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**Displacement and Blocking
Force Measurements of
Piezoelectric Macroactuators**

**M J Lodeiro, M Stewart and
M G Cain**

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M J Lodeiro, M Stewart and M G Cain
Division of Engineering and Process Control

ABSTRACT

A novel experimental system has been designed to characterise the static blocking force of piezoelectric macroactuators. These measurements are of particular interest to manufacturers and end-users alike for comparison of devices for specific applications. This was successfully used to map the full force-deflection response of new commercially available macroactuator designs: a co-fired multilayer actuator, a stress-biased domed unimorph and a planar bimorph.

The system design, instrumentation, software automation, calibration and compliance are discussed in detail, highlighting important aspects of the measurement procedure and precautions/pitfalls.

The results obtained showed some non-linear actuator behaviour in the load-displacement traces, as well as providing invaluable insights into hysteresis and time dependent properties of the actuation load or displacement produced with driving voltage.

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National Physical Laboratory
Hampton Road, Teddington, Middlesex, TW11 0LW

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Approved on behalf of the Managing Director, NPL,
by Dr M G Gee, Knowledge Leader, Materials Performance Team
authorised by Director, Engineering and Process Control Division

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1 INTRODUCTION

Piezoelectric actuators are fast replacing more traditional electromechanical devices due to their small size, high strain, low power consumption, linear response and fine control. A small selection from the constantly expanding list of applications includes:

- optical component positioning;
- pneumatic and hydraulic valve control – flow management or pumping e.g. fuel injection;
- oscillation control or vibration damping;
- movement in small robotic systems.

The relationship between displacement and actuation force is an important characteristic of any piezoelectric actuator. These devices convert electrical energy into force or motion or, under the varying load conditions experienced in many applications, the energy is split between generating displacement and force simultaneously when coupled to an external mechanical system. The proportion depends on the relative stiffnesses of the actuator and the resisting mechanics, with the energy conversion being a maximum where these are exactly matched.

In practice, the actual achievable stroke/load depends on mounting conditions, loading area and morphology, temperature, ageing, field-related material non-linearity etc. Often manufacturers' specifications and performance data are defined in different ways, making it impossible to compare directly between actuators from several alternative suppliers. In addition, it is difficult to predict performance reliably, so that measurements made under anticipated operating conditions are usually recommended to assist in the choice between several actuators for a particular application. As a result, not only is design data scarce, but without any standard measurements its usefulness is severely limited.

This report presents the conception, design and construction of instrumentation to map the force-displacement characteristics of a wide range of macroactuators, evaluating the complete response from fully blocked to fully free displacement.

The scope includes a broad discussion of macroactuator architectures; terminology definitions and analytical predictions of key performance parameters. Relevant studies on bulk and MEMS devices and alternative measurement systems for blocking force determination are summarised to place the current study within a relevant framework. More detailed discussion relating to the operating principles of the new NPL design, the instrumentation, software automation, calibration and compliance corrections is provided. Data from several actuators subsequently used to test the system are presented, highlighting necessary precautions, remaining problems and further work required to improve the final design. Some topics of interest for future investigation are also offered.

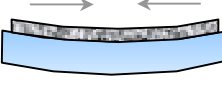
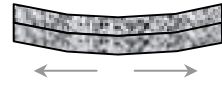
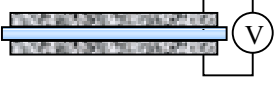
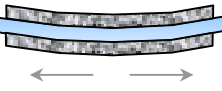
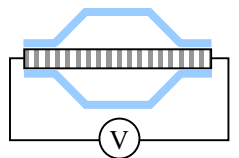
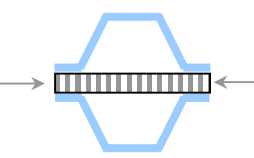
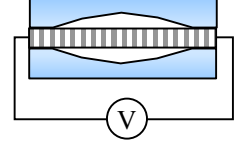
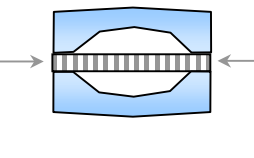
1.1 PIEZOELECTRIC MACROACTUATORS

Piezoelectric macroactuators are relatively large-scale actuators with novel architectures, usually incorporating either external or internal displacement amplification mechanisms since the direct extensional strain in piezoelectric ceramics is small, of the order of a few microns. They typically take the form of compliant benders, transforming small planar or linear movements into much higher bending travel, this amplification being at the cost of a much reduced force. For these, the flexural displacement under applied voltage is the result of the piezoelectric effect in the perpendicular direction to the poling axis, related to the piezoelectric constant d_{31} . They include actuators with dimensions in the range 5 to 40 mm with

displacements of up to about 2 mm and pushing forces of up to 100 N, i.e. having generally high actuation but low pushing force.

Typical examples of macroactuators [1] include the devices shown in Table 1. Other more complex/innovative actuator constructions are described briefly in Table 2.

Table 1 – Architecture and Function of Most Common Macroactuators.

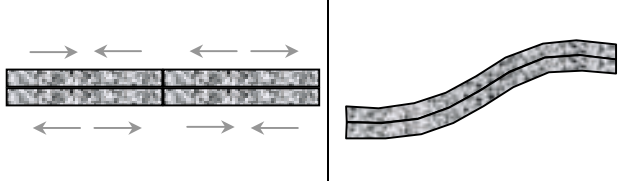
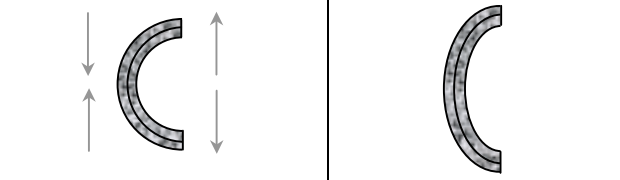
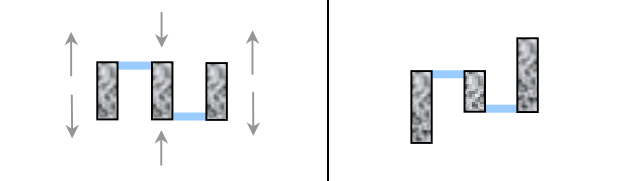
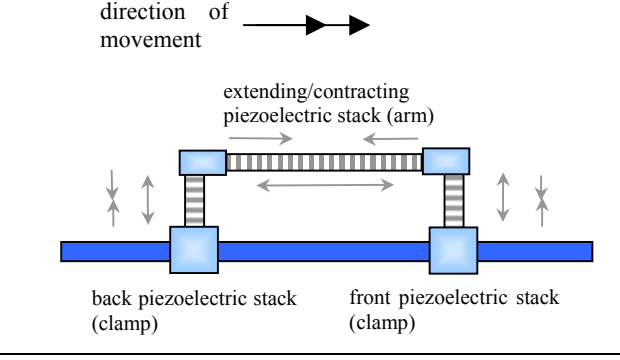
Generic title	Construction	Actuation	Comments
Unimorph			Single layer of active piezo. Passive substrate creates asymmetrical contraction of the structure.
Bimorph			Double layer of active piezo. Opposing layer motion (either through poling or applied field) produces and reinforces bending.
Trimorph			Combination of unimorph and bimorph designs.
Cymbal			Metal caps, connected at the ends to create hollow cavities, amplify displacement as stack contracts.
Moonie			Similar operating principle as cymbal.
Key:  - represents active piezoceramic layers  - represents an active piezoceramic multi-layer stack  - represents inactive material substrates, layers or components, usually metallic  - represents the direction of the dimensional changes in the piezoelectric material			

In addition to these increasingly commonplace actuator designs, newer stress-biased actuators are commercially available which are based on the functioning principle of unimorphs but are domed or saddle-shaped. They exhibit both fairly large displacements and blocking forces at reasonable driving fields and include:

- CRESCENT,
- THUNDER (THin UNimorph DrivER),
- RAINBOW (Reduced And INternally Biased Oxide Wafer) and
- CERAMBOW (CERAMic Biased Oxide Wafer).

RAINBOWs are piezoelectric wafers subjected to heat-induced chemical reduction on one side to effectively create an integral inactive substrate (d_{31} functionally graded material) with the commensurate benefits associated with monolithic structures, whilst CRESCENT, THUNDER and CERAMBOW use high temperature and/or pressure conditions to adhesively bond inactive substrates to the piezoelectric layer. For all these actuators, the difference in thermal contraction between the active and passive layers and the modulus mismatch between them causes the characteristic distortion curvature which occurs on cooling to room temperature from the fabrication temperature, locking the stress differential into place. This produces a more rugged and robust device as the piezoceramic is held in a favourable stress state throughout its displacement/force range.

Table 2 – Architecture and Function of More Complex Macroactuators.

Generic title	Construction/Actuation	Comments
Recurve		Operation similar to bimorph. Polarisation and thus bending moment inverted halfway along the length. Segments can be integrated into larger actuation structures.
C-block		Operation as bimorph but pre-fabricated in curved shape.
Telescopic		An array of simply interconnected elements combine direct displacement for increased linear motion.
Inchworm		Rapidly repeated small actuation steps move actuator in a “shuffling” motion along a rod. A possible sequence entails: • back end clamps • front end unclamps • arm extends - pushes front clamp forward, • front end clamps, • back end unclamps • arm contracts - pulls back clamp forward.
Key:  - represents active piezoceramic layers  - represents an active piezoceramic multi-layer stack  - represents inactive material substrates, layers or components, usually metallic  - represents the direction of the dimensional changes in the piezoelectric material		

1.2 BLOCKING FORCE

The *blocking force* is a term used to define the maximum force which can be applied by an actuator when generating zero displacement at full driving voltage or, sometimes, the force required to return an actuator from its maximum amplitude displacement to the zero position. Conversely, the *free displacement* is the maximum displacement which can be produced by an actuator when acting against a zero load. This information is commonly presented as shown in Figure 1, i.e. assuming a linear response between the two measured datapoints [1, 2].

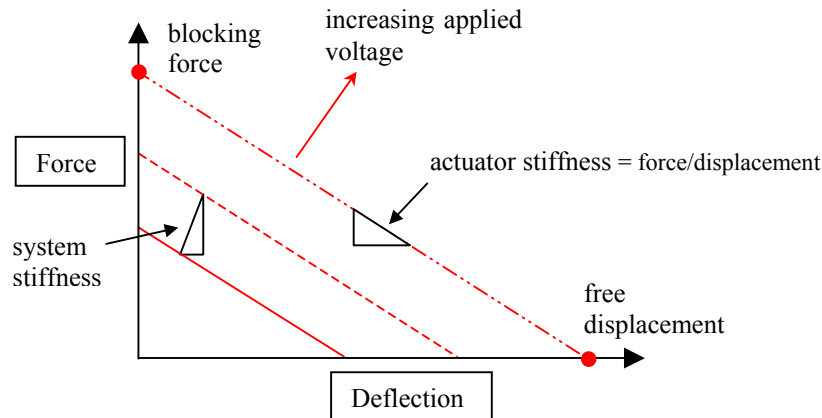


Figure 1 – Typical actuator blocking force diagram showing measured endpoints, key stiffnesses and lines of constant drive voltage.

However, it is the real performance between these two extremes which is important for most practical applications. The objective then is to map the true and full actuation response under realistic conditions - not just the free and fully blocked datum points with the assumption of linearity between them.

In order to achieve this, there are a number of possibilities available which should in principle generate equivalent data, shown in Figure 2. Measurements can be taken for several applied voltages along lines of constant resisting stiffness, or along lines of constant actuator force or constant actuator deflection. Therefore by measuring both actuator force and deflection over a range of stiffnesses, or by measuring the actuator deflection generated for several resisting forces (up to a maximum given by the blocking force) or by measuring the actuator force generated for several restricted displacements (up to a maximum given by the free displacement), the complete actuator response can be determined.

The procedures to apply for the different routes are:

- Constant stiffness – as marked in green in Figure 2. The actuator is placed in contact with a known system stiffness, the applied voltage is increased incrementally and both load and deflection are measured at each increment. This process is repeated for different system stiffnesses.
- Constant displacement – as marked in light blue in Figure 2. A given voltage is applied to the actuator to produce a known free displacement. A nominally infinite stiffness load cell and loading system is placed in contact with the actuator and displaced incrementally to return the actuator to zero deflection with the load being measured at

each stage. Alternatively, the infinite stiffness load cell and loading system can be placed in contact with the actuator at zero deflection, after which a given voltage is applied and the loading system incrementally backed away to maximum displacement with the load being measured at each stage. These are repeated for different initial voltages, in both cases.

- Constant force – as marked in dark blue in Figure 2. A known infinitely compliant force or a mass is applied to the actuator. The applied voltage is then incrementally increased and the deflection is measured at each stage. This process is repeated for different loads.

There is as yet no direct experimental proof that all these routes produce equivalent data or constitute equivalent measurement conditions.

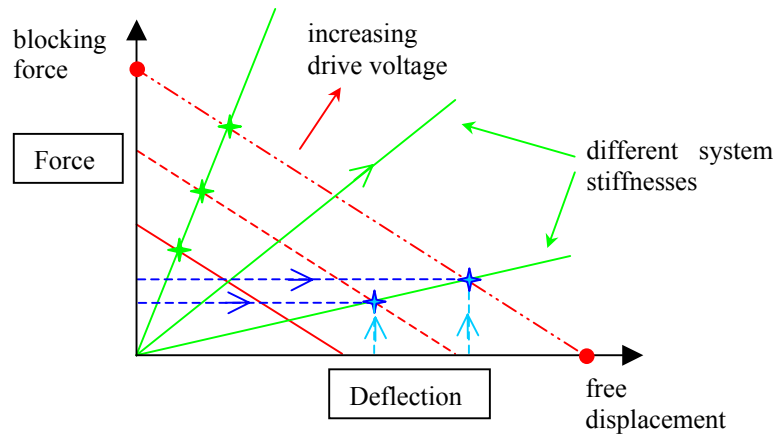


Figure 2 – Actuator blocking force diagram showing alternative routes for data measurement: constant stiffness (marked in green), constant force (marked in dark blue) and constant displacement (marked in light blue).

1.2.1 Analytical Predictions of Blocking Force and Free Deflection

Simple analyses are available for the most basic 2 or 3-layer bending actuators [3], i.e. unimorphs and bimorphs, to enable the characteristic properties of the actuator to be predicted from the constituent material properties and dimensions, derived by considering both the general constitutive piezoelectric equations and the system mechanics. The expressions for free deflection, δ , and blocking force, F , are presented in Table 3. The expressions are generally based on the hypotheses of classical beam theory and linear elastic response.

Table 3 – Expressions for characteristic unimorph/bimorph actuator performance parameters.

Actuator type	Free deflection	Blocking force
Unimorph	$\delta = \frac{3L^2}{2t_p} d_{31} V_3 \frac{2AB(1+B)^2}{A^2B^4 + 2A(2B + 3B^2 + 2B^3) + 1}$	$F = \frac{3wt^2 E_p}{8L} d_{31} V_3 \frac{2AB}{(AB+1)(1+B)}$
Bimorph	$\delta = \frac{3L^2}{2t} d_{31} V_3$	$F = \frac{3wt^2 E_p}{8L} d_{31} V_3 \frac{2B+1}{(B+1)^2}$
Bimorph with internal metal shim (Trimorph)	$\delta = \frac{3L^2}{2t} d_{31} V_3 \frac{(1+B)(2B+1)}{AB^3 + 3B^2 + 3B + 1}$	$F = \frac{3wt^2 E_p}{8L} d_{31} V_3$
where t_p =total piezoelectric thickness, t_m =total metal thickness, t =overall actuator thickness, E_m =metal modulus, E_p =piezoelectric modulus, d_{31} = piezoelectric coefficient, V_3 =applied field, $A=E_m/E_p$ and $B=t_m/t_p$		

These expressions consider electric field induced bending only. For stress-biased actuators in particular, the non-uniform internal mechanical stresses superimposed across the thickness and their natural domed-shapes result in a much higher loading capability than might be expected from their simple unimorph/bimorph structures. This is believed to be due to enhanced piezoelectric extrinsic contributions, produced as a result of combined field (ferroelectric) and stress (ferroelastic) effects, which alter domain configurations and improve switching (particularly of 90° domains present at the actuator tensile upper surface giving an increased strain response) [4]. The polarisation characteristics are also affected by the presence of a pre-stress during poling, increasing d_{31} via the formation of specific domain structures [5].

Consideration of the non-linearity of the piezoelectric coefficient, d_{31} ; elastic compliance, s_{11}^E , and dielectric constant, ϵ_{33}^T , for high electric fields and mechanical stresses would also need to be added for completeness. This non-linearity is believed to be principally due to the extrinsic piezoelectric response caused by non-180° domain wall motion under these conditions. Previous work has shown that [6]:

- d_{31} decreases with increasing uniaxial compressive stress (once poled),
- d_{31} increases with increasing electric field,
- s_{11}^E increases and so E_p decreases with increasing electric field.

These effects are more pronounced in soft PZT ceramics, which are commonly used in macroactuators because of their greater intrinsic displacement capabilities. Additionally, for these materials it has been found that the blocking force normalised with respect to the field applied is proportional to the ratio of d_{31} to s_{11}^E and moreover that the expected increase in s_{11}^E is lower than that of d_{31} . This would result in a proportionately larger increase in the blocking force with applied field rather than the increase in compliance which might otherwise be observed [7].

Modified analytical equations [6] have been developed to account for the pre-curvature and temperature sensitivity, in RAINBOW actuators in particular, but inclusion of non-linearity associated with the electric field and internal stress have yet to be implemented.

