Design for Precision: Current Status and Trends

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Abstract

'Design for Precision' reviews the status quo in Precision Engineering and concludes that today's precision engineers put repeatability at the top of their list. The design rules, patterns or principles, quoted here from various authors, are all time-proven insights, to get reproducible results with ultra precision machines and instruments. Modelling and analysis of different concepts, systems, and components is required to adapt the progressing design or to confirm its adequacy. Expenditure on such analysis is worthwhile to avoid realisation of an inadequate design. However, creativity is more important in keeping the cost down by finding other than locally optimised solutions. World-wide, precision engineers agree on design principles, the challenge is to apply them creatively to obtain a thought-out design. In today's most accurate machines, advanced techniques are applied for compensation of e.g. residual geometric errors, errors caused by machine dynamics, or thermo-mechanically induced errors. Future developments in Precision Engineering require nanometre- or even subnanometre positioning- and measuring accuracy, demanding new design concepts with integrated control and error compensation systems.

Keywords: Design principles and features, predictive design, accuracy, repeatability, reproducibility.

Acknowledgements

The authors would like to thank the following contributors:

K. Blaedel	H. van Brussel
J. Bryan	D. DeBra
J. van Eijk	C. Evans
G. Goch	R. Hocken
P. McKeown	V. Portman
S. Sartori	H. Spaan
C. Teague	E. Thwaite
A. van Tooren	D. Trumper
R. Weill	G. X. Zhang

1. Introduction

Currently, many activities employed in industry and institutes are aimed at creating high precision products and involve high precision processing. The manufacturing of these products demands a highly specialised science called Precision Engineering, which is based on the following disciplines:

- Design for Precision
- Optical and Mechanical Metrology
- Precision Manufacturing

'Design for Precision' here is meant as total design including materials, mechanics, electronics, control, thermo-mechanics, dynamics and software. It may be indicated as high precision mechatronic design. Design for higher precision is becoming more and more important due to the rapidly increasing need for high accuracy machines, instruments and consumer products. [McKeown, 1987], [Evans, 1989], [Gardner, 1991], [Nakazawa, 1994]. Today, this tendency is highly influenced by developments in computer technology, data processing and data storage. This process started in 1958, when the integrated circuit was invented. The need to have more and more transistors on a chip [Hutcheson, 1996] demanded special machines with an extremely low positioning uncertainty of a few nanometres. An example of such a machine is the waferstepper, used to position the image of the IC on the surface of a silicon wafer. Such a machine can be achieved only by highly developed design- and manufacturing technologies. Also the increasing need for high density optical recording systems (DVD), and as a result for optical disk mastering systems, asks for advanced machines with nanometre uncertainty. The parts used in these machines, like bearings, drives and beam shaping optics, have to be manufactured to submicron accuracy. For the manufacturing process, machines with submicron even down to nanometre accuracy have to be developed.

In Metrology, the technology for high precision measurements is developed e.g. measurement software, error modelling [Soons, 1993], measurement techniques and measurement strategies. To measure parts and products with sufficient accuracy, measuring machines for submicron to nanometre accuracy have to be developed, demanding new design skills since existing design-methods for high precision hardly allow to reach these levels [Teague, 1989, 1997]. Therefore, Metrology as a basic discipline has faced an enormous growth resulting in accurate co-ordinate measuring machines, laser interferometers and nanosensors such as STM and AFM. Additionally much analysis software and software for error compensation was developed and implemented.

Precision Manufacturing concerns the realisation of products with high shape accuracy and surface quality. The accuracy may be at the nanometre level, so both machine design and process behaviour must be well understood as well as the interaction between process and machine, i.e. the interaction between tool and workpiece. Several machining techniques can be mentioned here like diamond turning, grinding, lapping, honing, polishing, ion- and electron-beam machining and chemical machining. Excellent overviews about machines and machine techniques are given by [Gardner, 1991] [Nakazawa, 1994], and [Taniguchi, 1996]. New developments in this field are in the nanometre region [Stix, 1996].

Although many interesting examples can be given in the field of precision metrology and manufacturing, this paper focuses on the 'Design for Precision' as a basic activity in Precision Engineering. This discipline has a significant history and has several origins. Nevertheless it is clear that the development of astronomy and metrology had a major influence on the development of 'Design for Precision' in the early days. An excellent overview about historical developments is given in [Evans, 1989]. The 19th century brought many inventions, especially in the field of design. Through the development of linear and circular dividing machines much knowledge was gained. Many precision machines were designed and built using famous design principles like 'kinematic design' and the 'Abbe-principle'. In the 20th century the development of knowledge of design was further increased, stimulated by the development of all kinds of measuring instruments and precision machines. An outstanding example of precision machine design in the USA is the Large Optics Diamond Turning Machine (LODTM) [Donaldson, 1983]. Also the excellent design work of Bryan should be mentioned here. He contributed to several projects like the 84-inch LDTM [Bryan 1979a]. More recently is the development of the Molecular Measuring Machine [Teague, 1989, 1997], shown in fig. 1.1.

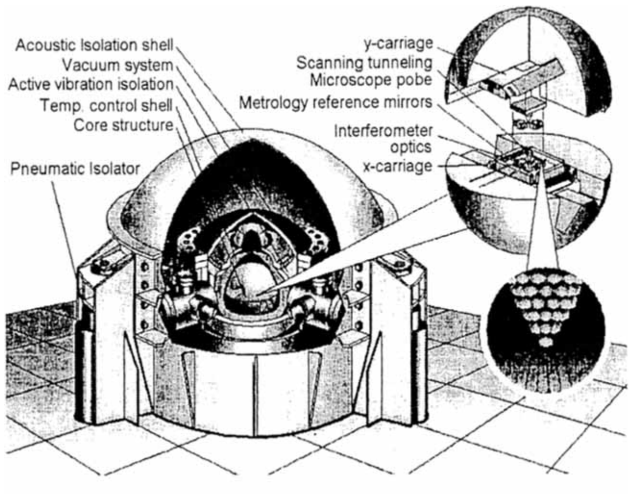

Fig. 1.1: Molecular Measuring Machine

In Europe, the Cranfield Unit for Precision Engineering and Rank Taylor Hobson developed a wide range of high precision machines including the 'Nanocentre' [McKeown, 1990] while at Philips Research Laboratories (Netherlands) several high precision machines for internal purposes have been realised since the early 1950's [Gijsbers, 1980]. In Germany and Switzerland, high precision measuring and manufacturing machines were developed by Zeiss and GSIP, the latter going back as far as 1875.

Japan also has a long history in the development of high precision machines and instruments. Nowadays, Japan plays a major role in this field [Nakazawa, 1994]. An excellent overview about the 'Japanese State of the Art' is presented by Taniguchi in the book 'Nanotechnology', [Taniguchi, 1996]. As an example the Canon Super Smooth Polishing machine (CSSP) may be mentioned here [Ando, 1992].

So we may conclude that there is an increasing need for high precision machines and products. The trend towards higher precision was shown by [McKeown, 1987], [Evans, 1989] and more specifically for machining accuracy by [Taniguchi, 1983] and [Taniguchi, 1996]. His famous graph predicting machining accuracy is depicted in fig. 1.2.

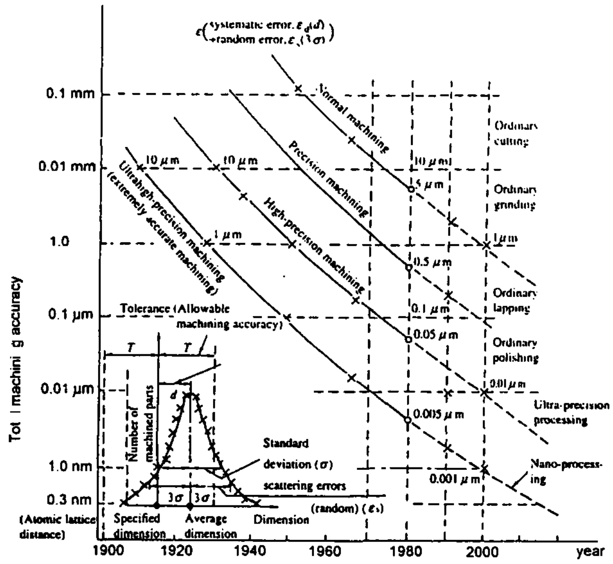

Fig. 1.2: Trends in precision machining by Taniguchi

Current trends are well predicted in this graph. Future trends in Precision Engineering will mainly be determined by trends in IC technology, information displays and storage, biomedical engineering, MEMS and consumer products needs. Developments will continue at a fast pace, spurred on by micro-engineering and nanotechnology developments so the need for high precision machinery will increase in future.

'Design for Precision' will play a major role in the development of future high precision products and machinery. Designs will be realised in multidisciplinary design groups using total design methodology. Due to the increasing cost of advanced designs, the design must be 'right the first time'. Therefore 'Predictive Design' is a necessity. This paper will summarise basic information on 'Design for Precision' and indicate the state of the art and future trends in this important area of Precision Engineering.

2. Elements of precision

In precision instruments and machines many parts interact to achieve a final accuracy. Each part contributes to the total accuracy due to deviations caused by geometric, kinematic and dynamic effects. Although in practice the interaction between these effects plays an important role in the overall system behaviour, here we will focus on these effects separately.

2.1 Definitions

Throughout this paper the terms instrument and machine will both be used to indicate an apparatus. Metrological terms are defined according to the 'International vocabulary of basic and general terms in metrology'. In 'Design for Precision' concerning machines and instruments rather than pure metrology, accurate positioning and machine tool operations are key items. Therefore, definitions given below, are extended from the International vocabulary mentioned above:

- Accuracy of an operation: closeness of the agreement between the actual value resulting from an operation and a target value of the quantity. Accuracy is a qualitative description.
- Uncertainty of an operation: parameter, associated with the result of an operation that characterises the dispersion of the values that could reasonably be attributed to the quantity.
- Resolution: smallest difference between indications of displaying device that can be meaningfully distinguished.
- Repeatability (of results of operations): closeness of the agreement between the results of successive operations of the same quantity carried out under the same conditions.
- Reproducibility: closeness of the agreement between the results of operations of the same quantity carried out under changed conditions.

Other definitions, for measuring as well as manufacturing machines are given in [ASME B89.4.1-1997] and [ASME B5.54, 1993] respectively, and quantitative descriptions are given in ISO guidelines. Bryan in his tutorial on 'Axis of rotation' describes practical verification methods for precision spindles and shafts, starting from the late 1930's till recently [Bryan, 1996].

2.2 Geometry

In the initial design of machines and instruments, geometry results from the designer's intentions on the functions that the machine has to perform. In that first stage, geometry usually consists of primitive shapes, i.e. cylinders or tubes for shafts, beams or closed box structures for supporting loads and flat or cylindrical surfaces for guiding parts. In practice however, these ideal shapes cannot be reproduced. Lines will never be perfectly straight and circles never completely round due to finite machining accuracy. Here, a careful selection of the machining operation may significantly improve the accuracy of the part [Knapp, 1998]. In general, more moving axes during machining give more errors, although small movements from an additional axis might be useful for compensation of geometric imperfections.
Accuracy is not only affected by macroscopic shape errors of a part, but also by microscopic deviations, i.e. surface finish. For many applications this is an essential factor in the total performance. In contact bearings, the effect of surface finish on wear and friction is obvious. The effect of surface finish on connections between clamped parts is less obvious but essential when properties like stiffness, damping, hysteresis and thermal conductivity and diffusivity are concerned. Geometry not only changes in (mechanical) machining operations. Unless sufficient isolation is applied, (e.g. seismic, acoustic and thermal isolation) geometry will be influenced by the environment. Most material components for instance, expand and deform under the influence of temperature change, and for unsealed natural granite, the shape of structures highly depends on moisture ingress. Other environmental effects that can affect geometry are vibrations, electric and magnetic fields. Some materials 'age', frequently resulting in dimensional change (secular instability).

Deviations of perfect shapes are also introduced due to the fact that machines are assembled from many parts. Here, considerations about form- or force closed constructions, as well as the choice between monolithic structures and bolted or glued assemblies are essential. In the case of assembling, parts can be machined very accurately on feature specific machines [Knapp, 1998], although hysteresis at the interface might negatively affect the overall reproducibility. In the classical form closed method of assembly each part must have tight tolerances, otherwise backlash occurs or, in the case of negative over-measure, a high and undefined stress level will be introduced during assembly. Force closed constructions on the other hand solve this problem by demanding statically correct connections, i.e. kinematic-, semi-kinematic- [McKeown, 1998] or pseudo-kinematic- [Smith, 1992] design connections, thereby highly reducing tolerances on geometrical shapes. Even in force closed constructions, some geometrical imperfections, like squareness and flatness errors of guiding beams, will affect the overall accuracy. However, these errors are reproducible and might therefore be reduced by software compensation.
Due to the fact that the mechanical structure of a machine has a finite stiffness, the geometry will change under loading. Especially if the load changes, both in place and size, as is the case with moving carriages, this can severely affect machine performance. With correct models, these errors can be predicted and compensated for [Spaan, 1995].

Another problem related to geometry, is the fixation of the workpiece itself. Both for manufacturing and measuring machines, the workpiece must be fixed in such a way that it is not significantly deformed by the fixation itself. At the same time the piece must be rigidly connected to the machine frame or table. Also, especially in the case of manufacturing, thermal expansion of the workpiece must be possible without intolerable stress levels. Comparable problems, related to fixation, apply to the mounting of sensors in high precision instruments. This is where kinematic- or semi-kinematic design has an important application.

2.3 Kinematics

Machines are usually not static. Different parts perform different motions, to be described by kinematic relationships. These mathematical descriptions of constructions and mechanisms only describe what happens in theory, based on theoretical lengths, positions and set-point curves. In practice however, these factors are realised with a finite accuracy and therefore, the path followed will be different from the ideal one with respect to its form, speed and acceleration.

In modern machines, displacements are often generated by a combination of mechanical parts, such as actuators and sensors under servo control. Properties like actuator power and speed, sensor resolution, control strategy and mechanical reproducibility together determine the accuracy of the prescribed path. In the case that more than one axis is under control, the synchronisation of axes is another factor that affects accuracy. An example is the case of milling a circular contour by simultaneously controlling two orthogonal linear axes.

2.4 Dynamics

The fact that machines are not static but contain accelerating parts, means that dynamic effects can play a major role in their behaviour. An effective way of minimising the effect of acceleration on relative place uncertainty is by choosing appropriate motion profiles, e.g. set-point curves that contain no jump in the second derivative, like an inclined sine function instead of a parabolic one. Preventing 'play' or lost motion is also very effective in reducing dynamic positioning errors. The parts themselves can also be designed for minimum forces. In case of rotating parts this means, that symmetry is important for reducing unbalance and also the mass moment of inertia should be minimised. In case of translations the mass should be kept small and driven as close through the axes of reaction as possible [McKeown, 1973-1997] (see section 3.6).
Another factor that determines the response of a machine to dynamic forces is stiffness. For minimising forces and maximising stiffness in general, it is not only the amount and sort of material that matters but how it is distributed. Often dynamic disturbances come from outside the instrument, like floor and acoustical vibrations. In these cases, the stiffness to mass ratio again is essential in minimising the response to these inputs. Isolating the machine from the disturbance (see section 5.7) can reduce the input itself.

3. Design principles

Design of high precision machines has been analysed by a number of people. Pollard described the mechanical design of scientific instruments in his Cantor Lectures [Pollard, 1922]. Loewen has put forth a list of major principles [Loewen, 1980]. McKeown has defined the 'Eleven Principles and Techniques' in [McKeown, 1986, 1987, 1997]. Teague and Evans have set out basic concepts, published as twelve 'Patterns for Precision Instrument Design' [Teague, 1989-1997]. On the basis of these surveys, section 3 and 5 have been set up.

3.1 Rules of Abbe and Bryan

The Abbe principle was first published in 1890 in [Abbe, 1890]: 'The measuring instrument is always to be constructed that the distance being measured is a straight line extension of the graduations on the scale that serves as the reference...'. The principle is also known as 'the principle of alignment' [Rolt, 1929], 'the Abbe comparator principle' [Reindl, 1967], and 'the first principle of machine design and dimensional metrology' [Bryan, 1979c]. As a restatement to cover those situations where it is not possible to design 'in line', Bryan defined a generalised Abbe principle as: 'A displacement measuring system should be in line with the functional point (i.e. centre of stylus ball or tool tip) whose displacement is to be measured. If this is not possible, either the slide ways that transfer the displacement must be free of angular motion, or angular motion data must be used to calculate the consequences of the Abbe offset'.

Another basic principle of measurement is the Bryan principle [Bryan, 1979a], formulated as: 'A straightness measuring system should be in line with the functional point at which straightness is measured. If this is not possible, either the slide ways that transfer the measurement must be free of angular motion or angular motion data must be used to calculate the consequences of the offset'.

Vermeulen developed a 3D-CMM, (see fig. 3.1), in which, by using intermediate bodies (A and B), Abbe errors are prevented in all three directions in the horizontal mid-plane [Vermeulen, 1998]. This machine satisfies the Bryan principle as well, making the machine non-susceptible for straightness errors of certain guides.

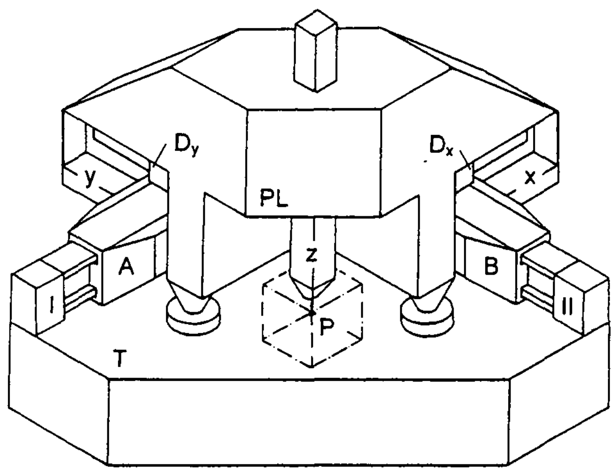

Fig. 3.1: 3D-CMM satisfying Abbe- and Bryan principle to a high degree

3.2 Kinematic design

Maxwell describes kinematic design as: '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' [Maxwell, 1890]. Lord Kelvin established Cambridge Scientific Instruments firmly on this design philosophy in the interest of achieving highest precision at lowest cost. GSIP used it extensively, especially in metrology instruments. Pollard stressed the importance of kinematic design as opposed to the usual machine tool design of couplings in instrument mechanisms, not only to the user and the minimising of variance, but also to economic manufacture [Pollard, 1922] [Pollard, 1929-1951]. The importance of kinematic design for today's precision engineers is clearly described in [Blanding, 1992]. McKeown emphasises its importance in his 'Eleven Principles'. Teague states it as part of his patterns. Less known world-wide is Van der Hoek, who was both an employee of Philips Electronics and professor at the Eindhoven University of Technology from 1962 till 1985. His lecture notebook contained about 200 examples of sound versus poor design, in which kinematic design is a key to solution [Hoek, 1962-1986], [Hoek, 1985-1989]. Other books that describe kinematic design are [Slocum, 1992], [Smith, 1992], [Nakazawa, 1994] and [Koster, 1996].

Kinematics evolved from mathematics and is hence somewhat 'idealised' in its components. e.g. rigid bodies, straight lines, perfect circles and 'point contacts' etc. Nevertheless it is a good starting point for mechanical design that is correct in principle. The basis for kinematic design is constraining the right number of degrees of freedom (d.o.f.), i.e. translations or rotations. For constraining one d.o.f. for rotation, necessary in e.g. coupling elements in feed drives, bellows are typically used [DeBra, 1998]. In fig. 3.2 some examples of fixing one d.o.f. for translation are given. The classical solution is the use of a slender rod (fig. 3.2a). Due to its finite length, sideways displacements result in a parasitic motion in the constrained direction. The same function is performed without this disadvantage by a folded leaf spring (fig. 3.2b). In fig. 3.2c an alternative consisting of 4 rods is shown.

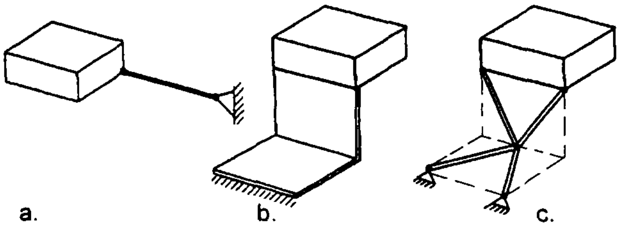

Fig. 3.2: Single d.o.f. constraints for translation

Fig. 3.3 shows some examples for constraining 2 d.o.f. for translation. This can be achieved by 2 rods (fig. 3.3a and b) or a hinged leaf spring (3.3c).

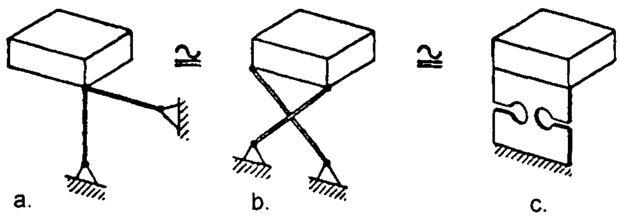

Fig. 3.3: Two d.o.f. constraints for translation

Two d.o.f. for translation and one for rotation are constrained by for example 3 rods (fig. 3.4a and b) or a normal leaf spring (fig. 3.4c).

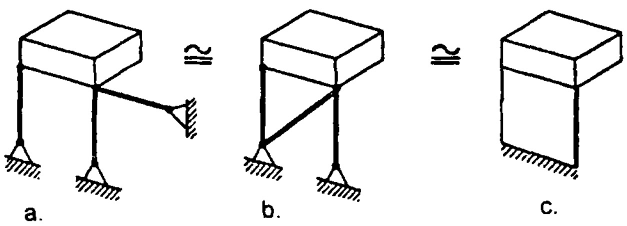

Fig. 3.4: Two d.o.f. constraints for translation, one for rotation

With combinations of these basic elements kinematic mechanisms or fixtures can be constructed. Fig. 3.5 gives an example of a fixation for a table-top. Due to the application of three hinged leaf springs, the table-top is constrained in 6 d.o.f., with a thermal centre at the intersection of the normals to the leaf springs.

