COURSE NOTES
PRINCIPLES ACTIVE IN THE DESIGN OF PRECISION MACHINES

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General

Over the years designers have evolved certain principles which are very commonly utilized when constructing precision machines. Evans has provided a history of these developments [Evans, 1989] and Slocum has created an excellent textbook on the subject [Slocum, 1992]. A landmark synthesis of these principles has been currently published by Teague and Evans [Teague, 1989]. This document is, however, available only as course notes and contains a prohibition against duplication and resale. Their important contribution is modeled after a similar document describing patterns for building construction [Alexander, 1977] and includes information from a number of texts [Jones, 1983; Braddick, 1963; Strong, 1943; Sydenham, 1980; Whitehead, 1954; Wilson, 1952; Moore, 1970; Pollard, 1929; McKeown, 1982], as well as references to individual papers. Much of this report is taken from that document, with additions taken from Slocum and contributed by myself. Wherever possible, I have tried to illustrate these principles in terms of how I would evaluate a precision machine, whether it be a machine tool, step-and-repeat camera, or measuring machine, etc. Teague and Evans call their principles "patterns". The patterns they report are as follows.

- Repeatability
- Kinematic Mounting
- Metrology Frame
- Structural Loop
- Probe Knowledge
- Error Budget
- Isolation
- Alignment Principle
- Materials Selection
- Kinematic Drives
- Energy Flow
- Symmetry

To this set I have added the following,[1]

- Resolution
- Appropriate Metric
- Broken Symmetry
- Force Control

while expanding their comments regarding the metrology frame to include, as sub-principles within this pattern, the concept of drive decoupling from the measurement and including explicitly elastic averaging as a viable option within their section on kinematic mounting.

[1]Most of these additions have been discussed with Teague and Evans.

When examining a machine to see if it is capable of a precision operation, I would normally go through this checklist mentally. I will discuss each of these principles in turn, though not in the order given above. At the end of each section regarding a principle, I will give a summary statement that can be treated as a rule. **I want to caution that following such rules should be viewed as a necessary, but not sufficient, condition for achieving proper precision or vice versa. Furthermore, I have attempted to make the principles quantitative, insofar as possible, but exact values are probably less important than conceptual compliance.**

Terminology

As discussed in the Machine Tool Task Force Report, Vol. 5 [Hocken, 1980], nearly all precision instruments/machines are composed of one or more moving elements designed to position a measurement probe or cutting tool with respect to a workpiece. It is customary, insofar as possible, to design each positioning element such that it behaves as a rigid body with five of its six degrees of freedom eliminated, then to drive the element in the remaining direction and measure its motion in that direction as accurately as possible. Normally the motion desired is a pure linear or rotary motion. Exceptions to this practice are few. Complicated forms, like gears, threads, turbine blades, and the like, are created by coordinated action of the different elements. Errors in the motion of an individual element are referred to as "parametric errors", and there are Standards dealing with the parametric errors of linear axes and rotary axes; most specifically, ANSI/ASME B5.54 [ANSI, 1991] and ANSI/ASME B89.3.4M [ANSI, 1985], as well as technical papers [Beckwith, 1991; Bryan, 1968; Bryan, 1968]. For example, a linear carriage has six degrees of freedom, five of which are constrained. Motion in the unconstrained direction may have an error which is normally referred to as a "scale error", since this motion is measured by some type of scale. Linear motions orthogonal to the intended motion direction are called "straightness errors" or "straightness motions", and rotations of the carriage are called "angular errors", usually yaw, pitch, and roll, analogous to terms used in aircraft. For rotary axes translational motions of the axis are usually called "radial error motions" and "axial error motions". Angular errors of the axis are called "tilt motions". Often a subsidiary measurement, called "face motion", is also used, since movement of the face of a spindle is different at different points and can lead to significant measurement or manufacturing errors.

Besides these parametric errors of individual axes, there are also errors in the squareness or parallelism required between these axes. Thus, for a typical three-axis machine, there are 21 degrees of freedom, six for each axis plus three squareness terms. All of the principles outlined below are applied to the reduction of these error terms, in order to ensure as perfect as possible positioning of the tool/probe with respect to the workpiece.

Appropriate Metric

Much of what is required in a precision machine can be based around the principles of Euclidian geometry. However, the scale of length is defined by man. Any precision machine must have linear and rotary scales whose accuracy with respect to national Standards can be established and rigorously maintained. Specific scales in common use on modern high-accuracy machines are line scales based on moiré or diffraction principles, laser interferometers [Hocken, 1979], and inductive transducers. I would expect to see one of the first two of these on a precision machine. This leads to the first principle.

PRINCIPLE: THE SCALES ON A PRECISION MACHINE SHOULD BE STABLE AND CALIBRATED WITH RESPECT TO INTERNATIONAL STANDARDS OF LENGTH.

Resolution

The first requirement of any precision machine built along normal principles is that it have sufficient measurement resolution in the directions of its intended motion, whether those motions are linear or rotary. In the "old days", screws with resolvers or other encoders were used for linear axes, and worm gear drives of essentially the same concept were used on rotary axes. Nowadays it is almost "de rigueur" that the measurement of the intended motion be separate from the drive mechanism. This is discussed in more detail in later sections. In any event, I would immediately examine the resolution of a machine, with the expectation that the least count of its scales be at least 10 times smaller than the stated machine precision. If this were not true, I would severely question whether the machine could meet the specified goal.

PRINCIPLE: THE RESOLUTION OF THE SCALES ON A PRECISION MACHINE SHOULD BE AT LEAST 10 TIMES THE REQUIRED MACHINE ACCURACY.

Repeatability

Repeatability is a premier requirement in a precision machine. There have been many discussions regarding repeatability and its role going back over a century. The basic physics has been expressed in many ways, but essentially the dominant concept now is one of determinism. To quote Donaldson [Donaldson, 1972; Donaldson, 1973], "A basic finding from our experience in dealing with machining accuracy is that machine tools are deterministic. By this we mean that machine tool errors obey cause-and-effect relationships and do not vary randomly for no reason. Further, the causes are not esoteric and uncontrollable but can be explained in terms of familiar engineering principles."

Once repeatability is achieved, error reduction can be accomplished by one of three basic approaches normally called the "brute strength", the "error correction", or the "error compensation" approach [Blaedel, 1980]. Regardless of the method chosen, a rule can be simply stated that no machine can be accurate unless it is repeatable. Teague and Evans summarize this in the following pattern: "design for repeatability; compensate and/or correct for motion errors to achieve accuracy". As a principle for evaluating machines, I would use the following.

PRINCIPLE: A PRECISION MACHINE SHOULD HAVE A REPEATABILITY OF NOT MORE THAN THREE TIMES THE RESOLUTION OF ITS SCALES.

Isolation

Precision machines are subject to disturbances from many sources. To quote Maxwell [Maxwell, 1890], "In designing an experiment the agents and phenomena to be studied are marked off from all others and regarded as the field of investigation. All others are called 'disturbing agents'. The experiment must be so arranged that the effects of disturbing agents on the phenomena to be investigated are as small as possible." In precision manufacturing the major disturbing agents are vibration, temperature changes, humidity, acoustic noise, and electric and magnetic fields. In specific situations other disturbances may become important, such as particle count in clean rooms. There are at least three design strategies for making a machine resistant to environmental disturbances. They are (1) decoupling the machine from the environment; (2) designing so that environmental disturbances are minimized; or (3) controlling the environment [ANSI, 1973]. For illustration of the differences, take the disturbing influence of temperature. An example of decoupling would simply be to insulate the machine from all external convective, conductive, and radiant heat transfer sources. Designing it to reduce temperature variation effects would probably consist of manufacturing the machine elements from some low-expansion material such as Invar or Zerodur. Finally, controlling might consist of either temperature controlled air or oil showers [Bryan, 1972]. Multiple examples exist in the literature.

For vibration isolation, analogous solutions would be (1) vibration isolation legs for decoupling; (2) designing a structural loop so stiff that the lowest resonant frequency is much greater than any exciting influence; and (3) active sensing and cancellation of vibration using actuators in the sensitive directions. Again, examples of all types exist.

Teague and Evans again summarize this in the following pattern: "Conceptually draw an envelope that totally encloses the instrument/machine; weigh the effects of mechanical, acoustical, temperature, light, humidity, atmospheric pressure, gravitational variations, and excitations on instrument performance; formulate design strategies to shield or to isolate the instrument from these and any other suspected environmental influences to the maximum extent justifiable by the demands of the instrument." In an attempt to make this more concise, I would propose the following [ANSI, 1991; ANSI, 1991].

PRINCIPLE: THE ENVIRONMENT SURROUNDING A PRECISION MACHINE SHOULD BE SUCH THAT TEMPERATURE, VIBRATION, RADIATION, HUMIDITY, AND CLEANLINESS ARE CONTROLLED TO A LEVEL COMMENSURATE WITH THE

MACHINE'S ACCURACY SPECIFICATION.[2] THE TEMPERATURE VARIATION ERROR (TVE) AND THE BROAD-BAND RELATIVE VIBRATION AMPLITUDE OF A PRECISION MACHINE SHOULD BE LESS THAN 50 PERCENT OF THE MACHINE'S REPEATABILITY.