1.2.2 Bulk Actuator and MEMS Blocking Force Measurements

Work carried out previously at NPL [2] used an in-house designed, instrumented prototype apparatus to measure the blocking force of bulk piezoelectric actuators. The principle conclusion from this study was that a substantial departure from linearity could be expected from any measured blocking force curve. It was noted that actuator stiffness is dependent on both applied voltage and stress levels. These effects combined to produce the non-linearity observed in equipotential lines where blocking forces at each level of applied field were well below linearly assumed values.

It was also noted that static pre-loads (below that required to depolarise the material) applied prior to the driving voltage do not change the displacement obtained but simply offset the origin of the displacement. Conversely, spring forces with constantly varying loads (load increasing linearly with displacement) applied with the driving voltage lead to a reduction in the total displacement.

A system using the constant stiffness method was devised to measure the blocking force/free displacement characteristics of PZT micro-cantilevers. This study also demonstrated a non-linear force/displacement response and similarly showed an increase in actuator compliance with increasing field and stress [8].

1.2.3 Macroactuator Blocking Force Measurements

Methods used previously to determine the end points of the actuator load/displacement curve, i.e. free displacement and blocking force, have often been simple and crude. These include dial or digital indicators to measure free displacement and weights hung from the actuator to determine the blocking force as shown in Figure 3.

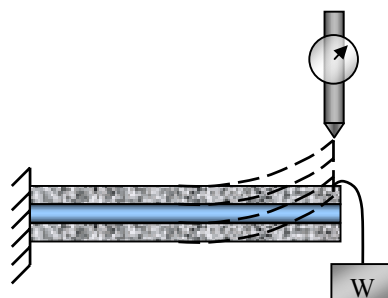


Figure 3 – Most basic measurement set-up for a bender actuator.

Recently, more sophisticated test configurations have been employed enabling more accurate measurements over the full deflection/force range. Researchers [6] have used an optical fibre displacement sensor, a sensitive load cell and a micropositioning system (to allow 3-D adjustment of the actuator on the sensor), in conjunction with lock-in amplifiers to determine the quasi-static response of RAINBOW actuators (frequency range 1-10 Hz). The free deflection (unloaded) and blocking force (loaded so that no deflection was observed) were determined separately for a range of driving electric fields. The same system has also been employed to measure the free deflection and blocking force of CRESCENT actuators as a function of applied field under quasi-static conditions (frequency $\leq 100\text{Hz}$) [5]. Schematics of the measurement system and of the data obtained are shown in Figure 4.

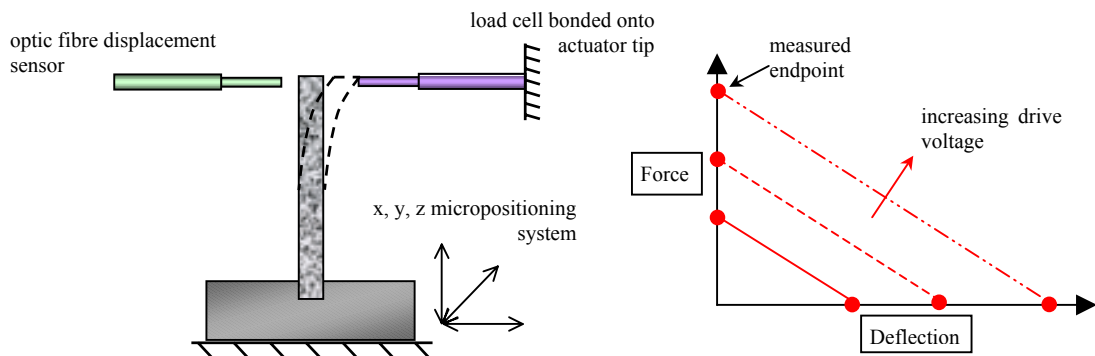


Figure 4 – Measurement system for blocking force and free displacement only.

More detailed static measurements have been carried out on Recurve actuators [9]. The force applied by the actuators was measured at incremental displacements, for several drive voltages. To achieve this, the force transducer was moved incrementally relative to the actuator using a precision stage, starting from the completely blocked state until the free deflection was reached and then returned to the original blocked position, monitoring the force at each step (i.e. the deflection route described in Figure 2). The system and data generated are shown in Figure 5.

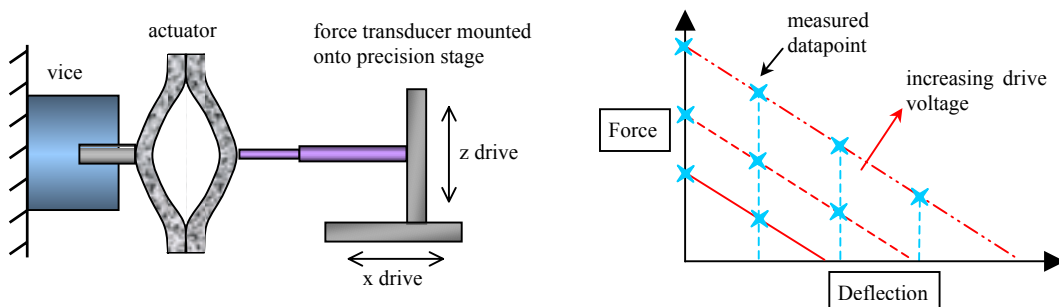


Figure 5 – Measurement system for blocking force, free displacement and intermediate conditions, using the constant deflection route.

2 BLOCKING FORCE RIG DESIGN

The preferred approach was to develop an automated system to measure both the actuator force and deflection at several applied voltages along a line of constant system stiffness, for a range of system stiffnesses. The objective behind adopting this particular approach was to employ a loading condition closer to that experienced by actuators in real industrial applications, by working against a given spring force [2], unlike the alternatives described previously.

The system was designed to perform static measurements, incrementing the DC drive voltage to the selected maximum and then back to the initial value. A cantilever beam was chosen to provide an easily quantified, variable resisting stiffness opposing the deflection of the actuator. The actuator displacement could then be measured directly from the cantilever movement which in turn would enable the partially blocked force to be determined using cantilever beam theory, eventually mapping the full actuator response from blocked to free displacement.

2.1 PRINCIPLES OF OPERATION

The basic components of the NPL design and their operation are demonstrated for a simple unimorph structure in Figure 6.

Variable stiffness resistance is achieved by altering the length of the cantilever, hence long cantilever lengths are flexible and easily displaced, whilst short cantilevers are stiffer and provide greater resistance to actuator motion. This effective change in cantilever stiffness can be controlled via a motorised z-drive, by moving the point of actuation progressively down along a fixed length cantilever. A ruby bead bonded onto the actuator at the location where maximum displacement is expected provides a unique contact point with the cantilever. A sensitive capacitance gauge monitors the deflection of the actuator for determining the point of contact between the actuator and cantilever. A long-range laser triangulation displacement sensor determines the deflection of the actuator/cantilever under the applied driving voltage which then enables the actuation force to be calculated indirectly.

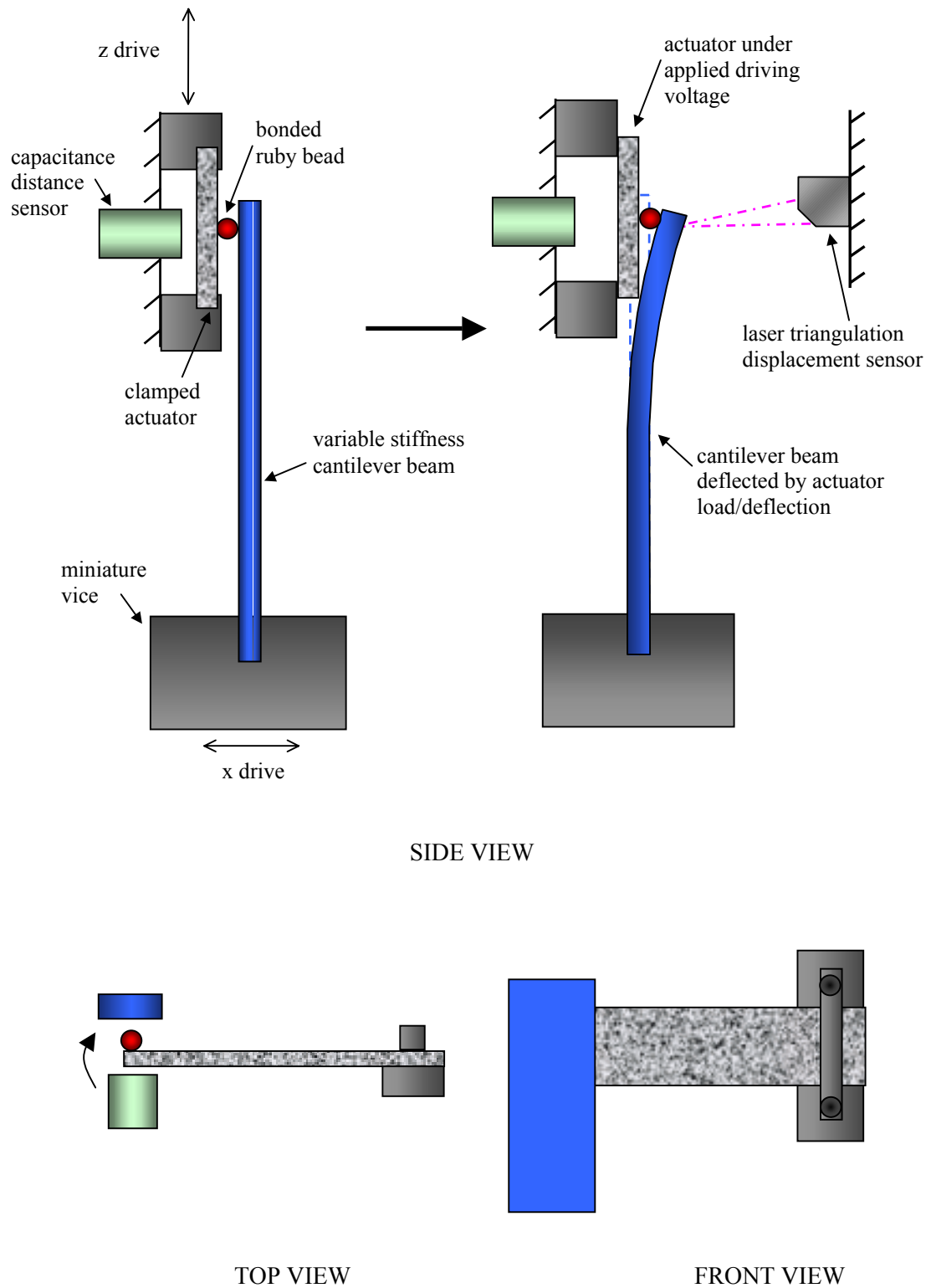


Figure 6 – Principle of blocking force measurement system using a simple cantilever as a variable stiffness loading mechanism.

2.1.1 Classical Beam Theory

Simple beam theory enables the displacement of a cantilever beam to be described for all points along its length with respect to the point of load application. For classical beam theory to apply the beam must be:

- straight
- narrow and long compared to its thickness
- comprised of homogeneous material, with the same modulus in tension as compression
- of uniform cross-section
- loaded perpendicular to the beam axis
- loaded below its linear elastic limit.

These hypotheses are important as otherwise additional factors need to be taken into account, such as shear corrections or anticlastic bending across the width. Similarly, the point of load contact should be along the axial mid-line of the beam to prevent unwanted torsion.

The typical cantilever loading scenario used within the design is presented in Figure 7, where the beam is built-in at one end and free at the other. For a transverse beam/actuator displacement δ (as measured at position z) caused by a force F (acting at position a) [10, 11], the displacement is given by the expressions in Table 4, where L is the length of the cantilever, E is the Young's modulus of the beam and I is the area moment of inertia of the beam ($I = wt^3/12$ for a solid rectangular beam of width w and thickness t).

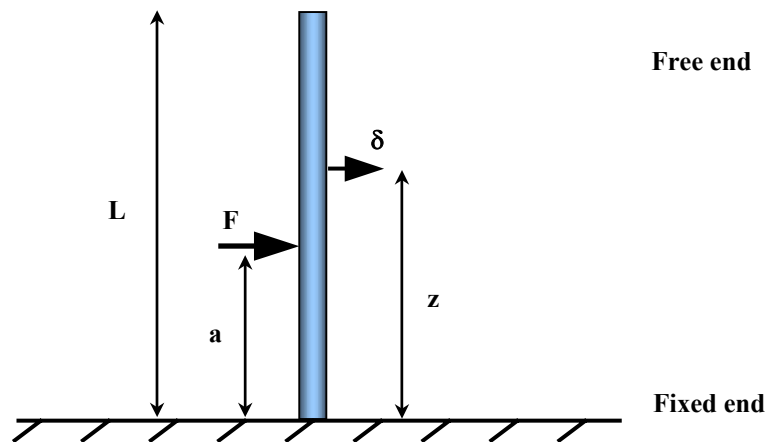


Figure 7 – Schematic of cantilever forces and deformations.

Table 4 – Cantilever deformations for the two different regions demarcated by the point of load application.

$0 \leq z \leq a$	$a \leq z \leq L$
$\delta = \frac{Fz^2}{6EI} (3a - z)$	$\delta = \frac{Fa^2}{6EI} (3z - a)$

At the measurement points, $z = a$ and $z = L$, these expressions can be simplified:

$$\text{for } z = a: \quad \delta = \frac{Fa^3}{3EI} \quad \text{and} \quad \text{for } z = L: \quad \delta = \frac{Fa^2}{6EI}(3L - a)$$

For the special case where load is applied at the end point of the beam $a = L$ and deflection is monitored at the same point $z = L$, the displacement and/or force can be calculated using:

$$\delta = \frac{FL^3}{3EI} \quad \text{or} \quad F = \frac{3\delta EI}{L^3}$$

In order to illustrate this method, the load per unit cantilever displacement for cantilevers of different lengths and cross-sections are given in Table 5 below. It is important to note the wide range of effective cantilever stiffnesses obtained, and consequently the dramatic differences in load/deflection sensitivity, with changing cantilever length and thickness which can be achieved.

Table 5 – Effective spring constant (N/mm) as a function of the distance from the built-in end to the point of load application for different cantilever materials/geometries.

Cantilever properties/lengths	25 mm	50 mm	75 mm	100 mm	150 mm	200 mm	250 mm
E = 210 GPa (steel) w = 6 mm t = 2 mm	161.3	20.2	6.0	2.5	0.75	0.32	0.16
E = 75 GPa (Al) w = 6 mm t = 2 mm	57.6	7.2	2.1	0.90	0.27	0.11	0.058
E = 75 GPa (Al) w = 12 mm t = 4 mm	921.6	115.2	34.1	14.4	4.3	1.8	0.92

2.2 DETAILED SYSTEM DESIGN

The final system design is presented in Figure 8. The cantilever is fixed into a vice which is mounted on the horizontal motorised drive for proximity adjustment to the actuator. The mounting bracket holds both the actuator and the 3-axis mount, which in turn holds the laser sensor, onto the vertical motorised drive. As a result, once the laser spot is located on the correct point of the actuator, the whole assembly can move up and down the cantilever in unison, so the laser remains locked onto the same point of the actuator at all cantilever heights tested. The constructed system is shown in Figure 9 and in more detail in Figure 10.

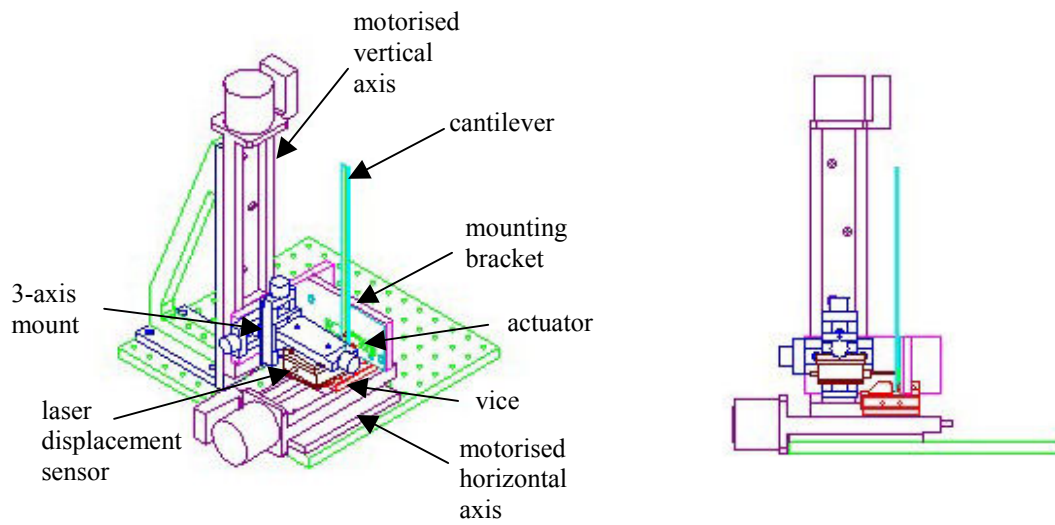


Figure 8 – Final system design drawings.

