Proc. of SPIE Vol. 1167, Precision Engineering and Optomechanics, ed. D Vukobratovich (Nov 1989) Copyright SPIE

An instrument for generation and control of sub-micron motion.

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1. ABSTRACT

This paper describes a new class of instruments that can reliably produce and control small motions. The instrument is intrinsically stiff, stable, athermal and adaptable to very harsh environments. Although the principles may be scaled for large motions, their current application is optimized for the optical wavelength range, a micron to an angstrom or smaller. The paper also describes the "calibrated elasticity" principles used in the development of the instruments.

2. INTRODUCTION

In precision instruments, especially optical instruments, it is often desirable to be able to adjust the position of the various elements of the instrument to be able to achieve the best possible performance. Similarly, it is often desirable to accurately position a specimen for observation by a microscope, X-ray source or other instrument. This involves making very small adjustments in the instrument. In precision instruments these motions may range from a fraction of a millimeter to a micrometer and perhaps less; in optical systems one may be interested in making adjustments that are a small fraction of the wavelength of the light for which the optical system is designed, requiring motions with magnitudes from a micrometer to an angstrom.

These motions and adjustments are currently being made with levers, micrometers, compound screws and piezoelectric crystals. Micrometers are limited to making the larger motions in the range of interest because the technology of making precision screw threads limits the thread pitch to about forty threads to the inch. (Eighty to 100 threads to the inch thread forms are available but are problematical in manufacture, idiosyncratic in assembly and are therefore generally not widely used.) One turn of such a micrometer produces about 635 micrometers of motion (.025 inches). If one can reliably produce a half-degree angular motion of the thimble of the micrometer he may then reliably produce a .88 micrometer motion. Use of optical encoders to read the angular position may improve this some but the increased precision is soon overwhelmed by the backlash and other imperfections in the mated threads and supporting structure of the micrometer. A precision micrometer is limited to resolutions and motions larger than about .50 micrometer.

To achieve smaller resolutions than this value the micrometer is often combined with some form of lever which further provides some reduction of the motion of the micrometer. Levers are limited in their ability to provide high reduction ratios in a small space. Control of the ratio requires accurate location of the fulcrum with respect to the acting ends of the lever. In high ratio levers it may be difficult to adequately control the dimension of the short end of the lever when it is less than some critical value, say .050 to .010 inches. To illustrate the space problems associated with high mechanical advantage levers assume that one wishes to produce a lever with a mechanical advantage of 10,000 and the short arm of the lever was .010 inches from the fulcrum, the long arm would need to be 10,000 x .010 = 100 inches (8.3 feet) long!

Compound screws offer much higher average mechanical advantages than single screws (i.e., micrometers) but imperfections in the manufacture of the screw threads lead to cyclic run-out errors. The cyclic run-out errors in compound screw threads generate an oscillating axial component of motion superimposed upon the desired linear translation of the spindle. The manufacturing limitations in compound screw threads limit their improvement over conventional screw threads to less than one order of magnitude.

Piezoelectric devices have been employed to generate small motions. These devices are quartz crystal which are exposed to an electrostatic field across them. If the axes of the crystal lattice are appropriately oriented with respect to the electric field a small (but perceptible) change occurs in the dimensions of the crystal lattice. This results in a dimensional change for the whole quartz crystal. These devices can produce motions down to very small quantities (an angstrom or less) but suffer sizeable hysteresis effects (five to ten percent of full scale) which prevent the device from returning to the same point every time the voltage is adjusted to the apparently appropriate value. To overcome the hysteresis effects piezoelectric devices are often fitted with optical encoders or capacitance micrometers to measure their motion (position) and the voltage is then adjusted to obtain the desired reading. In these applications the lower limit of accuracy (about .001 micrometer) is controlled by the accuracy with which optical encoders or capacitance micrometers may be produced. The stability of these devices is further limited by the thermal coefficients of expansion, permitivity and other physical properties of the materials of which the sensors and associated electronics are manufactured. Thermal drift in such systems may be as great as .015 micrometer per degree Celsius.

Accordingly, the object of the present discussion is to describe a device for accurately and repeatedly making small motions (position adjustments) and to provide features that allow the attachment of the resulting motion transducer into instruments and systems of all kinds that require control of small motions. The device is also mechanically stiff, thermally stable and repeatable with high precision.

3. PRINCIPLES OF CALIBRATED ELASTICITY DEVICES

The instruments discussed in this paper take advantage of the elastic properties of an extended body to generate small motions. Because of their dependence on elastic properties they are called "calibrated elasticity devices." When an elastic body is loaded with forces or deformed by displacements it will experience a change of shape throughout its volume. Each body with a specific applied load or displacement will have a specific deformed shape. This makes it possible to define a set of deformations for the body associated with the applied loads or deflections.

