Precision Machine System Design- Design strategy

Design strategy for precision machine system:

1.Conceptual Design

2.Detail Virtual Design

3.Simulation and Modeling

with continuous improvement

4.Prototype Manufacture

5.Test/Evaluation

6.Validation and Commissioning

To achieve precision machine design, Key design parameters should be identified and applied.

Main Key Design Principles:

1. High structural stiffness, damping, and thermal stability of machine structure
2. Kinematic Design Principle for bearings and guideways in view of non-influencing drive

coupling and clamping

3. Abbe principle for Minimization of Abbe errors
4. Suitable metrology to minimize the Abbe offset and metrology frame to be independent from machine distortion.
5. Drives/actuation for high stiffness/accuracy
6. Control scheme for high axial stiffness, high response, high bandwidth
7. Error compensation and/or force compensation

Conceptual Design

-Initial investigation is extremely important; it is to check functionality of the specification

-Clear understanding of functionality required/clear understanding of specifications, various concepts can be drawn that may secure all functions.

-All aspects of the design discussed before starting any modeling and simulation.

-Several variants achieving the same objectives, but a systematic evaluation comparison for the best variant.

-'House of Quality', or 'System Matrix' can be a good systematic tool to compare between variants, giving the order of importance.

Case) House of Quality or System Matrix

	Factors				
Requirements	Material	Bearing	Control	Guides	Load
Stiffness(4)	5	5	3	2	3
L.Weight(3)	5	5	1	1	3
Accuracy(5)	1	5	5	5	5
Reliability(2)	1	3	3	1	0
Agility(1)	3	3	5	2	5
Sum	**46**	**69**	**51**	**40**	**51**

(Modified from S. Mekid, Introduction to precision machine design and error assessment, CRC Press)

Structural analysis

(By Analytical method and/or Finite Element Method)

1)Static Structure Analysis

Stiffness, Force-deformation, Stress-Strain, Yield and Fracture

2)Dynamic Structure analysis

Natural Freq, Vibration modes, Damping Coeffcient, Dynamic response

3)Thermal analysis

Thermal distribution, Thermal deformation

Machine Elements

-Carriage non-influencing drive coupling and clamping

-Position of drives on axes of reaction

-Bearing averaging, friction, wear, effect of temperature

-Drive/actuation: to avoid any loss or sub-transformation of movement that is sometimes completely non-repeatable

Metrology

-Sensing system independent from machine distortion

Control

-Servo-drive stiffness, position/velocity loop synchronization

-Error compensation

Useful Analytical Methods

(Modifed from Mekid's Introduction to precision machine design and error assessment, CRC Press)

Maxwell's Reciprocity theorm:

For two identical forces applied at distinct points i and j on a linear structure, the displacement at i caused by force at j is the dame as the displacement at j caused by the force at i. Thus, the stiffness matrix, or the inverse of the compliance matrix, is symmetric. That is,

$K_{ji}=F_i/X_j=F_j/X_i=K_{ij}$

Betti's Interpretation:

The indirect or mutual work done by a loading system i during the application of a new loading system j is equal to the work done by the loading system j during the application of the loading system i.

Ex) For forces P, Q applied to simple beam at 1 and 2 locations $\rightarrow P\Delta_2=Q\Delta_1$

Castigliano Theorem

: Energy method for calculating displacements

If the strain energy of a linearly elastic structure can

be expressed as a function of generalized force Q_i ;

Then the partial derivative of the strain energy with respect to generalized force gives the generalized

displacement q_i in the direction of Q_i.

Thus $q_i=\partial U/\partial Q_i$,

where U=total strain energy

=Strain energy due to Tension/Compression

+Strain energy due to Bending

+Strain energy due to Torsion

+Strain energy due to Shear

$=\int P^2/2EAdx+\int M^2/2EIdx+\int T^2/2GJdx+\int V^2Q^2/2Gb^2I^2Adx$

Where, P=Tensile/Compressive force, M=Bending

moment, T=Twisting moment, V=Shear force,

E=Young's modulus;

I=2nd area moment of Inertia$=\int y^2dA$

A= Cross sectional area

J=Polar moment of Inertia=$\int r^2 dA$

G=Shear modulus =E/2(1+ν)

Q=1st area moment of inertia=$\int y dA$

B=width of beam

L=$\int dx$=Length of Beam

<u>Lagrange Equation</u>:

A system of rigid bodies is in dynamic equilibrium when the virtual work of the sum of the applied forces and the inertial forces is zero for any virtual displacement of the system. This results in sets of motion of equations (Lagrange equation)for the rigid body dynamics

$$d/dt\{\partial L/\partial(dq_j/dt)\}-\partial L/\partial q_j = Q_j,\ j=1,2,..m$$

where L is the Lagrangean which is the kinetic energy minus elastic energy, T-U.

q_i is the displacement, Q_i is the applied force.