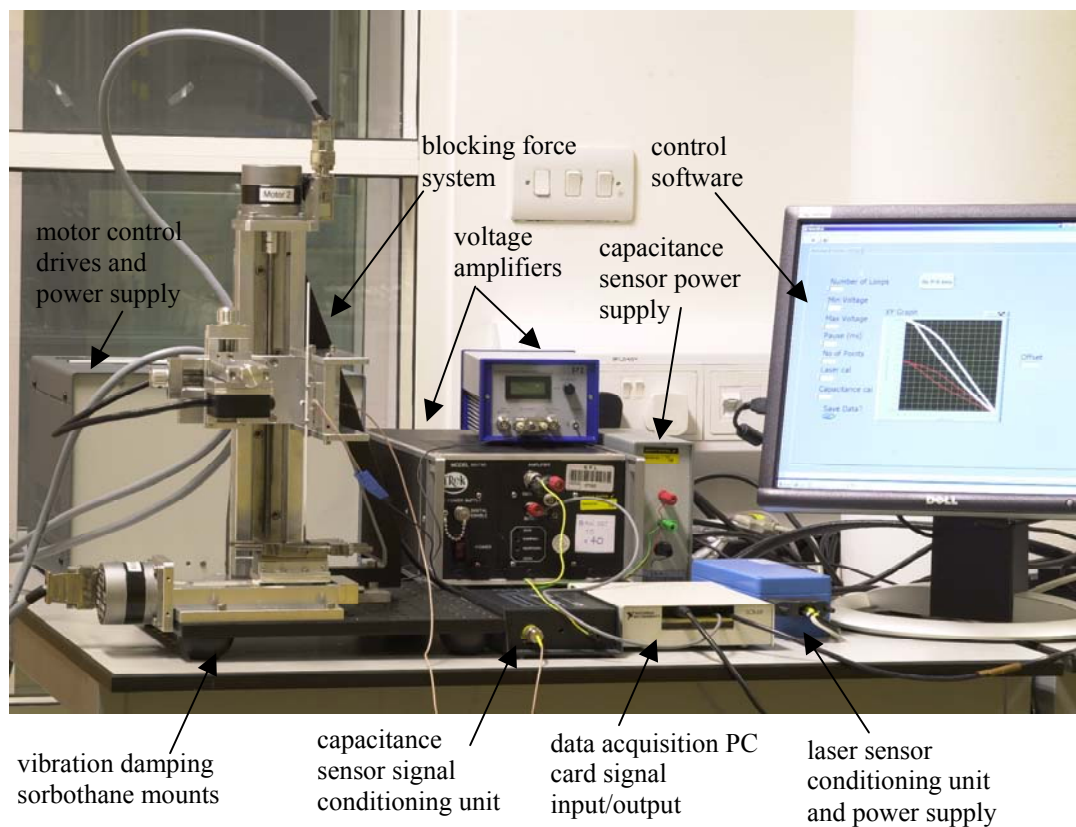


Figure 9 – The experimental system and principal components.

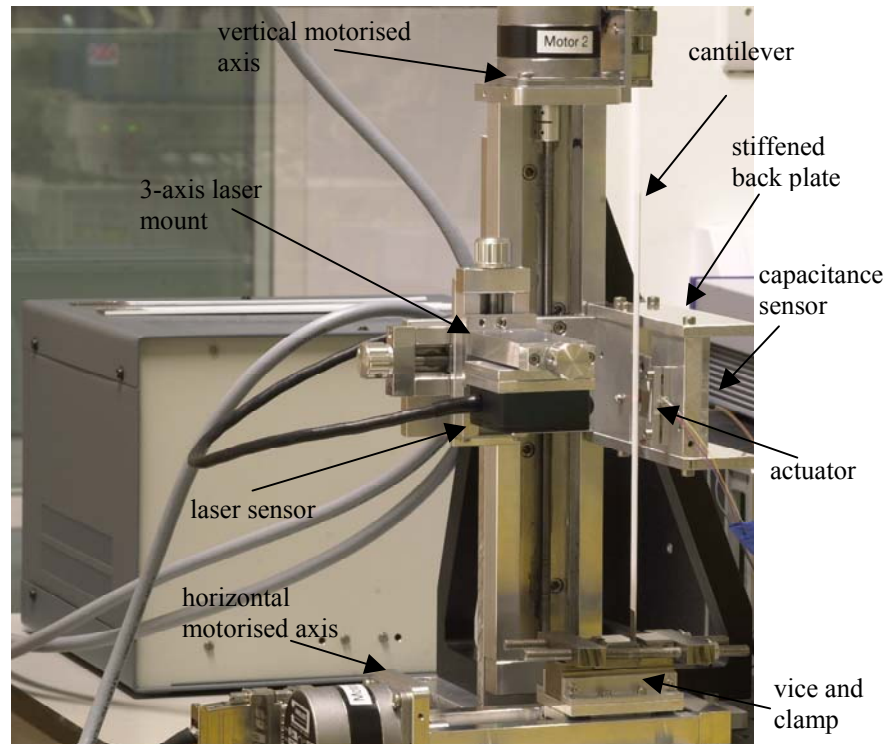


Figure 10 – Detail of the experimental system: modifications included stiffening ribs attached to actuator backing plate, side brace on vertical motorised axis, reduction of lever arm on vertical axis bearings and additional clamp on vice jaws.

The system was constructed from commercially available components:

- motorised stepper axes
 - vertical – 12 " leadscrew, 320 mm travel, 1 mm thread pitch
 - horizontal – 6 " leadscrew, 50 mm travel, 0.5 mm thread pitch
- 80V DC /3 A (24 V / 1.8 A logic) power supply unit
- 2 x control indexer drives 2.5 A /phase, 4000 steps/revolution, RS232 communication
- 3 x small manual adjustment axes, 3 " leadscrew, 30 mm travel, 1 mm thread pitch
- precision ground, flat-faced jaw, hardened steel miniature vice (0 – 40 mm opening)
- standard metric optical component breadboard and right angle brackets
- 2.5 mm range capacitance gauge and conditioning unit (0 – 10 V output)
- 10 mm range laser triangulation displacement sensor and conditioning unit (± 10 V output).

Adaptor plates and attachments, to connect the components as necessary, were manufactured from 10 mm thick aluminium plate. The power supply unit and control drives were mounted in a vented metal equipment housing case with through connectors for the motor power and communication cables, mains supply and computer communication.

Peripheral components included power amplifiers for increasing the control software signal to required actuator driving levels and a conditioning unit connecting the sensor outputs (single-ended referenced) and amplifier inputs (analogue -10 V – $+10$ V) to the control software via a 6036E National Instruments data acquisition card.

2.2.1 Software Control and Automation

The basic procedure for automating this system is shown in Figures 11 and 12 and listed below:

- calibration of the system,
- measure free displacement at several applied fields,
- measure partially blocked displacement at several applied fields for one cantilever length/stiffness,
- repeat partially blocked measurements for a range of cantilever lengths/stiffnesses.

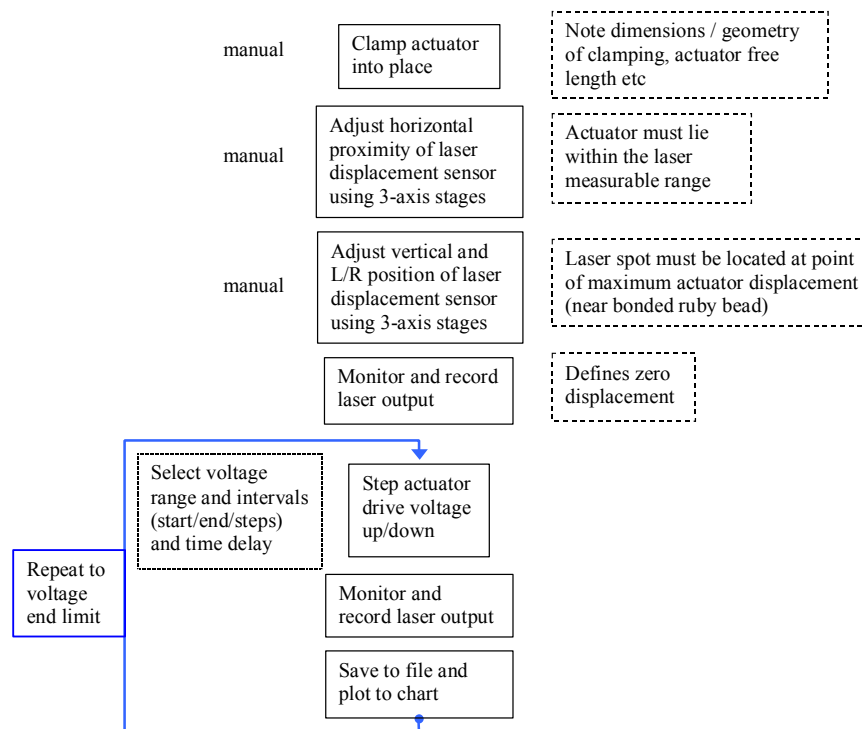


Figure 11 – Basic procedure for automating free displacement measurements.

The control software was written and compiled in LabView v. 6.0. Screen views from the software are shown in Figures 13, 14 and 15.

Figure 13 shows the operator-controlled screen for adjusting the position of the motorised axes prior to testing. This allows fixed relative displacements in either direction of each of the motorised axes or a direct move to a pre-specified referenced position. In addition, live readings from both the displacement sensors enable the sensor proximities to the cantilever and actuator, respectively, to be adjusted to within the appropriate working range. The current motorised stage positions with respect to the reference position (set to mid-travel for each axis) are also displayed.

Figure 14 shows the input parameters for setting the voltage-displacement loop including:

- minimum and maximum actuator drive voltages
- number of points - which defines the number of voltage steps between minimum and maximum voltages and hence the voltage increment/interval
- pause - which defines the time interval between the voltage increment and the laser displacement measurement
- laser and capacitance calibration values (mm/V)
- number of loops – which defines the number of times the system runs through the voltage cycle before moving on to the next cantilever position.

Also shown on this screen is a trace of the output from both capacitance and laser sensors, for each voltage cycle in turn. The software runs from the minimum voltage to the maximum and back again, and repeats for the number of loops specified by the user to allow settling of the system to occur. This ensures the data obtained is consistent and repeatable by making the relaxation time, electric field and mechanical history the same for each loop to be analysed.

Figure 15 shows the screen view for the automated test including contacting the actuator with the cantilever and subsequent partial blocking force measurement. Again, the real-time capacitance gauge and laser displacement outputs are displayed, in both numerical and graphical forms, as the cantilever (clamped in the vice) moves towards the actuator. The required operator inputs include:

- contact volts – which defines the capacitance gauge output threshold level associated with contact (generally set 0.2 V below the uncontacted level)
- max search – which defines the maximum distance within which the threshold level should be found (prevents the cantilever driving too far into the actuator)
- x step size – which sets the horizontal increments used to search for the threshold contact level.

Once the contact level is found on the first pass, the cantilever is backed off a small distance and approaches the actuator once more but this time in micron steps until the threshold level is found precisely. At this point, the voltage-displacement loop subroutine is run to give the partially blocked data at the current cantilever height, the plot of which is shown on screen. Thereafter the process continues for several more cantilever positions defined by:

- z step size – defines the increments in cantilever length from the initial position at which measurements will be made
- number of z steps – sets the number of partially blocked cantilever height measurement positions.

The z step value is updated after each measurement position to indicate the number of z steps which have been performed.

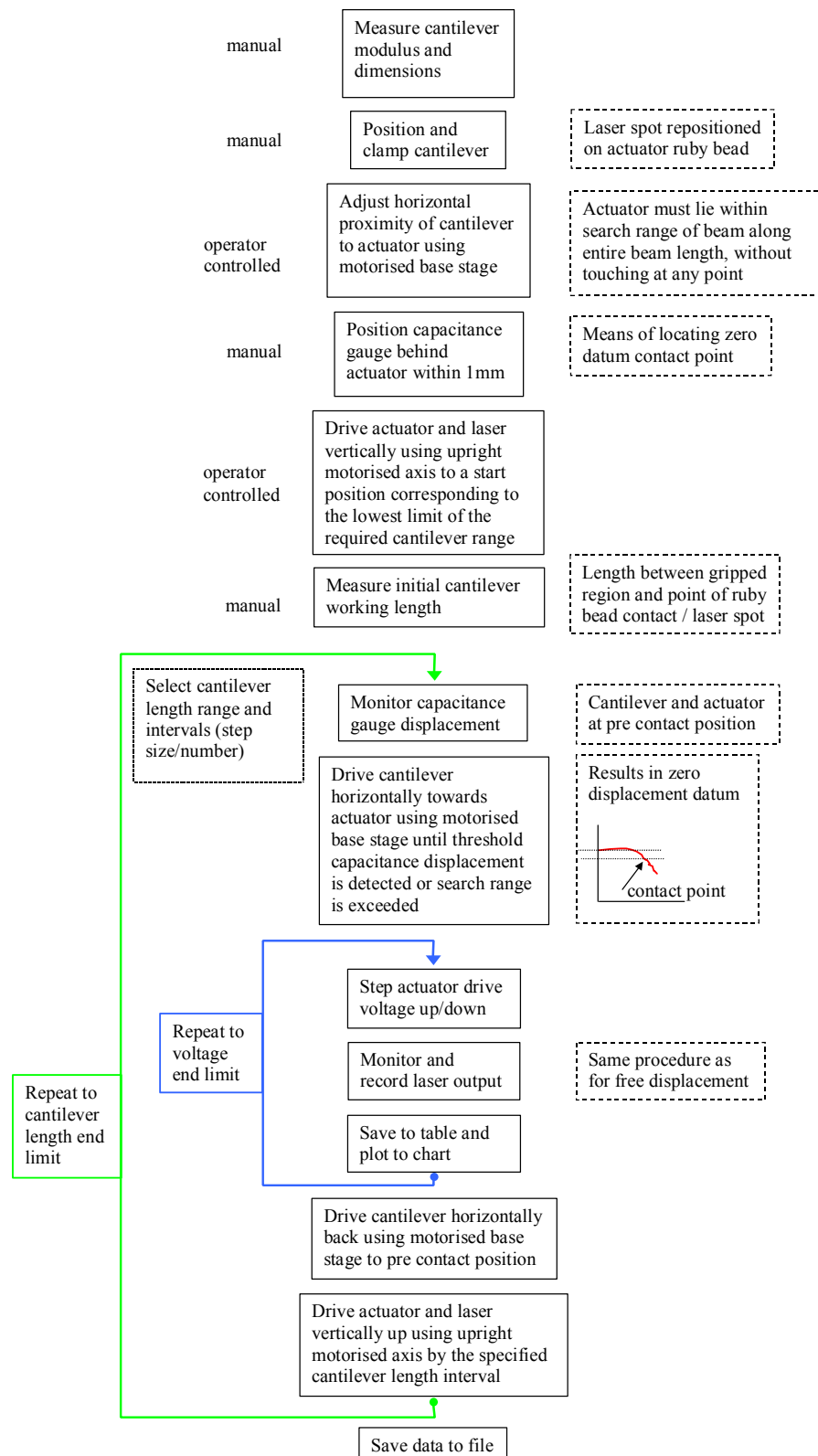


Figure 12 – Basic procedure for automating partially blocked displacement measurements.

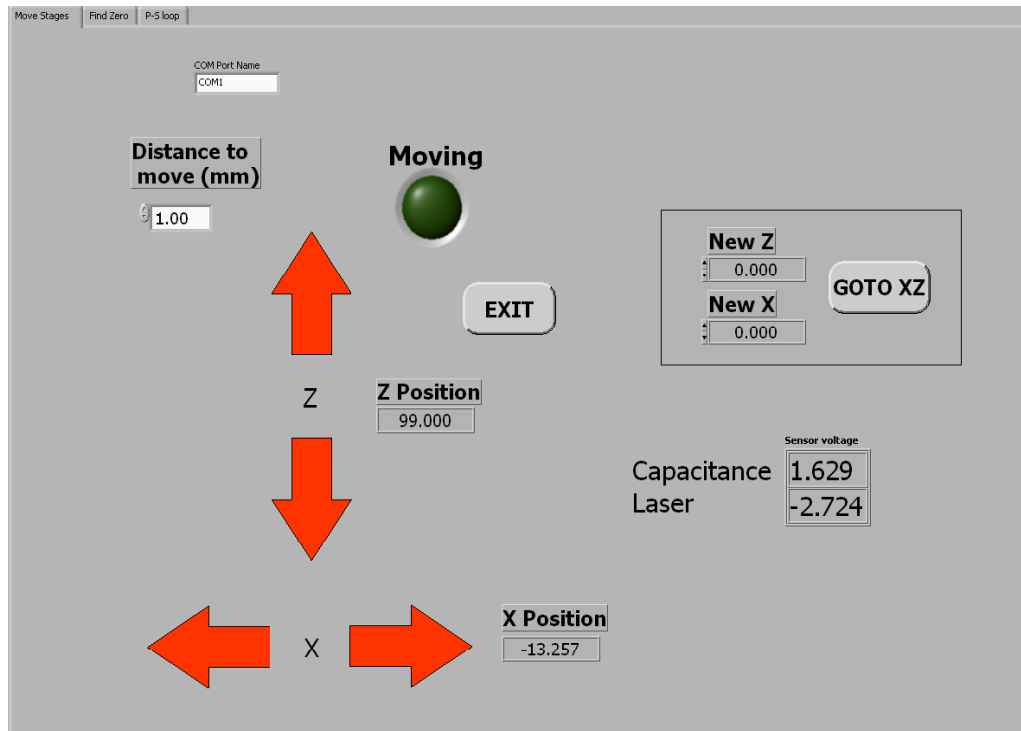


Figure 13 – Screen view of software control for operator adjustment of motorised axes.

