3

DESIGN OF MACHINE TOOL STRUCTURE AND ANALYSIS

3.1 Introduction

Beds, bases, columns and box type housings are called “structures” in machine tools. In machine tools, 70-90% of the total weight of the machine is due to the weight of the structure [3]. In this chapter classification and functions of machine tool structure is described. Researchers [7, 9] have worked with different types of materials like cast iron, mild steel, granite and epoxy concrete for machine tool structure for different applications. Profile of the machine tool and selection of different stiffeners/ribs are suggested by researchers. Quality of the job produced on these machine tools depends directly on the quality and performance of machine tools. To develop good products, design engineers need to study how their designs will behave in real-world conditions.

The limitations of physical model techniques have led to the development of mathematical models representing a variety of mechanical structures. As in this approach, whole structure is divided into finite elements, it is known as ‘Finite Element Analysis’. The FEA is a very useful tool in engineering today and same has proved to be an important technique in machine tool structural analysis. Thus, Computer is an invaluable tool for a designer in his task for evaluating alternative designs to arrive at the optimum design and also predicting the static, dynamic and thermal behavior of the machine before arriving at the final design.

3.2 Functions of Machine Tool Structure and Their Requirements

Machine tool parts, such as beds, bases, columns, box-type housings, over arms, carriages, table etc. are known as structures. Basic functions of machine tool structure are as follows:

a) To provide rigid support on which various subassemblies can be mounted i.e. beds, bases.
b) To provide housings for individual units or their assemblies like s gear box, spindle head.
c) To support and move the work piece and tool relatively, i.e. table, carriage, tail stock etc.

Machine tool structures must satisfy the following requirements:

a) All important mating surface of the structures should be machined with a high degree of accuracy to provide the desired geometrical accuracy;
b) The initial geometrical accuracy of the structures should be maintained during the whole service life of the machine tool; and
c) The shapes and sizes of the structures should not only provide safe operation and maintenance of the machine tool but also ensure that working stresses and deformations do not exceed specific limits; it should be noted that the stresses and deformations are due to mechanical as well as thermal loading.
d) Efficient thermal control on machine element such as spindle, ball screw and bearings for better part accuracy.
e) Faster tool change system.
f) Very high rapid traverse rates of round 40-60 m/min for faster tool positioning and very high cutting feed rates for increased metal removal rates.

The design features that provide for ease of manufacture, maintenance, etc. are peculiar to each structure and will, therefore, be discussed separately for different structures. However, there are two common features, which are fundamental to the satisfactory fulfillment of above requirements for all structures. These are:

1. Proper selection of material.
2. High static and dynamic stiffness.[23]

3.3 Classification of Machine Tool Structure

Classification of machine tool structures which can be subdivided by various characteristics into the following groups:

a) By purpose into:

1. Beds, frameworks, carrying bodies.
2. Bases, bedplates etc.
3. Housing, boxes, columns, pillar, brackets.
4. Castings and covers.

b) By the method of manufacture into:

1. Cast.
2. Welded.
3. Combined cast and welded.

c) By functions they perform:

1. Beds and bases, upon which the various subassemblies are mounted.
2. Box type housings in which individual units are assembled.
3. Parts those serve for supporting and moving work piece and tool i.e. table, carriage etc.

3.4 Materials of Machine Tool Structure

The structure of a machine tool forms the vital link between the cutting tool and work piece on a metal cutting machine. The machine tool's metal removal rate, accuracy, overall cost, method of production and lead times, depend upon the type of structural material and its properties.

The commonly used materials for machine tool structures are cast iron and steel. While in some applications Granite and 'Epoxy Concrete', newly developed material, is also introduced. Cast iron structures were almost exclusively used in machine tools till a decade or so ago, but lately welded steel structures are finding wider application due to advances in welding technology. The choice of whether the structure should be made from cast iron or steel depends upon a number of factors, which are discussed as follows:

3.4.1 Material Properties

Important material properties of relevance are as under:

- **Modulus of elasticity:** For high stiffness it is necessary to choose materials with a high value of E. For instance, the high strength nodular graphite cast iron has doubled the modulus of elasticity than the normal cast iron, apart from its high internal damping. All steels have practically the same E and therefore mostly the non-expensive good commercial quality steel is used for machine tool structures.