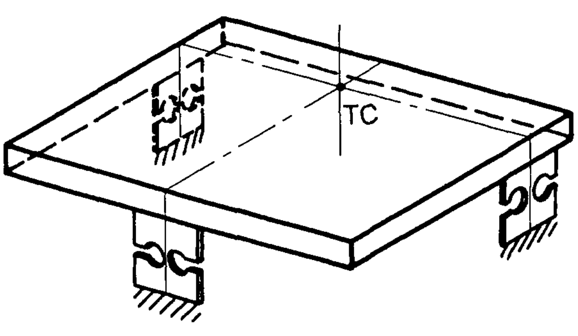

Fig. 3.5: Six d.o.f. constraints with thermal centre (TC).

Fig. 3.6 shows how an xyθ-stage can be constructed from three folded leaf springs.

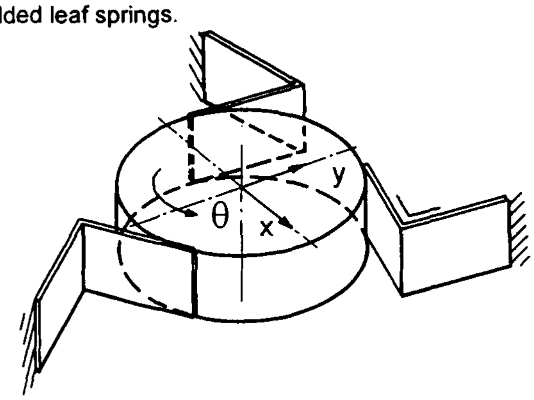

Fig. 3.6: xyθ-stage with three folded leaf springs

An illustrative example of the application of kinematic design is the kinematic support shown in fig. 3.7. The classic version with six fixed supporting surfaces is enhanced by making the surfaces flexible around an elastic hinge [Schouten, 1997]. In this way the friction force along the surface is greatly reduced while the contact stiffness normal to the surface is only slightly reduced. Due to the improved ratio between stiffness and friction, the hysteresis (see section 5.1)was reduced from 0.42 for the classic non-hinged version to 0.03 μm for the new version.

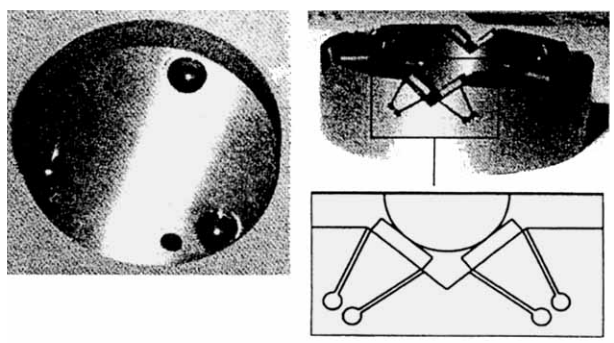

Fig. 3.7: Symmetrical kinematic support with TC, where friction is recognised as principal cause of hysteresis

3.3 Thermal loop

A thermal loop is defined as: 'a path across an assembly of mechanical components, which determines the relative position between specified objects under changing temperatures'. In principle, a thermal loop should be as small as possible to minimise the influence of spatial thermal gradients. Thermal expansion in a thermal loop of a machine can be compensated mechanically by adapting the effective length of machine components and by choosing materials with the appropriate coefficient of thermal expansion, e.g. Harrison's 'grid iron' pendulum [Evans, 1989]. The fixed point or axis, from where components expand with respect to each other, can be chosen by creating a thermal centre (TC), such as in fig. 3.5. Although coefficients of thermal expansion are not better known than within $0.5 \cdot 10^{-6}$ °C^{-1} [Breyer, 1991], the influence of thermal expansion can be reduced by measuring expansion of machine components due to controlled temperature changes [Kunzmann, 1988] and subsequently creating equal thermal lengths by choosing the proper fixed point.
To obtain thermal stability of ± 0.5 °C/day in an air-conditioned assembly hall and ± 0.1 °C/day in a precision-climatised cabin is quite a problem [Breyer, 1991]. Heat sources localised internally or externally of the machine

lead to changes in the temperature profile in the machine. This may cause unequal thermal expansion rates in a thermal loop, due to different thermal time constants of machine components, (see section 5.5). Hence Donaldson strongly recommends, as a principle in his publication about Machine Tools [Donaldson, 1980] to take the heat out at the source. Wetzels experienced a thermo-mechanical stability problem with an integrated heat source. After having removed this source thermal drift was reduced with one order [Wetzels, 1998].
Internal heat disturbances have responses that can be middled and compensated partially (See section 6). However environmental temperature variations can only be responsive and not predictive, because the input is unknown [de Bra, 1998].

3.4 Structural loop

According to [ANSI, 1992] 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 a cutting tool and a workpiece: the structural loop includes the spindle shaft, the bearings and housing, the slide ways and frame, the drives, and the tool and work-holding fixtures'. All mechanical components and joints in the propagation path from the drive to the point of reaction, e.g. the end-effector (cutting tool or probe) or the centre of gravity, must have a high stiffness to avoid deformations under changing load. The design of a machine or an instrument contains one or more structural loops.
Essential in recognising good structural loop design is the split-up in series and parallel paths. Along a series path stiffness should not change to abruptly. Improvement of the series path is possible by stiffening the most compliant part preferably by 'transferring' material from more rigid parts. Parallel path improvement is in contrary done by improving the stiffest part further, preferably –for a system with equal mass– at the cost of more compliant parallel paths.
Due to physical limitations, the measuring system of a closed loop system is unavoidably located at a certain distance from the end effector. In addition to good structural loop design, the path between measuring system and end-effector has to be as stiff as possible to minimise deviations, e.g. by minimising the length of this path, called the 'measuring circle' [Kunzmann, 1996].

3.5 Metrology frame

A metrology frame is a reference frame for displacement-measurements, independent of the machine base, i.e. the external forces upon the metrology system must be constant [Bryan, 1979b]. DeBra suggests to see the metrology frame as an example of a broader principle, i.e. the principle of 'separate functions' [DeBra, 1998]. In fact the routes for force and position information are disconnected, an idea which is present as well in the design of the rotary table of fig. 5.8 [Philips, 1994].
In [Teague, 1989-1997] historical applications of the 'metrology frame' are discussed, to overcome the problem of deformation of machine parts. The first example of a metrology frame found so far is on the Rogers-Bond Universal Comparator [Rogers, 1883]. More recent examples are found at NIST in measuring machines by Hocken et al, at NPL in interferometric time standard comparators, at LLNL in the 'Ultimat' CMM [Bryan, 1979b], the 84" SPDTM [Bryan, 1979a] and the LODTM [Donaldson, 1980], and at Cranfield Precision in the Nanocentre by McKeown et al (see fig. 3.8) and Wills-Moren [Wills-Moren, 1982] and [Wills-Moren, 1989].

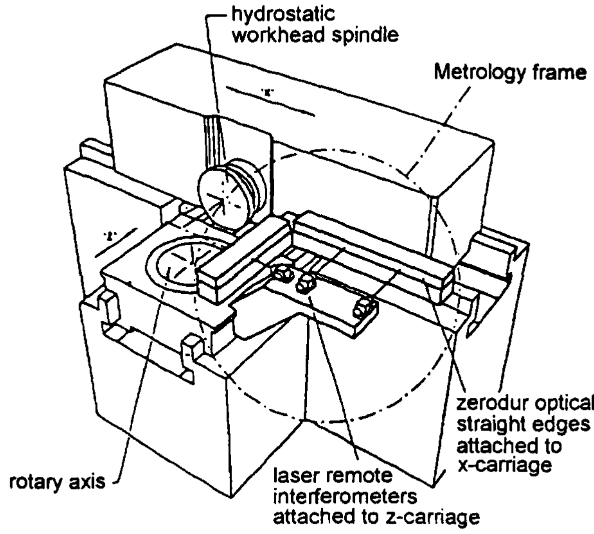

Fig 3.8: Nanocentre, using a Zerodur® metrology frame for complying with Abbe in two axes and direct measurement of horizontal straightness error motion.

[Teague, 1989-1997] advises to make the metrology frame as small as possible, to minimise environmental influences. [Bryan, 1979b] suggests either to build the metrology frame in zerodur or to use temperature control (e.g. an oil shower). Furthermore the plane of supports of the metrology frame should coincide with the bending neutral axis of the machine base.

3.6 Drive offset

By combining sound mechanical design with closed loop control, enhanced operating speed, accuracy and flexibility of movement can be achieved. Typical examples are the Compact Disc player, the waferstepper, advanced CNC milling and turning machines and fast component mounting (FCM) machines. In the development of servo-controlled positioning devices it is essential to consider the place where the actuator force is loading the slide to supply the (resulting) force generated by inertia, tool- or measuring forces, friction, etc. As stated in one of the 'Eleven Principles' [McKeown, 1986, 1987, 1997], drives should be placed to operate through the axes of reaction. If this is not possible, the deviation from the axis of reaction –called drive offset– induces moments on the machine slides. The effect of resulting rotations of a slide on it's controllability is minimised, if both the drive- and measurement-axis are at the same side of the centre of rotation [Rankers, 1997].

3.7 Force compensation

3.7.1 Weight compensation

In many 3D-CMM's vertical rams are used. In order to avoid loading the vertical slide's driving system with the (constant) weight of the ram, force compensation might be applied, thereby eliminating undesired heat dissipation in the motor. Constant forces can be obtained in many ways, e.g. using additional weights, which is unfavourable from a dynamic point of view. Alternatives are magnetic fields, cylinders under pressure or vacuum, or 'constant rate springs', e.g. tensators [Tensator, 1997], [Rosielle, 1998]. Depending on the stroke of the slide and the specifications on admissible force variations, one type of weight compensation is more suitable than another. Weight 'relieving' in which Coulomb friction is reduced on guide ways, was used in many high precision machines designed by GSIP.

3.7.2 Reaction force compensation

Due to a limited mass and support stiffness of stationary machine parts, reaction forces as a result of a driving forces, cause a motion of these stationary parts [Rankers, 1994], [Rankers, 1997]. Excitation of machine frames caused by reaction forces becomes increasingly important in situations of direct drive actuation at high frequency, e.g. fast tool servo cutting [Patterson, 1985] and cutting of non-rotational symmetric surfaces [Renkens, 1997]. In addition to commonly used approaches to reduce the motion of the frame, such as increasing stiffness and mass of the frame, or increasing the (active) damping, Rankers mentions more fundamental approaches [Rankers, 1997]. A first example is to accelerate a counter mass between the load and the frame in opposite direction [Weck, 1995b] and [Weck, 1997]. Secondly a simultaneous counteracting force can be applied to the frame by a second motor without tight requirements with respect to positioning accuracy.

3.7.3 Parasitic stiffness compensation (negative stiffness)

Instruments and mechanisms based on elastic elements have the advantage that actual backlash and friction, causing virtual play [Koster, 1996] are absent. Sizing of flexures is based on allowable stress in combination with desired stroke, considering manufacturability and necessity to suppress the fixed motions rigidly. However, the stroke is restricted by the elastic limit of material and the stiffness of the elastic elements results in an opposing force linear proportional to the displacement. In those instances where the driving force becomes too high to handle, be it through the sheer size of actuators required or their heat production as a disturbance to the machine functions, either flexures have to be discarded as a design option or this undesired effect of force buildup has to be compensated. This is best done with passive elements for obvious reasons. The classical approach is referred to as creating 'negative stiffness'. With a device also containing flexures and rigidly connecting this with the rotational or translational flexure stage so as to obtain a near zero stiffness design, this problem can be overcome at the cost of increased complexity. Van Eijk gives several examples of creating negative stiffness [Eijk, 1985].

3.8 Symmetry

In [Teague, 1989-1997] it is recommended to incorporate symmetry to the maximum extent possible in properties of machine elements (e.g. mass- and force distribution or stiffness), the entire instrument and properties of the environment. In designing, manufacturing, assembling and operating a precision instrument, any departure from symmetry has to be weighed against the resulting compensation to overcome the problems produced by the asymmetry. To avoid thermal asymmetry, inducing significant distortions of the machine components, a thermal centre as symmetry axis for thermal expansions (see section 3.3) can be applied [Vermeulen, 1998]. To overcome the effect of asymmetry about a horizontal plane, caused by gravitational forces, machines can be equipped with a vertical axis, e.g. the LODTM, [Donaldson, 1983]. Three-dimensional symmetry is superbly achieved by a tetrahedral structure, e.g. the Tetraform by Lindsey of NPL [Lindsey, 1988], [Slocum, 1992], [Corbett, 1997]. In addition to its proponents Hocken mentions some adversaries of symmetry [Hocken, 1995], e.g. vibrational energy is not reduced by symmetric design, in fact it is often enhanced.

3.9 Repeatability

According to the definition, given in section 2, repeatability means that the results of the same machine action carried out under the same conditions are equal. The machine action can either be a measurement on a CMM or manufacturing a product on a machine tool. [Bryan, 1993] propounds determinism i.e. the deterministic performance of machines under automatic control, as a sound basis for design, build and performance testing: 'The basic idea is that automatic machine tools and measuring machines are perfectly repeatable just like the stars*. They obey cause and effect relationships that are within our ability to understand and affordably control. There is nothing random or probabilistic about their behaviour. Everything happens for a reason. The list of reasons is small enough to manage by common sense, good metrology, and a reasonable investment of resources'. Practically, design for repeatability demands:

- Application of statically determinate high stiffness design, minimised hysteresis in the parts-connections (section 5.1)
- Minimising of friction and maximising stiffness of the bearing systems, (section 5.2)
- Optimisation of performance of drives (section 5.3) and control systems (section 5.4)
- Consideration of sensor qualities including sensor mounting (section 5.5)
- Attention to the thermal stability of the design and adequate isolation against vibrations (section 5.6).

Repeatability is essential to predictive modelling (section 4). The closer the instrument is resembled by the model, the better the prediction of it's behaviour and the more scope for software error compensation (section 6).

* Here Bryan is quoting [Loxham, 1970]. He goes on to summarise seven exceptions to the deterministic nature of classical physical laws, according to Kidder (LLNL) and Hocken [Bryan, 1993]. These exceptions apply to molecular and atomic size masses and have no practical significance in the field of manufacturing.

4. Modelling for predictive design

As indicated in section 1, there will be an increasing need for ultra precision machinery in the near future. As a result, design approaches are changing dramatically in this field. The complete understanding of the behaviour of precision machines is necessary to predict the component dimensional errors. By adding all the component errors together in an error budget, the machine designer can make a prediction of the accuracy of the overall machine [Blaedel, 1998], [Thompson, 1989]. A very good overview of nowadays machinery and processes is given in the recently published book 'Nanotechnology' [Taniguchi, 1996]. As the development of ultra precision machinery generally is very expensive, 'right-the-first-time'- design is getting more and more important. Although extensive analysis of design concepts is expensive, a lot of money can be saved by systematic analysis early in the design process. Accuracy of machines and instruments is mainly determined by five error sources, i.e. deviations concerning kinematic-, thermo-mechanic-, static- and dynamic- and control system behaviour (see fig. 4.1).

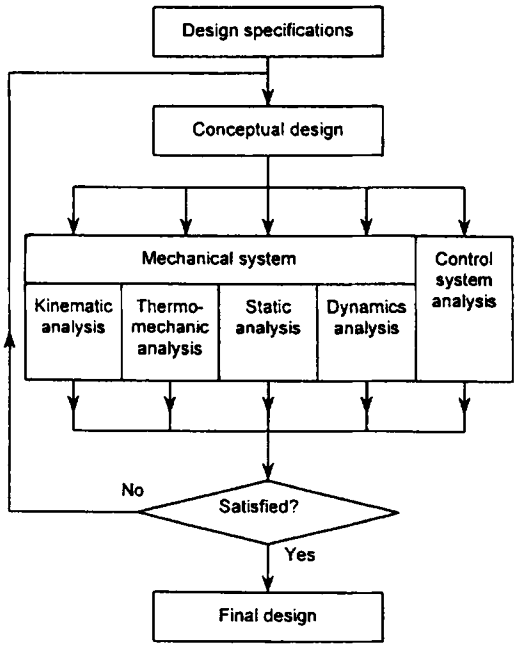

Fig. 4.1: Design flow chart for (components of) machines and instruments

Starting from the design specifications, one can create a conceptual design, which must then be analysed by modelling and simulation and improved. A flowchart can be drawn, showing the kind of modelling- and simulation activities, necessary to check the conformance to it's functional specifications. In addition to kinematic models for estimating effects of motion errors, FEM part modelling can be used to study (thermo)-mechanic- and static aspects including thermal expansion and diffusivity respectively stiffness and strength. For both the first approximation and necessary validation, calculations based upon simple elastic and thermal theories can be used. Dynamic aspects of the mechanical system concerning effects of both inertia and stiffness have to be analysed as well. Additionally the control system and the mechanical system have to be tuned in order to get good dynamic behaviour of the closed loop system. Fig. 4.1 shows design to be an iterative process. The further the design process evolves, the more detailed models and simulations get. Hereafter, the most important analysis activities and approaches are discussed in brief.

4.1 Kinematic analysis

Having developed a design concept for a precision machine or measuring instrument, error modelling strategies, being developed mainly during the last two decades, can be used to predict kinematic errors [Soons, 1993], [Krulewich, 1995a,b]. These modelling techniques are based on estimating the effects of motion errors of e.g. linear carriages, rotary tables and spindles. Successful use of lumped parameter models to predict and compensate for bending deflections were used as early as 1972 [Wills-Moren, 1982]. For each axis i, motions errors can be described by means of a translation- ${}_i\vec{T}$ and rotation ${}_i\vec{R}$ error vector. The effect of rotations of carriages at specified positions in the structural loop may be calculated by the outer vector product of rotation vector ${}_i\vec{R}$ and local position vector ${}_i\vec{P}$, being defined in the Cartesian co-ordinate system of specific carriage. Here, the origin of the co-ordinate system is connected to the sensor which is reading the scale position of belonging axis i. Vector ${}_i\vec{R}$ contains a fixed contribution from squareness errors between machine axes as well. The total kinematic error at a specified machine position may be described by an error vector to be calculated from contributions of all axes [Spaan, 1995]: $d\vec{P}(x,y,z)=\sum_i\left({}_i\vec{R}\times{}_i\vec{P}+{}_i\vec{T}\right)$.

Fig. 4.2 contains a schematic representation of this modelling approach for one axis i.

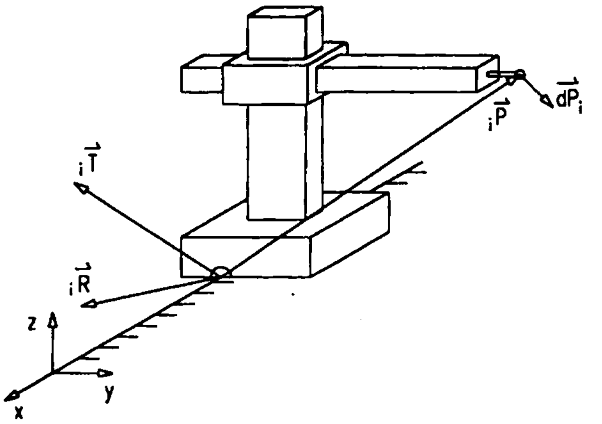

Fig. 4.2: Modelling approach for kinematic errors

Kinematic modelling must be applied to the subsequent structural loops of a machine, i.e., one can start at the workpiece position passing through all the elements and joints of machine carriages, ending at the probe- or tool position. Rotary tables and spindles are treated in the same way as linear carriages. In [Spaan, 1995], the modelling technique has been applied to a five axis milling machine. Its relatively simple approach rapidly offers the designer information about the effects of a design proposal for a change in the structural loop.
Another modelling method based on a description of kinematic errors using homogeneous transformation matrices is widely used as well. Here, the error parameters ${}_iR_j$, ${}_iT_j$ are used as matrix elements [Paul, 1981], [Soons 1993], [Portman 1997]. In the latter, a description of second order effects for high precision applications is presented as well.

4.2 Thermo-mechanic analysis

As indicated by many authors, thermal effects may have a significant contribution to the uncertainty of precision machines and instruments [Yoshida, 1967], [Camera, 1976], [Attia, 1979], [Balsamo, 1990], [Cresto, 1991], [Schellekens, 1992]. An excellent overview of work in this field is given by Bryan et al in their keynote papers for CIRP - General Assemblies of 1968 and 1990 [Bryan, 1968], [Bryan, 1990].
Generally, the temperature profile in a machine can be described by an instationary function $T=T(x,y,z,t)$. As a result of these spatial and temporal gradients, in combination with thermal properties of the material used, shape and size of a machine will change in time. Especially the instationary behaviour makes error estimations very difficult to describe. Therefore, numerous attempts have been made to describe (quasi) stationary situations $T=T(x,y,z)$ [Soons, 1993], [Trapet, 1997a]. Here, as a first approach, error modelling is mainly based on expansion and bending of (large) machine elements in the thermal loop, considering a stress-free deformation (otherwise modelling becomes very difficult) [Boley, 1960],

[Trapet, 1997a]. After having calculated expansions and deformation, the kinematic model discussed in section 4.1 can be used to calculate the thermal error vector $d\bar{P}(T(x,y,z))$. In [Soons, 1993], this approach was successfully applied to a milling machine, giving a thermal error reduction of about 70%.
As machine tools unavoidably have internal heat sources, it may sometimes be necessary to use instationary descriptions $T = T(x,y,z,t)$. From this relationship, thermal effects on the tool position relative to the workpiece can be calculated. Nowadays, finite element (FE) and finite difference (FD) modelling techniques are widely used [Soons, 1993]. Soons used state-space models as well, to predict the temperature field in a five axis milling machine due to heat generation in the main spindle drive. However, there are still difficulties in specifying thermal boundary conditions. In contrast to disturbances as a result of internal heat sources, the modelling of environmental effects, such as door opens, can only be responsive, not predictive.