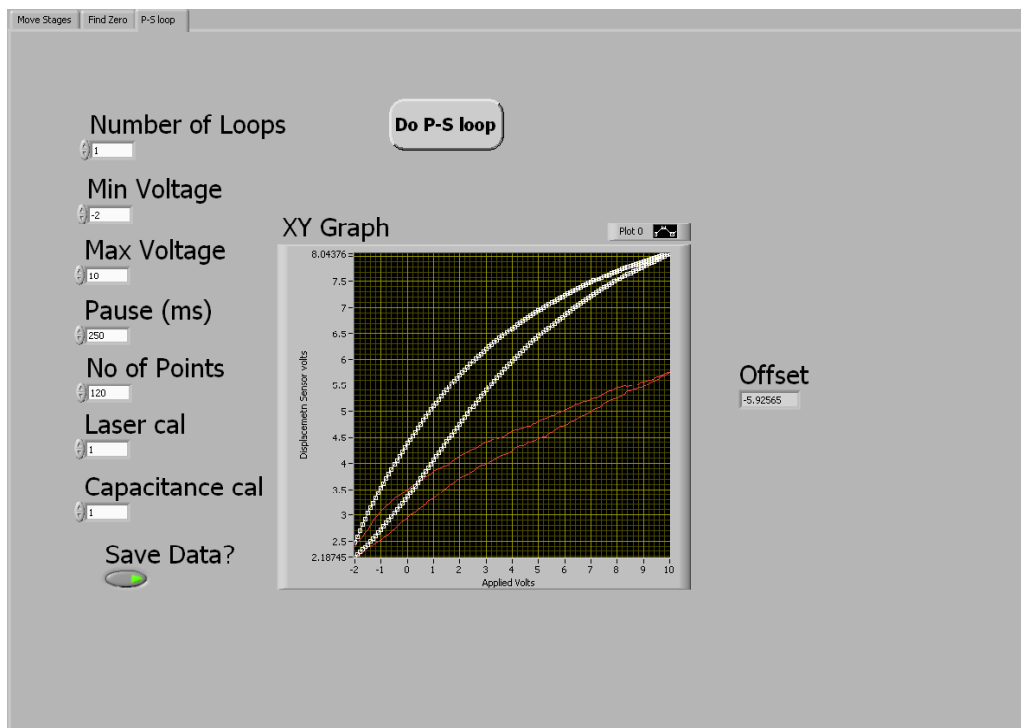


Figure 14 – Screen view of software control for setting voltage-displacement loops and parameter input: plot shows laser (red) and capacitance gauge (white) output with applied actuator voltage for a single cycle.

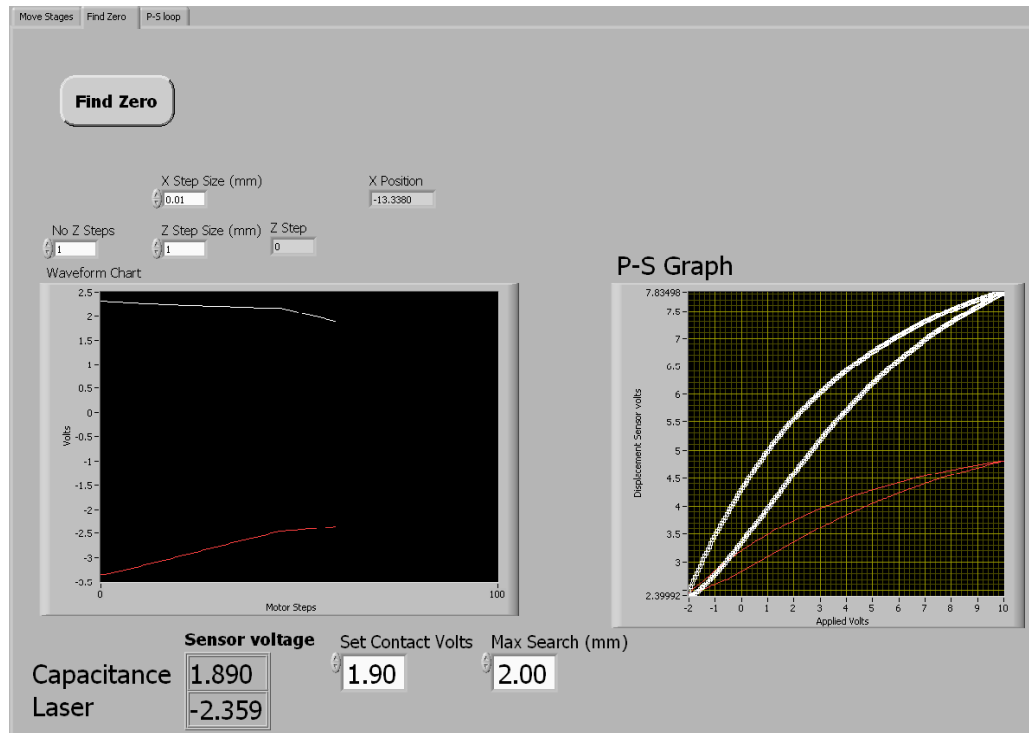


Figure 15 – Screen view of software control for parameter setting the zero contact position subroutine and partially blocked voltage-displacement loops: plot shows laser (red) and capacitance gauge (white) output as contact is made (left) and the same with applied actuator voltage for a single cycle (right).

2.2.2 Laser and Capacitance Gauge Calibration Checks

Simple calibrations of the two sensors were carried out to determine their performance within the system. A thick steel cantilever, painted matt white on one face, was clamped in the vice and stepped in 1 μm increments towards the sensor. The white face was used as the optimum laser reflective surface and the metallic face was on the capacitance gauge side. The increments were produced by the horizontal motorised stepper axis, initially checked to the nearest micron using a digital indicator. Five sets of measurements were made over the full sensor range. The data from these checks are shown in Figures 16 and 17. The nominal error in the output compared to the best fit line are also given. Both sensor outputs were stable to ± 1 mV.

The capacitance gauge output was non-linear, so best fit lines were determined for the output range 0.2 – 7 V giving an average conversion factor of 232.91 $\mu\text{m}/\text{V}$. The error plot shows more non-linearity and errors of up to 5 μm from the straight line prediction in the range 0.2 - 6.5 V. This reduced range was used for subsequent contact point measurements during the set-up stage of testing. The capacitance gauge was not used for absolute displacement measurements and so the associated errors were insignificant.

The laser sensor output was linear over the whole range with an average conversion factor of 502.85 $\mu\text{m}/\text{V}$. The error plot showed inconsistent errors with repeat tests, but generally the negative output range is better than the positive range, with displacement errors within 10 μm from the straight line prediction. The laser was therefore operated in the range -2 to -10 V during the cantilever measurements.

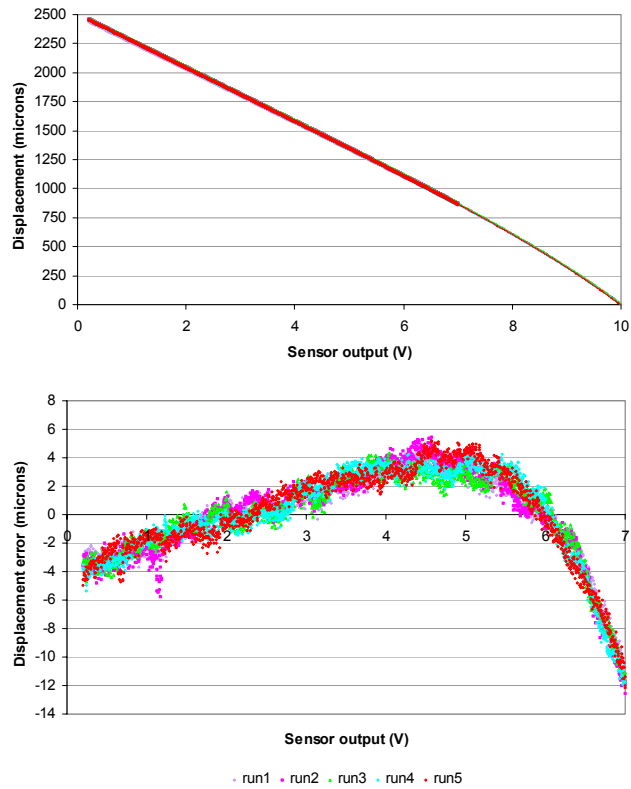


Figure 16 – Calibration curve and error plot for capacitance gauge with displacement.

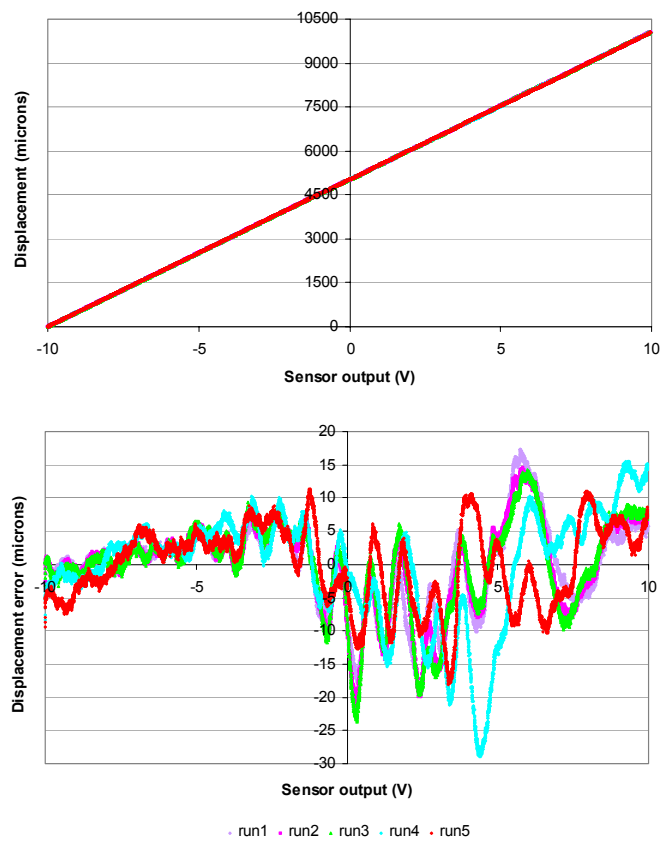


Figure 17 – Calibration curve and error plot for laser sensor with displacement.

2.2.3 Compliance Corrections

Before the system was tested with real actuators, the mechanical performance was investigated for the load range likely to be encountered for the majority of macroactuators. In this case, a small 100 N load cell was used to apply and monitor loads in place of an actuator. The load cell was screwed into the end of a push-rod which was clamped rigidly into place on a manual linear translation stage. The stage was, in turn, fixed behind the back plate of the measurement system through the stiffening plate. This set-up is shown in Figure 18.

By manually turning the screw on the linear stage, the load cell was pushed forward, both mimicking the displacement of an actuator and also providing a measurable load for the system to react against.

On initial inspection, problems were discovered with compliance non-linearities throughout the system. This was improved upon by the modifications shown in Figure 10. Firstly, a side brace was added to the vertical motorised axis to prevent any possible rotations at longer cantilever lengths. Secondly, stiffening plates were added to the back support plate to reduce the cantilever bending at high blocking forces. Thirdly, the connection between the U-section supporting the laser arm and actuator and the vertical axis was moved closer to the actuator back support to minimise the moment on the axis bearing and reduce movement when the actuator applies a load. Finally, a clamp was added to provide horizontal loading across the faces of the vice to ensure the cantilever was rigidly fixed and that simple beam theory would apply as expected.

Four different measurements were made, using the load cell and laser displacement system together, on the modified system. A 10 mm thick plate of hardened steel was used as a very stiff cantilever; by keeping the point of load contact to a short length, no bending displacement would be expected for the load range considered. This acted as a nominally fixed point such that any deflection in the surrounding system could be measured with reference to it.

Initially, the cantilever block was loaded and its front face was monitored by the laser to determine the displacement of the vice and laser support arm combined. Secondly, the cantilever block was loaded and the back support plate was monitored by the laser to determine the displacement of the laser arm and back support plate together. Thirdly, a second plate was positioned in front of the loaded cantilever block, but disconnected from it or the surrounding mechanical system, and was monitored by the laser to determine the compliance of the laser arm in isolation. Finally, the laser was repositioned to monitor the movement of the vice only as the cantilever block was loaded. The compliance curves for these experiments are shown in Figure 19.

With an actuator in place, the same compliances could be corrected for by running two sets of tests: the first measuring the displacement of the cantilever for different voltages and stiffnesses, and the second measuring the displacement of the back support plate under nominally identical conditions to the first test. In this instance, the back plate effectively operates as a load cell, allowing the load to be determined from the measured displacement and enabling appropriate corrections to be made to the measured cantilever displacement from the compliance measurements for the surrounding system. Obviously, the optimum solution is to minimise or reduce the compliance by stiffening the system still further, but for the initial investigation compliance corrections were sufficient.

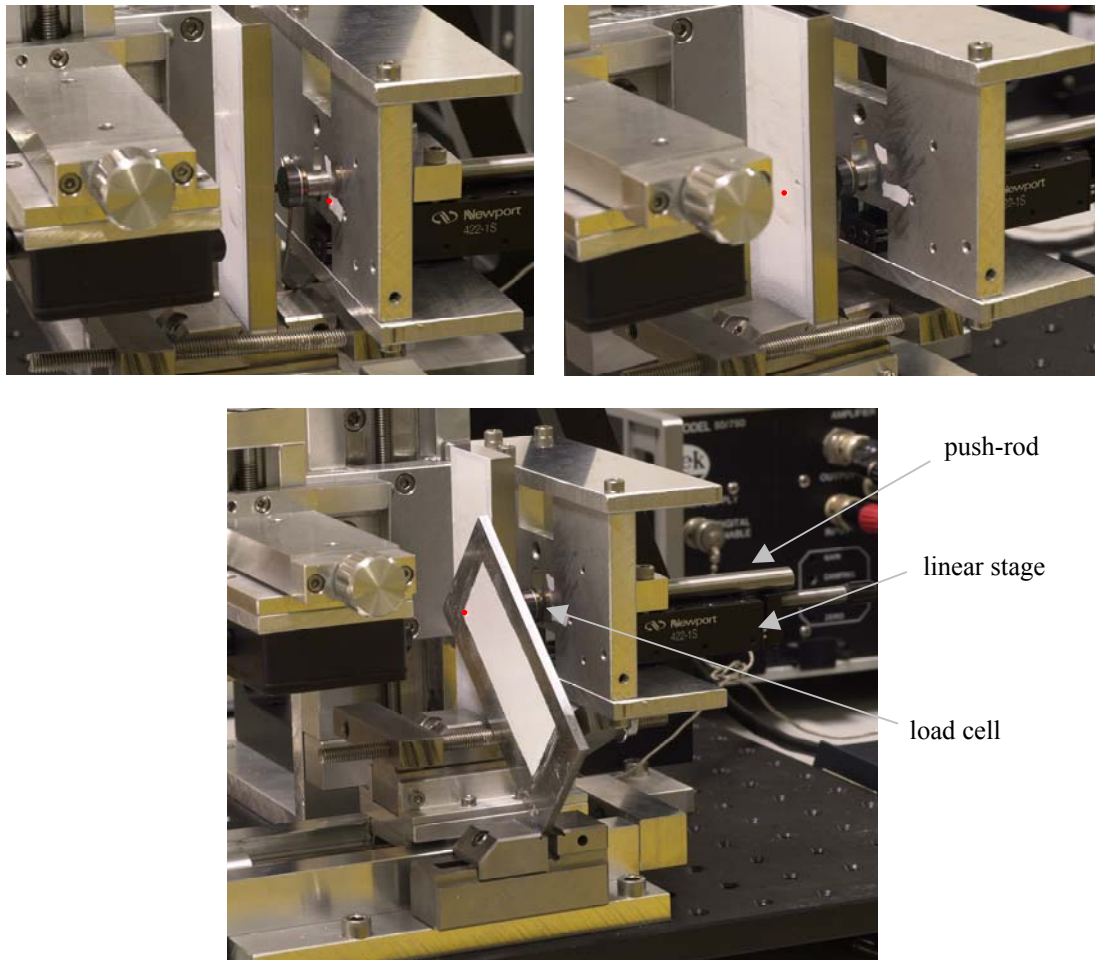


Figure 18 – Set-ups for system compliance measurement: laser and back support (top left), laser and vice (top right) and laser only (centre) where the laser-monitored surface is marked with a red dot.

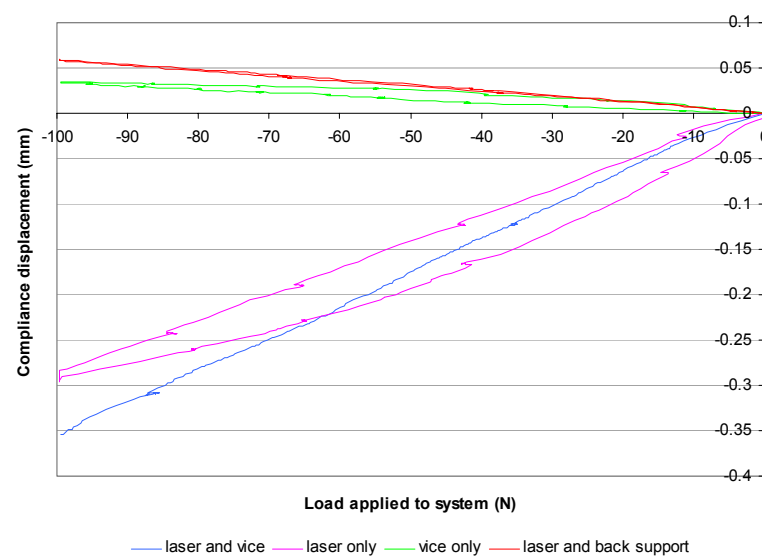


Figure 19 – Results for compliance corrections in the blocking force system.

