-22: a) Image of the 3D vibrating microprobe by NPL [36], b) schematic diagram shows the main parts of vibrating micro-CMM probe [91].

2.3.4.3 Three-axis vibrating micro-CMM probe by Ilmenau University of Technology

Ilmenau University of Technology developed recently a tri-axial microprobe which oscillates in three independent directions to avoid coupling between the axis of

motion. As observe for the NPL micro vibrating probe, coupling between transducers can cause deviation in the vibration directions [36]. The Ilmenau probe has been developed with a focus on overcoming that challenge. This vibrating probe is MEMS based as shown in Figure 2-23(a) [87]. The electrostatic actuators and sensors were integrated into the probing system. The comb drive is used to vibrate the stylus tip in both z and y axis as can be seen in Figure 2-23 (b). To excite the stylus tip in x axis, the parallel-plate actuator is used which consist of movable structure and stationary substrate. To decrease the squeeze film damping in the x-direction, the moveable structure in the x-axis consist of perforations.

The system has two styluses where the first one was used to detect the measurement surface and the second one was used to balance the movement and to avoid system wobbling in y axis. The diameter of stylus tip is 200 μm and the stylus has 5 mm length. This system is operating in multiple natural frequencies which can be considered as anisotropic where the frequency in x, y and z direction are 2000 Hz, 990 Hz and 880 Hz, respectively. The resolution of the measurement system is reported to be 30 nm [87]. The repeatability test is not published anywhere.

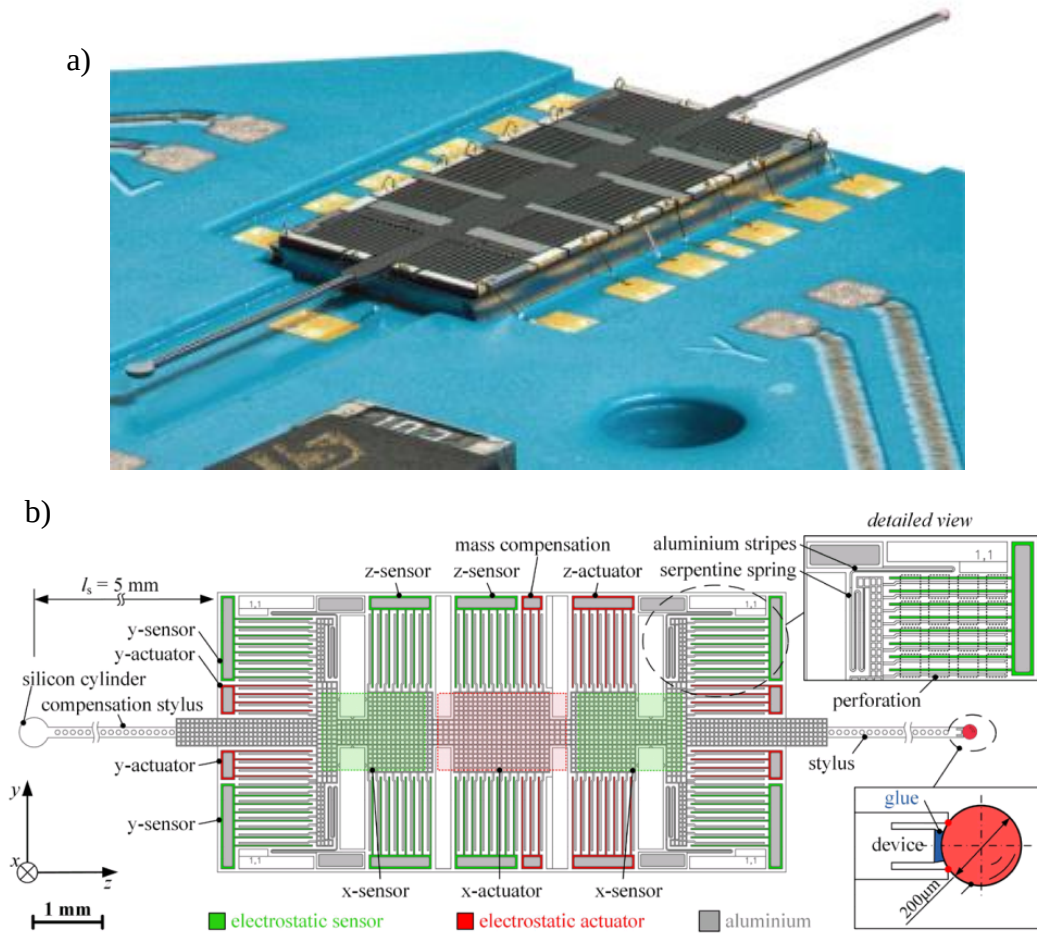


Figure 2-23: (a) Photograph of triaxial microprobe, (b) schematic diagram shows working principle of the triaxial microprobe [87].

2.3.5 Summary of current state-of-the-art in micro-CMMs probes

A wide variety of low contact force probing systems have been reported in this section and summarised in Table 2-1. These tactile probing systems have been classified in four categories which include miniaturized 3D probing system, MEMS-based probing system, opto-tactile sensor and vibrating micro-scale probing system.

The miniaturized 3D probing systems, such as the METAS system, can achieve reasonably low isotropic stiffness, but suffer with high probe mass, sensitivity to inertial triggering a high dynamic force that can cause damage to the work piece. Using these probes with smaller stylus tip diameters can lead to increase the contact pressure which can damage the measurement surface.

Probing systems that replace the metallic structures of miniaturized probes with silicon structures with integrated circuitry have been shown. These can offer high sensitivity and low effective mass. However, most of MEMS-based probing systems have a high and anisotropic stiffness. The single crystalline silicon that is mostly used in MEMS design has an anisotropic mechanical behaviour. This can make the design of these probing systems difficult to achieve the isotropic stiffness [92]. Also, these probing systems are fragile and susceptible to damage.

It has also been shown that opto-mechanical probing systems can provide low contact force, in range of micronewton, and with very small stylus tip diameter ($15\text{ }\mu\text{m} \sim 25\text{ }\mu\text{m}$). However, this probe system is significantly affected by surface forces [21], and anisotropic stiffness. Another observation is that the uncertainty of this probe is quite high compared to other categories, in range of 300 nm.

The vibrating probing system in part overcomes a number of these shortcomings, by counteracting surface attraction force by using the dynamic inertia, and reducing contact forces, to micronewton range, by detecting proximity rather than full contact. However this can be considered as a slow process compared with scanning probe where more points can be measured with short time. Also, a complex control strategy is required for the operation of these probing systems. To date there are no examples of vibrating probes that have the necessary motion control, or well defined contact mechanics, required for low uncertainty measurement.

It is observed that when designing micro-scale tactile probes, a design trade-off must be made between the stiffness and flexibility of the probing element. Flexibility is needed to ensure contact with a sensitive part without causing damage but, the probe must be stiff enough to overcome attractive surface forces and to ensure it is not fragile, easily damaged or sensitive to inertial loads. This leaves a requirement for a viable stiffness probe that can be used as easily as a conventional probe and is capable of scanning.

To aid in the design and development of a variable stiffness probe the literature was reviewed for methods of varying suspension structure stiffness. Various methods for achieving variable stiffness are presented in the next section.

Table 2-2: Summary of current state-of-the-art in micro-CMMs probes. (*) Indicates the measurement range of the probing system, (+) Indicates the resolution of the sensor. (---) Indicates the value is not published anywhere

Category	Sensor name	Force at the stylus tip	Mass / mg	Range* / μm	Res ⁺ / nm	Measurement error / nm
Miniature	METAS	Isotropic $\approx 20 \text{ N m}^{-1}$	7000	100	1	Repeatability ≈ 5
	NPL	Isotropic $\approx 10 \text{ N m}^{-1}$	350	20	3	Uniaxial uncertainty ≈ 100 ($k = 2$)
	IBS	Isotropic $\approx 70 \text{ N m}^{-1}$	160	10	2	3D uncertainty ≈ 15 ($k = 2$)
	Hefei	$k_y \approx 118 \text{ N m}^{-1}$ $k_z \approx 11 \text{ N m}^{-1}$	2500	5	2.5	Repeatability ≈ 9
		Isotropic $\approx 500 \text{ N m}^{-1}$	660	20	1	Repeatability ≈ 30
MEMS-based	PTB	$k_y \approx 3 \text{ kN m}^{-1}$ $k_z \approx 55 \text{ kN m}^{-1}$	---	10	$y \approx 3$ $z \approx 5$	Repeatability ≈ 20
	TUE	$k_y \approx 18 \text{ N m}^{-1}$ $k_z \approx 100 \text{ N m}^{-1}$	20	20	1.2	Repeatability ≈ 4
		Isotropic $\approx 450 \text{ N m}^{-1}$	25	5	1	3D uncertainty ≈ 10 ($k = 2$)
Opto-tactile	PTB	Anisotropic, micronewton	---	---	50	Uncertainty $\approx (200 \sim 300)$
	NIST	Anisotropic, $< 0.2 \mu\text{N}$	---	$y \approx 15$ $z \approx 0.3$	10	Uncertainty $\approx 110 \text{ nm}$ ($k = 2$)
	HIT	Anisotropic, $< 10 \mu\text{N}$	---	$y \approx 5$ $z \approx 1.5$	5	Uncertainty $\approx (24 \sim 50)$
Vibrating	Mitutoyo, uniaxial vibrating sensor	Anisotropic, micronewton Resonant frequency $\approx 349 \text{ kHz}$	---	---	10	Repeatability in micrometre range
	NPL, 3D sensor	Isotropic, micronewton Resonant frequency $\approx 1.5 \text{ kHz}$	0.2	---	---	Repeatability ≈ 9 Uncertainty ≈ 58 ($k = 2$)
	TU Ilmenau, 3D sensor	Resonance frequency : $x = 2 \text{ kHz}$, $y = 0.99 \text{ kHz}$, $z = 0.88 \text{ kHz}$	less than 1.5	---	30	---

2.4 Methods for variable stiffness suspension structure

Tuning the stiffness of suspension structures is not a new concept, and it has been used for many applications, but has not been applied to micro-probing systems. It is important to understand the method that has been used to modulate the stiffness element. This will help to implement and compare different techniques to tune the stiffness of the micro-probing system as presented in Chapter 4. Therefore, this section will present various designs for variable stiffness that have been proposed based on the literature review.

2.4.1 Tuning stiffness element by using shape memory alloy elements

The modulus of elasticity plays an important role in determining stiffness. Shape memory alloy (SMA) has an interesting property where the elastic modulus can be modified by heating up or cooling down the stiffness element. This application has been used in a tuned vibration absorber (TVA) [93]. The effective TVA suppresses the forcing frequency that acts on the primary structure through adjusting the natural frequency of the vibration absorber. To eliminate the vibration, the forcing frequency has to be equal to the natural frequency of the vibration absorber. This system does not need an actuator to modulate the stiffness element. However, the maximum tuning change in frequency was found to be around 21 % and the time taken to reach a steady state was 2 min [93]. This system has a low response time due the heating and cooling limitations.

2.4.2 Tuning stiffness element by modifying the effective geometry of the structure

This strategy is based on modifying the effective geometry of the structure; for example, adjusting the ratio between the beam separation and the beam thickness [94]. In this method, the stiffness reduction is limited by making the separation gap equal to zero.

Figure 2-24 illustrates another technique for tuning the stiffness of TVA by controlling the geometry [95]. The working principle of this technique is based on varying the distance (b) between flexural stiffness as shown in Figure 2-24. This can be achieved by changing the static load (F), where increasing the distance (b) leads to a decrease in the stiffness and vice versa.

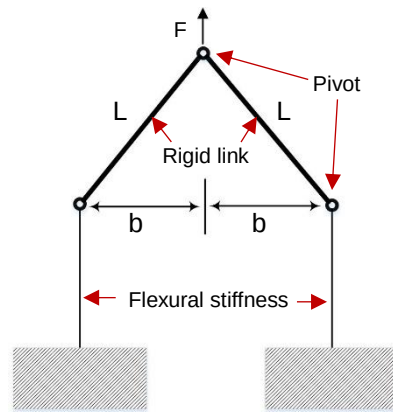


Figure 2-24: Schematic of modulating the stiffness by modifying the effective geometry of the structure. Applying static force (F) leads the two rigid links (L) to cause lateral deflection at the pivot of the flexural stiffness, where the distance (b) can be either increased or decreased [95].

2.4.3 Tuning stiffness element by adjusting the internal strain

The third strategy is to subject the suspension structure to an internal strain. This results in either strain stiffening for tensile loads, or reduced stiffness for compressive loads. There are two ways to produce a compressive force in order to develop stress to the structure to reduce its inherent stiffness, Joule heating to cause thermal expansion and the external application of load.

2.4.3.1 Joule heating effect

This strategy is based on internal strain energy tuning. The stiffness element is subject to a stress due to thermal expansion. This results in either strain stiffening for tensile loads (cooling), or reduced stiffness for compressive loads (heating). This application has been used in the microelectromechanical system (MEMS) to tune the frequency of a resonant sensor [96]. The requirement is that the spring element needs to be a clamped-clamped beam to allow the developing stress built up in the structure to reduce its inherent stiffness. In this application, the overall stiffness at the movable plate, which has been driven by the comb-drive actuator, is tuned as can be seen in Figure 2-25. Heating up the spring element causes thermal expansion, which leads to reduce the stiffness. However, the comb-drive actuator that generates force in the lateral direction (F_y , see Figure 2-25) leads to increase the overall stiffness as the tensile stress increases in the beam. The resonant frequency has been increased by

24 % using the comb-drive actuator and reduced by 27 % using the joule heating effect. Increasing the compressive stress makes the potential energy curve wider, as can be seen in Figure 2-26(b) compared to Figure 2-26(a), where the compressive stress is zero. It is clear that increasing the compressive force creates two stable positions as the beam can be deviated to either the right or left region. The movable structure can travel over the range where the barrier energy is less than the maximum energy stored in the beam, as can be seen in Figure 2-26(b). Figure 2-26(c) shows that the barrier energy has limited the travel of the movable structure where the frequency starts to increase at this point [96].

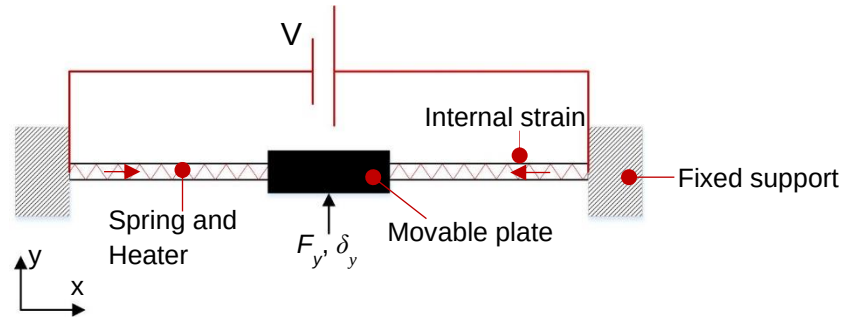


Figure 2-25: Schematic illustration of integrating the thermal actuator to provide compressive force for stiffness reduction. F_y is the lateral force that is generated by the comb-drive actuator to produce a tensile stress for increasing the stiffness element.

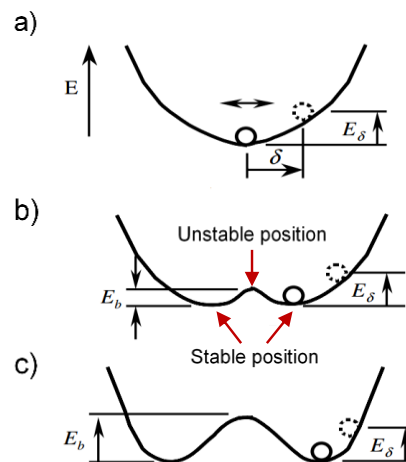


Figure 2-26: Maximum potential energy (E_δ) versus stable position (δ) at different situations: (a) the compressive force is equal to zero and there is one stable position; (b) the barrier energy is less than the maximum potential energy; and (c) the barrier energy is larger than the maximum potential energy [96].

2.4.3.2 *Effects of compressive axial preload*

Loads can be applied to the ends of each clamped-clamped beam to modulate the stiffness using a piezo-electric or other similar form of actuator. This method has been implemented to tune the resonant frequency for a piezoelectric energy scavenger to produce a power along a range of driving frequencies [97]. The compressive force has been produced using a differential micrometer placed at one end of the beam as can be seen in Figure 2-27. The maximum reduction of the frequency that has been achieved was around 24 %.

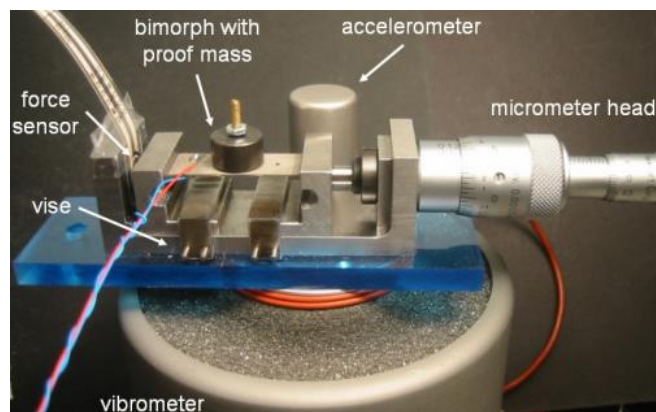


Figure 2-27: Experiment set up which shows the differential micrometer placed at one end of the beam to generate a compressive force [97].

An axial compressive force applied in a different way at the free end of cantilevers beam to tune the lateral frequency for energy harvesting [98]. The compressive force has been controlled to straining the structure at the free end using a magnetic force (see Figure 2-28). The two separate permanent magnets are facing each other with adjustable distance d . Adjusting the distance (d) between two magnets assists to reduce the frequency by around 70 %.

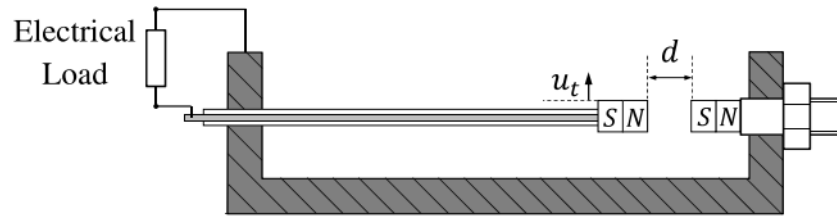


Figure 2-28: Schematic diagram of tuning the lateral stiffness of cantilever beam using axial compressive force [98].

2.4.4 Tuning stiffness by applying additional force to either augment or reduce the action of external loads

This technique is based on applying an additional non-contact force which in turn induces an additional stiffness to the structure to tune the resonance frequency. For example, Figure 2-29 shows the electromagnetic force applied to the free end of the cantilever beam where the additional lateral stiffness can be either increased or decreased. The maximum reduction that has been achieved was 20 % [99]. The stiffness reduction is limited as the attractive force increases with decreasing distance between the two magnets. The attractive force at a certain distance becomes larger than the lateral stiffness of the cantilever beam, which leads the two magnets to collapse against each other.

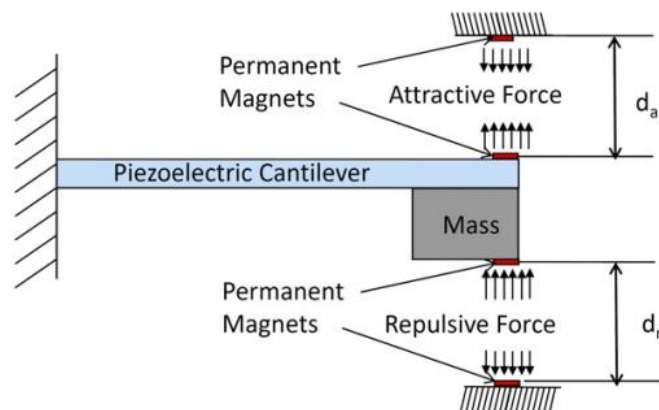


Figure 2-29: Schematic of harvesting device that used attractive electromagnetic force to decrease the stiffness at the free end of cantilever beam. The repulsive force was applied at the bottom to increase the stiffness [99].

Modulating the stiffness of the probe membrane has been implemented into atomic force microscopy (AFM) for biological applications [100]. A microcantilever of the AFM is actuated using a piezoelectric actuator to bring a tip into and out of a contact with a surface. However, there is a dynamic limitation of using this technique to image a specimen in fluid with high-speed. Therefore, the active probe membrane is electrostatically actuated instead of using the piezoelectric actuator to drive the cantilever. Moreover, to increase the sensitivity of the force sensor, while the membrane actuated to the tip of the cantilever, the stiffness of probe membrane can be modulated as shown in Figure 2-30. The top electrode was located inside the membrane, and the second electrode was at a diffraction grating. To measure the membrane displacement; the membrane and diffraction grating was illuminated from underneath of a transparent substrate, using incident laser beam, as shown in Figure 2-30. The maximum reduction achieved is around 54 %. The stiffness reduction and actuation range are limited by the pull-in phenomenon where the two electrodes collapse on each other when the system becomes unstable.

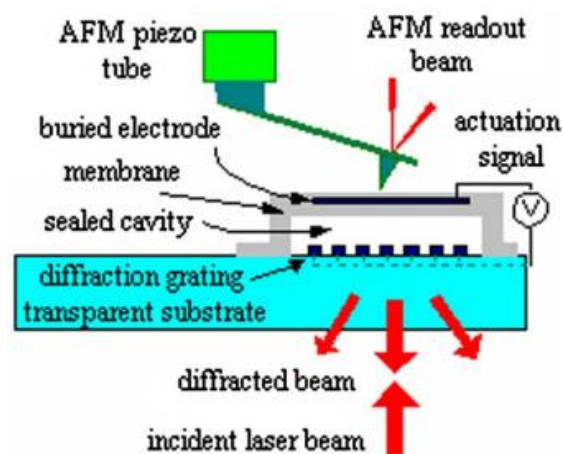


Figure 2-30: Schematic of integrating the active probe membrane with atomic force microscopy. The incident laser beam was used to measure the displacement of the active membrane [100].

A system for modulating the stiffness is a force sensor using an electrostatic actuator was developed by PTB. Figure 2-31 shows the working principle of modulating the stiffness of the force sensor. This system consists of gold-plated disk pendulum which suspended by two gold-plated tungsten wires and has a mass of 4 g. The central disk was grounded and located between two conductive outer gold-plated

plates with distance d_1 and d_2 as illustrated in Figure 2-31. Applying two additional springs between the central disk leads to decrease the stiffness of the disc-pendulum. By changing the applied voltage between the capacitors, the system is able to reduce the stiffness from $1.3 \times 10^{-1} \text{ N m}^{-1}$ to $7.4 \times 10^{-3} \text{ N m}^{-1}$. To detect the deflection of the central disk that is generated by the force to be measured (F_m), an interferometer was focused on the centre of the disk to measure its displacement [101-103].

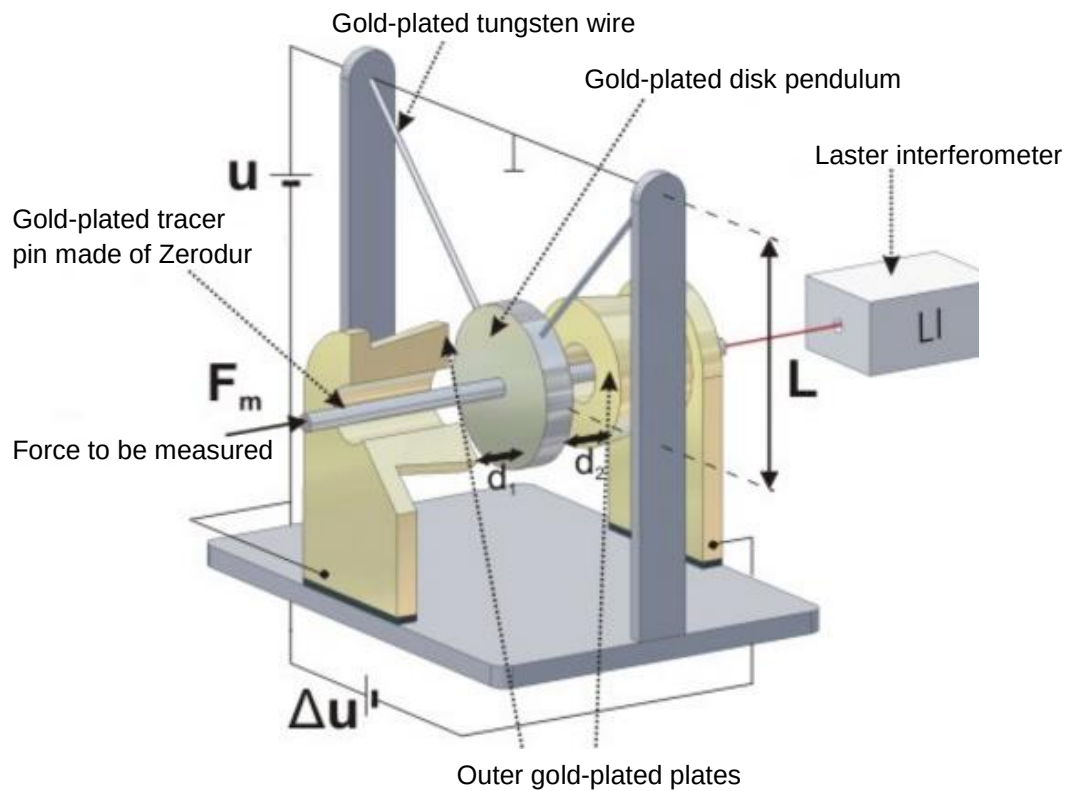


Figure 2-31: Schematic of integrating the electrostatic actuator with nanoforce sensor [101].

4.2.5 Summary of stiffness modulating techniques

Based on the literature, there are a number of stiffness modulation applications that are similar in nature to probing systems. A wide range of strategies were considered, and each type of strategy has a different operating principle, which plays an important role in determining the performance of stiffness reduction. The percentage of frequency reduction that has been achieved by different techniques is illustrated in

Table 2-3. It has been shown that the SMA has low response time, where 2 min has been taken to reduce the frequency by 20 %. The frequency response of the actuator is an important parameter to control and adjust the stiffness of the suspension structure.

It is obvious that the approach of modulating geometry can decrease the frequency by around 40%. The boundary condition for the flexural stiffness is changed during the actuation. However, this technique needs an actuator with a large stroke to vary the distance between flexural stiffness [95]. The main concern when using this technique is that as large load is applied to the beams, therefore, they suffer some degree of bending which results in a deflection of the probe tip. An error that is more difficult to deal with would arise if the large force results in a lateral in-plane displacement of the suspension structure. The transducer sensor arrangement typically used for measuring probe suspensions would not be able to detect this type of error as they are only sensitive to vertical displacements.

It has been demonstrated that modulating stiffness by adjusting the internal strain can be achieved by two ways. Firstly, joule heating can produce a thermal expansion which leads to adjust the internal strain of the spring beam. It has been shown that using this technique, the resonance frequency can be reduced by 27 %. Exerting an axial compressive force is the second way to manipulate the internal strain. It has been shown that the resonance frequency of the clamped-clamped beam structure is tuned manually using a micrometer to provide a compressive axial preload. Continuous modulation can be achieved using an actuator such as a piezoelectric to generate sufficient axial force. Using this method it is possible to reduce the stiffness to zero theoretically at the buckling point.

Applying additional non-contact force has shown the overall stiffness can be reduced or increased. This method can be achieved either using the electromagnetic or electrostatic actuator. It has been shown that integrated the permanent magnetic at the end of the cantilever beam assist to reduce the frequency by 20 %. Continuous modulation can be achieved using a planar coil with permanent magnetic to control the electromagnetic force.

Different strategies for tuning frequency have been reviewed in this section that may be integrated into a variable suspension structure of the micro-probe. In order to evaluate the appropriateness of each approach they must be carefully evaluated against the requirement of the probing system. A detailed evaluation of these techniques will be provided in chapter 4

Table 2-3: Summary of stiffness modulating methods.

Strategies	Applications	Frequency reduction,%	Tuning voltage or Ampere	Mechanical effect
Thermal effect [93]	(SMA) Tuning vibration absorber	≈ 21	0-9 A	Low response. Tuning temperature 67 °C.
Modify geometry [95]	Tuning vibration absorber	≈ 40	Manually adjusted	Linear.
Adjust internal strain	Joule heat effect (resonator) [96]	≈ 27	0-2 V	Thermal buckling. Low response.
	Apply axial force (harvesting) [97]	≈ 24	Manually adjusted (micrometer)	Damping increase. Spring beam buckled at critical load.
Add electrical spring	Electromagnetic spring (harvesting) [99]	≈ 20	Manually adjusted distance between magnets	Pull-in Non-linear.
	Electrostatic spring (AFM) [100]	≈ 29	0-60 V	Pull-in effect Non-linear.
	Electrostatic spring, force sensor by PTB [103]	≈ 94	0-5.8 V	Pull-in effect Non-linear.

2.5 Literature review summary and identification of knowledge gaps

This chapter has provided background information which included the area of coordinate metrology, the anatomy of CMMs and the challenge of measuring micro-parts. The types of micro-CMMs, which are available commercially, and their specifications have been reviewed as well in this chapter. A review of existing

micro-CMM probes focusing on the working principle and limitations of the probing system is presented. Then an overview of the methods for variable stiffness suspension structure has been covered.

From the literature review, it is possible to highlight three knowledge gaps, which are described as follows:

2.5.1 There are no examples of variable stiffness probing systems

When choosing the optimum micro-scale probing system, there are several conflicts that must be considered. Probing systems need to be flexible enough to be sensitive and maintain low contact force, whilst being robust enough for long life and insensitive to external influence such as inertial loads and surface forces. So far the problem has been tackled by making very low stiffness probes, using ultra-sensitive MEMS type probes, or using vibrating probes. No attempt has been made to create a probe with variable stiffness, so that stiffness can be reduced or increased as required during the measurement process. While there are examples of variable stiffness devices for applications such as tuned vibration energy harvesting, this concept has never before been considered in the context of micro-scale probing systems.

2.5.2 There is no easy way to minimise contact-force anisotropy of the probing system without being limited by stylus length

There are several commercially available micro-CMM probes that have an isotropic stiffness. However, the isotropic stiffness is only maintained when operating a stylus of a fixed length, determined at the time the probing system was designed. As such, to increase the stylus length or make it shorter, the suspension structure needs to be redesigned to have isotropic stiffness at the stylus tip. Also, a range of micro-CMM probes which are available commercially, yet suffer from anisotropic stiffness. Achieving isotropic stiffness is difficult due to the relatively long stylus that acts as a lever which can measure high aspect ratio features, causing stiffness in a direction coaxial to the stylus to be significantly higher than the stiffness in directions perpendicular to the stylus. As yet there is no satisfactory solution to this challenge; highlighting a need for further research in this area.

2.5.3 There are no established methods for operating an active suspension micro-probing system

Currently the only available work that was conducted to overcome some of the challenge of measuring the micro-parts has resulted in the development of a vibrating probing system. However, these probing system lack a well-defined operational strategy and suffer from coupling and crosstalk between transducers which leads to anisotropic measurement sensitivity. Therefore, in order to avoid these problems that have been observed with vibrating probing system, a variable stiffness probe will also require to have a simple operational strategy to be effective and must be supported by an effective controller. These operational methods may be used to modulate the probe stiffness that can switch between a stiff state and a flexible state as required by the stage of the measurement cycle.

Resolving these knowledge gaps will require a combination of design, analysis and experimentation, and that a detailed explanation of the research methods which has been followed is presented in the next chapter.

Chapter 3 Research Methodology

The three knowledge gaps were presented in previous chapter literature review were used in the present chapter to define the objective of the original research described in the thesis. An overview of the research approach that describes activities including mathematical modelling and experimental work was presented in this chapter. Following this boundary conditions that provide the base line for the scope of this project are described.

3.1 Research approach

The central aim of this project was to develop a tactile micro-probing system that makes use of an active suspension structure that can be tuned to have either low or high stiffness as required, in order to improve accuracy of measurement.

There were three areas of investigation in the project can be summarized as follows:

- 1) Modulating probing force of the micro-CMM probe,**
- 2) Ability to achieve isotropic probing force without being limited by stylus length,**
- 3) Strategies for operating a variable stiffness probing system in two distinct modes.**

The research strategy employed in this project and the structure of the present thesis are shown in Figure 3-1. The three main contributions in the project have been broken down into six objectives to achieve the central aim. These objectives are described in the following subsections:

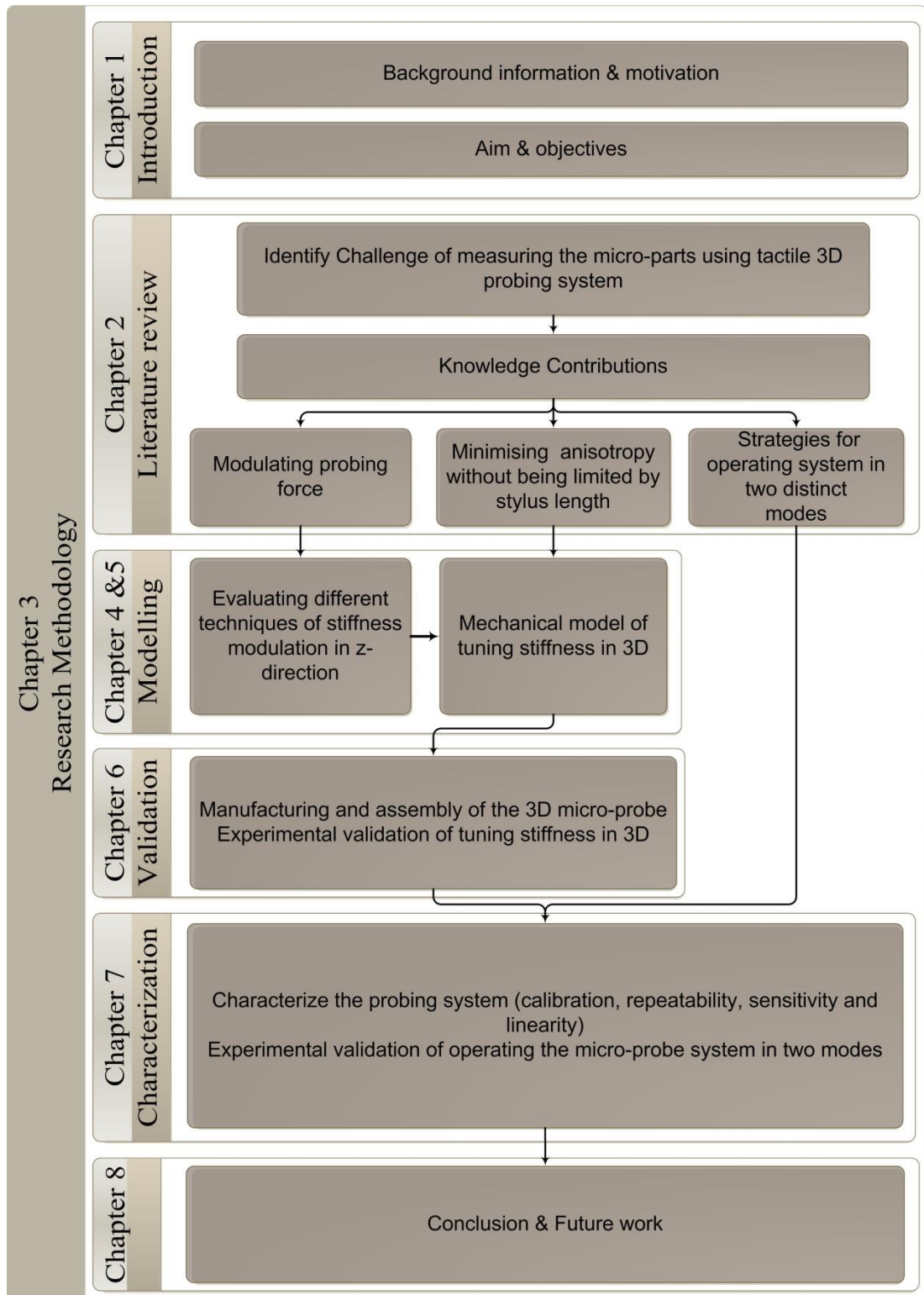


Figure 3-1: Presentation of research methodology which describes and links the main chapters.

3.1.1 Develop a realistic concept for applying variable stiffness

The idea of modulating the stiffness or probing force of micro-probe system for CMMs, is novel. Using the variable stiffness suspension structure it is proposed that a probe may be designed to operate in two distinct modes, either stiff, or flexible. These modes may then be selected at suitable points within a measurement cycle. This new concept was developed based on the limitations of the micro-CMM probe which was presented in **chapter 1** and **2**. However, when developing a realistic concept that enables the modulation of probing force of a micro-probing system, it is essential to investigate suitable stiffness modification strategies that may be employed in such a micro-probe. This will be the second object and presented in the next section.

3.1.2 Investigate into suitable stiffness modification strategies that may be employed in such a probe

Three approaches can be adopted to modify the stiffness of the suspension structure during use. The first approach is to modify the effective geometry of the structure; for example, by moving the fixing points of the suspension structure to create either a shorter or longer beam [95]. The second approach is to apply additional force to the structure to augment the action of external loads applied to the stylus tip [99]. The third approach is to subject the suspension structure to an internal strain [97]. This results in either strain stiffening for tensile loads, or reduced stiffness for compressive loads.

The first strategy has not been considered in the evaluation. In practise, controlling geometry is difficult to implement with suspension structure. This technique needs a high force to change the dimensions which may lead to in-plane displacement of the probe tip. This error would not be detectable by the common sensor systems used for monitoring suspension position, and therefore would present a significant additional challenge to overcome.

The second and third approaches require a micro-actuator to act as external force for tuning the probing force during use. The manner in which the thermal, piezoelectric, electrostatic and electromagnetic actuators were integrated with the suspension structure in the present study as presented in **chapter 4**. The behaviour of proposed

variable stiffness suspensions have been considered analytically and by employing finite element analysis (FEA) carried out using COMSOL Multiphysics 4.3a software.

To keep the analysis manageable for a large number of conditions, the work in chapter 4 only considers change in stiffness in the vertical direction. From this simplified analysis a stiffness variation concept has been selected and general design characterisations identified.

3.1.3 Investigate the ability to achieve isotropic probing force without being limited by stylus length

Achieving isotropic stiffness is often a difficult design task. In order to measure high aspect ratio features a relatively long stylus is often employed in micro-CMM systems. This can lead to significantly higher stiffness along the axis of the stylus compared to axes perpendicular to the stylus, as the length of the stylus acts as a lever. One aim of the project was to achieve a means to adjust stiffness parameters independently of stylus length, and therefore permit flexibility in choice of stylus.

In order to achieve isotropic stiffness, reduction in the vertical stiffness of the probe must be proportional to and greater in rate than the reduction in the lateral stiffness. This was investigated in **chapter 5** by creating a full mathematical model of the stiffness of the selected probe sensor with different stylus lengths. This model included analytical modelling using elasticity theory and numerical analysis (FEA) of the proposed system for modulating the stiffness of the probe. In order to assess the feasibility of tuning the stiffness of the probe within different configurations of the suspension structure, two possible designs of probe sensor are considered in chapter 5. The main geometric parameters which include length, width, thickness of the spring beam and the effect of the actuator on the vertical and lateral stiffness of the probe were examined in order to assess the performance of the proposed sensor.

To validate the concept of modulating the stiffness of the micro-probing system a prototype device needed to be developed.

3.1.4 Investigate a method of manufacturing for a prototype probe system

Different manufacturing techniques for building a prototype of the probing system were investigated and a series of probes produced in **chapter 6**. This chapter considers micro-milling, micro-electro-discharge machining and chemical etching for production, and a number of important designs revisions are made. To confirm the quality of production optical metrology was performed to verify part geometry and flatness. Production quality was related to aspects of the suspension performance in objective 5.

3.1.5 Compare a numerical solution with experimental result in order to validate the simulation, and understand the limits of the manufacturing.

The probing system was then assembled to validate the first and second knowledge gaps; namely modulating probing force of the micro-CMM probe, and ability to achieve isotropic probing force without being limited by stylus length. In order to demonstrate the predicted performance, and validate the modelling suspension, the stiffness of the system was experimentally measured.

To accurately measure the vertical and lateral stiffness of the probe sensor, a system for controlled displacement in the micrometer range and a force sensor that can read in the micronewton range are required. This would allow force displacement curves to be generated for each axis. Accurate force sensors, such as the Femtotools FT-8 (Femtotools, Buchs, Switzerland) [104], are expensive, susceptible to damage and unaffordable in this research. Therefore, an alternative method was devised to evaluate the stiffness along different axes. This method is based on analysis of the suspension structures' modal frequencies. Frequency information could then be used to determine the potential of the structure to modulate both vertical and lateral stiffness. To check the capability and robustness of the designed test rig for measuring the resonant frequencies of suspension, a laser vibrometer was initially used to measure the amplitude of the oscillations (see **chapter 6**).

To measure resonant frequencies using test rig, sinusoidal excitation frequencies was applied, sweeping from the vertical to the lateral mode using a linear chirp signal. For a range of static loads, used to modulate the stiffness, the response of the structure to the dynamic excitation could then be determined using a displacement

transducer (capacitive sensors). A finite element model was used to predict the modulated stiffness as the frequencies monitored in measurement were in the same range as frequencies estimated in FEA using COMSOL 4.3a. The reduction ratio between the vertical and lateral stiffness was determined to identify the characteristics of the isotropic stiffness point. The ratio between vertical and lateral stiffness values that was reduced is identified to characterise the isotropic stiffness point.

The final area of investigation was to establish a suitable approach to switch between the stiff and flexible modes during use. This will be the sixth objective and discussed in the next section.

3.1.6 Develop a strategy that allows the control system switch between stiff and flexible stiffness

The main operational stages in the use of the probe to take measurements are presented and examined in **chapter 7**. The complete measurement system was created before validating the proposed approaches to switching between the flexible and stiff modes (third contribution). A novel controller was developed for NI LabVIEW and allowed the probing system to easily switch between controlled flexible and stiff modes as required. The repeatability of flexible mode probing was assessed from two perspectives. First repeatability in the traditional sense of variation in response are repeated controlled displacement as performed in flexible mode. Second the ability to make repeated switches from stiff to flexible mode, and maintain probe offset position, was determined; referred to as switching repeatability. Assessment of the method also included consideration of factors affecting measurement stability, such as long- and short-term off-set drift, sensitivity drift.

3.2 Chapter summary

An overview of the research methodology that was followed in this thesis was described in this chapter. The three knowledge gaps identified in chapter 2 were breakdown into six objectives:

- Develop a realistic concept for applying variable stiffness,

- Investigate into suitable stiffness modification strategies that may be employed in such a probe,
- Investigate the ability to achieve isotropic probing force without being limited by stylus length,
- Investigate a method of manufacturing for a prototype probe system,
- Compare a numerical solution with experimental result in order to validate the simulation, and understand the limits of the manufacturing,
- Develop a strategy that allows the control system switch between stiff and flexible stiffness.

Now that the central aims and objectives have been defined the technical work will be covered in **chapters 4** to **chapter 7**. The first objective and second objectives, *develop a realistic concept for applying variable stiffness*, and *investigate into suitable stiffness modification strategies* will be addressed in next chapter

Chapter 4 Methods for varying probe stiffness

When developing a realistic concept that enables the modulation of probing force of a micro-probing system, it is essential to investigate suitable stiffness modification strategies that may be employed in such a micro-probe. This investigation of the strategies will be established based on the literature review as presented in chapter 2. This chapter will present an evaluation of a range of methods to manipulate the stiffness of the suspension structure, such that it might be flexible during probing yet stiff after probing, in order to overcome some of the limitations faced by existing technology, as illustrated by Figure 4-1. For example, when the stylus tip is being driven towards a workpiece, the probe will be in the stiff mode (Figure 4-1a) to prevent false triggers caused by inertial loads generated when the probing system is moved by the micro-CMM. Immediately prior to making contact, when the probe enters a predetermined distance from the expected surface, it is set to the low stiffness condition (Figure 4-1b) to ensure minimal contact force. After a measurement has been made, when the probe must be removed from the surface, it can be switched back to the stiff mode (see Figure 4-1c). In this state, it is possible to develop sufficient force at the stylus tip to overcome surface attraction forces, without overstraining, and potentially damaging, the suspension structure.

After defining the strategies, a design study of different techniques that allow the micro-actuator to be integrated into the suspension structure will be reviewed. This study has addressed the ability to reduce the probing force and compares the relative performance of each actuator in regard to mechanical and thermal effects, and the applied voltage required. Finally, once the potential of each of these strategies is compared, the most suitable method for use with micro-scale probing systems will be selected.

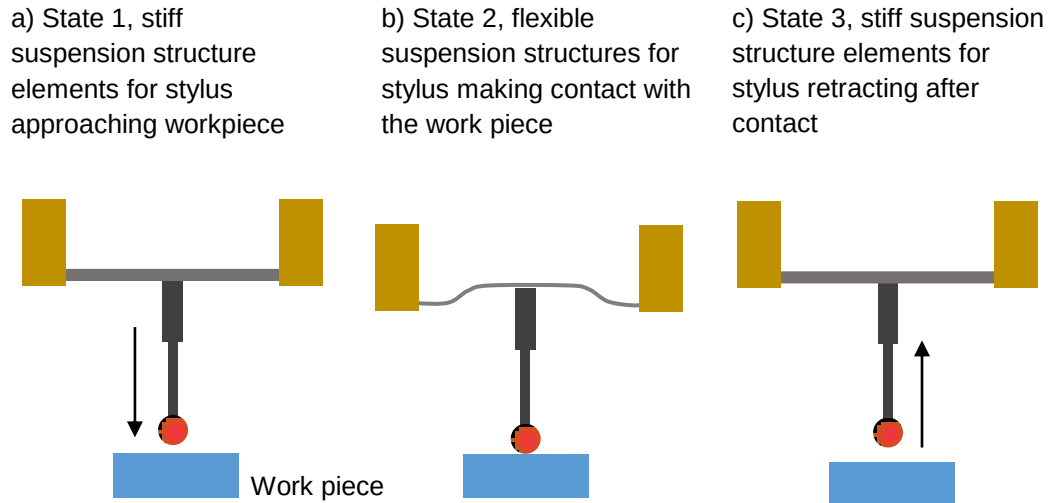


Figure 4-1: Illustration of the application of a variable stiffness probing system.