Properly-constrained Mounting

There is more confusion and dissention amongst practitioners of precision engineering about the proper mounting of machines and carriages than in any other area. There are two quite separate schools of thought, each of which has its proponents and each of which is convinced of the correctness of its approach. The two approaches can best be called "kinematic mounting" and "elastic averaging". My opinion is that each of these has its place, and properly designed mounts of either type have specific advantages. Besides these two apparently contradictory approaches, a third approach, which is called "semi-kinematic", combines kinematic principles with some aspects of elastic averaging and deserves discussion.

Kinematic mounting. Kinematic mounting is scientifically based in the work of Maxwell [Maxwell, 1890]. He stated, "The pieces of our instrument are solid, but not rigid. If a solid piece is constrained in more than six ways it will be subject to internal stress, and will become strained or distorted, and this in a manner which, without the most exact micrometrical measurements, it would be impossible to specify." In more modern terminology, in order to stably mount a rigid body without unpredictable and changing internal distortions, it is necessary to constrain it in the minimum number of points. The philosophy is applied to mounting whole machines and to mounting carriages on machines. Implementations are fairly diverse and some types of mountings use kinematic principles to provide equal loading at a large number of points. As a secondary consideration, proper kinematic design can greatly reduce the required manufacturing accuracy and still produce machines with repeatable performance.

Kinematic and semi-kinematic machine mounting. The classical "kinematic machine mount" is simply the three-point support, as represented by the tripod used in surveying and the three "feet" of many well-designed small machines. A clever variation on this uses five legs, two of which do not actually touch the ground. This is done because a three-point support is unstable for loads placed outside the triangle formed by the three points. Therefore, on a machine which is basically square, there is nowhere to place the three legs such that the machine would be stable with a heavy load at any of the four corners. The two extra legs are used to remedy this situation and generally clear the foundation by only fractions of an inch. It should be noted that it is extremely rare to see a real kinematic mount on a machine tool or measuring machine, as in the presence of

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[2]Due to differences in levels of what is considered precision in industry, this can not, in general, be better quantified.

friction a three-point support is not kinematic. (For example, if a three-legged steel machine is set on a concrete base and the dimensions of the machine or base change with temperature or humidity, considerable stress could be set up in the machine since the support points are not free to slide.) This problem was addressed by Bryan [Bryan, 1979] on the support of the metrology frame for DTM-3 which sits on kinematically-designed hydrostatic-bearing pads.

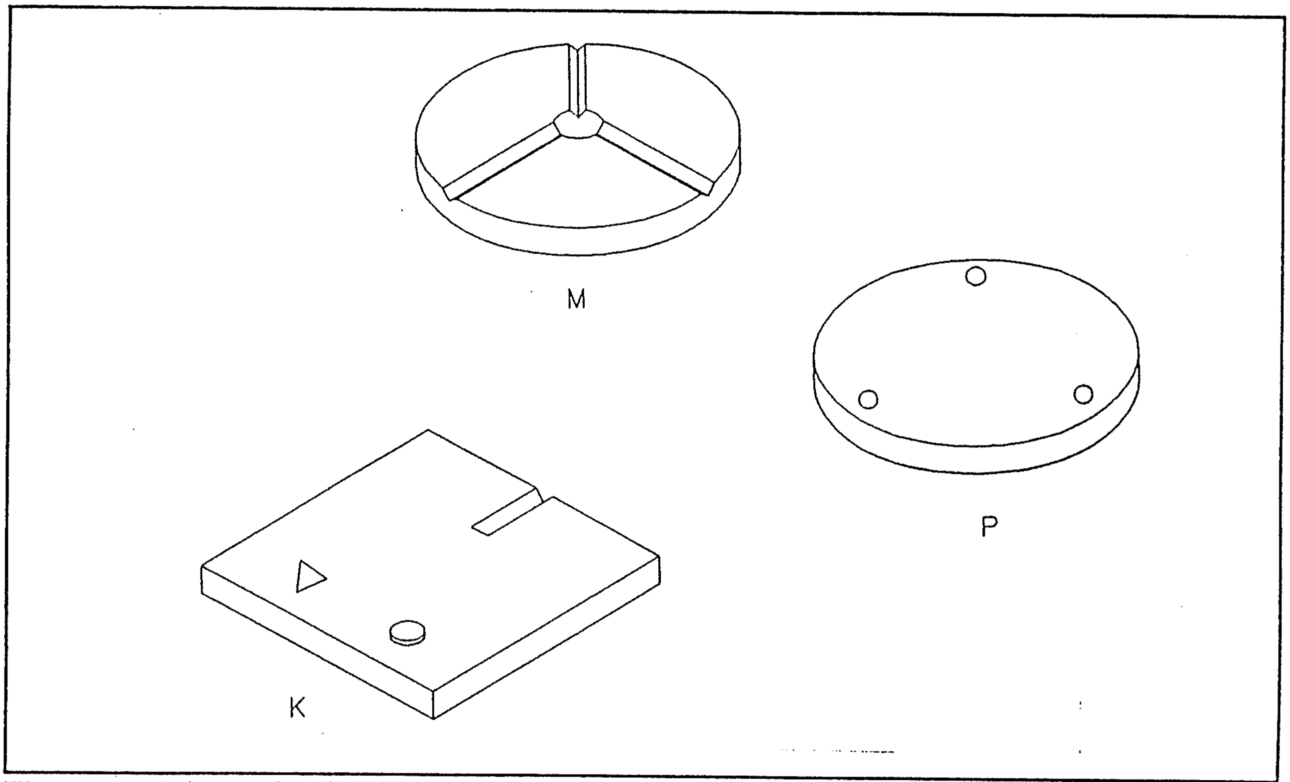

Figure 1. Schematic diagrams of the two types of kinematic mounts.

The classic kinematic mount takes two forms: it is either three Vs pointing towards a common center in which three balls rest or the V, trihedral, flat design, with the V pointing towards the trihedral. Mounts of both kinds are shown in Fig. 1. Both of these date from the 1800s and are discussed by John Strong in his book, <u>Procedures in Experimental Physics</u>. Historically speaking, the three-V design is called the "Maxwell clamp" and the V, trihedral, flat design the "Kelvin clamp". I have often seen these used constraining optical elements, but rarely on a machine base. The Stewart platform [Stewart, 1965] is also, in some sense, kinematic, if the mounting for the linear actuators is so designed so as to provide force in only one direction. Most actual Stewart platforms that I have seen do not conform to this restriction.

A variation on the kinematic designs above which provides for multi-point support of a structure is commonly used for the suspension of very long length standards and for support of large optical elements. Such a mount has been called a "wiffle tree" mount in analogy to a similar mechanism used on horse-drawn carriages and wagons to balance the pull from multiple horses with different levels of strength.[3] Teague and Evans contend that these designs are kinematic. To my mind they are elastically averaged but with the elastic averaging kinematically controlled. By this I mean that although the load is supported on more than the appropriate number of points the final distribution of this load to the load-bearing frame is accomplished in the correct kinematic fashion. An illustration of such a mount holding a long length standard (in two dimensions) is shown in Fig. 2 and holding the telescope mirror of the Keck telescope (in three dimensions) is shown in Fig. 3. These mounts merely ensure that one is distributing the load very carefully on multiple points. Obviously floating an element in liquid would provide similar support but with the load more uniformly distributed.

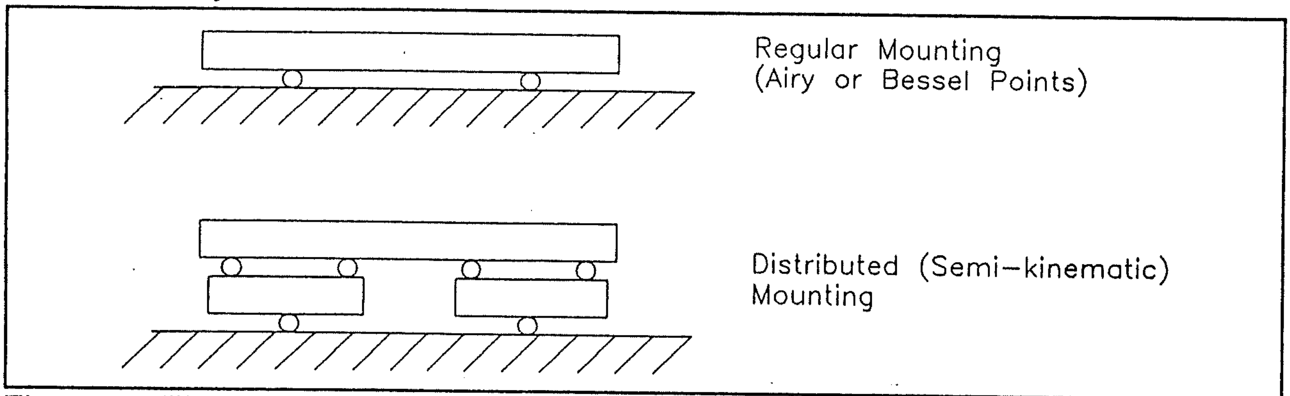

Figure 2. Illustration of a two-dimensional "wiffle tree" type mounting used on length standards.

In an attempt to summarize these comments on machine mounting, I have arrived at a principle.

PRINCIPLE: IF AT ALL POSSIBLE, A PRECISION MACHINE SHOULD BE MOUNTED ON THREE POINTS OR VIA A STRUCTURE THAT ENSURES THAT THE LOAD DISTRIBUTION WILL BE UNCHANGED FOR SMALL DIMENSIONAL CHANGES OF THE MACHINE OR ITS FOUNDATION.