2.2.4 System Performance Calibration

Once the system compliances were analysed, the system was checked for load and displacement accuracy using a beam with known stiffness. An aluminium cantilever was loaded and displaced via the load cell and monitored using the laser at different lengths, as shown in Figure 20, from which the modulus was calculated using the classical beam theory equations introduced in section 2.1.1. The linear force/displacement traces obtained for the different cantilever lengths are shown in Figure 21.

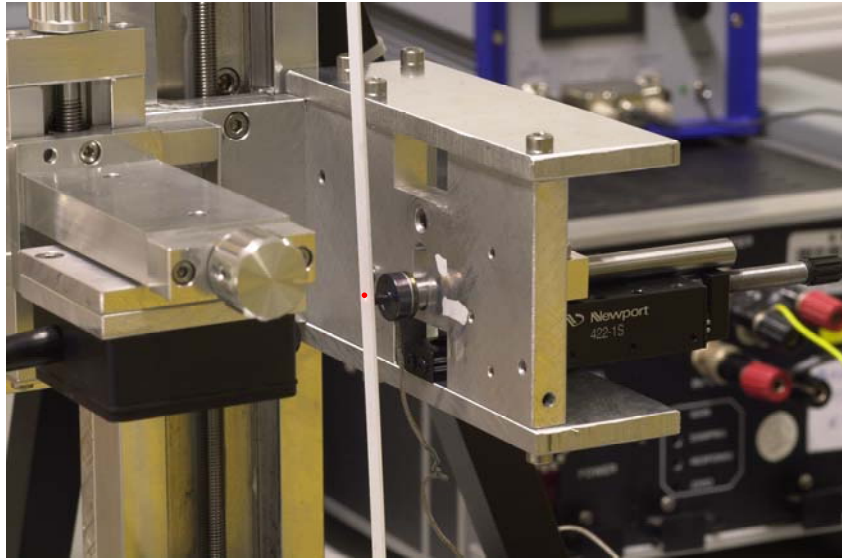


Figure 20 – Experimental set-up for calibration of system performance.

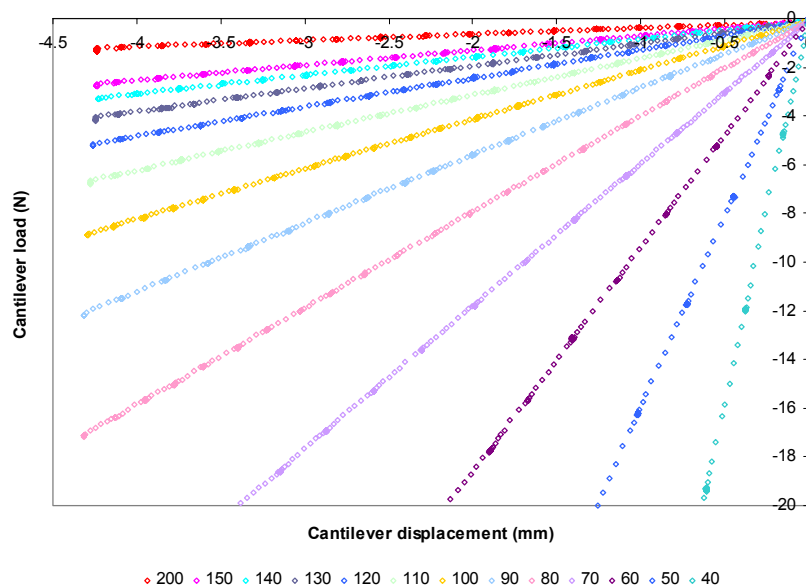


Figure 21 – Raw force/displacement data for aluminium cantilever 15.15 x 1.89 x 250 mm for increasing distances between load application point and fixed cantilever end, as indicated (in mm).

The modulus thus calculated was compared with the modulus as determined by an independent method: an axial strain gauge was bonded close to the root of the cantilever to monitor the strain produced by the displacement applied at the point of load contact, as shown in Figure 22 for two different positions of load application. The stress in the beam was calculated using the equation below, where L is the effective cantilever length given by the distance between the mid-line of the strain gauge and the loading point, wt^2 is determined from the cantilever dimensions and F is the load applied as measured by the load cell.

$$\sigma = \frac{6FL}{wt^2}$$



Figure 22 – Strain-gauged cantilever for system calibration.

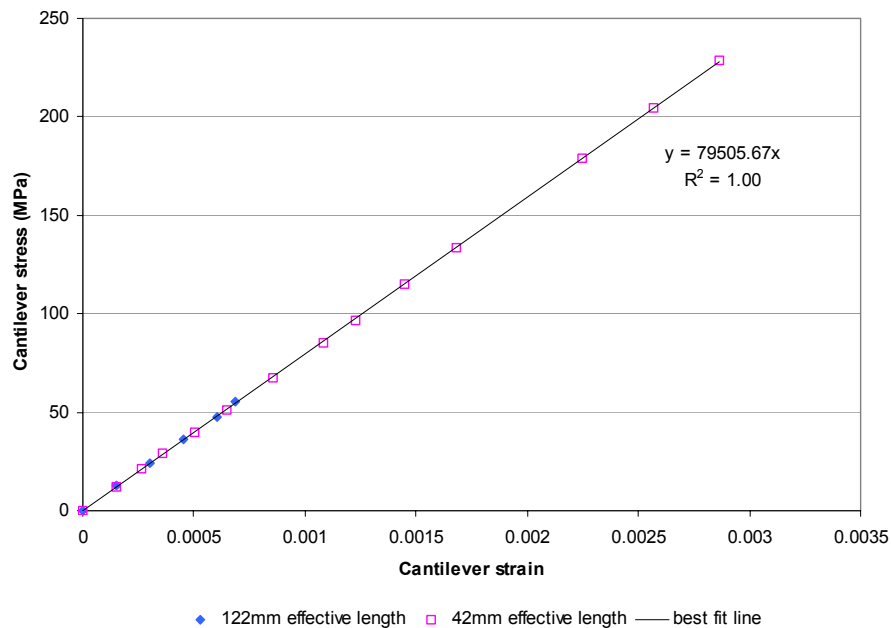


Figure 23 – Stress-strain plot from strain-gauged aluminium cantilever.

The stress-strain data presented in Figure 23, gave a modulus value for aluminium of 79.6 GPa (book value ~ 76 GPa). The modulus calculated from the force-displacement data is presented in Figure 24, which once corrected for compliance shows an improved scatter with changing cantilever length (~ 1 GPa) and an average value close to that found from strain measurements. These values were also shown to be repeatable to within the same level of scatter. This provided some confidence that the system would perform as expected and the blocking force data would not be an artefact of the manner in which it was measured.

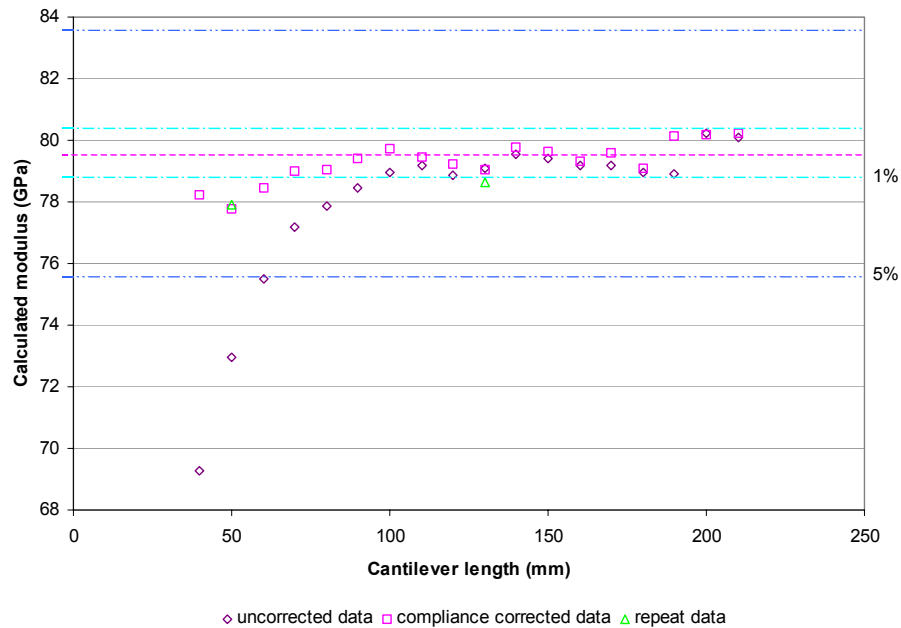


Figure 24 – Calculated aluminium modulus with cantilever length, both uncorrected and compliance corrected (dotted line represents modulus value from stress-strain).

For the final blocking force tests a much stiffer material was required to provide an appropriate cantilever stiffness range. Stainless steel was chosen and a similar set of modulus checks were carried out, the results of which are shown in Figure 25, on a cantilever $4.061 \times 2.003 \times 250$ mm. The stress-strain value of modulus was 187.8 GPa (book value ~ 193 GPa) and that from the force-displacement data showed a variation with length of ~ 10 GPa about a similar value. For the subsequent blocking force tests a modulus value of 187.8 GPa was used.

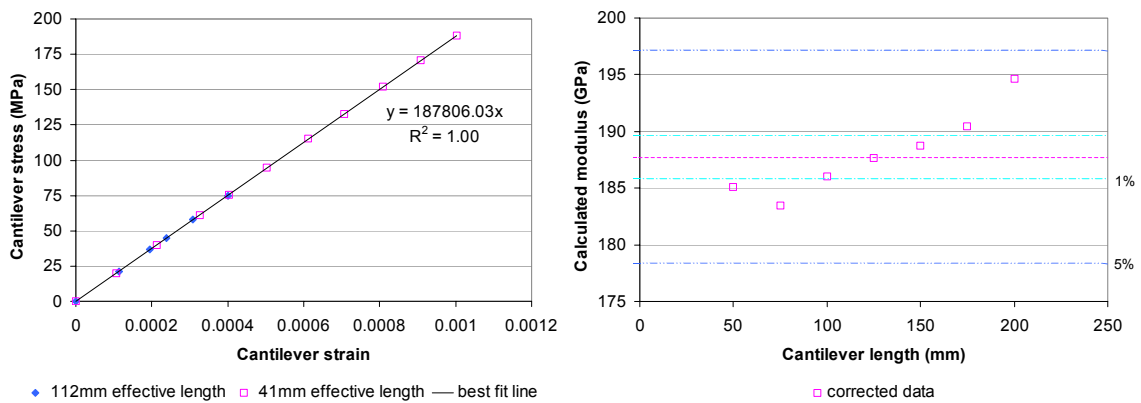


Figure 25 – Stainless steel modulus results: stress-strain plot from strain-gauged cantilever (left) and compliance corrected force-displacement derived modulus (right).

3 EXPERIMENTAL RESULTS AND DISCUSSION

3.1 ACTUATORS INVESTIGATED

The final test system design was used to characterise the full force/deflection response for several commercially available actuators, described below and in Table 6 and shown in Figures 26 - 28. Figure 26 shows the region on each of the actuators from where load (marked by the ruby bead) and where free displacement were measured (marked with white paint).

- THUNDER (FACE International Corporation) [12] – stress-biased domed actuator with a PZT layer bonded between a stainless steel substrate and an aluminium top layer, using a high performance epoxy adhesive under a precise temperature and pressure cycle.
- Co-fired multilayer bender actuator (Physike Instrumente Ceramic GmbH) [13] – based on the functioning principle of bimorphs but with reduced operating voltage, faster response time and higher stiffness; manufactured from ceramic layers 25 μm thick with internal electrodes and fully surrounded by ceramic insulation applied in a co-firing process.
- Planar bimorph (Servocell Ltd) [14] – based on the functioning principle of bimorphs but within a more compact area, with greater displacement and using low cost processing; consists of two adjacent bimorphs mechanically connected at one end to produce a U-shape.

Table 6 – Manufacturer supplied actuator characteristics.

Actuator	Type	Blocking force (N)	Free displacement (mm)	Operating voltage (V)	Dimensions (mm)
PICMA PL 140	Multilayer bimorph	0.5	1	0 $\rightarrow$ 60	45 x 11 x 0.6
THUNDER TH-8R	Stress-biased unimorph	67	2	-240 $\rightarrow$ 480	63.5 x 12.7 x 0.483
PBT F740	Planar bimorph	0.17	1.6	-80 $\rightarrow$ 400	38 x 11 x 2

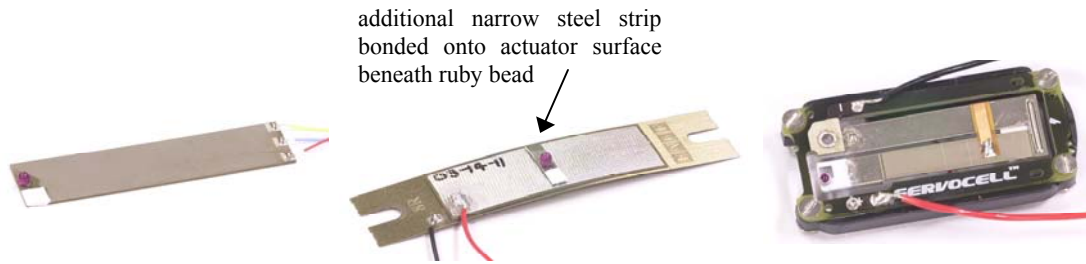


Figure 26 – Novel macroactuators investigated: multilayer cofired bimorph (left), pre-stressed unimorph (centre) and planar bimorph (right), point of load contact marked by bead and free displacement measured from white painted region.

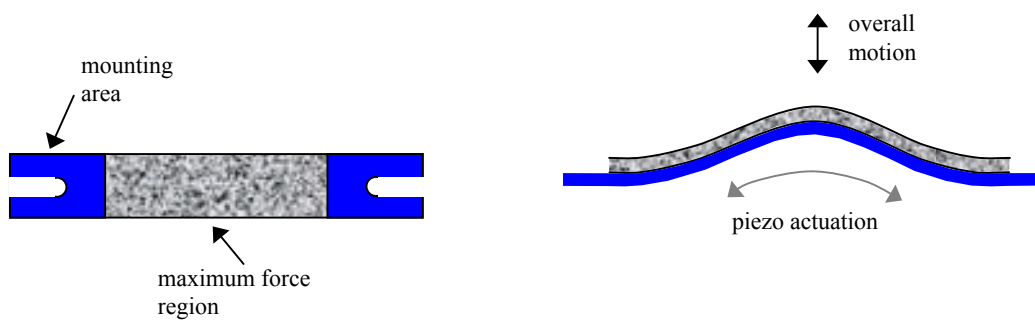


Figure 27 – Pre-stressed unimorph actuation: top and side views.

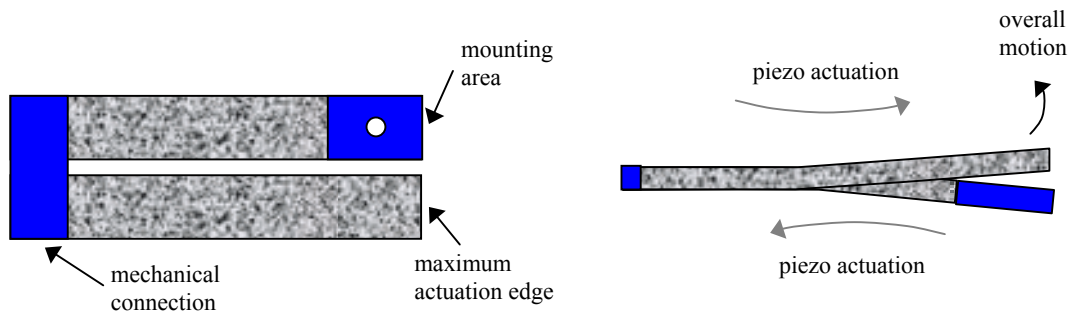


Figure 28 – Planar bimorph actuation: top and side views.

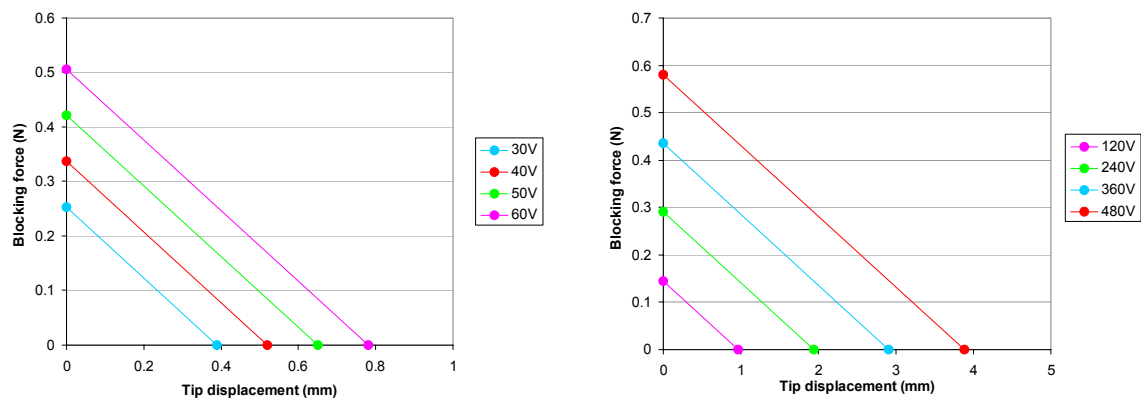
3.1.1 Prediction Data

The predictions discussed in section 1.2.1 are considered here for two of the actuators studied. The prediction was based approximately on the manufacturers' specified structure, dimensions and typical material properties. The values employed for the calculations are given in Table 7.

Table 7 – Structure and properties of multilayer bimorph and stress-biased unimorph actuators used in prediction examples.