4.1 Evaluation of a range of methods to manipulate the vertical stiffness of the probing system

The strategies that have been used to modulate the stiffness are based on the literature. In order to select an appropriate method to adjust the stiffness of the probing system, a range of micro-actuation methods have been considered in this section to compare the relative performance of each one.

4.1.1 Methodology

To compare and evaluate the feasibility of modulating the stiffness of the probing systems over a range of strategies, it is possible at this stage to eliminate the method of tuning the stiffness element by using SMA and modifying the effective geometry of the structure which presented in chapter 2, section 2.4.1 and 2.4.2. The response time of the SMA is not acceptable due to long time to cool down [105]. This strategy is not preferable as reducing measurement speed is generally thought of as a very negative outcome; in fact efforts are normally made to increase measurement speed as much as possible.

Also it would appear in the literature that controlling geometry to control stiffness is difficult to implement with a suspension structure. This technique needs a high force to change the dimensions, which may lead to producing an in-plane displacement of the probe tip. It would not be possible to detect this error as the displacement would occur out-of-plane of deflection for typical sensor configuration.

This analysis modulating the vertical stiffness when evaluating will only consider the range of strategies. This is because a long stylus acts as a lever, causing stiffness in a direction coaxial to the stylus to be significantly higher than the stiffness in directions perpendicular to the stylus. Displacing the probe vertically causes the spring beams to deflect symmetrically. The stylus length does not affect the deflection resulting from the vertical displacement. Thus, performing the analysis in this way allows the parameter of stylus to not be considered. The initial stiffness in the vertical direction is assumed to be stiff (1045 N m^{-1}) related to probes that were presented in the literature review. The initial stiffness can be presented as state 1 and 3 in Figure 4-1 (a) and (c) respectively and calculated based on the parameters detailed in Table 4-1.

The targeted probing force is chosen as 1 mN as it represents an upper limit of the range of achieved probing forces for many micro-scale probing systems [16]. According to Boss [76], the over travel distance in which the motor needs to stop after receiving a contact signal between the probe tip and work piece is $7 \text{ }\mu\text{m}$. Thus, in order to achieve 1 mN probing force, the stiffness of the suspension structure must be able to be modulated down to 142 N m^{-1} . The flexible stiffness can be presented as state 2 in Figure 4-1 (b).

Finally the geometry of the suspension structure must be considered. For modulation by controlling strain, the load applied to the beam must be constrained such that the load vector is coincident with the axis of the beam. In addition, the structure must be symmetrical along the z-axis and free from any bias effects that may predispose the beam to bend as a result of an axial load (see Figure 4-2).

Figure 4-3 illustrates four different examples of suspension structure that meet the mentioned requirements. For each suspension structure, an intermediate body links the spring beams and provides a mounting point for the stylus. Qualitatively, the four structures have different strengths and weaknesses, which are presented in chapter 5. For simplicity, a cross-beam suspension structure design has been considered in the evaluation as shown in Figure 4-3(a). Titanium was chosen for the material of the suspension structure, as it has a high strength to weight ratio, which assists to reduce the effective mass of the suspension structure. Titanium also has a low elastic modulus, which is beneficial for reducing stiffness.

The micro-actuation methods considered are thermal, piezoelectric, electrostatic and electromagnetic actuators. The analysis of these actuators involves the investigation of their performance and mechanical behaviour using finite element analysis (COMSOL) and analytical methods.

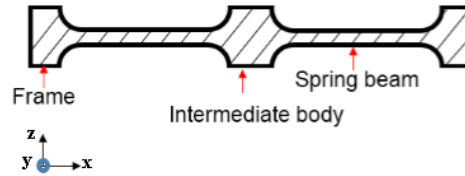


Figure 4-2: Schematic diagram of the cross-section of a beam to demonstrate the symmetry along the z-direction.

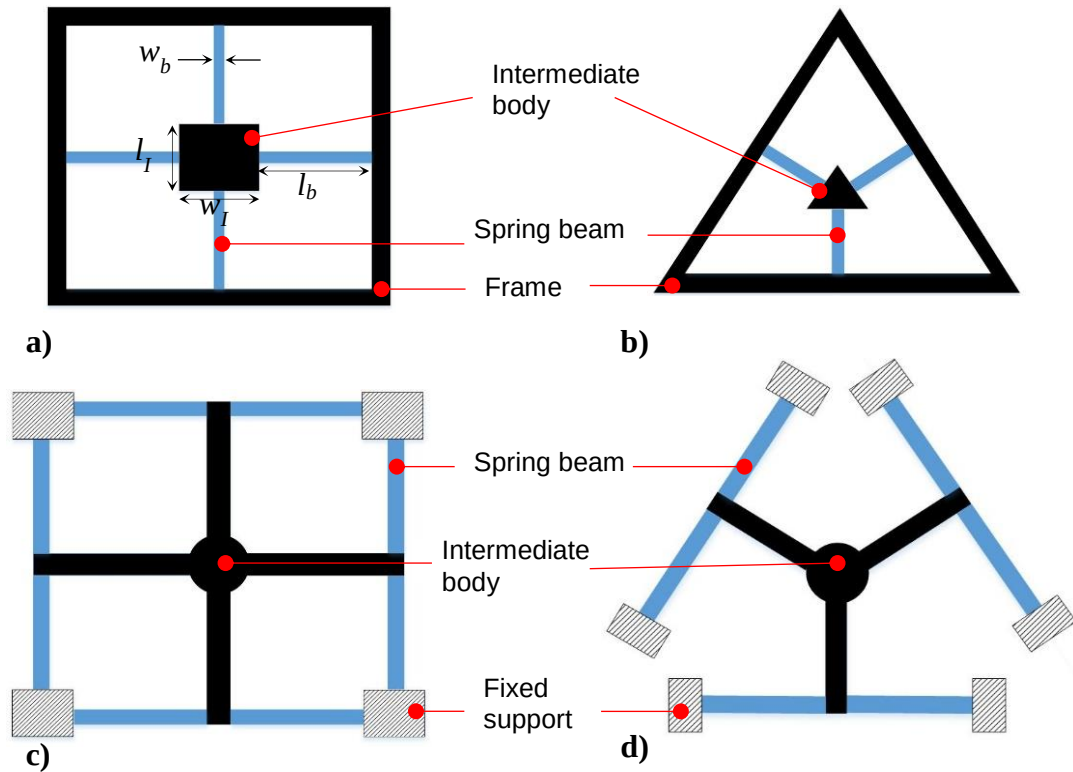


Figure 4-3: (a) cross-beam and (b) tri-beam designs which consist of four and three suspension beams respectively, joined to the intermediate body. (c) rectangular-beam and (d) delta-beam designs consisting of eight and six suspension beams respectively connected to a larger intermediate body.

Table 4-1: Structure main parameters and material properties of cross-beam design used to evaluate a range of micro-actuator.

Symbol	Definition	Value
l_b	Length of spring element	7 mm
w_b	Beam width of spring element	700 μm
t_b	Beam thickness of spring element	100 μm
$l_I \times w_I$	Dimensions of the intermediate body	5 mm x 5 mm
E	Young's modulus of Titanium	113.8 GPa
$\tilde{E}$	The effective Young's modulus for wide beam = $E/(1-\nu^2)$ [106]	128.8 GPa
ρ	Density	4430 kg m^{-3}
ν	Poisson's ratio	0.342
I_x	Second moment of area of the beam with respect to x axis, $I_x = w_b t_b^3/12$	$5.8 \times 10^{-17} \text{m}^4$
k_{iz}	Initial vertical stiffness of the cross-beam design, $k_{iz} = 4(12 \tilde{E} I_x / l_b^3)$	1045 N m^{-1}

4.1.2 Thermal actuator

The resistance of the beams is increased by decreasing the cross sectional area, which for a constant voltage leads to an increase in the current density; this causes increased Joule heating. The current can flow in the beam as shown in Figure 4-4. The intermediate body is grounded in order to control the thermal stress generated in each beam, which assists to modulate the stiffness individually for the four beams.

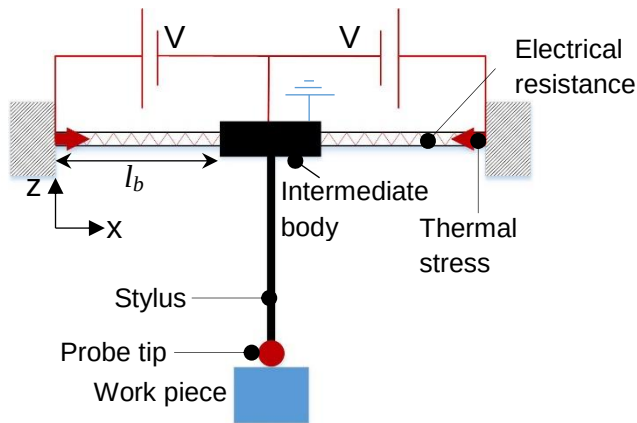


Figure 4-4: Schematic illustration of integrating a thermal actuator with the suspension system, where the stiffness of each beam can be modulated individually by grounding the intermediate body.

The beam warms up and dissipates more heat as a result of increasing the current, which leads to a greater thermal expansion. The thermal expansion, which can

generate compressive loads to the beams of the suspension system, can be obtained using equation (4-1) [106]:

$$\Delta l_b = \int_0^{l_b} \alpha (\Delta T) dx \quad (4-1)$$

where Δl_b , α , ΔT and x denote the thermal expansion along the beam length, the coefficient of thermal expansion, the difference between the ambient and maximum temperature and the beam length respectively.

By assuming the suspension system is placed in a vacuum, no convection heat transfer will be considered. This approximation is made to simplify the evaluation of the thermal actuator. The temperature difference along the beam can be obtained as a function of the square of the voltage as describes in equation (4-2) [106]

$$\Delta T = \frac{V^2}{2 R A_b K} \left[x - \frac{x^2}{l_b} \right] \quad (4-2)$$

where R , A_b and K represent electrical resistance, cross-section area of spring beam and thermal conductivity, respectively. The material properties are defined in Table 4-2.

Substituting equation (4-1) into (4-2) and integrating the equation to obtain the axial force as a function of square voltage as described in equation (4-3)

$$P_o = \frac{\alpha \tilde{E} l_b}{12 K R} V^2. \quad (4-3)$$

Solving the beam equation, including the axial force as can be done in chapter 5, the effective vertical stiffness of the cross-beams design can be written as a function of voltage, see equation (4-4).

$$k_z = 4 \tilde{E} I_x \frac{\left(\frac{\alpha l_b}{12 I_x K R} \right)^{\frac{3}{2}} \sin\left(\sqrt{\frac{\alpha l_b^3}{12 I_x K R}} V \right)}{2 - 2 \cos\left(\sqrt{\frac{\alpha l_b^3}{12 I_x K R}} V \right) - \sqrt{\frac{\alpha l_b^3}{12 I_x K R}} V \sin\left(\sqrt{\frac{\alpha l_b^3}{12 I_x K R}} V \right)} V^3 \quad (4-4)$$

Table 4-2: Thermal and electrical properties of titanium (Grade 5) [107].

Sample	Definition	Value
α	Coefficient of thermal expansion	$8.6 \times 10^{-6} \text{ K}^{-1}$
K	Thermal conductivity	$6.7 \text{ W m}^{-1} \text{ K}^{-1}$
ρ_e	Electrical resistivity	$1.78 \times 10^{-6} \Omega \text{ m}$

As can be shown in Figure 4-5, the stiffness of the suspension system can be tuned to 124 N m^{-1} by applying approximately 49 mV. Moreover, buckling occurs when the effective stiffness reaches zero as a result of the continuous load. However, by assuming the ambient temperature to be 20°C , the maximum temperature that can be achieved in the middle of the beam is approximately 45.1°C . The four beams buckle at around 53 mV, where the stiffness becomes equal to zero and is unstable.

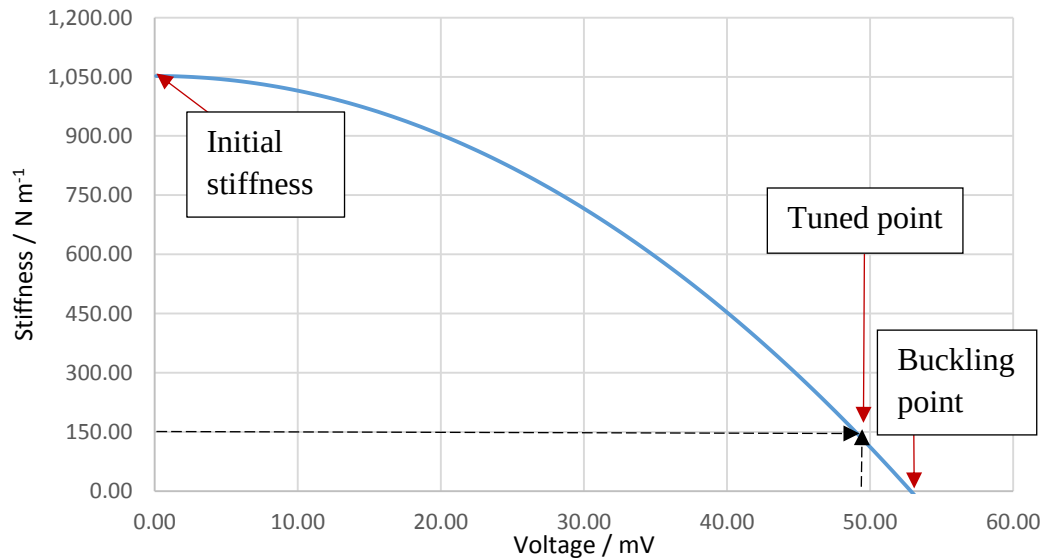


Figure 4-5: Modulated vertical stiffness using thermal actuator versus applied voltage.

4.1.3 Piezoelectric actuator

Piezoelectric materials have become widely used, where they can be either an actuator or a sensor [108].

The stiffness elements are actuated by piezoelectric stacks, positioned in intimate contact with the end of spring beams clamped against rigid fixing; the position of these actuators is shown in Figure 4-6. The actuator is 2 mm long with 4 mm² cross-section area. These dimensions were selected based on currently commercially available actuators (PI, PL022.30 [109]). The parameters of a piezoelectric linear stacks actuator that has been used for the evaluation are shown in Table 4-3

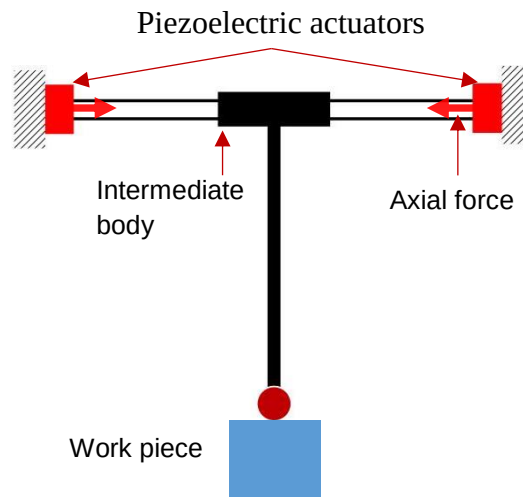


Figure 4-6: Illustrates the schematic of the integration the piezoelectric actuator with suspension system.

Table 4-3: Dimensions and properties of piezoelectric actuator.

Symbol	Definition	Value
$w_a \times t_a$	Actuator cross-sectional area	2 mm x 2 mm
l_a	Actuator length	2 mm
n	Number of piezoelectric layers	40
E_3	Young's modulus of PZT-5H [108]	50 GPa
d_{33}	Strain coefficient of PZT-5H [108]	$650 \times 10^{-2} \text{ m V}^{-1}$

The axial stiffness of the piezoelectric actuator ($100 \text{ N } \mu\text{m}^{-1}$) is greater than the axial stiffness of the spring beams ($1.3 \text{ N } \mu\text{m}^{-1}$). Thus, the displacement generated by the actuator is almost equal to the maximum displacement that can be generated without

loading the actuator. The generated axial force against the axial stiffness of the spring beam can be written as a function of the applied potential as described in equation (4-5) [108]

$$P_o = \frac{\tilde{E} w_b t_b n d_{33}}{l_b} V. \quad (4-5)$$

Solving the beam equation including the axial force, the effective stiffness of the suspension system in the z direction can be written as a function of voltage, see equation (4-6):

$$k_z = 4 \tilde{E} I_x \frac{\left(\frac{w_b t_b n d_{33}}{I_x l_b} \right)^{\frac{3}{2}} \sin \left(\sqrt{\frac{w_b t_b n d_{33} l_b}{I_x}} V \right)}{2 - 2 \cos \left(\sqrt{\frac{w_b t_b n d_{33} l_b}{I_x}} V \right) - \sqrt{\frac{w_b t_b n d_{33} l_b}{I_x}} V \sin \left(\sqrt{\frac{w_b t_b n d_{33} l_b}{I_x}} V \right)} V^{\frac{3}{2}} \quad (4-6)$$

The working principle of the piezoelectric to tune the stiffness of the probing system is the same as the thermal actuator. However, one advantage that distinguishes piezoelectric actuators is that the input voltage increases linearly with axial force as can be seen in equation (4-5). Moreover, the relationship between the stiffness and applied potential is linear compared to the thermal actuator, as can be shown in Figure 4-7. The stiffness of the suspension system can be tuned to 124 N m^{-1} by applying approximately 38.5 V.

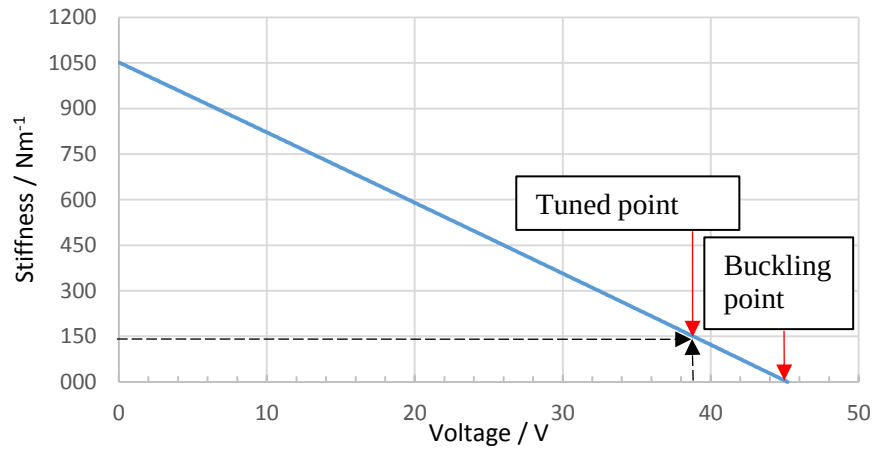


Figure 4-7: Effective stiffness vs. applied voltage for integrating piezoelectric actuator with the micro probing system.

4.1.4 Electrostatic actuator

Nowadays, micro electrostatic actuators are widely used in MEMS that create more difficult design challenge but allows the precise reliable production of such small devices. The working principle of this actuator is that the attractive force between electrodes can be utilized to actuate the system. However, there are several kinds of micro-electrostatic actuators such as cantilever beams, parallel plates, fixed-fixed beams, torsional and comb-drive actuators. All these actuators, except the comb-drive actuator, may experience the pull-in phenomena where two plates collapse on each other when the system becomes unstable.

Tuning the frequency of the resonator has been achieved in [110] by employing a different position of the electrostatic actuator to tune the system in the axial and transverse directions. However, integrating the parallel plate electrostatic actuator with the probing system also can assist to modulate the vertical stiffness, as can be seen in Figure 4-8.

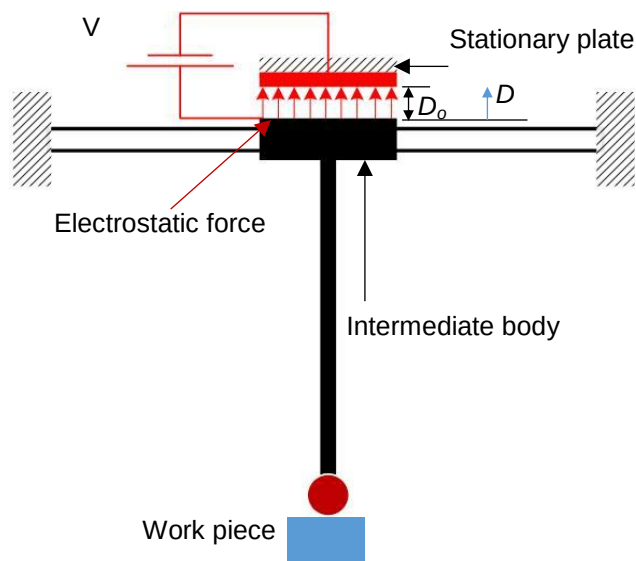


Figure 4-8: Shows the schematic of the integration of the electrostatic actuator with the probing system.

The gap between electrodes reduces as a result of increasing the capacitive potential, which can lead to a softening of the stiffness structure of the probing system. The effective stiffness can be estimated by taking the derivative of the total (residual) force that acts on the parallel plate as described in (4-7) [106]

$$\frac{\partial F}{\partial z} = k_z = k_{iz} - \frac{\epsilon_0 w_i l_i}{D^3} V^2 \quad (4-7)$$

where $\partial F/\partial z$, k_z , k_{iz} , ϵ_0 , V , D , w_i , and l_i represent the derivative of the restoring force with respect to the vertical displacement, effective stiffness in vertical direction, initial stiffness in vertical stiffness, the permittivity of air, applied voltage, the gap between the intermediate body and stationary plate after actuation, and the width and length of the intermediate body.

FEA (COMSOL) has been used to model and evaluate the integration of the electrostatic actuator with a probing system. The boundary conditions that have been used in the simulation to predict the effective stiffness of the probing system are shown in Figure 4-9(a). The properties and the definition that have been employed in the simulation are illustrated in Table 4-4.

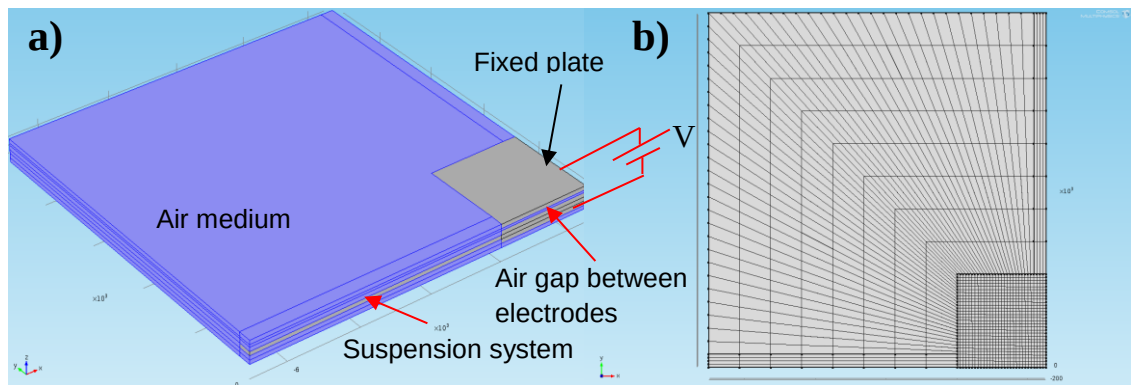


Figure 4-9: (a) Shows the schematic of the one-fourth of the model integration of the electrostatic actuator with probing system and the boundary conditions. (b) The high aspect ratio mapped mesh employed.

Table 4-4: Dimensions and properties of electrostatic actuator.

Symbol	Definition	Value
ϵ_0	Permittivity of air	$8.85 \times 10^{-2} \text{ C}^2 \text{ N}^{-1} \text{ m}^{-2}$
D	Displacement of the suspension system due to the actuator	m
D_o	Initial gap distance	$60 \text{ }\mu\text{m}$
$w_i \times l_i$	Surface area of the intermediate body	25 mm^2
$X_{standstill}$	Over-travel distance	$7 \text{ }\mu\text{m}$

Figure 4-10 illustrates the effective stiffness in the vertical direction and the electrostatic force acts on the intermediate body against the applied voltage. The reduction in the stiffness is limited by a pull-in phenomenon, where increasing the voltage beyond 526 V leads to collapse the two plates on each other. Thus, the effective stiffness can be modulated down to 500 N m^{-1} with an electrostatic force of 12 mN.

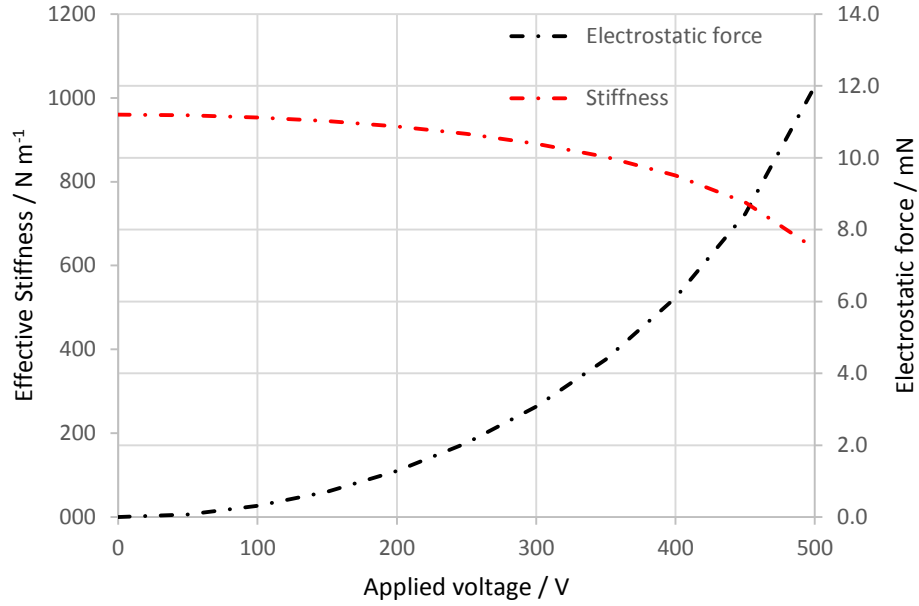


Figure 4-10: Effective stiffness and electrostatic force against applied voltage for electrostatic actuator.

However, this actuator can be used to reduce the probing force by applying additional force feedback (electrostatic force) to the structure that is a function of probe displacement. Increasing the electrostatic force leads to reduce the gap between the intermediate body and the stationary plate, where the intermediate body is displaced or lifted by ($z = D_o - D$). By assuming the displacement for which the motor needs to come to a standstill after receiving the signal is around $7 \mu\text{m}$, the desired scanning force can be estimated as presented in equation (4-8):

$$F_{\text{desired scanning force}} = k_z(X_{\text{standstill}} - z) \quad (4-8)$$

Figure 4-11 shows that the probing force at zero applied voltage is around 6.95 mN which is equal to $(k_{iz} / X_{standstill})$. This probing force can be reduced to 1 mN by applying 380 V to counteract the structures natural restoring force. The vertical stiffness is also reduced to around 820 N m⁻¹ as result of the electrostatic force acting on the intermediate body.

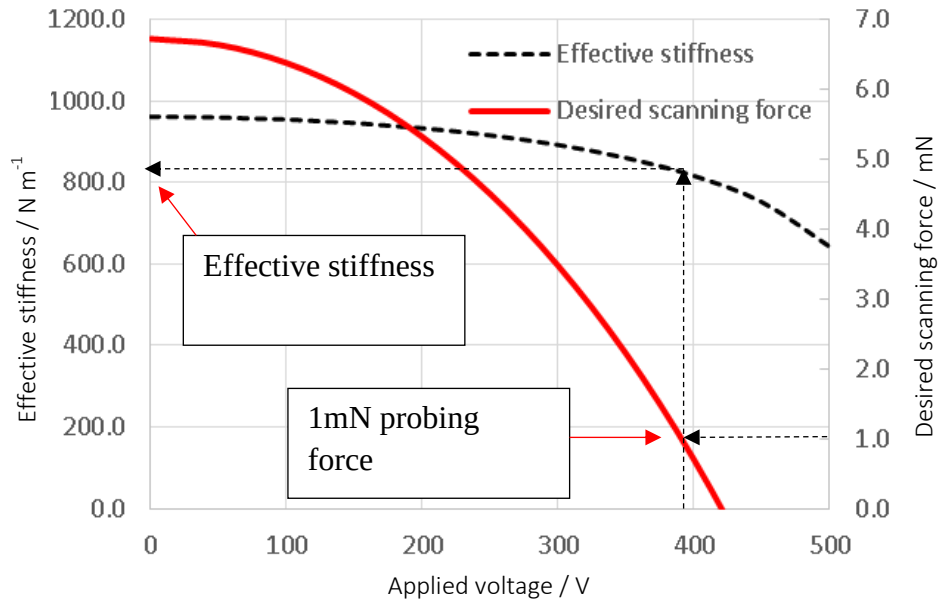


Figure 4-11: Scanning force and effective stiffness vs. applied voltage where the over-travel distance is 7 μm .

4.1.5 Electromagnetic actuator

The working principle of the electromagnetic actuator is that a force can be produced by the interaction between the permanent magnet and magnetic field which is provided by applying the current in the coil. The technique of modulating the probing force is based on applying an additional force to the suspension structure to counteract the structure's natural restoring force. The schematic diagram of integrating an electromagnetic actuator to modulate the probing force in the z-axis is shown in Figure 4-12.

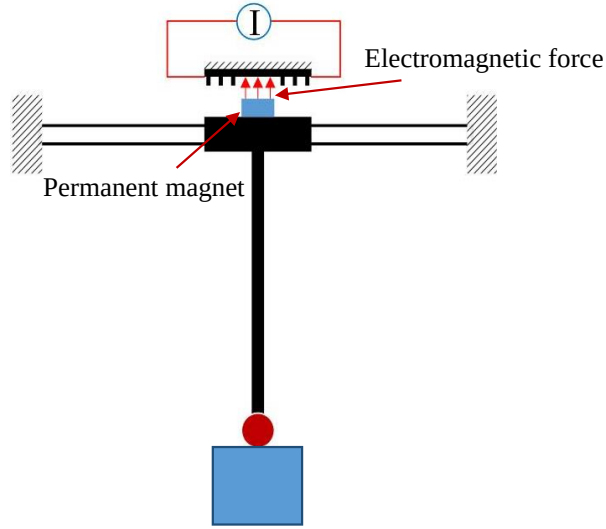


Figure 4-12: Shows the schematic of the integrating an electromagnetic actuator with the probing system.

The electromagnetic force that is generated by applying a magnetic field on a permanent magnetic positioned at distance from the planar coil is described in equation (4-9) [111].

$$F_{e,z} = B_r A_{PM} \int_{D_o}^{D_o+H} \frac{\partial H_z(z)}{\partial z} dz \quad (4-9)$$

where B_r , A_{PM} , D_o , H and H_z are the remanence of the magnetic material, surface area of permanent magnet, distance between the permanent magnet and the 2D coil, height of the permanent magnet and magnetic field along the central axis ($\hat{z}$). The type of planar coil is assumed to be Archimedean spiral due to it being easy to fabricate with a simple analytical model to predict the magnetic field [112]. Utilising the Biot-Savart law as presented in equation (4-10) to estimate the magnetic field that is generated by the planar coil [113].

$$\vec{H}(r, \theta, z) = \frac{I}{4\pi} \int \frac{d\vec{l} \times \vec{R}}{|\vec{R}|^3} \quad (4-10)$$

where ($\vec{R} = -r \hat{r} + z \hat{z}$) is displacement vector that specifies the distance from point of desirable magnetic field to ($d\vec{l} = r d\theta \hat{\theta}$), the thin wire element. The radius of the planar coil, which assists in estimating the electromagnetic field, is a function of θ and can be described in equation (4-11)

$$r(\theta) = a_0 + c\theta$$

$$c = \frac{b_0 - a_0}{2\pi N} \quad (4-11)$$

Where a_0 , b_0 and c represent inner mean radius of Archimedean coil, outer mean radius of Archimedean coil and the distance between complete turnings of the spiral.

The magnetic field along the central axis ($\hat{z}$) of the Archimedean coil is given by equation (4-12)

$$H_z(z) = \frac{I}{4\pi} \int_0^{2\pi N} \frac{(a_0 + c\theta)^2}{(z^2 + (a_0 + c\theta)^2)^{\frac{3}{2}}} d\theta \hat{z} \quad (4-12)$$

$$= \frac{IN}{2(b_0 - a_0)} \left[\ln \left[\frac{\sqrt{b_0^2 + z^2} + b_0}{\sqrt{a_0^2 + z^2} + a_0} \right] + \frac{a_0}{\sqrt{a_0^2 + z^2}} - \frac{b_0}{\sqrt{b_0^2 + z^2}} \right] \hat{z}$$

The geometric parameters of the permanent magnet and planar coil that are used in the analysis are presented in Table 4-5.

Table 4-5: Dimensions and properties of electromagnetic actuator.

Symbol	Definition	Value
a_0	Inner mean radius of Archimedean coil	1 mm
b_0	Outer mean radius of Archimedean coil	5.5 mm
g_0	Gap of spiral coil	0.3 mm
A_c	Cross sectional area of the planar coil	0.5 mm x 0.2 mm
N	Number of turns	9
ρ_{coil}	Resistivity of the coil = $\frac{R_{coil} A_{coil}}{L_{coil}}$, copper [114]	$1.67 \times 10^{-8} \Omega \text{ m}$
A_{PM}	Surface area of the permanent magnet	1 mm x 1 mm
H	Height of the permanent magnet	1 mm
B_r	Remanence of the magnetic material, NdFeB [115]	1.16 T

The permanent magnet should be positioned at a certain height from the planar coil for the optimal performance where the maximum of the magnetic field gradient is achieved. To predict the performance of the electromagnetic actuator, the

electromagnetic force at the centre axis of the coil versus distance between the coil and permanent magnet is plotted in Figure 4-13, using equation (4-9). Clearly, the maximum performance that can be achieved is placed around $450\ \mu\text{m}$ from the planar coil. The electromagnetic force after this point starts to decrease, where the preference is to work at a maximum of its magnetic field gradient.

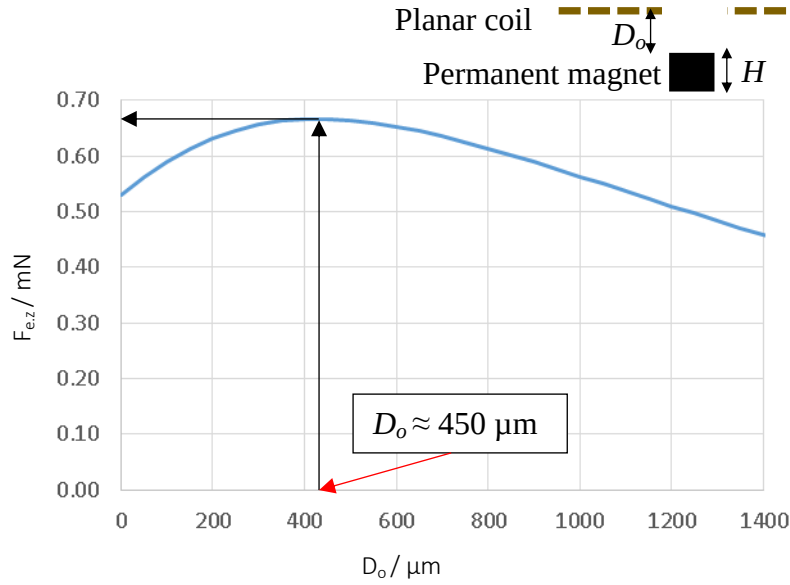


Figure 4-13: Electromagnetic force ($F_{e,z}$) at the centre axis of the Archimedean coil versus the distance (D_o) between the coil and the permanent magnet where the applied current is 1 A.

In order to compensate the probing force of 1 mN at the over travel distance of $7\ \mu\text{m}$, the electromagnetic actuator should exert a vertical force on the permanent magnet of around 6.315 mN. The amount of current that is required to minimise the probing force was found to be 9.5 A, which is equivalent to 0.21 V.

4.2 Discussion

Each type of micro-actuator has a different operating principle, which plays an important role in determining the performance of the probing system. The performance for each micro-actuator is feasible as a method to integrate into a tactile microprobe to overcome the limitations of a static stiffness response. Table 4-6 illustrates the comparison between actuators to achieve a 1 mN probing force.

The thermal actuator is capable of tuning the stiffness in the z direction with a low voltage, but it can be an obstacle as the system warms up to a high temperature. The thermal effect can reduce the repeatability of the probing system; which is likely to be a critical issue in this application. As regards the electrostatic actuator, it has been shown that the effective stiffness only declines slightly with increasing of the applied potential as compared to thermal and piezoelectric actuators. Even though reducing the gap or increasing the area of the electrode can lead to enhance the performance of the electrostatic actuator, the stiffness of the structure that the actuator softens is very high. The mechanical work of the electromagnetic actuator is produced by a non-contact force compared with piezoelectric. However, the induced currents in planar coils produce undesirable heat energy which can cause deviation in the measurements. Placing the 2D coil above the suspension system and attaching the permanent magnet on the middle of the intermediate body assists to minimise the thermal effect. Increasing the distance between the permanent magnet and the coil to minimize the thermal effect can lead to reduce the efficiency of the electromagnetic actuator.

One clear advantage of the piezoelectric actuator is that the input voltage increases linearly with displacement and vice versa. Moreover, the compensating voltage of the piezoelectric actuator is quite high compared with the electromagnetic actuator without adding heat energy into the system. The piezoelectric actuator has a good capability to be integrated into the measuring device compared with other actuators, as it is comparatively compact, has a strong effect on stiffness all the way to a buckling point, and has less unwanted thermal effects.

Table 4-6: Micro-actuators performance for compensating probing force of 1 mN.

The type of technique	Mechanical work	Linearity	Thermal effect	Mechanical effect on the probing system	Probing force voltage / V
Thermal	Thermal expansion	Nonlinear actuator	Temperature increased by 25.1 °C.	Thermal buckling at ≈ 51 mV	$\approx 4 \times 49 \times 10^{-3}$
Piezoelectric	Introducing an axial force	Linear actuator	Unaffected	Buckling at ≈ 860 mV	$\approx 4 \times 38.5$
Electrostatic	Noncontact force	Nonlinear actuator	Unaffected	Pull in at 516 V	≈ 380
Electromagnetic	Noncontact force	Nonlinear actuator	Planar coil dissipates heat	-----	≈ 0.21

4.3 Chapter summary

In this chapter a realistic concept for applying a variable stiffness micro-scale probing system has been presented, and practical strategies that may be employed to produce a variable stiffness suspension have been investigated.

Different micro-actuators, which include thermal, piezoelectric, electrostatic and electromagnetic types, have been considered with the suspension system of the micro-probe to reduce the probing force in the z-direction. This evaluation was performed to compare between the micro-actuators in order to allow the effective integration with the probing system. It has been shown that the use of compressive strain to reduce stiffness is the most suitable strategy for use with micro-scale probing systems. This strategy demonstrates significant stiffness reduction is possible with the proposed concept, and show it is theoretically possible to reduce stiffness to zero at the buckling point. It has been demonstrated that the thermal and piezoelectric actuator can be used to provide axial compressive force. However, piezoelectric actuators were shown to be the most promising technique for stiffness control as the input voltage increases linearly with displacement and has less unwanted thermal effects. Therefore, tuning the probing system stiffness in all directions by using a piezoelectric actuator, which is based on developing stress to the structure to reduce its inherent stiffness, will be addressed in the next chapter.

Chapter 5 Design of a variable stiffness suspension structure

In chapter 4, only methods to change stiffness in the vertical direction were considered. The actual probing system will require stiffness to be controlled in all directions. In this chapter, a probing system that uses a suspension structure that can be modulated to have either low or high stiffness in 3D as required using axial compressive force is presented. To illustrate the concept, two examples of suspension structure are presented for comparison. A range of stylus lengths is considered to achieve isotropic stiffness with the same geometric parameters for the two kinds of suspension structure. The investigation considers the performance and mechanical behaviour of the proposed sensors using analytical methods and finite element analysis using COMSOL4.3a. The chapter also covers the design of a sensor unit to detect and measure a movement of the suspension structure during operation of the probe.

5.1 Suspension structure performance considerations

Suspension structures with a range of geometric configurations are potentially possible as illustrated by the four very different examples shown in chapter 4 (Figure 4-3). Rectangular-beam and delta-beam designs have the same configuration, where a large intermediate body links between the spring beams. However, increasing the number of spring elements leads to increase the stiffness, such as the rectangular-beam design has eight spring elements compared to six spring elements in the delta-beam design. The vertical stiffness ratio of the rectangular-beam design to the delta-beam design is 1.333. Also, the vertical stiffness for the cross-beam design is higher than the tri-beam design by one third. Therefore, tri-beam and delta-beam designs are considered in this study. Figure 5-1(a) illustrates the tri-beam design which allows an axial force to be applied through each beam with an intersect point at the centre of the intermediate body. Figure 5-1(b) shows the delta-beam design which consists of three segments, each with two beams. In the delta-beam design, axial load is applied to the ends of each beam, with equal magnitude and in opposite directions. Both configurations are also designed such that, if equal loads are applied, there is no resulting displacement of the central intermediate body.

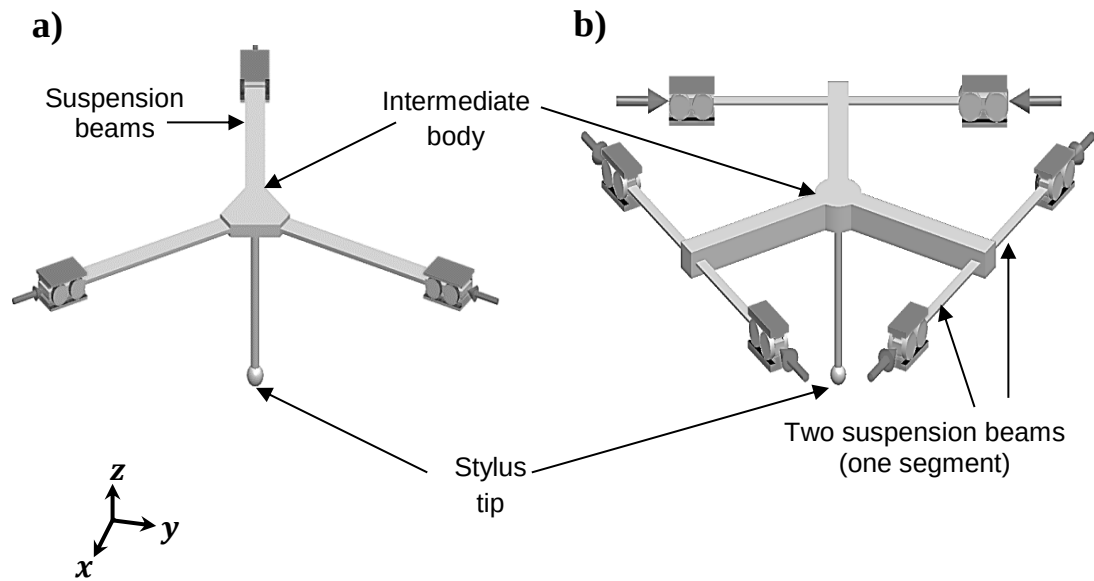


Figure 5-1: (a) Tri-beam design which consists of three suspension beams, joined to the intermediate body, (b) delta-beam design consisting of six suspension beams connected to a larger intermediate body. Guided roller constraints are shown at the ends of each of the beams for illustration only to indicate a degree of freedom allowing axial load to be applied, with the arrows used to indicate applied load vectors.

Qualitatively, the two structures have different strengths and weaknesses. The tri-beam structure has the advantage of being constrained in only three positions, and only three actuators are needed to load the structure, whereas six are needed for the delta-beam structure. An additional benefit of the tri-beam structure, as compared to the delta-beam structure, is that the three beams are spaced to allow sufficient room for actuators to be integrated. The disadvantage of the tri-beam structure is that an equal load must be applied to each beam to avoid translation of the intermediate body. This is not the case for the delta-beam design, which only requires equal loads to be applied to each individual beam.

The ability to vary the loads applied to each beam may be beneficial as it would allow independent stiffness control of each beam. For example, independent stiffness control would be useful to compensate for errors in manufacturing that resulted in beams with different geometry. In this case, different loads could be independently applied to each beam, which would allow each beam to be tuned to a uniform stiffness. A negative consequence of the delta-beam structure, as compared to the tri-

beam structure, is that it requires a larger, stiffer, intermediate body which increases probe inertia.

Both designs have three degree of freedom which can be considered as out-plane displacements. The three degrees of freedom are one linear displacement in the z axis and two angular displacement of the intermediate body around x and y axis.

To provide a more detailed understanding of how the two presented structures behave, theoretical and finite element models of both types were created using COMSOL 4.3a finite element software. Using the finite element models, the vertical and lateral stiffness of each suspension structure were estimated as a response to axial displacement.