Saint Venant Principle:

The localized effects caused by any load acting on the body will dissipate or reduce within regions that are sufficiently away from the location of the load.

Centre of mass and permanent equilibrium:

Permanent equilibrium is usually determined by the position of centre of mass(CM). When the CM is a point within the structure, the structure can be balanced at that point.

Finite Element Analysis: (*modified from Wikipedia*)

: The origin of finite method can be traced to the matrix analysis of structures where the concept of a displacement or stiffness matrix approach was introduced. Finite element concepts have been further developed based on engineering methods. It can provide very useful and powerful tool for numerical calculation for static analysis (force-deformation, stress-strain, thermal deformation, etc), dynamic analysis (natural frequency, vibration modes, etc), thermal analysis(temperature, heat flux, conduction, convection, radiation), fluid analysis(fluid flow, fluid dynamics, etc), electro-magnetic analysis(electric filed, magnetic field, field flux, etc).

Elements

- Straight or curved one-dimensional elements with physical properties such as axial, bending, and torsional stiffness. This type of element is suitable for modeling cables, braces, trusses, beams, stiffeners, grids and frames. Straight elements usually have two nodes, one at each end, while curved elements will need at least three nodes including the end-nodes. The elements are positioned at the centroidal axis of the actual members.
- Two-dimensional elements that resist only in-plane forces by membrane action (plane stress,

plane strain), and plates that resist transverse loads by transverse shear and bending action (plates and shells). They may have a variety of shapes such as flat or curved triangles and quadrilaterals. Nodes are usually placed at the element corners, and if needed for higher accuracy, additional nodes can be placed along the element edges or even within the element. The elements are positioned at the mid-surface of the actual layer thickness.

- Torus-shaped elements for axisymmetric problems such as membranes, thick plates, shells, and solids. The cross-section of the elements are similar to the previously described types: one-dimensional for thin plates and shells, and two-dimensional for solids, thick plates and shells.
- Three-dimensional elements for modeling 3-D solids such as machine components, dams, embankments or soil masses. Common element shapes include tetrahedrals and hexahedrals. Nodes are

placed at the vertexes and possibly in the element faces or within the element.

Element Interconnection and Displacement

The elements are interconnected only at the exterior nodes, and altogether they should cover the entire domain as accurately as possible. Nodes will have nodal (vector) displacements or degrees of freedom which may include translations, rotations, and for special applications, higher order derivatives of displacements. When the nodes displace, they will *drag* the elements along in a certain manner dictated by the element formulation. In other words, displacements of any points in the element will be interpolated from the nodal displacements, and this is the main reason for the approximate nature of the solution.

Practical consideration

From the application point of view, it is important to model the system such that:

- Symmetry or anti-symmetry conditions are exploited in order to reduce the size of the model.
- Displacement compatibility, including any required discontinuity, is ensured at the nodes, and preferably, along the element edges as well, particularly when adjacent elements are of different types, material or thickness.

  Compatibility of displacements of many nodes can usually be imposed via constraint relations.

- Elements' behaviours must capture the dominant actions of the actual system, both locally and globally.
- The element mesh should be sufficiently fine in order to produce acceptable accuracy. To assess accuracy, the mesh is refined until the important results shows little change. For higher accuracy, the aspect ratio of the elements should be as close