Actuator	Multilayer bimorph	Stress-biased unimorph
Length L (mm)	38	63.5
Thickness t (mm)	0.6	0.48
Width w (mm)	11	12.7
Piezoelectric layer thickness (mm)	0.05	0.2
Metal layer thickness (mm)	n/a	0.28
Piezoelectric modulus E_p (GPa)	60	65
Metal modulus E_m (GPa)	n/a	195
Piezoelectric coefficient d_{31} (pm/V)	180 (PIC 255 PZT, manufacturer)	320 (soft PZT, [3])

The results of the example predictions are shown in Figure 29. The analysis only provides the free deflection and blocked force for different driving voltages, all points in between are assumed linear.

**Figure 29 – Predicted force/deflection characteristics of a multi-layer actuator (left) and a stress-biased actuator (right) for different applied fields.**

These predictions show reasonable agreement with manufacturers' values for blocking force and free deflection for the multi-layer bimorph (0.5 N and 1 mm), but overestimate the free deflection whilst simultaneously underestimating the blocking force for the stress-biased unimorph (65 N and 2 mm). This highlights the need for accurate measurements covering the complete force-displacement capabilities of the actuator.

3.2 FREE DISPLACEMENT MEASUREMENTS

Initially, the free displacement of the three actuators was investigated in detail using the set-up shown in Figure 30. The clamping for each of the actuators is described below:

- Planar bimorph – supplied built into a rectangular plastic frame which was then screwed into the back plate at the four corners with M2 countersunk screws,
- Stress-biased unimorph – bolted onto the back plate at one end through one of the slots supplied in the steel substrate with an M3 bolt, the other end was left free to move but simply supported against the back face,
- Multilayer bimorph – bonded between chamfered edge aluminium clamps using room temperature curing two-part epoxy adhesive, shown in Figure 31, whole assembly then bolted into the back plate (to protect against mechanical/clamping damage).

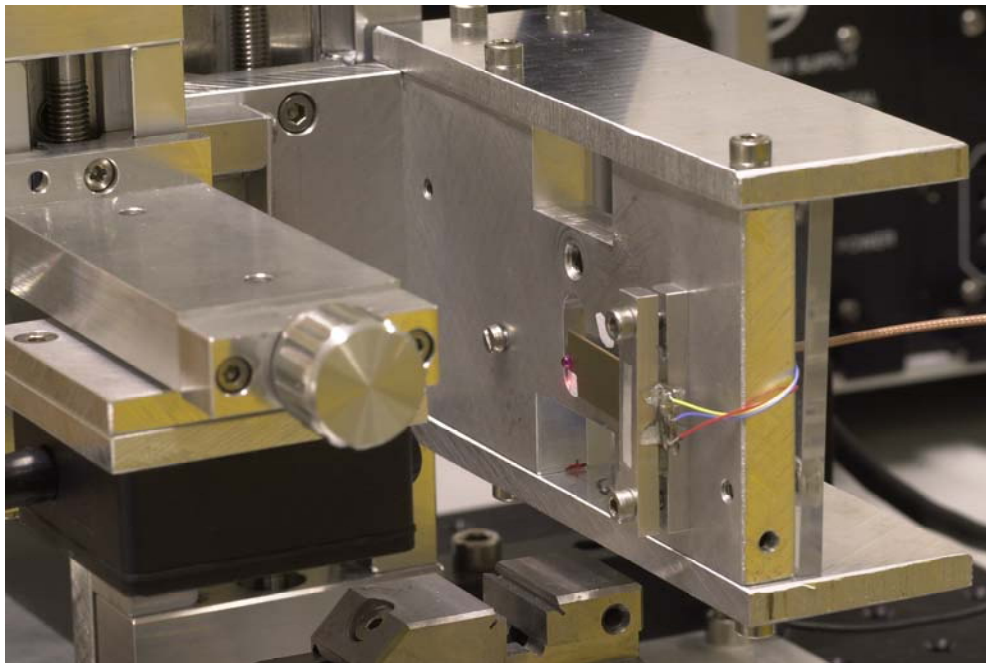


Figure 30 – Test set-up for free displacement measurements: planar bimorph shown.



Figure 31 – Clamping arrangement for planar bimorph; 35 mm free length.

3.2.1 Maximum Displacement

For the free displacement measurements, a large number of consecutive voltage loops (15 in this case) were run immediately after the actuators were connected to the power amplifier to check the consistency/repeatability of the measurements. The voltage range used was based on manufacturers' data on the allowable positive and negative voltage limits. The displacement with incrementing voltage was monitored continuously using the laser displacement sensor and the total displacement of each loop was established. The results are shown in Figure 32. This shows that the first loop is significantly different to subsequent cycles in terms of both total displacement and, in the case of the planar bimorph, in the displacement response with voltage. The hysteresis loops for each of the actuators are also significantly different. The planar bimorph shows the greatest degree of both non-linearity and asymmetry, with the multilayer bimorph showing the smallest hysteresis and the best degree of symmetry between the outward and return halves of the voltage loop. The greatest hysteresis was observed with the stress-biased unimorph. For subsequent measurements, data from either the 3rd or 5th cycles was used for analysis to ensure consistency of the data.

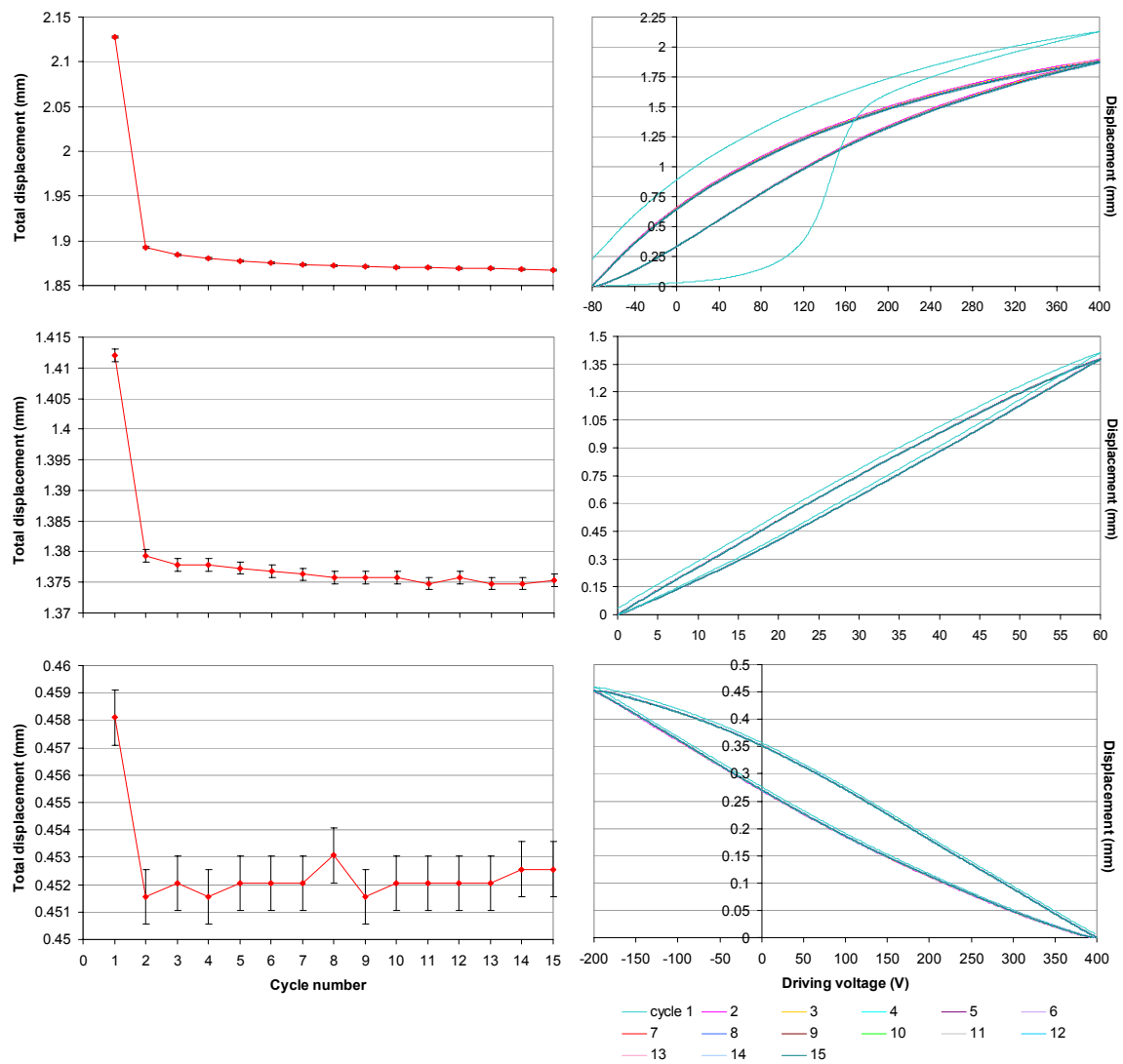


Figure 32 – Free displacement repeatability measurements for planar bimorph (upper), multilayer bimorph (centre) and stress-biased unimorph (lower).

The planar bimorph was subsequently used to investigate further factors affecting the free displacement measurements.

3.2.2 Hysteresis

The effect of applying different voltage cycles to the planar bimorph actuator on the hysteresis loop obtained was studied and is shown in Figure 33. The hysteresis loops were found to superimpose directly over one another, whether the loop was run up or down in voltage, over the same range. Cycles performed over smaller voltage ranges gave narrow symmetric hysteresis curves which became wider and more non-linear as the voltage range increased (to over 100 V) to the maximum recommended. This might limit the voltage range of operation of the actuator to where the hysteresis is linear/symmetrical. It was also noted that the starting voltage level dictated the form of the outward section of the cycle, all cycles starting at the same level following the same path; whilst the return portion of the cycle was dependent on the maximum voltage level reached.

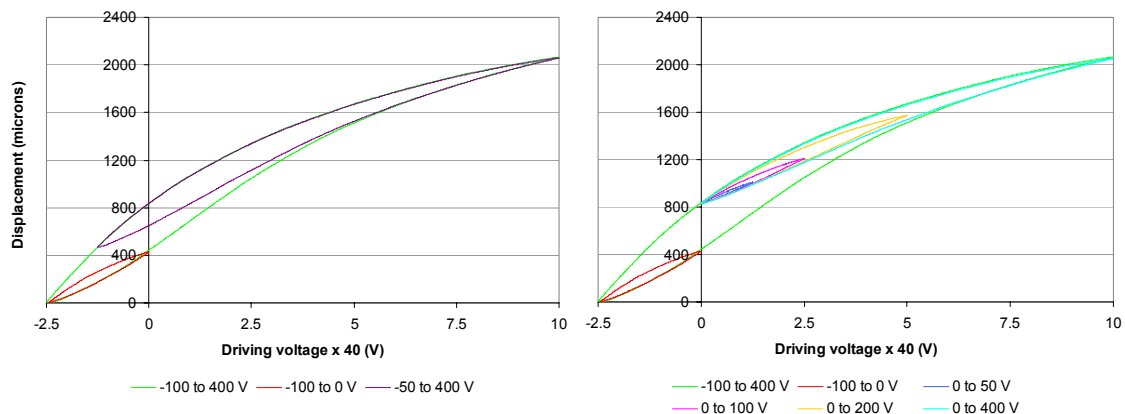


Figure 33 – Hysteresis loops for different voltage ranges and start/end levels (–100 V minimum to +400 V maximum) for the planar bimorph actuator.

3.2.3 Time Dependence

Measurements were also carried out on the time dependence of the free deflection measurements. Both (i) the time lapse between the application of the driving voltage and the measurement being made and (ii) the size of the voltage increments were found to affect the overall displacement. Results from a series of tests are shown in Figure 34. Voltage steps were varied from 0.2 V to 20 V and measurement pause times from 10 ms to 10 s. The two factors were found not to interact with each other, i.e. changes made to one affected the final displacement in a prescribed way regardless of the level of the other factor. Figure 34 shows a family of curves, calculated from the separate assessments, which map the total actuator displacement for any combination of voltage and time interval within the respective ranges studied. This was proven by calculating the time intervals required to give a displacement level of 1925 μm for a series of voltage intervals and running tests on the combinations established. The data are shown in Figure 35 and show equivalence for all the test combinations to within 10 μm .

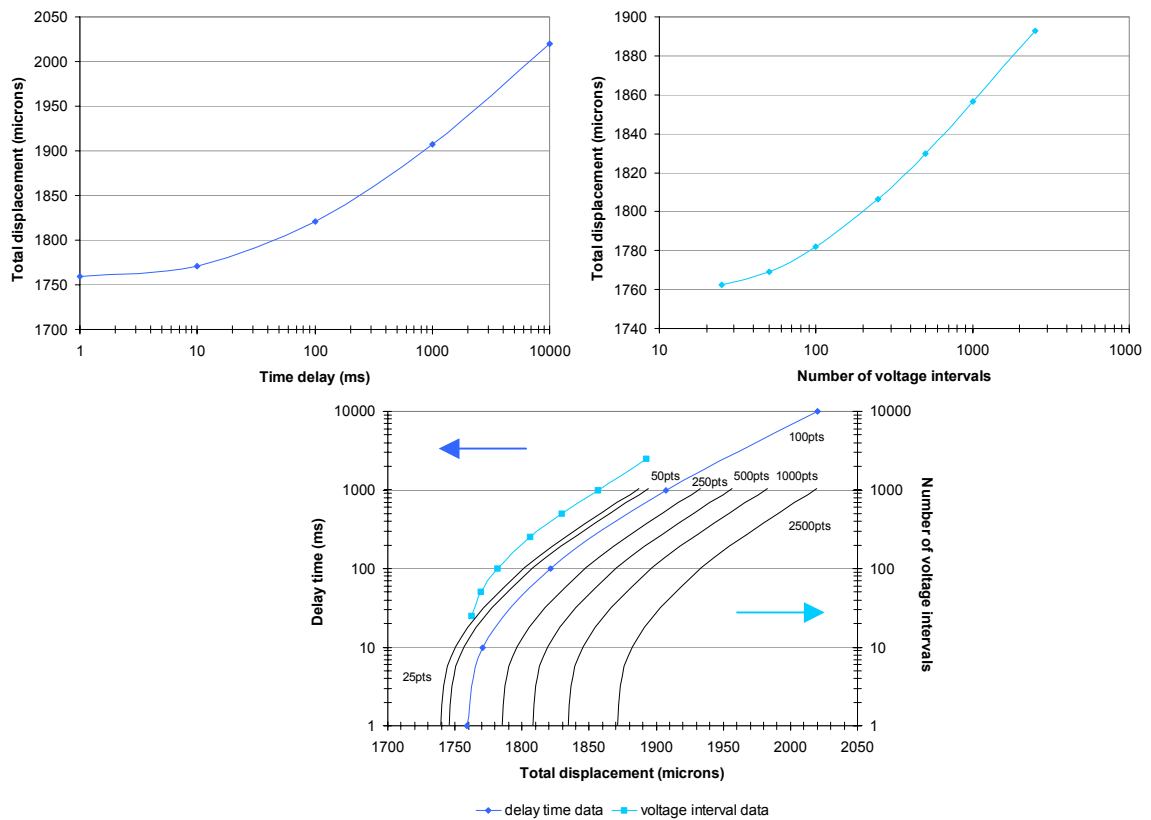


Figure 34 – Free deflection measurements for changes in time delay (upper left), changes in voltage intervals (upper right) and curves describing the output based on combinations of the two factors.

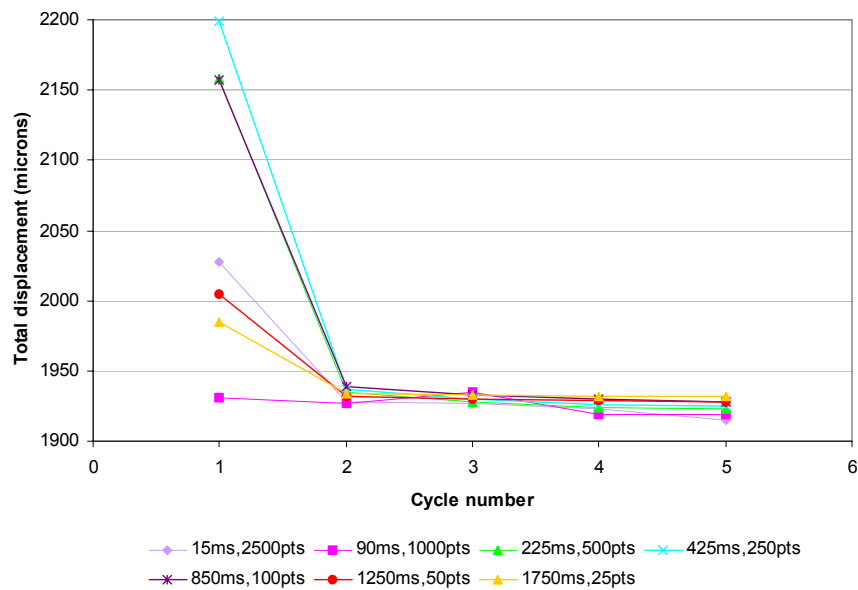


Figure 35 – Equivalent measurements using different time and voltage intervals.

The effect of relaxation occurring during potential pauses in the measurement procedure was also checked for time intervals in between identical tests. Five voltage loop experiments were run with 2, 10 and 50 minute delays between them. Figure 36 shows the results from these tests. Whilst the initial loop displacement is different, as was seen previously, immediately subsequent loops show identical deflections regardless of the pause between the last loop of the previous experiment and the first loop of the next.