5.2 Analytical performance model

In order to understand the capability of the proposed probing system, full analytical as well as numerical models have been created. The following section presents the analytical models that may be used to determine the influence of the axial compressive force on both tri-beam and delta-beam geometries. As there are subtle differences required for the analytical models of the two proposed probes, these will also be covered. To simplify the analysis the following assumptions have been made. The first assumption is that the deformations remain small enough that the beam stiffness is constant [116]. Therefore, geometric nonlinearity was neglected as all linear and angular displacements are small. The second assumption is that the stiffness of the intermediate body and the stylus have been assumed to be high enough compared to the suspension beams that they may be considered infinitely stiff. Thirdly, the suspension structure is symmetric along the z -axis (see Figure 4-2) where the unwanted bending stress is not considered. Thus, the stiffness in the vertical (k_z) and lateral (k_y) directions can be derived as follows:

5.2.1 Vertical and Lateral Stiffness of the Tri-beam design

The coordinate system used throughout this work is illustrated in Figure 5-1. The local coordinate for each spring (see Figure 5-3) has been converted to the global coordinate system.

5.2.1.1 Tri-beams vertical stiffness

The suspension structure for the tri-beam is symmetric about both the x - and y -axes. Thus, loading the stylus tip vertically (F_z) displaces the intermediate body in the z -direction with no angular displacement. The beam deformation can be modelled as a single beam that is clamped and guided in the y direction, while the other end is clamped and guided in the z direction (see Figure 5-2).

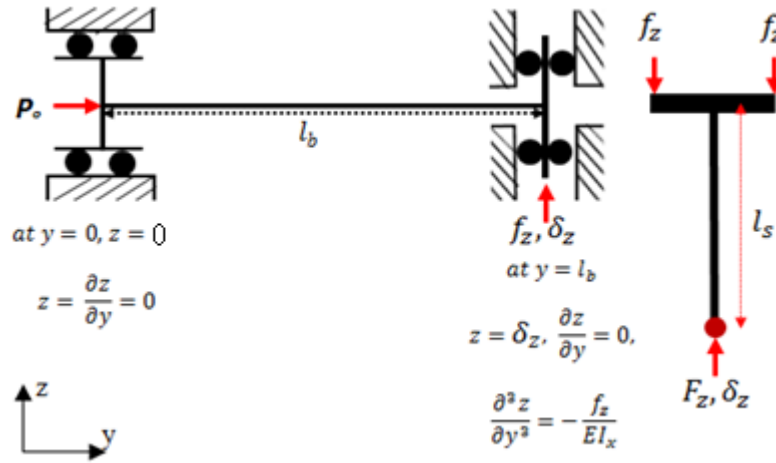


Figure 5-2: Shows the boundary conditions for estimating the stiffness in the z direction that is subjected to a compressive force and vertical force at the probe tip.

The displacement response to the probing force can be modelled by using the theory of elasticity, leading to equation (5-1) [116]:

$$\tilde{E}I_x z''''(y) + P_0 z''(y) = 0 \quad (5-1)$$

where $\tilde{E}$ is the effective Young's modulus for the wide beam [116], I_x is the second moment of area of the beam with respect to x axis, $z''''(y)$ is the fourth order derivative of deflection of the beam in the z direction, $z''(y)$ is the second order derivative of deflection of the beam in the z direction and P_0 is the axial compressive force.

The roots of the characteristic equation are $r = 0$, $r = \pm i\beta$, therefore the complementary solution can be written as shown in equation (5-2).

$$z(y) = c_1 \cos(\beta y) + ic_2 \sin(\beta y) + c_3 y + c_4 \quad (5-2)$$

where β is the dimensionless form of the axial compressive force ($\beta = \sqrt{P_o/\tilde{E}I_x}$). Applying boundary conditions that are specified in Figure 5-2 allows constants c_1 , c_2 , c_3 and c_4 to be determined. Thus equation (5-3) gives the deflection at any point of the suspension beam that is subjected to both a vertical force and an axial compressive load.

$$z(y) = \frac{f_z}{\tilde{E}I_x \beta^3} \left[\frac{(\cos(\beta l_b) - 1)(\cos(\beta y) - 1)}{\sin(\beta l_b)} + \sin(\beta y) - \beta y \right]. \quad (5-3)$$

The vertical force that acts on the probe tip can be divided by the number of beams to obtain the shear force on each beam ($f_z = F_z/3$). Using equation (5-3) the effective stiffness of the probing system in z -direction under the effect of the axial compressive force for the tri-beam design is given by equation (5-4).

$$k_z = 3\tilde{E}I_x \beta^3 \frac{\sin(\beta l_b)}{2 - 2\cos(\beta l_b) - \beta l_b \sin(\beta l_b)}. \quad (5-4)$$

5.2.1.2 Tri-beams lateral stiffness

Loading the probe tip at any direction in the lateral plane can be considered as two perpendicular forces acting in the x and y directions at the same time. To simplify the analysis, it is assumed that the probe tip is loaded (F_y) in the y direction. To analyse the lateral stiffness of the tri-beam first, consider each beam as a separate spring to show the relative magnitude and direction of all forces acting on the intermediate body. From this basis, Figure 5-3 (free body diagram) illustrates the scenario when the probe tip is displaced in the y -direction. The cantilever beam springs 2 and 3 will be subjected to the shear force (f_z), bending (M_ϕ) and torsional moments (M_η). Spring 1 is just deflected in the z -direction (as it is aligned with the lateral deflection) with bending about the x -axis, as can be shown in the free body diagram.

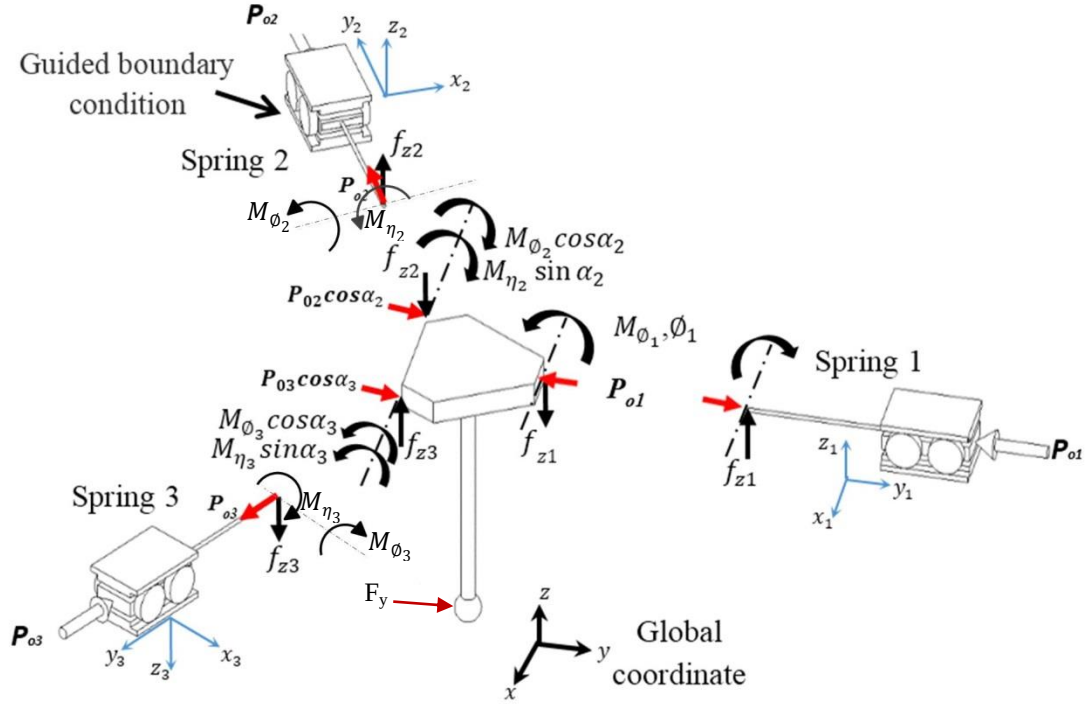


Figure 5-3: Free body diagram for the intermediate body and the spring beams under a lateral force F_y applied at the probe tip. The free body diagram for the spring beams corresponds to the local coordinate for each beam. The shear force (f_z), bending (M_ϕ), torsional moments (M_η) and axial compressive force (P_o) on the intermediate body are analysed corresponding to the global coordinate.

To account for the presence of the intermediate body, the position where each beam is fixed to the intermediate body with respect to the centre of rotation of the intermediate body must be specified. This is illustrated in Figure 5-4(a), where R and S are the distances along the y -axis from the point O to the end of spring beam number 1 and the distance from the point O to end of spring beams number 2 and 3, respectively. Figure 5-4(b) shows moment loads are applied at the centre of the intermediate body. These moments represent the lateral force (F_y) acting at the end of a long stylus (l_s).

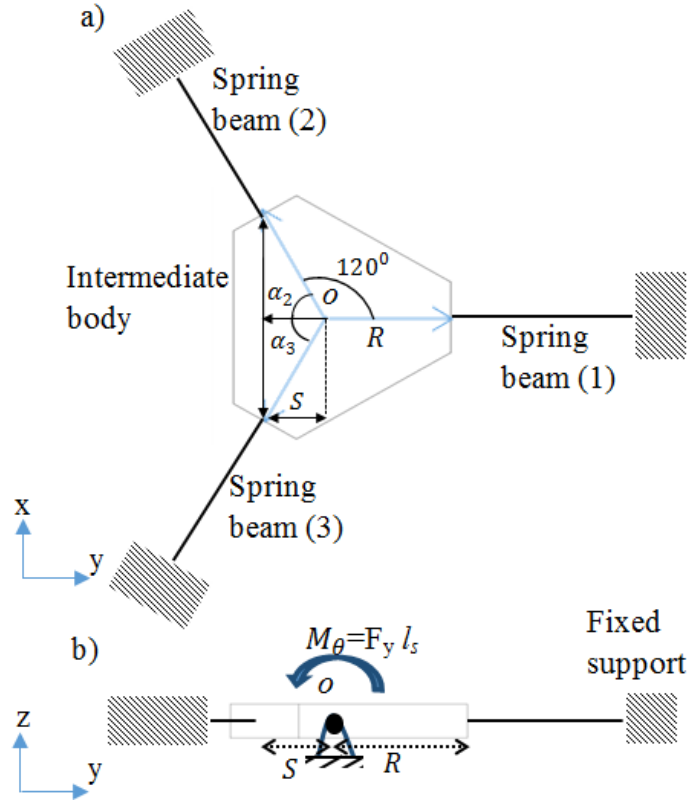


Figure 5-4: (a) Top view of the intermediate body illustrating the distance between centre of rotation (O) about x -axis and the end of spring beams. (b) Illustrating the intermediate body is pinned at the centre where a lateral in-plane displacement of the central intermediate body is zero due to the symmetry about the x and y directions.

The free body diagram under the effect of axial load for the intermediate body and three beams is illustrated in Figure 5-3. Taking moments about the pivot ' O ', which is around the x -axis, in order to derive the lateral stiffness of the probing system yields equation (5-5), where $M_\theta, M_{\phi_1}, M_{\phi_2}, M_{\phi_3}, M_{\eta_2}, M_{\eta_3}, f_{z1}, f_{z2}$ and f_{z3} represent applied resultant moments to the intermediate body; bending moments for spring 1, 2 and 3; torsional moments for springs 2 and 3; and the shear forces for springs 1, 2 and 3, respectively.

$$M_\theta = -M_{\phi_1} + f_{z1}R + M_{\phi_2}\cos\alpha_2 + M_{\eta_2}\sin\alpha_2 - f_{z2}S - M_{\phi_3}\cos\alpha_3 - M_{\eta_3}\sin\alpha_3 + f_{z3}S \quad (5-5)$$

$$- P_{01}d_{z1} - P_{02}\cos\alpha_2 d_{z2} - P_{03}\cos\alpha_3 d_{z3}$$

To include the applied axial compressive force in the shear force f_z and bending moment M_ϕ , the theory of elasticity has been used, see equation (5-1).

Applying the boundary conditions that are specified in Figure 5-5 (spring number 1) yields expressions for shear force (f_z) and the bending moment (M_ϕ) as shown in equations (5-6) and (5-7) respectively.

$$f_z = -\left[\frac{\partial M_\phi}{\partial y} + P_0 \frac{\partial z}{\partial y}\right] = \frac{\tilde{E}I_x\beta^3}{2 - 2\cos(\beta l_b) - \beta l_b \sin(\beta l_b)} \left[d_z [\sin(\beta l_b)] - \frac{\phi}{\beta} [1 - \cos(\beta l_b)] \right] \quad (5-6)$$

$$M_\phi = \tilde{E}I_x \frac{\partial^2 z}{\partial y^2} = \frac{\tilde{E}I_x\beta^2}{2 - 2\cos(\beta l_b) - \beta l_b \sin(\beta l_b)} \left[-d_z [1 - \cos(\beta l_b)] + \frac{\phi}{\beta} [\sin(\beta l_b) - \beta l_b \cos(\beta l_b)] \right] \quad (5-7)$$

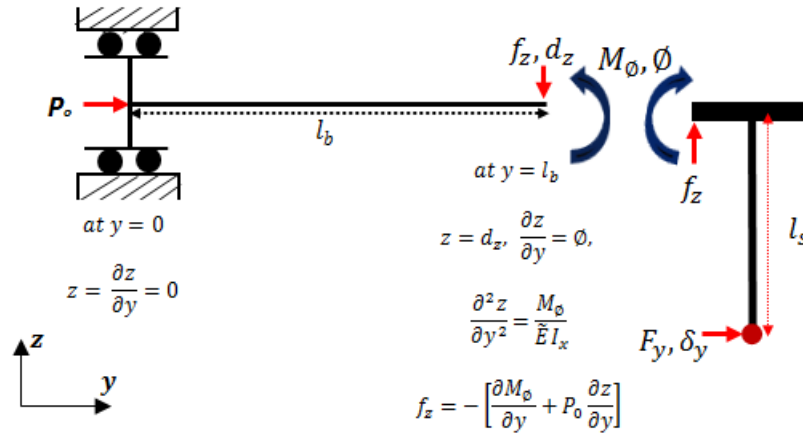


Figure 5-5: Shows the boundary conditions of loading the probe tip at y-direction and free body diagram for the spring 1. This spring is subjected to shear force (f_z) (as it is aligned with the lateral deflection) and bending moment (M_ϕ) about the x -axis. The analysis is considered to find the shear force (f_z) and the bending moment (M_ϕ) at the end of the beam as function of the axial compressive force (P_0).

To account for small torsional deflections of the beams as can be seen in spring 2 and 3, the combined axial and torsional load on the end of the beam is considered. The torsional stiffness can be reduced to zero by increasing the axial compressive force. The total torsional moment for a narrow rectangular section which is affected by compressive axial load is given by [117] , see equation (5-8):

$$M_\eta = \left(\frac{GJ}{l_b} - \frac{I_z}{A_c l_b} P_0 \right) \eta \quad (5-8)$$

where J, G, I_z, l_b, η and A_c denote the polar moment of inertia of the uniform cross-sectional area as in [116], shear modulus, the second moment of area of the beam with respect to z-axis, length of spring beam, angular displacement and the area of cross section, respectively. However, the effect of axial load on the torsional stiffness (M_η) is not appreciable compared with other parameters, as the slope value ($w_b^2 / (12l_b)$) is quite small.

As can be shown in the free body diagram (see Figure 5-3), the torsion and bending moments in springs 2 and 3 have been converted from the local to the global coordinates (see equation (5-5)). Therefore, transferring the linear and angular displacements for both springs 2 and 3 from the local to the global coordinate has been considered to find a relation between parameters, as can be seen in Table 5-1.

Table 5-1: Geometric relationship of the linear and the angular displacements.

Spring	The displacement in z direction(δ_z)	The angular displacement about x direction (ϕ)	The angular displacement about y direction (η)
1	θR	$-\theta$	No exerted torque
2	$-\theta S$	$\theta \cos(\alpha_2)$	$\theta \sin(\alpha_2)$
3	θS	$-\theta \cos(\alpha_3)$	$-\theta \sin(\alpha_3)$

Substituting the stiffness equation and geometric condition equations (5-6), (5-7), (5-8) and the relation between parameters in Table 5-1 into the moment equilibrium equation (5-5) to obtain the torsional stiffness as in equation (5-9):

$$k_{ty}(P_0) = \frac{M_\theta}{\theta} = \frac{3}{2} \left[\frac{\tilde{E} I_x \beta}{2 - 2 \cos(\beta l_b) - \beta l_b \sin(\beta l_b)} [\sin(\beta l_b) [R^2 \beta^2 + 1] - A \cos(\beta l_b) [2R + l_b] + 2R\beta] + \left(\frac{GJ}{l_b} - \frac{P_0 w_b^2}{l_b 12} \right) - P_0 R \right] \quad (5-9)$$

The probe tip displacement can be calculated as a function of the angular displacement ($\delta_y = l_s \sin \theta$) as can be seen in Figure 5-6. The angular displacement is quite small where it can be approximated to ($\theta \approx \delta_y / l_s$). Therefore, the torsional

stiffness about the x and y directions of the suspension structure (equation (5-9)) can be converted to lateral stiffness that includes a stylus, as can be shown in equation (5-10).

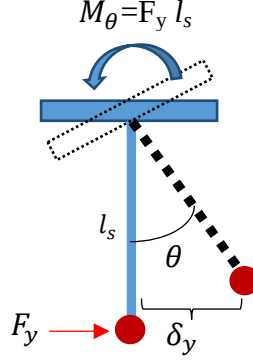


Figure 5-6: Converting the torsional stiffness to the lateral stiffness of the micro-probe CMM.

$$k_y = \frac{F_y}{\delta_y} = \frac{M_\theta / l_s}{\theta l_s} = \frac{k_{t_y}}{l_s^2} \quad (5-10)$$

Where k_y , F_y and δ_y denote the lateral stiffness, probing force in the y -direction and displacement of the probe tip at the y -direction.

To find the initial stiffness for lateral direction, the limit of $k_y (P_o)$ as P_o approaches zero is considered, see equation (5-11)

$$\lim_{P_o \rightarrow 0} k_y (P_o) = \frac{3}{2l_s^2 l_b^3} [\bar{E} I_x [12R^2 + 12l_b R + 4l_b^2] + l_b^2 GJ] \quad (5-11)$$

Then the sensitivity of lateral stiffness that shows the change of the stiffness as a result of change in the axial compressive force can be obtained using a second-order Taylor series expansion [116], which allows a simplified expression to be determined using [118] as given by equation (5-12). Apparently, the effect of the axial compressive force on the initial stiffness depends on the geometric parameters of the suspension structure rather than on the material properties.

$$S_{P_o} = \frac{1.8(R^2) + 1.8(l_b R) + 0.2l_b^2 + 0.125w_b^2}{l_b l_s^2} \quad (5-12)$$

Then the approximation of lateral stiffness can be rewritten as a function of the axial compressive force as given by equation (5-13):

$$k_y(P_o) = \lim_{P_o \rightarrow 0} k_y(P_o) - S_{P_o} P_o \quad (5-13)$$

5.2.2 Vertical and Lateral Stiffness of the Delta-beams design

In this section, an analytical model is used to predict the vertical and lateral stiffness of the delta-beam design. The global coordinate system used throughout this analysis is illustrated in Figure 5-7. The delta-beams design has three segments that hold the intermediate body, where each segment consists of two beams as can be seen in Figure 5-1 (b).

5.2.2.1 Delta-beams vertical stiffness

The derivation of the vertical stiffness for the delta-beams design is similar to the tri-beam design. Loading the stylus tip vertically (F_z) leads to a displacement of the intermediate body in the z-direction without any angular displacement. The vertical force that acts on the probe tip can be divided by the number of beams to obtain the shear force on each beam ($f_z = F_z/6$). The maximum vertical displacement at the end of the beam can be obtained as function of the shear force (see equation (5-3)). Thus, the vertical stiffness can be described by equation (5-14).

$$k_z = 6 \tilde{E} I_x \beta^3 \frac{\sin(\beta l_b)}{2 - 2 \cos(\beta l_b) - A l_b \sin(\beta l_b)} \quad (5-14)$$

5.2.2.2 Delta-beams lateral stiffness

Using symmetry to simplify the analysis of the delta-beams design, half the structure is considered, as shown in Figure 5-7. Displacing the stylus tip in the y-direction will cause a deflection in z-direction and bending around the x-axis for spring 1, while springs 2 and 3 will be subjected to the shear force (f_z), bending (M_θ) and torsional

moments (M_η) as can be seen in Figure 5-7. Following the same procedure as used in section (5.2.1.2), the overall torsional stiffness can be described by equation (5-15).

$$k_{ty} = \frac{M_\theta}{\theta} = 3 \left[\frac{\tilde{E} I_x \beta}{2 - 2 \cos(\beta l_b) - \beta l_b \sin(\beta l_b)} \left[\beta^2 \sin(\beta l_b) \left[r^2 + O^2 + \frac{1}{\beta^2} \right] - \beta \cos(\beta l_b) [2r + l_b] + 2r\beta \right] + \left(\frac{GJ}{l_b} - \frac{P_0 t_b w_b^2}{\beta l_b 12} \right) - P_0 r \right] \quad (5-15)$$

where O and r denote the distance from the centre of the intermediate body to suspension beams and the half-width of the intermediate body, respectively.

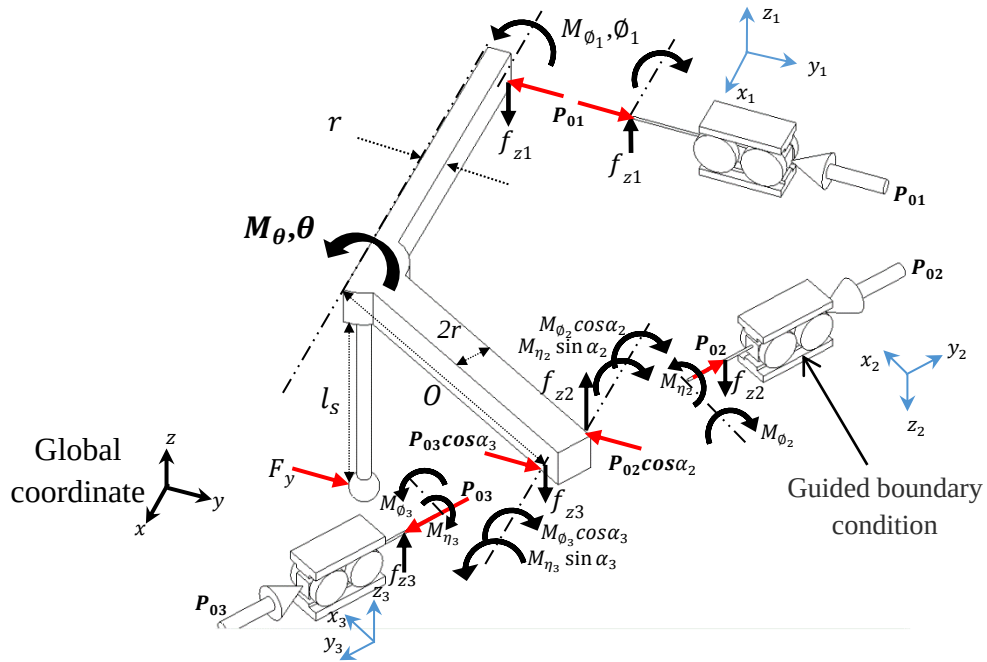


Figure 5-7: Half-model of the delta-beams design showing the guided boundary conditions at the end of each beam, the compressive loads applied to the beams, and all the resulting forces and moments. The free body diagram for the spring beams corresponds to the local coordinate for each beam. The shear force (f_z), bending moment (M_ϕ), torsional moment (M_η) and axial compressive force (P_o) on the intermediate body are analysed corresponding to the global coordinate.

To find the lateral stiffness without the axial compressive force, the limit of $k_y(P_o)$ as P_o approaches zero is considered, see equation (5-16):

$$\lim_{P_o \rightarrow 0} k_y = \frac{3}{l_s^2 l_b^3} [\tilde{E} I_x [12(r^2 + O^2) + 12rl_b + 4l_b^2] + l_b^2 GJ] \quad (5-16)$$

Then the sensitivity of the lateral stiffness that shows the change of the stiffness as a result of change in the axial compressive force can be obtained using a second-order Taylor series expansion, which allows a simplified expression to be determined as given by equation (5-17).

$$S_{P_o} = \frac{3.6(r^2 + O^2) + 3.6(l_b r) + 0.4l_b^2 + 0.25w_b^2}{l_b l_s^2} \quad (5-17)$$

To rewrite and linearise the lateral stiffness as a function of the axial compressive force, the second-order Taylor series has been used and can be given as:

$$k_y = \lim_{P_o \rightarrow 0} k_y - \frac{[3.6(R^2 + O^2) + 3.6(l_b r) + 0.4l_b^2 + 0.25w_b^2]}{l_b l_s^2} P_o \quad (5-18)$$

5.3 Numerical modelling of the suspension structure

The analytical model presented in the preceding section provides a sound theoretical basis for understanding the performance of the smart suspension structure. However, to model the full detail of the structure a numerical solution is required. The combination of numerical and analytical models of system performance also provides increased confidence in the predicted results. Therefore, finite element models of both types of suspension structure were created using the COMSOL 4.3a finite element software. The models use a pre-stressed analysis to compute the stiffness for the structure which are the result of the structural properties (geometry and material properties), as well as the stress developed from applied compressive loads that are used to modulate stiffness.

For each structure, either a force or a moment load was applied to the centre of the intermediate body, and the finite element models were run to determine probe tip displacement. A direct force of 1 mN was applied to assess stiffness in the z-direction

(as defined in Figure 5-1). To assess stiffness in the x and y directions, a moment load was applied. The applied moment was selected to be representative of a lateral load at the stylus tip of 1 mN. As stated in section 4.1.1 the load value of 1 mN was chosen as it represents an upper limit of the range of achieved probing forces for many micro-scale probing systems [16]. According to Pril [74], a stylus length of 4 mm should provide good accessibility and visibility to stylus monitoring cameras common to typical micro-CMM applications. However as this probe is intended to be useful for a range of stylus lengths a range of stylus lengths, 3.8 mm, 6.6 mm and 8 mm, were considered. Therefore three moment loads of 3.8 $\mu\text{N m}$, 6.6 $\mu\text{N m}$ and 8 $\mu\text{N m}$ were applied at the centre of the intermediate body respectively. The geometric parameters that define each of the structures are presented in Figure 5-8 and detailed in Table 5-2. The material properties used are those of titanium, Grade 5 (see Table 5-3). Titanium was chosen for the material of the suspension structure, as it has a high strength to weight ratio, which assists to reduce the effective mass of the suspension structure. Titanium also has a low elastic modulus, which is beneficial for reducing stiffness.

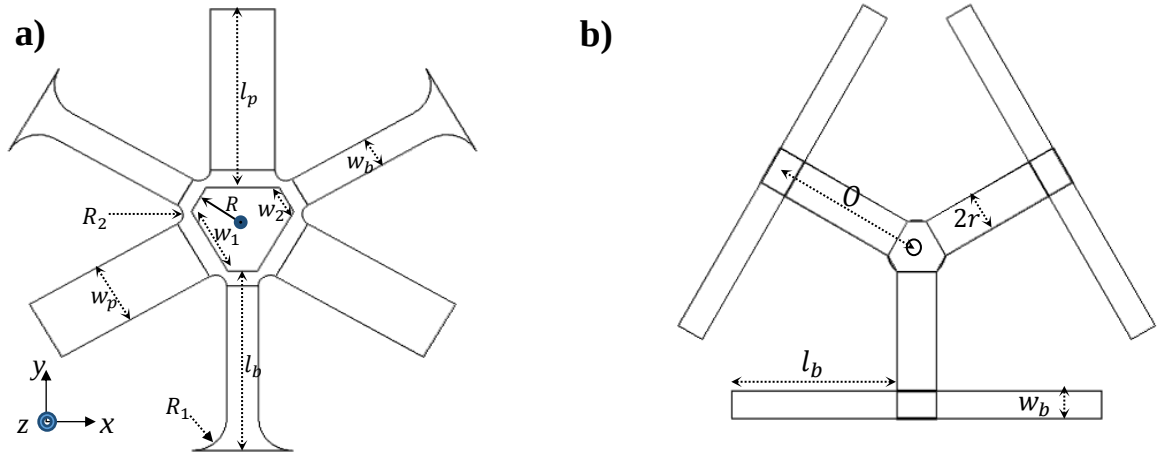


Figure 5-8: The main geometric parameters of the two considered design variants. (a) The tri-beam design and (b) The delta-beam design.

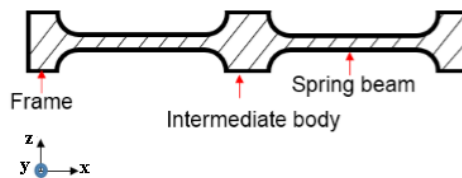


Figure 5-9: Schematic diagram of the cross-section of a beam to demonstrate the symmetry along the z -direction.

Table 5-2: Structure main parameters of the tri-beam and the delta-beam design (define all terms).

Item	Tri-beams design	Delta-beam design
Elastic spring	$l_b = 4215 \mu\text{m}$, $R_1 = 720 \mu\text{m}$, $w_b = 630 \mu\text{m}$, $t_b = 110 \mu\text{m}$ (thickness),	$l_b = 4215 \mu\text{m}$, $w_b = 630 \mu\text{m}$, $t_b = 110 \mu\text{m}$ (thickness)
Sensor paddles	$l_p = 4184.6 \mu\text{m}$, $R_2 = 450 \mu\text{m}$ $w_p = 1300 \mu\text{m}$, $t_b = 110 \mu\text{m}$ (thickness)	----- -----
Intermediate body	$R = 1085 \mu\text{m}$, $w_1 = 1574.2 \mu\text{m}$, $w_2 = 610 \mu\text{m}$	$O = 3926 \mu\text{m}$, $r = 500 \mu\text{m}$

Table 5-3: Material properties of the two configuration designs (titanium, Grade 5) [119].

Elastic modulus, E / GPa	Poisson's ratio, ν	Density, ρ / kg m^{-3}	Yield strength, / MPa
113.8	0.342	4430	880

The finite element model was constructed so as to ensure geometric nonlinearity was accounted for; this was necessary as the effect of the axial compressive force was required to be considered for the modal frequencies to be calculated [120]. In the model, these loads were applied in the form of forced axial displacements of the beam loading points.

A mapped uniform mesh for each suspension beam was created to ensure an equivalent model of the properties in each beam. As such, a mapped quadrilateral mesh has been implemented for all suspension beams, with $154 \mu\text{m}$, $130 \mu\text{m}$ and $27.5 \mu\text{m}$ elements along the length, width and thickness of the beams respectively. These mesh sizes have been chosen based on convergence of the solution. For practicality the more complex structural features, such as the parts that connect beams or the fillets in tri-beams design, were modelled with an unstructured triangular swept mesh. The compressive stress is concentrated on the intermediate body for the tri-beams design compared with the delta-beam design. Thus, the intermediate body has been meshed with fine mesh compared with the delta-beam

design. The mesh and boundary conditions for each model can be seen in Figure 5-10 and Figure 5-11.

To model the response of each structure to compressive loads applied to the beams, a ramped displacement was applied to each of the beam loading points. For comparison with the analytical solution, the axial force can be converted to an equivalent applied displacement using equation (5-19)

$$d_0 = \frac{l_b}{\tilde{E}A_b} P_o \quad (5-19)$$

where $\tilde{E}$, A_b , d_0 , and l_b denote the effective Young's modulus for wide beam, cross section area, axial compressive displacement and beam length, respectively [116].

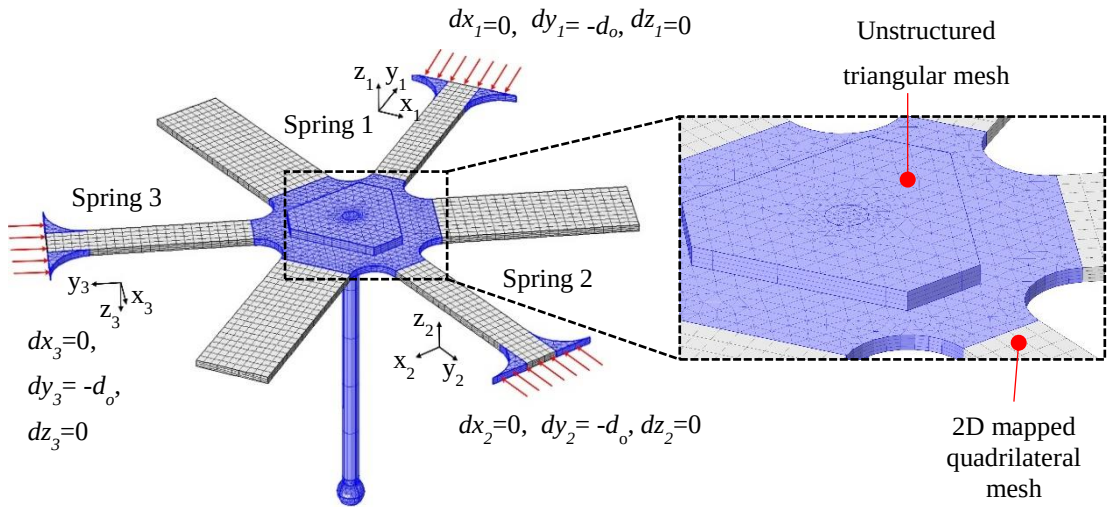


Figure 5-10: Shows finite element mesh that is generated for the tri-beam design and boundary conditions at loading point where all 3 rotational degrees of freedom are considered.

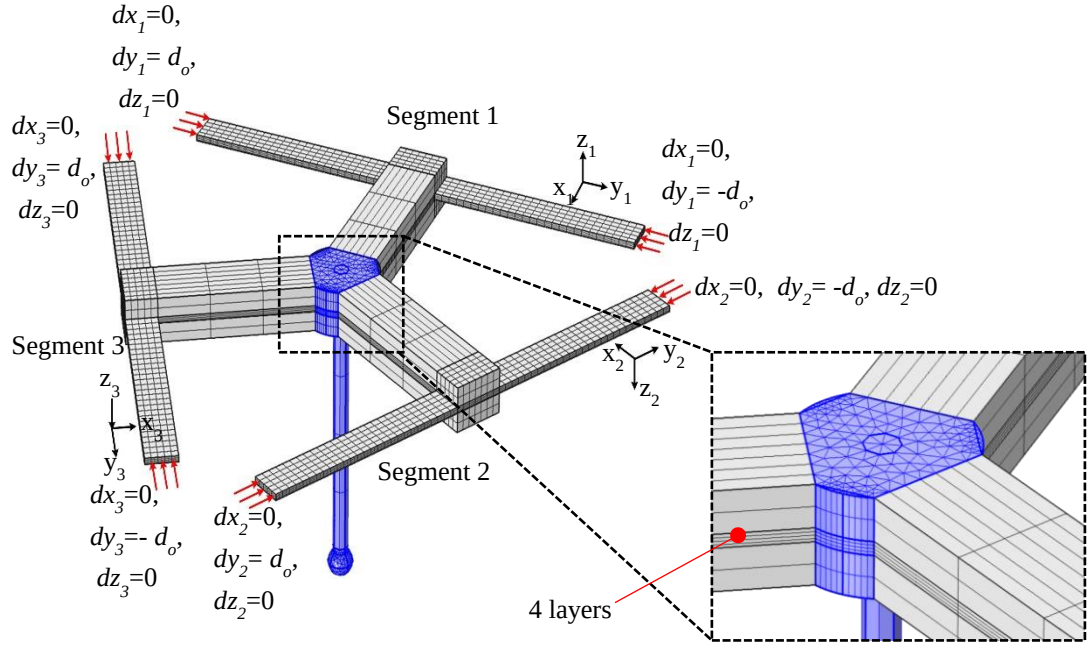


Figure 5-11: Finite element mesh that is generated for the delta-beam design and boundary conditions at loading point where all 3 rotational degrees of freedom are considered. Each segment has two springs beam that share the same local coordinate system.

5.4 Performance analysis

Using both the analytical and numerical models the lateral and vertical stiffness for each structure was evaluated under a range of forced axial displacements. This allowed the relationship between applied compressive load and stiffness to be determined over the full working range of the suspension structures, i.e. from no applied load (maximum stiffness condition) to the beam buckling point (minimum stiffness condition). The analysis was completed for the stiffness in both the vertical and lateral directions, and as such in this analysis the terms k_z and k_y are used to describe vertical and lateral stiffness respectively.

Three key factors were considered: firstly, what impact changing parameters of the suspension structure has on performance; secondly, how using two models (analytical and FEA) has provided confidence; and thirdly, what impact changing stylus length has on performance of the probing system.

5.4.1 Stiffness parametric study on suspension structure for tri-beam and delta-beam design

The suspension structure consists of two key elements. Firstly, the spring beams which connect the intermediate body to the frame of the probing system. It plays an important role in determining the flexibility of the system. The intermediate body is the second part where the stylus can be mounted to transfer the displacement from the probe tip to the spring beams. To study the impact of changing the parameters of the spring beams and the intermediate body on performance, each parameter value in Table 5-2 is changed by $\pm 20\%$ for both designs. In each study, one parameter was changed and the rest remained constant. Figure 5-12 illustrates the parametric study of the spring beam on the performance of the probing system. The vertical and the lateral stiffness versus axial displacement for both designs are plotted with changing thickness, width and length of beam. Results from both designs are presented, considering the case of a suspension structure with a 3.8 mm long stylus.

Changing the thickness of the spring beam has major effect on the probing system for both tri-beam and delta-beam designs (see Figure 5-12(a) and (d), respectively). The vertical and lateral stiffness are directly proportional to the cubic power of the thickness (t). Even though reducing the thickness of the spring leads to decrease the stiffness, it is limited by manufacturing technique. Very thin beams become difficult to manufacture with accuracy and consistency along their length. Figure 5-12(b) and (e) show the impact of changing the width of the spring beam for both designs. It is obvious that the vertical and the lateral stiffness change linearly with the width of the spring beam and buckle at the same point. Another observation is that there is no significant change in the slope curve for both vertical and lateral stiffness with changing the beam thickness. The sensitivity of lateral stiffness as described in equation (5-12) and (5-17) shows that the effect of the axial compressive force on the initial stiffness does not depend on the beam thickness and material properties.

The beam length is inversely proportional to the vertical and lateral stiffness for the tri-beam and delta-beam designs (see Figure 5-12(c) and (f), respectively). Decreasing the beam length to 3372 μm leads to an increase in the vertical stiffness by $\sim 100\%$ for both the tri-beam and delta-beam designs. Increasing the beam length in order to reduce the stiffness can lead to increasing inertial loads. These inertial

loads must not result in stylus tip displacements that are mistaken for real surface contacts. Another observation is that the intersection point of the curves must exist to achieve the isotropic stiffness. In order to achieve the intersection point, the initial stiffness in the vertical direction has to be greater than in the lateral direction and the buckling point for the vertical direction has to be less than for the lateral direction, and vice versa. For example, an intersection point between the vertical and the lateral stiffness for the delta-beam design does not exist when the beam length is increased to 5058 μm (see Figure 5-12 (f)).

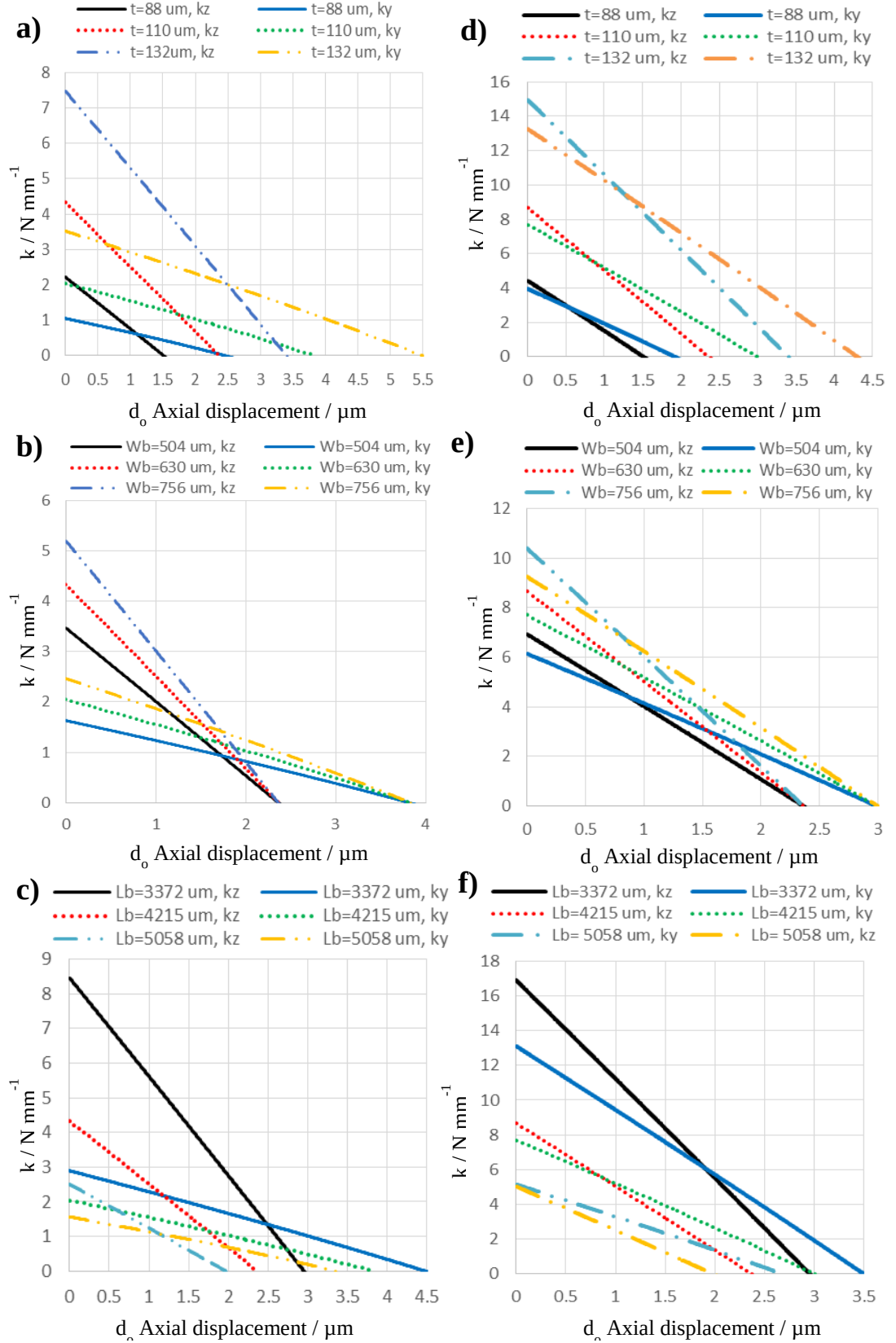


Figure 5-12: Stiffness versus axial displacement determined by analytical model. (a), (b) and (c) varying thickness (t), width (w_b) and length (l_b), respectively for the tri-beam design, and (d), (e) and (f) thickness (t), width (w_b) and length (l_b), respectively, for the delta-beam design.

Figure 5-13 shows the impact of changing the parameters of the intermediate body where results from both designs are presented considering the case of a suspension structure with a 3.8 mm long stylus. It is obvious that the vertical stiffness is not affected by changing the parameters of the intermediate body. The lateral stiffness of the tri-beam design is directly proportional to the radius of the intermediate body as can be seen in Figure 5-13(a). Increasing the radius of the intermediate body assists to have curves for both the vertical and lateral stiffnesses that are closely matched in slope. This is due to the buckling point decrease and the initial stiffness increase, where it can be an advantage as discussed in detail in section 5.4.3. However, increasing the radius of the intermediate body can lead to an increase in the inertial loads. The effect of changing the half width of each arm in the intermediate body (r) is not appreciable compared to the distance between the centre of the intermediate body and the spring beam for the delta-beam design, as can be seen in Figure 5-13(b) and (c).