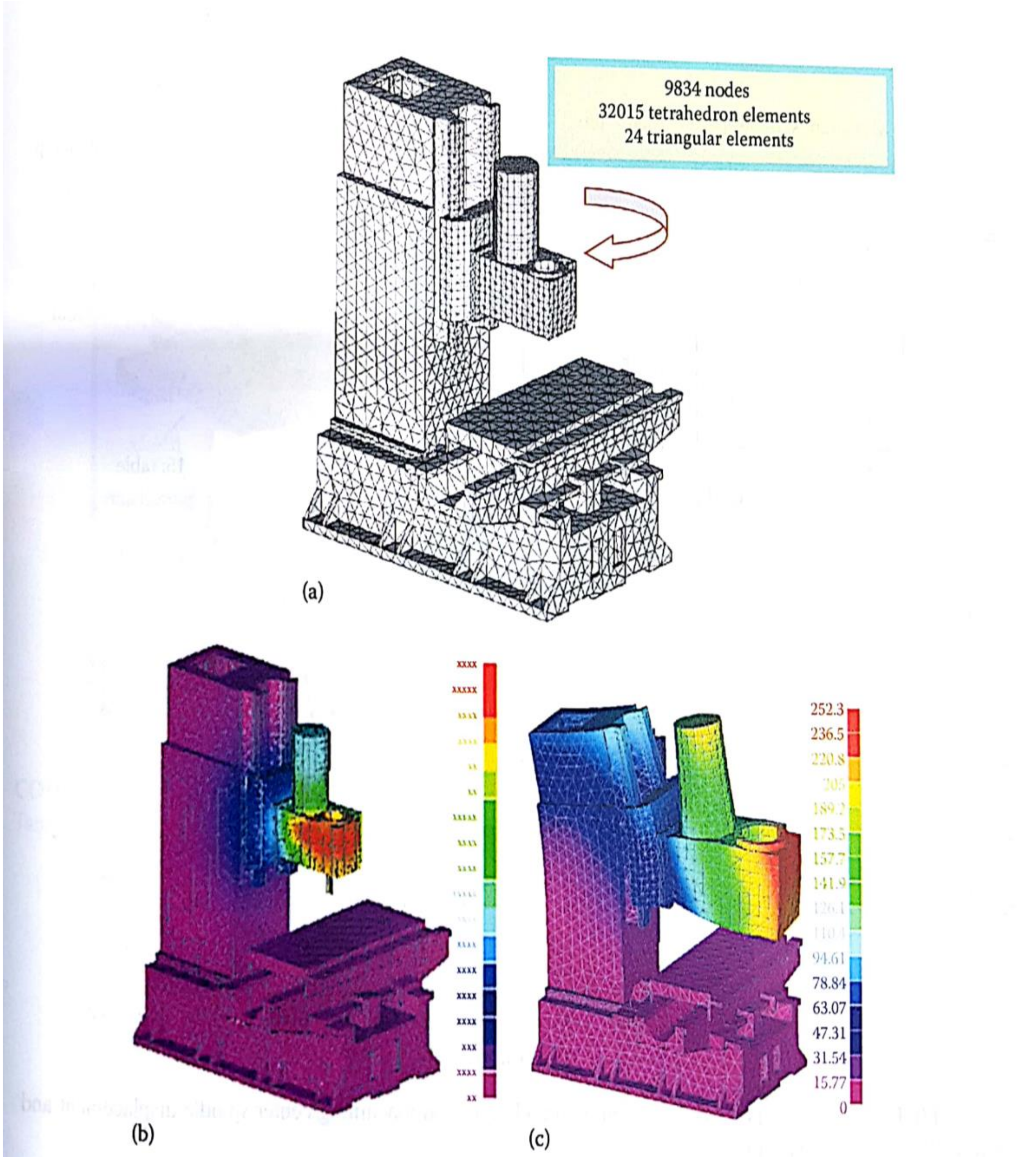
to unity as possible, and smaller elements are used over the parts of higher stress gradient.

- Proper support constraints are imposed with special attention paid to nodes on symmetry axes.

Large scale *commercial* software packages (such as ANSYS) often provide facilities for generating the mesh, and the graphical display of input and output, which greatly facilitate the verification of both input data and interpretation of the results.

Thus, all students/engineers are should be familiar with using the Finite Element Analysis for most engineering applications.

The following figure shows a typical example of Finite Element Analysis for a Machine Tool.



FEM analysis of a Machine Tool:(a) Mesh Generation (b)Temperature Distribution (c)Thermal deformation

(from S. Mekid, Introduction to precision machine design and error assessment, CRC Press)