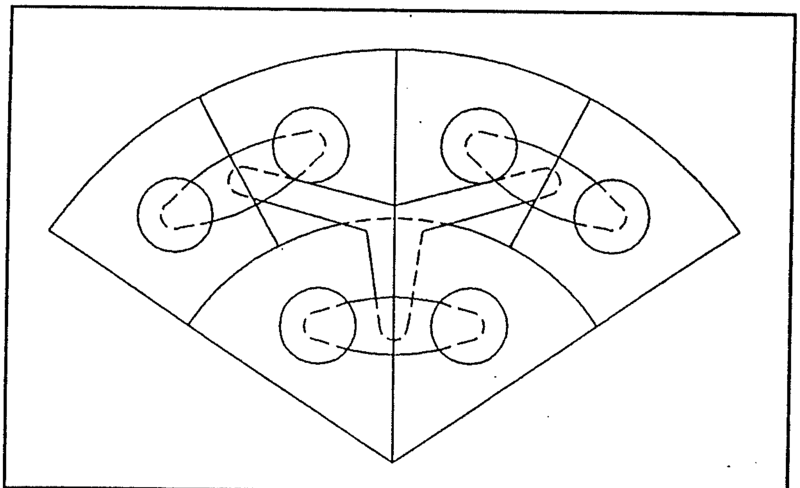

Figure 3. Illustration of "wiffle tree" mounting on the Keck telescope.

[3] We will return to the wiffle tree concept when dealing with kinematic drives.

Elastically-averaged machine mounting. Although the principles of kinematic mounting have been well established in the scientific community since the 1800s, it is still common to find precision machines using elastically-averaged mounts. The primary purpose for this practice is to allow the foundation to provide some of the necessary stiffness and alignment for a large machine structure. Although such a practice may seem to violate the principles discussed above, I personally have seen many machines behave quite accurately and stably over periods of years when such a mounting is used.

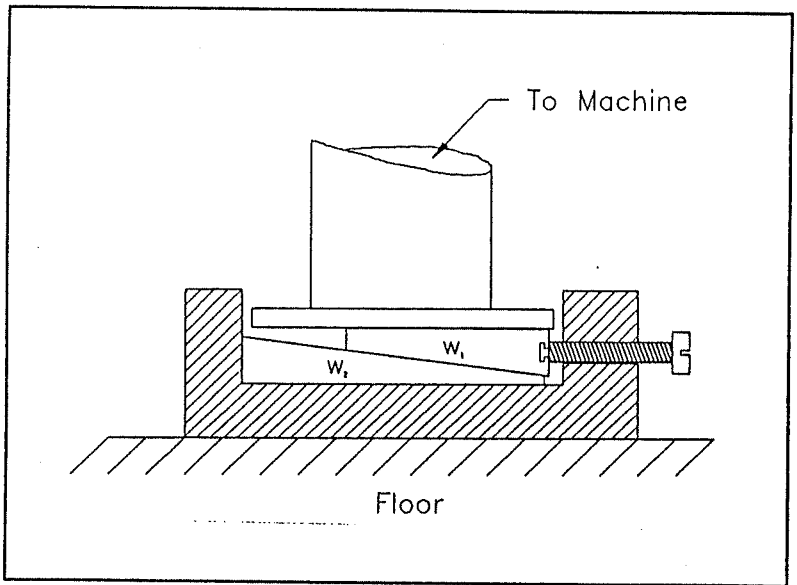

Figure 4. Typical arrangements used for the leveling of large machines.

The classical technique is to pour a well-designed and reinforced concrete foundation more than thick enough to accommodate the weight of the machine. Here we get into an area more appropriate for a civil engineer, but on foundations that I have designed I have specified a minimum of 18 inches thick (heavily reinforced) concrete set in a bed of undisturbed soil or sand. It is also necessary to provide a good moisture barrier between the foundation and the sub-soil, in order to minimize the hygroscopic effects of the concrete. Next, the foundation should be cured for a long period of time before mounting of the machine. I arbitrarily use six months; however, a year may be better. After the concrete is fully cured, the machine mounts are attached by various techniques. On one machine I used simple concrete anchors; however, epoxied plates can also be quite successful. Next, the machine is mounted on the plates and, with alignment tools, load is applied to each of the mounting jacks in order to "true up" the machine frame. Many different types of jacks have been designed. A dual-wedge configuration is shown in Fig. 4. Often the alignment procedure is coupled with either scraping or grinding to bring the way bed to the desired geometry. Although I have watched this done, I am not personally an expert at doing it. It is, however, in my opinion, an art form. One large way bed that I installed in this manner (22 feet long) took an expert technician from a precision engineering company two weeks to bring into geometric compliance. When finished, the carriage traversing this way bed had a pitch of less than 1 arc second. In a previous installation of the same design this bed remained stable for many years; however, it is important on such configurations to re-check at regular intervals. The intervals should be short at first (say, three months) and then lengthened if the system proves to be stable.

PRINCIPLE: IF A MACHINE IS TO BE MOUNTED ON MULTIPLE POINTS IN ORDER TO PROVIDE SUPPORT FROM THE FOUNDATION, THE FOUNDATION MUST BE SPECIFICALLY DESIGNED, CAREFULLY CURED, AND OF SUFFICIENT DEPTH TO PROVIDE THE REQUISITE SUPPORT. FURTHER, THE MACHINE ALIGNMENT SHOULD BE RE-CHECKED AT REGULAR[4] INTERVALS AND READJUSTED AS REQUIRED.

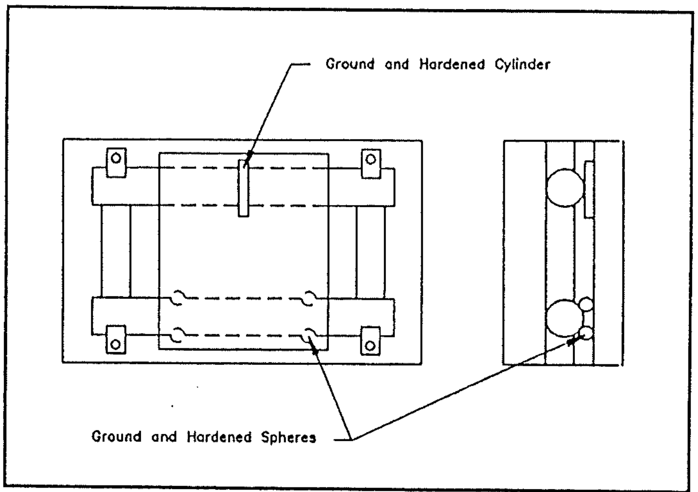

Figure 5. A linear guideway using ball contacts and a cross roller (from [Teague, 1990]; original figure from [Rolt, 1929]).

Kinematic and semi-kinematic mounting of carriages. Just as machines must be constrained in such a way so as their internal distortions are predictable, so also must the machine carriages, both linear and rotary. As with machine mounting, however, there is also a school of thought that advocates elastic averaging. This will be discussed further below. In kinematic and semi-kinematic mounting, the idea is that there should be only five degrees of freedom constrained on a moving carriage. A classical design is shown in Fig. 5 [Rolt, 1929]. Here the carriage really rests only on five points and thus as it is moved along the guideway will not distort internally due to changing loads. A similar design for a rotary carriage is shown in Fig. 6 [Pollard, 1929], where again each of the two carriages rests on five contact points. It is easy to see from these examples that although kinematic design provides for low cost and high performance as the load increases the ideal point contacts deform and Hertzian stresses increase, which can lead to friction, wear, and damage to the ways or the contacting element. To quote, "Even with a pure kinematic design where theoretically one has a point contact, the effect of an indentation of a ball location cannot be ignored. In the case of linear translation, for example, the point contact on cylinders or balls rolling in Vs will not carry much load without serious loss of accuracy due to indentation or deformation of parts. Thus in instruments required for heavier duties, the point or line contacts must at times give way to plane surfaces. Moreover, in use these surfaces some wear will occur, flats will be worn on the bars forming the guideways and indentations will appear in the V grooves..." [Furse, 1981]. This approach to design is called "semi-kinematic". Teague and Evans define semi-kinematic as follows: "If the area of each locator in a coupling is reduced to a theoretical point contact, the design should become one of pure kinematics and not contain over-constraints."

[4]The intervals should be short at first (for example, six months) and then may be lengthened if the installation is stable.

Two classic examples of semi-kinematic coupling are shown in Figs. 7 and 8. Figure 7 shows the design of the way system and support bearings used by SIP for their measuring machines over a period of many years. Five pre-loaded ball bearings are used to provide line contact between the carriage, which is connected to the bearings, and the flat and V way structure. There are, in reality, five line contacts: four on the V and one on the flat. Figure 8 shows the "Gothic arch" design developed by Slocum [Slocum, 1992] for use in the kinematic mounting of fixtures in turning, where high forces are encountered. The large radius of curvature of the Gothic arch increases the area of contact to the ball and thus provides greater load bearing capability without damage to the surface. In both cases the extension of the kinematic point to an averaged line or area greatly increases the wear resistance and load-carrying capability of the design.