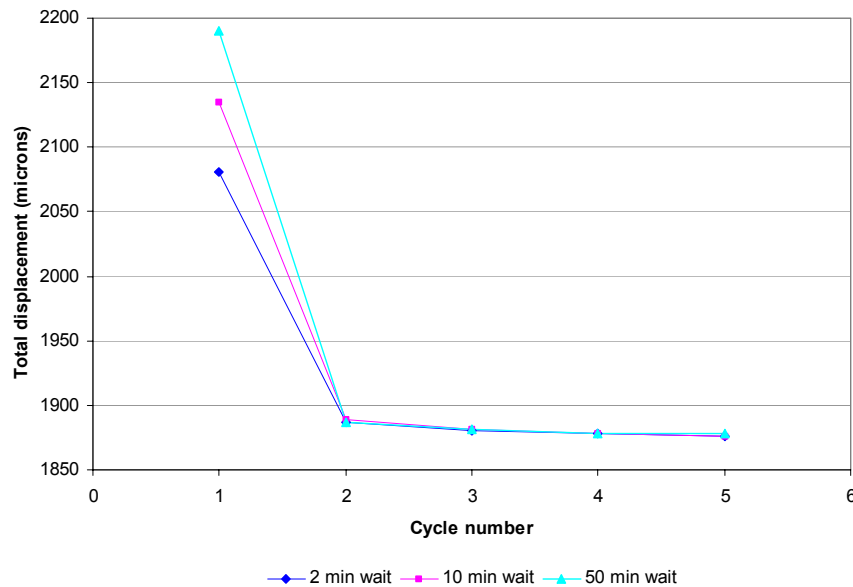


Figure 36 – Effect of increasing the time delay between series of voltage cycles.

These time dependencies affect the displacement data obtained and consequently the corresponding force determined, so for accurate predictions of performance, the actuator must be characterised using the same drive voltage and timing conditions which are intended to be applied in-service.

3.3 PARTIALLY BLOCKED MEASUREMENTS

Full range blocking force measurements were made on all three actuators using appropriately dimensioned cantilevers to cover the full range of force and displacement expected. The typical experimental set-up is shown in Figure 37.

The repeatability of the partially blocked measurements was checked by performing a large number of consecutive cycles (10 in this case) with the actuator motion being opposed by different cantilever stiffnesses for cantilever lengths of 40 mm and 220 mm, the two extremes of the range of lengths available. These results are presented in Figure 38 and show a similar trend to the free displacement repeatability results. *The first cycle is always different to subsequent repeats as the most recent pause time, electric field and mechanical history is then the same for each loop.* All the loops analysed were therefore 3rd or 5th repeats to enable the device response to stabilise. Also as an aim to improve consistency, it was decided to run free displacement ‘refresh’ cycles in between each cantilever height/partially blocked measurement.

This ensured that the immediate actuator history prior to each partially blocked measurement was identical.

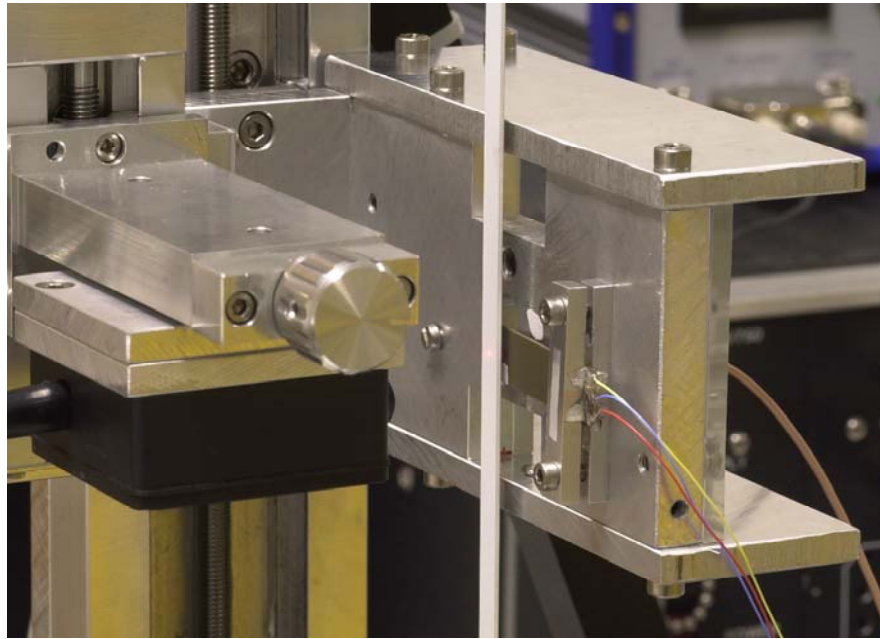


Figure 37 – Test set-up for partially blocked measurements: planar bimorph shown.

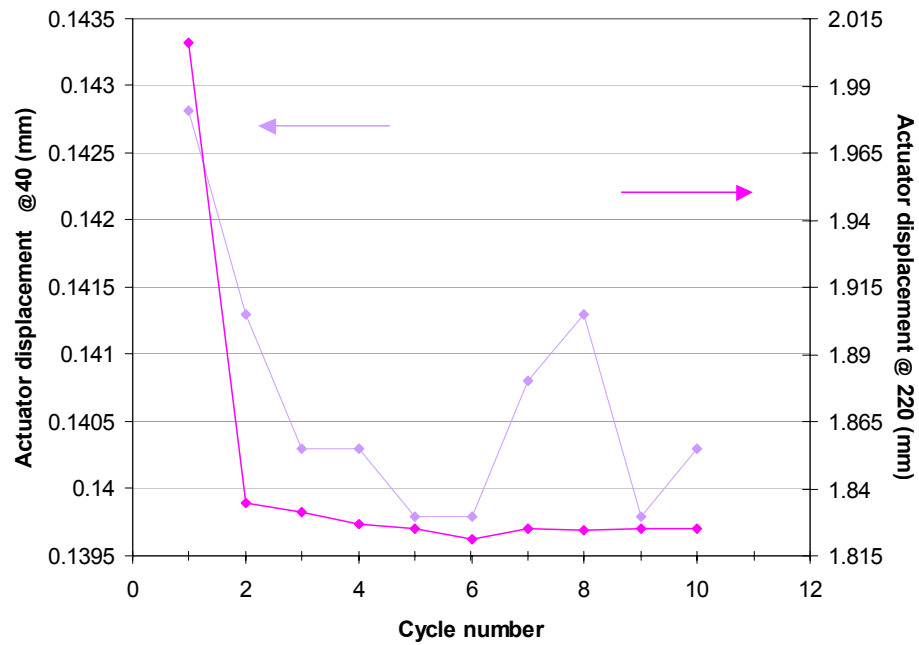


Figure 38 – Partially blocked repeatability measurements for planar bimorph actuator.

The effect of measuring the displacement from above, below and exactly on the point of contact between the actuator and cantilever was checked. These different regions of curvature are covered by different equations and experience different deflection magnitudes. The results are given in Figure 39 showing both the raw data at the different positions and the data once corrected (using the beam equations to calculate the displacement at the point of contact from that measured at the other two positions).

Figure 39 shows (i) the overall consistency achieved in the measured and corrected displacements and (ii) the disparity between the data obtained from directly over the contact point and that from either of the other two zones (some evidence was found to suggest an issue with the laser measuring in line with a high voltage region).

All partial blocking force measurements were subsequently made from 10 mm above the contact point. This ensured consistency of the data and, since deflections at this point were greater, minimised measurement errors. The force at the point of contact was calculated from the displacements measured 10 mm distance from the load contact point, enabling the displacement at the point of contact to be calculated in turn.

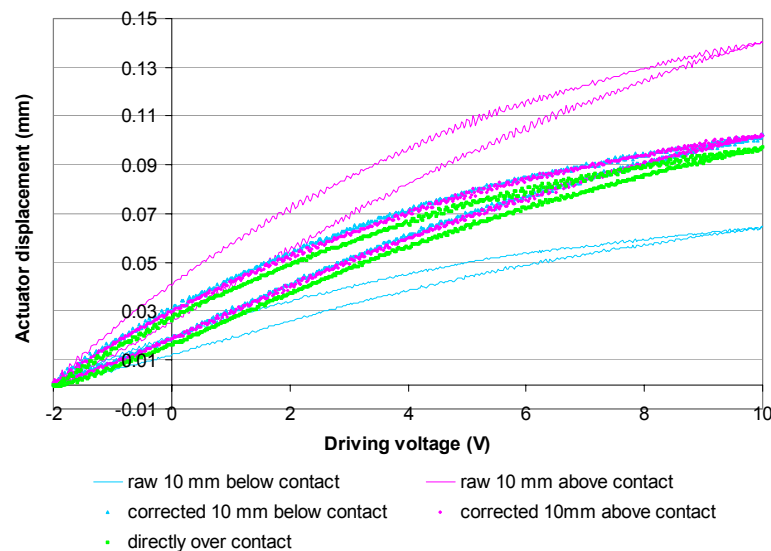


Figure 39 – Effect of measurement position with respect to the loading point of contact (40 mm cantilever) for the planar bimorph, both raw and corrected data.

Figure 40 shows the displacement with driving voltage for full cycles over a range of cantilever stiffnesses. This clearly shows a greater hysteresis for lower resistance actuation, i.e. larger loop for free displacement measurements.

Figure 41 shows the same data as in Figure 40 but with equivalent force calculated for the actuation displacements measured. The slope of each line corresponds to the stiffness of the cantilever resisting the actuator's motion.

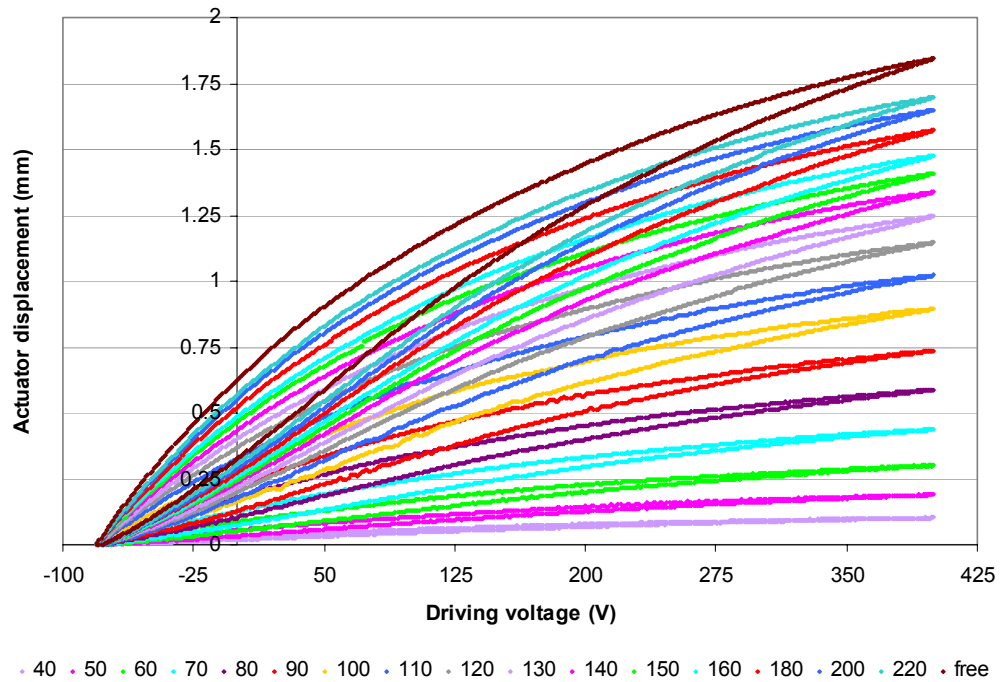


Figure 40 – Partially blocked displacement loops with driving voltage for cantilever lengths ranging from 40 mm to 220 mm, as well as free displacement, (480 pts 100 ms delay) for the planar bimorph actuator using a 1 x 3 mm cantilever.

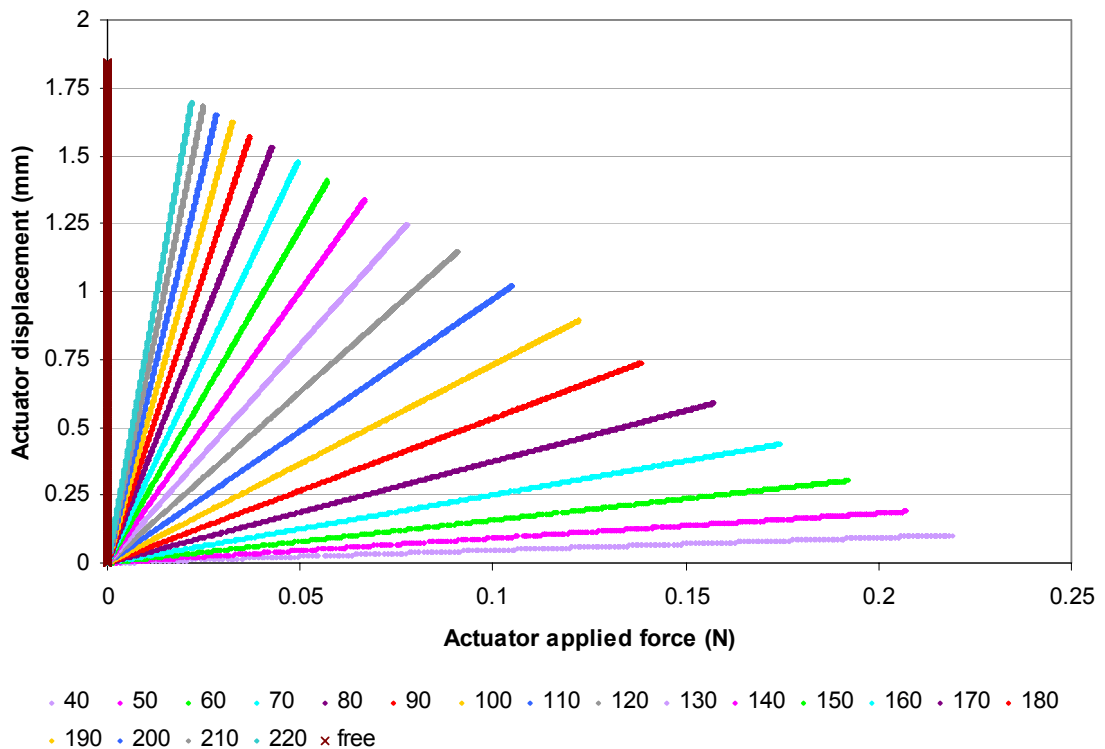


Figure 41 – Partially blocked force-displacement for cantilever lengths ranging from 40 mm to 220 mm, as well as free displacement, (480 pts 100 ms delay) for the planar bimorph actuator using a 1 x 3 mm cantilever.

The results for the full force-deflection characterisation of the actuators are given in Figures 42-44. Each set of measurements was carried out twice as a simple check of repeatability. These show a linear force-displacement response for the planar bimorph despite its significantly non-linear displacement-voltage response, whilst the stress-biased unimorph and multilayer bimorph both showed greater levels of non-linearity in their force-displacement responses, with the force increasing with increasing electric field and stress. The load levels reached in the trials were too low to make sensible in-situ compliance measurements and corrections, but these were expected to be below $5\text{ }\mu\text{m}$ based on spot measurements made during the highest load loop and from the compliance curves in Figure 19. It is also interesting to note that the high blocking force expected for the stress-biased unimorph was not achieved although this may be related to the actuator clamping/support or due to load application (narrow strip of steel bonded to the front face electrode to distribute load from the bead over a larger area to protect the piezoceramic from damage), but was observed to be similarly below expectations in previous studies [4].

Errors associated with a nominal $5\text{ }\mu\text{m}$ error in displacement for each of the actuators are given in Figure 45. Additional errors include uncertainty in the cantilever modulus, length and dimensions which are not considered in this example.

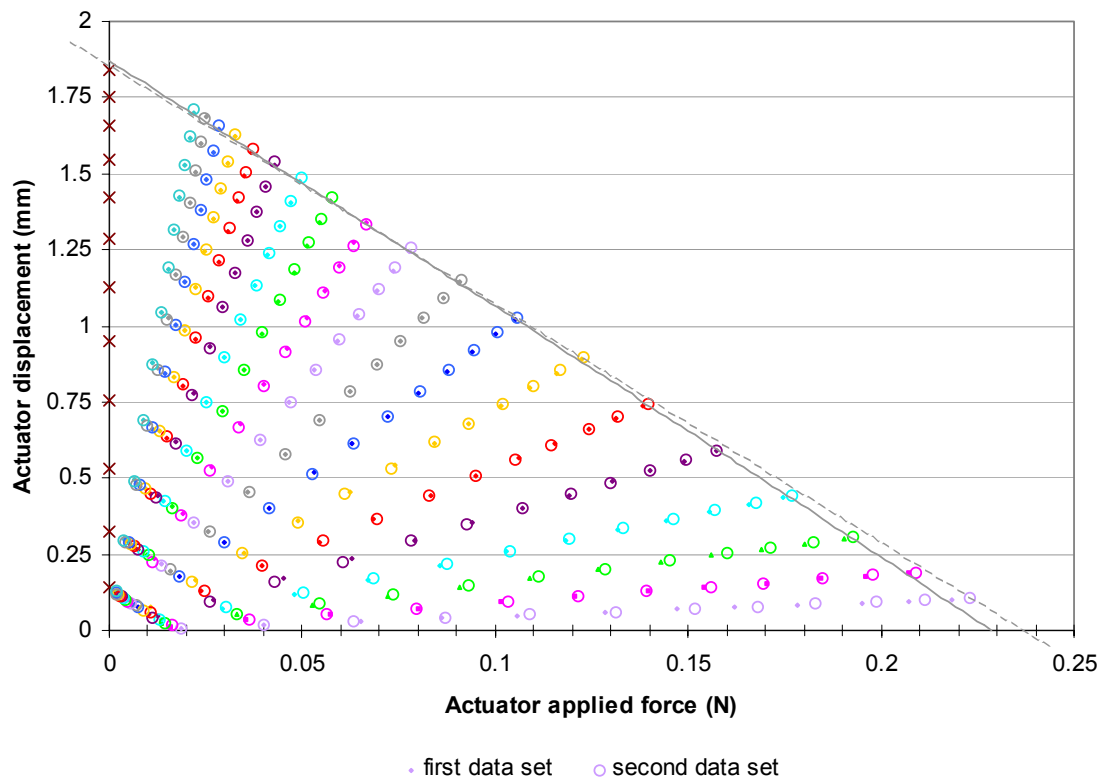


Figure 42 – Planar bimorph: force-displacement map (960 pts/loop, 100 ms delay, -80 V to $+400\text{ V}$, $1 \times 3\text{ mm}$ cantilever); dotted/solid lines correspond to linear/ 2^{nd} order polynomial.