4.3 Static analysis

The structural loop of machines and instruments can be affected by (quasi) static forces, e.g. changing weight forces of slowly moving machine parts, (nominal) machining forces [Spaan, 1995], and forces as a result of cabling, air- and vacuum hoses. Acceleration forces, having much higher frequencies, will be considered in the next paragraph.
Due to finite stiffness of machine components, e.g. beams, spindles, carriages, including bearing systems and joints, the forces mentioned above will cause positioning errors at tools or probes. As a first approach, deviations can be estimated fairly easy by means of stiffness calculations based on simple linear elastic theories or theories about Hertzian contacts. Nowadays advanced software packages are available, such as Unigraphics®, I-DEAS®, Algor®, Pro-Engineer® [FEM, 1998] to be used for both linear and non-linear analysis at e.g. complex plate frame type structures, beams, trusses and 3D solids, made of homogeneous isotropic as well as composite materials (see section 5.1). Interesting approaches in this field are reported in [Reinhart, 1997], which describes 3D-CAD with integrated FEM analysis early in the design stages, called 'Virtual Prototyping'.

4.4 Dynamic analysis

Since machine structures generally are assembled out of many different components, to be seen as a combination of masses and springs, the total structure will behave according to an interaction of these elements [Timoshenko, 1974]. Many good textbooks are published about this subject, e.g. [RaO, 1990].

Since most driving systems do not act in line with the centre of mass, inertial forces cause rotations of machine parts, mainly due to finite stiffness of bearing systems and joints. In high precision machines, such as 3D CMMs, even low accelerations values may have significant influence on measuring accuracy [Weekers, 1997].

Machine dynamics may have significant influence on system performance. Position- and tracking accuracy as well as on smoothness of motion are highly reduced by mechanical step response in the structure. In addition to nominal acceleration levels as a result of inertial effects of slides and spindles, vibrational accelerations may continue after an external force or a set point motion profile has been applied to a mechanical component. In addition to the way the set point is applied (acceleration or 'jerk'), the vibrational behaviour highly depends on the system's natural frequencies and the amount of damping (see section 5.1.2).
In order to foresee deviations as a result of internal vibrations, it is of vital importance to model and analyse machine dynamics, i.e. to find the lowest natural frequencies and vibration modes of the mechanical structure in the relevant frequency range (limited by several times the bandwidth of the servo system). Here, lightweight- and (dynamically) stiff design is particularly important for those machine components that determine the lowest natural frequencies and the static stiffness.
Even reaction forces of actuators on machine frames may cause intolerable deviations, especially for high precision machines, such as wafer steppers or SPDT machines using fast tool servo systems for the manufacturing of non-rotationally symmetric parts. As the mass of the frame is limited and its connection to the world suffers finite stiffness, resonance due to reaction forces may occur (see section 3.7) [Weck, 1995b] [Weck, 1997], [Renkens 1997], [Rankers 1997]. A schematic diagram summarising the indicated effects is given in fig. 4.3.

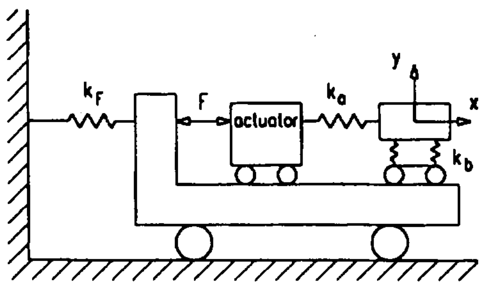

Fig. 4.3: Simple dynamic model of driving system with frame (k_F), actuator (k_a) and bearing stiffness (k_b) (no damping considered).

For multibody systems like manufacturing machines and wafer steppers, modelling and evaluating the machine as a whole may be inefficient and excessively time consuming. Therefore, [Rankers, 1997] proposes to divide the total system in substructures and components successively, and model and analyse them separately. Afterwards these models are added to an overall system model, to be used for mode shape analysis resulting in natural frequencies and vibration modes of the overall system.
Dynamic models, such as lumped mass models, are very helpful to analyse the dynamic behaviour of a chosen concept. Here, the model, which consists of a number of masses connected by a series of mechanical compliances, can be described by a set of ordinary differential equations. Model reduction techniques [Hoek, 1962-1986], [Nijs, 1988], [Koster, 1996], [Rankers, 1997], [Rosielle, 1998] are used to diminish the complexity of the description, advantageous for mathematical treatment and interpretation of results.
For rather complex (sub)-structures 'mode shape analyses' offer an excellent tool to analyse the behaviour [RaO, 1990], [Ewins, 1984], [Rankers, 1997]. For that purpose, the static model of section 4.3 can be used, for determining natural frequencies, as well as mass, mass moment of inertia, and the location of the centre of gravity. For a ceramic plate frame type slide construction, to be applied in an SPDT machine, the result of a modal shape analysis is depicted in fig. 4.4 [Vermeulen, 1996a].

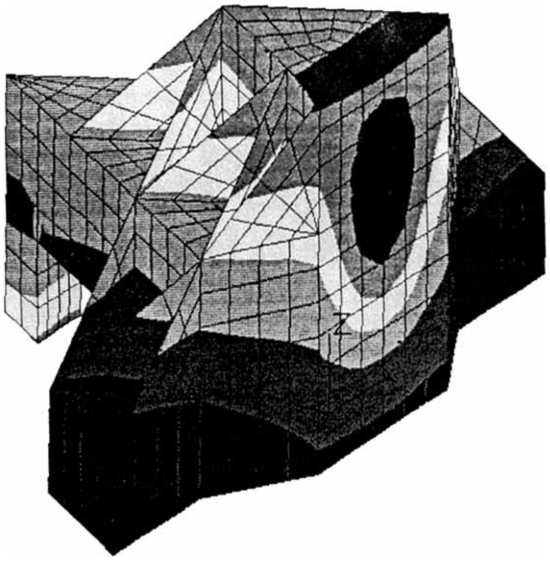

Fig. 4.4: Mode shape analysis of plate-frame-type slide construction

4.5 Control system analysis

In most cases a closed loop control system will be added to the mechanical system for positioning and multiaxis contour control. Examples of closed loop control in ultra precision machines are SPDT machines, ceramic grinding machines operating in 'ductile mode' as for mag-disc substrates, wafer steppers, and optical disk mastering machines. In order to get good dynamic behaviour, the control strategy should be matched very carefully to the dynamics of the mechanical system. The optimised design of the control system highly depends on the behaviour of the mechanical system, e.g. its natural frequencies, the presence of friction and other disturbing forces.

Generally the control system contains position and velocity feedback, and sometimes for the very highest performance, velocity and acceleration feedforward loops are added as well (see section 5.6). As indicated in [Groenhuis, 1991], the mechanical design mainly determines the order of the controlled system. The more degrees of freedom (determined by the number of lumped masses and springs), the higher the order of the model. [Rankers, 1997] indicated three main sources of dynamic influence, i.e. flexibility in the actuator system, guide way systems flexibility, and mass and stiffness of the machine base. All these effects may increase the order of the controlled systems since they introduce additional degrees of freedom. As depicted in [Groenhuis, 1991] a fourth order model (two degrees of freedom) is the minimum and generally adequate order to describe the dynamic behaviour of a mechanical closed loop system.

5. Design features for precision

5.1 Stiffness of constructions and mechanisms

In addition to (dynamic) stiffness requirements of high precision components to obtain a desired kinematic function, stiffness is necessary to overcome additional side effects such as positional errors due to friction, essentially Coulomb friction. Friction forces whenever present, will load the stiffness of components to a maximum before sliding in any direction. The positional uncertainty associated is called 'virtual play' [Hoek, 1962-1986], [Koster, 1996], [Rosielle, 1998], and amounts to twice the maximum friction force divided by the stiffness. Uncertainty in position is hence uncertainty in the internal potential energy content of the construction. Designing for high stiffness can be conceived as designing for a minimum potential energy content. 'Bigger is better' might be true for stiffness reasons, for example when a force transmitting area of a cross section is concerned. But if the efficient use of material in a design is also aimed for, e.g. in high precision slide constructions or aerospace applications, the mass and thereby the kinetic energy content should be minimised as well [Riven, 1989], [Slocum, 1992], [Smith, 1992], [Rivin, 1996]. DeBra emphasises the importance for a design to get it's functional dependence right, but meanwhile to separate 'material' from 'geometry' [DeBra, 1998]. On the basis of the latter principle, this section about stiffness of constructions and mechanisms is further divided in 'Geometry at the component level' on the one hand, and 'Material selection' on the other.

5.1.1 Geometry at the component level

Certain movements are desired to obtain functions, others are preferably avoided. This because the latter result in undesired energy storage, e.g. kinetic energy, or potential energy in elasticity of material components. Vibrations are a typical example of such energies flowing back and forth. If there are many free movements within the design, energy flow becomes chaotic, different vibration modes store energy from or release energy to each other. Because this energy flow eventually may affect movements, which are necessary for functional reasons, it is most desirable only to allow functional movements and to suppress deliberately all other movements of components in a design. This calls for high stiffness in the directions in which movements are not functional. Such stiffness can best be obtained by correctly placing material in the right shape and direction, and by using as little material as possible (i.e. stiff-lightweight design), especially on functional moving parts. Stationary frames are affected to a much smaller extent by this idea, although here too, saving mass has certain benefits. A design, in which the material involved is used poorly in terms of deriving stiffness from solids, is often characterised by unfavourable loading. If the loading is very localised, large deformations and high local stresses occur, and unfavourable lever ratios result in high forces and undesirably induced bending moments (see fig. 5.2). In precision equipment, the often-unintended load is usually caused by inertial forces, moving-weight forces, and internal friction, which should be minimised. Good design has uniformly distributed loading. Ideally, the stress level under load should be the same value for all material used. However, even if there is a uniform stress level over the construction, it definitively does not mean that the structure is optimised to a point where improvement is no longer possible. One could have attained a local design optimum and a different design may simply be more favourable for the loads and purposes intended [DeBra, 1998].

Designs are either assembled from separate parts or inversely produced by removing material from a volume leaving the required components in place. Regardless of the method, the outcome can be regarded to consist of truss, beam, plate or solid volume-like parts. Solid volumes should be avoided where possible, i.e. they should be hollowed out. Stiff and lightweight structures can then be obtained by using closed boxes of plate material. These boxes should only to be joined and loaded at the ribs and corners. Large unsupported plate areas will usually resonate due to lack of out-of-plane bending stiffness in thin plates. Here thicker plates, made of material with a lower density may beneficial. Alternatively, the plates are made of a lightweight sandwich design, offering two thin stress skins separated by a shear resistant web (honeycomb or foam). Alternatively, a statically determined truss structure can be used, which offers high stiffness at the system nodes, especially for relatively large protruding- or span lengths in a construction. Optimising truss structures involves preventing sharp angles between the trusses at the joints. Tube-like trusses help both to improve buckling resistance as well as natural frequency, which is higher than for solid rods of equal mass. Tetraform is a prime example. Node connections however become more complex.

5.1.2 Material selection

The choice of materials for a machine design is a key factor in determining final machine performance. Many criteria may be considered, including temporal stability, (specific) stiffness, homogeneity, thermal expansion and diffusivity etc. Often, these criteria vary for differing machine elements and the approaches are numerous [Hocken, 1995]. Smith and Chetwynd introduce property groupings and use charts and profiles in order to compare different materials in the performance of precision instruments and machines [Smith, 1992]. Teague and Evans have listed several criteria to be considered [Teague, 1989-1997]. Criteria used for thermal loop design are described in section 3.3. For structural loop design, specific stiffness (the response to static and dynamic loading defined as the ratio of the modulus of elasticity to the density), internal damping (the specific damping capacity), and deformation under point loading (determined almost exclusively by the modulus of elasticity) are crucial. The former two are further discussed in this section.

5.1.2.1 Specific stiffness

As a general conclusion from section 5.1.1, it can be stated that good design of high precision machines and instruments, requires design for stiffness instead of admissible stress or lifespan, and as a result, for a readily optimised geometry, it is the modulus which determines selection of the material. Moreover, in situations where components are accelerated, e.g., CMMs, next generation step-and-scan wafersteppers, and SPDT machines, high moving masses cause high inertial forces as well as a substantial heat dissipation in the slide's driving system, having a negative influence on the systems accuracy. In addition, high masses lower the values of the subsequent natural frequencies. Therefore, optimising the stiffness per kg for a given geometry means selecting a material with high specific stiffness.

Metals, like cast iron, steel and aluminium, have in spite of their moderate specific stiffness, the advantage that conventional joining techniques, can be applied, e.g., welding and thread tapping. Granite, having a comparable specific stiffness, is generally used as a structural material in machines, which operate in dry environments, e.g., for filtered and dried air bearing slide ways. As granite can absorb moisture and swell, it may not be appropriate to use it in a machine where cutting fluid splashes all over it [Slocum, 1992], [Meijer, 1989]. For this reason many builders seal the granite with very thin epoxy resin.

Technical structural ceramics are commonly used in applications where high resistance is required. Ceramics like B_4C, SiC, Si_3N_4 and Al_2O_3 have high refractory values and extremely high resistance against wear, corrosion, and erosion. Until now, designing in technical ceramics meant performing calculations of stress, and stress intensity factors [Klostermann, 1987], [With, 1996]. The low toughness, in comparison to metals, resulted in a statistical approach of the fracture mechanics (Weibull statistics). Technical ceramics however, have a high specific stiffness as well (factor of 3 to 7 compared to steel), and therefore, application of technical ceramics in high precision machines is interesting [Slocum, 1994b]. However, in consequence of transformations at high temperature sintering, ceramic parts may have internal defects, and show considerable non-uniform shrinkage deformation. Therefore, uniform sheet thickness, relatively simple structures, and the absence of sharp edges and point or line forces are essential.

Vibrations caused by slide movements, or coming from a cutting process or the environment, will negatively influence performance (see section 5.7.2). Whilst the specific stiffness of massive ceramics is good, vibration damping is poor. Therefore, instead of massive ceramic structures, a laminated build-up of thin ceramic tiles is highly preferable. Here, glued lap joints are barriers for vibrations, so that damping is increased (see section 5.1.2.2). However, for stiffness reasons, a very small lap joint thickness is required, in practice mainly depending on the flatness of the tiles used. Semi-manufactured alumina substrate tiles (applied in the manufacture of IC's), having a flatness within a few 0.01 mm, enable the laminate stiffness to be mainly (>95%) determined by the stiffness of the tiles [Vermeulen, 1996a,b], [Rosielle, 1996]. Thereby, almost a factor of 3 in mass reduction is possible, compared to an aluminium or steel plate frame structure with the same stiffness.

As a result of the air space program of NASA during the 1960's, advanced fibre composites were introduced. During the last decade, new high-tech fibres like high modulus (HM) carbon, aramid, and polyethylen, as well as advanced thermohardening and thermoplastic matrices were developed. However the strength of metals can be improved with additional alloy-elements or heat treatments, its modulus of elasticity is nearly constant, except for AlLi alloys [Rivin, 1996]. Here, fibre composites differ from metals, as the former's modulus changes dramatically with the change in fibre fraction. In addition to the fibre quantity in a matrix, its orientation can be changed dependant on the load case of the structure. For a tensile-compression load, mainly unidirectional (UD) fibres can be applied. Especially UD HM carbon fibres have a substantially higher specific stiffness than conventional construction materials. However, for getting in-plane stiffness, which is required when building plate-frame type structures, only some 40% of the UD modulus remains as in-plane quasi-isotropic modulus [Lubin, 1982]. Even then, for HM carbon fibres a gain factor of 2.5 in specific stiffness can be obtained.

In [Kerstiens, 1990], structural components for high-speed grinding and milling machines (slides, spindles), and bridge-type robot centres (beams), made of carbon fibre are described and compared to metal alternatives. Dependent on the application and fibre orientation, a specific stiffness gain with a factor of 2 to 3 is possible. In addition, as a result of internal damping, an increase of the dynamic stiffness is obtained. Manufacturing costs are doubled, but for equal machine performance, costs are about the same.

Reinforced ceramic and metallic composites have performance characteristics beyond those of conventional massive ceramics or metals [Lanxide, 1990]. Reinforced composites allow for a density and thermal conductivity approximating those of aluminium, combined with steel like values for modulus and tensile strength. In addition to a high specific stiffness, reinforced composites are tough and, unlike ceramics, have tolerance for damage. Reinforced ceramic matrix composites are made by a directed metal oxidation process. Another infiltration process (without oxidation) is used to make reinforced metal matrix composites.

5.1.2.2 Internal damping

The response of a structure to a time-varying input depends on the stiffness, damping, and mass of the structure. According to [Slocum, 1994a], high stiffness and damping are each necessary, but are not individually sufficient requirements for a precision machine.

Although extensively studied, and heavily relied upon in machine tool design, the mechanism of damping in a material is difficult to quantify and one must generally rely on empirical results [Lazan, 1968]. In fact, damping highly depends on alloy composition, frequency, stress level and type (i.e. tension or shear), temperature, and joint preload. Traditionally, machine tools have been built of cast iron, which has moderately good damping properties, or steel weldments, whose damping properties are not as good. Polymer concrete was then developed, and typically had on the order of 5 to 10 times the material damping of cast iron and was easier to cast. However, the use of polymer concrete led to marginal increase of machine weight and decrease of thermal diffusivity. However it is quite widely used for precision grinding machines.

Structural joints in machine tools have long been known to be a source of damping by the mechanisms of friction and microslip. Numerous theories are available for predicting structural joint damping by these mechanisms [Tsutsumi, 1979], [Murty, 1982], [Ferri, 1995]. In [Slocum, 1992] and [Slocum, 1994a], the Tetraform structural concept for machine tools and instruments, developed by Lin, is described. Here, besides viscous shear and squeeze film damping in the legs, damping at the spherical joints is achieved by application of adjustable sliding bearing technology, developed for the Nanosurf 2, in which thin (<2 μm) PTFE layers are used.
However, the amount of damping obtained in structural joints is still an order of magnitude less than what is desired for optimal performance. In addition, as far as the accuracy of the machine is required, it would be best if the joints behaved as rigid interfaces, instead of slip planes [Slocum, 1994a].

In machines with rotating components (e.g. grinding wheels), there is often enough energy at multiples of the rotational frequency (harmonics) to cause resonant vibrations. The amplification at a particular frequency can be reduced using a tuned mass damper, which pulls the resonant peak down into two smaller peaks. A tuned mass damper consists of carefully selected mass (at least 3-10% of the structure's mass), spring, and damper elements attached to a structure. The design of a tuned mass damper is quite straightforward [Meirovitch, 1975]. However, the limitation of these dampers is that it can only remove energy at a small frequency range. As the machine configuration or boundary conditions change (e.g. the moving parts in a CMM, or the depth of cut taken by a boring bar), the damper will lose some of its effectiveness.

By contrast to tuned mass dampers, constrained layer dampers add damping at all frequencies and are insensitive to the vibration amplitude. This type of damping mechanism has been used in many systems, among others consumer products, helicopter blades, and machine tools. Damping is achieved by imposing shear strains on the alternate layers of visco-elastic and structural materials, which is caused by relative motion. Motion of a structure is generally greatest far from its neutral axis, especially when interface surfaces move in opposite directions. The design can be developed using numerical methods [Haranath, 1987], or closed form solutions that were formulated many years ago [Nashif, 1985].
Viscous shear forces stretch the structural constraining layer, so thin layers are not as effective.

In the interest of reducing vibration in precision machines and components, e.g. fixtures, [Slocum, 1994a] proposes a concept of a replicated internal viscous damper. The design principle is similar to that of a constrained layer damper. The replicated internal viscous damper uses internal beams covered with a viscous fluid that are replicated in place inside structural members. In the case where the centre of a structure is cored to make room for the damper, although the static stiffness is decreased by up to 25% depending on the design, the dynamic stiffness can be increased by an order of magnitude. Design theory as well as experimental results show that Q factors on the order of 20 are obtainable. This is one to two orders of magnitude lower than would normally occur in the material of the structure or in its joints.
[Marsh, 1996] describes a passive damping method based on both constrained layer and replicated viscous damping. Here, visco-elastic layers are embedded within the structure as opposed to being applied externally. An analytical solution to the amount of damping in modes is presented using a modal strain energy approach. Several full-scale structures have been built and tested, that offer modal loss factors (1/Q) between 0.1 and 0.3, depending on the constraints of added weight and available space for shear members.

5.2 Bearing systems

5.2.1 Full film bearings

5.2.1.1 Static bearing stiffness

Full film bearings, mostly aerostatic or hydrostatic bearings, are increasingly used in PE applications to provide a high accuracy long-stroke motion. The latter is mainly as a result of low limiting friction and a high averaging effect of surface imperfections. The choice of lubricant depends upon the type of application. Hydrostatic bearings exhibit much greater film stiffness than aerostatic bearings, usually by a factor of about five, due to the higher supply pressures. Important differences lie in the performance under dynamic loading. Hydrostatic bearings have superior damping characteristics to aerostatic bearings and this may be a critical feature for some application. Instability phenomena in the latter are mainly caused by the compressibility of air and molecular transit times in the bearing gap, in combination with gap thickness variations and pressure changes [Blondeel, 1980]. In [Plessers, 1988a,b], a stability (Nyquist) criterion is described to evaluate instability phenomena. Both numerical and experimental methods are used to derive dynamic stiffness and damping coefficients.
Compared to hydrostatic bearings, aerostatic bearings have negligible friction and hysteresis. Viscous friction is

proportional to both relative speed and viscosity of the fluid. The latter is extremely low for air compared to oil (factor of 10^2 to 10^5), and therefore, air spindles can be applied in a much wider speed range. When designed for equal stiffness, the friction in an air spindle might be a factor of 3 to 10 lower compared to a hydrostatic system, but the mass moment of inertia of hydrostatic spindles is about a factor of 30 less, being advantageous for application in CNC production machines [Kraakman, 1997], [Schouten, 1997]. However, in addition to a minimum required surface area for a high bearing stiffness, the spindle stiffness has to be high enough to attain a favourable dynamic behaviour. A drawback of hydraulic systems is the need for return pipes and much more complex filtering and conditioning systems.