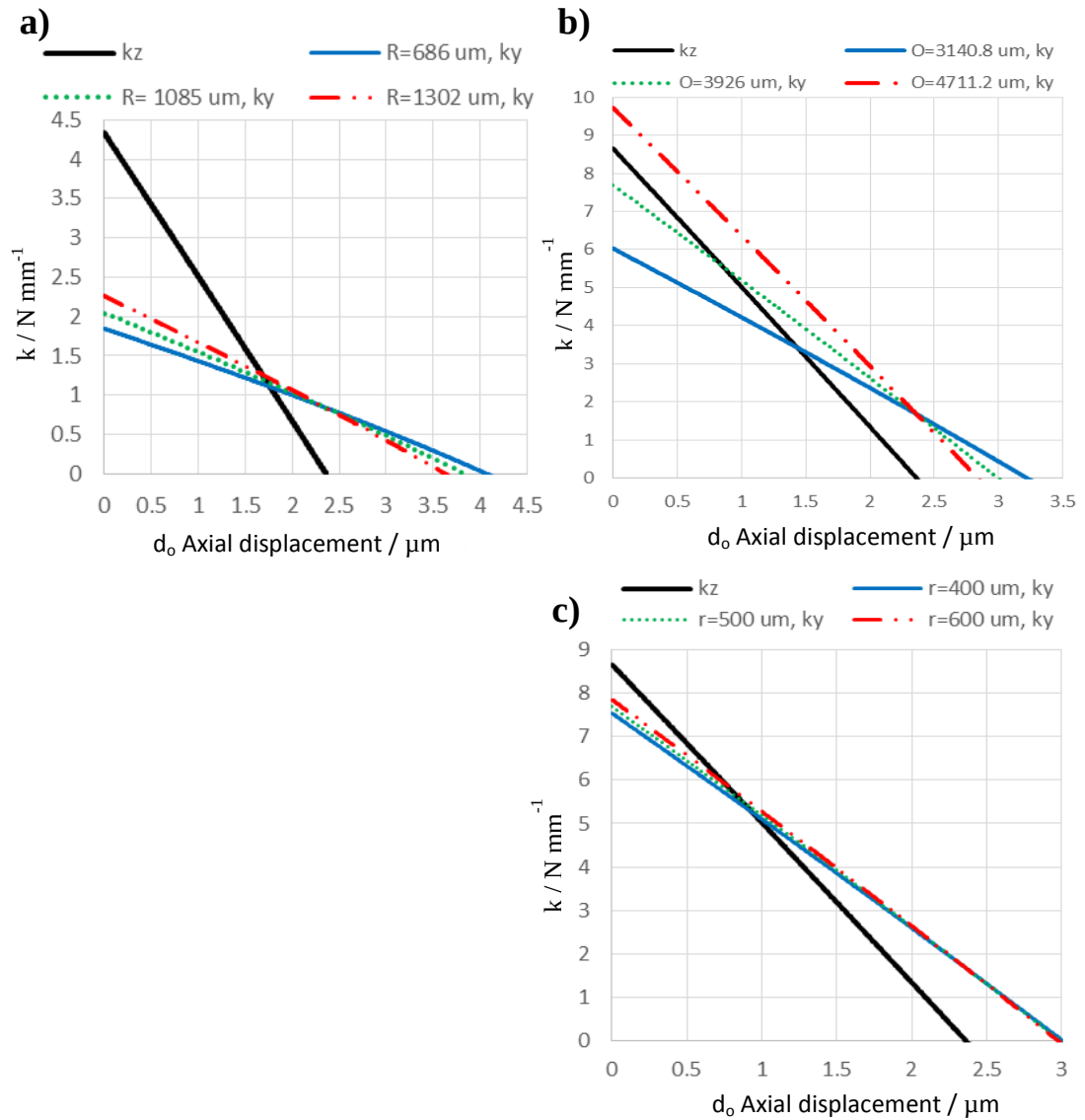


Figure 5-13: Stiffness versus axial displacement determined by analytical model (a) A plot showing the effect of changing the radius of the intermediate body (R) for the tri-beam design. (b) and (c) show change in the distance between the centre of the intermediate body and the spring beam (O) and change in the half width of each arm (r) for the delta-beam design, respectively.

5.4.2 Comparison between the analytical and numerical models

The second case to consider was a direct comparison between the analytical and numerical models. Therefore results from both models are presented considering the case of a suspension structure with a 3.8 mm long stylus. This direct comparison of the two models shows that the trends are the same, leading to high levels of

confidence that the models are valid. At first inspection the curves generated by both models as shown in Figure 5-14 and Figure 5-15 are in close agreement. More detailed analysis shows that the maximum deviation between the models is at the zero load condition, where stiffnesses in the vertical and lateral directions for tri-beams and delta-beam structures are all within 9 %, see Table 5-4. The second observation is that the initial stiffness of the FEA model is greater than the analytical solution for the tri-beam design compared to the delta-beam design. This is due to the simplification that has been made to the analytical model where the fillets in the tri-beam design are not considered. This leads to make the stiffness in the FEA model higher than for the analytical solution. It can also be seen that the error at the buckling point is around 9.1 %. However, the difference between the analytical and FEA models for the delta-beam design tends to decrease as the load is increased. Comparing the predicted axial displacement at the first and second buckling points shows a good agreement, with predictions of buckling load falling within 2 %, see Table 5-4. The direct comparison of the two models provides a high confidence that the behaviour of the suspension structure is well understood and the models are both effective. This error can be due to several reasons. Firstly, the stiffness of the intermediate body has not been included in the analytical model as it is assumed that the stiffness is not the same order of magnitude as that of the spring beams. Secondly, nonlinear FEA is approximate solution where the response of the structure, and therefore predicted stiffness is obtained through an iterative process.

The approximate solution for estimating the lateral stiffness using equations (5-13) and (5-18) is also plotted in Figure 5-14 and Figure 5-15. Obviously, the approximate solution coincides with the exact solution at low axial compressive force due to approximations to the second-order Taylor polynomial.

Table 5-4: Comparison between theoretical and finite element analysis (COMSOL) results for the tri-beams and delta-beam design.

	Tri-beam design			Delta-beam design		
	Analytical	FEA	Error	Analytical	FEA	Error
$k_z / \text{N mm}^{-1}$ at $d_o=0$	4.33	4.60	5.8 %	8.65	7.92	8.4 %
$k_y / \text{N mm}^{-1}$ at $d_o=0$	2.04	2.09	2.4 %	7.69	7.04	8.4 %
d_o at the first buckling point / μm	2.36	2.63	9.1 %	2.36	2.40	1.6 %
d_o at the second buckling point / μm	3.83	4.10	6.6 %	2.99	3.04	1.6 %

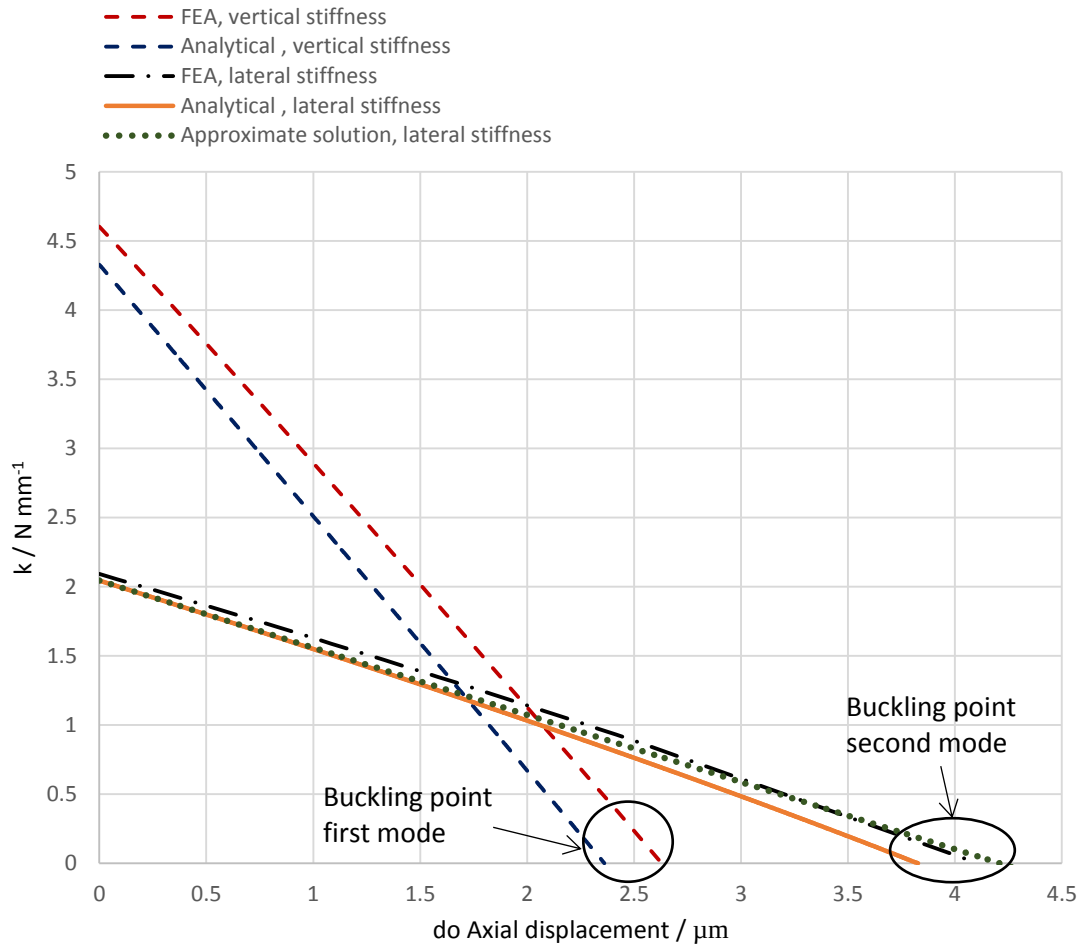


Figure 5-14: A plot of the vertical to lateral stiffness as a function of the axial displacement (d_o) for the tri-beam designs, with 3.8 mm stylus length, as calculated in FEA analysis (COMSOL), analytical and approximate solution.

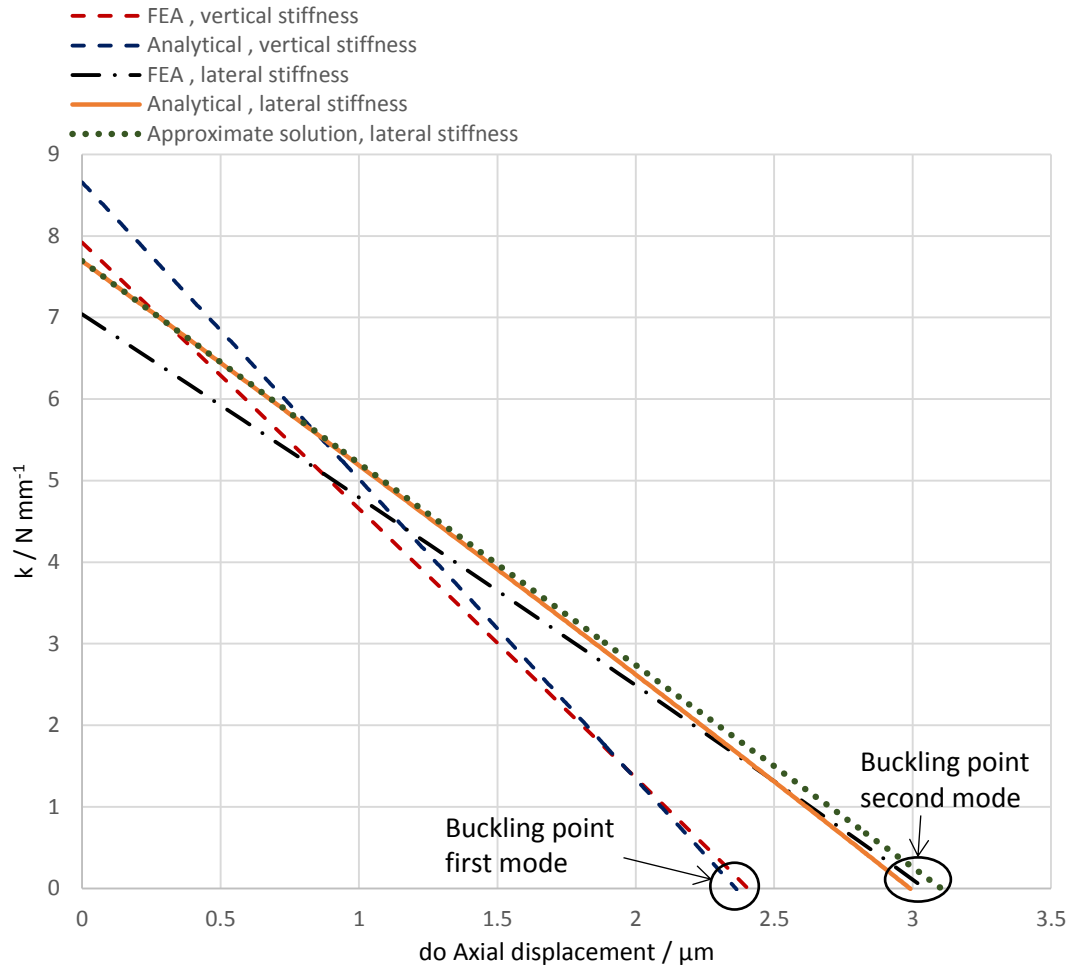


Figure 5-15: A plot of the vertical to lateral stiffness as a function of the axial displacement (d_o) for the delta-beam design, with 3.8 mm stylus length, as calculated in FEA analysis (COMSOL), analytical and approximate solution.

5.4.3 Minimising the anisotropy of the probing system

The previous result showed that the finite element analysis provides a reasonable estimation to stiffness of the two design approaches compared to the simplified analytical model. Therefore, there is good confidence that the finite element analysis can be used for further investigation into the means for controlling anisotropy.

The third investigation is to study the impact of three different stylus lengths on the performance of the probing system. Thus, to allow a range of stylus lengths to be considered, three moment loads of 3.8 $\mu\text{N m}$, 6.6 $\mu\text{N m}$ and 8 $\mu\text{N m}$, were applied at the centre of the intermediate body. These moments represent a force of 1 mN acting at the end of a 3.8 mm, 6.6 mm and 8 mm long stylus respectively, as mentioned in

(section 5.3). Using the finite element models, the vertical (k_z) and lateral (k_y) stiffness for each design were evaluated for a range of axial loads applied to the suspension structure beams. In the model, these loads were applied in the form of forced axial displacements of the beam loading points. The geometric parameters that define each of the structures are presented in Figure 5-8 and detailed in Table 5-2. The material properties used are those of titanium, (see Table 5-3). The results from this analysis are illustrated in Figure 5-16, which shows the relationship between applied axial displacement and stiffness, for three stylus lengths.

The effect of load on lateral and vertical stiffness for both structures is not equal. From Figure 5-16 this is clearly observed by the change in slope of the curves for vertical and lateral stiffness. An important consequence of this is that an intersection point of the curves must exist, and at this point the probe stiffness is isotropic. In, the Figure 5-16 isotropic stiffness point with a 3.8 mm stylus length occurred at 1140 N m^{-1} and 5100 N m^{-1} for the tri-beams and delta-beam design respectively (see Table 5-5). This is an important observation, as achieving isotropic stiffness by geometry design alone is not easy and in some cases not possible. Therefore, a major benefit for probe sensors with tuneable stiffness is the ability to operate at the isotropic stiffness point. It is also evident from Figure 5-16 that changing the stylus length leads to a shift in the position of the isotropic stiffness point. Therefore, for any future application of variable stiffness suspension structures, consideration would be required to ensure the correct isotropic stiffness point was achieved for each stylus length required.

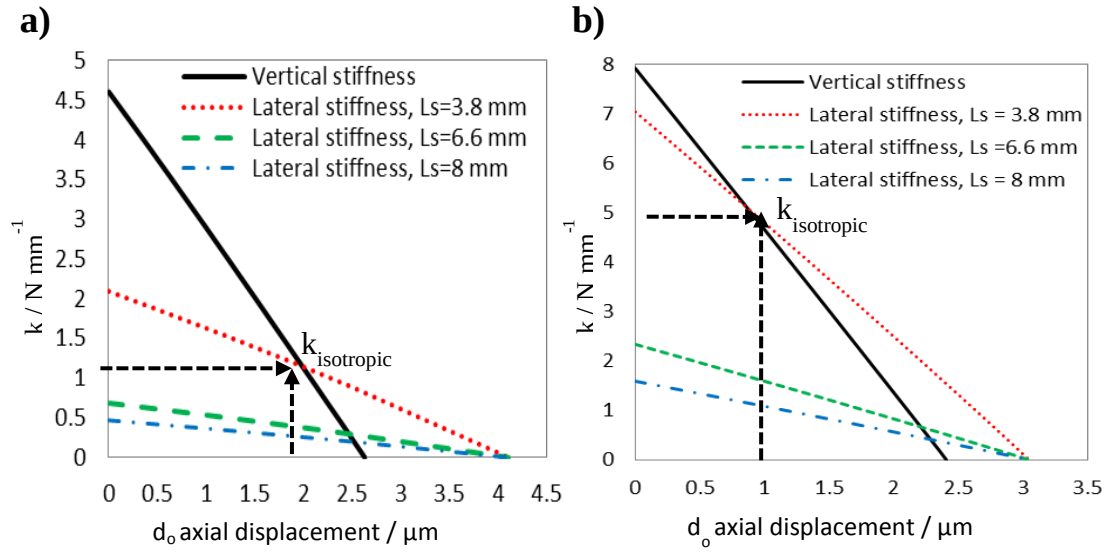


Figure 5-16: Stiffness versus axial displacement determined by FEA for (a) the tri-beam and (b) delta-beam design. Different effective stylus lengths ($L_s = 3.8$ mm, 6.6 mm and 8 mm) have been considered.

It is also advantageous to design a suspension such that the curves for both the vertical and lateral stiffness that are closely matched in slope. This means that, for a given difference in vertical and lateral stiffness, the range of suitable applied loads is increased. To illustrate this, Figure 5-17 shows the ratio of the vertical to the lateral stiffness as a function of applied load. From Figure 5-17, the delta-beam design (3.8 mm stylus length) remains within a stiffness mismatch of 10 % or less for a larger range of applied displacements as compared to the tri-beam design.

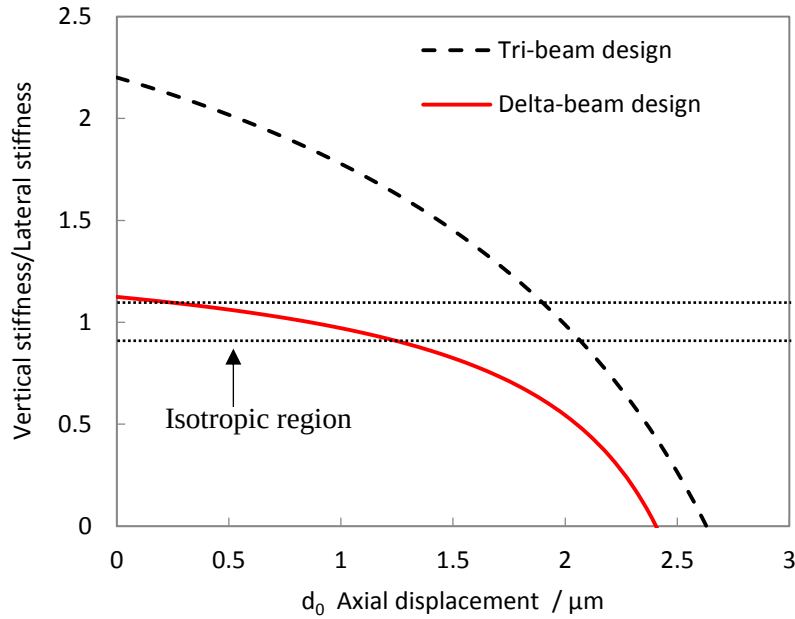


Figure 5-17: A plot of the ratio of the vertical to lateral stiffness as a function of the axial displacement (d_o) for both delta- and tri-beam designs, with 3.8 mm stylus length.

An additional consideration that can be investigated with these models is the operating range of the probe suspension. The measuring range of the probing system should be greater than the over-travel distance that the micro-CMM needs to stop in after receiving the contact signal. The critical factor to consider for this is that because when displacing the probe tip by a certain amount, the stress on the spring beam exceeds the yield stress which can cause damage to the probing system. To avoid damage to the probing system, the von Mises yield criteria with safety factor of 2 is considered where the upper limit of the stress on the spring beam is 440 MPa. Table 5-5 shows the maximum (critical) probe tip displacement at the isotropic point in vertical and lateral directions for three different stylus lengths. Clearly, the measurement range for the tri-beam design is larger than for the delta-beam design for different stylus lengths. This is due to the fact that tri-beam design has lower stiffness than the delta-beam design.

Table 5-5: Finite element analysis results of isotropic stiffness for the tri-beams and delta-beam design.

Stylus length, l /mm	Tri-beams design				Delta-beam design			
	$k_{isotropic}$	d_o	Critical displacement at the probe tip / μm		$k_{isotropic}$	d_o	Critical displacement at the probe tip / μm	
	/Nmm ⁻¹	/ μm	Vertical	Lateral	/Nmm ⁻¹	/ μm	Vertical	Lateral
3.8	1.14	1.98	168	200	5.1	0.86	133	121
6.6	0.302	2.45	170	340	0.664	2.2	152	238
8	0.201	2.5	173	410	0.411	2.28	154	290

5.5 Measurement principle of the suspension structure deformation

To sense movement of the suspension structure during operation of the probe in the vertical and lateral directions, non-contact displacement transducers, such as capacitive [121] and optical [66] types, can be used as noted in section 2.3 of chapter 2. The displacement transducer must be positioned such that its datum is not a part of flexible suspension assembly. For example, the intermediate body has a stiffness that is two orders of magnitude greater than the spring beams. Measurements taken from references on the intermediate body are therefore more reliable while maintaining the sensitivity to the displacement at the probe tip. This must be balanced against the effort to minimise the probes effective mass.

The intermediate body has three cantilevered paddles for tri-beam design, which provide a suitable surface for measurement with non-contact displacement transducers (see Figure 5-18(a)). Another consideration is that the sensor paddles for the tri-beam design must have a high resonant frequency which is greater than the effective vertical and lateral frequencies for the probing system. This is to ensure that the sensor paddle is not excited to vibrate by any motions typically observed in the system. The resonant frequency of the cantilevered paddles (f_p) was calculated at 5.5 kHz based on equation [122] (5-20) where the geometric parameters and the material properties are defined in Table 5-2 and Table 5-3.

$$f_p = \frac{1}{2\pi} \sqrt{\frac{35 \bar{E} t_p^2}{33 l_p^4 \rho}} \quad (5-20)$$

The displacement sensor can be located above the arms of the intermediate body for the delta-beam design as can be seen in Figure 5-18(b), and additional paddles are therefore not required.

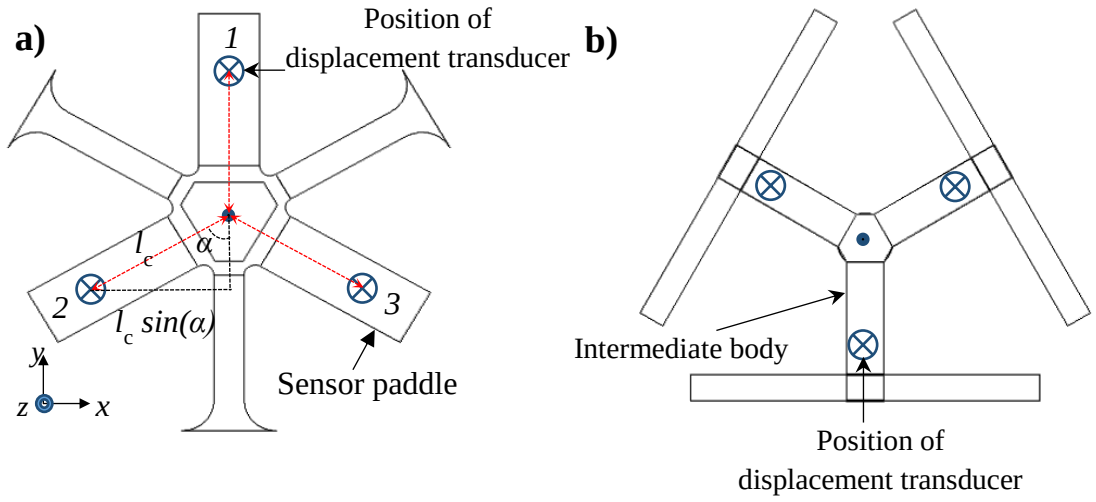


Figure 5-18: Schematic view showing the position of the displacement transducer for (a) the tri-beam design and (b) the delta-beam design.

5.5.1 Displacement of the stylus tip in the vertical direction

The working principle of the sensor unit that was used in this project to detect the contact force is based on measuring the gap distance between the capacitive sensor and the sensor paddles. Figure 5-19 shows the position and the alignment of three capacitive sensors. Theoretically, loading the probe tip in the vertical direction results in changing the gap distance equally for three capacitive sensors which can be measured as seen in equation (5-21):

$$\delta_z = (\Delta S_1(V_{01} - V_1) + \Delta S_2(V_{02} - V_2) + \Delta S_3(V_{03} - V_3))/3 \quad (5-21)$$

where δ_z , ΔS_1 , V_{01} and V_1 denote the vertical displacement at the probe tip, sensor sensitivity (m/V), the voltage reading at the initial gap distance and the voltage reading after the displacement for the first capacitive sensor.

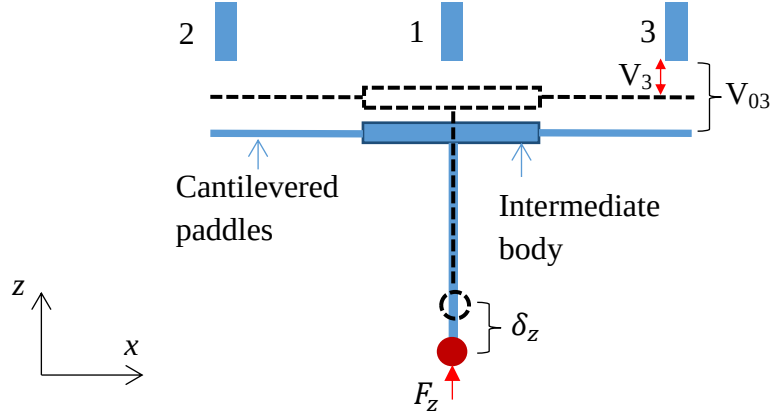


Figure 5-19: Layout of the three displacement sensors used to measure the displacement of loading the probe tip at vertical direction.

5.5.2 Displacement of the stylus tip in the lateral direction

Loading the probe tip in the lateral direction leads to rotation in the intermediate body. To measure the lateral displacement, the angle of rotation (θ) has to be considered. Figure 5-20(a) shows the probe tip loaded in the x -direction where the rotation can be measured using the second and third sensor (see equation (5-22)). The capacitive sensor (1) is located in the middle, which makes it less sensitive to the x -direction.

$$\delta_x = \frac{l_s}{2} \left[\sin \left(\tan^{-1} \left(\frac{\Delta S_2 (V_{02} + V_2)}{l_c \sin(60)} \right) \right) + \sin \left(\tan^{-1} \left(\frac{\Delta S_3 (V_{03} - V_3)}{l_c \sin(60)} \right) \right) \right] \quad (5-22)$$

where δ_x , l_s , ΔS_2 , V_{02} , V_2 and l_c represent the displacement of the probe tip in the x -direction, effective length of the stylus, sensor sensitivity (m/V), the voltage reading at the initial gap distance, the voltage reading after the displacement for the second sensor and the distance between the centre of the rotation and the capacitive sensors (l_c).

Loading the probe tip in the y -direction leads the three sensors to read the rotation of the suspension structure as can be seen in Figure 5-20(b). It is obvious that sensors 2 and 3 are located to output one-half of sensor 1 due the distance in the y -direction between the centre of the rotation and the capacitive sensors. The displacement of the probe at the y -direction can be measured as shown in equation (5-23).

$$\delta_y = \frac{l_s}{3} \left[\sin \left(\tan^{-1} \left(\frac{\Delta S_1 (V_{01} - V_1)}{l_c} \right) \right) + \sin \left(\tan^{-1} \left(\frac{\Delta S_2 (V_{02} + V_2)}{l_c \cos(60)} \right) \right) + \sin \left(\tan^{-1} \left(\frac{\Delta S_3 (V_{03} + V_3)}{l_c \cos(60)} \right) \right) \right] \quad (5-23)$$

The noise of the probing system in the z -direction depends on many factors, such as distance between sensor and target, surface roughness and the area of capacitive sensors. However, the noise in the lateral direction is greater than in the vertical direction, as can be observed in equations (5-22) and (5-23). The noise in the lateral displacement signal relies on three parameters: noise in z -direction, the effective length of the stylus (l_s), and the distance between the centre of the intermediate body and the capacitive sensor (l_c). The lateral noise can be reduced by increasing the length of (l_c) or even reducing the length of (l_s).

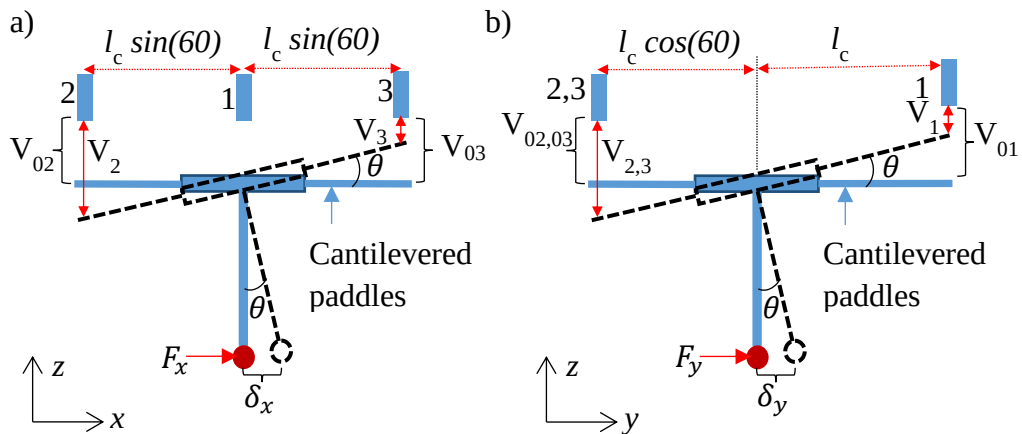


Figure 5-20: Layout of the three displacement sensors used to measure the displacement of loading the probe tip at (a) x -direction and (b) y -direction.

5.6 Chapter summary

Two possible configurations of probe sensor have been described, and a model of their performance is presented. Analytical and finite element analysis has been carried out using COMSOL. It has been shown that the stiffness for both configurations can be theoretically reduced to zero, where the difference between the two the solutions is less than 9 %. As such the analytical model provides a useful tool to perform parametric studies of variable stiffness suspension structures. Using the model it is demonstrated that the effect of the axial displacement on lateral and vertical stiffness for both structures is not equal. This is clearly observed by the change in slope of the curves for vertical and lateral force extension curves.

As a result of the different sensitivities to applied load, there is intersection point of the curves must exist to achieve the isotropic stiffness. The probe is able to operate in a fully isotropic stiffness regime without being limited by stylus length where three different stylus lengths (3.8 mm, 6.6 mm and 8 mm) were considered. It is illustrated that it is advantageous to have curves for both the vertical and lateral stiffness that are closely matched in slope to extend the isotropic stiffness regime. The maximum measuring range at isotropic stiffness (3.8 mm stylus length) was shown that the probe can be displaced of 168 μm and 133 μm at the vertical direction for tri-beam and delta-beam design.

The sensing system to measure the movement of the suspension structure is presented for both designs. Three capacitive sensors are mounted above the cantilevered paddles for the tri-beam design and the intermediate body for the delta beam design to measure the displacement of the probe tip.

To provide and validate the concept of a probing system that makes use of a variable stiffness suspension structure, the tri-beams design will be fabricated and basic evaluation of that device is presented in the next chapter. To achieve the isotropic stiffness of around 300 Nm^{-1} with 6.6 mm stylus length, the spring beam that has thickness of 110 μm , width of 630 μm and length of 4 mm will be implemented into the embodied design which made from titanium alloy (grade 5).

Chapter 6 Embodiment, fabrication and experimental validation of a variable stiffness suspension

To validate the concept of modulating the stiffness of the micro-probing system and achieving the isotropy of the probing force that are described in chapter 5, the tri-beam design has been manufactured and tested. A range of manufacturing techniques were considered in this chapter to build the prototype of the suspension structure for the micro-probing system. To demonstrate a significant improvement in device flatness and analyse the resulting structure, surface 3D microscopy was used.

Following the manufacture of the suspension structure, the assembly of this part with three piezoelectric micro actuators, capacitive sensors and stylus is considered in this chapter.

Experimental procedures used in measuring the vertical and lateral frequencies, which are related to vertical and lateral stiffness, will be presented. These procedures include a description of the setup, experimental method and the procedure that was used to analyse data. Moreover, the experimental data is compared with FEA analysis in order to validate the simulation, and understand the limits of the manufacturing process.

6.1 Design considerations for integrating a micro-actuator with the suspension structure

The working principle of the device is based on a beam that is subjected to a compressive load, which is used to reduce the stiffness of the centre point of the beam. Theoretically it should be possible to reduce stiffness in this way to zero, which occurs at the buckling load for the beam as shown in Chapter 5. To make it possible to reduce stiffness close to this point in a controlled way requires a precisely manufactured miniature mechanical structure. For example, load applied to the beam must be constrained such that the load vector is coincident with the axis of the beam. In addition, the structure must be symmetrical and free from any bias effects that may tend to predispose the beam to bend as a result of an axial load as illustrated in Figure 4-2. To allow load to be applied to a suspension beam in a controlled manner, there are three main features of the design as can be seen in Figure 6-1. These are the

delicate inner suspension structure, the outer bulkier frame (compliant guide mechanisms) and the micro-actuators. The inner suspension system consists of an intermediate body, three suspension beams and three sensor paddles. The probe stylus is rigidly fixed to the intermediate body that is supported by three suspension beams. When the stylus tip is deflected from its neutral position it exerts a moment on the intermediate body and changes its orientation. The changes in orientation are also experienced by the paddles and sensors (capacitive) located above them that sense the change and signal to the computer that the surface of the specimen has been contacted.

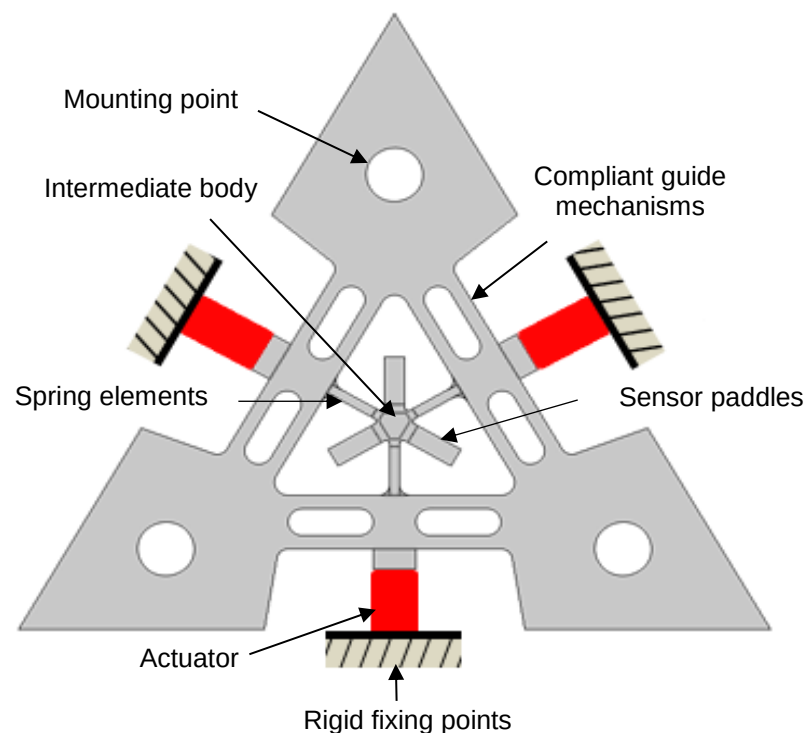


Figure 6-1: Position of the three piezoelectric actuators used to apply load to the spring beams via compliant mechanisms.

The outer frame supports the spring beams and provides sturdy mounting points at the corners of the structure. These mounting points allow the structure to be assembled with the displacement sensors and mounted to a coordinate measurement machine, or in the case of the work undertaken here, to a test rig. The outer frame also provides well-defined load paths to transfer compressive loads to the ends of the spring beams. This is achieved by the use of the three compliant guide mechanisms that are directly connected to each spring beam. The guide mechanisms must be

designed to be flexible in one degree of freedom, and relatively stiff in all others. This allows them to selectively transmit uniaxial displacements along the axis of the spring beam. Figure 6-2 shows the schematic diagram of the compliant mechanism. The stiffness of the compliant mechanism in any direction can be calculated as in equations (6-1) and (6-2), respectively:

$$C_y = 4E \frac{w_c t_c^3}{l_{cc}^3} \quad (6-1)$$

$$C_z = 4E \frac{t_c w_c^3}{l_{cc}^3} \quad (6-2)$$

where C_y , C_z , t_c , w_c , l_{cc} and E represent the stiffness of the compliant mechanism in the y and z directions, beam thickness, beam width, beam length and elastic modulus of the compliant mechanism.

To make the stiffness of the compliant mechanism in the y direction (along the axis of the spring beam) flexible compared to the vertical stiffness, the ratio of the beam width to thickness $(w_c / t_c)^2$ must be greater than one. For example, if the width and thickness are 2 mm and 1 mm respectively, the stiffness in the z direction will be 4 times greater than the stiffness of the complaint mechanism in the y direction. Also, the parallel beams in the compliant mechanism assist to increase the torsional stiffness about the x-axis compared with a single beam which has a thickness and width equal to $2 t_c$ and w_c , respectively. This can help to increase the rigidity of the frame to prevent movement while the probe tip is in contact with the measurement surface.

The third main component is the micro-actuator. The compliant mechanisms are actuated by piezoelectric stacks with at least 4 μm maximum displacement based on the estimation in chapter 5, and positioned in intimate contact with the load application points clamped against rigid fixing locations on the test rig; the position of these actuators can be seen in the top down view shown Figure 6-1. A further requirement is that the piezoelectric actuators must have a sufficient stiffness on their longitudinal axis to avoid a variation of the axial displacement during the operation.

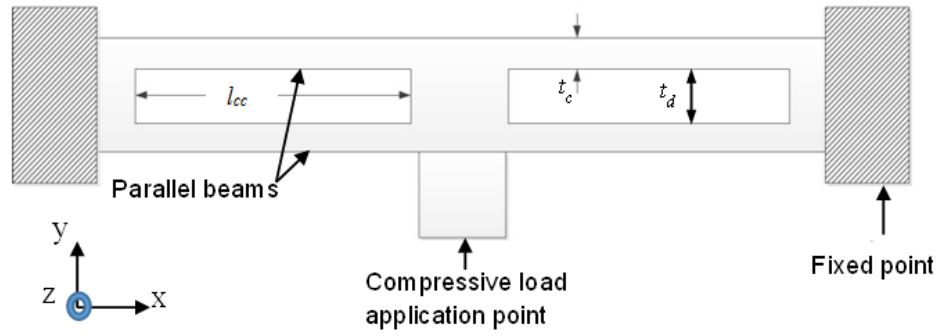


Figure 6-2: Schematic diagram showing the compliant mechanism which consists of two parallel beams.

6.2 Manufacturing of the smart suspension structure

Initial prototype devices were manufactured using a combination of micro-milling and micro-electro-discharge-machining (μ EDM). While these manufacturing techniques were able to provide valuable insight for device prototyping, they were subject to some limitations. They were found to be time consuming, costly, and also resulted in the development of unwanted residual stress in the structures. The prototypes of the suspension structure have been manufactured in two steps. Firstly, the micro-milling machine (Kern EVO CNC Machine) has been used to mill the titanium plate of thickness 2 mm from two sides to reduce the thickness of the spring beams and sensor paddles down to 100 μ m. The intermediate body was reduced to 150 μ m. Then three pinholes were made to insert the wire of the EDM (Sodick AP200L) that has a 100 μ m diameter in order to cut the plate into the required shape, as can be seen in Figure 6-3(a). Each hole was used to shape one third of the suspension structure and the process was repeated three times in order to complete the final structure (see Figure 6-3 (b)). Surface 3D microscopy (Bruker Contour GT) was used, to analyse the resulting structure, to demonstrate a significant improvement in device flatness. The 2.5 X objective was used and stitching measurements were setup to measure the suspension structure. The vertical scanning interferometry (VSI) measurement mode was chosen as it is suitable and effective for rough surfaces [123]. The surface 3D microscope was operated with vertical and lateral resolution of around 6 nm [124] and 7 μ m, respectively. Figure 4-6 shows a contour plot of a suspension structure that was manufactured using conventional micro-milling and micro-electro-discharge-machining. It clear that this method was not successful as the milling force and heat deformed the structure and introduced

residual stresses into the part, where the centre of the part is deflected upwards by around $350\text{ }\mu\text{m}$ in the vertical direction, as can be seen in Figure 6-4.

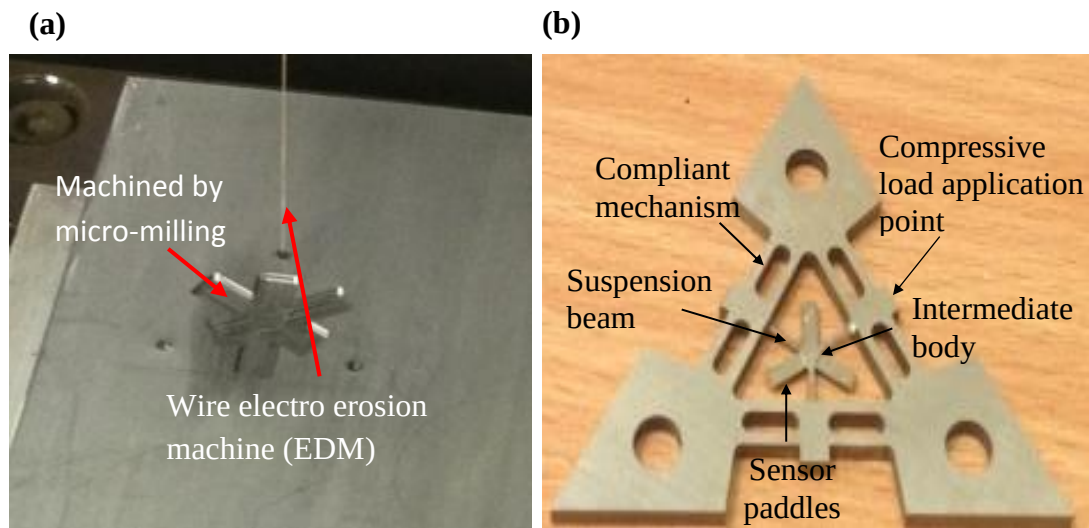


Figure 6-3: (a) Shows the micro-milling and erosion process to fabricate the suspension structure for the micro-tactile probe. (b) Initial prototypes of the suspension structure.

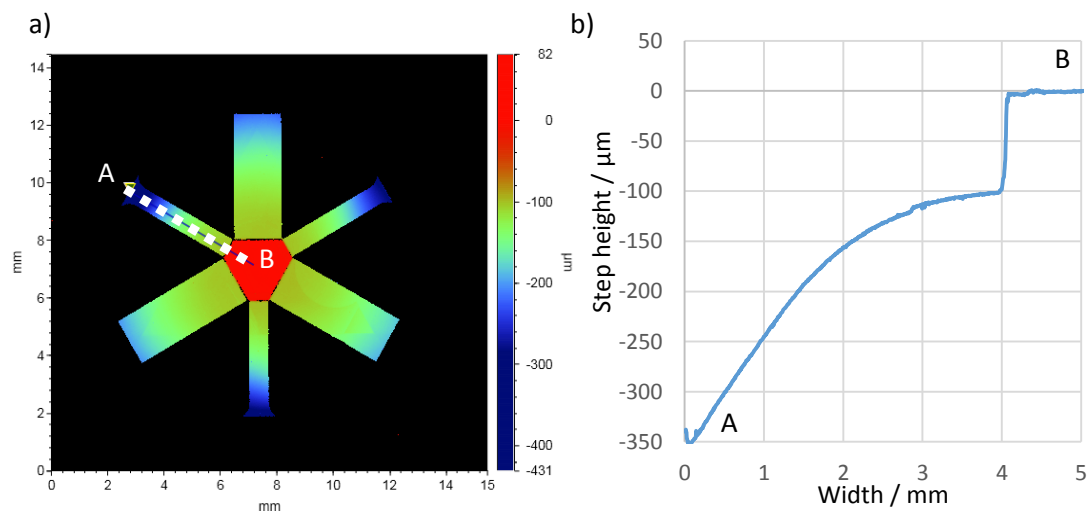


Figure 6-4: (a) Contour plot of measuring the thickness of the suspension structure that is manufactured based on conventional micro-milling and micro-electro-discharge-machining. (b) Shows the profile plot of the step change between the frame (A) and the intermediate body (B).

The main purpose of the second method was to avoid deficiencies resulting from the micro-milling process as illustrated in Figure 6-4. Thus, another fabrication process

was introduced. The same procedure was followed where the thickness was initially reduced to 400 μm from each side. Laser-ablation (LaserTec 60 HSC Laser Machining Centre) was used to reduce the thickness of the titanium to 100 μm without causing the plate to vibrate or even cause deformation. The parameters that have been used in the laser-ablation is summarised in Table 6-1. However, one of the elastic elements (suspended beam) was broken during the process, the position of the fine crack can be seen in Figure 6-5. The exact reason is unknown; however it is thought to be due to local stresses caused by thermal shock during the laser process.

Table 6-1: Parameters that was set of using LaserTec 60 HSC.

Power / W	Frequency / kHz	Velocity / mm s^{-1}	Spot size / μm	Path step over / μm
20	20	500	30	10

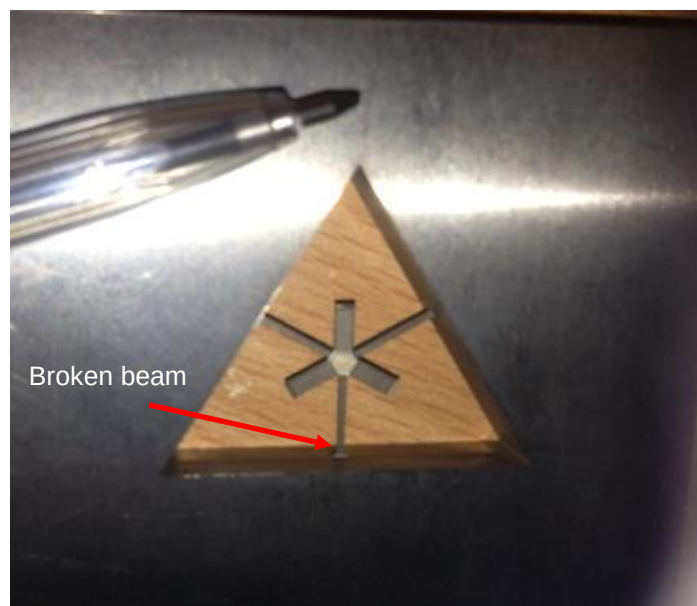


Figure 6-5: Beam thickness was reduced to 100 μm using laser-ablation process.