Furthermore, even though the applied loads or displacements may be very large, the deformations in other parts of the body, away from the points where loads or displacements are applied, may be very small. This fact allows one to design an elastic body that has the desired ratio between the applied loads or displacements at one point and the deformations at another point on the body. The performance of such an elastic body will have two performance parameters called "Effectiveness Ratios": one Effectiveness Ratio for applied loads, and another for applied displacements. The Effectiveness Ratio for an applied load (ER_L) is the response deformation divided by the applied load,

$$ER_L = \frac{\text{response deformation}}{\text{applied load}} .$$

Since the response deformation may be either a translational or a rotational quantity and the applied load may be either a force or a moment, the ER_L may have a variety of units.

The Effectiveness Ratio for an applied displacement (ER_D) is the response deformation divided by the applied displacement,

$$ER_D = \frac{\text{response deformation}}{\text{applied displacement}} .$$

ER_D will also have a variety of units depending upon whether the deformations and displacements are translational or rotational quantities.

In one of the simplest cases ER_D will be based upon translations and the resulting quantity may be thought of as a "mechanical advantage" similar to those used to define the effect of a rigid lever system. However, a lever system is reciprocal in that it may be operated from either end, whereas, the elastic body is not necessarily reciprocal, as will be discussed.

Consider a simple cantilever beam, Figure 1. An applied displacement at the free end (1) will produce a small elastic deformation at another point (2) near the fixed end. The ratio of these two values will be the ER_D for this configuration

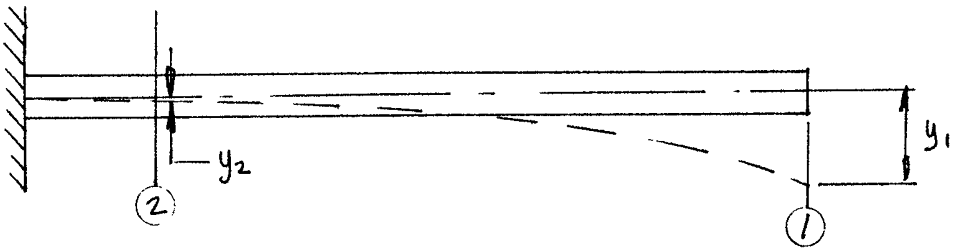

Figure 1. displacements in an elastic cantilever beam.

of elastic body. However, if one applies at point (2) a displacement equal to the original elastic deformation at that point and observes the resulting elastic deformation at the free end (1) he will see that this value is not equal to the original displacement applied to the free end above as one might expect using the reciprocal principles of the lever. Nor are the ER_Ds of these two observations reciprocals of each other as one might expect from the performance of a lever. The reason for this is simple, elastic bodies do not observe the geometric laws of rigid bodies. The shapes of the elastic curves of the cantilever beams in these two cases are not similar and therefore the ratios will not be reciprocal.

A similar analysis of the ER_Ls (applying loads in stead of displacements) for elastic cantilever beams will lead to the same conclusion for force or moment loaded elastic bodies. Therefore, unlike rigid bodies, reciprocally loaded elastic bodies do not have similar deflected shapes and their ER_Ds and ER_Ls will not have reciprocal values.

The cantilever beam may be a useful elastic body for transducer applications since it extends the Effectiveness Ratio somewhat beyond that available from a lever system of about the same size. However, the range of the Effectiveness Ratios may be even more greatly extended by the use of two and three dimensional bodies.

Consider the solid body in Figure 2. In addition to the proportions shown it also has thickness through the plane of the

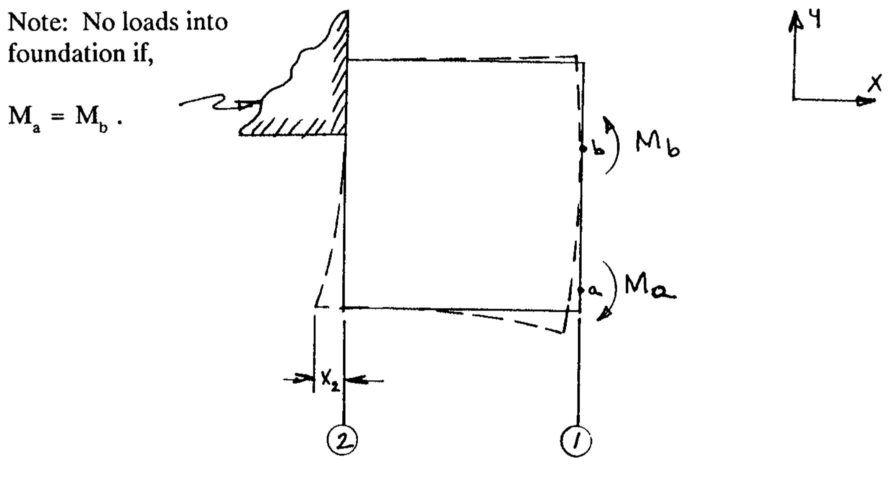

Figure 2. Elastic deformations in a solid body.