On account of its simplicity, both aerostatic and hydrostatic thrust as well as journal bearings with parallel gap shapes (both inherent and orifice types) are often used in PE slide constructions and spindles. In addition to analytical descriptions characterising fluid film flows, design methods and software are available to determine bearing dimensions, necessary for a certain load capacity or stiffness [Holster, 1967], [Kraakman, 1976], [Kraakman, 1977], [Blondeel, 1980], [Stout, 1981], [Plessers, 1988a,b]. In [Wang, 1993], bi- and tri-conical gap shapes are used, giving a stiffness gain of about 2 to 3, compared to parallel gap shape bearings.

As air is highly compressible compared to oil, aerostatic bearings need far more surface area and much smaller film thicknesses (5 instead of 30 μm) to achieve comparable stiffness values. In consequence of the relatively small modulus of air (about a factor of 10^3 less than oil), conventional aerostatic bearing systems tend to determine the overall obtainable static machine stiffness. However, using a compliant bearing surface (membrane), the compressibility of air might be compensated passively [Al-Bender, 1992]. Thereby, even infinite static stiffness can be obtained over a large range of the load capacity, provided that a stiffness of the membrane and the air gap are equal. This principle of stiffness compensation can be used in axial as well as journal bearing systems [Blondeel, 1976], [Snoeys, 1977], [Bryant, 1986], [Snoeys, 1987], [Holster, 1987]. Additionally, the load-displacement characteristic of the bearing may even compensate for the (static) elastic deformation of connected machine parts, e.g. slides and spindles [Kazimierski, 1992]. At Philips Research Laboratories, double hydrostatic bearings are used with pressure feedback to a double-restriction membrane, to create infinite stiffness for Single Point Diamond Turning (SPDT) operation [Kraakman, 1969].

5.2.1.2 Dynamic bearing stiffness

To improve the dynamic aerostatic bearing stiffness as well, active stiffness compensation can be applied. In [Al-Bender, 1994], three methods of active control are described to enhance dynamic stiffness, viz. support control, conicity control, and supply pressure control. Support control is most simple and applicable to all bearing types. However, it is not compact and quite expensive. In conicity control the deflection of a compliant convergent surface is additionally controlled by a piezo-electric actuator. As described in [Al-Bender, 1997], this method of active compensation is found to be most effective, owing to the low actuated mass and the large bearing force gain and bandwidth. It achieves infinite static stiffness and still at 100 Hz, a tenfold increase in stiffness is possible compared to the passive value. [Al-Bender, 1998] describes the development and application of active front radial bearings in HF electro-spindles, as and effective and competitive alternative to both magnetic- and ball bearings, for its infinite static stiffness and active control bandwidth of 700 Hz.
Supply pressure control finally, has in contrary a limited bandwidth and relatively low cross over frequency. Until now, for a system with a great number of aerostatic bearings, the costs for active stiffness control are significant, since at least an equal number of sensors and actuators is required.

5.2.2 Active magnetic bearings

In many high precision applications, the limitation of friction losses in bearings is an engineering challenge. As magnetic levitation is contact-free, there is virtually no friction and wear, so magnetic bearings offer long life and reduced maintenance. The rotational speed is only limited by the strength of the rotor material (400 N/mm^2 for soft magnetic iron), and peripheral speeds of up to 350 m/s have been reached. Magnetic bearing systems are particularly suitable for applications in which there should be no contamination from lubricants, as in vacuum. In addition, it is fairly easy to adjust the position of the bearing forces and suppress the effects of unbalance. A difficulty however, is that in practice, magnetic levitation is always accompanied by an unstable equilibrium of forces, so that height control with feedback is necessary. Active magnetic bearing (AMB) systems were built as early as 1938 by Kemper for experiments and later for momentum wheels in space applications. Due to the enormous progress in electronics, the number of industrial applications has considerably increased during the last 15 years. [Schweitzer, 1988], [Higuchi, 1990], [Allaire, 1992], [Trumper, 1993], [Schweitzer, 1994], [Matsumara, 1996].

In general, five control loops are required for an AMB system with one d.o.f. The associated electronics, however, make such bearing rather complicated and expensive, and for this reason, AMBs have been designed that have one or more non-controlled d.o.f. [Kamerbeek, 1983, 1984]. The drawback here is that the load-carrying capacity and stiffness corresponding to the non-controlled d.o.f. are reduced [Hendrikson, 1974], [Sabnis, 1974], [Kamerbeek, 1997]. Electromagnets are usually arranged in two radially opposed coils for each d.o.f., giving an eight-pole radial bearing system. However, a minimum of six poles with six windings would suffice, but, contrary to systems using passive elements, such as flexures and fluid film bearings, control becomes more difficult in AMB systems, since force generation in the x and y directions is then coupled.

For small rotor displacements and small control currents, (compared to the nominal air gap and the bias current respectively), the AMB force is quite linear. However, due to a negative coefficient of proportionality of the rotor displacement (the open-loop stiffness), AMB systems are characterised by open-loop instability. Besides position feedback, Kamerbeek applied velocity feedback, having the advantage of less heat dissipation in the control coils [Kamerbeek, 1983]. Although the losses in AMBs are usually less then 20% of that in fluid film bearings, and less then 5% of the loss in mechanical bearing systems, they may be examined in some applications. Especially for larger rotor displacements, non-linear and non-ideal effects appear, such as copper losses, hysteresis effects, stray fields, saturation of the iron, and eddy current losses. Especially for high performance AMBs, these non-ideal properties, which mainly depend on the material used in the flux-loop and the bearing arrangement (hetropolar or homopolar configuration), have to be considered. For conventional soft magnetic iron, such as silicon iron, the maximum flux density is limited to about 1.5 T, thereby limiting the maximum static bearing force density to about 20-40 N/cm^2 (comparable to air bearings).

For the general class of linear permanent magnet electric machines, [Trumper, 1996] presents a design and analysis framework, with which complex magnet arrays and winding patterns can be addressed. The models resulting from the analysis, serve as design tools for the development of high-resolution, power efficient, linear magnetic levitators, such as used in wafer steppers (fig. 5.1). To improve power efficiency, the levitator uses a permanent magnet Halbach array, primary developed for use as an optical element in particle accelerators. In [Trumper, 1993], useful geometries of such magnet arrays for synchronous machines are presented in different coordinate systems.

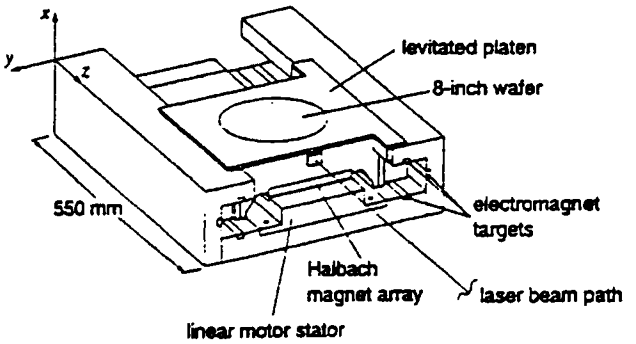

Fig. 5.1: Six d.o.f. magnetically levitated stage [Trumper, 1996].

Another six d.o.f. magnetically suspended stage is described in [Holmes, 1996]. Here, additional neutrally buoyant oil floatation is used in order to achieve extremely high disturbance rejection and position resolution, competitive with piezoelectric actuators. This hybrid stage, which was designed for STM applications, attains atomic-scale motion control inside 100 μm cube of travel.

5.3 Flexures

The advantages of flexures over classical hinges are the absence of play, and sliding- or rolling friction as a source of structural hysteresis [Koster, 1996], [Rosielle, 1998]: hence, there is no need for lubrication systems and maintenance. Even for ultra precision mechanisms, the level of material hysteresis in flexures, e.g. as a result of non-elasticity, or dislocation mechanisms in metals, is insignificantly low, and extremely precise measurements are necessary to detect this phenomenon [Lazan, 1968].
The linear operating force of flexures presents no big challenge in motion control (see section 5.6). Care must be taken to avoid mechanical flip flops due to internal over constraints as a result of for example design or manufacturing defects, undesired damage such as pre-buckling, external assembly- or thermal miss-alignment. Provided that such problems are avoided, overload protection is envisaged, and buckling-, as well as modal analysis is required. Although the length of stroke is limited, flexures are often advantageous in 'Design for Precision', featuring predictability and repeatability [Eastman, 1935], [Eastman, 1937], [Jones, 1951], [Jones, 1956], [Geary, 1954], [Hoek, 1962-1986], [Hoek, 1985-1989], [Paros, 1965], [Eijk, 1985], [Smith, 1992], [Koster, 1996], [Rosielle, 1998].
Single motion flexures can be divided in two main groups, i.e. rotation and translation. For rotation, fig. 5.2 contains some execution examples (a,b,c) as well as the schematic representation (d).

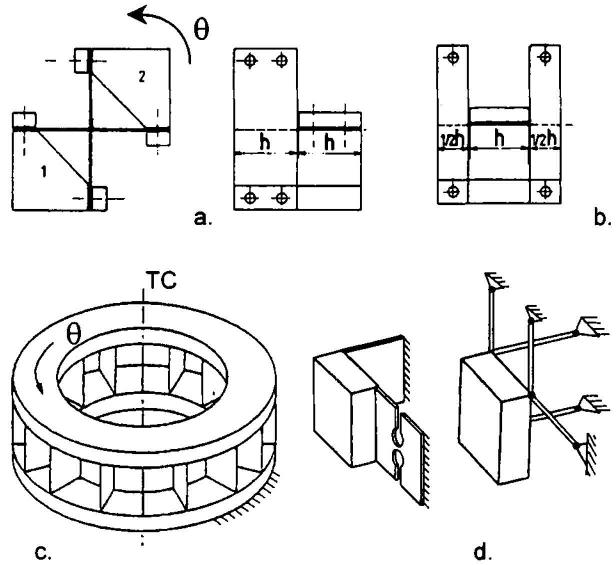

Fig. 5.2: Execution examples of one rotation.

The alternative 5.2b is preferable to 5.2a from the point that a symmetrical loading results in four times higher stiffness. This becomes clear from fig. 5.3, where the dimensionless stiffness reduction c(a)/c(0) is set against the dimensionless distance of the force's line of action to the cross section's neutral axis a/h*. Alternative 5.2c is (thermally) symmetrical about the rotational axis (TC). The schematic representation is shown in fig. 5.2d.

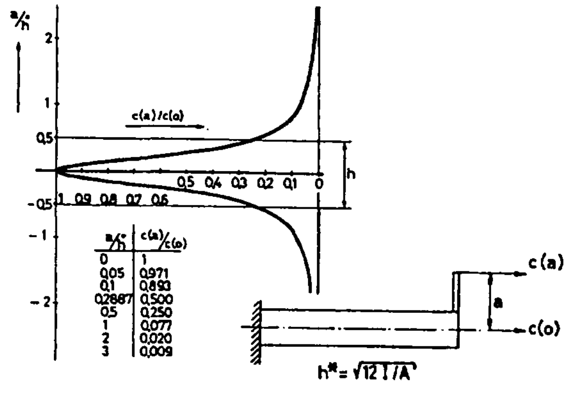

Fig. 5.3: Dimensionless stiffness dependent on the force's line of action.

Fig. 5.4a contains an example of an elastic universal joint, (with it's schematic representation shown in fig. 5.4b), leaving two rotational d.o.f. free, whilst constraining the remaining rotation and three translations.

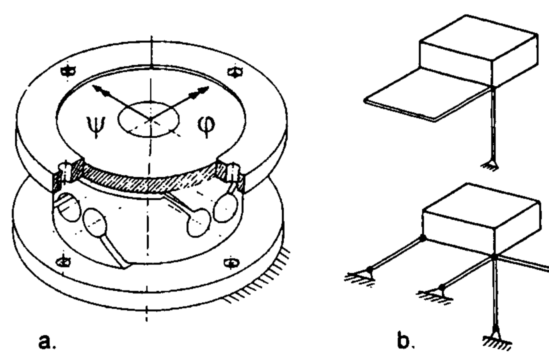

Fig. 5.4: Elastic universal joint: two rotational d.o.f.

An elastic spherical pivot is shown in fig. 5.5a. Different from the schematic representation (fig. 5.5b), where the three translational d.o.f. are fixed with three rods, the execution example contains three 'folded' leaf springs (cut by wire EDM), as explained in section 3.2. As the fold-lines are orthogonal, translation stiffness at the (fictitious) intersection point P is equal in all directions (580 N/μm) [Rosielle, 1996].

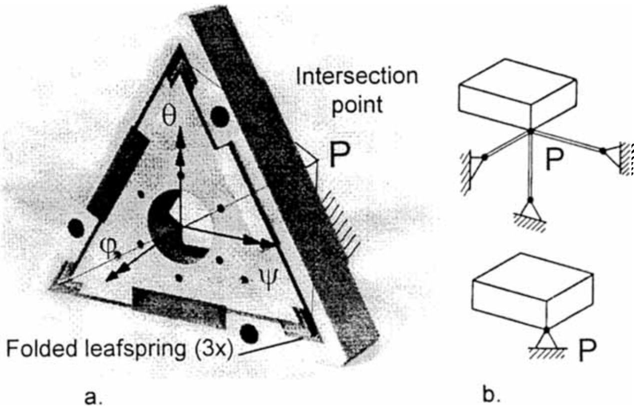

Fig. 5.5: Elastic spherical pivot: three rotational d.o.f.

For translation, fig. 5.6 contains examples of one d.o.f. When applying the driving force F halfway length l of the leaf springs, as in alternative 5.6b, these springs are free of (opposed) normal forces. The disadvantage of the lowering effect ($0.6x^2/l$) of the parallel motion alternative of fig. 5.6 might be compensated by a double parallel mechanism, which was among others implemented in the roughness measurement instrument by Rank Taylor Hobson.

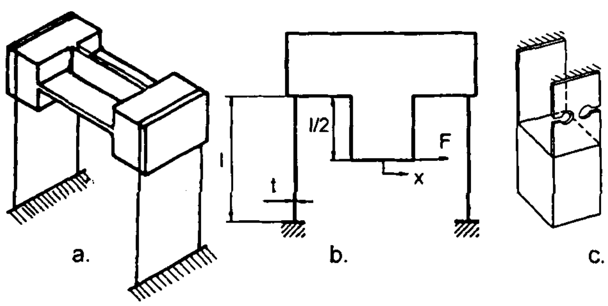

Fig. 5.6: Five d.o.f. constraint parallel motion without symmetry.

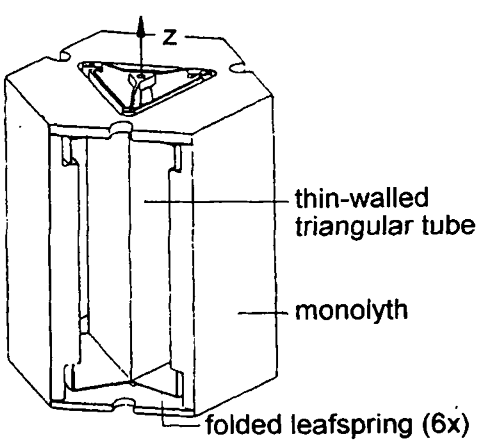

Fig. 5.7: Five d.o.f. axis-symmetric linear motion produced by wire EDM.

The elastic guide in fig. 5.7 on the other hand, has a true axis of symmetric translation without lowering effects. An additional 'folded' leaf spring is applied for optimal symmetry. Internal stress is avoided by the monolithic fabrication process, using wire EDM [Rosielle, 1994a,b], [Haitjema, 1997].

5.4 Drives: properties and accuracy

Machine tool feed drives control the position and velocity of slides or axes in accordance with commands generated by CNC interpolators. In [Srinivasan, 1997], a list of requirements on feed drive performance is given: 'control over a wide range of speeds, precise control of position, ability to withstand machining loads while maintaining accuracy, rapid response of drive system to command inputs from the machine tool CNC system, and precise co-ordination of the control of multiple axes in contouring operations'. These requirements are best met by closed loop control of feed drives based upon real-time sensing of position.

Demands on feed drive performance have become more stringent as machining technology has evolved to meet the requirements of a broader range of applications, e.g. high speed machining. Precision machining applications, such as SPDT have similarly tightened positioning accuracy requirements on feed drives. In [Smith, 1992], a *variety of actuators is described, that are capable of* operating at the nm level. Each actuator is discussed in terms of its relative merits, such as dynamic response, repeatability, size, linearity and cost.

5.4.1 Short stroke actuators

Whereas recent low voltage type piezos based on thin layers are giving more extension at typically 100 V, many precision applications still apply high voltage piezos, which, at present, tend to be stiffer under static conditions. Compact actuators can generate large forces. Piezo elements typically have 0.1% strain under 1000 V/mm. Hence, single piezo elements are limited in maximum stroke to the μm range, which often necessitates two stage actuation. The total length of stroke obtainable depends on the number of discs in a piezo stack, and amounts typically to a maximum of about 100 to 200 μm. Here, typical material expansion rates are 1 mm/s.

Piezo material itself may have an internal hysteresis, which can be in the order of 20% of the stroke length. Therefore, accurate positioning over large strokes is best done with closed loop control. In [Ge, 1996], an approach is described for modelling the hysteresis non-linearity of piezo-ceramic actuators under non-cyclic input variations. By implementing the model in a feedforward control loop, tracking control accuracy was improved significantly (see section 5.6.1).

Manufacturers tend to clamp piezo stacks with a preloaded spring to avoid mechanical hysteresis in the many contact surfaces. A preload itself, e.g. by a weight, shifts the range of piezo movement without reducing it, whereas stiffness, e.g. from an external elastic *mechanism will reduce the stroke that can be obtained.*

Nowadays, linear piezo actuators are used in numerous applications like positioning systems [Wulp, 1996], shape control of optical components [Schothorst, 1997], vibration control, smart structures [Koster, 1997], laser cavities [Schellekens, 1986], measuring lasers [Wetzels, 1998], optical delay lines [Mierlo, 1993], AFMs and STMs [Vorburger, 1997], piezo driven Stewart Platform [Beltman, 1997], etc.

5.4.2 Long stroke actuators

5.4.2.1 Piezosteppers

The above limitations tend to shadow the otherwise clean electrical to mechanical transfer. Designers have thought around the obstacles and created actuators for longer strokes like the classical inchworm®, a stepping device which can be used for long stroke actuation by repeated stepping motion, as well as for single movement with nanometre resolution. Besides linear motion types, designers have come up with numerous rotational drive systems, e.g. for rotary tables and mirrors. Large rotations, when required are mostly step type or inertial drive systems, providing tangential motion at large radii. Though all bearing types mentioned in section 5.2 are potential candidates for high precision piezo driven rotary tables, it can be useful to consider driving more d.o.f. than just the rotation. [Philips, 1995] describes the design of a rotary table, depicted in fig. 5.8, that can be moved within 0.1 μrad and some nm position accuracy of the rotation axis, due to closed loop control and calibration facilities. Three piezo stepper drive-units are equally spread around a rotary table with an axial air bearing. All mechanisms are cut into a single plate by wire EDM, which is freely programmable, and thereby other than traditionally circular holes can be applied in order to make a more compact design [Rosielle, 1991]. Due to the application of high accuracy sensors, many disturbances including temperature change, can be compensated for in closed loop control. The relatively low rotational speed of 1 rpm is a challenge to increase further.

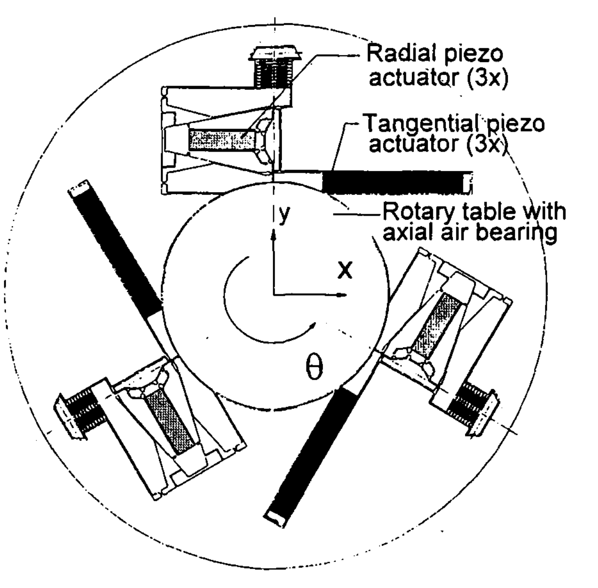

Fig. 5.8: rotary table with nm range positioning system for the axis of rotation.

5.4.2.2 Electromagnetic actuators

Currently, electrical actuation is predominantly used in machine tool feed drives. Rotary motors are used in conjunction with rotary-to-linear power transmission elements. The need for the latter is eliminated using linear DC or AC motors. While brushed DC motors retain a purchase price advantage over the other motors because of simpler associated electronics, they are subject to brush wear and require higher maintenance. Furthermore, brush friction in such motors is non-linear and can limit control loop performance. AC motors are powered by AC line power, but rely upon sophisticated electronic control of frequency and amplitude. Smoothness of generated force, i.e. low force- and velocity ripple, is an important performance measure for all motor types and requires specific design features and electrical power control to minimise force ripple [Bollinger, 1982], [Slocum, 1992].