To address these issues, the design of the suspension structure was modified to make it suitable for manufacturing using a chemical etching process. The advantage of chemical etching is that it is able to remove material precisely, with no residual stress imparted to the structure. To allow the structure to be successfully manufactured using this method required it to be broken into three sub-components. This is due to limitations with the chemical etching process that restricted the thickness of the

components, as the variability of the etching process increases as the etch depth increases. Therefore, the suspension structure had to be manufactured in three separate layers. These consisted of a middle layer containing the spring beams, sensor paddles and central platform, and the top and bottom layers, which are identical and only contain the features associated with the outer frame.

The middle layer was manufactured from 406 μm thick spring steel sheet (AISI 1095). The geometry of the structure was defined using photolithography, and was etched using a commercially available proprietary process (Meggitt PLC, Precision Micro Ltd) [125]. The etching was completed in multiple stages. Two processes produce the final shape: a through-etch defines the outline of the structure, and a blind-etch defines the thickness of the spring beams and sensor paddles. A photograph of the middle layer is given in Figure 6-6(a) that shows the spring beam and sensor paddles with a darker surface as a result of the chemical etching process used to thin these sections of the structure (the blind-etch step). The final beam thickness was chosen to be 100 μm . The top and bottom layers were simpler to manufacture, requiring only one through-etch to define the outline of the structures. They were manufactured from 800 μm thick spring steel sheet (also AISI 1095) to provide a more substantial and robust layer, as can be seen in Figure 6-6(b).

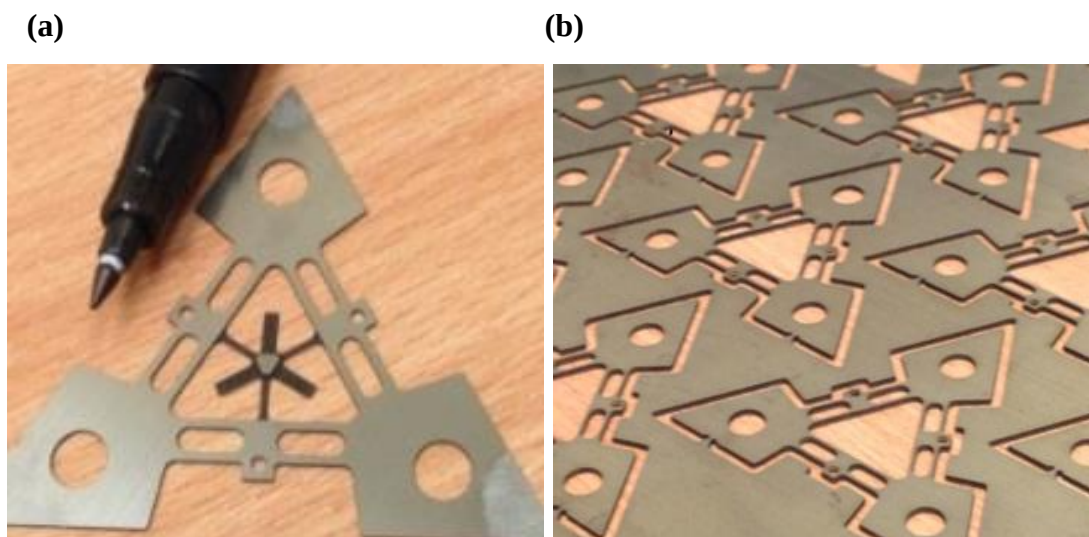


Figure 6-6: (a) Photograph of the middle layer of the suspension structure which made the spring beam. The thin sections of spring beams and paddles appear to be darker in colour due to the chemical etching process. (b) The spring steel sheet has been etched to produce the top and bottom layers.

Figure 6-7 (a) shows outline illustrations of the three layers where the thicker 800 μm layers sandwiched a thinner 406 μm layer. The three layers of the smart suspension structure were then fixed together using thin layers of UHU PLUS 300 epoxy resin adhesive to create a laminar construction. Also, to ensure that the three layers transfer the load equally to the suspension beams, pin holes were incorporated into the compressive load application points and M1.0 x 8 brass hex bolts were used to assemble the structure as shown in Figure 6-7(b).

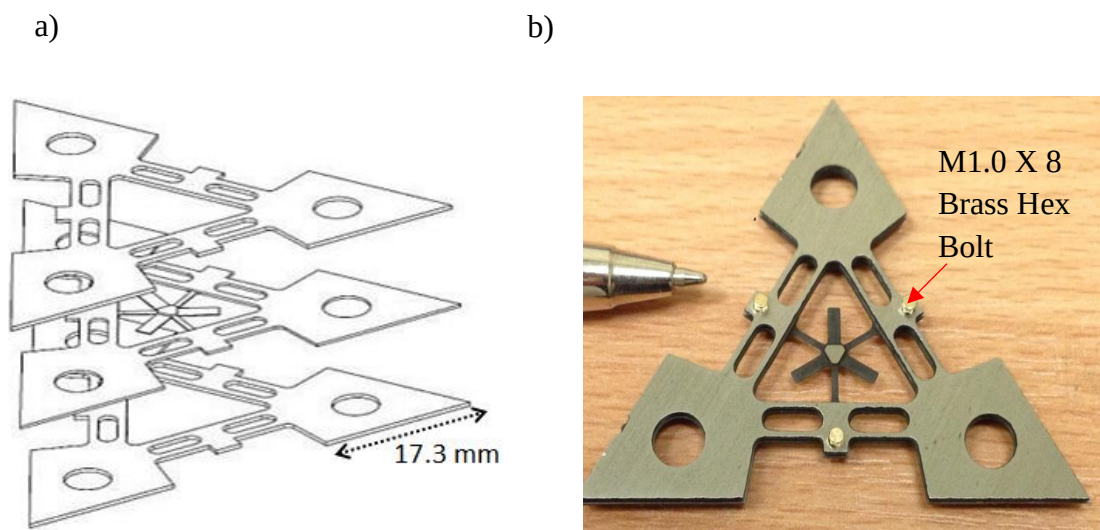


Figure 6-7: (a) Outline illustration of the three spring steel layers defined using chemical etching that form the smart suspension structure. (b) The assembled prototype of the smart suspension structure for 3D-probing system.

The geometry and flatness of the suspension structure that has been manufactured by the chemical etching technique were measured using an optical microscope (Bruker Contour GT), as can be seen in Figure 6-8. The suspension beam has a good flatness compared with a conventional machine. Bending occurs at the end of three sensor paddles where the beams deflection from ideal flat case is less than 16 μm over 4 mm length.

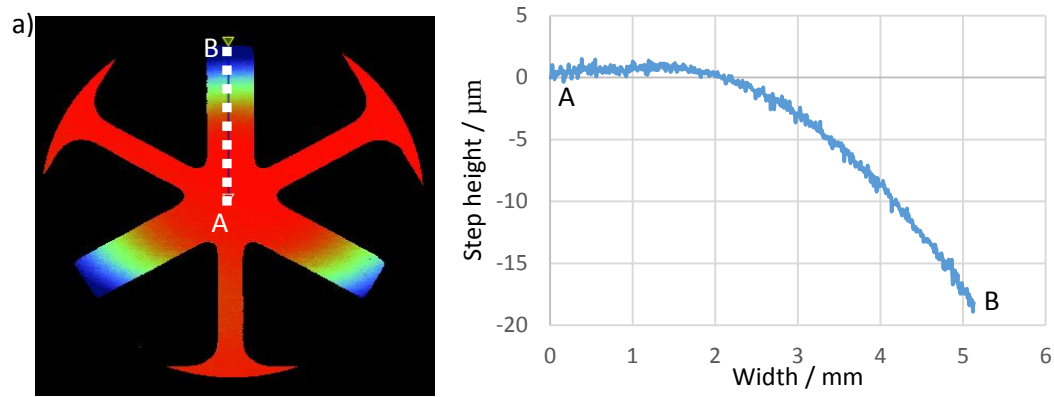


Figure 6-8: (a) Contour plot of measuring the form of the suspension structure that is manufactured based on chemical etching. (b) Shows the profile plot of the step change between the intermediate body (A) and the end of sensor paddle (B).

6.3 Suspension structure, evaluating variable stiffness (spring steel, AISI 1095)

To evaluate and validate the concept of the variable stiffness probing systems, the suspension structure that was made of spring steel as mentioned in section (6.2) needed to be mounted on a coordinate measurement machine, or in the case of the work undertaken here, a test rig. Therefore, an experimental setup was designed and the main part in a test rig assembly will be described in detail in the following section.

The main procedures that were used in the experimental validation will be described. These procedures include a description of the setup, preparation and the methodology that was used to evaluate the performance.

6.3.1 Assembly of the suspension structure on the test rig

The variable stiffness micro-scale probing system can be seen in Figure 6-9. The assembled suspension structure is the triangular shaped object that is also shown in exploded diagram format. During probing, displacements of the stylus tip are measured as a function of suspension structure displacement; to measure this displacement, the suspension structure is mounted above three high precision capacitive sensors. To allow the three sensors to be accurately positioned relative to the suspension structure, they are mounted within the probe fixture. The probe fixture contains three mounting points to allow the suspension structure to be

securely fixed in place relative to the sensors. The sensors can also be positioned vertically within the probe fixture, so the optimum working gap of approximately $100\text{ }\mu\text{m}$ can be set. To allow stiffness modulation of the suspension structure, three piezoelectric stacks (PICMA, P-882.11) with $8\text{ }\mu\text{m}$ maximum displacement are also mounted to the probe fixture. The actuators must be positioned in intimate contact with the load application points on the suspension structure. To allow them to be securely clamped in place, between the probe fixture and the suspension structure, a set of three sliding clamps was added to the probe fixture.

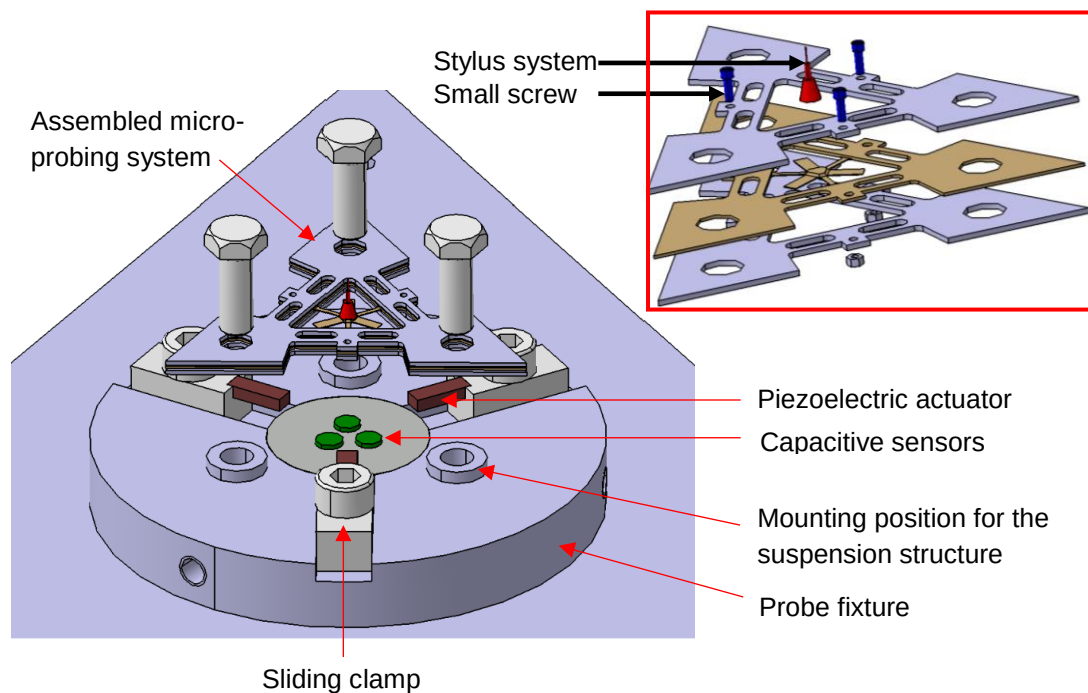


Figure 6-9: Schematic drawing of the assembly of the variable stiffness micro-probing system, with exploded view inset.

The prototype of the probe fixture with three capacitive sensors used to measure the position of the suspension structure, as well as three piezoelectric micro-actuators used to apply compressive load to the structure, are shown in Figure 6-10.

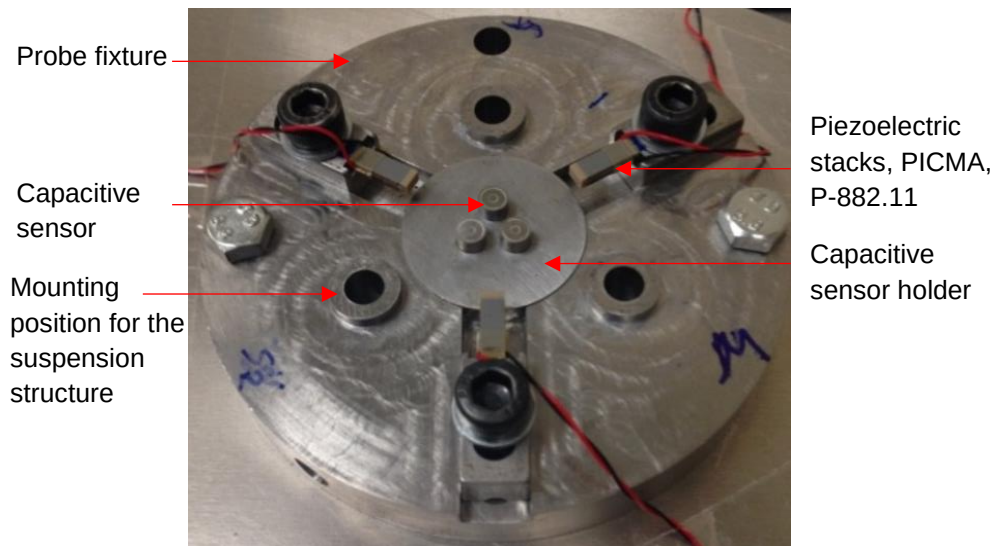


Figure 6-10: Assembled probe fixture, with three mounted piezoelectric actuators for compressive force application, and three capacitive sensors.

6.3.2 Experimental methodology for measuring stiffness of the suspension structure

To accurately assess the vertical and lateral stiffness of a probing system requires a fully assembled probe to be tested, using an integrated precision displacement actuator and low force sensor. This would allow force displacement curves to be generated for each direction. An alternative methodology was designed to assess both vertical and lateral stiffness of the manufactured suspension structure. This method was based on measuring modal frequencies of the structure, which are related to vertical and lateral stiffness. Structural mode shapes that are closely representative of both the vertical and lateral probing operations were selected using the results of a finite element modal analysis (details of the model are given in section 6.3.5). The selected modes can be seen displayed also in section 6.3.5, in Figure 6-18. These modes were selected as they correspond to either pure vertical displacement of the intermediate body (which would result from vertical probing), or pure rotational displacement of the intermediate body (which would result from lateral probing). Therefore, as the frequency of each of these modes is a function of stiffness, these frequencies may be monitored to provide an indication of how both the vertical and lateral stiffness is changing.

6.3.3 Measuring vertical and torsional frequency of the suspension structure using a laser vibrometer

To check the capability and robustness of the designed test rig for measuring the resonant frequencies, a PolyTec OFV-055 laser vibrometer was initially used to measure the amplitude of the oscillations at zero applied compressive loads. The laser vibrometer was calibrated in the previous project via pointing a laser onto a known base motion to generate the relationship between the displacement and voltage [126]. The suspension system as shown in Figure 6-7(b) was excited using a shaker table from 2200 Hz to 3400 Hz and the laser was pointed to the middle of the intermediate body as can be seen in Figure 6-11. The data could then be plotted in a graph and the natural frequencies identified as shown in Figure 6-12. The first peak at 2469 Hz is the vertical natural frequency (f_v) and the second, smaller peak at 3270 Hz is the torsional natural frequency (f_t).

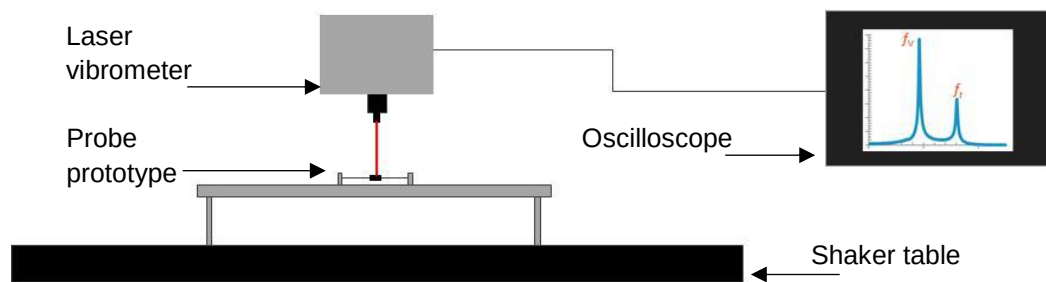


Figure 6-11: Shows the schematic diagram of the setup used in the initial experiment to find the natural frequency.

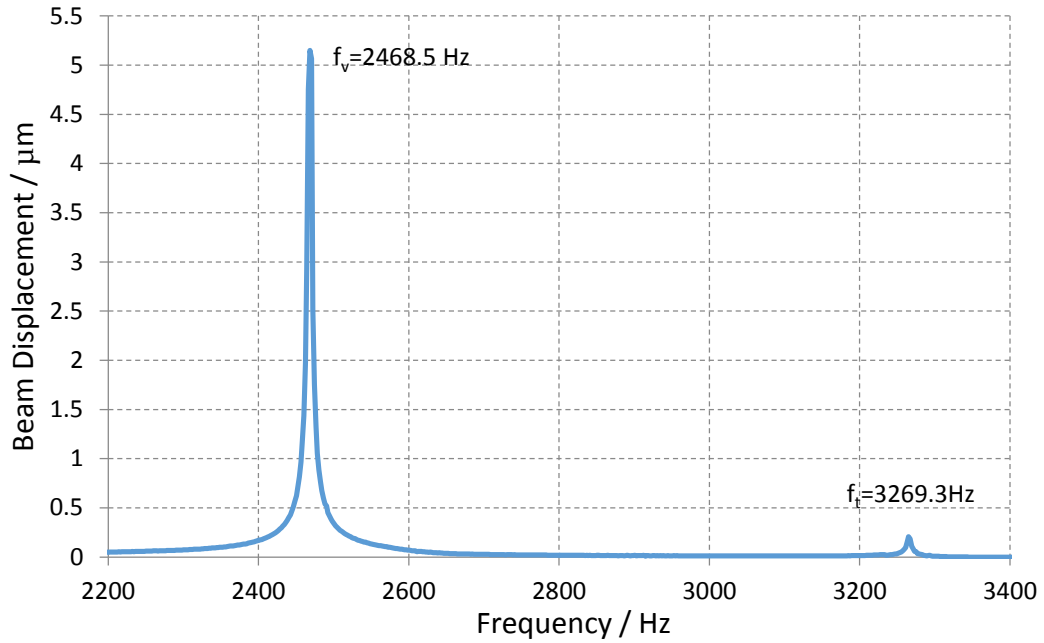


Figure 6-12: Sweep the frequency experimentally from the 2200 Hz to 3400 Hz to find beam resonant frequencies using a laser vibrometer where the vertical and torsional natural frequencies are 2469 Hz and 3270 Hz, respectively.

6.3.4 Measuring vertical and torsional frequency of the suspension structure using a designed test rig

In this experiment a complete probe was not available as only the suspension structure was manufactured. The suspension structure as illustrated in Figure 6-7(b) was mounted on the probe fixture as shown in Figure 6-13. The sensor paddles were used to provide a target surface for three capacitive displacement sensors aligned above the sensor paddles at a stand-off distance of approximately 100 μm . The piezoelectric actuators were used to apply a superposition of a static load to modulate the stiffness, and a dynamic sinusoidal load to excite the natural frequencies of the structure. Sinusoidal excitation frequencies were applied sweeping from 500 Hz to 3500 Hz using linear chirp signal. The response of the structure to the dynamic excitation for a range of static loads could then be determined using the three capacitive sensors. The full equipment setup is displayed in Figure 6-14. This test equipment made use of a dedicated amplifier to control the voltage applied to the piezoelectric actuators (E-663, Physik Instrumente), and signal synchronisation was achieved using National Instruments hardware (NI cDAQ-9172) and LabVIEW. The

Embodiment, fabrication and experimental validation of a variable stiffness suspension

maximum measuring rate of the capacitive sensor was 10 kHz, which is well above the required frequency sweep range.

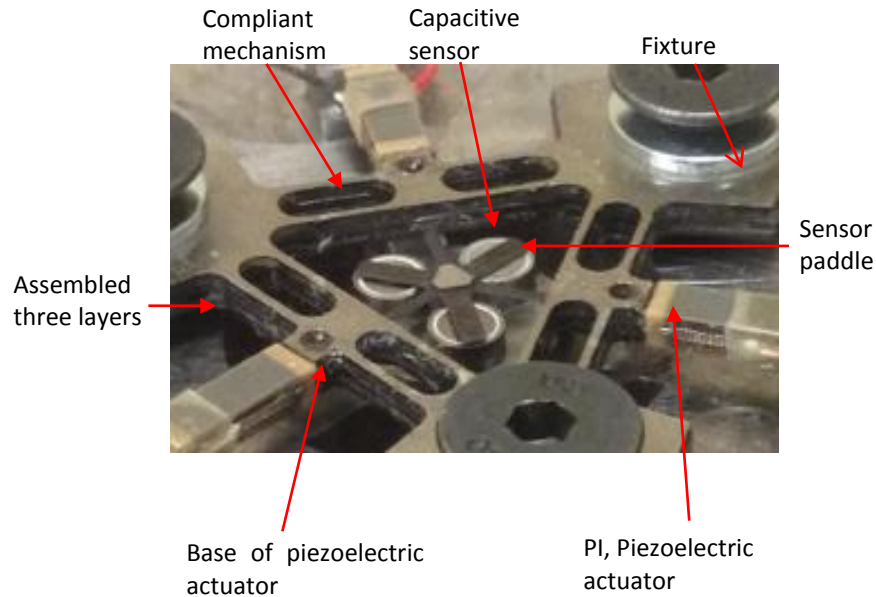


Figure 6-13: A prototype of the active suspension structure where the suspension structure was made from spring steel (AISI 1095).

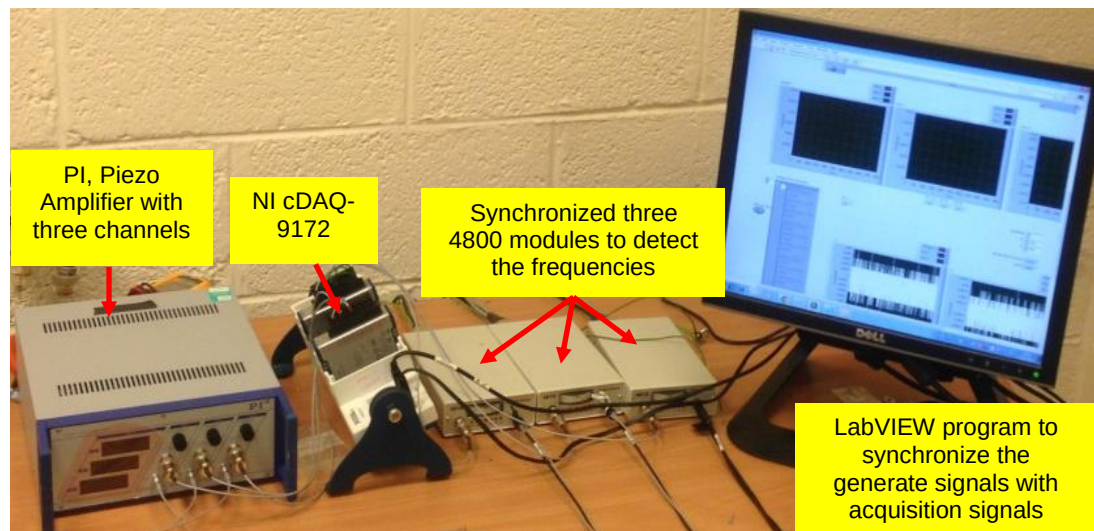


Figure 6-14: Configuration and experimental setup.

To analyse the response of the suspension structure to the chirped input signal, a Fast Fourier Transform (FFT) was performed using LabVIEW. These frequency values were then recorded with the corresponding applied static load.

For the actuators used the applied force was proportional to the applied voltage; therefore, the applied voltage was used to represent the applied force as an accurate calibration for this was not available. These results can be seen in the chart shown in Figure 6-15, which shows two curves that initially decrease with respect to the applied voltage, and then a point of inflection is reached at approximately 60 V for the vertical frequency, and a plateau in torsional frequency. The minimum frequency reduction, and hence stiffness reduction, is therefore limited to that achieved at 60 V applied to the actuators. The measured initial frequencies using a test rig shows a good agreement with the measured frequency using a laser vibrometer, where the difference in both frequencies is less than 49 Hz. This comparison between measured frequencies also provides increased confidence in the rig capability and robustness of measurement method.

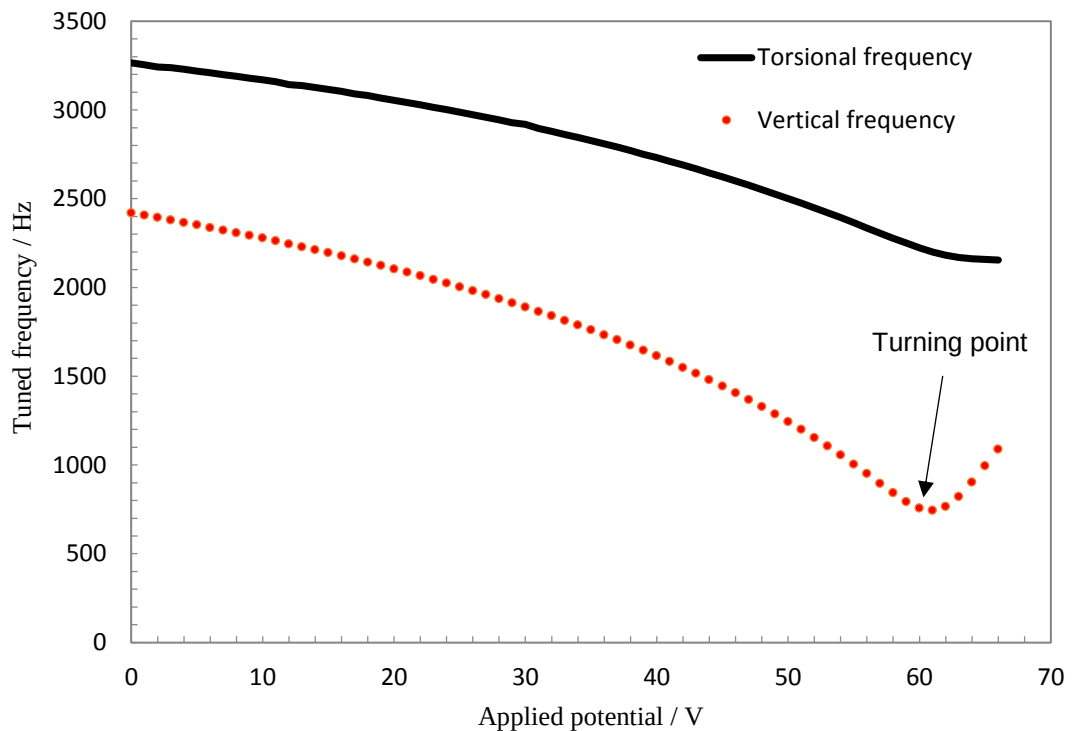


Figure 6-15: A plot of the vertical and the torsional frequencies of the suspension structure against the applied voltage (DC voltage), as calculated experimentally.

6.3.5 Finite element simulation of suspension structure performance

To understand the behaviour of the structure, a finite element model of exact geometry of the fabricated structure was created. The finite element model was

created to match the geometry of the manufactured smart suspension structures as closely as possible and as such is able to reflect any fabrication intolerance. The geometry was measured using the 3D microscope (Bruker GT-I). The dimensional parameters that define this geometry are presented in Figure 6-16 and Table 6-2. Material properties used in the model are based on values for spring steel AISI 1095 [127], which can be seen in Table 6-3. The finite element model was constructed so as to ensure that geometric nonlinearity was accounted for. As such the effect of the axial compressive force was considered [120]. A mapped uniform mesh for each suspension beam was created to ensure an equivalent model of the properties in each beam. Quadrilateral elements were used for all suspension beams, with 154 μm , 140 μm and 25 μm elements along the length, width and thickness of the beams respectively. To simplify the model, only the three spring beams and the intermediate body with sensor paddles were considered, as illustrated in Figure 6-16. This reduced the problem from a complex multi-layer structure to a simple monolithic one. Also, for practicality the more complex structural features, such as the parts that connect beams or the fillets, were modelled with an unstructured triangular swept mesh. An example of the output from the finite element model is given in Figure 6-17, which shows results from vertical and torsional stiffness analysis of the suspension structure at zero axial compressive force, where the vertical and torsional stiffnesses are 6640 N m^{-1} and $43.9 \times 10^{-3} \text{ N m rad}^{-1}$ respectively. The vertical stiffness has been estimated based on a load of 1 mN in the vertical direction as chosen to be representative of typical probing forces for this type of sensor [128]. For lateral stiffness a moment of 6.6 μNm was chosen, based on a 1 mN contact force applied to the end of a 6.6 mm long stylus.

In addition to the static analysis described above, the finite element model was also used to predict the modal frequencies of the suspension structure as a function of axial displacement. To simulate the deflection that was introduced by the piezoelectric actuator, a series of modal analyses was carried out with a range of pre-stressed conditions. This allowed the modal frequencies for the structure under a compressive load to be calculated. These modal frequencies were calculated to provide a direct comparison with the results of the physical characterisation of the structure.

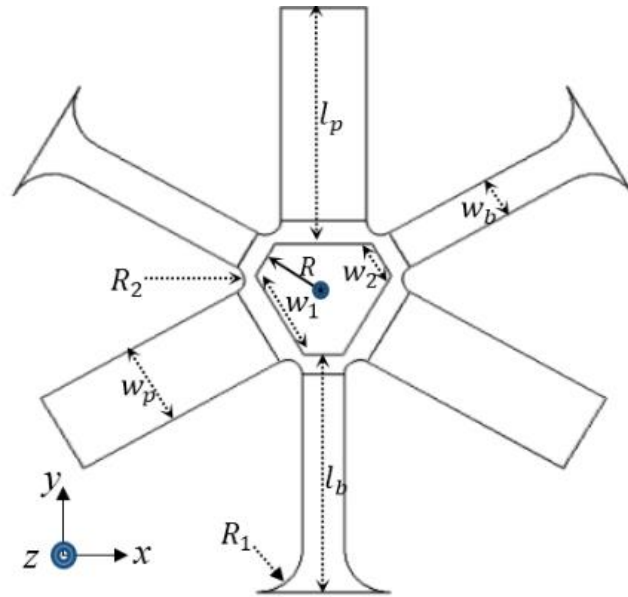


Figure 6-16: The main geometric parameters of the tri-beam design.

Table 6-2: Structure main parameters of the tri-beam design

Item	Specifications / μm
Spring beams	$l_b = 4188.6$, $w_b = 700$, $t_b = 100$ (thickness), $R_1 = 750$
Central platform	$R = 1111.3$, $w_1 = 1600$, $w_2 = 650$, $R_2 = 250$
Sensor paddles	$l_p = 4162.9$, $w_p = 1420$

Table 6-3: Material properties of the tri-beam design (spring steel, AISI 1095).

Elastic modulus, E / GPa	Poisson's ratio, ν	Density, ρ / kg m^{-3}
210	0.3	7850

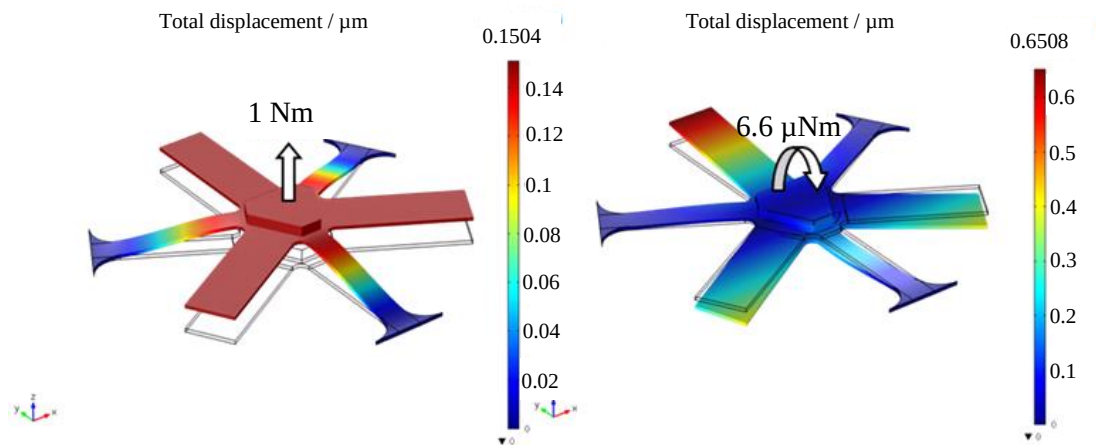


Figure 6-17: Shows results for zero applied compressive load: (a) Vertical displacement resulting from an applied vertical load of 1 mN at the centre of the structure, (b) Torsional stiffness analysis of the suspension structure resulting from 6.6 $\mu\text{N m}$ at the centre of the structure.

The first two mode shapes were estimated numerically using FEA to identify the expected modal frequencies and their associated mode shapes, as shown in Figure 6-18, which shows that the suspension structure vibrates vertically at 2402.6 Hz, with the second modes representing a torsional motion with a frequency of 3061.5 Hz.

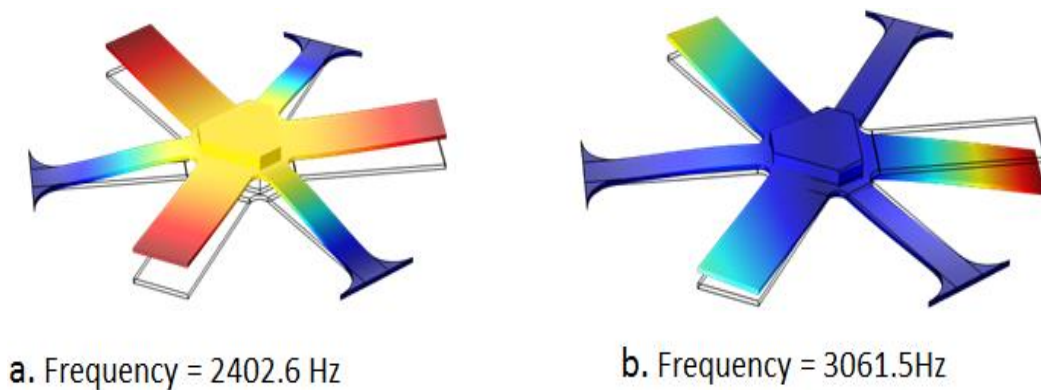


Figure 6-18: Shows mode shapes for zero applied compressive load: (a) First mode, where the suspension structure vibrates vertically, and (b) Second mode where the centre of the structure has a torsional motion.

This analysis was run over a range of applied axial displacements. The results can be seen in the chart in Figure 6-19. The solid lines on the chart represent the model of the manufactured suspension structure; by inspection it can be seen that the shape of

the curves for the torsional and vertical modes are similar to those collected experimentally (see Figure 6-15). The simulation was able to predict with good agreement the behaviour of the structure and in particular the limited frequency reduction beyond a displacement limit, as was seen for both modes in the physical device. A potential cause of this was thought to be non-symmetrical beam geometry as a result of the chemical etching process, which required the suspension structure beams to be manufactured with single-sided blind etching only as illustrated in Figure 6-20. To understand the impact of this on the performance of the structure, a slightly modified version of the finite element model was created. This second model was designed to have overall the same dimensions as the first; however, the beams were modelled as being completely symmetrical as would be produced from a double sided blind etch (see Figure 6-20). The results collected from this model are also shown on the chart in Figure 6-19 as the broken lines. For the symmetrical model it can be seen that the structure is able to reduce modal frequencies down to zero.

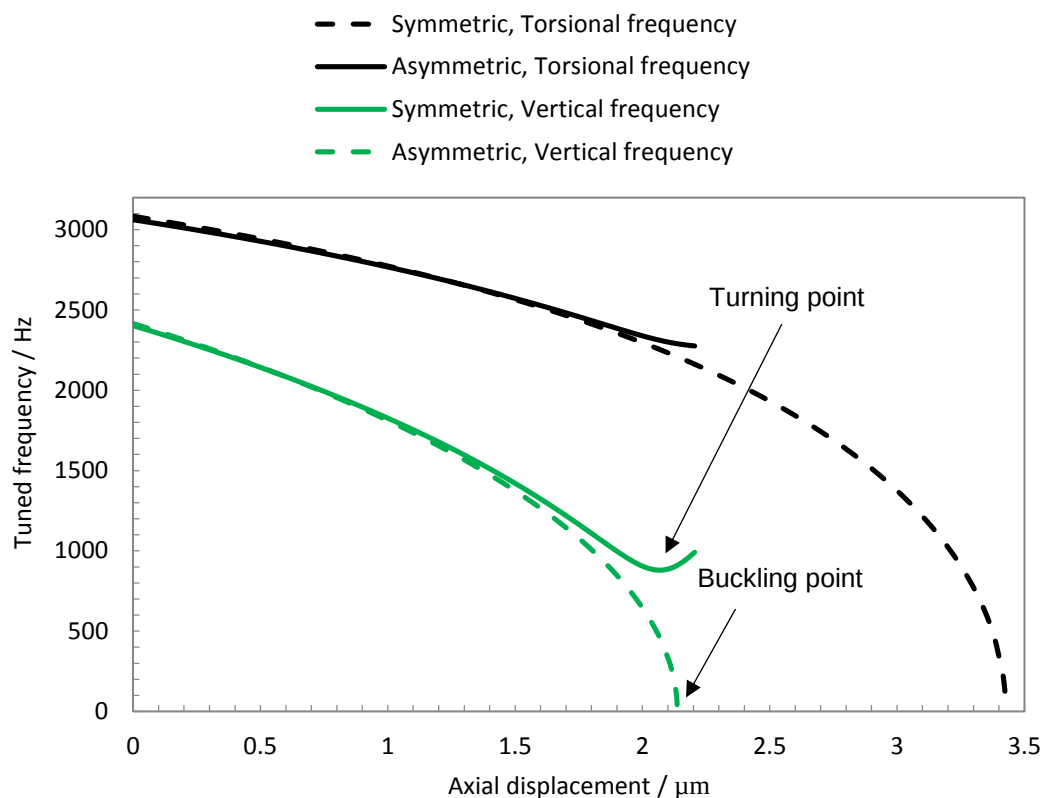


Figure 6-19: Graph of modal frequency against axial displacement of the suspension structure, as calculated for symmetric and asymmetric beam in FAE analysis (COMSOL).

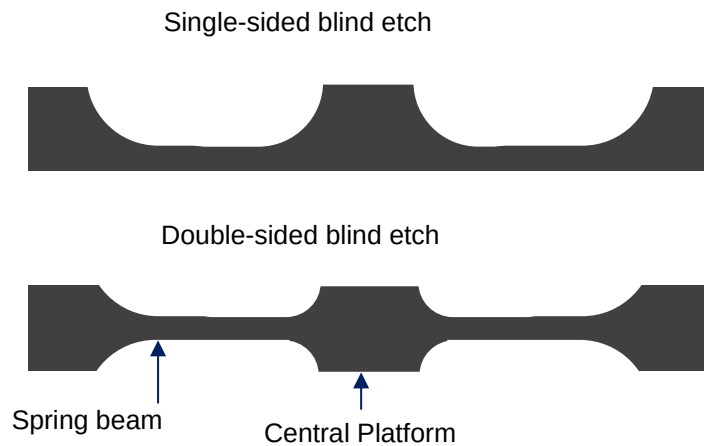


Figure 6-20: Schematic diagram of the cross-section of a beam etched using two different strategies to demonstrate non-symmetrical nature of a beam manufactured with a single-sided etch as compared to a double-sided etch.

6.3.6 Analysis of results and discussion

Considering the experimental test results, it is clear that the structure is able to successfully modulate stiffness. The results in Figure 6-15 show that for both the torsional and vertical modes of the suspension structure it was possible to reduce the frequency by applying a control voltage to the actuators. As the applied voltage was proportional to applied load, these results show that as load is applied to the structure, the frequency and therefore its stiffness is reduced. However, the reduction in frequency was limited as a turning point was reached at a load corresponding to a frequency of approximately 743 Hz for the vertical mode. This response was also seen in the finite element modal analysis for asymmetric beams as can be seen in Figure 6-19. To allow direct comparison of the model data with the experimental data, the applied voltage used in the experimental setup and the applied displacement used in the finite element model were both normalised to one scale. This can be done if we assume the voltage applied scales linearly with applied load and displacement. The scaling was done by normalising both sets of data to a value of one at the observed minimum vertical frequency. The results from this can be seen in Figure 6-21, where it can be seen that the finite element model is in reasonable agreement with the experimental data, with the vertical model frequency showing particularly close agreement, and the torsional mode having agreement in observed and predicted frequencies to better than 6 %. The observed error may be attributed to the simplified

monolithic model, or minor differences in beam cross section caused by the etching process; however, no direct cause can be attributed based on these results. Further detail of the results is given in Table 6-4, which presents a comparison of the measured frequencies and those predicted by the finite element model for the suspension structure, by considering the initial frequency and the lowest possible frequency as a result of applied displacement or voltage.

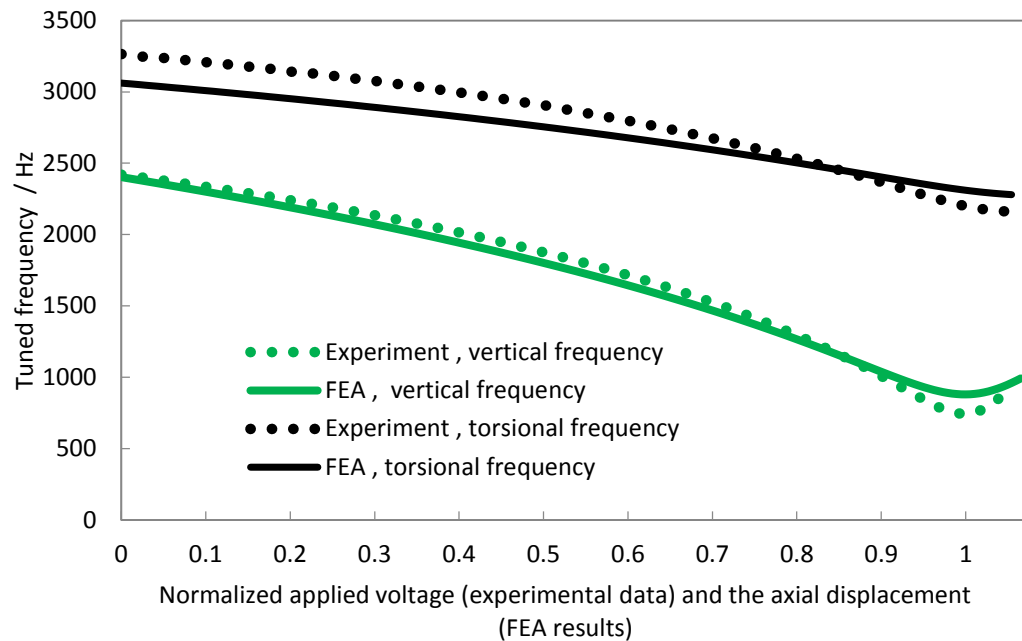


Figure 6-21: Graph of modal frequency of the suspension structure against the normalized scale at the observed minimum frequency for the vertical direction, as calculated for the asymmetric beam in experimental result and FAE analysis.

Comparison of data clearly also shows that the maximum level of frequency reduction possible is limited by the structural asymmetry that resulted from the prototype manufacture. To explain this consider any buckling beam system. If the system is completely symmetrical the beam will be able to have a compressive load applied until the buckling point is reached, at which point it will become unstable and the stiffness will reduce to zero. If asymmetry is introduced to the system, either in the form of additional off-centre loading, or as in this case non-symmetrical geometry, then the structure will be naturally subjected to bending stresses. These bending stresses predispose the system to bend before it buckles, and when bending occurs there is a small relief in the applied compressive load. This is a highly non-linear effect, and as such it is not felt until a significant level of compressive load has

been applied. In the case of the smart suspension structure it is visible at the turning point as the frequency stops decreasing as a result of applied load, and then starts to increase as the induced bending negates the compressive loads.

To quantify the observed frequency change in terms of stiffness, the finite element model was used to predict the stiffness change made possible once the turning point was reached. This was done assuming the suspension structure would be used as part of a probing system with a 6.6 mm stylus length. From the model the minimum vertical stiffness was found to be 788 N m^{-1} , approximately 12 % of the initial stiffness of 6640 N m^{-1} . While, for the lateral stiffness a minimum of 464 N m^{-1} was found which is 46 % of the initial stiffness of 1007 N m^{-1} .

As a key consideration for probe system design is to have isotropic stiffness, the ratio of the vertical to lateral stiffness is considered. For the initial stiffness of the structure this ratio would be 6.6, while for the minimum stiffness condition this ratio is reduced to 1.7, which is close to the isotropic stiffness.

Table 6-4: Comparison of the measured and the finite element predicted frequencies for the suspension structure. (*) Indicates the percentage reduction in frequency.