Figure. If one fixes the body to a foundation it will be unable to move freely in space. One may now apply a force or moment to the body at point (a) thereby causing the body to deform elastically. In this case it will be desirable to apply an equal and opposite force or moment to the area (b) in order to prevent reaction loads from entering the foundation. The load at (b) will also influence the distribution of the deformations of the elastic body. The resulting deformations of the body are shown as dashed lines. As can be seen, one may select from a variety of response quantities depending upon the location on the surface that one decides to use. The corner on surface (2) of the body is a convenient location to select. Any object brought to rest on this corner of surface (2) will experience a motion equal to the elastic deformation caused at that point by the loading condition.

Each point on the elastic body will experience a maximum of six simultaneous motions; three translations (parallel to the x, y and z axes) and three rotations (around the x, y and z axes). If one is interested in selecting only one of these components of motion at, say, the corner of surface (2), he can incorporate in the elastic body a system of flexures which are designed to transmit only the desired component of motion to another portion of the body.

These flexures may be designed to either minimize or exaggerate the rotational components of motion. Many arrangements of output flexural systems are possible.

Also the proportions of the elastic body may be adjusted to achieve a great range of Effectiveness Ratios.

4. AN ACTUATOR USING CALIBRATED ELASTICITY PRINCIPLES

Figure 3 shows the design for a simple manually operated actuator using the above principles. In Figure 3 we have

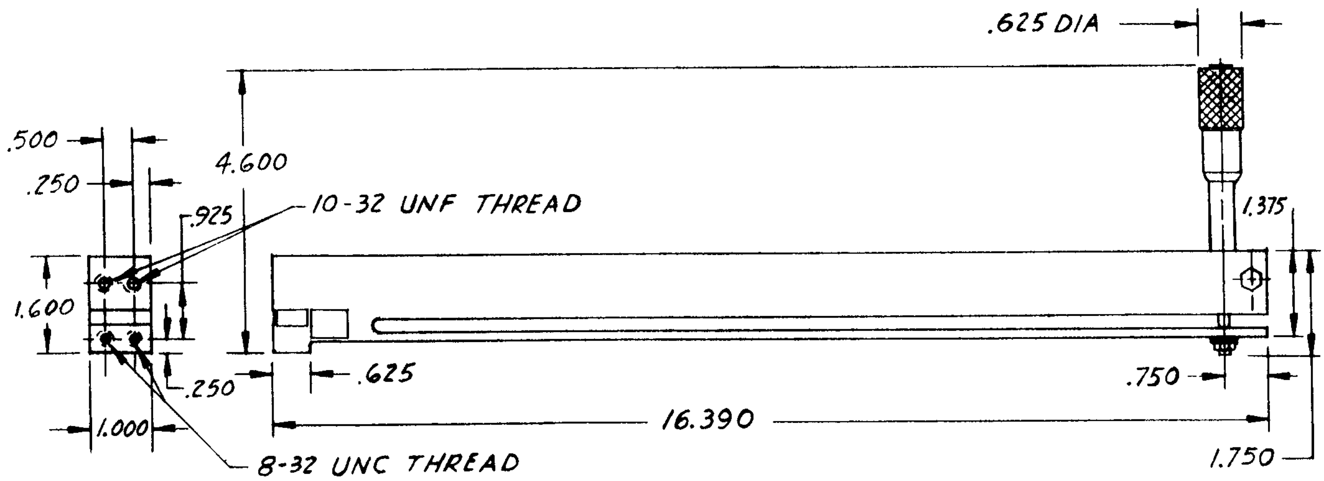

Figure 3. A one micron range, one angstrom resolution actuator using calibrated elasticity principles.

extended the elastic body to the left to incorporate flexures for selecting the desired components of motion. These flexures support a block with two #8-32 tapped holes at the output surface. These screws attach whatever device will be moved. The other screw holes (tapped #10-32) are used to secure the whole assembly to the frame or foundation of the surrounding instrument. The flexure system as described in Figure 3 selects the horizontal translational component of motion in the elastic transducer body and transmits it to the block containing the #8-32 tapped holes. This translational component is transmitted directly by the flexural system while attenuating the rotational component of the elastic body motion by more than two orders of magnitude, thereby imparting a nearly pure translational motion to the movable block. The magnitude of the rotational component of motion is controlled by the design of the flexure system.