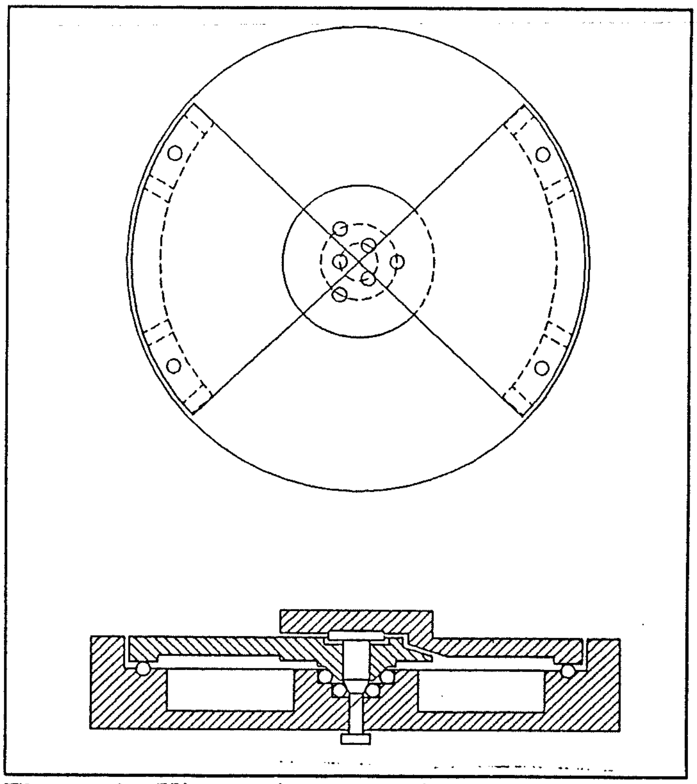

Figure 6. Kinematic mount for two rotary axes stacked on top of each other (from [Pollard, 1929], via [Teague, 1989]).

PRINCIPLE: WHERE LOAD PERMITS, KINEMATIC OR SEMI-KINEMATIC DESIGNS FOR MACHINE CARRIAGES ARE PREFERRED. THEY ARE MORE REPEATABLE AND CAN BE RELIABLY PRODUCED WITH LOWER TOLERANCE.

<u>Elastically-averaged mounting of carriages</u>. Teague and Evans give elastically-averaged carriage mountings a "black star". Many examples exist, though, where elastically-averaged systems work remarkably well. The most common for a linear guideway is the double V (or twin V) design used by Moore Special Tool on their measuring machines, jig borers, and diamond turning machines [Moore, 1970]. Many grating ruling engines also use this design. It is extremely robust, if properly constructed, provides for excellent load distribution, resistance to wear, and can be remarkably repeatable. A drawing of such a system for a linear guideway is shown in Fig. 9. In later years, Moore used extremely large numbers of caged roller bearings between the double Vs and the carriage for ease of control (reduced friction).

Many hydrostatic bearing designs also use elastic averaging. In some cases the correct number of pads are used to provide a semi-kinematic structure [Slocum, 1992]; however, in other cases, such as air-bearing spindles [P.I., 1991], no attempt is made to apply kinematic principles. Some of the best air-bearing spindles in the world are overconstrained.

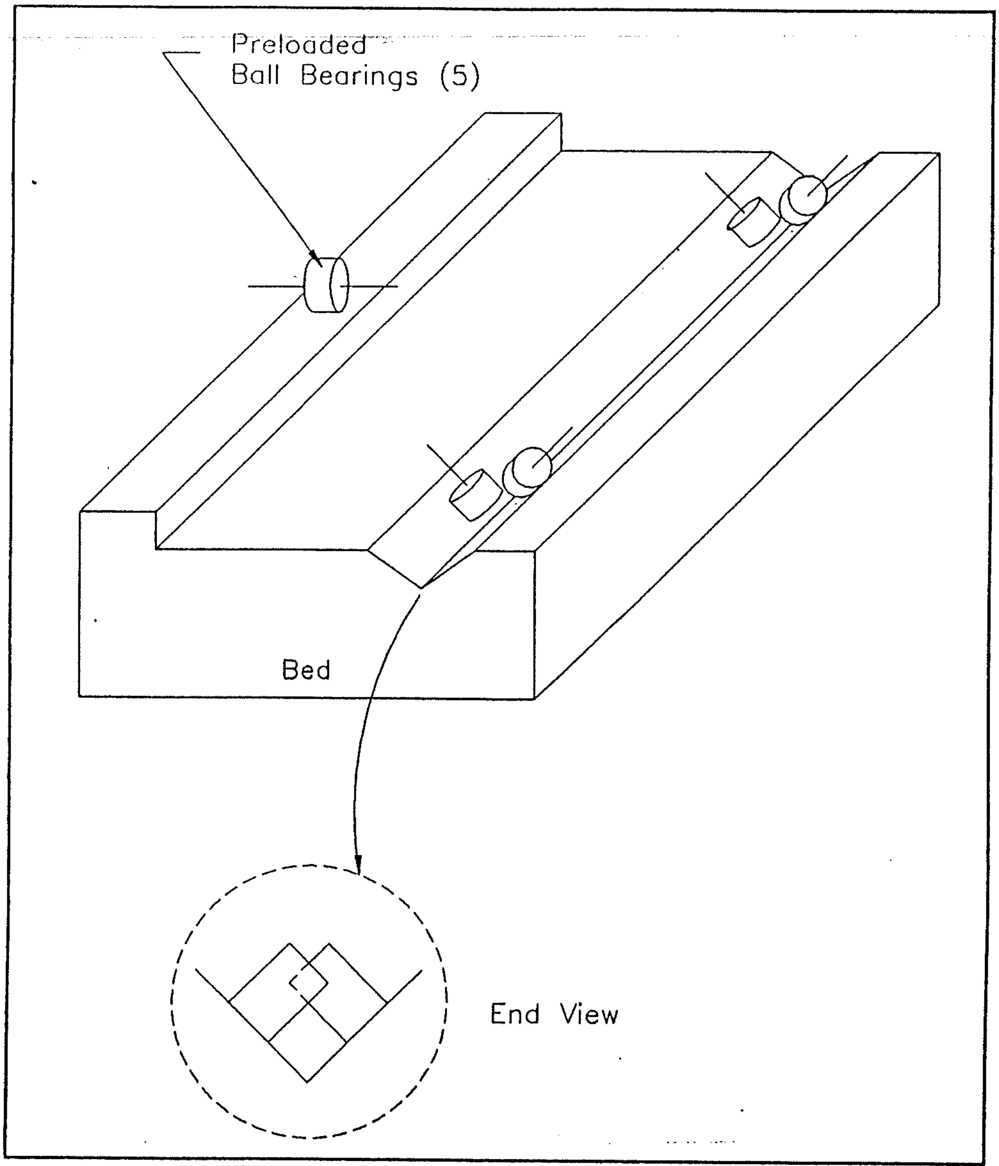

Figure 7. A semi-kinematic guide system for a linear carriage used by Société Genevoise for over a century.

Another common design which uses elastic averaging is incorporated in most indexing tables. Indexing tables are basically face gears which contact at numerous points. When they are lifted and re-rotated they return to the same position with extremely high accuracy. A diagram of such a coupling is shown in Fig. 10. Angular repeatability of indexing tables exceeds that of any other mechanism known to the author and may be as good as ±0.02 arc second [Reeve, 1968]. Curvic couplings, which are essentially the same as indexing tables, also provide for the transmission of large loads and the centering between different rotating elements. They are in common use in many jet aircraft engines, for the same reasons as mentioned above.

Finally, another type of elastically-averaged way structure is used for short-range motion in a wide variety of precision instruments. This elastically-averaged structure is normally called a "flexure". An example of such a structure is shown in Fig. 11 [Jones, 1988].

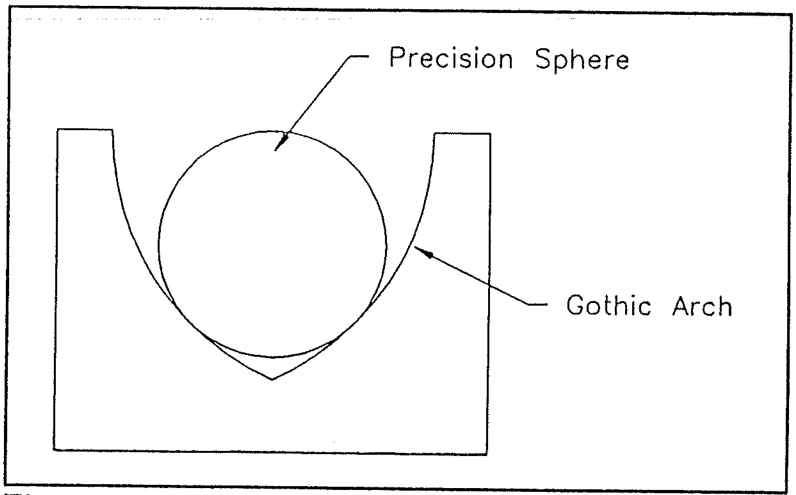

Figure 8. A Gothic arch used to replace a V in a kinematic clamp.

Given the success of the above designs, one can not arbitrarily rule out elastic averaging as an appropriate design strategy. The principle then is, in analogy to the similar principle for machine mounting, as follows.

PRINCIPLE: IF CONDITIONS OF LOAD, STIFFNESS, AND STABILITY CAN NOT BE SATISFIED BY KINEMATIC DESIGN, ELASTIC AVERAGING MAY BE USED. A CORRECTLY DESIGNED SYSTEM USING ELASTIC AVERAGING CAN HAVE HIGH ACCURACY, STABILITY, AND STIFFNESS.