Type of frequency	Experiment				COMSOL (FEA)				
	Laser vibrometer	Test rig			Asymmetric			Symmetric	
		f_o /Hz	f_o /Hz	f_{min} /Hz	Red* (%)	f_o /Hz	f_{min} /Hz	Red* (%)	f_o /Hz
Vertical	2469	2420	743	69	2403	880	63	2415	0
Torsional	3270	3265	2200	33	3062	2311	25	3083	0

6.4 Probing system, evaluating variable stiffness (titanium, Grade 5)

The prototype device that was made of spring steel remains relatively stiff at the turning point as mentioned in the previous section. However, the principle of a variable stiffness probing systems has been shown and further reduction in stiffness could be achieved through a change in materials. Therefore, titanium was chosen as it has a low elastic modulus, which is beneficial for reducing stiffness. Titanium also has a high strength to mass ratio which assists to reduce the effective mass of the suspension structure.

To provide an example of a full probing system that is made of titanium, a tri-beam variable stiffness suspension structure was integrated with a stylus system and displacement sensors. The suspension structure was manufactured using chemical etching following the same procedure as explained in section 6.2. The middle layer consist was produced in grade 5 titanium. The top and bottom layers were produced to sandwich the middle layer and are both identical, as seen in Figure 6-6(b), are both made of spring steel. The main features of the outer frame with the suspension structure in place can be seen in Figure 6-22.

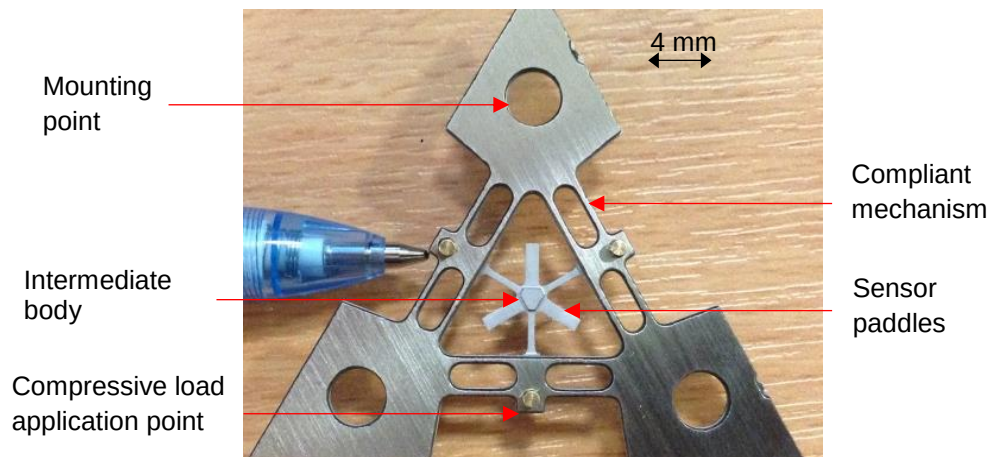


Figure 6-22: The prototype of the smart suspension structure, with main structural elements labelled where the middle layer (suspension structure) is made of titanium.

The geometry of the manufactured suspension structure is as described in Table 5-2. Figure 6-23(a) shows the geometry and flatness of the suspension structure (middle layer) that was measured using the 3D microscope (Bruker GT-I). The thickness of the thin elastic spring was measured and was around $110\text{ }\mu\text{m}$ as can be seen in Figure 6-23(b) where the overall thickness of the middle layer is $400\text{ }\mu\text{m}$.

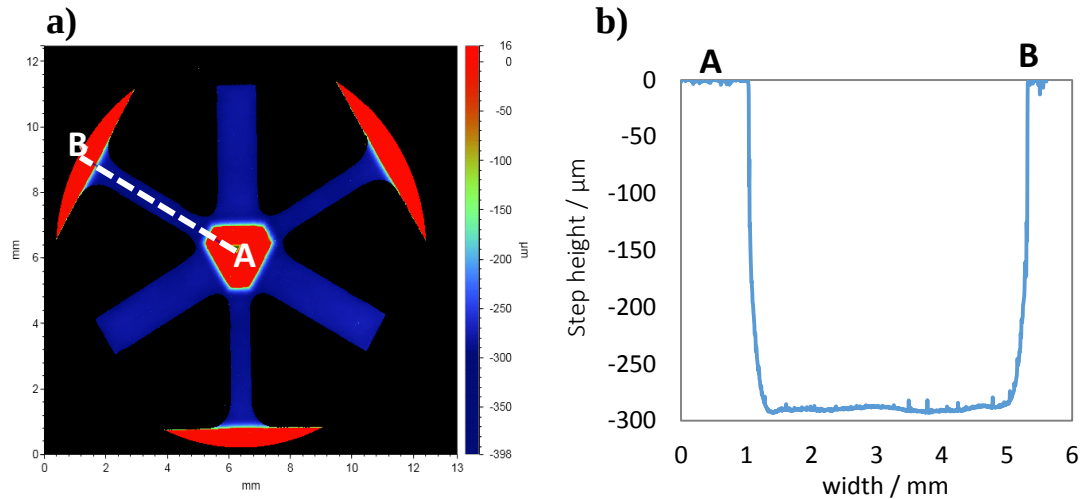


Figure 6-23: (a) Contour plot of measuring the thickness of the suspension structure (titanium part) that was sandwiched between two layers (spring steel) where the total thickness is 400 μm . (b) Shows the profile plot of the step change between the intermediate body (A) and the frame (B).

To provide a reasonable representation of a full probing system, a modified commercial stylus system (Renishaw stylus, serial number A-5000-7800) was bonded to the intermediate body using a fast acting cyanoacrylate adhesive. This stylus has total length of 6.6 mm and a conical neck which holds the stem and is made of steel. The modified geometry of the stylus is illustrated in Figure 6-24, and the dimensional parameters and material properties are defined in Table 6-5 and Table 6-6 respectively.

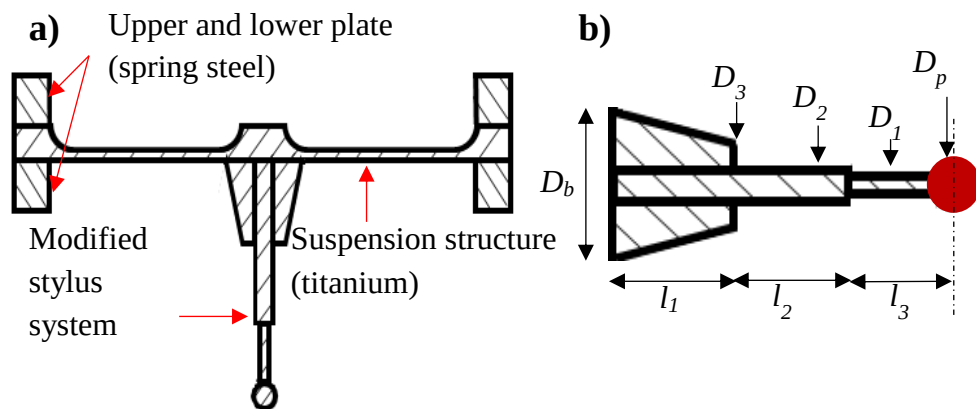


Figure 6-24: (a) Schematic cross-sectional diagram of the probing system. (b) The main geometric parameters of the modified stylus system.

Table 6-5: Main parameters of the modified micro-probe.

Item	Specifications / μm
Ruby ball	$D_p = 300$,
Stem	$D_1 = 210, D_2 = 675, l_2 = l_3 = 2000$
Stem holder	$D_3 = 1314.2, D_b = 2560, l_1 = 2600$

Table 6-6: Material properties of the micro-probe (Renishaw, A-5000-7800) [21].

Material	Elastic modulus, E /GPa	Poisson's ratio, ν	Density, ρ /kg m ⁻³
Ruby ball	375	0.22	3890
Tungsten carbide	345	0.28	19300

The assembled probing system was mounted on a fixture containing three piezoelectric actuators, and the actuators were aligned to the load application points on each compliant mechanism. A photograph of the probing system with actuators (piezoelectric stacks, PICMA, P-882.11) attached is shown in Figure 2-29.

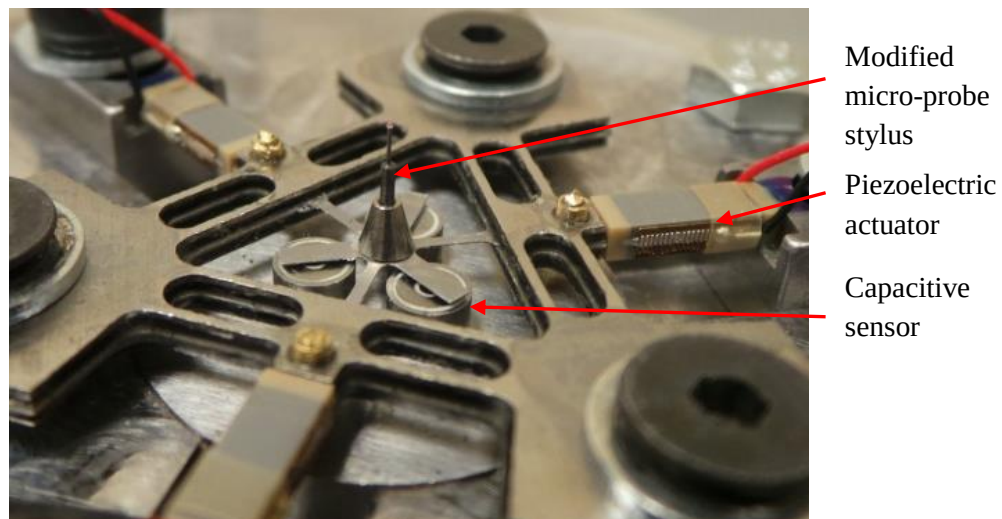


Figure 6-25: Assembled probing system, with three mounted piezoelectric actuators for compressive force application, and beneath the suspension structure three capacitive sensors.

To test the ability of the actuators to reduce the stiffness of the suspension structure, a range of loads was applied using the actuators, and the resulting modal frequencies

of the structure were monitored using the three capacitive sensors. This provided a simple and effective method for monitoring structural stiffness as a function of applied load, as both the vertical and lateral stiffness can be easily related to specific modal frequencies of the structure. To identify the expected vibration modes of the probing system, modes representative of vertical and lateral stiffness were estimated using finite element analysis of the full structure including the stylus system, and the selected mode shapes can be seen in Figure 2-30. Under zero applied load, the vertical and torsional frequencies were estimated to be 1114 Hz and 1423 Hz respectively.

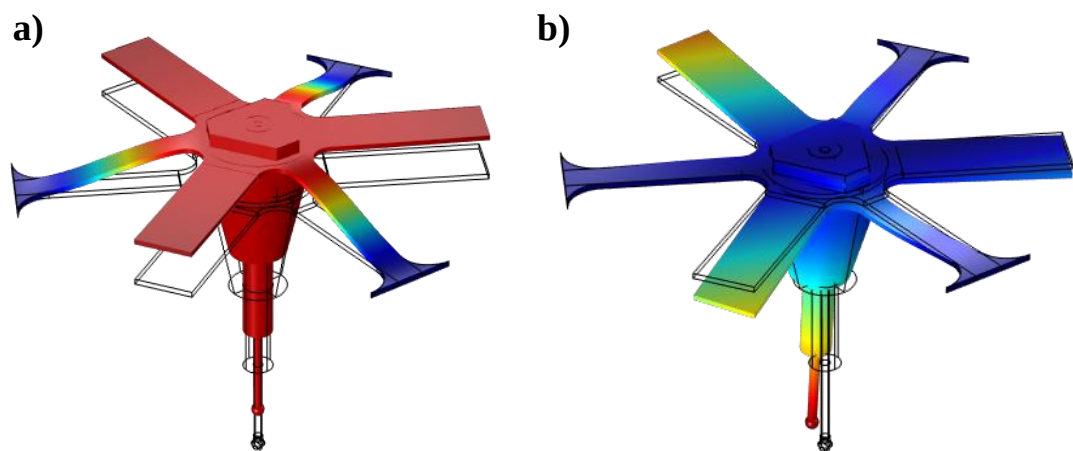


Figure 6-26: Shows mode shapes for zero applied compressive load: (a) The first mode ($f = 1114$ Hz), where the full micro-probe vibrates vertically, and (b) The second mode ($f = 1423$ Hz), where the centre of the micro-probe has a rotational motion.

The model was then used to predict how the vertical and torsional frequencies would change as compressive load was applied to the spring beams. To experimentally validate the finite element model, the vertical and torsional frequencies were also measured using the prototype device. To make this possible, the actuators were used to apply a combined static and dynamic load. The static load was a voltage that was sequentially increased from zero to 90 V to apply an increasing compressive load to the beams. The dynamic component was a sinusoidal chirp signal of 0.5 V peak-to-peak. This chirp signal was used to excite the structure through a range of frequencies from 100 Hz to 2000 Hz. The capacitive sensors, located below the sensor paddles as illustrated in Figure 2-29, were used to detect the resulting resonant frequencies. Both the experimental data and the data generated from the finite

element model are shown in Figure 6-27. A direct comparison is not possible as the transfer function of the actuators was not determined; however, both sets of data present the same underlying characteristics. Most interestingly, the vertical and torsional frequency reduction is limited by a turning point, beyond which the structural frequencies increase.

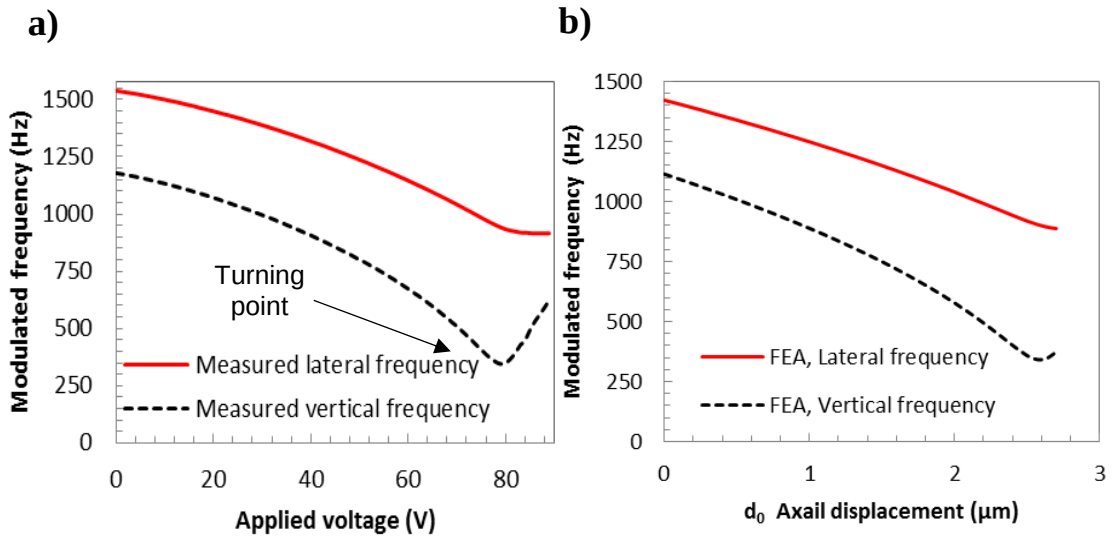


Figure 6-27: (a) A plot of the vertical and the torsional frequency of the full micro-probe against the applied static voltage determined experimentally for 6.6 mm stylus length. (b) The modulated frequencies of the full probe as estimated by FEA analysis plotted as a function of axial displacement (d_0).

The cause of this turning point was determined to be a combination of factors that result in a non-symmetrical strain state being created within each suspension beam as discussed in section 6.3.6. However, two potential causes of the non-symmetrical strain state were considered. The first was the non-symmetrical beam geometry that results from the chemical etching process, and the addition of the stylus system to the suspension structure, as illustrated in Figure 6-24(a). Note that the intermediate body when coupled to the stylus system presents an asymmetric stiffness. The stylus base is larger and provides more reinforcement when compared to the intermediate body itself. This results in a small rotation at the ends of the spring beams in reaction to the applied compressive load, causing the beams to bend. The second was the influence of gravity on the structure providing additional non-symmetrical load, especially as the stylus system is relatively massive compared to the suspension

beams. To explore the impact of these factors, the finite element simulation was adjusted to demonstrate the sensitivity of the structure to variations in both the mass of the stylus as well as structural asymmetry. Considering gravitational loading first, the simulation was adjusted to zero gravity load. The results for zero gravity are compared to the previous results in Figure 6-28, and it can be seen that gravity has a negligible effect on the device. This leaves structural asymmetry as the most likely cause of the turning point.

To compensate for the asymmetrical geometry, the finite element model was adjusted by trial and error until a perfectly balanced load was achieved in the suspension beams. In this context a balanced load in the suspension beams is one where the forces exerted by the beams on the intermediate body result in zero motion of the intermediate body. The end points of the beams therefore remain stationary and as the system approaches buckling it can be considered to be fixed-fixed. It was found that this can be achieved by changing parameter D_b (as shown in Figure 6-24), and that a value of 2146 μm resulted in vertical and lateral frequencies that could be reduced down to zero, as shown in Figure 6-28. It must be noted that the effective mass of the probe was reduced (72 mg) as compared to the initial model (104 mg), and this reduction in mass and the stiffness (see Figure 6-29) leads to slightly increased initial frequencies. Nevertheless, the finite element model clearly indicates that, through careful mechanical design, it is possible to design a structure capable of generating plain compressive stress in the suspension beams, and that this structure does not have to be geometrically symmetrical.

In addition to the modal frequency analysis described above, the finite element model was also used to predict the stiffness in the vertical and lateral directions of the suspension structure as a function of axial displacement. Figure 6-29 shows the predicted stiffness as a function of axial displacement for two cases. The first case illustrates the reduction of the stiffness for the manufactured geometry, with parameters of the modified probe in Table 6-5. The lowest stiffnesses that can be achieved in the vertical and lateral directions at the turning point are around 440 N m^{-1} and 340 N m^{-1} respectively. The initial ratio between the vertical and lateral stiffness is reduced from 7 to 1.3, which is close to the isotropic stiffness. However, to achieve the fully isotropic stiffness, the structure with optimised

geometry for a balanced load must be used. As can be seen in Figure 6-29, this results in a crossing point for the lateral and vertical stiffness curves.

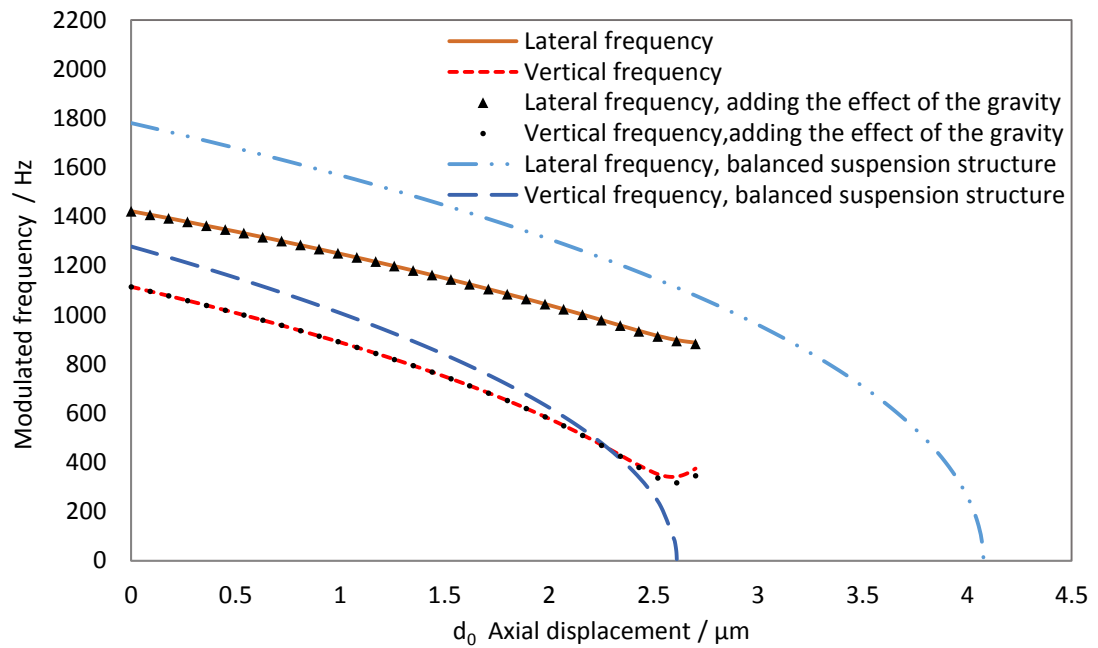


Figure 6-28: Modulated frequencies of the full micro-probe, estimated by FEA and plotted as a function of axial displacement (d_0). Three conditions are considered: manufactured geometry, manufactured geometry with gravity and a structure with optimised geometry to support a balanced load.

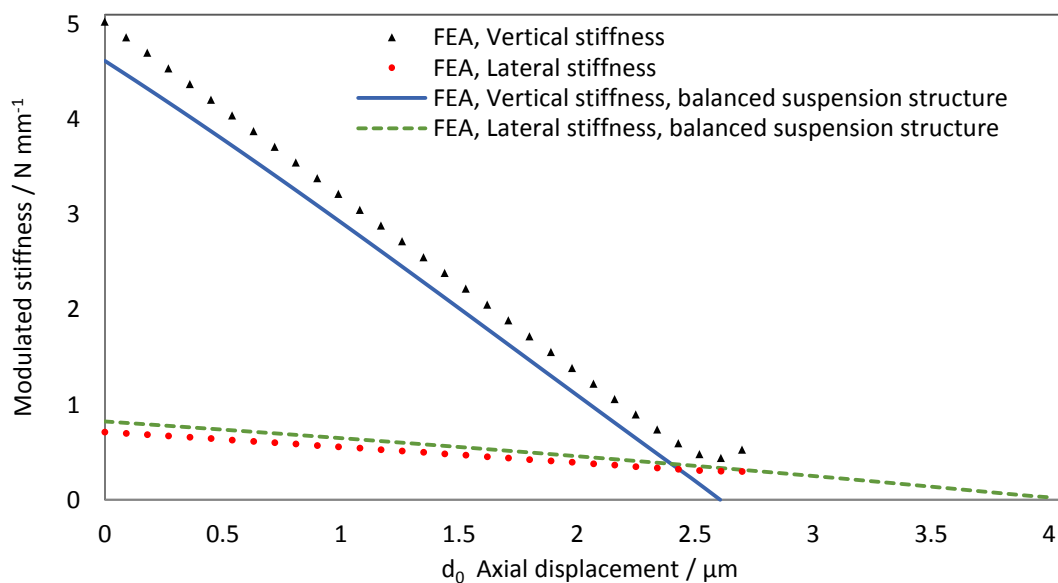


Figure 6-29: Modulated stiffness of the full micro-probe, estimated by FEA and plotted as a function of axial displacement (d_0). Two conditions are considered: manufactured geometry and a structure with optimised geometry to support a balanced load.

6.5 Probe performance over time

The data presented up to this point considers only the performance of the probing system over short periods of time. For real probing applications it is necessary to consider the stability of the probe over a longer period. To assess whether the probing system exhibits creep or hysteresis an experiment was conducted where the stylus tip was deflected by a nano-positioning stage (Physik Instrumente, P-611.ZS), to a known distance and held for an extended period. The position of the probe, as measured by the capacitive sensor signal, was monitored over time. The data was evaluated for stability of the measured displacement and the consistency of return to initial unloaded condition when the nano-positioning stage was retracted. Figure 6-30 shows an example of the experiment, with the output displacement from capacitive sensor for two steps displacement, plotted against time. The example illustrates that in its current state the probe exhibits creep, for the first and second displacement equal to $0.65\text{ }\mu\text{m}$ and $0.31\text{ }\mu\text{m}$, respectively, and after 5 minutes of loading and unloading offset error of around 160 nm .

The observed error may be attributed to a reliance on epoxy to hold the relative positions of the layers. When a load is applied over a long period of time at the stylus tip, this may cause a relative displacement between the layers.

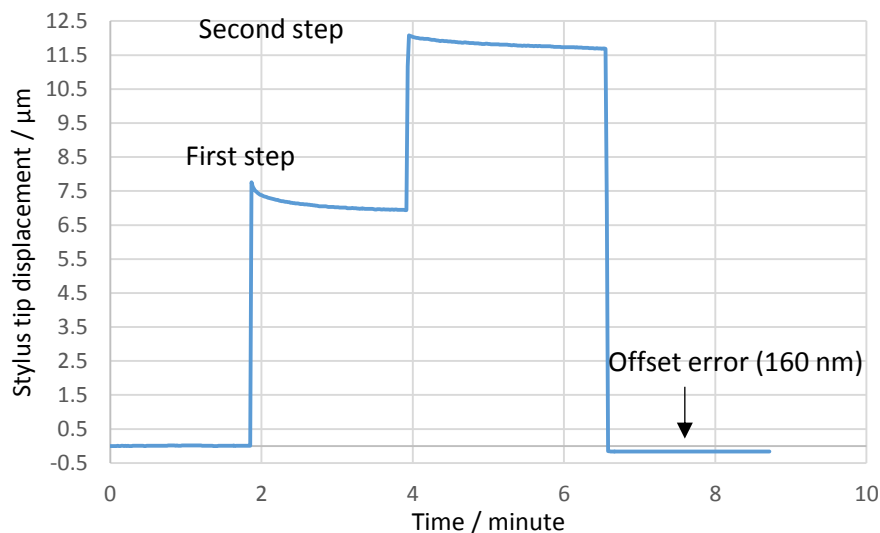


Figure 6-30: Output displacement from capacitive sensors against the displacement of the stylus tip in z direction.

To tackle these problems, some modifications have been made on the probing system. Firstly, the stylus has been glued on another side of the intermediate body. This modification has been made to assure that there is no glue on elastic beams during the assembly (see Figure 6-31 (a)) where the thickness of the intermediate body and elastic beams are 400 μm and 110 μm , respectively.

The second modification is to remove the glue between the layers. The layers that are located in the compliant mechanism have not been glued and small steel screws were used to fix them together (see Figure 6-31 (b)) instead of using the brass one. This also assists to reduce the creep of lateral movement that produce the axial compression of the beams. Figure 6-32 shows the initial test was conducted on the modified probing system. The system was improved where after 18 minutes the system was able to return to the original position within 10 nm. However, there is still a drift as capacitive displacement sensors are known to be affected by the conductivity or the air gap, and conductivity closely relates to humidity. This will be investigated in detail in chapter 7 section 7.3.1.1.

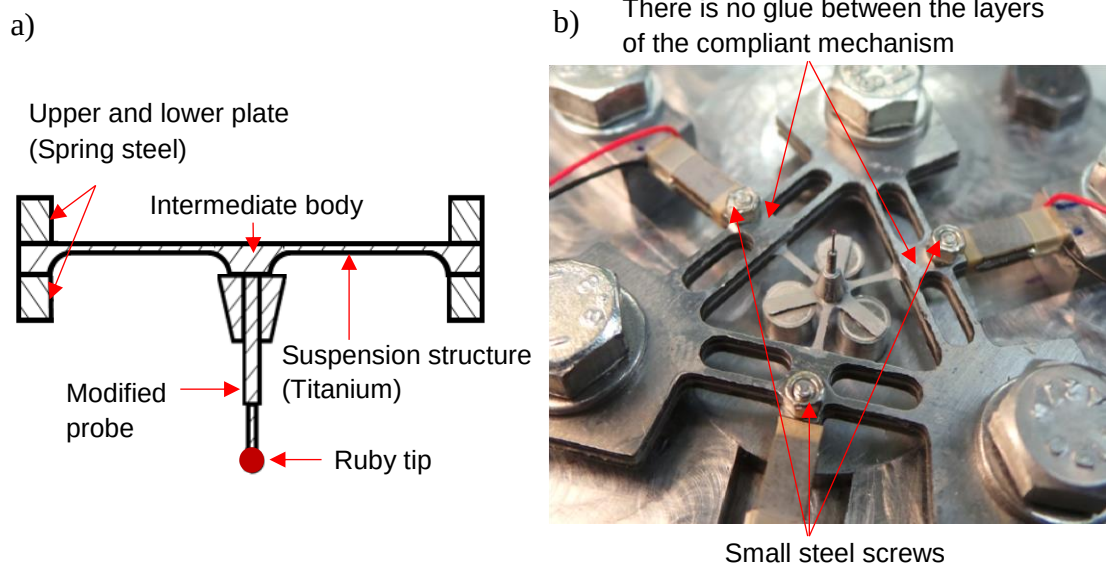


Figure 6-31: (a) Schematic cross-sectional diagram of the probing system with stylus length 5.6 mm and 300 μm ruby tip, (b) a modified probing system, with three mounted piezoelectric actuators for compressive force application, and beneath the suspension structure three capacitive sensors.

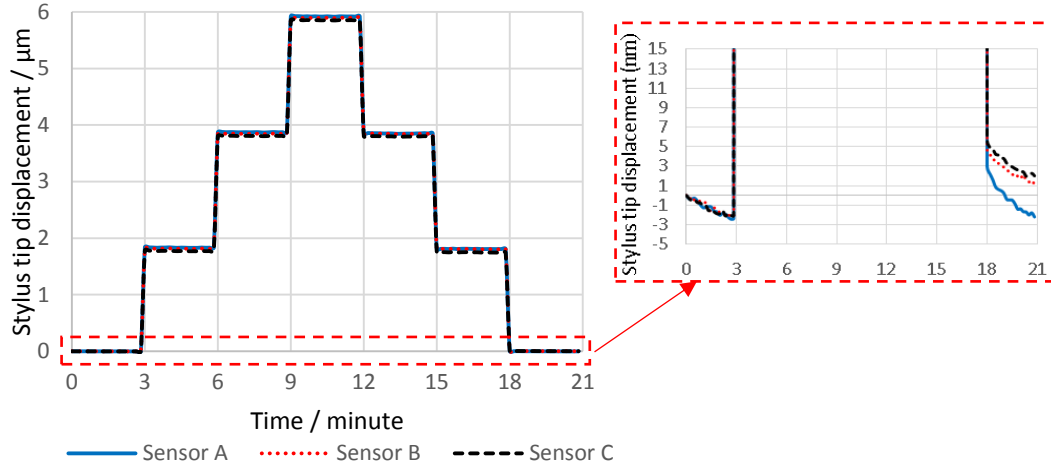


Figure 6-32: Output displacement from three capacitive sensors against the displacement of the stylus tip in z direction for modified probing system.

6.6 Chapter summary

In this chapter, a variable stiffness suspension structure for micro-tactile probing systems has been designed, fabricated and tested. Using this technique allows the stiffness of the suspension structure to be modulated during probing, avoiding many practical issues related to the stiffness of the probe being either too high or too low at different phases of a measurement.

A range of manufacturing techniques were considered to manufacture a prototype suspension structure, and chemical etching was selected as a suitable precision manufacturing technique. It has been shown that the suspension beam has a good flatness compared with production by conventional machining.

An analysis of the modal frequencies of the structure was undertaken to demonstrate that the stiffness modulation method was able to reduce the frequency of the vertical mode by approximately 70 %, while the torsional mode was decreased by approximately 33 % for a suspension structure which was made of spring steel. This level of reduction was shown to be in close agreement with a finite element simulation of the structure. Using the finite element model it was shown that vertical and lateral stiffnesses could be reduced to 788 N m^{-1} and 464 N m^{-1} of their initial value respectively using a 6.6 mm stylus length. Moreover, the ratio between these two stiffness values was reduced from 6.6 to only 1.7.

While the presented reduction in stiffness would represent a significant benefit, it has also been shown that the factors limiting the further reduction of stiffness are well understood and can be effectively modelled using the finite element method. It is therefore anticipated that further design and manufacturing process optimisation will result in a significant capability to provide even more stiffness reduction, and achieve perfect isotropic stiffness.

An optimized suspension structure that provides higher stiffness reduction was manufactured using titanium alloys (grade 5). Integrating a stylus (Renishaw, A-5000-7800), which has a 6.6 mm length, with a suspension structure has been demonstrated and tested. It has been shown that the lowest stiffnesses that can be achieved in the vertical and lateral directions at the turning point are around 440 N m^{-1} and 340 N m^{-1} respectively. The initial ratio between the vertical and lateral stiffness is reduced from 7 to 1.3, which is close to the isotropic stiffness.

Probe performance over longer periods was also assessed and the initial, glued, design was found to creep under applied load from stylus contact. A small modification of the probing system and the associated fixtures was presented and it was shown that the creep was largely eliminated, being reduced from 150 nm after 6.5 minutes of loading, down to only 10 nm after 18 minutes of loading. All data presented in the chapters that follow was generated using the unglued suspension.

Chapter 7 Operational strategies and Performance assessment of developed micro-CMMs probe

In this chapter, a performance evaluation of variable stiffness probing system is carried out. To set the context for the experimental evaluation, the intended measurement strategy for the probe is defined. In addition, the novel control system used to set, monitor and control probe stiffness is also described. The method used to complete a 3D calibration of the probe is then presented. Based on the calibration, a number of significant probing system errors are evaluated, including linearity, repeatability, and long-term stability. To investigate the capability of the probe to modify stiffness during use, the probe is assessed in both a stiff and flexible state, and also following repeated switching between these states.

7.1 The variable stiffness micro-scale probing system

The stiffness of the probing system can be modulated at two operating modes, stiff and flexible, which can be selected by switching the actuator force supplied by the three actuators between set point values. To investigate the relationship between applied actuator load and probe stiffness, a finite element model of the probe was created. In the finite element model, for simplicity, the effect of the actuators was modelled as a static displacement. For a range of applied actuator displacements, the stiffness and first two resonant modes of the probe were evaluated using a previously validated modelling process (see Chapter 6, section 6.3.6). Figure 7-1 shows the simulated axial displacement (d_o) that is introduced by the piezoelectric actuator plotted against the frequency and the stiffness of the probing system. The nominal working position of each mode is also shown on Figure 7-1, indicating the stiffness of the probe and the corresponding natural frequencies for each mode. From what has been observed, the vertical frequency is more sensitive to the axial displacement than the lateral frequency. Therefore, the vertical frequency will be used as feedback to monitor and control the stiffness; more detail will be discussed in section 7.1.2. The stiff mode is set at an actuator displacement of around 1 μm , resulting in modulated frequency in the vertical direction of 1057 Hz and a stiffness of approximately 2853 N m^{-1} in the vertical direction and around 610 N m^{-1} in the lateral direction. For the flexible mode, the displacement is set approximately at 2.2 μm and

the vertical frequency is 600 Hz, the vertical stiffness of 914 N m^{-1} and lateral stiffness of 410 N m^{-1} , as illustrated in Figure 7-1.

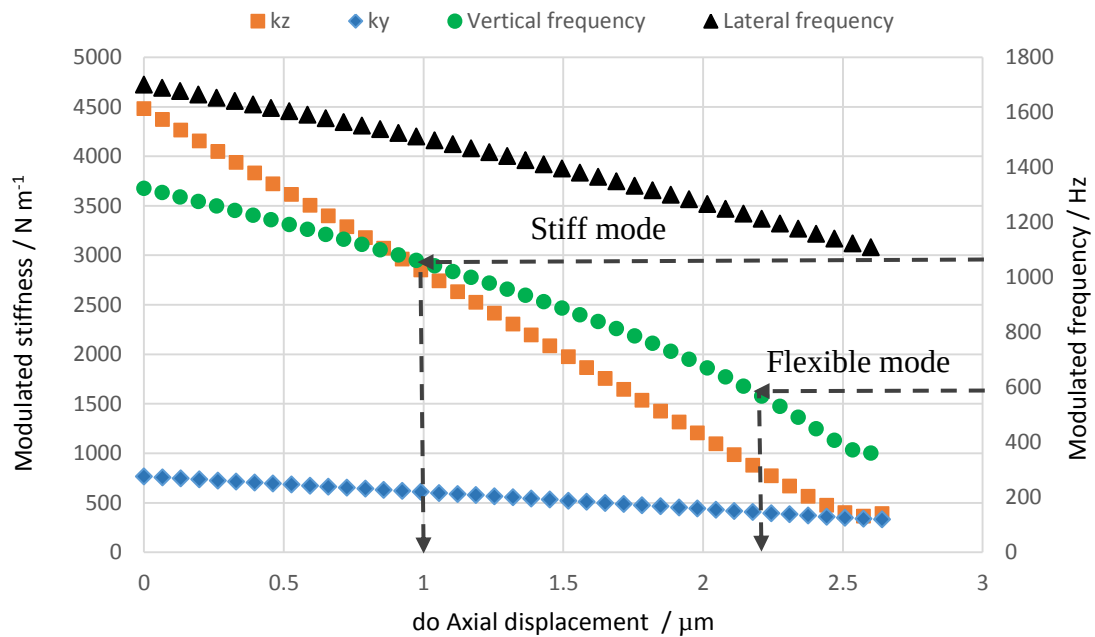


Figure 7-1: Simulated stiffness and frequency against the axial displacement showing the stiff and the flexible mode of the modified probing system, as calculated FEA analysis.

7.1.1 Operating sequence for the variable stiffness probing system

The sequence of operation for a variable stiffness probing system consists of four steps. The first step is to drive the probing system to a position where the stylus tip is close to the measurement surface, but not yet in contact. During the first step, it is advantageous for the micro-CMM to move the probing system as fast as possible towards the target surface. Therefore, the probe will be in the stiff mode to prevent inertial loads, generated as the probing system is moved by the micro-CMM, from causing misleading deflections of the probe tip. The second step is to switch the probe to be in a flexible mode, so it is ready to make contact with the target surface. During this step, a potential difference is applied across the piezo-electric actuators to achieve the desired reduction in stiffness. To ensure that a repeatable stiffness value is reached, a feedback control loop is used (see section 7.1.2). The third step is to carefully move the probing system towards the target surface, so that the stylus tip makes a controlled contact with the surface. During this step, the frequency-based

control loop is switched off, and most recent value of drive to the piezoelectric actuators is maintained, effectively operating the probe in an open loop condition for the brief period of the stylus contact. The fourth step occurs after a measurement has been made, when the probe must be removed from the surface. During this step, the probe is switched back to the stiff mode, still under open loop control, which makes it easier to develop sufficient force at the stylus tip to overcome attractive surface forces. The increased probe stiffness reduces the risk of overstraining the probe, which could result in measurement errors or damage to the probing system. Once stylus retraction has been achieved, closed loop control is reinstated.

7.1.2 Control strategy for the variable stiffness probing system

The open-loop control capability of piezoelectric stack actuators is significantly limited due to hysteresis and creep effects [129, 130]. Therefore, to use piezoelectric actuators in a high precision application requires the use of a closed-loop control system. For the variable stiffness probing system, a novel feedback control loop has been designed. The first modal frequency of the probe, which is a function of probe stiffness, is used as the control signal for the feedback loop. This is achieved by exciting the probe into resonance, and using the capacitive sensors to detect the resulting resonant frequency. The probe resonant frequency provides a control signal, which allows very repeatable probe stiffness values to be set. Figure 7-2 is the block diagram of the frequency-based proportional integral derivative (PID) control system. The feedback system also ensures that a constant probe tip position is set each time the probe is switched into flexible mode, which is important for ensuring accuracy and repeatability each time the probe is switched between the two modes.

To operate the flexible probing system in the two modes requires the use of two PID control loops. The two loops were implemented in LabVIEW software from National Instruments; Table 7-1 shows the specifications of the control system for tuning the frequency in both modes. In each case, the piezoelectric actuators were used to apply a superposition of a large static load combined with a small sinusoidal load where the small amplitude for the stiff and flexible modes is 0.3 V and 0.1 V, respectively. The static load is applied to provide a constant level of compressive force to the suspension structure, which is used to control the probe stiffness. The sinusoidal load is used to excite, at the first mode of vibration of the probing system into resonance.

To achieve this, a linear chirp signal is used to sweep the frequency from 1050 Hz to 1060 Hz for the stiff mode, and from 595 Hz to 605 Hz for the flexible mode. To analyse the resulting response of the suspension structure, a fast Fourier Transform (FFT) algorithm, also implemented using LabVIEW Real-time, was used.

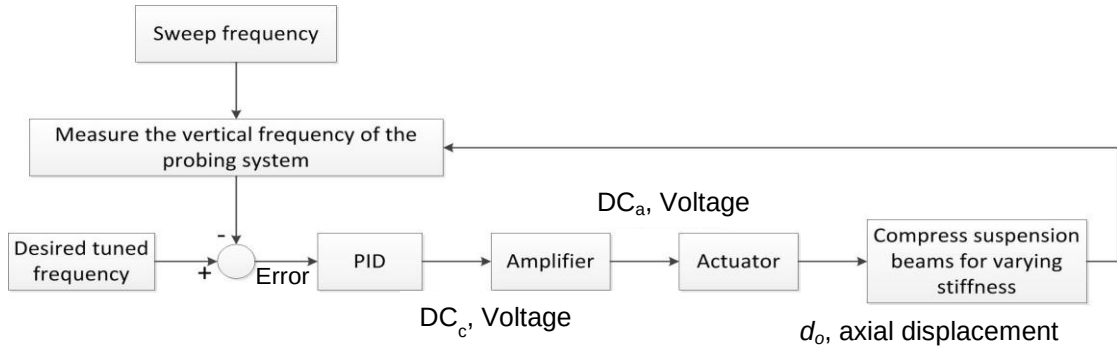


Figure 7-2: Block diagram of tuned stiffness for two modes (stiff and flexible). Note that DC_c and DC_a are the controlled voltage and amplified voltage, respectively.

Table 7-1: Specification of the control system used to tune the frequency in both modes

Mode	Time period /ms	DC voltage range/V	AC voltage /V	Tuned frequency/Hz	Sweeping frequency range/Hz
Stiff	51	5-10	0.3	1057	1050-1060
Flexible	51	41-55	0.1	600	595-605

7.2 Experimental setup for the probing system characterisation

To provide a stable environment for testing the capability of the variable stiffness probing system, the experimental work was completed in a temperature controlled laboratory ($20\text{ }^{\circ}\text{C} \pm 0.5\text{ }^{\circ}\text{C}$). An overview of the experimental setup is shown in Figure 7-3. The probing system was mounted on a three-legged platform placed on a pneumatic vibration isolation table; this minimised any disturbance due to ground vibration and air turbulence.

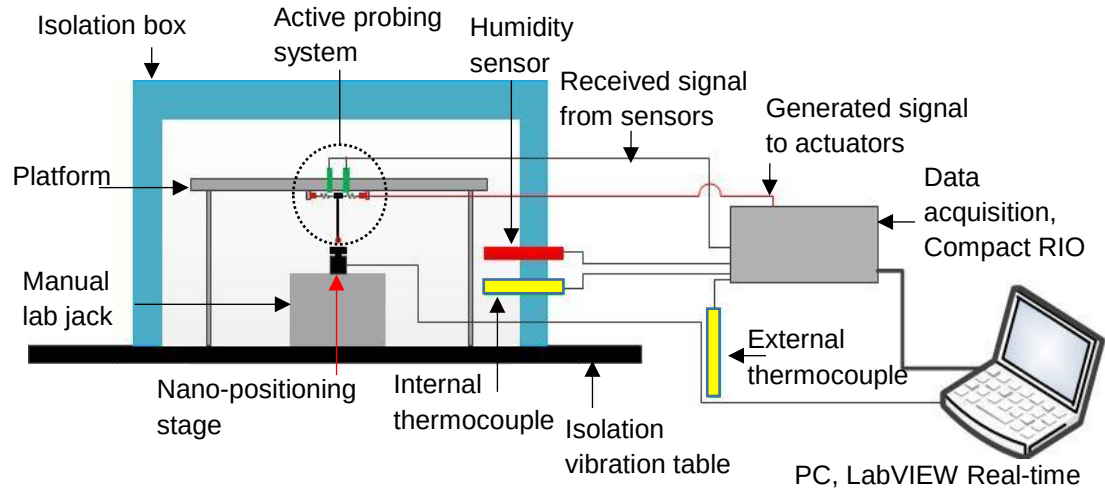


Figure 7-3: Schema of the setup for characterisation of the micro-probing system in z direction.

To simulate the measurement process, a nano-positioning stage (Physik Instrumente, P-611.ZS) was used to displace the probe tip while the probe fixture remained static. The nano-positioning stage has a 100 μm maximum displacement, nonlinearity error of 0.1 % and a repeatability of less than 10 nm [131]. In this work the repeatability and linearity of the PI stage affects the determination of the sensor sensitivity matrix coefficients, the measurement of probe linearity and the measured repeatability of the probing system; as such the measurements presented should be regarded in relation to the quoted performance for the PI stage. Figure 7-4 (a) is a photograph of the nano-positioning stage which was assembled with a manual lab jack to adjust the displacement between the stylus tip and the stage; in Figure 7-4 (a), the stage surface is perpendicular to the stylus, so vertical probe tip displacements can be applied (see Figure 7-4 (b)). To characterise the performance of the probing system in the lateral directions, the nano-positioning stage could be repositioned using the L-bracket, which is shown in Figure 7-4 (a), so lateral displacements could be applied, as shown by the schemas in Figure 7-4 (c). A polished aluminium surface was attached to the head of the nano-positioning stage which acted as the contacting surface ensuring repeatable contact to the stylus tip could be made.

To provide additional environmental stability, the actuators and probing system were housed within a foam enclosure. This enclosure provided an insulating barrier to reduce thermal fluctuations, and reduce the impact of acoustic vibrations and air

turbulence. Two thermocouples were used (see Figure 7-3) to measure the ambient temperature of the lab and the equipment setup inside the enclosure. A humidity sensor was also installed within the enclosure to account for the impact of humidity, which is known to affect the performance of the piezo-electric actuators [132] and capacitive sensors [133].