Linear power transmission elements are used in feed drives both to convert rotary actuator motion and to achieve speed reduction. Lead screw drives, using nut transmission elements, are most commonly used at this stage of development [Gross, 1983], [Weck, 1984]. Many of the problems with lead screw drives are the result of an improper kinematic mounting of the nut. When properly designed however, they are capable of submicron-level motion repeatability. The advantage of lead screw drives in some applications is that stiffness is present above the bandwidth of control, as opposed to linear motors, which therefore require excessive bandwidth. Backlash in lead screw drives is reduced by preloading the nut, at the expense of increasing Coulomb friction, and thereby, increasing virtual play and lowering drive efficiency. Smoothness of motion in ball screw drives is limited due to entry and exit of the balls in the ball-nut, causing velocity ripple.

The behaviour of friction wheel drives – also known as traction drives, or capstan drives – on the other hand, is more predictable and reproducible due to a prescribed level of preload at the statically determinate wheel contacts, thereby superior for the machining of optically smooth surfaces [Takahashi, 1990], [Rosielle, 1994]. At CUPE, LLNL [McKeown, 1998] and Philips Research Laboratories, friction wheel drives have been used in SPDT machines since the 1980's, in combination with laser interferometer measuring systems [Bryan, 1979a], [Gijsbers, 1980], [Donaldson, 1984], [Donaldson, 1986]. In [Bispink, 1992], ball screw as well as friction wheel drives are investigated in both constant velocity and reverse tests. Bispink found that friction wheel drives have a factor of 10 smaller position error signal. However, they have lower stiffness and a lower maximum force level. In [Taniguchi, 1996], experiments with nano-positioning systems are described. Here, an aerostatic nut-and-screw transmission is used in combination with a nano-servomotor with air bearings as well. The nut surface is made slightly concave, giving a 50 N/μm thrust stiffness. Applying motor encoder feedback as well as scale feedback, result in a deviation of the feedback signal from sine wave commands of some nm. CUPE/CPE have done the same with oil hydrostatic lead screws to 2 nm resolution and control [McKeown, 1998].

According to [Klafter, 1989], one of the primary causes of poor tracking in indirect driven machines, especially at low speed, is the mechanical link between the actuator and the output. Hence a design without such transmission, i.e. a direct drive system, is desirable in some cases. For these systems, e.g. linear motors, voice coil actuators, and torque motors, it is essential to employ very high resolution encoders, being available nowadays for linear as well as rotational measurements. Direct drive systems, are advantageous for their ease of maintenance, little friction and higher stiffness due to fewer compliant parts. Direct drives include lower drive inertia, higher acceleration capability, elimination of transmission resonance, and transmission related frictional effects and (virtual) play [Pritschow, 1990]. Pritschow indicated higher positioning performance achievable using direct drive motors.

An advantage of indirect drive systems is the accurate tachometer velocity information, even at a very low slide

travelling speed due to the application of a transmission element. Moreover, the inertia of the load as well as external- and internal forces converted to the motor shaft are reduced by the transmission ratio, thereby reducing the size of the actuator. Depending on motor constant and electrical resistance, heat dissipation of the actuator, might be smaller, especially when hold currents are supplied. In addition, the source of heat might be placed at a greater distance and thereby isolated from the workpiece.
Transmission elements affect the nature of the feed drive control problem significantly. Feed drives including gears or ball screws for speed reduction generally have high transmission ratio values. Consequently, the feed drive control loop is effectively uncoupled from the machining process and the structural dynamics loop. In the case of direct linear actuation however, where the ratio equals unity, the structural loop and the feed drive control loop interact significantly. Robust controllers are needed here, such as H^{∞} synthesis and sliding mode control [Brussel, 1998].
In [Renkens, 1997], a spindle unit is described, using a direct drive voice coil actuator. With respect to current spindles the objective was to reduce the axial error motion down to ±0.01 µm, increase the axial stiffness up to 1000 N/µm with a bandwidth of 700 Hz, and create the ability to produce non-rotationally symmetric parts with optical surface quality (R_a < 10nm).

Direct drive systems have the important drawback that load variations are directly felt by the motor, and in reverse, variations in motor force, such as cogging and force- and torque ripple, directly influence the accuracy at the load. Cogging results from end effects of steel laminations, crossing the alternate magnetic poles. Force- and torque ripples arise from higher-order field harmonics due to both the magnet array geometry and the stator current distribution. In addition to the application of non-ferrous core motors, state feedback, feedforward, and motor ripple compensation can be applied to reduce the variation in actuator force. [Braembussche, 1996] describes that only a simple ripple compensation method is necessary and sufficient in order to achieve acceptable tracking errors. In [Kim, 1996], an analysis of the force ripple in a surface-wound permanent magnet linear motor with Halbach array is presented, as being highly dependant on the number of phases. The force ripple is found to be less than 0.1% for motors using four or more phases. Disc armature motors, normally used in high acceleration applications for their small mass moment of inertia, have many self supporting copper coils, running through an axial magnetic field. For the very high number of coils (order of 10^2), compared to traditional radial field permanent magnet DC motors, the torque ripple is negligibly small and in consequence, these motors might be applied as direct drive in applications where high smoothness of motion is required, e.g. SPDT operations [Vermeulen, 1996a].

5.5 Sensors

Sensors can be incorporated in precision machines for two reasons: either in measuring instruments for which the sensor is a prerequisite, e.g. CCD cameras in vision system, or to obtain higher system accuracy, for example using a position sensor in a control loop or a temperature sensor for thermal error compensation. The sensors, which are dealt with in this section are limited to the latter group. These are position and velocity sensors for control, acceleration sensors for active vibration isolation, and temperature sensors for thermal error compensation.

Parameters characterising the behaviour of a sensor are the uncertainty, repeatability, resolution, bandwidth and measuring range. Below, a short list of most important sensors is dealt with [Doebelin, 1990], [Smith, 1992], [Slocum, 1992].

5.5.1 Digital position sensors

5.5.1.1 Laser interferometers

Laser interferometers are based on the interference of two beams, i.e. a reference and a measuring beam, emitted by a coherent laser source. These beams are split in the interferometer either by a polarising beam splitter or by an amplitude beam splitter. Both possibilities are used by different manufacturers. To increase stability often slightly different frequencies (heterodyne interferometry) are used. Whether this is done and which frequency difference is used also varies for different manufacturers. One or more photodiodes detect changes in the optical path difference of the beams by counting interference fringes, generated in the measurement after the measuring- and reference beam are made to interfere. Interpolation between the fringes is used to increase the resolution to the nm level By using different optics, in addition to position, also angle, flatness, and straightness can be measured [VDI, 1985], [Kunzmann, 1993], [Flügge, 1996].
The accuracy of laser interferometers is limited by the stability of the wavelength of the laser light, which changes due to variation in the refractive index of air. This index varies with temperature, pressure, humidity, CO_2 content and contamination by other gasses, and can be computed from Edlén's formula [Edlén, 1966], [Birch, 1994]. An other source of error is non-linearity in the interpolation process, due to misalignment of and imperfections in optics (beam splitters) optics [Quenelle, 1983], [Hou, 1992], and [Wu, 1996]. When laser interferometers are used as a measuring tool for measurement of length, rotation, straightness or flatness, the equipment used, e.g. laser source, optics, mechanics, electronics and software, have to be calibrated [Schellekens, 1986], [Kunzmann, 1992]. Although nm resolution is easily achievable nowadays the relative uncertainty will –for measurements in air– not come far below 10^{-7} due to the error sources indicated above.

5.5.1.2 Scales

Scales are based on a repetitive pattern of a reflective- transmissive- conductive- or magnetic material embedded in a substrate, giving a periodic signal. The accuracy is mostly limited due to deviations in the embedded pattern and lies in the µm to sub-µm range. Conventional optical encoders typically operate on the principle of counting Moiré lines by means of a light source and a photodiode, they can be linear or rotary.

Diffractional optical scales used as grating, increase the resolution to the nm level, whilst an uncertainty of about 10 nm. Here, two beams split from a laser of LED fall on the ruler and diffract with different orders. After that, the beams are brought to interfere. The interference signal is essentially counting the lines of the scale passing by. To improve the interpolation, other signals with different phases are generated, e.g. by means of quarter wavelength plates. Main advantages to conventional optical scales are the higher line density, up to 2000 instead of 125 lines per mm, and the improved possibilities for interpolation [Kunzmann, 1993], [Flügge, 1996], [Spies, 1997]. Travelling speed at maximum resolution however, is presently limited by electronics in commercially available systems.

Other scales, based on a magnetic, inductive or capacitive measuring principle have resolutions and uncertainties in the same range as conventional optical scales.

5.5.2 Analogue position sensors

In general, analogue position sensors like LDVTs, eddy current- and photonic sensors, have dynamic ranges (defined as the ratio of position measuring range and resolution) in the order of about 10^3 to 10^4. Only capacitive position sensors can have dynamic ranges comparable with optical scales. In fig. 5.9, the dynamic range of most common position sensors is depicted against the bandwidth. All sensors regarded here are non-contact sensors avoiding friction and hysteresis.

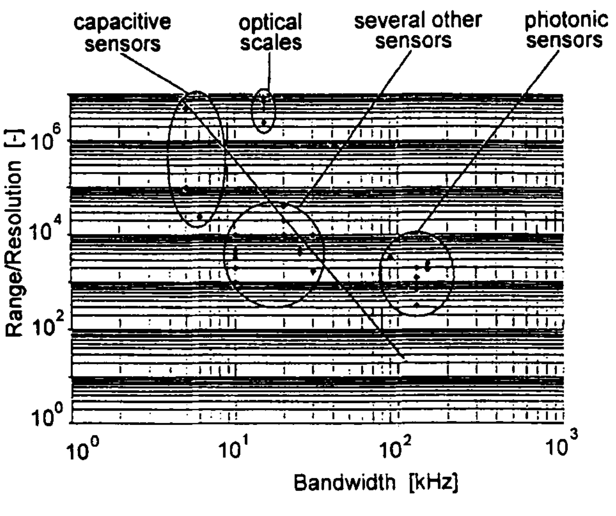

Fig. 5.9: Dynamic range against sensor bandwidth [Nijsse, 1997]

5.5.2.1 Capacitive sensors

Two electrically conducting parallel plates or concentric cylindrical parts, having a certain voltage difference and overlapping surface may form a capacitor. By changing the distance or the overlapping surface of the two parts, the capacitance value will change, and thereby capacitors can measure distance or displacement. By adding a third parallel plate in between the two others, being attached elastically to the sensor housing, capacitive sensors can be used as well for accurate angle measurement (differential method). Practical values vary from 0.01 pF for small probes, to 100 pF for the larger ones. Capacitive sensors can have very high resolutions down to the picometre level at about 10 kHz bandwidth [Bonse, 1995]. The range however, is limited to at most 1 mm. Capacitive sensors are highly temperature sensitive. They are used for velocity and acceleration measurement as well. [Doebelin, 1990].

5.5.2.2 Inductive sensors

Inductive sensors measure the variation of inductance between two or more coils, which depends on the position of a ferromagnetic core. There are two coil configurations. The first type uses two coils in a Wheatstone measuring bridge. The second configuration type uses three coils: a primary coil, inductively coupled to two secondary coils. The use of two measuring coils allows for differential measurements; these sensors are called LVDT or RVDT for linear- and rotary- variable displacement measurement respectively. From the two measuring coils additional information about the moving direction is obtained, and in the mid-position of the core, the sensors are independent of temperature changes. Inductive sensors may have very large ranges, differing from 0.1 mm to about 5 m, and linearities between 0.05 and 1%. The resolution can be very low down to one tenth of a nm and the accuracy can be as good as a few tenth on a nm [Wetzels, 1998]. The bandwidth is mostly limited to several hundreds of Hz.

5.5.2.3 Eddy current sensors

In varying magnetic fields, currents are induced at the surface of a conductive material (for aluminium: typically 0.1 mm skin depth). Measuring the change of self-induction of the coils generating the alternating magnetic field can sense these so-called eddy currents. Although these probes have a significant non-linearity of 0.1 to 10%, resolutions of a few nm can be obtained, and the maximum stoke amounts to about 10 mm [Doebelin, 1990]. The best target material for eddy current sensors is aluminium; copper and brass are good alternatives. The use of steel however, is better avoided because of the varying permeability due to the microstructure of steel or surface layers [Weekers, 1997]. The disadvantage of all high-magnification, eddy current sensors is their sensitivity to micro cracks, sub-surface stresses, and lack of homogeneity in the target material. If the target moves in and out with respect to the head, there is no problem, but transverse movements can cause errors as high as 2.5 μm. [Bryan, 1967]. The advantage of eddy current sensors is their relative immunity to oil and dirt on the surface.

5.5.2.4 Optical sensors

A wide range of optical sensors is available including laser triangulation sensors [Möhrke, 1991], focal error detection [Bouwhuis, 1985], [Renkens, 1994], and fibre optic sensors, CCDs, PSDs and array detectors [Doebelin, 1990]. The range and resolutions depend on the applied hardware and software: Focal error detection sensors have nm resolution and μm range, whereas triangulation sensors, CCDs and PSDs have sub μm resolution and mm range. A clean target surface is a prerequisite for these types of sensors.

5.5.3 Velocity sensors

Velocity sensors are often used in servo systems to implement damping as will be explained in section 5.6. Linear velocities can be measured with voice coils [Renkens, 1997], [Goossens, 1997]. For angular velocity information in both direct and indirect drive systems, mostly tachometers are used. In voice coils as well as tachometers, a back-EMF signal is induced, linear proportional to the speed. Tachometers are available in versions with or without brushes. Increasing the number of coils can reduce voltage ripple in tachometers, which increases costs as well, or by filtering which causes time delay. Resolution and accuracy depend on the application: in measuring instruments or to obtain higher system accuracy. Linearity is typically in the order of 0.1%.

5.5.4 Accelerometers

Measuring and differentiating the displacement of an accelerating body relative to a compliantly suspended mass can derive acceleration information. Here, most commonly strain gauges and piezo electric transducers are used. During the last decade however, silicon acceleration sensors have become commercially available as well, being produced in large quantities at small costs, (using etching technology) for the application in air bags. For high accuracy measurements however, force balanced accelerometers are applied. Here, a mass-spring combination is used in a feedback loop with a voice coil actuator keeping the mass in zero position. The current (force) needed is a linear proportional measure for acceleration.

5.5.5 Temperature sensors

Temperature detectors are usually based on either a temperature dependent resistance (Resistance Temperature Detector, RTD) or on the Seebeck effect, which generates a thermo-voltage proportional to the measured temperature change (thermo-couple).
Two types of RTDs are mainly used for practical temperature measurement i.e. thermistors and platinum resistance thermometers (Pt-RTD). Thermistors have a high negative temperature coefficient dR/dT and are non-linear but very stable. They show high sensitivity and resolution and uncertainty down to the mK range at normal workshop conditions. No complicated electronics are necessary. Pt-RTDs show better linearity but sensitivity is much lower and asks for complicated electronics means to reach mK resolution and uncertainty. Thermocouples may be used too but they ask for a very stable reference temperature and require very sensitive and stable voltage measurement equipment to get mK accuracy.

5.5.6 Sensor mounting

As sensors are often part of a structural loop, kinematic, stiff-, and thermo-mechanically stable sensor mounting is essential, as close as possible to the place of which information is required. The location of sensors is of vital importance for the accuracy of high precision machines, and sometimes, even a metrology frame is essential. The Abbe and Bryan principles have already been mentioned in section 3.1. Besides, sensors should be isolated from heat sources and vibrations (see section 5.7). Sometimes it is advantageous to measure displacements before the transmission for the higher resolution at lower costs. All the errors in the transmission however, will affect the output. A final rule for sensor mounting here, is that a slide's measurement and driving system should be at the same side of the centre of rotation. Otherwise, an additional phase shift of 180° will occur, which is disadvantageous for stability [Slocum, 1992].

5.6 Control

As disturbances acting upon high precision machines introduce inadmissible deviations from prescribed functions, machine control is essential. Control of machine tools commenced in the early 1950's with the invention of Numerically Controlled (NC) machines by Parsons [Koren, 1997]. A major progress, especially on flexibility, occurred with the introduction of the Computer Numerical Control (CNC) and adaptive control in the early 1970's.

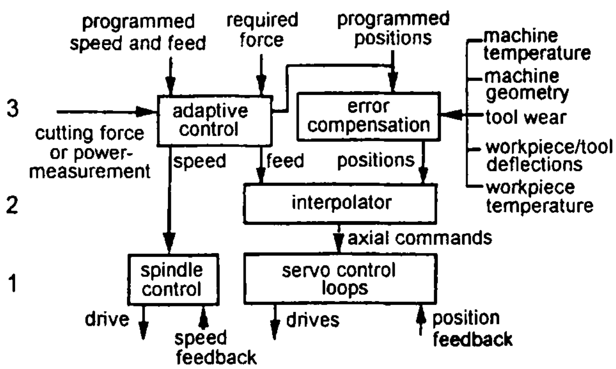

Fig. 5.10: Hierarchical levels in CNC controllers, from [Koren, 1997].

In a review about the last three decades, Koren describes three types of motion control, depicted in fig. 5.10, i.e. servo control of individual machine-axes, interpolators to co-ordinate the motion of several axes, and adaptive control to adjust real time the process variables in order to maximise system performance.

5.6.1 Servo control

In the past few decades, many types of feedback controllers have been developed, both analog and digital, as well as hybrid implementations [Srinivasan, 1997]. In digital controllers the effects of limited cycle frequency of sampling might reduce system performance [Renkens, 1997], [Goossens, 1997]. In addition, controllers applying state space estimation are used. The majority of industrial feedback control systems contains a PID-controller, having proportional gain (P-action) for servo stiffness, integral gain (I-action) to eliminate steady state errors and to reject external disturbances, and differential gain (D-action) to provide damping, which gives the system stability. For the latter either velocity feedback, using tachometer information, can be applied, or the differentiated position feedback signal. In the case that transmissions are used in combination with feedback of the actuator position, a resonance/anti-resonance pair is induced inside the control loop near to the transmission resonance frequency [Srinivasan, 1997]. Here, transmission related deviations are not dealt with inside the control loop, unlike feedback of the position of the load.
For effective rejection of disturbances, such as motor torque- and tachometer voltage ripple, as well as disturbances from contact bearings, a sufficiently high control loop bandwidth is required [Donaldson, 1984]. The effect of disturbances on the output position is expressed by the sensitivity [Bosch, 1998]. H^{∞}-controllers simultaneously filter disturbances and handle model uncertainties, for example at the subsequent resonance frequencies [Damen, 1997], [Braembussche, 1998]. In the latter case, a notch filter, used in classical controllers to filter a small frequency band, will not avail.
The required servo stiffness of the control loop depends on the application. For diamond turning, a static stiffness of 10^8 to 10^9 N/m is acceptable [Kouno, 1984], but in precision grinding of brittle materials, more than 10^{10} N/m is required [Kanai, 1990]. Classical PID-controllers are Single-Input-Single-Output (SISO) systems. More advanced controllers have Multiple Inputs and Multiple Outputs (MIMO), e.g. observer based controllers, which apply a state space approach [Eijk, 1998]. MIMO systems can be simplified to multiple SISO systems by conversion of dependent to independent variables.
In the design and analysis of feedback systems, various optimisation methods can be used. A rule of thumb to guarantee sufficient stability tells the ratio of the first and second undamped resonance frequency to be in between 3 and 10. A very powerful tool is the frequency response method, using Bode- and Nyquist diagrams, and having a close link to experimental information [Rankers, 1997]. As an example, in order to investigate the frequencies of interest, e.g. bandwidth, notch filter frequency, and tamed action frequency, the Bode diagram of the open loop response can be used. Two main problems with PID controllers in contouring applications are the relatively poor tracking of corners and non-linear contours, and significant overshoot [Koren, 1997]. To reduce these effects, a smooth motion profile can be chosen, e.g. an inclined sine, instead of a trapezoidal function [Rosielle, 1998]. Another effective method is the use of different controller strategies, for example, feedforward controllers, sophisticated axial controllers, like fuzzy logic and

repetitive controllers, or cross coupling controllers [Koren, 1997].

5.6.2 Feedforward control

In a feedback system, controller action is generated only after an error signal has occurred. Application of feedforward control however, can lead to significant error reduction up to a factor of 20 [Bosch, 1998], [Eijk, 1998]. Feedforward control –this can be velocity or acceleration feedforward– uses a priori knowledge of the mechanical system, e.g. inertial effects in a machine, or cogging effects in a motor. Application of intelligence however, is at the expense of calculation time, resulting in lower sampling frequency and lower control power. Feedforward can be implemented as the inverse of the feedback loop [Koren, 1997]. However, if the inverse function includes unstable poles, alternative forms of feedforward controllers have to be used. Tomizuka and others propose the use of frequency domain criteria [Tomizuka, 1989], e.g. zero phase error tracking controllers (ZPETCs). These controllers however, act very well in tracking complex trajectories but have poor performance in machine systems with large unmodelled disturbances [Tomizuka, 1987].

5.6.3 Interpolators

Interpolators are used successfully, in contouring systems, where a co-ordinated movement of separately driven axes is necessary [Koren, 1975], [Koren, 1981]. The interpolators, located at level 2 of fig. 5.10, generate command signals for each segment of the produced part, based upon an initial and final point, and some type of curvature, which is typically linear, circular and occasionally parabolic. Precision and productivity can be improved using real-time interpolators, instead of conventional CAD/CAM off-line decomposition [Shpitalni, 1994].

5.6.4 Adaptive control and compensation

Adaptive control and compensation, located at level 3 of the control architecture of fig. 5.10, improves performance, either by enlarging productivity, e.g. in rough cutting, or by improving accuracy, e.g. in optical quality finishing operations. Ro and Hubbel applied two different types of adaptive control in ball screw driven positioning systems, for both micro and macro movements, having very different underlying friction mechanisms [Ro, 1993]. For high precision purposes, real-time compensation techniques are used for errors caused by varying machine temperature, geometry errors of the machine, tool wear, etc. [Peklinik, 1970], [Soons, 1993], [Spaan, 1995], (see also section 6).