Alignment Principle

For linear axes, unwanted angular motions of a machine's carriages always occur. These angular errors can seriously affect the positioning accuracy of the carriage. In cases where the scale which measures the motion of the carriage is offset from the line of motion to be measured, this effect is magnified. Thus, for many years it has been one of the first principles of machine design to, if at all possible, align the scale of the machine such that there is no offset between the scale and the line of measurement. This is called the "Abbe principle" or the "alignment principle" [Abbe, 1890]. The Abbe principle requires that the line of action of the displacement measuring system be co-linear with the displacement to be measured.

Bryan [Bryan, 1979] reflected current practice when he modified the original Abbe principle to allow for compensation of the measurement line through external means. This is routinely done on NC machine tools where it is normally almost impossible to position the scale along the line to be measured, which not only is where the cutting tool is present but also is quite variable. Traditional practice compensates for this effect by calibrating the machine through lines that go through the center of the work zone and storing the errors in a look-up table used by the controller for correction (so-called "lead screw comp"). This is only part of the problem, however, since the effect of angular errors varies continuously throughout the work volume. More elaborate schemes that correct for all of the angular errors were developed in the mid-'70s [Hocken, 1977] and are now used on most production coordinate measuring machines [Fix, 1988]. The use of these correction schemes on machine tools has been developed in the laboratory [Donmez, 1984] but is still not common in production, although one prototype diamond turning machine [Donaldson, 1983; Estler, 1985] routinely uses full-scale volumetric correction.

PRINCIPLE: WHENEVER POSSIBLE, MINIMIZE THE OFFSET BETWEEN THE MEASUREMENT AXIS AND THE MOTION AXIS OF A LINEAR CARRIAGE. IF THIS IS NOT POSSIBLE, CORRECT FOR SUCH OFFSETS THROUGH PARAMETRIC CALIBRATION.

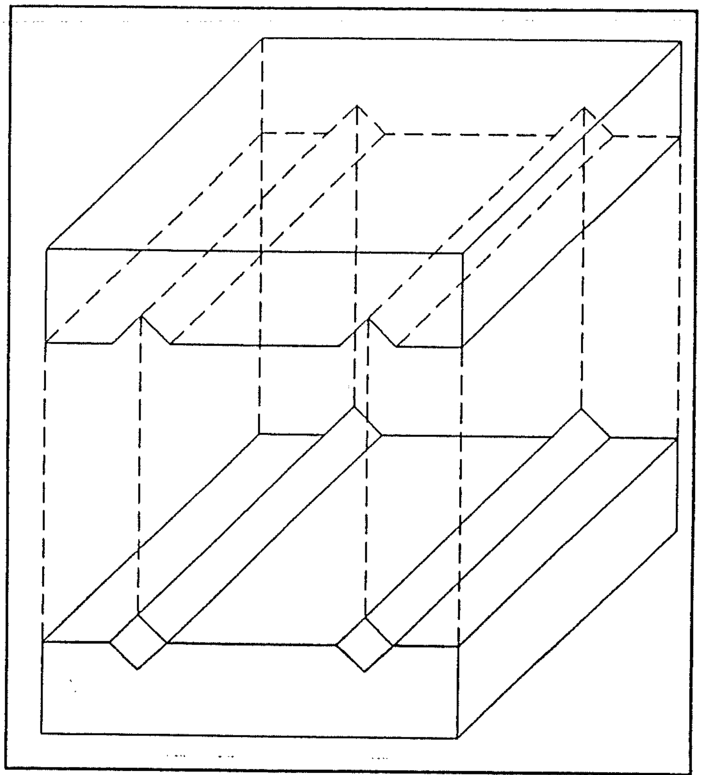

Figure 9. A double-V way system used commonly by Moore Special Tool. It provides excellent lateral stiffness but must be very carefully manufactured.

Metrology Frame

On the scales of interest in precision manufacturing, all real materials are subject to distortions which are sometimes large in comparison to the tolerances sought. In many machines these distortions cause significant errors due to unwanted relative displacements between the measurement sensors and the moving elements. The solution to this problem has been to carefully separate those elements that perform the measurement (the metrology system) from the other machine elements that bear the load. Several modern designs developed in the laboratories during the '70s and '80s [Hocken, 1977; Bryan, 1979; Donaldson, 1983; McKeown, 1987] reinforced this concept. It was, however, previously used on several metrology machines in the past and has been commonly practiced by surveyors doing first-order precision land surveys for decades.[5]

[5]A first-order survey is done with Invar tapes that are loaded with a 10 kg weight. This loading causes the structure holding the tapes to be deformed. For that reason this structure is always separate from the tripods that are set over the monuments where the tape is "read".

Again, very few production machines have adopted this principle; however, one measuring machine [Fix, 1988] addresses this issue by making the load-bearing table completely separate from the metrology frame that controls the motion of the bridge. A diagram of this machine is shown in Fig. 12. Note that as the machine bridge moves its weight is borne by the substructure, not the table. Further, if the table is distorted by a load it will distort the bottom machine frame only indirectly since supports are under pure compression.

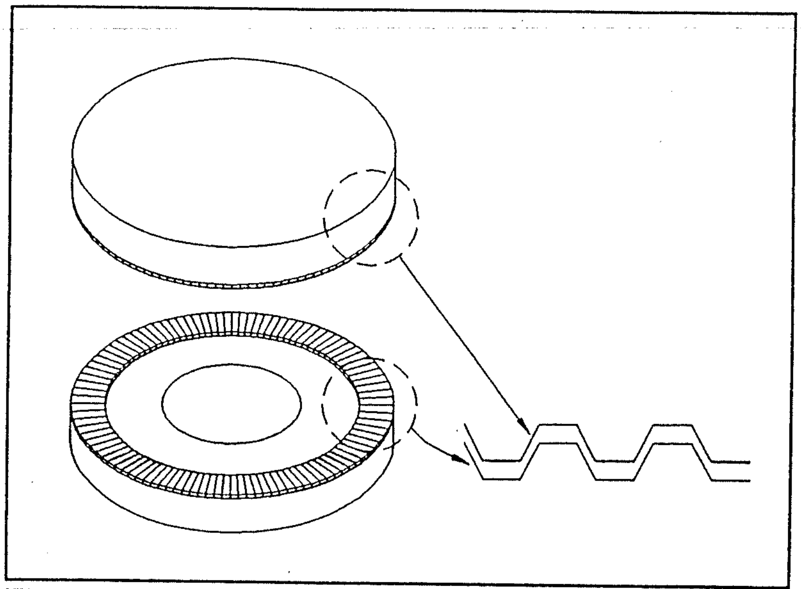

Figure 10. Face gears used on indexing tables. If the teeth are curved the same device is called a curvic coupling.

In many cases it is economically unviable to produce a machine with a complete metrology frame. As a subset of this principle one can obtain some independence between the measurement and the drive system by simply making them separate. The distinction is as follows. On a machine tool that uses resolvers on the end of the ball screw to provide position information, the drive and the measurement system are tightly coupled. If the same machine is equipped with lasers or line scales and still driven by the ball screw, the drive and the measurement system are, to a certain extent, decoupled. (Coupling can still exist through distortions of the machine frame.) In this case the latter solution is preferred.

PRINCIPLE: EXAMINE THE INFLUENCE OF VARYING LOADS ON THE MACHINE BASE DUE TO CARRIAGE MOTION AND DIFFERING WORKPIECE WEIGHTS. IF NECESSARY, ISOLATE THE METROLOGY MOUNTING SYSTEM (METROLOGY FRAME) FROM VARIABLE LOADS.[6]

Materials Selection

The choice of materials for a machine is a key factor in determining final machine capability. Many criteria may be considered, including temporal stability, homogeneity, thermal expansion, etc. Often these criteria differ for differing machine elements and the number of approaches that have been taken are numerous. Teague and Evans have listed

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[6]Metrology frames are subject to most of the other principles, including isolation, material selection, etc., as their stability is essential to correct machine function.

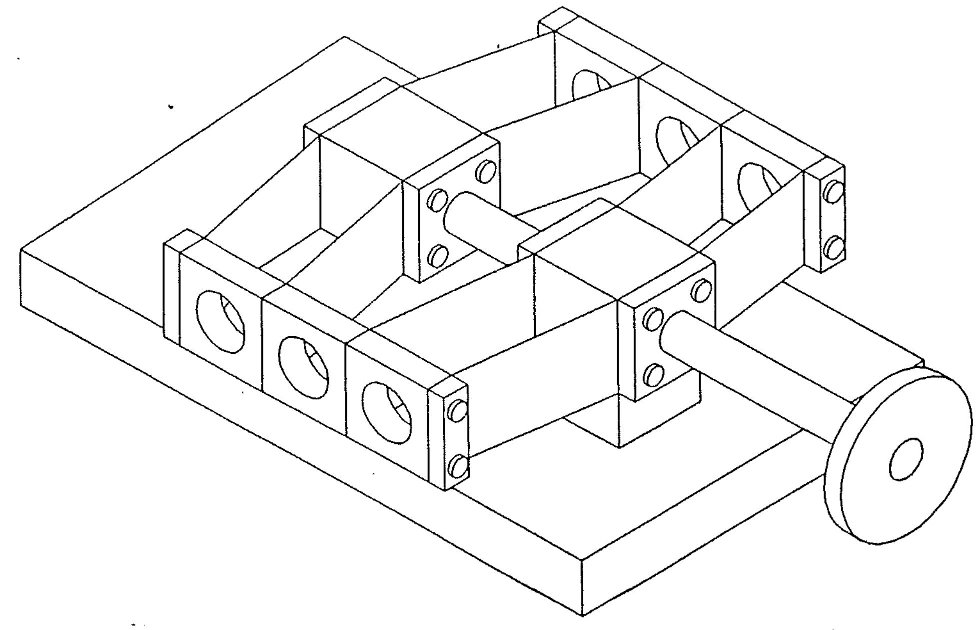

Figure 11. A flexure design for linear motion. The large attachment areas for the springs provide elastic averaging. Such a flexure also illustrates symmetry.

several criteria which should be considered. They are as follows.