Signal synchronisation and excitation were achieved using National Instruments hardware (NI cRIO-9022) and LabVIEW FPGA (field programmable gate array). A lookup table was used to store waveforms for the linear chirp signal in the FPGA. A control system was designed to operate the probe using LabVIEW Real-Time. Coordination of the FPGA and Real-Time controller was achieved using Direct Memory Access, and the resulting control loop had a cycle time of approximately 51 ms. The maximum measuring rate of the capacitive sensor was 10 kHz, which is well above the maximum expected resonant frequency for the probing system (approximately 1 kHz).

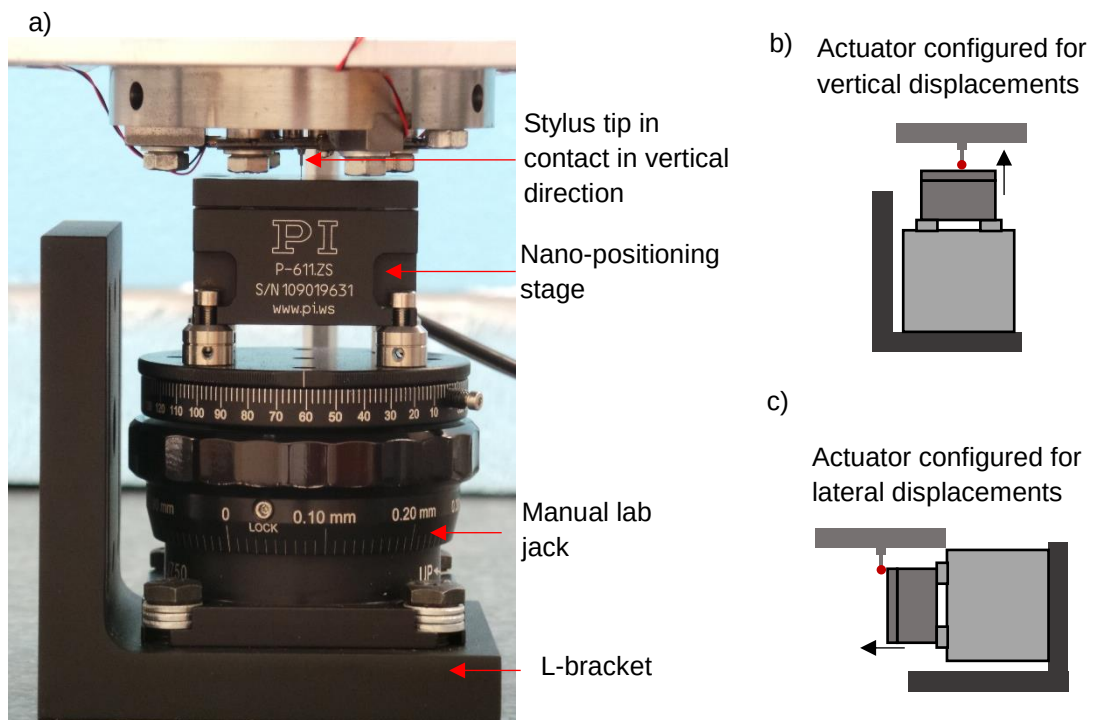


Figure 7-4: (a) An image of the experimental setup used for testing the micro-probe in the vertical direction, (b) actuator configured for lateral displacements, (c) actuator configured for vertical displacements.

7.3 Suspension structure performance evaluation

Two kinds of experiments have been designed to investigate the performance of the micro-probing system. First, the behaviour of the probing system in the stiff mode, where the system can work as a normal micro-scale probing system, is considered. This initial investigation determines long-term stability, sensitivity of the probe in 3D and repeatability. Second, the probing system is investigated in the flexible mode, and the ability of the probe to switch between stiff and flexible modes is examined. For example, is the probing system able to repeatedly switch from stiff to flexible mode with no resultant offset errors, and will the sensitivity of the probing system be changed as a result of switching between modes?

7.3.1 Characterisation of the probing system

7.3.1.1 *Environmental long-term stability*

The stability of the probing system in the lab environment was assessed over a continuous eighty hour period. During this time, the probing system output, the temperature and the humidity in the isolation enclosure (see Figure 7-3) were recorded. Within three hours of introducing the isolation enclosure, the rate of change of temperature was reduced to 0.005 °C per hour for the duration of the eighty hour test as shown in Figure 7-5. However, a more dramatic change was observed for the relative humidity over the same sample period, increasing from 35 % to 47 %. During this period, the probing system output was shown to drift by an amount equivalent to an 11 µm displacement of the stylus tip position. It can be seen in Figure 7-6, that there is an inverse correlation to stylus tip displacement error. As capacitive displacement sensors are known to be effected by the conductivity of the air gap [133], and conductivity closely relates to humidity, it would be reasonable to assume causation. The maximum observed rate of change of stylus tip displacement was shown to be 6 nm per minute, where the relative humidity was changed from 45 % to 39 % over sixteen hours. No attempt was made to compensate for this drift; however, to mitigate the impact of humidity, the time period of all subsequent tests was kept to a minimum. The time periods for each experiment are given with the detailed experimental description provided in the following sections.

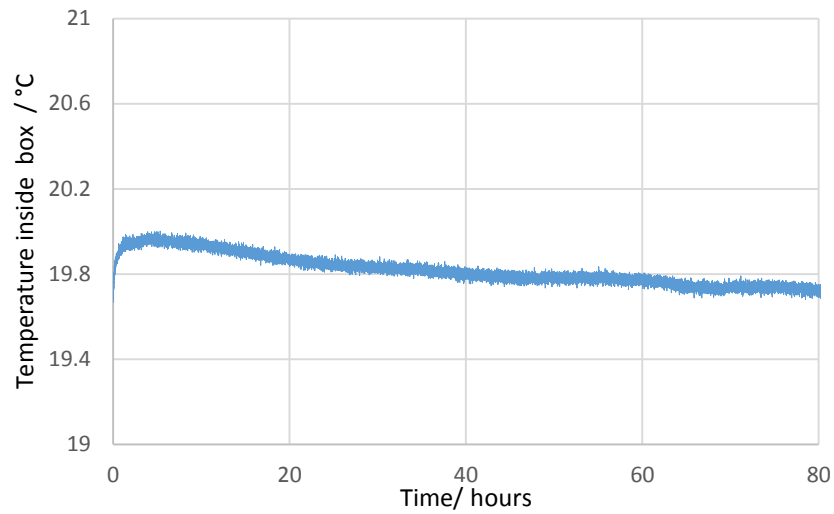


Figure 7-5: Temperature within the isolation box containing equipment setup 80 hours.

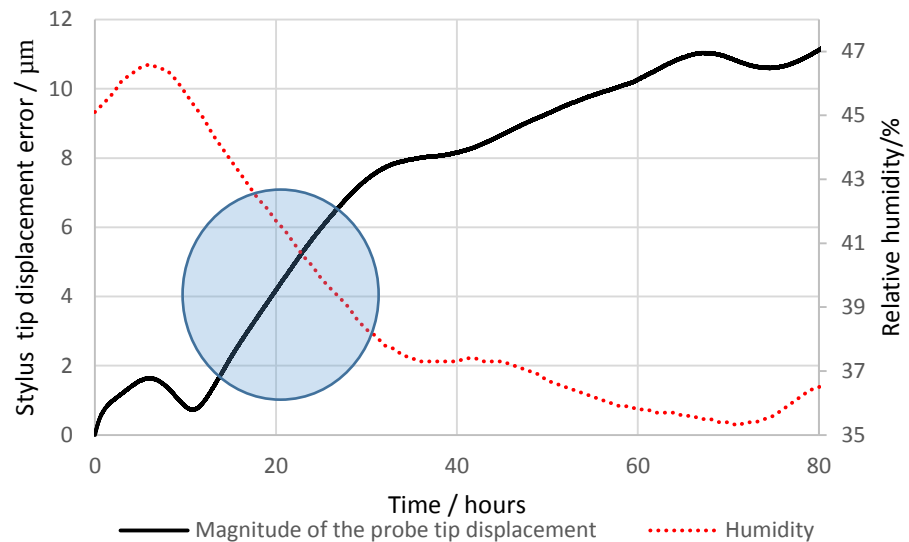


Figure 7-6: A plot of the magnitude of the stylus tip drift and relative humidity within the isolation enclosure over an eighty hour period. The steepest gradient area is indicated with an overlaid circle; in this region the rate of change of stylus tip position reaches a maximum of 6 nm per minute.

7.3.1.2 Performance in stiff mode

In order to accurately measure the position of a probed surface, a set of coefficients are required that can relate the sensor measurements to the Cartesian position of the stylus tip. For the probing system presented, it is not convenient to directly observe

Cartesian displacements, i.e. displacement along the x , y and z axes, as shown in Figure 7-7. Instead, it is necessary to first relate sensor response to motion along the more convenient δ_1 , δ_2 , δ_3 and δ_4 displacement vectors described below, and then relate these to motion in a Cartesian frame (this is hereafter referred to as calibration). To calibrate the probing system in the stiff mode, the stylus tip was displaced over an incremental range using the nano-positioning stage. To minimise the effect of humidity-related drift, the calibration was conducted over a one minute period for each direction. To correctly align the nano-positioning stage for the three lateral displacements, the triangular body of the probe fixture was used to provide datum faces (see Figure 7-7(b)). To approach the probe tip, 100 nm incremental steps were carried out until contact was registered by the micro-probing system. Once contact had been made with the stylus tip, the output from the three capacitive sensors was recorded over a displacement range of 14 μm , with incremental steps of 2 μm .

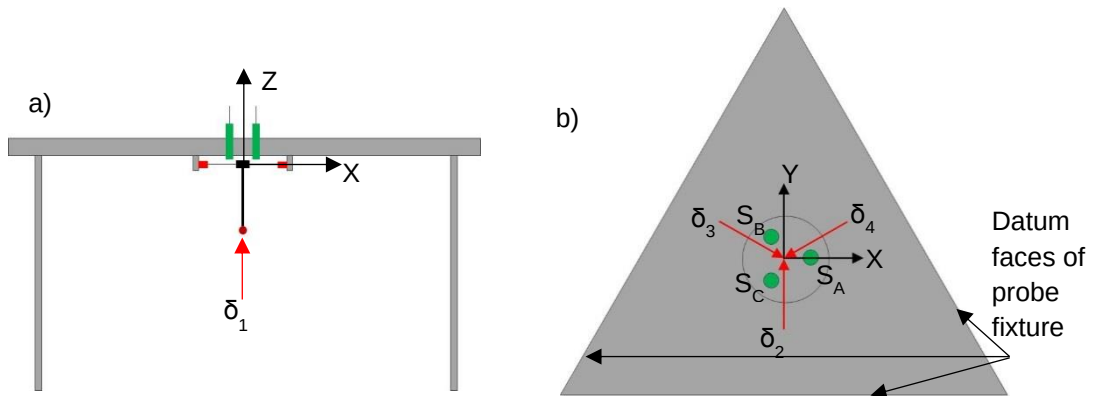


Figure 7-7: Schema showing (a) a front view of the platform that was used to calibrate the probing system in z axis, (b) a top view of the platform that indicate the position of three capacitive sensors and the calibration directions of the probing system in x and y axes.

The output from the three capacitive sensors, for δ_1 , δ_2 , δ_3 and δ_4 displacement vectors, can be seen in Figure 7-8(a), (b), (c) and (d) respectively. While all capacitive sensors read almost the same output voltage for displacements in the z direction, suggesting that the structure does not twist when displaced vertically. However, the displacement of the stylus tip in the y direction leads to rotation of the intermediate body about the x axis, where sensor B and C respond in opposite sign,

but similar magnitude, and sensor A remains constant due to its positioned at the centre of rotation.

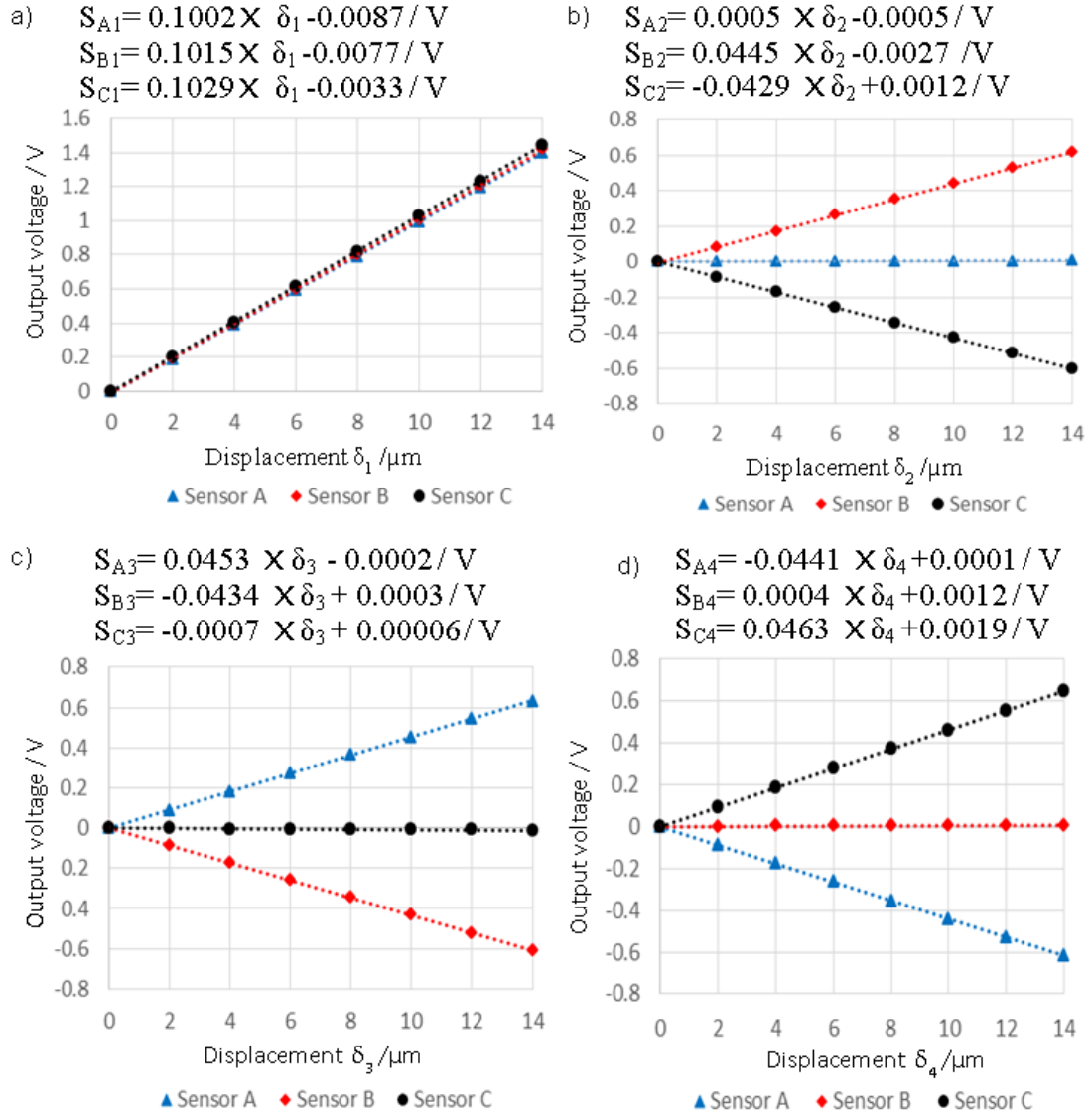


Figure 7-8: Output voltage from three capacitive sensors against the displacement of the probe tip in each of the four displacement ranges.

The data in Figure 7-8 was used to form a 3×4 matrix of sensitivity coefficients based on a least-squares linear fit. A sensitivity matrix can be approximated using the first-order Taylor series when the working range is small [134], see equation (7-1)

$$\begin{bmatrix} \Delta S_A \\ \Delta S_B \\ \Delta S_C \end{bmatrix} (V) = \begin{bmatrix} \frac{\partial S_{A1}}{\partial \delta_1} & \frac{\partial S_{A2}}{\partial \delta_2} & \frac{\partial S_{A3}}{\partial \delta_3} & \frac{\partial S_{A4}}{\partial \delta_4} \\ \frac{\partial S_{B1}}{\partial \delta_1} & \frac{\partial S_{B2}}{\partial \delta_2} & \frac{\partial S_{B3}}{\partial \delta_3} & \frac{\partial S_{B4}}{\partial \delta_4} \\ \frac{\partial S_{C1}}{\partial \delta_1} & \frac{\partial S_{C2}}{\partial \delta_2} & \frac{\partial S_{C3}}{\partial \delta_3} & \frac{\partial S_{C4}}{\partial \delta_4} \end{bmatrix} \begin{bmatrix} \delta_1 \\ \delta_2 \\ \delta_3 \\ \delta_4 \end{bmatrix}. \quad (7-1)$$

The coefficients in equation (7-1) relate the displacement of the stylus tip in the four directions described (see Figure 7-7) to the three capacitive sensor voltage responses. Equation (7-2) [135] was used to give the inverse to the coefficients in equation (7-1), thus

$$\begin{bmatrix} 0 & 0 & \sin 60 & -\sin 60 \\ 0 & 1 & -\cos 60 & -\cos 60 \\ 1 & 0 & 0 & 0 \end{bmatrix} = \begin{bmatrix} T_{XA} & T_{XB} & T_{XC} \\ T_{YA} & T_{YB} & T_{YC} \\ T_{ZA} & T_{ZB} & T_{ZC} \end{bmatrix} \begin{bmatrix} \frac{\partial S_{A1}}{\partial \delta_1} & \frac{\partial S_{A2}}{\partial \delta_2} & \frac{\partial S_{A3}}{\partial \delta_3} & \frac{\partial S_{A4}}{\partial \delta_4} \\ \frac{\partial S_{B1}}{\partial \delta_1} & \frac{\partial S_{B2}}{\partial \delta_2} & \frac{\partial S_{B3}}{\partial \delta_3} & \frac{\partial S_{B4}}{\partial \delta_4} \\ \frac{\partial S_{C1}}{\partial \delta_1} & \frac{\partial S_{C2}}{\partial \delta_2} & \frac{\partial S_{C3}}{\partial \delta_3} & \frac{\partial S_{C4}}{\partial \delta_4} \end{bmatrix}. \quad (7-2)$$

In equations (7-1) and (7-2), the first subscript denotes the Cartesian axis they relate to, and the second subscript denotes the sensor. Finally, equation (7-3) relates the sensor voltage directly to the displacement of the stylus tip with respect to x, y and z axes

$$\begin{bmatrix} \delta_x \\ \delta_y \\ \delta_z \end{bmatrix} = \begin{bmatrix} -12.9344 & 6.3118 & 6.3644 \\ -0.2157 & 11.5794 & -11.2016 \\ 3.3145 & 3.2608 & 3.2725 \end{bmatrix} \begin{bmatrix} \Delta S_A \\ \Delta S_B \\ \Delta S_C \end{bmatrix} \quad (7-3)$$

7.3.1.3 Repeatability in the stiff mode

It is important for accurate probe measurements that the stylus tip has a well-known and repeatable position before and after being displaced by contact. Therefore, the repeatability of the stylus tip initial position was evaluated for both lateral (y) and vertical (z) displacements (x direction data is omitted for brevity as it is fair to assume that it is equal to that in the y direction based on structural symmetry). For each displacement direction, the repeatable return to a datum position, henceforth called the ‘zero offset condition’ was recorded for three nominal displacements of 4 μm , 8 μm and 14 μm . For each of the nominal displacements, the probe was

displaced and then returned to the zero offset condition, with ten repeats being made at each displacement. Data is presented for both the return to zero offset, and the deviation between readings for the nominal displacements. Figure 7-9(a) and (b) show the result of repeatability in the z and y directions respectively. Note that the repeatability values were estimated by calculating the standard deviation of ten readings taken over ten repeated displacements of the probe tip followed by a return to the zero offset condition. It is clear that the suspension structure of the probing system has good repeatability, even for the larger, 14 μm displacement, where the zero offset was maintained at less than 3 nm vertically and 1 nm laterally. The zero offset error slightly increases as the nominal displacement for the z direction increases. The exact reason for the increase in zero offset error is not known; however, it is reasonable that as nominal displacement increases mechanical hysteresis and material related non-repeatability errors are also likely to increase.

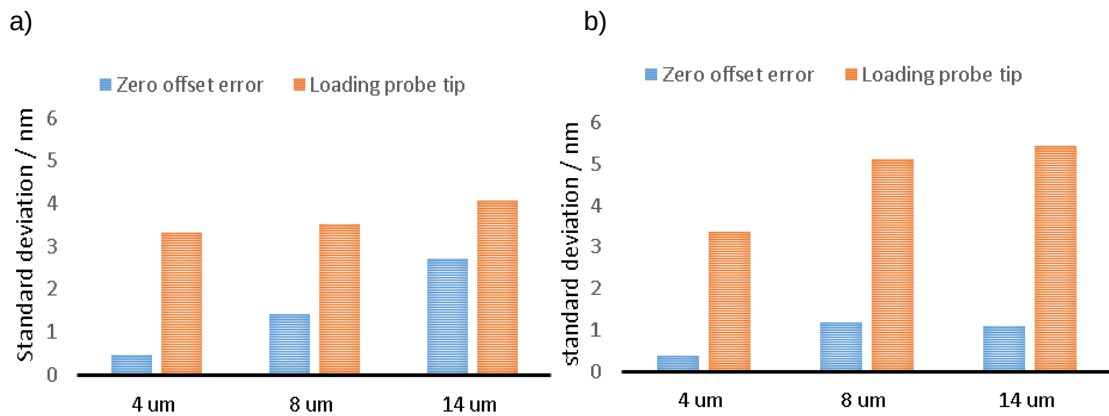


Figure 7-9: Bar charts showing repeatability in stiff mode for three nominal displacements (4 μm , 8 μm and 14 μm). (a) Shows results for the z direction, (b) shows results for the y direction.

7.3.1.4 Performance in flexible mode

As described in section 2.2, stiffness control in the flexible mode is achieved using a frequency-based closed-loop control system, which requires the probing system to be vibrating. To avoid any potential errors due to probe tip vibration, the control system must be stopped prior to contact. This has the unwanted effect of allowing the piezo-electric actuator to drift in an open loop condition. The amount of drift depends on the recent drive history of the actuators and creep can, therefore, be reduced by

initialising the piezo-electric actuator using a set of load cycles prior to use [136]. The number of cycles needed to reduce the influence of the creep was investigated. The closed loop control was run for one minute to maintain the frequency at 600 Hz, and then the controller was switched to open loop to allow the drift to be measured. Figure 7-10 shows the result of an experiment where the vertical frequency was controlled to (599.997 ± 0.013) Hz and after one minute, the closed-loop control was switched off. The probe tip amplitude of the vertical mode of vibration was estimated from the AC voltage of the capacitive sensors and calculated to be approximately $2.4 \mu\text{m}$. The sweeping chirp signal remains on to monitor the resonant frequency. It can be clearly seen that the piezoelectric actuators drift, which results in the first steep downward slope shown in Figure 7-10. Once the control system is restarted, the frequency rapidly returns to the 600 Hz set point. The test was repeated after a further minute, and it can be seen that the drift-related frequency change was greatly reduced. This trend was found to continue, and after a three minute period, drift was around 7 nm, as shown in Figure 7-11. Considering these results, ongoing tests of the probe in the flexible mode were conducted following an initial period of five minutes to reduce actuator creep.

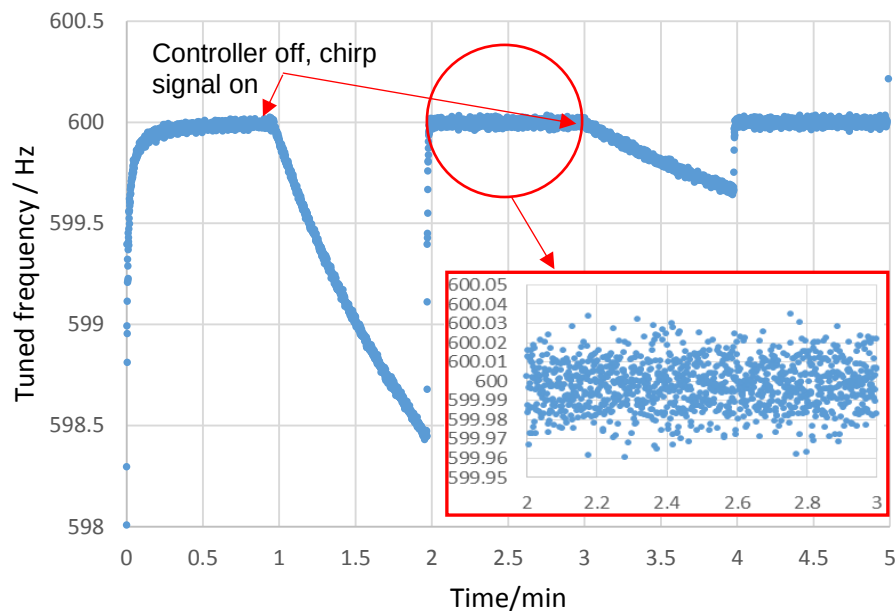


Figure 7-10: Frequency plot illustrating how actuator drift reduces following a series of actuator load cycles.

Using the transfer function described in equation (7-4) (see below), the impact of actuator creep on the stylus tip displacement was calculated. From Figure 7-11, it can be seen that after one and three minute periods, the actuator creep is around 35 nm and 7 nm at the stylus tip respectively. The creep in the three piezo-electric actuators was reduced as a result of repeating and recycling after a further minute.

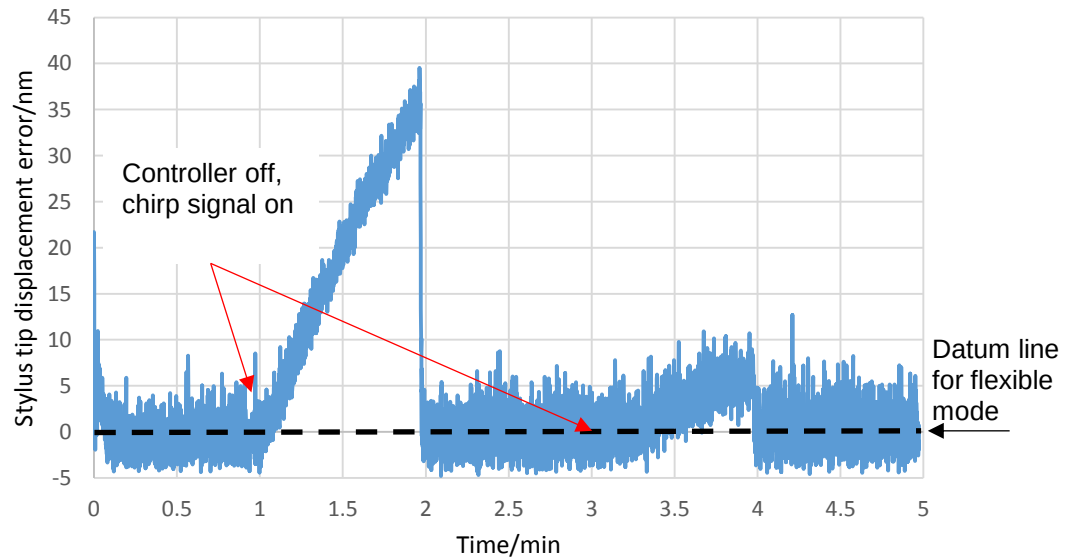


Figure 7-11: Frequency plot showing the magnitude of the stylus tip drift as a result of the impact of actuator creep at the flexible mode (600 Hz). This drift at the stylus tip and the tuned frequency in Figure 7-10 were measured at the same time.

In order to investigate the sensitivity of the probing system in the flexible mode, it was calibrated using the same procedure as for the stiff mode. The position of the nano-positioning stage was maintained between the calibration of the stiff and flexible mode in each direction to minimise error. The calibration was completed within one minute, reasonably limiting drift to less than 6 nm, as described above. Figure 7-12(a) and (b) show the calibration time against the output voltage from the three capacitive sensors for displacement in the first and second directions respectively. The step height was 1 μm , and the datum point was taken as 100 nm beyond the first detected contact, to ensure a reliable contact between stylus and surface.

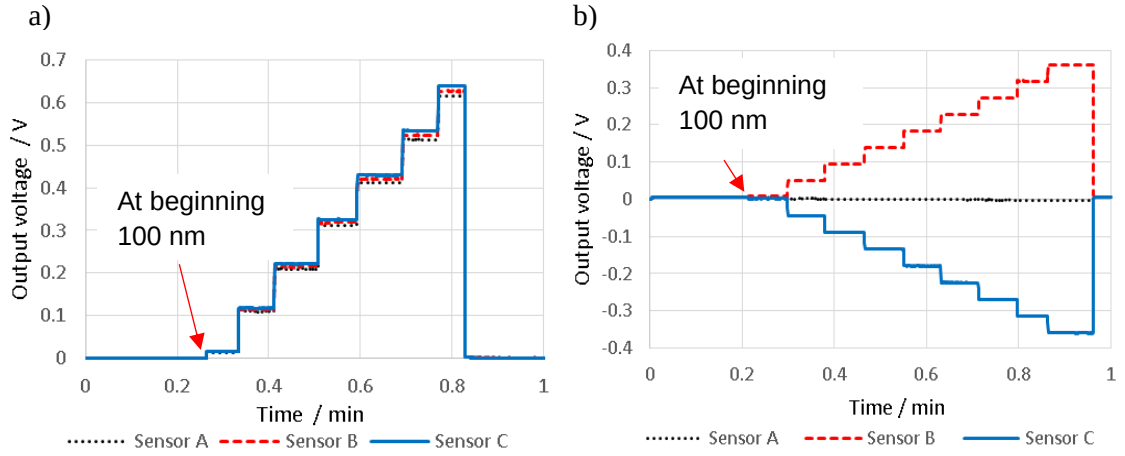


Figure 7-12: Output voltage from three capacitive sensors against the displacement of the probe tip in (a) z direction and (b) y direction, which are equal to 1 μm to generate probe calibration in vertical and lateral direction respectively.

The transfer function resulting from the calibration of the four directions is given in equation (7-4)

$$\begin{bmatrix} \delta_x \\ \delta_y \\ \delta_z \end{bmatrix} = \begin{bmatrix} -12.7610 & 6.1509 & 6.2700 \\ -00.1463 & 11.3783 & -11.0249 \\ 3.6676 & 3.2576 & 2.8760 \end{bmatrix} \begin{bmatrix} \Delta S_A \\ \Delta S_B \\ \Delta S_C \end{bmatrix}. \quad (7-4)$$

In order to illustrate the linearity of the probe's response to displacement, the residual is plotted in Figure 7-13(a) and (b) for the z and y directions respectively. The maximum residuals in the z and y directions are approximately 23 nm and 4 nm respectively.

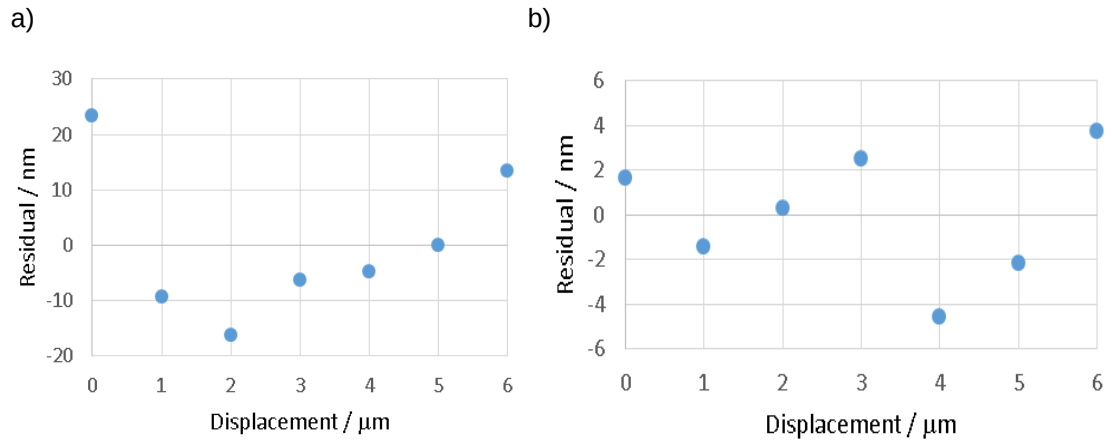


Figure 7-13: Plot shows the residual against fitted value, where the maximum nonlinearity is (a) 23 nm in z direction and (b) 4 nm in y direction for full range (6 μm).

To demonstrate that changing between modes does not significantly affect the calibration, a calibration was first performed in flexible mode, and then the probing system was set to stiff mode before returning to flexible mode for a repeat calibration. The coefficients generated from each calibration cycle are presented in in Table 7-2. There is a small change in the coefficients, approximately 0.4 % and 0.9 % in the z and y directions respectively. To illustrate the effect of the repeated calibrations, the second set of calibration coefficients for direction 1 was applied to the first set of data, and a new residual error was calculated. For the z direction, the maximum residual errors were found to be 23 nm, and changed to 17 nm. In the y direction, the maximum residual errors were initially 4 nm and changed to 5 nm, which is an acceptably low change in residual error.

Table 7-2: Sensitivity coefficients of the three capacitive sensors at the probe tip for displacement in $z(\delta_1)$ and $y(\delta_2)$ directions, before and after switching between flexible and stiff operating mode.

Item	Calibration coefficients for direction 1/ $\text{mV}\mu\text{m}^{-1}$		Calibration coefficients for direction 2/ $\text{mV}\mu\text{m}^{-1}$	
	First trial	Second trial	First trial	Second trial
Sensor A	$\frac{\partial S_{A1}}{\partial \delta_1} = 100.5$	$\frac{\partial S_{A1}}{\partial \delta_1} = 100.7$	$\frac{\partial S_{A2}}{\partial \delta_2} = 0.6$	$\frac{\partial S_{A2}}{\partial \delta_2} = 1.1$
Sensor B	$\frac{\partial S_{B1}}{\partial \delta_1} = 101.9$	$\frac{\partial S_{B1}}{\partial \delta_1} = 101.7$	$\frac{\partial S_{B2}}{\partial \delta_2} = 44.6$	$\frac{\partial S_{B2}}{\partial \delta_2} = 44.2$
Sensor C	$\frac{\partial S_{C1}}{\partial \delta_1} = 103.9$	$\frac{\partial S_{C1}}{\partial \delta_1} = 103.5$	$\frac{\partial S_{C2}}{\partial \delta_2} = 45.1$	$\frac{\partial S_{C2}}{\partial \delta_2} = 45.5$

As well as determining the effect that switching from stiff to flexible mode has on sensitivity, it was also necessary to determine any unwanted stylus tip position offset error. This was determined with the knowledge that the load applied to the beams will result in a deflection of the stylus tip, which is most dominant in the z direction. If the probing system was not able to return to the same initial zero offset when switched to flexible mode, it would directly impact the probing system error. The only option to reduce this error would be to recalibrate the probe each time the flexible mode was switched on, which would not be practical. As both natural frequency and displacement are functions of applied compressive force, it is possible to define displacement as a function of frequency; so for a given frequency there is an unambiguous corresponding displacement. The ability of the probing system to maintain a consistent zero position when set to flexible mode will therefore be limited by two factors. These factors are the ability of the control system to maintain a stable frequency, and the sensitivity of frequency to displacement change; for example, if the control system is able to maintain a perfect frequency with no error, then the displacement must also be perfectly controlled. However, if there is some error, then the degree to which this causes the stylus tip to be deflected will depend of the sensitivity of stylus tip deflection to frequency change; a low sensitivity will result in a low positional error and vice versa.

To overcome this issue, the frequency-based probe stiffness control system (section 7.1), was used. To determine the capability of the frequency-based probe stiffness control system to maintain stylus tip position, the relationship between stylus tip position (displacement in the z direction) and the natural frequency of the structure

was determined; the results can be seen in Figure 7-14. This data was collected by setting the frequency control system to target 600 Hz, and then switching off the control system so only a steady state voltage is applied to the actuators. Under these conditions the actuators are free to creep under the influence of environmental conditions. This results in a steady increase in the applied force which results in the observed reduction in frequency from 600 to 598.4 Hz. While the frequency was reducing, the output from the capacitive sensors was monitored to allow the resulting displacement of the stylus tip to be measured.

The relationship is effectively linear over the small range of frequency deviations from the nominal 600 Hz control position. From a linear least-squares fit, the sensitivity of the deflection of the probe tip in the z direction as a function of tuned frequency is 26 nm Hz^{-1} . The control system is able to tune the frequency within a range of 0.075 Hz (see inset on Figure 7-10), so based on this the deflection, the probe tip can be controlled to within 2 nm in the z direction. Using frequency-based control, therefore, provides an effective method of controlling the flexible mode stiffness.

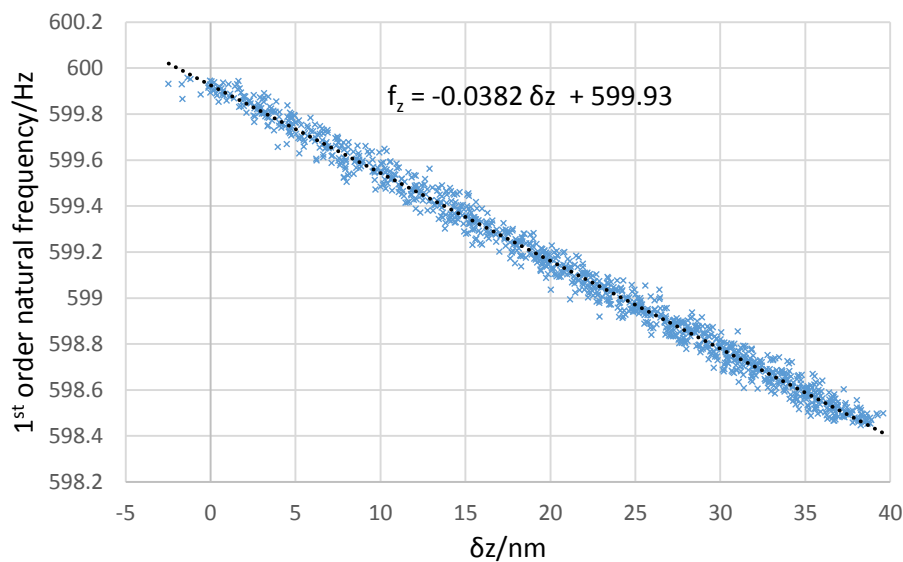


Figure 7-14: Plot showing the sensitivity of vertical frequency with respect to the displacement at the probe tip in the z direction.

The displacement that occurs between the ‘at rest’ unloaded positions when making repeated switches between stiff to flexible control modes, henceforth referred to as

switching repeatability, was determined. To measure the switching repeatability, the control mode was switched every ten seconds for a total of nine cycles, and the tip displacement monitored throughout. For this experiment the stylus tip was not constrained, ensuring true 'at rest' position were measured. The results are shown in Figure 7-15. To obtain the data plotted in Figure 7-15, the DC voltage from the three capacitive sensors was recorded, in stiff and flexible mode, and then flexible and stiff calibrations were used to calculate the stylus tip positions.

The stylus tip was deflected by a total magnitude of $18.54\text{ }\mu\text{m}$ when transitioning to flexible mode. This magnitude displacement is equal to $-12.78\text{ }\mu\text{m}$, $4.36\text{ }\mu\text{m}$ and $12.51\text{ }\mu\text{m}$ in x , y and z direction. This lateral deviation is caused by the rotation of the intermediate body as a result of unequal displacement in three actuators. The deflection of the stylus tip by this amount could cause large contact forces to occur when the probe is changed from flexible to stiff mode. To accommodate this displacement, three displacement sensors could be integrated to make sure three displacements are equal from each actuator, which would avoid the lateral deflection of probe tip during the translation between modes. Then to account for the motion in vertical direction, the probe tip could be manipulated by the CMM in synchronisation with switching of the mode. This would limit the contact force generated by the probing system. Even if the displacement sensors were not applied, and a complex displacement in three directions was observed, the CMM could still be controlled in synchronisation with the probing system such that as the stiffness is changed the CMM retracts the stylus tip to compensate for any resulting deflection. This compensation procedure would be learned as part of the calibration routine for the probe, by slowly switching stiffness and monitoring the stylus tip deflection as a function of actuator voltage; the learned compensation displacement path would be applied by the CMM during stiffness mode transitions.

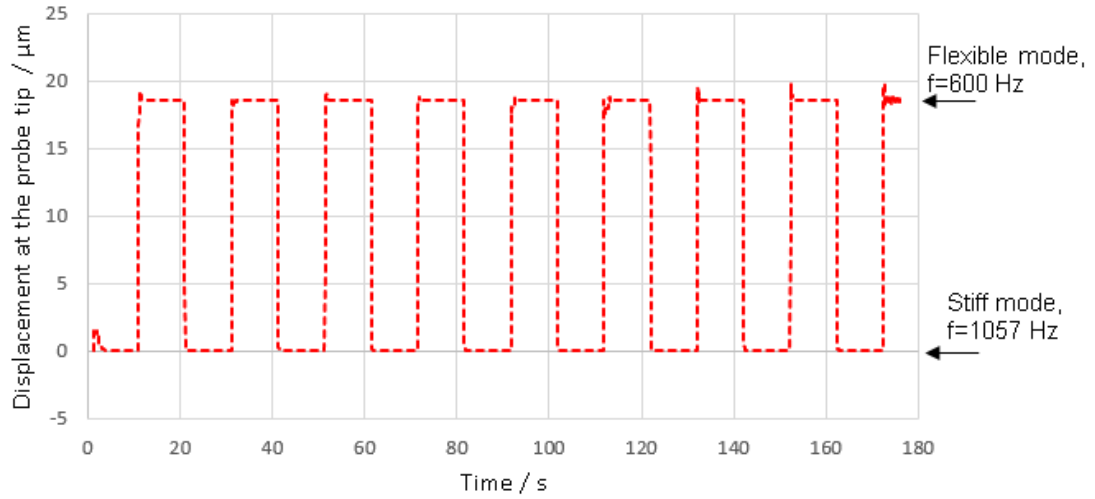


Figure 7-15: Plot showing control of the position of the probe tip in three dimensions between stiff and flexible mode using resonant frequency (the frequency of the stiff and flexible mode are 1057 Hz and 600 Hz respectively).

The frequency according to Figure 7-15 can be controlled within 0.12 Hz (standard deviation). Therefore, the stiffness variability in the flexible mode can be calculated based on Equation (7-5); where σ_k , σ_f and m_{eff} denote the standard deviation of the vertical stiffness, the standard deviation of the tuned frequency and the effective mass. The effective mass in the vertical direction is estimated based on the FEA (65 mg), this gives a stiffness variability of 0.04 N mm^{-1} , which is 4.5 % of the nominal value at 600 Hz.

$$\sigma_k = (2\pi\sigma_f)^2 m_{eff} \quad (7-5)$$

Table 7-3 shows the repeatability for the performance of the control system switching between stiff and flexible mode. The maximum standard deviation is 12 nm in the z direction. This standard deviation is relatively high compared with the estimated capability to control the deflection within 2 nm. It is expected that, if longer periods than ten seconds were used to allow increased settling time for the control loop, the standard deviation would be reduced.

Table 7-3: Stylus tip positional repeatability following repeated stiffness mode switching, calculated based on a series of eight repeated switches from flexible mode to stiff mode and then back to flexible mode, as illustrated in Figure 7-15.

Axis	<i>x</i>		<i>y</i>		<i>z</i>	
Mode	Stiff	Flexible	Stiff	Flexible	Stiff	Flexible
Standard deviation/nm	3	2	3	1	6	12

7.3.1.5 Repeatability of flexible mode probing

The repeatability of the flexible mode was determined, in a similar manner to that in section 7.3.1.2, by setting the probing system to flexible mode and taking a series of repeat measurements.

The repeatability of the probing system was evaluated at zero displacement, and at a nominal displacement of 4 μm applied in the *z* and *y* directions. A total of ten repeat measurements were made over a one minute period. Table 7-4 shows the results for the repeatability test; the probe tip has a zero offset repeatability, after loading, with a standard deviation of 1 nm, and at the 4 μm applied displacement, the repeatability was 3 nm.

Table 7-4: Repeatability of the flexible mode for the *x* and *y* directions. The measurement was repeated ten times for each displacement (nominal displacement is 4 μm).

Axis	<i>y</i>		<i>z</i>	
Type of error	Zero off-set position	Nominal 4 μm displacement position	Zero off-set position	Nominal 4 μm displacement position
Standard deviation/nm	1	3	2	3

The errors observed for the stiff condition, flexible condition and the effect of any offset during switching are considered together to arrive an uncertainty value for the probing system. This data is presented, along with suggestions for improvements in the probes design, in the following chapter summary.

7.4 Chapter summary

In this chapter, a performance of variable stiffness for micro-CMM probing systems has been evaluated. The key novelty in the micro-probing system presented here, is

the ability to switch between the stiff and flexible operating modes. To investigate this capability of the probing system, a novel control system based on measuring resonant frequency has been developed and a series of experiments were performed to characterise the stability and repeatability of the probing system.

The data produced from these experiments allowed for a complete a 3D calibration of the probes sensitivity characteristic, considering repeatability and application as a scanning type probe. As a starting point, the stability of the probing system was characterised in both stiff and flexible modes. A small drift resulting from relative humidity variation was observed. This drift was expected due to the use of capacitive sensors within the probe, which are sensitive to changes in relative permittivity caused by humidity changes. To mitigate for this source of drift, an alternative sensing technology could be used, such as optical or piezo-resistive, both of which have been successfully demonstrated in micro-scale probing systems [37, 68]. Alternatively, a compensation circuit using a reference capacitor or humidity sensor could be used to account for the drift. In this work, no compensation for the drift was made; instead all experimental work was conducted in a controlled laboratory environment, which limited humidity-related drift to less than 6 nm per minute (see Figure 7-6). A second source of drift, resulting from the piezo-actuators used in the probe, was also observed. To minimise actuator-related drift in future work, it is suggested that piezo-actuators with integral strain gauges should be used. These are commonly used in high precision nano-positioning stages, and can effectively eliminate actuator drift. In this work, standard actuators were used, and a period of actuator cycling was conducted to reduce actuator related drift to 5 nm per minute (see Figure 7-11). To further reduce the influence of both forms of systematic drift, all experimental procedures were conducted over as short a period of time as possible, typically one minute or less.