5.7 Isolation

For many different reasons, precision machines have to be isolated against external disturbances. In [Teague, 1989-1997], Hocken mentions a quotation of Maxwell [Hocken, 1995]: '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 must be so arranged that the effects of disturbing agents on the phenomena to be investigated are as small as possible' [Maxwell, 1890]. The major disturbing agents are vibration, temperature changes, humidity, acoustic noise, and electric and magnetic fields. The former two are discussed below in more detail. In specific situations, other effects, like variations of gravitational load, atmospheric pressure and EMC might have significant influence. Teague and Hocken mention three design strategies for machines to become resistant to environmental disturbances: 1) uncouple the machine from the environment, 2) design the machine so that environmental disturbances are minimised, and finally, 3) control the environment [ANSI, 1973]. Other references regarding this matter are: [Pohl, 1986], [Young, 1972], [Teague, 1986] and [Binnig, 1982].

5.7.1 Thermal isolation

Machine drives are one of the major sources of internal heat dissipation. To minimise this effect, the drive systems have to be adjusted to the load (see section 5.4). Small cycle times can reduce power dissipation, but at the same time limit productivity. For external heat sources, other techniques can be applied, e.g. encapsulation of heat source, and insulation of temperature sensitive materials (see section 3.3) from its surroundings [Trapet, 1997b]

In [Smith, 1992], the physical meaning of thermal material properties is dealt with. The smaller a material's coefficient of thermal expansion α, the smaller the effect of differential temperatures within a component. Temperature differences depend reciprocally proportional to the thermal conductivity λ, and as a result, it is the quotient α/λ that expresses the material sensitivity for spatial gradients. Mainly therefore, and for its good availability and machinability as well, aluminium is a favourable construction material. Therefore, CMM manufacturer Zeiss developed a special ceramic-surface-coated ageing resistant aluminium (CARAT®). In scales, exotic materials with negligible α are used, such as Invar® and Zerodur® glass ceramics. The latter is applied for structural components as well, e.g. in the Nanosurf by Lindsey, Smith and Robbie at NPL [Lindsey, 1988].
In the case of an time dependent temperature distribution in a machine, different time constants of components become predominant. The exponential path temperature change with time is governed by the volumetric thermal diffusivity $\lambda/\rho c_p$ (ρ: density, c_p: specific heat). For rapid response, this coefficient should be large. A mixture of different materials however, might cause potential problems [Lingard, 1991]. From temperature experiments, performed at PTB by rampwise increasing the environmental temperature with 20°C, the time constants of different CMM components appear to vary from typically 4 to more than 30 hours [Kunzmann, 1988].
When a body is over-constrained, internal stresses arise; For a small stress level the material's modulus should be small, which is in conflict with the demand for high stiffness (see section 5.1). Therefore, kinematic design, when possible, is highly preferable (see section 3.2).

Other frequently applied manners to minimise the effect of environmental temperature changes are temperature controlled rooms [Neumann, 1988], [Kunzmann, 1988], and oil – and air-showers, with which a thermal stability within a few mK is possible [Bryan, 1979b], [Bryan, 1982]. This approach to thermal stability is widely used today on ultra precision machines, e.g. the LDTM [Bryan, 1979a] and the LODTM [Donaldson, 1984].

5.7.2 Vibration isolation

In [DeBra, 1992], successful precision engineering is described as 'The balance of robustness of the machine and how benign the environment can be made through isolation to minimise the strains caused by vibration that compromise a machine's accuracy'. Seismic disturbances as a result of natural or cultural ground motions can compromise measuring accuracy and affect surface finish in diamond turning operations. In machine shop floors, typical ground vibrations have frequencies of 10 to 20 Hz

and amplitudes of a few μm [Riven, 1979a].
In addition to ground motions, vibrations might result from machine utilities, such as electricity, signal wires, air, vacuum, hydraulics and cooling fluid. Even acoustic coupling from neighbouring machine tools and fans can be significant, since acoustic energy is broadband in most cases.
Passive vibration isolation implies the application of a high mass on low stiffness springs, i.e. a second order system. Therefore, the attenuation of the amplitudes of vibration of the isolation on the one hand, and, for example, the tool or measuring system on the other, is given by the reciprocal squared ratio of their first natural frequencies. Multiple isolation stages can have higher natural frequency and give more attenuation above the critical frequency due to a steeper slope.
In [DeBra, 1992] and [Riven, 1979a,b], different kinds of isolators for precision machines are explained, some of them commercially available. For many applications polymer isolators may provide adequate isolation at low cost. Riven, for example, describes rubber-metal equifrequential isolators, based on the volumetric incompressibility of rubber and a proportionality between load and stiffness. However, when isolation requirements are tighter, pneumatic isolators are better. Air springs do not have the distributed property of slinky modes, present in metal helical springs. Hysteresis can be minimised using rubber roll seals between the piston and the isolation cylinder.
In isolation systems, additional damping is required to improve transient settings. Pneumatic isolators have the advantage that damping can be applied easily, by connecting each air spring through a laminar restriction to a different chamber, which has, in the ideal case, eight times the volume of the compliance chamber. In addition to damping, automatic height levelling is easy by individually changing the pressure in the legs [DeBra, 1992]. However, as air is filled through the damping tanks to the air spring chamber, the levelling system might become too slow.
For faster system response, active suspension and isolation can be applied [DeBra, 1992], [DeBra, 1994]. Here, a feedback loop is used to influence the height and orientation of the machine by means of a position or acceleration sensor and a magnetic or electrostatic actuator. Permanent magnet DC actuators based on Lorentz forces are easily controllable due to the linear reaction between current and the supplied force [Vermeulen, 1998]. In fully active isolation systems –which will be used much more in future– the natural frequency can be adjusted electronically, so that the frequency response of the vibration transmission can be shaped arbitrarily.

6. Error reduction and compensation

In Precision Engineering nowadays two methods of error minimisation are applied, as mentioned in [Sartori, 1995], [Weck, 1995a] and [Teague, 1989-1997]. In the first method, called 'error reduction', positioning uncertainty is minimised by proper design, based on design principles (section 3), modelling (section 4) and design features (section 5). Since a design will always be a compromise of many design philosophies, certain errors remain. A second method, called 'error compensation', can be applied to compensate for residual errors, for instance using geometric calibration- temperature measurement data. This type of error compensation may be applied in measuring machines. However, errors in positioning-machines and machine-tools have to be compensated real time in order to get an accurate result. The effect of error compensation strongly depends on the design of the machine and the ability to measure the errors of the machine which are in principle deterministic in a major sense as Bryan indicated in his papers [Bryan, 1984], [Bryan, 1993]. Therefore high repeatability is a need, asking for an excellent design.
As the first method of error minimisation has been described in previous sections, this section focuses on the latter method of error compensation.

6.1 Error compensation of measuring machines

Error compensation in measuring machines is generally based on the addition of a compensation vector $d\bar{P}(x,y,z,..)$ to scale- or interferometer readings $\bar{P}(x,y,z,..)$. The vector $d\bar{P}$ may be a vector function of co-ordinates $d\bar{P}=\bar{F}(x,y,z,..)$. The functions $\bar{F}(x,y,z,..)$ depend on the mechanical machine structure and the accuracy of the scales and the probe and geometry errors like straightness, squareness and waviness. These properties however may be influenced by the temperature distribution within the machine.
Also dynamic errors due to the inertial and modal behaviour of measuring machines can be corrected [Weekers, 1997]. It may be clear that compensation of this type of errors is much more difficult than the (quasi)-static errors discussed before. [Weekers, 1997] and [Knapp, 1988] showed that machine dynamics i.e. inertial effects and vibrations may cause considerable errors during probing.
To apply error compensation, different error models and measurement techniques were developed by many researchers [Hocken, 1977], [Zhang, 1985], [Jouy, 1986], [Sartori, 1988], [Teeuwsen 1989], [Balsamo, 1990], [Kunzmann, 1990], [Soons, 1993], [Kruth, 1994], [Sartori, 1995], [Lingard, 1991]. They showed that it was possible to develop very good error compensation for measuring machines. The compensation techniques are nowadays commercially available and used by many manufacturers of CMMs.

The techniques to analyse the error structure of CMM may be divided in parametric and volumetric techniques. In the parametric description the error parameters ${}_iR_j$, ${}_iT_j$ (as indicated in chapter 4) are measured directly at discrete positions and from these results, the error vector can be calculated, using adequate models. For these calibration techniques, laser interferometers, levelling instruments, autocollimators and hole- and ball plates may be used [Kunzmann, 1990], [Soons, 1993].

The volumetric approach to determine the error structure of a CMM is based on volumetric length measurements with artefacts like a double ball bar (DBB), a checking gauge of Renishaw or with laser interferometer, slip- or step gauge. It is important to measure sufficient lengths throughout the measuring volume [Krulewich, 1995a,b], [Jouy, 1986], [Soons, 1993], [Florussen, 1998], [Krulewich, 1998]. Using sophisticated modelling techniques, parametric errors ${}_iT_j(x,y,z)$, ${}_iR_j(x,y,z)$, may be estimated from the volumetric errors. In return, these parametric errors are used to calculate the error vector $d\bar{P}$ by an appropriate model [Florussen, 1997]. Impressive results are reported by some authors [Soons, 1993], [Spaan, 1995], [Krulewich, 1995a,b] and [Krulewich, 1998], but is has been proven too that it is not easy to estimate all the error parameters correctly from volumetric measurements using the DBB or the checking gauge as manufactured by Renishaw.

Recently activities are reported dealing with the so-called 'Virtual CMM' (VCMM) [Trapet, 1997a]. In this approach extensive error compensation of geometric-, thermo-mechanic- finite stiffness- and probing errors is applied. Residual errors are estimated from measurements of well-known artefacts and careful analysis. As a result each (corrected) measurement point $\bar{P}_i$ has a residual error field dP_i where $|dP_i| < A_p$, where A_p is the upper error limit. From $\bar{P}_i$ the measurement result M is calculated in the usual way for CMMs, but now many different values of M may be calculated using dP_i and simulate the measurement many times. From these results a mean value $\bar{M}$ and an statistical uncertainty value $S_{\bar{M}}$ can be calculated. In this way an approach for traceable measurements of products on CMMs may be realised.

6.2 Real time error compensation of machine tools

Real time error compensation of CNC machine tools and positioning machines (e.g. wafersteppers) by programmed positions or trajectories has been applied since the early 1970's [Hocken, 1986], [Soons, 1993], [Krulewich, 1995a,b], [Spaan, 1995], [Ni, 1997]. The main problem is, that the compensation values for each position have to be known beforehand. Particularly error compensation of thermo-mechanic and finite stiffness effects of manufacturing machines is difficult since these errors are time- and task dependent. Many authors have reported about different approaches to overcome these problems [Hocken, 1986], [Donmez, 1991], [Schellekens, 1992], [Spaan, 1995], [Weck, 1995a], and [Krulewich, 1995a,b].
Thermo-mechanic errors are the worst to compensate for due to their complicated structure [Spaan, 1995]. Most approaches use many temperature sensors to measure the outer machine temperature distribution and as a result to calculate the resulting tool drift. Here statistical techniques as well as neural network approaches are used to get robust relations between measured temperatures and tool drift [Spaan, 1995], [Veldhuis, 1994], [Ziegert, 1994], [Srinivasan, 1992], [Moriwaki, 1996]. Impressive improvements of machine accuracy were reported, showing that both approaches are effective in thermal error compensation [Spaan, 1995], [Krulewich, 1995a,b]. Fig. 6.1 shows the effect of geometrical and thermal error compensation on a milled workpiece with and without online error compensation. Here the extracted zero drift from the measured workpieces as a function of manufacturing time, is depicted.

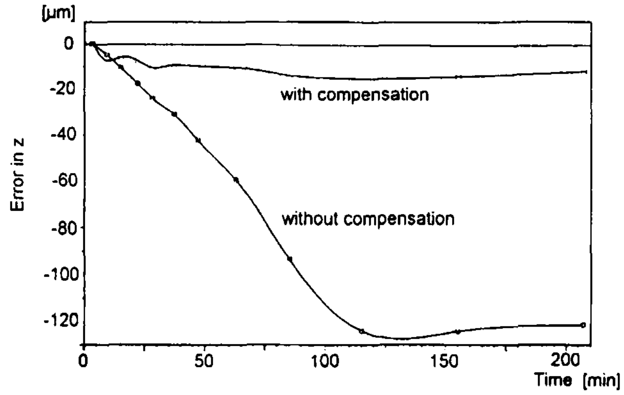

Fig. 6.1: Zero-point drift (z-direction) extracted from two workpieces milled with and without compensation [Spaan, 1995].

Compared to measuring machines, error compensation of machine tools has been developed not so well due to both measurement- as well as implementation problems of the algorithms and while additional measurement equipment is needed. For instance, due to the internal heat sources in a milling machine, ten to fifteen temperature sensors have to be connected to the machine body to estimate the temperature distribution of the machine efficiently. Sensor placement and effect analysis is of major importance [Spaan, 1995], [Ni, 1997].
Now first error compensation of geometrical and some thermal effects are commercially available for machine tools. Also some finite stiffness effects (straightness) may be compensated. Recently, first results were published about compensation of cutting force induced errors [Yang, 1996]. These errors may become significant in hard machining.
Generally the real time software error compensation is embedded in the controller of the machine tool. An interesting development is the so-called NC-code compensation [Spaan, 1995]. This method adapts the NC-code to compensate the programmed trajectory for the modelled errors.

7. Future trends in precision design

As indicated before, Precision Design is one of the basic disciplines in Precision Engineering. Precision Design offers the tools to improve the accuracy of precision machinery.
During the last decades Precision Design has been developed very rapidly which resulted in numerous high precision machines and instruments, such as CMMs and SPDT machines. Knowledge on Precision Design further developed in order to thrust back the frontiers of accuracy and resolution. The development of supporting software for design activities was of great importance, such as tools to predict the behaviour of machines, and foresee problems, without hardware realisation. Moreover, the integration of high accuracy sensors, closed loop control systems and advanced control software led to more accurate and flexible machinery.

Nowadays, needs for precision are mainly driven by developments in information technology. Research on IC-technology and technology for data handling and storage on tapes and discs, but also liquid crystal and plasma display technology are driving forces for Precision Engineering in general and Precision Design in particular.

The growth of the IC-technology is well indicated by the expansion of DRAM capacity from 245 kb in 1982 to 256 Mb in 1997. Line width and distance of traces have been reduced from 2 to 0.2 µm. Very large- and even ultra large scale ICs (VLSIs and ULSIs respectively) are common. All the many processing and illumination steps of the photo lithography process on silicon wafers of up to 12" diameter, ask for extreme requirements on velocity, acceleration, and especially reproducibility of the scanning and positioning stages, down to some tens of nanometres.

Next step will be x-ray lithography to overcome the restraints of the illumination wavelength, limiting the line width of traces to about 0.15 µm. Due to the short wavelength of about 13 nm for soft x-ray lithography, only mirrors can be used to deflect the x-ray beam from the source, through the mask at the wafer surface. Here, the uncertainty of the mirror shape has to be below 20 nm and a surface roughness less than 0.5 nm R_a is required at 200 mm diameter. In order to get enough x-ray reflection, high precision multi-layer coatings are necessary [Taniguchi, 1996]. As a result, new technologies have to be developed, not only for the stepper itself, but also for machines for mask making and inspection, as well as the manufacturing of high precision optical components. Here, also 'Design for Low Contamination' is getting part of concern of 'Design for Precision'.

Research and development of high density optical disk systems further shift the limits for data handling and storage. For systems like CD-Audio, CD-ROM, CD-Video and recently DVD-systems as well, maximum tolerances of track pitches and pit dimensions further decrease. Both recordable and rewritable versions are available. Here again, decreasing wavelengths, in combination with increasing numerical apertures (NA) are required. Recent developments like dual layer (DVD9) and even double-sided dual layer systems (DVD18) further increase the information density. The writing accuracy of laser beam recorders asks for nanometre accuracy rotary tables and mastering beam systems. Consumer DVD rewritable systems (DVD-RW) will soon reach the tens of nanometre accuracy level for writing reproducibility.

For PC terminals and HDTV, new flat screen systems are recently developed. For the former, thin-film-transistor colour LCDs are available, and for HDTV, plasma display panels (HD-PDP) were developed recently with new fabrication techniques. Both LCD and PDP systems need very accurate alignment techniques of opposite electrodes and well-controlled etching and film deposition processes [Taniguchi, 1996].

Finally, developments in astronomy have to be mentioned here. In the next decade, astronomic measurements will require stability in length of nanometres and nanoradians in angular position [Sartori, 1998].
Precision Engineering and therefore Precision Design as well, faces a decade of fast growth and striking new challenges.

References

[Abbe, 1890]
Abbe, E., Messapparate für Physiker, Zeits, Für Instrumenten-Kunde, Vol. 10, pp. 446-448.

[Al-Bender, 1992]
Al-Bender, F., 1992, Contributions to the design theory of circular centrally fed aerostatic bearings, Ph.D. Thesis, Katholic University of Leuven, Belgium, ISBN 90-73802-22-9.

[Al-Bender, 1994]
Al-Bender, F., Brussel, H. van, 1994, Active Dynamic Compensation of Aerostatic Bearings, Proc. of the ISMA 19, Tools for Noise and Vibration Analysis, Leuven, Belgium, pp. 187-197.

[Al-Bender, 1997]
Al-Bender, F., Brussel, H. van, 1997, Active Aerostatic Bearing through Control of Film Geometry, Proc. of the 9th IPES-4UME, Braunschweig, Germany, Vol. 2, pp. 389-392.

[Al-Bender, 1998]
Al-Bender, F., Brussel, H. van, 1998, Development of high frequency (HF) electrospindles with passive and active air bearings, Proc. of the International Seminar on Improving machine tool performance, San Sebastián, Spain, Vol. 1, pp. 175-184.

[Allaire, 1992]
Allaire, P. (ed.), 1992, Proc. of the 3rd International Symposium on Magnetic Bearings, Washington, DC.

[Ando, 1992]
Ando, M.,1992, Super smooth surface polishing on aspherical optics, Proc. of the SPIE, 1720, pp. 22-33.

[ANSI, 1973]
ANSI/ASME B89.6.2-1973(R1979), Temperature and Humidity Environment for Dimensional Measurement.

[ASME B89.4.1-1997]
Methods for performance evaluation of coordinate measuring machines, revision of ASME B89.1.12 M-1990

[ASME B5.54, 1993]
Methods for performance evaluation of computer numerically controlled machining centres

[Attia, 1979]
Attia, M.H., Kops, L., 1979, Calculation of thermal deformations of machine tool, in transient state, with the effect of structural joints taken into account, Annals of the CIRP, Vol. 28/1.

[Balsamo, 1990]
Balsamo, A., Margues, D., Sartori, S., 1990, A Method for thermal deformation correction of CMMs, Annals of the CIRP, Vol. 39/1.

[Beltman, 1997]
Beltman, A.J., 1997, A Piezo driven Stewart Platform, Proc. of NVPT Jubileumcongres Precisie Technologie, Leusden, The Netherlands.

[Binnig, 1982]
Binnig, G., Rohrer, H., Gerber, C.H., and Weibel, E., 1982, Surface studies by scanning tunnelling microscopy, Physical Review Letters, Vol. 49, p 57.

[Birch, 1994]
Birch, K.P., Downs, M.J., 1994, correction to the updated Edlén equation for the refractive index of air, Metrologia, Vol. 31, pp. 315-316.

[Bispink, 1992]
Bispink, T., 1992, Neue Wege zur Beurteilung hochgenauer Vorschubantriebe für Ultrapräzisions-maschinen, Ph.D. Thesis, Technische Hochschule Aachen, Germany, ISBN 3-86111-275-2.

[Blaedel, 1998]
Blaedel, K.L., 1998, Lawrence Livermore National Laboratory, CA, *Private communication*

[Blanding, 1992]
Blanding, D.L. 1992, Principles of Exact Constraint Mechanical Design, Eastman Kodak Company, Rochester, NY, 14650-1822 716-722-6382

[Blondeel, 1976]
Blondeel, E., Snoeys, R., Devrieze, L., 1976, Aerostatic Bearings with Infinite Stiffness, Annals of the CIRP, Vol. 25/1, pp. 291-296.

[Blondeel, 1980]
Blondeel, E., Snoeys, R., Devrieze, L., 1980, Dynamic Stability of Externally Pressurised Gas Bearings, Transactions of the ASME, Journal of Lubrication Technology, Vol. 102, pp. 511-518.

[Boley, 1960]
Boley, B., Weiner, J., 1960, Theory of thermal stresses, Krieger Publishing Company, Malibar, FL.

[Bollinger, 1982]
Bollinger, J.G., Stute, G., Van Brussel, H., 1982, Digital Controls and Feed Drives: State of the Art and New Developments, Annals of the CIRP, Vol. 29/2, pp. 497-506.