1. Thermal equilibration efficiency as measured by the ratio of the thermal conductivity to the thermal expansion coefficient.

2. Speed of thermal equilibration as measured by thermal diffusivity. The diffusivity is the ratio of the thermal conductivity to the product of the density and the specific heat.

3. Specific stiffness; that is, the response to static and dynamic loading defined as the ratio of the modulus of elasticity to the density.

4. Internal damping; that is, the specific damping capacity.

5. Deformations under point loading. This is determined almost exclusively by the modulus of elasticity.

6. Radiant coupling which is determined by the emissivity of the machine's surfaces, as well as the ratio of the surface area to volume.

7. Secular stability.

8. Corrosion resistance.

9. Processing costs.

10. Materials costs.

To this list most precision engineers would add simply:

11. Thermal expansion coefficient.

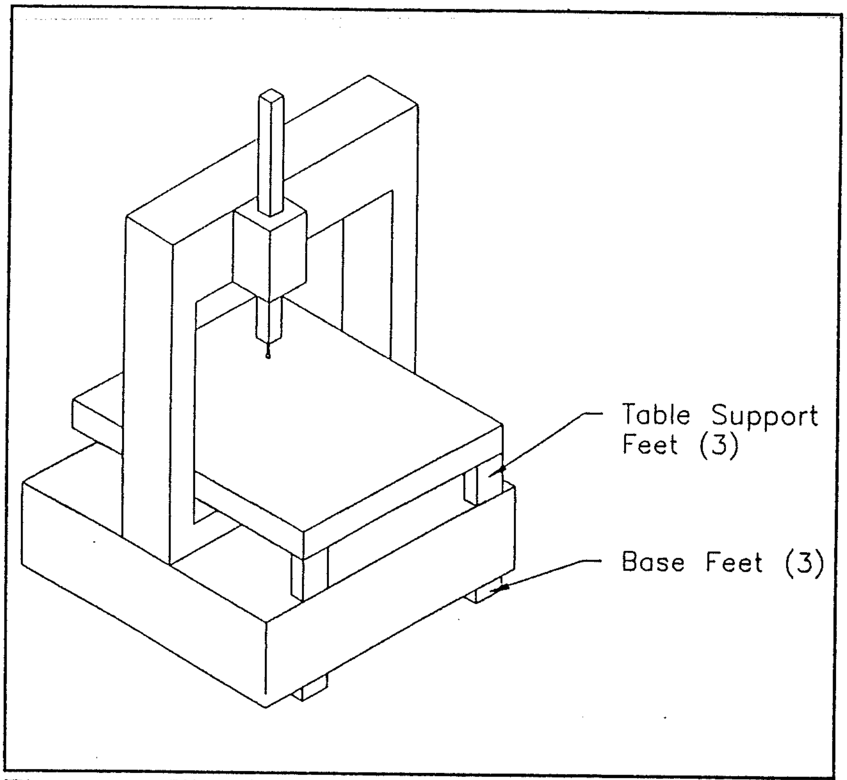

Figure 12. The ring bridge design pioneered by Sheffield.

In another report, Amatucci and Teague [Amatucci, 1989] propose a quantitative method for evaluating the relative merits of different materials; however, there is no uniform approach to these problems. According to Teague and Evans, "The ideal material ... would have the thermal conductivity of copper, the specific stiffness of beryllium, the heat capacity of Invar, the ratio of thermal diffusivity to thermal expansion coefficient of silicon, the modulus of elasticity of silicon carbide, and the specific damping capacity of magnesium...". To that I would add "the secular stability of Zerodur".

The material properties of many structural materials are not well quantified in terms of the criteria mentioned above. Also, there exist considerable differences of opinion as to which property is most important. The simplest example of this relates to machine tools, where one school of thought contends that if the machine tool and its scales are made of a material that is being machined, then thermal effects are minimized. Another school of thought attempts to minimize expansion of the machine tool structure. In either event, I am currently suspicious of any but the well-established structural materials. These, to my mind, are cast iron, granite, steel weldments, and Granitan. In encountering a machine using other materials I would examine closely the reasons for their selection and the method of their application.

PRINCIPLE: THE MATERIALS SELECTION FOR A MACHINE SHOULD BE BASED ON ENGINEERING PRINCIPLES SUCH AS STABILITY, STIFFNESS, DAMPING, THERMAL PROPERTIES, AND CORROSION RESISTANCE.

Structural Loop

A structural loop is defined as "An assembly of mechanical components which maintain relative position between specified objects. A typical pair of specified objects is the cutting tool and the workpiece; the structural loop would include the spindle shaft, the bearings and housing, the slideways and frame, possibly the foundation, and the tool and work-holding fixtures." [ANSI, 1991]. An illustration of the structural loop, taken from this Standard, is shown in Fig. 13. In any precision machine the structural loop and its stability are of great concern. The structural loop should be such that it is stable in time, temperature, and under vibrational excitations. The latter condition is particularly important in machine tools, where self-excited vibrations, i.e., chatter, can significantly degrade machine performance.

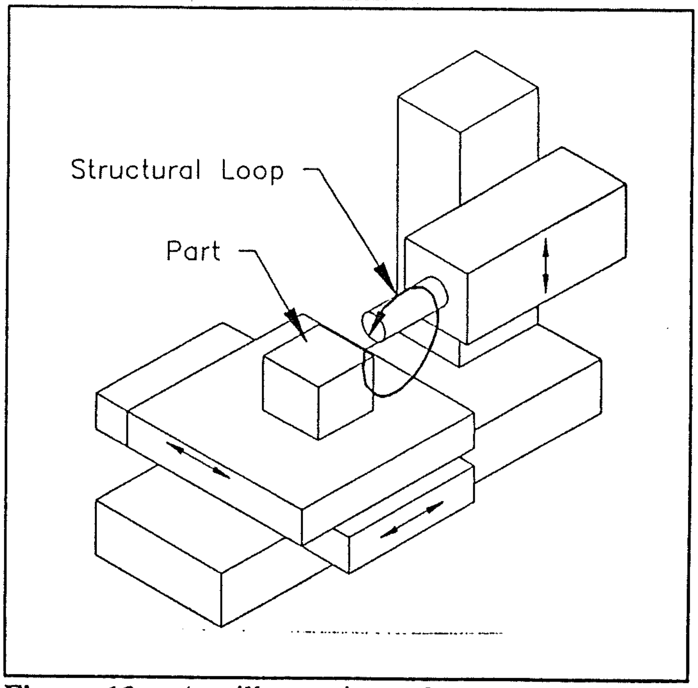

Figure 13. An illustration of a structural loop taken from ANSI B5.54.

The subject of chatter and its avoidance is really sufficiently complex that a separate document should be created which explains the complexities of the phenomena. Here I only mention that structural loop stiffness is only one part of the problem, as cutting stability is determined by the appropriate selection of feed, speed, depth of cut, and tool geometry. Even flexible machines can be quite stable if an appropriate region of stability, as defined by a lobing diagram, can be discovered (see ANSI B5.54 and publications referenced therein for a more complete explanation).

As an anecdotal note it should be mentioned that many instrument designers do not fully understand the requirement for stiffness in the structural loop. In recent history the builders of the first scanning tunneling microscope (STM) [Binnig, 1982] went to elaborate lengths to shield their "flimsy" structure from vibrations. Most of this shielding was ineffective. Later designs simply shortened and stiffened the structural loop, enabling experimenters to obtain images with atomic resolution without any vibration isolation [Pohl, 1986]. The lesson here is summarized in the principle below.

PRINCIPLE: THE RESISTANCE OF A MACHINE TO VIBRATIONAL AND THERMAL VARIATIONS SHOULD BE MINIMIZED BY MAKING THE PATH FROM THE TOOL TO THE WORKPIECE AS SHORT, STIFF, AND THERMALLY STABLE AS POSSIBLE.

Kinematic Drives

As mentioned previously, all machines are composed of elements that either move in a linear or rotary direction. A great deal of effort has been concentrated on how to drive these elements by applying a force only in the direction of intended motion. This is not as easy as it may sound, as even a simple ball screw, because of friction, misalignment of the drive nut, out-of-straightness of the ball screw, misalignment of the ball screw bearings, etc., can readily transmit forces orthogonal to the intended direction of motion and thus produce errors. Ideally one would prefer to apply a force only at a point and many metrology machines in the past have attempted to do this by using a ball contact against a flat plate. Even this can cause problems if the line of action of the ball is not perpendicular to the face of the plate. This is illustrated in Fig. 14. This design was used recently for a very high-accuracy surface profilometer, called the NanoSurf [Chetwynd, 1987]. Of course, in real machine tools significant forces need to be applied, and this leads to much more elaborate designs such as those of McKeown [McKeown, 1982; McKeown, 1989] and Slocum [Slocum, 1992]. A design, probably taken from McKeown, is shown in Fig. 15. Slocum has recently obtained a patent on his hydrostatic screw design shown in Fig. 16. Special drives, such as the Capstan drive [Donaldson, 1983], have been used in attempts to overcome these difficulties.