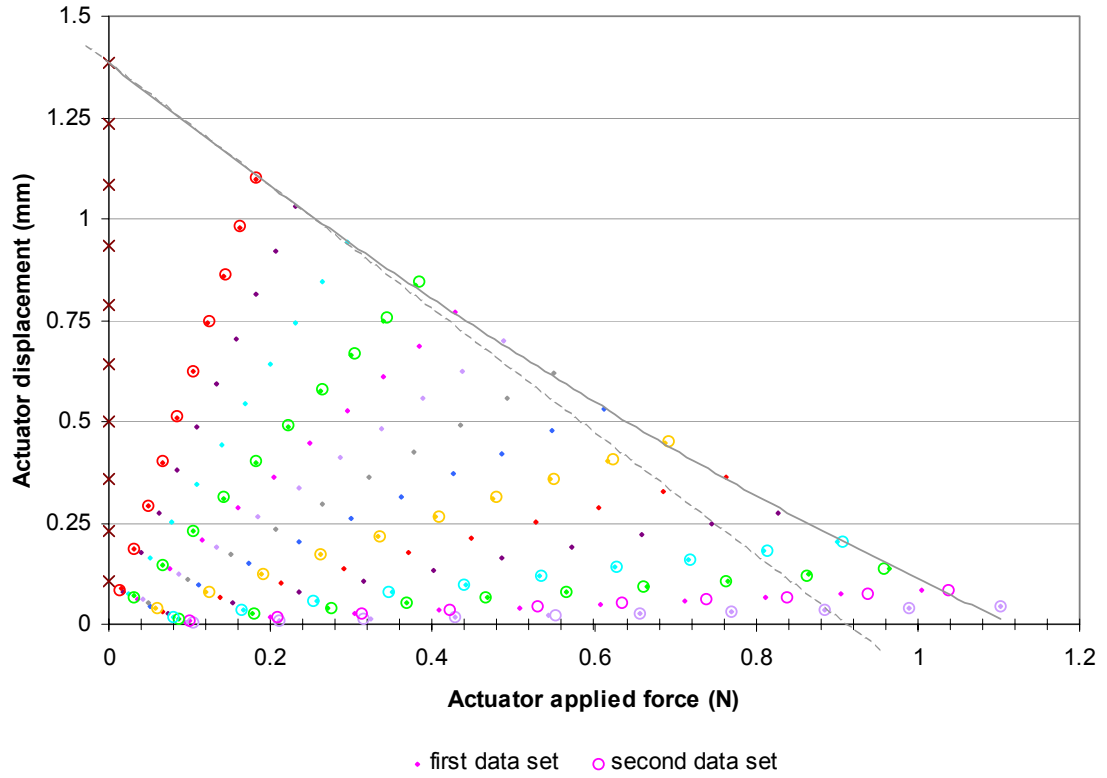


Figure 43 – Multilayer bimorph: force-displacement map (960 pts/loop 40 ms delay, 0 V to +60 V, 2 x 4 mm cantilever); dotted/solid lines correspond to linear/2nd order polynomial.

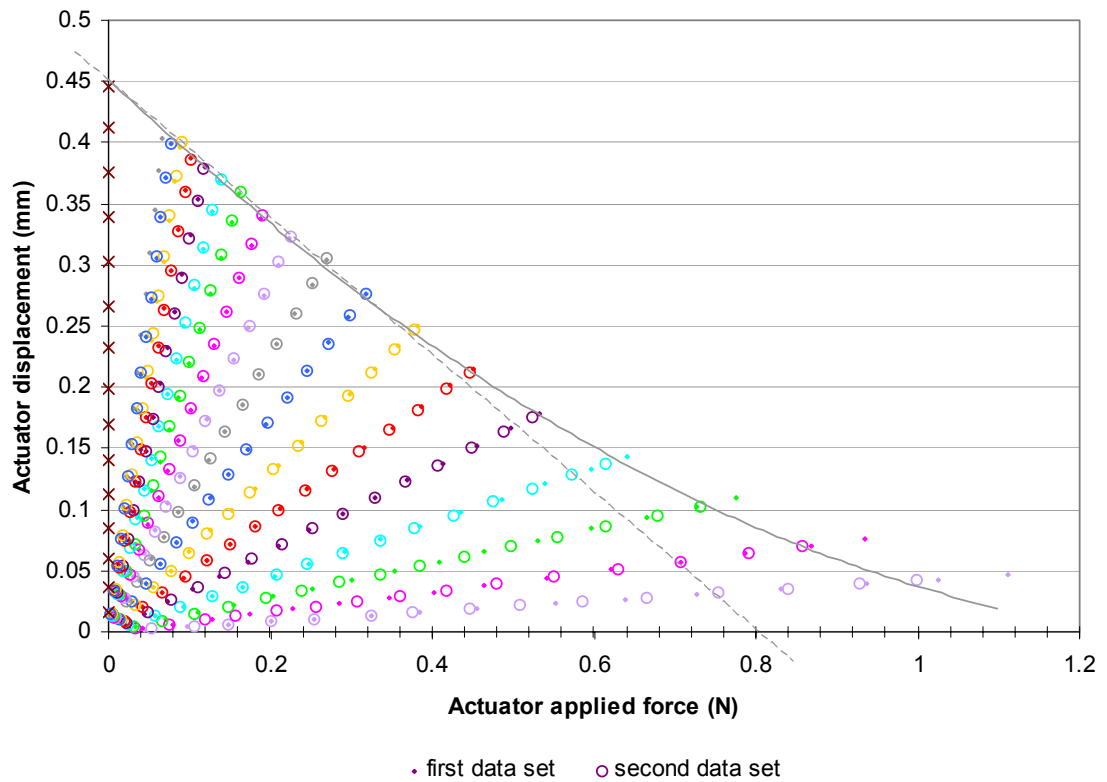


Figure 44 – Stress-biased unimorph: force-displacement map (1200 pts/loop, 50 ms delay, +400 V to –200 V, 2 x 4 mm beam); dotted/solid lines relate to linear/2nd order polynomial.

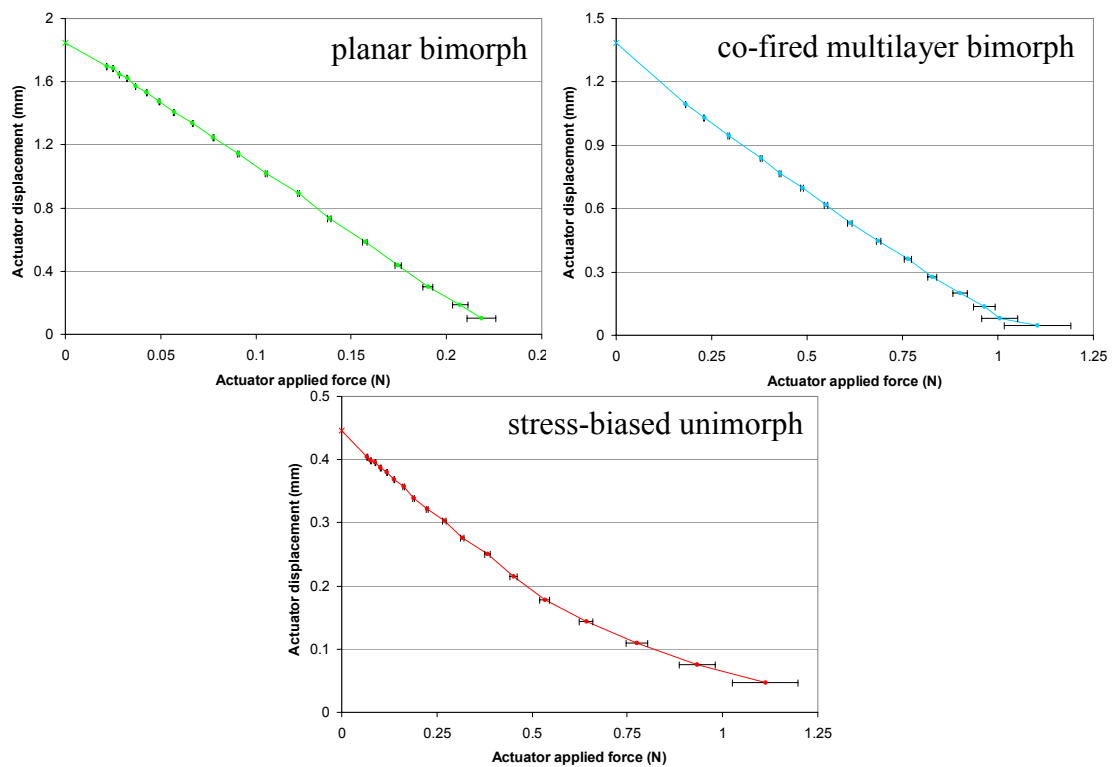


Figure 45 – Errors in calculated force associated with a 5 μm measured displacement error, corresponding errors in calculated displacement at contact point are 3.5 – 5 μm .

The scatter in nominally identical points was originally assumed to be caused by different levels of contact between the actuator and cantilever. In order to assess this, an experiment was run where the contact threshold was progressively changed for successive loops (with a ‘refresh’ free displacement loop in between each). The results are presented in Figure 46. Perhaps counterintuitively, this shows no measurable difference in the actuator performance between pre-load levels. This makes the experiment easier to conduct as knowing the precise point of contact is unnecessary. In fact, the only requirement seems to be that contact is maintained throughout the test, as the threshold set closest to the uncontacted value shows some mechanical relaxation resulting in a lack of contact at the start of the loop and a lower total displacement as a result.

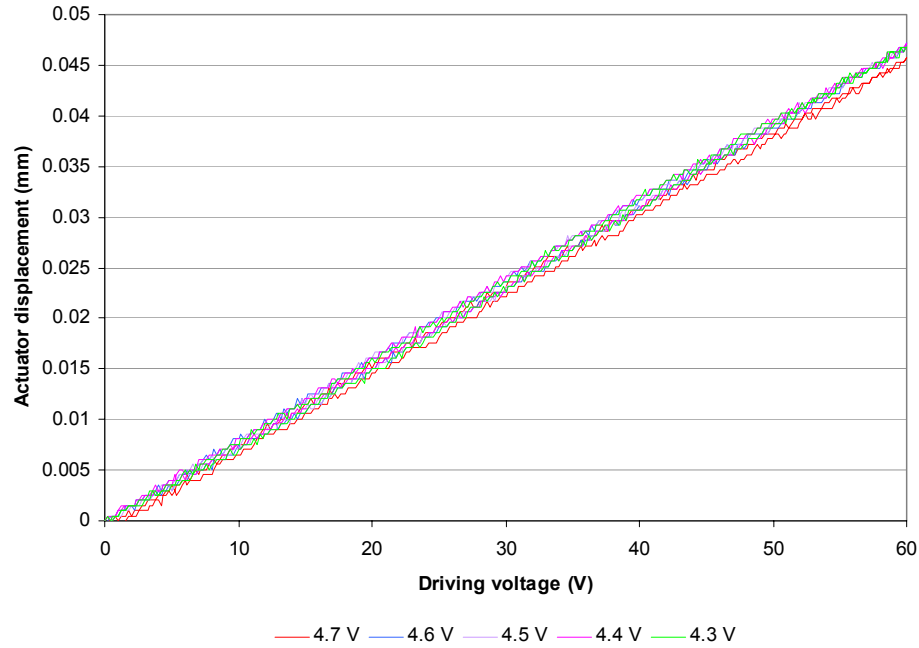


Figure 46 – Effect of contact threshold level/pre-load on measured displacement for 40 mm cantilever and multilayer bimorph, uncontacted sensor reading 4.85 V.

4 CONCLUSIONS

The detailed design and testing of novel instrumentation to characterise the static blocking force of piezoelectric macroactuators has been demonstrated. The equipment was successfully employed to map the full force-deflection response of several commercially available high-displacement/low-force actuators, making this instrument of equal interest to both actuator manufacturer's and end-users. The system is of particular value since the measurements are made under conditions which closely mimic those experienced in industrial applications and can be adjusted to accommodate different actuator sizes and architectures. The equipment also has potential for standardisation of the measurement and definition of these key characterisation parameters, making manufacturers' specifications comparable, from one supplier to another. This issue is highlighted by the lack of correlation between the supplied values and those measured here.

The actuator force-deflection data produced thus far substantiates the notion that the assumption of linearity for the actuator stiffness or force/displacement response may be incorrect in many cases due to the non-linear effects produced by mechanical stress, electric field and temperature on the material, preceeding depoling levels. We see a decrease in compliance with increasing load and field in the examples studied here, which is attributed to either non-linearity in the system compliance, specific non-linearity in the response of the individual actuator clamping arrangements (as Poisson's constraints change) or intrinsic non-linear response of the piezoelectric.

The free displacement data show a greater degree of hysteresis with increasing voltage [9], as well as with lower resistance actuation, i.e. larger loops observed for free displacement measurements than for partially blocked.

The intuitive relationship expected between voltage steps/size and interval pause time, i.e. overall displacement related to the total test time (given by number of steps multiplied by the pause time), was found not to be applicable. The two factors were observed to independently affect the total cycle displacement.

4.1 EXPERIMENTAL ISSUES

The design of the system and subsequent testing uncovered a number of issues which need to be considered when performing these measurements. Several of these are summarised below:

- For the beam calculations to apply, the beam must satisfy rigorous conditions based on dimensions, properties and linearity.
- The cantilever clamping at the fixed support is very important for the beam theory to be valid. Ideally, clamping should act perpendicular to the direction of load application to ensure sufficient rigidity at the root of the cantilever.
- Gripping or mounting conditions change actuator performance characteristics, e.g. unclamped actuator length, stiffness of clamping material, clamp profiles/chamfers, clamping load/pressure etc. These should be described and reported with the results for each test.
- As a result of the cubed dependence of the effective cantilever length on load calculations, it is very important to know this length precisely, particularly for shorter cantilever lengths where the effect on the calculated load is much more pronounced.
- Free displacement ‘refresh’ cycles should be carried out between partially blocked measurements to ensure the most recent mechanical and electrical history of the actuator is identical in each case, leading to more consistent data.
- The actuator should be subjected to more than one voltage cycle prior to cycle capture/analysis to guarantee representative and reproducible data. The first loop is always dissimilar to subsequent cycles.
- It is necessary to ensure good contact between the cantilever beam and actuator at the datum position (zero deflection/load) and throughout the partially blocked measurement cycles.
- The force is determined indirectly through the displacement measurement therefore the fully blocked (no displacement) value cannot be obtained. It would be difficult to produce a resistance of infinite stiffness as required, only a point very close to fully blocked can be achieved e.g. corresponding to 5 μm displacement.
- The forces determined are very sensitive to the displacement measurements, even to the nearest micron; accurate calibration of the displacement devices is essential. For this reason, it may be necessary to dissociate the force and deflection measurements to eliminate the error dependence.
- It is important to ensure the test system has sufficient stiffness to withstand typical actuator blocking forces. Slack in leadscrews and bearings should be eliminated and bending in other parts minimised and characterised/corrected for to improve the accuracy and reproducibility of the measurements.
- System should be checked with materials of known response, e.g. cantilever of known modulus, to ensure appropriate performance can be achieved.
- Poor reflectivity may require additional surface preparation of the cantilever or actuator for the laser displacement sensor to function properly.

- When displaced the cantilever is no longer perpendicular to the laser beam, but the angle is small in all cases and so has little effect on the measurements.
- It is important to reduce vibration noise on optical displacement measurements so damping feet should be employed.

4.2 FUTURE WORK

Future work will focus on reducing the compliance of the system and redesigning where necessary. This will then facilitate more challenging measurements on actuators which can produce higher loads e.g. cymbal or moonie. This stiffness redesign could also be used to extend the system to quasi-static ($<100\text{Hz}$) or dynamic measurements.

A comparison of the three principle routes to generate force-displacement data, i.e. constant force, constant displacement and constant stiffness, to determine whether these yield equivalent data or whether there are obvious differences between them.

Further investigation of several interesting aspects noted in this study:

- Loading configuration – dependence of the actuator applied force on the geometry or area distribution or location of the loading point on the actuator.
- Time dependence – definite time dependence of force-deflection properties, further study to determine whether these are dependent on the piezoceramic, peripheral materials or architecture of the actuator.
- Non-linearity – revisit bulk materials and follow through to a novel actuator designed with the same material to investigate the effect of material and/or structure.
- Importance of contact point or datum deflection/load – thus far seems not to be important but more work needs to be carried out to confirm this.

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