[Bonse, 1995]
Bonse, M.H.W., 1995, Capacitive position transducers, theoretical aspects and practical applications, PhD-thesis, Delft University of Technology, The Netherlands, ISBN 90-370-0135-1

[Bosch, 1998]
Bosch, P.P.J. van den, 1998, Eindhoven University of Technology, The Netherlands, *Private communication*.
[Bouwhuis, 1985]
Bouwhuis, G., Braat, J, Huijser, A., Pasman, J., Rosmalen, G. van, Schouhamer Immink, K., 1985, Principles of optical disc systems, Adam Hiler LTD, ISBN 0-85274-785-3.
[Braembussche, 1996]
Braembussche, P. van den, Swevers, J., Brussel, H. van, Vanherck, P., 1996, Accurate Tracking Control of Linear Synchronous Motor Machine Tool Axes, Mechatronics, Vol. 6, No. 5. Pp. 507-521.
[Braembussche, 1998]
Braembussche, P. van den, Swevers, J., Brussel, H. van, 1998, Control of the structural flexibility of a machine tool, Proc. of the International Seminar on Improving machine tool performance, San Sebastián, Spain, Vol. 1, pp. 339-350.
[Breyer, 1991]
Breyer, K.H., Pressel, H.G., 1991, Paving the way to thermally stable Coordinate Measuring Machines, Progress in Precision Engineering, pp. 56-75.
[Brussel, 1998]
Brussel, H. van, Braembussche, P. van den, 1998, Robust control of feed drives with linear motors, Annals of the CIRP.
[Bryan, 1967]
Bryan, J.B. et al, 1967, American Machinist, Special Report.
[Bryan, 1968]
Bryan, J.B., International status of thermal error research, Annals of the CIRP, Vol. 16/1, 1968.
[Bryan, 1979a]
Bryan, J.B, 1979, Design and Construction of an Ultra-Precision 84 inch Diamond Turning Machine, Precision Engineering, Vol. 1, No. 1, pp. 13-18.
[Bryan, 1979b]
Bryan, J.B., Carter, D.L., 1979, Design of a new error-corrected co-ordinate measuring machine, Precision Engineering, Vol. 1, No. 3, pp. 125-128.
[Bryan, 1979c]
Bryan, J.B., 1979, The Abbe Principle revisited: an updated interpretation, Precision Engineering, Vol. 1, No. 3, pp. 129-132.
[Bryan, 1982]
Bryan, J. et al, 1982, An order of magnitude improvement in thermal stability with use of liquid shower on a general purpose measuring machine, ASME Technical Paper MR82-936.
[Bryan, 1984]
Bryan, J.B., 1984, The Power of Deterministic Thinking in Machine Tool Accuracy, The first International Machine Tool Engineers Conference, Tokyo, Japan, Lawrence Livermore National Laboratory Report UCRL-91531.
[Bryan, 1990]
Bryan, J.B., 1990, International status of thermal error research, Annals of the CIRP, Vol. 39/2.
[Bryan, 1993]
Bryan, J.B., 1993, The deterministic approach in metrology and manufacturing, Proc. of the ASME International Forum on Dimensional Tolerancing and Metrology, Dearborn, Michigan.
[Bryan, 1996]
Bryan, J.B., 1996, Tutorial on Axis of Rotation, ASPE Annual Meeting, Monterey, CA.
[Bryant, 1986]
Bryant, M.R., Velinsky, S.A., Beachley, N.H., Fronczak, F.J., 1986, A Design Methodology for Obtaining Infinite Stiffness in an Aerostatic Thrust Bearing, Transactions of the ASME, Journal of Mechanisms, Transmissions, and Automation in Design, Proc. of Design Engineering Technical Conference, Columbus, Ohio, Paper No. 86-DET-32.
[Camera, 1976]
Camera, A., Favareto, M., D'Aprile, F., 1976, Analysis of thermal behaviour of a machine tool table using finite element method, Annals of the CIRP, Vol. 25/1.
[Cresto, 1991]
Cresto, P.C., Di Ciommo, M., Kancheva, T., Margues, D., Mudronja, V., Sartori, S., 1991, A method for the identification and correction of thermal deformations in 3D-CMMs, Measurement, 9(1).
[Corbett, 1997]
Corbett, J., McKeown, Course notes in Precision Engineering, School of Industrial and Manufacturing Science, Cranfield University, United Kingdom.
[Damen, 1997]
Damen, A., 1997, Complete system/control strategies, PATO Curcus Atieve Magnetische Lagering, Eindhoven University of Technology, The Netherlands.
[DeBra, 1991]
DeBra, D.B., 1991, Vibration isolation, Stanford University, CA, Tutorial Notes, ASPE Annual Meeting.
[DeBra, 1992]
DeBra, D.B. 1992, Vibration Isolation of Precision Machine Tools and Instruments, Annals of the CIRP, Vol. 41/2, Keynote Paper.
[DeBra, 1994]
DeBra, D.B., Rivin, G., 1994, Active vibration isolation, Stanford University, CA, Wayne State University, Detroit, Michigan, Tutorial Notes, ASPE Spring Topical Meeting, Cincinnati, OH.
[DeBra, 1998]
DeBra, D.B., 1998, Stanford University, CA, *Private communication.*
[Doebelin, 1990]
Doebelin, E.O., 1990, Measurement systems, Application and Design, McGraw-Hill International Editions, ISBN 0-07-100697-4
[Donaldson, 1980]
Donaldson, R.R., 1980, Error Budgeting in Technology of Machine Tools, Vol. 5, Machine Tool Accuracy (ed. R.J. Hocken, 1980).
[Donaldson, 1983]
Donaldson, R.R., Patterson, S.R., 1983, Design and Construction of a Large Vertical Axis Diamond Turning Machine, (LODTM), Proc. SPIE Tech. Symposium, pp. 433-438, UCRL. 89738.
[Donaldson, 1984]
Donaldson, R.R., Maddux, A.S., 1984, Design of a High-Performance Slide and Drive System for a Small Precision Machining Research Lathe, Annals of the CIRP, Vol. 33/1, pp. 243-248.
[Donaldson, 1986]
Donaldson, R.R., Thompson, D.C., 1986, Design and Performance of a Small Precision CNC Turning Machine, Annals of the CIRP, Vol. 35/1, pp. 373-376.
[Donmez, 1991]
Donmez, M.A., 1991, Progress Report of the Quality in Automation Project for F490 NIST, Gaithersburg, MD, NISTIR 4536.
[Eastman, 1935]
Easman, F.S., 1935, Flexure pivots to replace knife edges and ball bearings, Univ. Wash. Engng Exp. Sta. Bull. No. 86.
[Eastman, 1937]
Easman, F.S., 1937, The design of flexure pivots, J. Aero. Sci., 5(1), pp. 16-21.
[Edlén, 1966]
Edlén, B., 1966, The refractive index of air, Metrologia,

Vol. 2, pp. 71-80.
[Eijk, 1985]
Eijk, J. van, 1985, On the design of plate-spring mechanisms, Ph.D. Thesis, Delft University of Technology, The Netherlands.
[Eijk, 1998]
Eijk, J. van, 1998, Department Mechatronics, Centre For Manufacturing Technology, Philips Electronics N.V., The Netherlands, *Private communication.*
[Evans, 1989]
Evans, C., 1989, Precision Engineering, An Evolutionary View, Cranfield Press, Bedford, ISBN 1 871315 01 8
[Ewins, 1984]
Ewins, D.J., 1984, Modal Testing: Theory and Practice, Research Studies Press Ltd, United Kingdom.
[FEM, 1998]
Unigraphics® EDS Plano Texas; I-DEAS® SDRC Milfid Ohio; Algor® Pitsburg PA; Pro-Engineer®
[Ferri, 1995]
Ferri, A.A., 1995, Friction Damping and Isolation Systems, Transactions of the ASME, Special 50th Anniversary Design Issue, Vol. 117, pp. 196-206.
[Florussen, 1997]
Florussen, G., 1997, Accuracy Analysis of Multi-Axis Machines by Double Ball-Bar method, Masters Thesis, Eindhoven University of Technology, The Netherlands.
[Florussen, 1998]
Florussen, G., Delbressine, F., Schellekens, P., 1998, Accuracy analysis of multi-axis machines using a double ball bar, Proc. of the International seminar on improving machine tool performance, San Sebastián, Spain.
[Flügge, 1996]
Flügge, J., 1996, Vergleichende Untersuchungen zur meßtechnischen Leistungsfähigkeit von Laserinterferometern und inkrementellen Maßstabmeßsystemen, PTB-Bericht F-23, Thesis Technischen Hochschule Aachen, Germany, *in German.*
[Gardner, 1991]
Gardner, J.W., Hingle, H.T., 1991, From instrumentation to nanotechnology, Gordon and Breach Science publishers USA, ISBN 2-88124-794-6.
[Ge, 1996]
Ge, P., Jouaneh, M., 1996, Modelling and Control of Hysteresis in a Piezoceramic Actuator, Proc. of the ASPE Annual Meeting, Monterey, CA, pp. 396-399.
[Geary, 1954]
Geary, P.J., 1954, Flexure devices, Published by the British Scientific Instruments Research Association, Research Report M.18.
[Gijsbers, 1980]
Gijsbers, T.G., 1980, Colath, a numerically controlled lathe for very high precision, Philips Technical Review, Vol. 39, No. 9, pp. 229-244.
[Goossens, 1997]
Goossens, H.J., 1997, Design of a long range tool servo, Mechatronics post-masters design Thesis, Eindhoven University of Technology, The Netherlands, ISBN 90-5282-785-0.
[Groenhuis, 1991]
Groenhuis, H., 1991, A design tool for electromechanical servo systems, Ph.D. Thesis, Twente University, The Netherlands, ISBN 90-9004089-1.
[Gross, 1983]
Gross, H., 1983, Electrical Feed Drives for Machine Tools, John Wiley & Sons Ltd, NY.
[Haitjema, 1997]
Haitjema, H., Rosielle, P.C.N.J., Steijaert, H.A.M., 1997, A Calibration Platform for Generating Parallel Displacements with Nanometre Accuracy, Proc. of the 9th IPES-4UME, Braunschweig, Germany, Vol. 2, pp. 456-459.
[Haranat, 1987]
Haranat, S., Ganesan, N., Rao, B., 1987, Dynamic analysis of machine tool structures with applied damping treatment, International Journal of Machine Tools and Manufacturing, Vol. 27, pp. 43-55.
[Henrikson, 1974]
Henrikson, C.L., Lyman, J., Studer, P.A., 1979, Magnetically suspended momentum wheels for spacecraft stabilization, AIAA 12th Aerospace Sciences Meeting, Washington, DC.
[Higuchi, 1990]
Higuchi, T. (ed.), 1990, Proc. of the 2nd International Symposium on Magnetic Bearings, Tokyo, Japan.
[Hocken, 1977]
Hocken D.J. et al, 1977, Three dimensional metrology, Annals of the CIRP, Vol. 26/2. Pp. 403-408.
[Hocken, 1986]
Hocken, R., et al. NIST, 1986, A general methodology for machine tool accuracy enhancement by error compensation, Precision Engineering, Vol. 8/4, pp. 187-196.
[Hocken, 1995]
Hocken, R., 1995, Course Notes, Principles Active in the Design of Precision Machines.
[Hoek, 1962-1986]
Hoek, W. van der, 1962-1986, Het voorspellen van het Dynamisch Gedrag en Positioneringsnauwkeurigheid van constructies en mechanismen, Lecture note-book, Eindhoven University of Technology, The Netherlands, *in Dutch.*
[Hoek, 1985-1989]
Hoek, W. van der, Koster, M.P., 1985-1989, Des Duivels Prentenboek, Lecture note-book, Eindhoven University of Technology, The Netherlands, *in Dutch.*
[Holmes, 1996]
Holmes, M., Trumper, D., 1996, Magnetic/fluid-bearing stage for atomic-scale motion control (the angstrom stage), Precision Engineering, Vol.18, No.1, pp. 38-49.
[Holster, 1967]
Holster, P.L., 1967, Gaslagers met uitwendige drukbron, P.T. Werktuigbouw, Vol. 22, No. 9, pp. 363-370, No. 10, pp. 415-425, No. 26, p. 1105, *in Dutch.*
[Holster, 1987]
Holster, P.L., Jacobs, J.A.H., 1987, Theoretical analysis and experimental verification on the static properties of externally pressurized air-bearing pads with load compensation, Tribology International, Vol. 20, No. 5, pp. 276-289.
[Hou, 1992]
Hou, W., Wilkening, G., 1992, Investigation and compensation of the nonlinearity of heterodyne interferometers, Precision Engineering, Vol. 14, p.p. 91-98
[Hutcheson, 1996]
Hutcheson, G., Hutcheson, J., 1996, Technology and Economics in the Semiconductor Industry, Scientific American.
[Jones, 1951]
Jones, R.V., 1951, Parallel and rectilinear spring movements, J. Sci. Instrum., Vol. 28, No. 38.
[Jones, 1956]
Jones, R.V., Young, I.R., 1956, Some parasitic deflections in parallel spring movements, J. Sci. Instrum., Vol. 33, No. 11.

[Jouy, 1986]
Jouy, F., 1986, Theoretical modelisation and experimental identification of the geometrical parameters of coordinate measuring machines by measuring a multidirected bar, Annals of the CIRP, Vol. 35/1.

[Kamerbeek, 1983/84]
Kamerbeek, E.M.H., 1983/84, Magnetic bearings, Philips Technical Review, Vol. 41, No. 11/12, pp. 348-361.

[Kamerbeek, 1993]
Kamerbeek, E.M.H., 1997, Magnetische velden, krachten en actuatoren, PATO cursus Actieve Magnetische Lagering, Eindhoven University of Technology, The Netherlands.

[Kanai, 1990]
Kanai, A., Sano, H., Yoshioka, J., Miyashita, M., 1990, Positioning of a 200 kg Carriage on Pain Bearing Guideways to Nanometre Accuracy with a Force-Operated Linear Actuator, Nanotechnology, pp. 43-51.

[Kazimierski, 1992]
Kazimierski, Z., Brzeski, L., Horodko, L., 1992, Gas Bearing of Infinite Stiffness, Transactions of ASME, Journal of Tribology, Vol. 114, No. 1, pp. 270-273.

[Kerstiens, 1990]
Kerstiens, P., 1990, Strukturelemente für Produktionsmaschinen aus carbonfaserverstärkten Kunststoffen, Ph.D. Thesis, Technische Hochschule Aachen, Germany, *in German*.

[Kim, 1996]
Kim, W-J., Trumper, D.L., 1996, Force Ripple in Surface-Wound Permanent-Magnet Linear Motors, Proc. of the Intermag, Seattle, WA.

[Klafter, 1989]
Klafter, R.D., et al., 1989, Robotic Engineering, an integral approach, Englewood Cliffs, Prentice-Hall Int. ISBN. 0-13-782053-4

[Klostermann, 1987]
Klostermann, J.A., 1987, Een metaalkundige visie op de mechnanische eigenschappen van keramische materialen, De Constructeur, Okt. 1987, No. 10, pp. 42-51, *in Dutch*.

[Knapp, 1988]
Knapp, W., 1988, Accuracy of Length Measurement and Positioning: Statical Measurement and Contouring Mode, Annals of the CIRP, Vol. 37/1.

[Knapp, 1998]
Knapp, W., 1998, Ingenieurbüro Knapp, *Private communication*.

[Koren, 1975]
Koren, Y., 1975, Interpolator for a CNC System, IEEE Trans. Comp. Vol. C-25 No. 1, pp. 32-37.

[Koren, 1981]
Koren, Y., Massory. O., 1981, Reference-Pulses Circular Interpolators for CNC Systems, ASME Journal of Engineering for Industry, Vol. 103, No. 1, pp. 131-136.

[Koren, 1997]
Koren, Y., 1997, Control of Machine Tools, Journal of Manufacturing Science and Engineering, Vol. 119, pp. 749-755.

[Koster, 1996]
Koster, M.P., Constructieprincipes, Uitgeverij Universiteit Twente, Enschede, The Netherlands, ISBN 9036508320, *in Dutch*.

[Koster, 1997]
Koster, M.P., 1997, Smart Disc, Proc. of NVPT Jubileumcongres Precisie Technologie, Leusden, The Netherlands, *in Dutch*.

[Koster, 1998]
Koster, M.P., 1998, Twente University, The Netherlands, *Private communication*.

[Kouno, 1984]
Kouno, E., 1984, A Fast Piezoelectric Actuator for Servo Correction of Systematic Errors in Precision Machining, Annals of the CIRP, Vol. 33, pp. 369-372.

[Kraakman, 1969]
Kraakman, H.J.J., Gast, d., J.G.C., 1969, Een precisiedraaibank met hydrostatische lagering en aandrijving, Philips Technisch Tijdschrift, Jaargang 30, No. 5, pp. 121-137, *in Dutch*.

[Kraakman, 1976]
Kraakman, H.J.J., 1976, Hydrostatische lagers met verhoogde stijfheid, De Constructeur, Nov. 1976, No. 11, pp. 51-57, *in Dutch*.

[Kraakman, 1977]
Kraakman, H.J.J., 1977, Hydrostatische lagers met verhoogde stijfheid I-III, P.T. Werktuigbouw, Vol. 32, No. 10, pp. 552-557, No. 11, pp. 611-616, No. 12, pp. 690-698, *in Dutch*.

[Kraakman, 1997]
Kraakman, H.J.J., 1997, Ontwerp van een precisie bewerkingsmodule, kracht, precisie en doelmatigheid door hydrauliek, Mikroniek, No. 6, pp. 182-187, *in Dutch*.

[Krulewich, 1995a]
Krulewich, D., 1995, Bibliography on Error Compensation, LLNL, 94551, Livermore, CA.

[Krulewich, 1995b]
Krulewich, D., Hale, L., Yordy, D., 1995, Rapid Mapping of Volumetric Errors, Proc. of ASPE Annual Meeting, Austin, USA, pp. 412-415.

[Krulewich, 1998]
Krulewich, D.A., 1998, Rapid Mapping of Volumetric Errors Using Distance Measurements, Proc. of International Seminar on Improving Machine Tool Performance, San Sebastián, Spain, Vol. 2, pp. 487-496.

[Kruth, 1994]
Kruth, J.P., Vanherck, P., Jonge, d. L., 1994, Self-calibration method and software error correction for three-dimensional coordinate measuring machines using artefact measurements, Measurement 14.

[Kunzmann, 1988]
Kunzmann, H., Wäldele, F., 1988, Performance of CMMs, Annals of the CIRP, Vol. 37/2, pp. 633-640.

[Kunzmann, 1990]
Kunzmann, H., Trapet, E., Wäldele, F., 1990, A uniform concept for calibration, acceptance test and periodic inspection of coordinate measuring machines using reference object, Annals of the CIRP, Vol. 39/1.

[Kunzmann, 1992]
Kunzmann H., Herrmann K., 1992, Calibration and testing of laser interferometers, Collection of papers and printed information of participants in the EUROMET-Workshop held on 11th and 12th March 1992, PTB-Bericht F-15, Berlin, Germany.

[Kunzmann, 1993]
Kunzmann, H., Pfeifer, T., Flügge, J., 1993, Scales vs. laser interferometers performance and comparison of two measuring systems, Annals of the CIRP, Vol. 42/2, pp 753-767.

[Kunzmann, 1996]
Kunzmann, H., 1996, PTB, Braunschweig, Germany, *Private communication*.

[Lanxide, 1990]
Lanxide™, Reinforced ceramics and metals, Product information, Lanxide, Newark, Delaware.

[Lazan, 1968]
Lazan, B.J., 1968, Damping of Materials and Members in Structural Mechanics, Pergamon, London, United Kingdom.