On rotary axes, particularly spindle axes, great effort has also been made to kinematically couple the driving motor to the spindle without introducing unwanted error motions. Here another version of the "wiffle tree" mount is used. Such a coupling is shown in Fig. 17. Note that as the drive member is rotated in the absence of friction it is impossible for it to impart other than rotary motion to the spindle.

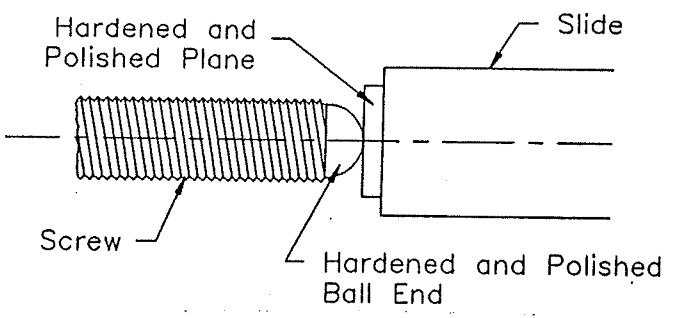

Figure 14. Ball on flat used to isolate a drive from lateral and angular disturbances.

Flexures are also commonly used and available commercially for performing this decoupling function [Slocum, 1992].

PRINCIPLE: DRIVE SYSTEMS MUST BE CONSTRUCTED SUCH THAT THE REQUIRED DEGREE OF FREEDOM, AND ONLY THIS DEGREE OF FREEDOM, IS COUPLED BETWEEN THE DRIVE MECHANISM AND THE MOVING ELEMENT.

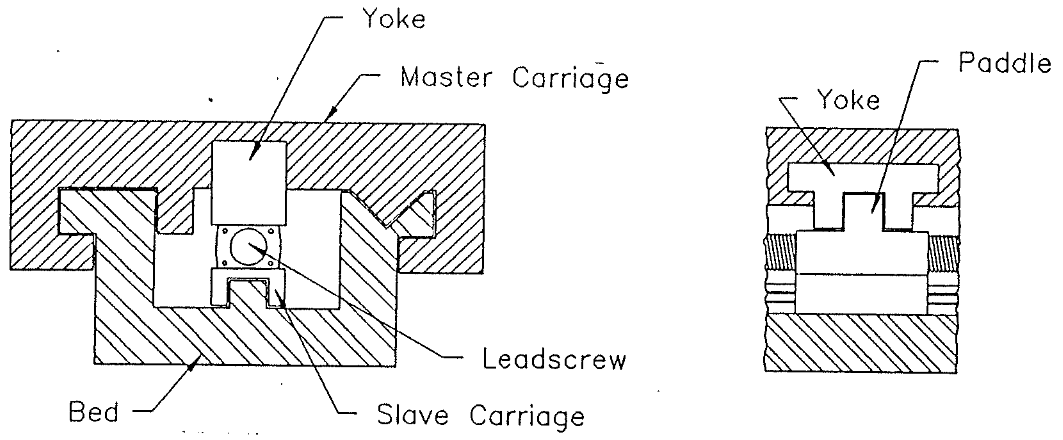

Figure 15. A drive mechanism using a secondary way structure and a "paddle" to decouple error motions of the drive from the carriage.

Probe Knowledge

The ultimate precision of any machine, whether it be a machine tool or a measuring machine, is normally limited by the capabilities of a localized process. Such a process would refer to measurement using a probe on a coordinate measuring machine, cutting with a diamond tool on a diamond turning machine, metal removal by an electric discharge on an EDM machine, etc. The properties of the process must be considered very early in the design and the general practice is to design a machine with intimate knowledge of these process parameters. Two examples come readily to mind. The first, in diamond turning, metal removal rates are extremely small, the tool is very sharp and is normally only used on soft materials, and therefore cutting forces are small. This enables the designer of a diamond turning machine to use air bearings (which are often of low stiffness) to take advantage of their smooth behavior. Decisions such as this can also lead to problems, however, since the machine so designed is certainly limited and cannot readily be used with other tools and materials. This problem was encountered many times when trying to convert diamond turning machines to precision grinding operations where the cutting forces are considerably higher. As a second example, consider the design of numerous measuring machines that use the so-called "touch fire" probe developed in the '70s. Since these machines only acquire data in a point-wise fashion and the probe is primarily sensitive to the velocity of approach, a whole generation of machines were designed that are incapable of following an accurate contour in space due to the compliance of their drives. Rather, they are designed to acquire data by approaching a point at a constant low velocity and latching the readings from their scales when the probe "fires". Again this probe knowledge greatly reduced the cost of these machines but limited their applicability when probe technology changed.

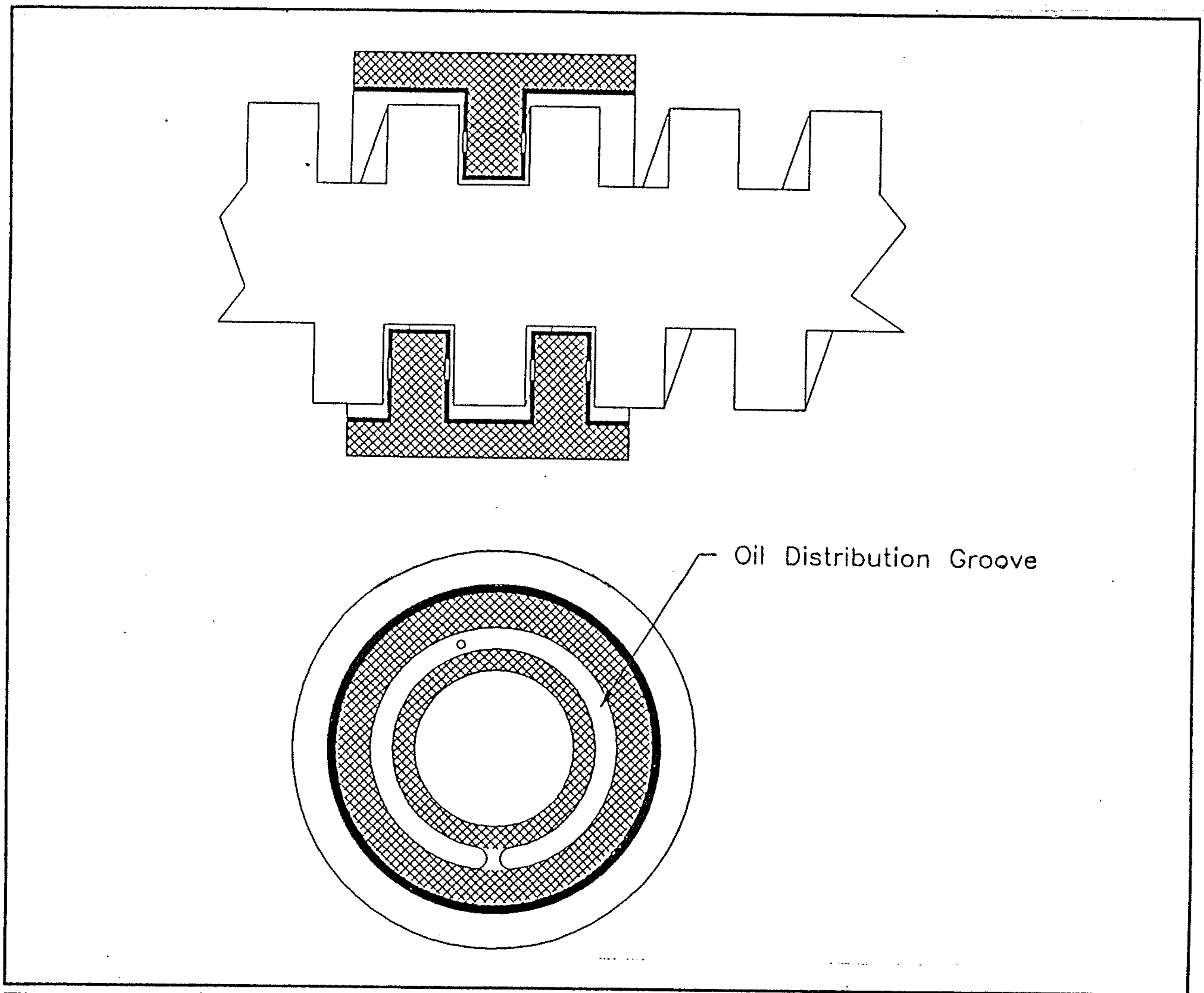

Figure 16. A hydrostatic lead screw, developed and patented by Slocum, for decoupling error motions of a drive from a carriage.

PRINCIPLE: THE DESIGN OF A PRECISION MACHINE SHOULD BE DONE WITH MAXIMUM UNDERSTANDING OF THE LIMITATIONS AND STRENGTHS OF THE PROBE/TOOL SYSTEM AND THE LOCALIZED INTERACTION BETWEEN TOOL AND WORKPIECE.