Once environmental stability was known a stiffness transformation matrix was constructed under 'stiff' conditions and then repeated after the probe was actuated 'stiff'>'flexible'>'stiff' condition. Results show the probing system has good repeatable sensitivity calibration with the maximum impact of calibration discrepancy calculated as 9 nm over a 14 μm displacement range for the stiff mode and 5 nm over a 4 μm range for flexible mode, respectively. When actuated from one mode to another the suspension geometry is altered, and the stylus tip does move, but

the rest position for both modes is shown to be predictable and repeatable to within 8 nm for the stiff mode and 12 nm for the flexible mode.

As the probe is intended to perform measurements of surface position in the flexible mode, the performance of the probing system in this mode was examined from three perspectives. First, the maximum linearity error of the probing system was found to be 24 nm, over a full working range 6 μm , as presented in Figure 7-13. This value represents the magnitude of the stylus tip displacement, and is based on the assumption that the linearity error in the y direction is equal to linearity error in x direction. Second, the variation in stylus tip offset position, following excursions from flexible to stiff mode and then returning, was found to have a standard deviation of 12 nm (see Table 7-3). Third, while in the flexible mode, the probing system was able to measure a nominal 4 μm displacement with a standard deviation of 5 nm (determined by taking the magnitude of the z , y and x values), the values for z and y are indicated in Table 7-4; again assuming that the repeatability in the y direction is equal to that in the x direction.

In addition to the probing system performance, the performance of the PI precision stage must also be considered. The precision stage used was quoted, by the manufacturer, to have 0.1 % nonlinearity error and 10 nm repeatability. As the probing system was calibrated over a 6 μm range the expected nonlinearity error over this range is 6 nm. The repeatability of the PI stage was quoted by the manufacturer to be better than 10 nm; assuming the stage repeatability is uniformly distributed then the expected standard deviation of stage position can be estimate to be 6 nm.

Combining stage errors with the probing system linearity error, the stylus tip zero offset position error and the probing system measurement repeatability, gives an estimate of a combined uncertainty for the probing system or of 29 nm. The dominant uncertainty comes from the linearity error of the probing system. The expanded uncertainty is 58 nm (coverage factor, $k = 2$). This validates the feasibility of variable stiffness probing systems; showing the ability to switch between stiff and flexible modes with minimal impact on probing system performance.

Chapter 8 Conclusion and future work

This thesis has been concerned with improvement in the practise of micro coordinate metrology, with the specific aim of **developing a tactile micro-probing system that offers variable suspension stiffness so as to operate optimally in each phase of a tactile measurement.** The need for a variable stiffness probe came from the observation that none of the currently published technologies had overcome the conflict between the requirement for flexible response during the approach and initial contact, and inflexible response during retracting and fast stage movements. Miniaturised and MEMS probes demonstrated refinements in sensitivity but only at the cost of ruggedness. Vibrating probe designs showed promise in overcoming this conflict but to date are unable to demonstrate adequate control and the mechanics of near-contact detection are undefined. As such there is still a need for a probe which overcomes the stiffness conflict while remaining simple to control, and understand.

The motivation for this development was expanded upon by reviewing literature of the current state of the art of micro-CMM technologies, and the following knowledge gaps were identified:

- 1) **There are no examples of variable stiffness probing systems**
- 2) **There is no easy way to minimise contact-force anisotropy of the probing system without being limited by stylus length**
- 3) **There are no established methods for operating an active suspension micro-probing system**

To address these three knowledge gaps six objectives were created;

Objective 1. Develop a realistic concept for applying variable stiffness

Objective 2. Investigate practical strategies that may be employed to produce a variable stiffness suspension.

Objective 3. Investigate the ability to achieve isotropic probing force without being limited by stylus length through the aid of analytical modelling and computer simulation.

Objective 4. Investigate manufacturing techniques that may be used to create prototype variable stiffness probing system, and employ these in the production of test probes.

Objective 5. Compare a numerical solution with experimental result in order to validate the simulation, and understand the limits of the manufacturing.

Objective 6. Develop a strategy that allows the control system switch between stiff and flexible stiffness.

Each of these six objectives have been achieved and their findings presented in the preceding chapters. From these activities the following outputs have been produced.

8.1 Key findings

An innovative, low force, micro-CMM probe with a novel variable stiffness mechanism has been presented, and validated with experimental data from a prototype probe.

In order to produce a variable stiffness suspension during use, two practical strategies were investigated. These concepts were based on either applying additional force to the suspension structure to counteract the structures natural restoring force, or by subjecting the suspension structure to an internal compressive strain along the suspension beams, reducing their flexural stiffness. It has been shown that it is possible to achieve a significant change in the stiffness of a suspension structure by applying an axial force to vary the internal strain energy of a beam, and has been presented, at least in principle, that the vertical and lateral stiffness can be reduced to zero at the buckling point. This response was described by analytical means, and cross validated by FEA, and can therefore be used to guide the process of suspension design. The two solutions were shown to have a good the agreement when estimating the stiffness and the displacement of the buckling point, with difference of only 9 %.

As long as the spring elements are made of long slender beams onto which a compressive load can be applied, then a wide range of geometries are possible. Of the two suspension structures presented, the geometric simplicity of the tri-beam design offers a lower stiffness and easier manufacturing. However, with the tri-beam structure there is a requirement to apply equal forces to each of the beams in order to

maintain probe position. The delta-beam design by comparison permits compressive forces to vary between beam elements. The delta-beam also demonstrates more closely matched vertical and lateral stiffness, thereby extending the isotropic stiffness regime. The two very different embodiments of the suspension structure highlight the flexibility of the concept.

An investigation into possible actuator technologies has demonstrated that using thermal expansion or thermally phase changing materials like shape memory alloys, are not appropriate because of their slow cycle times. Electrostatic actuation methods are unable to provide the necessary range of motion to modulate the stiffness, and while electromagnetic actuators could be successful they had a number of issues. If the coil element is placed on the suspension then heat dissipated in the coil can cause distortion and drift. If the permanent magnet is placed on the suspension, this will dramatically increase the effective mass. For either condition an electromagnetic solution is relatively large in size and therefore difficult to integrate with sensing elements of the probing system. Piezoelectric actuators have been shown to provide a fast response time, compact form, have linear behaviour and have less significant thermal effects. However there remain challenges to be overcome due to hysteresis and drift. To minimise the drift of the piezoelectric actuator during the experiment, initialising the actuator by cycling through a range of movement was found to reduce drift to 5 nm per minute.

To provide an example of variable stiffness probing systems, different manufacturing techniques for building a prototype of the tri-beam design were investigated and a series of probes produced. It has been shown that the micro machining produced too much distortion because of machining induced stresses, and that the laser ablated suspension failed due to thermal stress induced cracking. The suspension structure used in the prototype was manufactured using chemical etching, as this technique has shown a high process control and allows precision parts to be manufactured in batches, making it a cost effective technology. It has been found that chemical etching is able to remove material precisely, with no residual stress imparted to the structure and, as multiple suspension elements can be produced in a single etches, the process is less time consuming and more applicable to commercial probe production. Chemical etching demonstrated a significant improvement in device flatness where the beams deflection from ideal flat case is less than 16 μm over 4 mm length

compared to conventional manufacturing methods of around $350\text{ }\mu\text{m}$. It was necessary to assemble the suspension structure from a three layered structure in order to get the best performance out of the chemical etching process. Production quality was related to aspects of the suspension performance shown a good agreement between the measured initial frequencies, which the compressive load is equal to zero, and FEA within 6.3 %.

The main novelty of this probing system is that the stiffness of the suspension could be varied such that a suspension may be tuned to be very nearly isotropic for a range of styli lengths. An initial investigation was completed regarding varying the stylus length where FEA and analytical models were used. It was shown that when force is applied to reduce stiffness, there is a difference in the rate of change of stiffness for the lateral and vertical directions. This results in the two stiffness's converging and eventually becoming equal at an intersection point; making perfectly isotropic stiffness possible. The theoretical analysis demonstrated a possible application of the variable stiffness probe to operate in a fully isotropic stiffness regime for three different stylus lengths (3.8 mm, 6.6 mm and 8 mm).

To validate the concept of variable stiffness probing systems the ideal would be to directly obtain the stiffness by measuring a force and a displacement at the probe tip however this was not possible with the available equipment and an alternative method was devised. This method is based on analysis of the suspension structures' modal frequencies. Frequency information could be used to infer the stiffness of the suspension and could be used as a process variable in the closed loop control of the structures vertical and lateral stiffness. The piezoelectric actuators were used to apply a combined static compressive load and a dynamic load to excite resonance and determine the resonant frequency. The first investigation was conducted on a spring steel part where there is no stylus system integrated with suspension structure. This system was examined and shown that frequencies that are related in the vertical and torsional stiffness of the suspension structure can be reduced of approximately 70 % in vertical direction, while the torsional mode was decreased by approximately 33 %. It has been found that the minimum frequency reduction, and hence stiffness reduction is limited by a turning point in the applied voltage vs frequency response. A finite element model of exact geometry of the fabricated structure was shown a

give good agreement with the experimental results, although the dissimilarities observed may be explained by the non-symmetrical beam geometry that occurs as a result of the chemical etching process. As the suspension structure beams were required to be manufactured with single-sided blind etching only. It has been shown this prototype device remains relatively stiff, around 800 N m^{-1} , at the turning point. However, an optimized suspension structure that provides further reduction in stiffness was achieved through change to material using titanium alloy (grade 5) where the stiffness was reduced down to almost 440 N m^{-1} at the turning point. The stylus length of 6.6 mm was integrated with optimized suspension structure. It has been demonstrated that the lowest lateral stiffness at the turning point that can be achieved are around 340 N m^{-1} . The initial ratio between the vertical and lateral stiffness is reduced from 7 to 1.3, which is close to isotropic stiffness. The vertical stiffness is not affected by the stylus parameter and lateral stiffness is inversely proportional to square of the stylus length. Therefore, reducing the stylus length to be $l_s < 6.6 \text{ mm}$ can increase the possibility to achieve exact isotropic point as the intersection point should be greater than the turning point. For a FEA model of a perfectly symmetrical suspension structure, unlike the one sided etched prototype, it has been demonstrated that modal frequencies may be reduced down to zero.

To establish methods for operating a variable suspension micro-probing system, a novel and sophisticated controller was developed for NI LabVIEW that is very easy to use and has the ability to monitor and control the stiffness based on measuring the first vibrational mode frequency, i.e. the vertical frequency. The control system has shown the ability to switch between flexible and stiff modes, consistently, controlled within a range of 0.075 Hz. This leads to control the deflection of the probe tip within 2 nm in the vertical axis. This is unlike the control problems that are seen for vibrating probes, where there appears to be a fundamental challenge in achieving a dynamic control that permits for isotropic motion of oscillation.

The method used to complete a three-dimensional (3D) characterisation is based on well-known conventional contact where the only difference is that the probe tested in the flexible mode. Three probing system errors were evaluated based on the characterisation results. First, the linearity error was shown to be the most dominant error of the probing system, equal to 24 nm over a full working range $6 \mu\text{m}$. Second,

the probing system was shown a good performance where it is able to measure a nominal 4 μm displacement with a standard deviation of 5 nm. Thirdly, when switching from stiff to flexible mode the probe tip was shown to move by 12.78 μm , 4.36 μm and 12.51 μm in x , y and z direction respectively, due in part to imperfect suspension beam geometry. While this motion is not ideal it was demonstrated to be highly repeatable with a standard deviation of 12 nm. Combining stage errors with the probing system linearity error, the stylus tip zero offset position error and the probing system measurement repeatability, gives an estimate of a combined uncertainty for the probing system or of 29 nm. The expanded uncertainty is 58 nm (coverage factor, $k = 2$).

8.2 Future work

The research in this project has been presented and the concepts have been validated experimentally to meet the project aim. To open up new opportunities for future work, several further investigations have identified in this section.

Several issues have been demonstrated during the use of probing system need further investigations. Firstly, the probing system as it is currently designed takes around 20 s to reach steady state when switching between stiffness conditions. For operation as a working probe this may need to be improved to work in range of millisecond. Second, there is a drift resulting from the piezo-actuators used in the probe which is around of 35 nm within 2 minutes. The third issue is that the probe tip deviates in lateral direction resulting of uneven displacements from three actuators. The displacement at the stylus tip as a result of switching between the stiff to flexible mode is equal to -13.78 μm , 4.36 μm and 12.51 μm in x , y and z direction. To speed up the response time of actuator, minimise the drift and lateral displacement of the probe tip, it is suggested that piezo-actuators with integral strain gauges should be used. These are commonly used in high precision nano-positioning stages, and can effectively eliminate actuator drift, independently control the deflection of each beam, and because there is no need to wait for a frequency analysis to estimate beam stiffness they can dramatically improve switching times.

A possible extension of the probes operation would be to use the frequency monitoring as a method of near contact surface detection, as is done in other vibrating probe systems [89, 137]. This would enable the system to switch measuring

mode from the stiff mode to the flexible prior to an unplanned contact, and therefore avoid the high contact forces of stiff mode operation. The suspension structure is excited at its resonant frequency where the interaction between the probe tip and workpiece leads to change dynamic response and therefore the measurement surface can be triggered. This can be achieved by generating a chirp signal sweeping from the vertical to lateral frequency to make the probe tip vibrate in the vertical and lateral direction. To determine the position of the probe tip during use, the output voltage from three capacitive sensors can be implanted on the transformation matrix. At this stage it is not clear how this probe structure would compare to other proposed vibrating probes.

One of the limitations of the probing system is that the reduction of the frequency is limited to its lowest value at the turning point. The vertical and torsional frequency can be reduced by around 70 % and 33 %, respectively. A potential cause of this was demonstrated to be non-symmetric beam geometry as a result of the chemical etching process. It is recommended that in order to overcome this limitation, and current deleterious effects that manufacturing tolerance has on balanced response, that the probe concept should be adapted for manufacturing by the semi-conductor fabrication techniques common to MEMS devices. Multi-layer lithographic semi-conductor fabrication methods would allow for tighter production tolerances, integration of actuators and sensing elements, providing a more symmetric and repeatable suspension system. The closer that a production device can be brought to the idealised model, the more feasible a buckling, zero stiffness probe becomes, which would be a dramatic development for micro-CMM applications.

The need for a new kind of low force micro-CMM probe was identified, and from that need the concept of a variable stiffness probe was developed, through concept generation, modelling, embodiment, fabrication, validation and calibration, demonstrating a probe with unique properties and a good candidate for a future commercial solution. While there are challenges remaining to be overcome, this work has contributed to the field of micro-CMM and precision metrology from three perspectives. The first contribution is that the probing system is able to modulate the stiffness to overcome the compromise between a need for low contact force and a robust probe. The second contribution is that the probe may adjust the ratio of

vertical and lateral stiffness during operation, managing anisotropy by means other than stylus length. The third contribution is a control system for the probing system that is simple and effective, easily integrated with micro-CMM systems and able to perform self-calibration of stiffness positional control during the switch between modes.

References

- [1] Y. Zhao, T. Kramer, R. Brown, and X. Xu, *Information Modeling for Interoperable Dimensional Metrology*: Springer Publishing Company, Incorporated, 2014.
- [2] A. Weckenmann, P. Krämer, and G. Akkasoglu, "Metrology–base for scientific cognition and technical production," *AIP Conference Proceedings*, vol. 1431, pp. 283-292, 2012.
- [3] J. Cui, J. Li, K. Feng, and J. Tan, "Three-dimensional fiber probe based on orthogonal micro focal-length collimation for the measurement of micro parts," *Optics Express*, vol. 23, pp. 26386-26398, 2015/10/05 2015.
- [4] D. Gaoliang, P. Frank, X. Min, K. Ludger, D. Hans-Ulrich, and W. Günter, "Accurate and traceable measurement of nano- and microstructures," *Measurement Science and Technology*, vol. 17, p. 545, 2006.
- [5] H. N. Hansen, K. Carneiro, H. Haitjema, and L. De Chiffre, "Dimensional Micro and Nano Metrology," *CIRP Annals - Manufacturing Technology*, vol. 55, pp. 721-743, // 2006.
- [6] ISO 10360-2-Geometrical product specifications (GPS) — Acceptance and reverification tests for coordinate measuring machines (CMM) — Part 2: CMMs used for measuring linear dimensions, 2009.
- [7] S. H. R. Ali, "Advanced Nanomeasuring Techniques for Surface Characterization," *ISRN Optics*, vol. 2012, p. 23, 2012.
- [8] R. Danzl, F. Helml, and S. Scherer1, "Focus Variation - a New Technology for High Resolution Optical 3D Surface Metrology," presented at the The 10th International Conference of the Slovenian Society for Non-Destructive Testing, Application of Contemporary Non-Destructive Testing in Engineering, Ljubljana, Slovenia, 2009.
- [9] F. Li, A. P. Longstaff, S. A Fletcher, and A. A Myers, "Integrated Tactile and Optical Measuring Systems in Three Dimensional Metrology," presented at the Proceedings of The Queen's Diamond Jubilee Computing and Engineering Annual Researchers' Conference 2012: CEARC'12, 2012.
- [10] S.-M. Kao and D.-Y. Sheu, "Developing a novel tri-switch tactile probing structure and its measurement characteristics on micro-CMM," *Measurement*, vol. 46, pp. 3019-3025, 11// 2013.
- [11] S. Carmignato and A. Voltan, "An industrial comparison of coordinate measuring systems equipped with optical sensors: the VideoAUDIT Project," 2009, pp. 72390F-72390F-12.
- [12] Bruker.Optical Coordinate Measurement Machine (CMM).Contour CMM Dimensional Analysis System. [access date 30/01/2016]; Available from: <http://bruker-nano.actonsoftware.com/acton/attachment/9063/f-04c0/1/-/-/-/-/Contour%20CMM%20Dimensional%20Analysis%20System%20-%20DS583%20-%20RevA2.pdf>.
- [13] M. Wagener, F. Bader, and K. Seitz, "Multi-sensor CMM, miniaturized sensor system," Dutch Society for Precision Engineering (DSPE)2010.
- [14] Xpress Precision Engineering.TriNano N100 nano-CMM. [access date 28/01/2016]; Available from: <http://www.trinano.eu/n100.php>.
- [15] G. Lukacs, J. Lockhart, and M. Facello. Non-contact whole-part inspection. 2012 [access date 25/04/2016]; Available from:

- http://www.geomagic.com/files/6113/3037/1396/NonContactwhole-part-inspection_Final.pdf.
- [16] A. Weckenmann, T. Estler, G. Peggs, and D. McMurtry, "Probing Systems in Dimensional Metrology," *CIRP Annals - Manufacturing Technology*, vol. 53, pp. 657-684, 2004.
 - [17] P. K. Kinnell and R. R. Habeb, "An evaluation of cleaning methods for micro-CMM probes," *Measurement Science and Technology*, vol. 24, p. 085603, 2013.
 - [18] D. Wang, X. Chen, Y. Xu, F. Wang, M. Kong, J. Zhao, and B. Zhang, "High-NA fiber point-diffraction interferometer for three-dimensional coordinate measurement," *Optics Express*, vol. 22, pp. 25550-25559, 2014/10/20 2014.
 - [19] M. Petz, R. Tutsch, R. Christoph, M. Andraes, and B. Hopp, "Tactile–optical probes for three-dimensional microparts," *Measurement*, vol. 45, pp. 2288-2298, 2012.
 - [20] A. Rabani, N. Senin, F. Helml, P. Butler-Smith, and R. Leach, "An integrated measurement method for complex micro-scale geometries," in *euspen's 16th International Conference & Exhibition*, Nottingham, UK, 2016.
 - [21] H. Murakami, A. Katsuki, T. Sajima, and T. Suematsu, "Study of a vibrating fiber probing system for 3-D micro-structures: performance improvement," *Measurement Science and Technology*, vol. 25, p. 094010, 2014.
 - [22] T. Luhmann, S. Robson, S. Kyle, and J. Boehm, *Close-Range Photogrammetry and 3D Imaging*: De Gruyter, 2014.
 - [23] IBS Precision Engineering. Ultra-precision coordinate measuring machine. ISARA 400. 10/10/2015 [access date 20/04/2016]; Available from: http://www.ibspe.com/public/uploads/content/files/Isara400_Webversion.pdf.
 - [24] W. Berggold, "Leaving the Inspection Lab," *Optik & Photonik*, vol. 10, pp. 50-53, 2015.
 - [25] J. D. Claverley and R. K. Leach, "A review of the existing performance verification infrastructure for micro-CMMs," *Precision Engineering*, vol. 39, pp. 1-15, 1// 2015.
 - [26] K. Chen-Chun and J. S. Albert, "Form measurements of micro-holes," *Measurement Science and Technology*, vol. 18, p. 3603, 2007.
 - [27] *Metrology & Measurement*: McGraw-Hill Education (India) Pvt Limited, 2009.
 - [28] H. Schwenke, W. Knapp, H. Haitjema, A. Weckenmann, R. Schmitt, and F. Delbressine, "Geometric error measurement and compensation of machines—An update," *CIRP Annals - Manufacturing Technology*, vol. 57, pp. 660-675, // 2008.
 - [29] R. Ramesh, M. A. Mannan, and A. N. Poo, "Error compensation in machine tools — a review: Part I: geometric, cutting-force induced and fixture-dependent errors," *International Journal of Machine Tools and Manufacture*, vol. 40, pp. 1235-1256, 7// 2000.
 - [30] H. Schwenke, M. Franke, J. Hannaford, and H. Kunzmann, "Error mapping of CMMs and machine tools by a single tracking interferometer," *CIRP Annals - Manufacturing Technology*, vol. 54, pp. 475-478, // 2005.
 - [31] Coordinate Measuring Machine. Orthopaedic Innovation Centre. [access date 20/04/2016]; Available from: <http://www.orthoinno.com/precise-measurement/cmm/>.

- [32] K. C. Fan, F. Cheng, W. T. Pan, and R. Li, "Analysis of the contact probe mechanism for micro-coordinate measuring machines," *Optoelectronics, Instrumentation and Data Processing*, vol. 46, pp. 340-346, 2010/08/01 2010.
- [33] P. A. Cauchick-Miguel and T. G. King, "Factors which influence CMM touch trigger probe performance," *International Journal of Machine Tools and Manufacture*, vol. 38, pp. 363-374, 3// 1998.
- [34] M. Felix and K. Alain, "AFM investigation on surface damage caused by mechanical probing with small ruby spheres," *Measurement Science and Technology*, vol. 18, p. 496, 2007.
- [35] A. Weckenmann, G. Peggs, and J. Hoffmann, "Probing systems for dimensional micro- and nano-metrology," *Measurement Science and Technology*, vol. 17, p. 504, 2006.
- [36] J. Claverley and R. Leach, "Three-dimensional characterisation of a novel vibrating tactile probe for miniature CMMs," presented at the LAMBDAMAP, Chichley, UK, 2013.
- [37] E. J. C. Bos, "Aspects of tactile probing on the micro scale," *Precision Engineering*, vol. 35, pp. 228-240, 2011.
- [38] J. Claverley and R. Leach, "A vibrating micro-scale CMM probe for measuring high aspect ratio structures," *Microsystem Technologies*, vol. 16, pp. 1507-1512, 2010/08/01 2010.
- [39] D. Flack, J. Claverley, and R. Leach, "Chapter 9 - Coordinate Metrology," in *Fundamental Principles of Engineering Nanometrology (Second Edition)*, R. Leach, Ed., ed Oxford: William Andrew Publishing, 2014, pp. 295-325.
- [40] K. Takamasu, S. Ozawa, T. Asano, A. Suzuki, R. Furutani, and S. Ozono, "Basic Concepts of Nano-CMM (Coordinate Measuring Machine with Nanometer Resolution)," presented at the Proceedings of the Japan-China Bilateral Symposium on Advanced Manufacturing Engineering 1996.
- [41] K. Takamasu, M. Hiraki, K. Enami, and S. Ozono, "Development of Nano-CMM and Parallel-CMM," presented at the International Dimensional Metrology Workshop, Tennessee, USA 1999.
- [42] A. J. Lewis, "Fully traceable miniature CMM with submicrometer uncertainty," 2003, pp. 265-276.
- [43] L. Richard, H. Jane, J. Keith, L. Andrew, O. Simon, and Y. Andrew, "Advances in traceable nanometrology at the National Physical Laboratory," *Nanotechnology*, vol. 12, p. R1, 2001.
- [44] G. N. Peggs, A. J. Lewis, and S. Oldfield, "Design for a Compact High-Accuracy CMM," *CIRP Annals - Manufacturing Technology*, vol. 48, pp. 417-420, // 1999.
- [45] J. D. Claverley, "Development and validation of a 3D vibrating contact probe for micro-CMMs," PhD thesis, University of Nottingham, 2014.
- [46] P. Yang, T. Takamura, S. Takahashi, K. Takamasu, O. Sato, S. Osawa, and T. Takatsuji, "Development of high-precision micro-coordinate measuring machine: Multi-probe measurement system for measuring yaw and straightness motion error of XY linear stage," *Precision Engineering*, vol. 35, pp. 424-430, 7// 2011.
- [47] K. C. Fan, Y. T. Fei, X. F. Yu, Y. J. Chen, W. L. Wang, F. Chen, and Y. S. Liu, "Development of a low-cost micro-CMM for 3D micro/nano measurements," *Measurement Science and Technology*, vol. 17, p. 524, 2006.

- [48] I. Widdershoven, R. L. Donker, and H. A. M. Spaan, "Realization and calibration of the "Isara 400" ultra-precision CMM," *Journal of Physics: Conference Series*, vol. 311, p. 012002, 2011.
- [49] H. A. M. Spaan, R. L. Donker, and I. Widdershoven, "Isara 400: development of an ultra-precision CMM for 3D measurement for large parts," in *Proc.ASPE Spring Topical, Chicago, IL*, 2009.
- [50] J. D. Claverley, M. Baas, I. Widdershoven, R. K. Leach, and H. A. M. Spaan, "Installation of a vibrating contacting probe onto an ultra precision CMM," in *ASPE*, Boston, MA, USA, 2014.
- [51] R. Leach, "Chapter 3 - Precision Measurement Instrumentation – Some Design Principles," in *Fundamental Principles of Engineering Nanometrology (Second Edition)*, R. Leach, Ed., ed Oxford: William Andrew Publishing, 2014, pp. 41-61.
- [52] M. M. P. A. Vermeulen, P. C. J. N. Rosielle, and P. H. J. Schellekens, "Design of a High-Precision 3D-Coordinate Measuring Machine," *CIRP Annals - Manufacturing Technology*, vol. 47, pp. 447-450, // 1998.
- [53] Carl Zeiss Industrial Metrology. F25 Measuring Nanometers. [access date 20/04/2016]; Available from: http://moncktons.com/docs/zeiss_f25.pdf.
- [54] G. Jäger, T. Hausotte, E. Manske, H. J. Büchner, R. Mastlylo, N. Dorozhovets, and N. Hofmann, "Nanomeasuring and nanopositioning engineering," *Measurement*, vol. 43, pp. 1099-1105, 11// 2010.
- [55] I. Schmidt, T. Hausotte, U. Gerhardt, E. Manske, and G. Jäger, "Investigations and calculations into decreasing the uncertainty of a nanopositioning and nanomeasuring machine (NPM-Machine)," *Measurement Science and Technology*, vol. 18, p. 482, 2007.
- [56] G. Jager, E. Manske, T. Hausotte, and W. Schott, "Operation and analysis of a nanopositioning and nanomeasuring machine," in *Proceedings of the ASPE*, 2002, pp. 229-304.
- [57] SIOS GmbH NMM-1. Nanopositioning and Nanomeasuring Machine. 10/10/2015 [access date 20/04/2016]; Available from: http://www.piezosystem.com/fileadmin/datasheets/Interferometer-SIOS/NMM1-Nanopositioning-and-Nanomeasuring-Machine-SIOS_Datasheet.pdf.
- [58] B. Edwin, M. Ton, and R. Martijn van, "Design and verification of an ultra-precision 3D-coordinate measuring machine with parallel drives," *Measurement Science and Technology*, vol. 26, p. 085904, 2015.
- [59] E. J. C. Bos, A. J. M. Moers, and M. C. J. M. Van Riel, "Trinano N100 3D measurements with nanometer repeatability and effects of probe-surface interaction," in *Proceedings - ASPE 2012 Annual Meeting*, 2012, pp. 85-88.
- [60] Meli F, Fracheboud M, Bottinelli S, Bieri M, Thalmann R, B. J. M, and C. R, "High precision, low force 3D touch probe for measurements on small objects," in *EUSPEN*, Germany, 2003.
- [61] A. Küng, A. Nicolet, and F. Meli, "Study of wear of diamond-coated probe tips when scanning on different materials," *Measurement Science and Technology*, vol. 26, p. 084005, 2015.
- [62] A. Küng and F. Meli, "Self calibration method for 3D roundness of spheres using an ultraprecision coordinate measuring machine. ," in *EUSPEN* Montpellier, France, 2005.

- [63] F. G. Balzer, T. Hausotte, N. Dorozhovets, E. Manske, and G. Jäger, "Tactile 3D microprobe system with exchangeable styli," *Measurement Science and Technology*, vol. 22, p. 094018, 2011.
- [64] Mecartex. 3D Nanotouch Probe Ultraprecision Coordinate Measuring Machines for Small Parts. [access date 20/04/2016; Available from: <http://www.mecartex.com/products/MeasuringSystems.cfm>.
- [65] IBS Precision Engineering . Triskelion ultra-precision touch probe. [access date 20/04/2016]; Available from: <http://www.ibspe.com/public/uploads/content/files/Triskelion-2012-Web.pdf>.
- [66] K.-C. Fan, F. Cheng, W. Wang, Y. Chen, and J.-Y. Lin, "A scanning contact probe for a micro-coordinate measuring machine (CMM)," *Measurement Science and Technology*, vol. 21, p. 054002, 2010.
- [67] L. Rui-Jun, F. Kuang-Chao, M. Jin-Wei, H. Qiang-Xian, T. Sheng, and G. Er-min, "An analogue contact probe using a compact 3D optical sensor for micro/nano coordinate measuring machines," *Measurement Science and Technology*, vol. 25, p. 094008, 2014.
- [68] R.-J. Li, K.-C. Fan, Q.-X. Huang, H. Zhou, E.-M. Gong, and M. Xiang, "A long-stroke 3D contact scanning probe for micro/nano coordinate measuring machine," *Precision Engineering*, vol. 43, pp. 220–229, 2016.
- [69] H. Haitjema, W. Pril, and P. H. J. Schellekens, "A silicon-etched probe for 3-D coordinate measurements with an uncertainty below 0.1 μm ," *Instrumentation and Measurement, IEEE Transactions on*, vol. 50, pp. 1519-1523, 2001.
- [70] S. Bütefisch, S. Büttgenbach, T. Kleine-Besten, and U. Brand, "Micromechanical three-axial tactile force sensor for micromaterial characterisation," *Microsystem Technologies*, vol. 7, pp. 171-174, 2001/11/01 2001.
- [71] U. Brand, T. Kleine-Besten, and H. Schwenke, "Development of a special CMM for dimensional metrology on microsystem components," in *ASPE*, 2000, pp. 361–364.
- [72] N. Vladimir and B. Uwe, "Modelling and investigation of the silicon twin design 3D micro probe," *Journal of Micromechanics and Microengineering*, vol. 15, p. 514, 2005.
- [73] N. Vladimir and B. Uwe, "Modelling and investigation of the mechanical and electrical characteristics of the silicon 3D-boss microprobe for force and deflection measurements," *Journal of Micromechanics and Microengineering*, vol. 16, p. 1116, 2006.
- [74] W. O. Pril, "Development of High Precision Mechanical Probes for Coordinate Measuring Machines," PhD thesis, Technische Universiteit Eindhoven, 2002.
- [75] H. Haitjema, W. O. Pril, and P. H. J. Schellekens, "Development of a Silicon-based Nanoprobe System for 3-D Measurements," *CIRP Annals - Manufacturing Technology*, vol. 50, pp. 365-368, // 2001.
- [76] E. J. C. Bos, "Tactile 3D probing system for measuring MEMS with nanometer uncertainty," PhD Theses, Department of Mechanical Engineering, Eindhoven University of Technology, 2008.
- [77] Xpress Precision Engineering. 3D tactile scanning probes with nanometer uncertainty. [access date 20/04/2016]; Available from: <http://www.xpresspe.com/download/Xpress-Gannen-series.pdf>.

- [78] H. Schwenke, F. Wäldele, C. Weiskirch, and H. Kunzmann, "Opto-tactile Sensor for 2D and 3D Measurement of Small Structures on Coordinate Measuring Machines," *CIRP Annals - Manufacturing Technology*, vol. 50, pp. 361-364, 2001.
- [79] B. Uwe and K. Juergen, "A micro-CMM with metrology frame for low uncertainty measurements," *Measurement Science and Technology*, vol. 16, p. 2489, 2005.
- [80] Werth Messtechnik GmbH. Werth Fiber Probe WFP. [access date 20/04/2016]; Available from: <http://www.werth.de/en/unser-angebot/products/sensors/werth-fiber-probe-wfp.html>.
- [81] R. Tutsch, M. Andräs, U. Neuschaefer-Rube, M. Petz, T. Wiedenhöfer, and M. Wissmann, "Three dimensional tactile-optical probing for the measurement of microparts," in *SENSOR+TEST*, 2009, pp. 139 - 144.
- [82] B. Muralikrishnan, J. A. Stone, S. C. Vemuri, C. Sahay, A. Potluri, and J. R. Stoup, "Fiber deflection probe for small hole measurements," in *In Proceedings of the 2004 ASPE Annual Meeting*, USA, 2004, pp. 24-27.
- [83] B. Muralikrishnan, J. A. Stone, and J. R. Stoup, "Fiber deflection probe for small hole metrology," *Precision Engineering*, vol. 30, pp. 154-164, 2006.
- [84] B. Muralikrishnan, J. A. Stone, and J. R. Stoup, "Enhanced Capabilities of the NIST Fiber Probe for Microfeature Metrology," in *In Proceedings of the 2006 Annual Meeting of the ASPE*, USA, 2006, pp. 3-6.
- [85] J. Tan, F. Wang, and J. Cui, "Fiber deflection probing method based on micro focal-length collimation," *Optics Express*, vol. 18, pp. 2925-2933, 2010/02/01 2010.
- [86] M. C. J. M. v. Riel, E. J. C. Bos, and F. G. A. Homburg, "Analysis of the measurement sensitivity of multidimensional vibrating microprobes," *Measurement Science and Technology*, vol. 25, p. 075008, 2014.
- [87] G. Boris, D. Lothar, and H. Martin, "Design and characterization of a resonant triaxial microprobe," *Journal of Micromechanics and Microengineering*, vol. 25, p. 125011, 2015.
- [88] Mitutoyo. UMAP Vision System. 2002 [access date 20/04/2016]; Available from: http://www.mitutoyo.co.jp/eng/new/news/2002/02_13.html.
- [89] K. Hidaka and P. H. J. Schellekens, "Study of a Small-sized Ultrasonic Probe," *CIRP Annals - Manufacturing Technology*, vol. 55, pp. 567-570, 2006.
- [90] K. Hidaka, A. Saito, and S. Koga, "Study of a micro-roughness probe with ultrasonic sensor," *CIRP Annals - Manufacturing Technology*, vol. 57, pp. 489-492, 2008.
- [91] S. Stoyanov, C. Bailey, R. Leach, B. Hughes, A. Wilson, W. O'Neill, R. A. Dorey, C. Shaw, D. Underhill, and H. J. Almond, "Modelling and prototyping the conceptual design of 3D CMM micro-probe," in *Electronics System-Integration Technology Conference, 2008. ESTC 2008. 2nd*, 2008, pp. 193-198.
- [92] E. J. C. Bos, R. W. P. Heldens, F. L. M. Delbressine, P. H. J. Schellekens, and A. Dietzel, "Compensation of the anisotropic behavior of single crystalline silicon in a 3D tactile sensor," *Sensors and Actuators A: Physical*, vol. 134, pp. 374-381, 2007.
- [93] E. Rustighi, M. J. Brennan, and B. R. Mace, "A shape memory alloy adaptive tuned vibration absorber: design and implementation," *Smart Materials and Structures*, vol. 14, p. 19, 2005.

- [94] M. R. F. Kidner and M. J. Brennan, "Varying the Stiffness of a Beam-Like Neutralizer Under Fuzzy Logic Control," *Journal of Vibration and Acoustics*, vol. 124, pp. 90-99, 2001.
- [95] P. Bonello, M. J. Brennan, S. J. Elliott, J. F. V. Vincent, and G. Jeronimidis, *Designs for an adaptive tuned vibration absorber with variable shape stiffness element* vol. 461, 2005.
- [96] K. B. Lee, A. P. Pisano, and L. Lin, "Nonlinear behaviors of a comb drive actuator under electrically induced tensile and compressive stresses," *Journal of Micromechanics and Microengineering*, vol. 17, p. 557, 2007.
- [97] E. S. Leland and P. K. Wright, "Resonance tuning of piezoelectric vibration energy scavenging generators using compressive axial preload," *Smart Materials and Structures*, vol. 15, p. 1413, 2006.
- [98] W. Al-Ashtari, M. Hunstig, T. Hemsel, and W. Sextro, "Frequency tuning of piezoelectric energy harvesters by magnetic force," *Smart Materials and Structures*, vol. 21, p. 035019, 2012.
- [99] V. R. Challa, M. G. Prasad, Y. Shi, and F. T. Fisher, "A vibration energy harvesting device with bidirectional resonance frequency tunability," *Smart Materials and Structures*, vol. 17, p. 015035, 2008.
- [100] H. Torun, K. K. Sarangapani, and F. L. Degertekin, "Spring constant tuning of active atomic force microscope probes using electrostatic spring softening effect," *Applied Physics Letters*, vol. 91, p. 253113, 2007.
- [101] PTB.Force measurements in the nano- and sub-nanonewton range. [access date 13/05/2016]; Available from: <https://www.ptb.de/cms/en/ptb/fachabteilungen/abt5/fb-51/ag-512/about-nano-force-standard-device.html>.
- [102] V. Nesterov, M. Mueller, L. L. Frumin, and U. Brand, "A new facility to realize a nanonewton force standard based on electrostatic methods," *Metrologia*, vol. 46, p. 277, 2009.
- [103] N. Vladimir, "Facility and methods for the measurement of micro and nano forces in the range below 10^{-5} N with a resolution of 10^{-12} N (development concept)," *Measurement Science and Technology*, vol. 18, p. 360, 2007.
- [104] FemtoTools. FT-S1000 Microforce Sensing Probe. [access date 20/04/2016]; Available from: <http://www.femtotools.com/products/accessories/ft-s-microforce-sensing-probe/>.
- [105] M. Elahinia, *Shape Memory Alloy Actuators: Design, Fabrication and Experimental Evaluation*: Wiley, 2015.
- [106] K. B. LEE, *Principles of Microelectromechanical Systems*: John Wiley and Sons, 2011.
- [107] R. Boyer, G. Welsch, E. W. Collings, and M. American Society for, *Materials properties handbook : titanium alloys*. Metals Park, Ohio: ASM International, 1994.
- [108] D. J. Leo, "Piezoelectric Materials," in *Engineering Analysis of Smart Material Systems*, ed: John Wiley & Sons, Inc., 2008, pp. 122-204.
- [109] Physic Instrument. Miniature Multilayer Piezo Actuators. PL022.30. [access date 20/04/2016]; Available from: <http://piceramic.com/product-detail-page/pl0xx-100800.html>.
- [110] J. J. Yao and N. C. MacDonald, "A micromachined, single-crystal silicon, tunable resonator," *Journal of Micromechanics and Microengineering*, vol. 5, p. 257, 1995.

- [111] D. de Bhailís, C. Murray, M. Duffy, J. Alderman, G. Kelly, and S. C. Ó Mathúna, "Modelling and analysis of a magnetic microactuator," *Sensors and Actuators A: Physical*, vol. 81, pp. 285-289, 2000.
- [112] S. Eroglu, B. Gimi, B. Roman, G. Friedman, and R. L. Magin, "NMR spiral surface microcoils: Design, fabrication, and imaging," *Concepts in Magnetic Resonance Part B: Magnetic Resonance Engineering*, vol. 17B, pp. 1-10, 2003.
- [113] J. W. Judy, "Batch-Fabricated Ferromagnetic Microactuators with Silicon Flexures," Doctor of Philosophy, Engineering-Electrical Engineering and Computer Sciences, UNIVERSITY OF CALIFORNIA, BERKELEY, 1996.
- [114] G. M. Rebeiz and J. Rizk, "Micromachined Inductors," in *RF MEMS*, ed: John Wiley & Sons, Inc., 2004, pp. 359-382.
- [115] S. C. Gottschalk, D. C. Quimby, K. E. Robinson, E. R. Moog, and R. T. Apparao, "Central field design methods for hybrid insertion devices," *Review of Scientific Instruments*, vol. 67, pp. 3347-3347, 1996.
- [116] K. B. Lee, "Statics," in *Principles of Microelectromechanical Systems*, ed: John Wiley & Sons, Inc., 2011, pp. 64-143.
- [117] N. S. TRAHAI, "Wagner's beam cycle," School of Civil Engineering The University of Sydney Sydney NSW 2006 Australia 2011.
- [118] Taylor Series Calculator. Wolfram Alpha. 2015 [access date; Available from: <http://www.wolframalpha.com/widgets/view.jsp?id=f9476968629e1163bd4a3ba839d60925>.
- [119] R. Boyer, G. Welsch, and E. W. Collings, "Materials Properties Handbook - Titanium Alloys," ed: ASM International, 1994.
- [120] COMSOL, "Multiphysics Reference Guide for COMSOL 4.3a," ed, 2013.
- [121] T. Liebrich and W. Knapp, "Improvements and experimental validation of a 3D-probing system for micro-components," *CIRP Annals - Manufacturing Technology*, vol. 61, pp. 475-478, // 2012.
- [122] S. S. Rad, *Mechanical Vibrations- Fourth Edition* United States of America: Pearson Prentice Hall, 2003.
- [123] Bruker. ContourGT-I. Optical Profiler. User Manual. [access date 17/05/2016]; Available from: <https://www.bruker.com/>.
- [124] W. Ding. Standard operating procedure. Bruker contourgt-k optical profiler. [access date 17/05/2016]; Available from: <http://sharedresearchfacilities.wvu.edu/r/download/178433>.
- [125] PrecisionMicro. 2014 [access date 20/09/2014; Available from: www.precisionmicro.com.
- [126] R. Patel, "Modelling analysis and optimisation of cantilever piezoelectric energy harvesters," PhD, Department of Mechanical, Materials and Manufacturing Engineering, University of Nottingham only, 2013.
- [127] S. Dhar, P. M. Dixit, and R. Sethuraman, "A continuum damage mechanics model for ductile fracture," *International Journal of Pressure Vessels and Piping*, vol. 77, pp. 335-344, 2000.
- [128] M. He, R. Liu, Y. Li, H. Wang, X. Lu, G. Ding, J. Wu, T. Zhang, and X. Zhao, "Tactile probing system based on micro-fabricated capacitive sensor," *Sensors and Actuators A: Physical*, vol. 194, pp. 128-134, 2013.
- [129] J. A. Main, E. Garcia, and D. V. Newton, "Precision position control of piezoelectric actuators using charge feedback," *Journal of Guidance, Control, and Dynamics*, vol. 18, pp. 1068-1073, 1995/09/01 1995.

- [130] M. Goldfarb and N. Celanovic, "Behavioral implications of piezoelectric stack actuators for control of micromanipulation," in *Robotics and Automation, 1996. Proceedings., 1996 IEEE International Conference on*, 1996, pp. 226-231 vol.1.
- [131] Physik Instrumente, P-611.Z Piezo Z Stage. [access date 20/04/2016]; Available from:
http://www.physikinstrumente.com/download/PI_Datasheet_P-611_Z_20150122.pdf.
- [132] I. P. Lipscomb, P. M. Weaver, J. Swingler, and J. W. McBride, "The effect of relative humidity, temperature and electrical field on leakage currents in piezo-ceramic actuators under dc bias," *Sensors and Actuators A: Physical*, vol. 151, pp. 179-186, 4/29/ 2009.
- [133] N. Lazarus, S. S. Bedair, C. C. Lo, and G. K. Fedder, "CMOS-MEMS Capacitive Humidity Sensor," *Microelectromechanical Systems, Journal of*, vol. 19, pp. 183-191, 2010.
- [134] P. Jae-jun, K. Kihwan, and C. Nahmgyoo, "Development of a coordinate measuring machine (CMM) touch probe using a multi-axis force sensor," *Measurement Science and Technology*, vol. 17, p. 2380, 2006.
- [135] E. J. C. Bos, "Tactile 3D probing system for measuring MEMS with nanometer uncertainty.," Phd thesis, Eindhoven University of Technology, Netherlands, 2008.
- [136] N. Henmi, "An Open-Loop Positioning Method for Reduction of Influence of Nonlinearities in Piezoelectric Actuator," *Journal of Advanced Mechanical Design, Systems, and Manufacturing*, vol. 5, pp. 70-79, 2011.
- [137] J. D. Claverley and R. K. Leach, "Development of a three-dimensional vibrating tactile probe for miniature CMMs," *Precision Engineering*, vol. 37, pp. 491-499, 2013.