[Lindsey, 1988]
Lindsey, K., Smith, S.T., Robbie, C.J., 1988, Sub-Nanometre Surface Texture and Profile Measurement with Nanosurf 2, Annals of the CIRP, Vol. 37/1, pp. 519-522
[Lingard, 1991]
Lingard, P.S. Purss, M.E., Sona, C.M., Thwaite, E.G., Mariasson, G.H., Temperature perturbation effects in a high precision CMM, Precision Engineering, Vol. 13. No. 1, pp. 41-51.
[Loewen, 1980]
Loewen, E.G., 1980, Metrology problems in general engineering, a comparison with precision engineering, Annals of the CIRP, Vol. 29/2, pp. 451-453.
[Loxham, 1970]
Loxham, J., 1970, The Commercial Value of Investigations into Repeatability, Cranfield Unit for Precision Engineering, Bedford, United Kingdom.
[Lubin, 1982]
Lubin, G., 1982, Handbook of composites, Van Nostrand Reinhold, NY.
[Marsh, 1996]
Marsh, E.R., Slocum, A.H., 1996, An integrated approach to structural damping, Precision Engineering, Vol. 18, No. 2/3, pp. 103-109.
[Matsumura, 1996]
Matsumura, F., Okada, Y., Futjita, M., Namerikawa, T., (ed.), 1996, Proc. of the 5th International Symposium on Magnetic Bearings, Kanazawa, Japan.
[Maxwell, 1890]
Maxwell, J.C., 1890, General considerations concerning scientific apparatus, The Scientific Papers of James Clerk Maxwell, W. D., Niven, Ed., Dover Press, Reprint from the Handbook to the Special Loan Collection of Scientific Apparatus, 1876.
[McKeown, 1973]
McKeown, P.A., Loxham, J., 1973, Some Aspects of the Design of high precision machines, Annals of the CIRP, Vol. 22/1 pp. 139-140.
[McKeown, 1973-1997]
McKeown, P.A., Dunsdale, J., and Wills-Moren, W.I., 1973-1997, The design of high precision machines and systems, Course Notes, College of Manufacturing, Cranfield, United Kingdom.
[McKeown, 1986]
McKeown, P.A., 1986, High Precision Manufacturing and The British Economy, James Clayton Lecture, Inst. Mech. E. Roc., Vol. 200, No. 76.
[McKeown, 1987]
McKeown, P.A., 1987, The role of Precision Engineering in Manufacturing of the Future, Annals of the CIRP, Keynote, Vol. 36/2
[McKeown, 1990]
McKeowm, P.A., Carlisle, K., Shore, P., Read, R.F.J., 1990, Ultra-precision, high stiffness grinding CNC grinding machines for ductile mode grinding of brittle materials, SPIE, Vol. 1320, Infrared Technology and Application, pp. 301-313
[McKeown, 1997]
McKeown, P.A., 1997, New and Advanced Materials, Special Article, Nanotechnology, Emerging Technology Series.
[McKeown, 1998]
McKeown, P.A., 1998, Cranfield University, United Kingdom, *Private communication,*
[Meijer, 1989]
Meijer, J., 1989, From straightness to flatness, Ph.D. Thesis, Twente University, The Netherlands.
[Meirovitch, 1975]
Meirovitch, L., 1975, Elements of Vibration Analysis, McGraw-Hill, NY, pp. 93-100, 115-117.
[Mierlo, 1993]
Mierlo, H.A., van, 1993, Ontwerpen en testen van delay line systemen voor optische apertuur synthese, Masters Thesis, Eindhoven University of Technology, The Netherlands, *in Dutch.*
[Möhrke, 1991]
Möhrke, G., 1991, Mehrdimensionale Geometrieerfassung mit optoelektronischen Triangulationssensoren, Ph.D. Thesis Technischen Hochschule Aachen, Germany, *in German.*
[Moriwaki, 1995]
Moriwaki, T., Schamamoto, E., Kawano, M., 1995, Estimation of Thermal Deformation of Machine Tool by Applying Neural network, Transaction of JSME, Part C, Vol. 61, No. 584.
[Murty, 1982]
Murty, A.S.R., Padmanabhan, K.K., 1982, Effect of surface topography on damping in machine joints, Precision Engineering, Vol. 4, pp. 185-190.
[Nakazawa, 1994]
Nakazawa, H., 1994, Principles of Precision Engineering, Oxford University press UK, ISBN 0-19-856266-7.
[Nashif, 1985]
Nashif, A.H., Jones, D.I.G., Herderson, J.P., 1985, Vibration Damping, John Wiley & Sons, NY.
[Neumann, 1988]
Neumann, H.J., et al, 1988, CNC-Koordinaten-meßtechnik, Expert Verlag, Ehningen, ISBN 3-8169-0220-0.
[Ni, 1997]
Ni, J., 1997, CNC Machine Accuracy Enhancement through Real Time Error Compensation, Journal of Manufacturing Science and Engineering, Vol. 119.
[Nijs, 1988]
Nijs, J. de, Schellekens, P., 1988, Modelling of Coordinate Measuring Machine for its Dynamic behaviour, Guyan Reduction Technique, Annals of the CIRP, Vol. 37/1.
[Nijsse, 1997]
Nijsse, G., 1997, Verplaatsingssensoren, PATO cursus Actieve Magnetische Lagering, Eindhoven University of Technology, The Netherlands, *in Dutch.*
[Paros, 1965]
Paros, J.M., Weisbord, L., 1965, How to design flexure hinges, Machine Design, Nov. 1965.
[Patterson, 1985]
Patterson, S., Magrab, E., 1985, Design and Testing of a Fast Tool Servo for Diamond Turning, Precision Engineering, Vol. 7, No. 3, pp. 123-128.
[Paul 1981]
Paul, R.P., 1981, Robot Manipulators: Mathematics, programming and Control, MIT Press, Cambridge, MA.
[Peklinik, 1970]
Peklinik, J., 1970, Adaptive Control of Manufacturing Systems, Annals of the CIRP, Vol. 18/1, pp. 265-272.
[Plessers, 1988a]
Plessers, P., Snoeys, R., 1988, Dynamic Identification of Convergent Externally Pressurized Gas-Bearing Gaps, Transactions of the ASME, Journal of Tribology, Vol. 110, pp. 263-270.
[Plessers, 1988b]
Plessers, P., Snoeys, R., 1988, Dynamic Stability of Mechanical Structures Containing Externally Pressurized Gas-Lubricated Thrust Bearings, Transactions of the ASME, Journal of Tribology, Vol. 110, pp. 271-278.
[Philips, 1995]
Philips, D.M.H., Rosielle, P.C.J.N., Schellekens, P.H.J., 1995, Design of a High Precision Rotary Table, Proc. of 8th IPES, Compiègne, France, pp.221-224.

[Pohl, 1986]
Pohl, D.W., 1986, Some design criteria in scanning tunnelling microscopy, IBM Journal of Research and Development, Vol. 30, p 428.
[Pollard, 1922]
Pollard, A.F.C., 1922, The Mechanical Design of Scientific Instruments, Cantor Lectures, Imperial College of Science and Technology, Parsons Ltd. Hastings.
[Pollard, 1929-1951]
Pollard, A.F.C., 1929-1951, The Kinematic Design of couplings in Instrument Mechanisms, Imperial College of Science and Technology, Hilger and Watts Ltd., London, United Kingdom.
[Portman, 1997]
Portman, V.T., Weill, R.D., Shuster, V.G., 1997, High order Approximation in Accuracy Computations of Machine Tools and Robots, Journal of Manufacturing Science and Engineering, Vol. 119.
[Pritschow, 1990]
Pritschow, G., Philipp, W., 1990, Direct Drives for High-Dynamic Machine Tool Axes, Annals of the CIRP, Vol. 39, pp. 413-416.
[Quenelle, 1983]
Quenelle, R.C., Wuerz, L.J., 1983, A New Microcumputer-Controlled Laser Dimensional Measurement and Analysis System, Hewlett-Packard Journal, 2-22.
[Rankers, 1994]
Rankers, A.M., Eijk, J. van, 1994, The Influence of Reaction Forces on the Behaviour of High Performance Motion Systems, Proc. of the Second International Conference on Motion and Vibration Control, Yokohama, pp. 711-716.
[Rankers, 1997]
Rankers, A.M., 1997, Machine dynamics in mechatronic systems, An engineering approach, Ph.D. Thesis, Twente University, The Netherlands, ISBN 90-365-0957-2
[RaO, 1990]
RaO, S.S., 1990, Mechanical Vibrations, Addison Wesley Publishing Company, USA, ISBN 0-201-50156-2.
[Reindl, 1967]
Reindl, R., 1967, Dr Otto Eppenstein – A Scientist of the Jena Zeiss works, Jena Review, No. 5, pp 263-267.
[Reinhart, 1997]
Reinhart, G., Weiszenberger, M., 1997, Eigenschaftsoptimierung am virtuellen Prototyp, ZWF 92, 1997,11, Germany.
[Renkens, 1994]
Renkens, M.J.M., 1994, Ontwerp van een geregeld axiaal lager, Mechatronics post-masters design Thesis, Eindhoven University of Technology, The Netherlands, ISBN 90-5282-329-4, *in Dutch.*
[Renkens, 1997]
Renkens, M.J.M., 1997, Design of an Axially Controlled Spindle Unit for High Precision Diamond Turning, Ph.D. Thesis, Eindhoven University of Technology, The Netherlands, ISBN 90-74445-35-7.
[Riven, 1979a]
Riven, E., 1979, Vibration Isolation of Precision Machinery, Sond and Vibration
[Riven, 1979b]
Riven, E., 1979, Principles and Criteria of Vibration Isolation of Machinery, ASME, Journal of Mech. Design, Vol. 101, p 682.
[Riven, 1989]
Riven, E., Kang, H., 1989, Improvement of machining conditions for slender parts by tuned dynamic stiffness of tool, International Journal of Machine Tools and Manufacturing, Vol. 29, pp 361-376.
[Riven, 1996]
Riven, E.I., 1996, Stiffness in Design, Wayne State University, Detroit, Michigan, Tutorial Notes, ASPE Annual Meeting, Monterey, CA.
[Ro, 1993]
Ro, P.I., Hubbel, P.I., 1993, Model Reference Adaptive Control of Dual-Mode Micro/Macro Dynamics of Ball Screws for Nanometre Motion, ASME, Transactions Journal of Dynamic Systems, Measurement, and Control, Vol. 115, pp. 103-108.
[Rogers, 1883]
Rogers, W.A., 1883, Studies in metrology, Proc. of the American Academy of Arts and Sciences, Vol. 18, pp 287-398.
[Rolt, 1929]
Rolt, F.H., 1929, Gauges and fine measurements, Macmillan, London, United Kingdom.
[Rosielle, 1991]
Rosielle, P.C.J.N., 1991, Een meer efficiënte vorm van Gatscharnieren, Mikroniek, No. 4, pp. 104-108, *in Dutch.*
[Rosielle, 1994]
Rosielle, P.C.J.N., 1994, Development and design rules for friction wheel drives, Proc. of ASPE Spring Topical Meeting, Mechanisms and Control for Ultra-precision Motion, Cincinnati, OH.
[Rosielle, 1996]
Rosielle, P.C.J.N., Vermeulen, J.P.M.B., Reker, E.A.G., 1996, Laminaten, vervaardigd uit epoxy-verlijmde, dunne Al_2O_3-keramische tegels, geschikt voor sledeconstructies in precisietoepassingen, Licence no. 1002369, *in Dutch.*
[Rosielle, 1998]
Rosielle, P.C.J.N., Reker, E.A.G., 1998, Constructiepricipes 1, Bedoeld voor het nauwkeurig bewegen en positioneren, Lecture note-book, Eindhoven University of Technology, The Netherlands, *in Dutch.*
[Sabnis, 1974]
Sabnis, A.V., Dendy, J.B., Schmitt, F.M., 1974, Magnetically suspended large momentum wheels, Proc. of AIAA Mechanics and Control Flight Conf., Anaheim, CA.
[Sartori, 1988]
Sartori, S., Cresto, P.C., Dicionimo, M., Kancheva, T., 1988, A way to improve the accuracy of a coordinate measuring machine, Measurement 6(2).
[Sartori, 1995]
Sartori, S, Zang, G.X., 1995, Geometric Error Measurement and Compensation of Machines, Annals of the CIRP, Vol. 44/2
[Sartori, 1998]
Sartori, A., 1998, Istituto di Metrologia "G. Colonnetti", Italy, *Private communication*
[Schellekens, 1986]
Schellekens, P.H.J., 1986, Absolute meetnauwkeurigheid van technische laserinterferometers, Ph.D. Thesis, Eindhoven University of Technology, The Netherlands, *in Dutch.*
[Schellekens, 1992]
Schellekens, P.H. et al, 1992, Modelling of thermal behaviour of machine tools, Development of Methods for numerical error correction of Machine tools, Bureau Communantaire de References, 3320/1/0/160/89/8 - BCR – NL, Brussels, Belgium.
[Schothorst, 1997]
Schothorst, G., 1997, Piezo aangedreven vervormbare spiegel, Proc. of NVPT Jubileumcongres Precisie Technologie, Leusden, The Netherlands, *in Dutch.*

[Schouten, 1997]
Schouten, C.H., 1997, Het ontwerp van een precisie spindel voor hoge snelheden, Mechatronics post-masters design Thesis, Eindhoven University of Technology, The Netherlands, ISBN 90-5282-754-0, *in Dutch.*
[Schweitzer, 1988]
Schweitzer, G. (ed.), 1988, Proc. of the 1st International Symposium on Magnetic Bearings, Springer-Verlag, Berlin, Germany.
[Schweitzer, 1994]
Schweitzer, G., Siegwart, R., Herzog, R. (ed.), 1994, Proc. of the 4th International Symposium on Magnetic Bearings, VdF-Verlag, Zurich, Switzerland.
[Shpitalni, 1994]
Shpitalni, M. Koren, Y., Lo, C., 1994, Real-Time Curve Interpolators, Computer Aided Design, Vol. 26, pp. 832-838.
[Slocum, 1992]
Slocum, A., 1992, Precision Machine Design, Englewood Cliffs, NJ, Prentice Hall.
[Slocum, 1994a]
Slocum, A.H., Marsh, E.R., Smith, D.H., 1994, A new damper design for machine tool structures: the replicated internal viscous damper, Precision Engineering, Vol. 16, No. 3, pp. 174-183.
[Slocum, 1994b]
Slocum, A., Scagnetti, P., Kane, N., 1994, Ceramic Machine Tool Axis with Self-Compensated Water-Hydrostatic Linear Bearings, Proc. of the ASPE Annual meeting, Cincinnati, OH, pp. 57-60.
[Smith, 1992]
Smith, S.T., Chetwynd, D.G., Foundations of Ultra-precision Mechanism Design, Gordan and Breach Science Publishers.
[Snoeys, 1977]
Snoeys, R., Devrieze, L., Vanherck, P., 1977, Self aligning aerostatic shoe bearing, Annals of the CIRP, Vol. 25/1, pp. 205-210.
[Snoeys, 1987]
Snoeys, R., Al-Bender, F., 1987, Development of improved externally pressurized gas bearings, KSME Journal, Vol. 1, No. 1, pp. 81-88.
[Soons, 1993]
Soons, H., 1993, Accuracy Analysis of Multi-Axis Machines, Ph.D. Thesis, Eindhoven University of Technology, The Netherlands.
[Spaan, 1995]
Spaan, H.A.M., 1995, Software Error Compensation of Machine Tools, Ph.D. Thesis, Eindhoven University of Technology, The Netherlands.
[Spies, 1997]
Spies, A., 1997, Linear and Angular Encoders for the High-Resolution range, Proceedings of the 9th International Precision Engineering Seminar.
[Srinivasan, 1992]
Srinivasan, N., Ziegert, J., Smith, S., 1992, Prediction of Positional Errors of a Three Axis Machine Tool Using Neural Network, Japan/USA, Symposium on Flexible Automation.
[Srinivasan, 1997]
Srinivasan, K., Tsu Chin Tsao, 1997, Machine Tool Feed Drives and Their Control – A Survey of the State of the Art, Journal of Manufacturing Science and Engineering, Vol. 119, pp. 743-748.
[Stix, 1996]
Stix, G., 1996, Waiting for Breakthroughs, Scientific American.
[Stout, 1981]
Stout, K.J., 1991, The design of externally pressurized gas bearings.
[Takahashi, 1990]
Takahashi, M., Otsuka, J., Ono, K., 1990, Study of Precision Positioning using a Friction Drive (I), Structure and Traction Characteristics, Bull. JSPE, Vol. 24, No. 1, pp. 21-26.
[Taniguchi, 1983]
Taniguchi, N., 1983, Current status in and future trends of ultra-precision machining and ultra fine materials processing, Annals of the CIRP, Vol. 32/2
[Taniguchi, 1996]
Taniguchi, N., 1996, Nanotechnology: Integrated processing systems for ultra precision and ultrafine products, Oxford University Press, UK, ISBN 0-19-8562837.
[Teague, 1986]
Teague, E.C., 1986, Room temperature gold-vacuum-gold-gold tunnelling experiments, NBS Journal of Research, Vol. 91, p 171.
[Teague, 1989-1997]
Teague, E.C., Evans, C., 1989, Patterns for precision instrument design (mechanical aspects), National Institute of Standards and Technology, Tutorial Notes, ASPE Annual Meeting, Norfolk, Virginia.
Teague, E.C., 1997, Basic Concepts for Precision Instrument Design: Designing Instruments and Machines to Have a High Degree of Repeatability, National Institute of Standards and Technology, Tutorial Notes, ASPE Annual Meeting, Norfolk, Virginia.
Teague, E.C., 1997, Basic Concepts for Precision Instrument Design: Some Draft Notes and Papers, National Institute of Standards and Technology, Tutorial Notes, ASPE Annual Meeting, Norfolk, Virginia.
[Teeuwsen, 1989]
Teeuwsen, J., Soons, J., Schellekens, P., 1989, A General Method for error description of CMM's using polynomial fitting procedures, Annals of the CIRP, Vol. 38/1.
[Tensator, 1997]
Tensator, Constant Force Springs, Catalogue 2nd Edition, Tensator Limited, United Kingdom.
[Thompson, 1989]
Thompson, D.C., 1989, The design of an ultra-precision CNC measuring machine, Annals of the CIRP, Vol. 39/1
[Timoshenko, 1974]
Timoshenko, S., Young, D.H., Weaver, W., 1974, Vibration Problems in Engineering, John Wiley, NY.
[Tomizuka, 1987]
Tomizuka, M., Zero Phase Error Tracking Algorithm for Digital Control, ASME Journal of Dynamic Systems, Measurement and Control, Vol. 109, pp. 65-68.
[Tomizuka, 1989]
Tomizuka, M., 1989, Design off Digital Tracking Controllers for Manufacturing Applications, ASME, Manufacturing Review, Vol. 2, No. 2, pp. 134-141.
[Trapet, 1997a]
Trapet, E., 1997, Physikalisch Technische Bundesanstalt, Braunschweig, Germany, Workshop on Traceability of Coordinate Measuring Machines, Oct. 9-10, 1997.
[Trapet, 1997b]
Trapet, E., 1997, PTB, Braunschweig, Germany, *Private communication.*
[Trumper, 1993]
Trumper, D.L., Williams, M.E., Nguyen, T.H., 1993, Magnet Arrays for Synchronous Machines, Proc. of the IEEE Ind. Appl. Soc. Annual Meeting, Toronto, Canada.

[Trumper, 1996]
Trumper, D.L., Kim, W.J., Williams, M.E., 1996, Design and Analysis Framework for Linear Permanent-Magnet Machines, IEEE Transactions on Industry Applications, Vol. 32, No. 2, pp. 371-379.
[Tsutsumi, 1979]
Tsutsumi, M., Ito, Y., 1979, Damping mechanism of a bolted joint in machine tools, Proc. of the International Machine Tool Design Research Conference, 1979, pp. 443-448.
[VDI, 1985]
VDI Berichte 548, 1985, Dokumentation Laserinterferometrie in der Längenmesstechnik, Verlag des Vereins Deutcher Ingenieure Düsseldorf, Germany, *in German*.
[Veldhuis, 1994]
Veldhuis, S., Elbestawi, M., 1994, Modelling and compensation for five-axis machine tool errors, International Mechanical Engineering Congress and Exposition Conference, ASME, Chicago, Vol. 68/2.
[Vermeulen, 1996a]
Vermeulen, J.P.M.B., 1996, Development of a lightweight and ultrastiff single point diamond turning machine with submicron machining accuracy, Mechatronics post-masters design Thesis, Eindhoven University of Technology, The Netherlands, ISBN 90-5282-702-8.
[Vermeulen, 1996b]
Vermeulen, J.P.M.B., Rosielle, P.C.J.N., Schellekens, P.H.J., 1996, Innovations in the design and development of ultraprecision machines, Lightweight and ultrastiff single point diamond turning machine with submicron accuracy, Proc. of ASPE Annual Meeting, Monterey, CA, pp. 631-636.
[Vermeulen, 1998]
Vermeulen, M.M.P.A., Rosielle, P.C.J.N., Schellekens, P.H.J., 1998, Design of a High-precision 3D-Coordinate Measuring Machine, Annals of the CIRP, Vol. 47/1
[Vorburger, 1997]
Vorburger, T.V., Dagata, J.A., Wilkening, G., Iizuka, K., 1997, Industrial Uses of STM and AFM, Annals of the CIRP, Vol. 46/2, Keynote Paper
[Wang, 1993]
Wang, J., 1993, Design of Gas Bearing Systems for Precision Applications, Ph.D. Thesis, Eindhoven University of Technology, The Netherlands, ISBN 90-386-0112-3.
[Weck, 1984]
Weck, M., 1984, Handbook of Machine Tools, Vol. 3, Automation and Controls, Wiley, NY.
[Weck, 1995a]
Weck, M., McKeown, P., Bonse, R., Herbst, U., 1995, Reduction and Compensation of Thermal Errors in Machine Tools, Annals of the CIRP, Vol. 44/2.
[Weck, 1995b]
Weck, M., Öezmeral, H., Mehlkopp, K., Terwei, T., 1995, New Hybrid Concept for long Stroke Fast-Tool-Servo-System, American Society For Precision Engineering, Proc. Vol. 12, pp. 211-214.
[Weck, 1997]
Weck, M. Oezmeral, H., Mehlkopp, K., Terwei, T., A Long Stroke Fast-Tool-Servo with Air Bearings, Progress in Precision Engineering and Nanotechnology, Proc. of 9th IPES-4UME, Vol. 2, pp. 393-394
[Weekers, 1997]
Weekers, W.G., 1997, Compensation for Dynamic Errors of Coordinate Measuring Machines, Ph.D. Thesis, Eindhoven University of Technology, The Netherlands, ISBN 90-386-0178-6.
[Wetzels, 1998]
Wetzels, S.F.C.L., 1998, Laser Based Displacement Calibration with Nanometre Accuracy, Ph.D. Thesis, Eindhoven University of Technology, The Netherlands, ISBN 90-386-0730-X
[Wills-Moren, 1982]
Wills-Moren, W.J., Modjarrad, A. Read, R.F.J., and McKeown, P.A., 1982, Some Aspects of the Design and Development of a large, High Precision CNC Diamond Turning Machine, Annals of the CIRP, Vol. 31, pp. 409-414.
[Wills-Moren, 1989]
Wills-Moren, W.J., Wilson, T., 1989, The Design and Manufacture of a large CNC Grinding Machine for Off-axis Mirror Segments, Annals of CIRP, Vol. 38/1
[With, 1996]
With, d., G., Dortmans, L.J.M.G., 1996, Constructief gebruik van brosse materialen, De Constructeur, Okt. 1996, No. 10, pp. 30-37, *in Dutch*.
[Wu, 1996]
Wu, C., Su, C., 1996, Nonlinearity in measurements of length by optical interferometry, Meas. Sci. Technol. Vol. &, pp. 62-68
[Wulp, 1996]
Wulp, H. van der, Beek, H.F. van, Mertens, B.M., 1996, A piezodriven TEM stage with subnanometre position stability, Proc. of ASPE Annual Meeting, pp. 26-31.
[Yang, 1996]
Yang, S., Yuan, J., Ni, J., 1996, Real Time Cutting Force Induced Error Compensation on a Turning Center, Proceedings of the Second S.M. Wu Symposium, Ann Arbor.
[Yoshida, 1967]
Yoshida, Y., Honda, F., 1967, Thermal deformations of machine tool structure-the bed of a lathe, Annals of the CIRP, Vol. 15/1.
[Young, 1972]
Young, R.D., Ward, J., Scire, F., 1972, The Topografiner: an instrument for measuring surface topography, Review of Scientific Instruments, Vol. 43, p 999.
[Zhang, 1985]
Zhang, G.J., et al, 1985, Error compensation of coordinate measuring machines, Annals of the CIRP, Vol. 34/1.
[Ziegert, 1994]
Ziegert, J.C., Mize, C.D., 1994, The laser Ball Bar: A new Instrument for Machine Tool Metrology, Journal of the American Society for Precision Engineering, Vol. 16, No. 4.