Energy Flow

Previously we discussed the concept of isolation which is related to this pattern. Energy flow is used here in a more general sense in that it refers to the flow of energy both to and from the external environment and to the machine structure from internal energy sources. Internal energy sources may include thermal sources and sinks, acoustic and vibration generation, electrical noise, etc., while external sources are those that were mentioned previously. In a precision machine design considerable attention must be paid to how to manage these internal sources. For example, on a diamond turning machine each of the drive motors, the spindle motor, any gear trains, the metrology laser (if supplied), and even the cutting "coolant" constitute heat sources (or sinks), and the energy flow from such sources must be carefully analyzed and controlled. Methods include isolation using insulating materials, fans, fins, etc.; however, for such provisions to be successful a thorough analysis must be performed. Cutting coolant is a particularly troublesome source or sink as, if improperly pumped, it can heat up or, if water based, can cool the machine structure and part in a poorly predictable fashion. Oil-showered diamond turning machines overcome this difficulty by using the temperature control medium as the cutting lubricant.

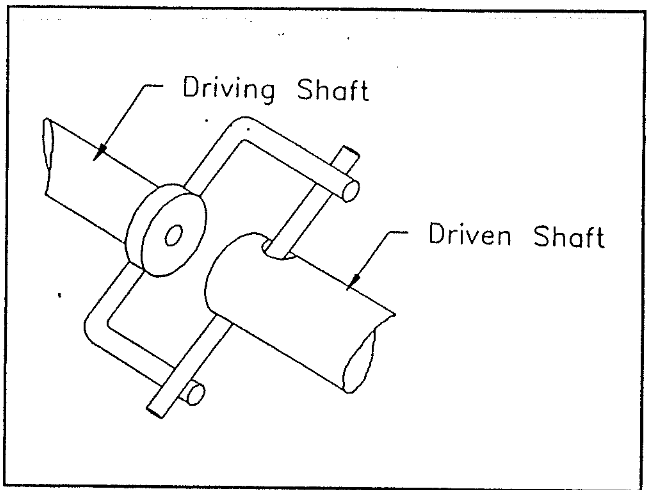

Figure 17. A schematic of a simple rotary "wiffle tree" used to decouple drive motor error motions from a spindle.

In machining, another important source of energy is vibration. Vibration can be caused by unbalanced loads, poorly functioning bearings, gear trains, and the cutting process itself. Even the roller bearings on the ways of a precision machine contribute "roller noise". Again, internal damping is a common method for absorbing energy from these sources, but attention must be paid to variations that can occur through wear.

PRINCIPLE: DETERMINE SUSCEPTIBILITY OF THE MACHINE TO ENERGY SOURCES. THIS WOULD MEAN THERMAL MODELING AND MODAL ANALYSIS. DESIGN TO MANAGE THIS ENERGY APPROPRIATELY. MEASURE THE MACHINE DRIFT AND VIBRATION DUE TO INTERNAL SOURCES.

<u>Symmetry</u>

Symmetry, like kinematic mounting, is a principle that has its proponents and its adversaries. Teague and Evans are strong proponents of symmetry, and in their treatment of this problem arrive at the following pattern: "Incorporate symmetry to the maximum extent possible into the design of each machine element, the entire machine and into the machine environment...". This concept is heavily incorporated into Teague's design of the new molecular measuring machine [Teague, 1989] and shows up commonly in flexure designs. Such a flexure design is shown in Fig. 11 [Jones, 1988]. The symmetry here keeps the forces normal to the direction balanced and thus provides for extremely linear travel.

A second example of this type of symmetry can be readily understood by examining the difference between a typical C section machining center and a bridge-type machining center (see Fig. 18(a) and (b)). The bridge machine is considerably more stable than the C section to changing thermal conditions, as the C has no balancing force on the opposite sides to prevent it from "opening" and "closing" with small temperature gradients. Another type of symmetry to reduce the effects of thermal coupling is that used in calorimeters. Here the common design is to have a series of nested spheres which control both the convective and the radiated coupling between numerous shells.

Of course on the earth's surface it is not possible to obtain ideal symmetry since the gravitational force is uni-directional. This leads to some interested effects that occur on horizontal spindle, high-precision turning machines. Donaldson [Donaldson, 1983] recognized this problem explicitly and thus chose a vertical spindle machine for LODTM. He also recognized the inherent stability of a bridge structure and thus the cross-slide on this machine is a bridge spanning the spindle. Large boring machines (which are sometimes also called lathes) have recognized this problem for many years. It would be difficult to find such a machine with a horizontal spindle.

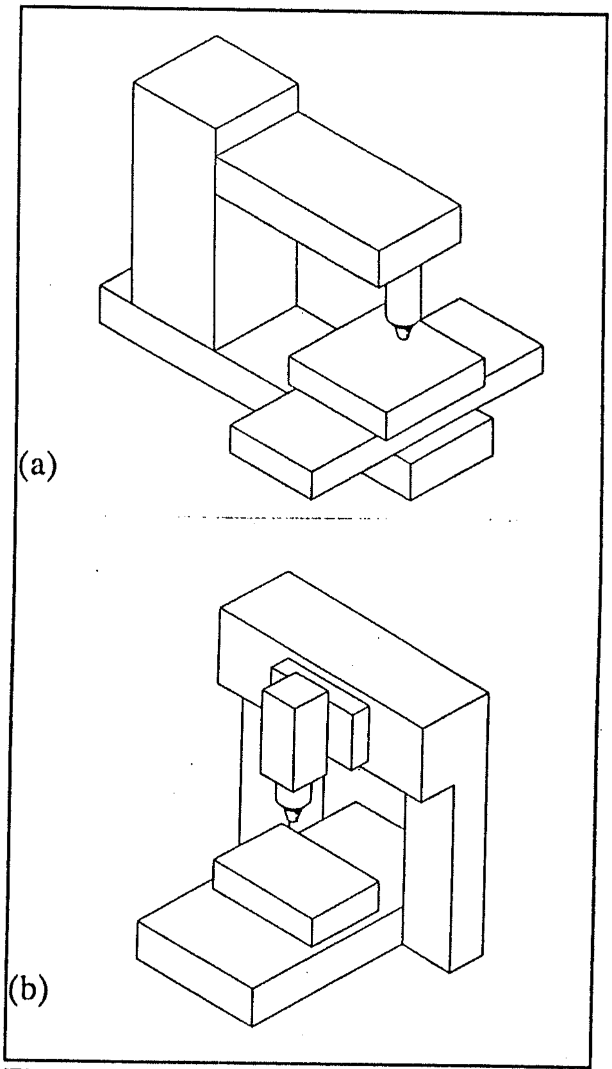

Figure 18. Typical machining centers showing (a) a C section and (b) a bridge-type structure.

In apparent contradiction to the beneficial effects of symmetry, vibrational energy is not reduced by symmetric design, in fact it is often enhanced. This is readily understood by examining a familiar phenomenon. Almost all people have created a resonance in a crystal wine glass by wetting their finger and running it carefully around the rim. In a piece of good glass this excites a vibrational mode which emits acoustic energy and can be readily heard. If the glass were rectangular in cross section (that is, asymmetric), this mode would not exist. Some designers of precision machines [Dahl, 1990] contend that in many applications the elimination of pure vibrational modes, through asymmetry, can be more efficacious than the balancing of other effects through symmetry. This leads to two principles.

PRINCIPLE: USE SYMMETRY TO THE EXTENT POSSIBLE FOR THE BALANCING OF THERMAL AND STRUCTURAL INFLUENCES. THIS REFERS TO THE MACHINE AND ITS ENVIRONMENT.

PRINCIPLE: CAREFULLY BREAK SYMMETRY, WHERE REQUIRED, FOR THE REDUCTION OF VIBRATIONAL RESONANCES.

Force Control

Most machines are composed of a number of carriages, many of which move with respect to each other as well as with respect to the machine base and require connections (electrical, hydraulic, pneumatic, etc.) to the outside world, as well as shielding (way covers, etc.). It is my opinion that the influence of these connections is one of the most often overlooked contributors to degraded machine performance. Examples abound, including a new million dollar circular geometry machine built for the measurement of X-ray optics, where the cabling to the moving gage head is one of the largest contributors to apparently random error motion of the measurement setup. In a well-designed precision machine, forces induced by these attachments are either carefully controlled at a constant level or balanced out with some other mechanism. For example, on LODTM servos were used to balance out the force due to atmospheric pressure on the evacuated laser path, since the designers carefully analyzed their machine for these effects.

PRINCIPLE: THE FORCES INDUCED BY ANY ATTACHMENTS TO A MOVING CARRIAGE (CABLES, WAY COVERS, ETC.) SHOULD BE CAREFULLY CONTROLLED TO BE CONSTANT OR BE PROPERLY BALANCED THROUGH FEEDBACK.

Error Budgeting/Analysis

Essential to the design of any high-technology, precision machine is a thorough analysis of the effects of each of the possible sources of error and an estimation as to how these errors will sum in the completed instrument. In the world of experimental physics this was always called "sensitivity analysis", and when performing an exacting experiment the sensitivity of the result to any possible external variations was computed and included in the error. In the area of precision machines the most complete exposition of this technique probably comes from Donaldson [Donaldson, 1980] and his use of these techniques on LODTM. In a modern factory the methods of Taguchi are similar in concept.

PRINCIPLE: QUANTIFY ALL IDENTIFIABLE ERROR SOURCES AND THEIR EFFECT ON PERFORMANCE PARAMETERS. COMBINE THESE RESULTS INTO A QUANTITATIVE ESTIMATE OF MACHINE PERFORMANCE.

Conclusions

The principles above apply, almost exlusively, to the mechanical aspects of precision machine design. In this day and age a large portion of any machine is in the electronic and computer control systems. To fully integrate such considerations into a similar format would be the subject of an extensive research project.

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