We may further extend the elastic body to facilitate the loading process. In Figure 3 the rectangular body has also been extended to the right and a slot has been cut into the extension. The slot creates two cantilever beams; a thick one and a thin one. Near the free end of the thick beam it has been bored with a hole to receive a standard micrometer. The tip of the micrometer is brought into contact against the surface of the thin beam. Thus, when the micrometer is extended, the slot width is locally increased. This slot width increase is accommodated by elastic deformations of the two cantilever beams. The resulting beam deformations are sustained by reaction loads at their fixed ends. The reaction loads thus result in the balanced loading of the elastic body. It can now be seen that by dialing in a value on the micrometer one may achieve a desired loading on the elastic body and a desired small deflection at the mounting block.

Hysteresis effects in the elastic body will be due to plastic yielding of the material. The onset of plastic yielding in a material is characterized by the stress level at which a specific amount of plastic strain is observable in test specimens. If the stress levels in the elastic body are maintained below the perceptible yielding points of the material, the transducer will exhibit imperceptible hysteresis effects. Since many engineering materials have sizeable Yield Point stresses (stress where plastic strain = .002) and micro-yield point stresses (stress where plastic strain = .000001) many materials may be used for construction of transducers with very small hysteresis effects. Some materials appear to have a sizable region where no plastic strain appears; these materials may produce transducers with no hysteresis effects at all when the stresses do not exceed the threshold level.

The stiffness of the transducer will be controlled by the stiffness of the flexures. Actual stiffness depends upon the details of mechanism design but flexure systems as illustrated here may be designed to be as stiff as the element or device whose motion is being controlled. When no mechanism is used and the device to be controlled is mounted directly to the elastic body, the transducer is even stiffer than when output flexures are used.

The thermal sensitivity of the transducer system has three sources: thermal expansion of the transducer body, thermo-elastic changes in the transducer body and thermal effects in the loading mechanism. In a displacement loaded transducer, the thermal sensitivity may be reduced nearly to zero if the transducer body and the loading mechanism are made of identical materials and if the foundation mounting point and the movable surface (refer to Figure 3) are co-planar. Since the thermal effects are proportional to length, a zero effect exists at the zero deflection position. As displacement loads are applied to the transducer body, the movable surface is displaced from its original plane by some small amount. This small amount will be subject to the effects of thermal expansion which, for most materials is in the fifth or sixth decimal place of the amount of deformation of the movable surface. If the loading mechanism is of a different material than the transducer body the loading mechanism will contribute some thermal sensitivity to the system due to the difference in the thermal expansion coefficients and the thermo-elastic coefficients in the two materials. These effects, which may be thought of as occurring directly at the loading mechanism, will be proportioned down according to the magnitude of the Effectiveness Ratio. If the Effectiveness Ratio is small (say, .0001 or .000001) then the resulting effect of these effects at the movable surface will be similarly small (.0001 or .000001) of the deformation motion at the movable surface.

Thermo-elastic effects are similarly small. For many alloys of aluminum and steel the change in Young's Modulus with temperature is about two percent in 100 degrees Celsius (near room temperature). This represents a shift of .0002 of full scale per degree Celsius. If better thermo-elastic stability is desired the transducer body may be constructed of specialty alloys which have very small thermo-elastic coefficients. This becomes important only in force actuated devices (as opposed to displacement actuated devices which may be independent of thermo-elastic effects in the materials of construction).

Similar analyses may be made for the thermal sensitivities in both displacement and force or moment loaded transducer systems. As a result the transducer systems that this paper describes may be made to have very low sensitivities to both thermal expansion effects and thermo-elastic effects as well as other thermodynamic effects in the loading mechanism and transducer body.

The proportions of the actuator shown in Figure 3 are optimized for a transducer body made of 6061-T6 aluminum deformed by a standard one inch micrometer. This actuator produces a full range of 1.0 micron (1×10^{-6} meters) for a 1.0 inch displacement at the micrometer. Furthermore, the vernier divides the 1.0 inch displacement into 10,000 equal parts providing a resolution of 1×10^{-10} meters (one angstrom) over the full range at the output surface. This transducer is capable of supporting movable devices, assemblies, specimens and instrument packages of up to 100 pounds without additional structural support. The temperature coefficient at the output surface is less than .02 nanometer/degree K at full range and the non-linearity caused by the finite displacement of the cantilever beam is less than .2 percent of full range. The structural stiffness at the output surface is about 1,000,000 pounds/inch.

In this design the micrometer may be replaced by a voice coil, a solenoid, a pneumatic bellows or any other device that spreads the beams with a calibrated force or displacement. All of the performance parameters are design variables and a wide variety of styles, shapes, ranges, resolutions, load capacities and environmental factors may be accommodated.

The design shown in Figure 3 has been built and is currently being tested. The initial tests have corroborated the theoretical principles and analyses outlined above.

5. CONCLUSION

This paper describes a stiff, stable and repeatable transducer system capable of reliably producing small motions. Calibrated elasticity overcomes the intrinsic limitations of threaded mechanisms (micrometers and compound screws), rigid levers, cranks and linkages, and piezoelectric devices of all kinds and offers a new regime of precision motion control.