- **Specific stiffness:** Material should have high specific stiffness.

- **Damping:** Cast iron has higher inherent damping properties, damping in steel structures occurs mainly in welds, if welded joints are properly designed, the damping of steel structure may approach that of cast iron.

- **Long-term dimensional stability:** The machine tool structural material must also have a good long-term dimensional stability. Locked in stress levels should be reduced to as close to zero as possible to achieve this.

- **Coolant resistance:** The material should be unaffected by coolant.

- **Wear rate and frictional properties:** Material should have low wear rate and low coefficient of friction.

- **Thermal expansion coefficient:** The material used should have a reasonably low coefficient of expansion. If several composite materials are used, each should have the same coefficient of expansion to avoid thermal bending/distortion. [7]

3.4.2 Different Materials Used for machine tool structure

As already stated, commonly used materials for machine tool structure are cast iron and steel. While in recent times, granite and epoxy concrete are also developed and used for structures. These materials are discussed here:

1. **Cast Iron:** From early times cast iron has been the most commonly used material for machine tool structures. It may be cast into complex and intricate shapes. It is easily

machined and may be hand-scraped and lapped to a high degree of accuracy. It has fairly good damping properties and also has reasonably good antifriction properties helped by the graphite contained in it. It can be given very good long-term dimensional stability by giving it a special long cycle stress relief annealing treatment. Cast iron should be preferred for complex structures subjected to normal loading, when these structures are to be made large in numbers.

It does, however, have several disadvantages. One major disadvantage is the time and cost taken to produce a finished casting. Again care has to be taken at design stage to ensure no abrupt changes in section thickness. Most manufacturing stages involve the moving of the component either in or outside the factory. Typical physical properties are given in the table.

2. **Mild steel weldments:** since 1950's mild steel weldments have been used more and more as a machine tool structural material. They have a high stiffness and the strength is also high. Values of properties of steel are listed in table. It has lower weight compare to cast iron. If necessary, in mild steel structures thin wall sections can be used. While with cast iron the wall thickness is limited by the accuracy of casting. Steel should be preferred for simple, heavily loaded structures, which are to be manufactured in small numbers; this is due to the fact that in lightly loaded structures the higher mechanical properties of steel cannot be fully exploited.

   This material too has some disadvantages. The material damping is low and mild steel weldments have a marked tendency to 'ring'. Friction points are sometimes built-in friction is high and cast iron or plastic insets have to be used to reduce friction to avoid 'pick-up'. Again for this material, manufacturing times are long. This material will rust, too. Long-term dimensional stability has not been verified to the same degree as cast iron. Finally, combined welded and cast structures are becoming popular, now days. They are generally used where a steel structure is economically suitable but is difficult to manufacture owing to the complexity of some portions; these complex portions are separately cast and welded to the main structure.

3. **Granite:** granite is used for surface tables and measuring machine structures. Its internal damping is better than that of cast iron. Its wear properties are good. It is reputed to be very stable dimensionally.
   Granite has a number of disadvantages. It is becoming more and more scarce. It takes a long time to cut it out to size, grind and lap it to shape. There are many types of granite, but most absorb water and the surrounding air humidity affects its dimensional stability and thus geometrical accuracy.

4. **Epoxy concrete:** it is a new material specifically developed over the past two decades for high precision machine tool structure. It is the mixture of binding agent reaction resin and the hardener together with carefully selected and mixed aggregates. It is completely new technology as compared with those of the materials mentioned above. Epoxy concrete offers great design freedom, similar to cast iron. It has outstanding damping properties – better than traditional concrete. It costs approximately the same as steel reinforced concrete or even less. Epoxy concrete does not expand and contract with change in humidity, as does ordinary concrete. Again various material properties can be controlled in epoxy concrete by the type of mixture chosen. Epoxy concrete has a very high long-term dimensional stability. Values of some properties and comparison of above discussed material are given in Table 3.1 and Table 3.2.

Table 3.1 Some properties of some structural material – app. Average value [7]

Material	Modulus of Elasticity N/mm^2	Specific Gravity	Specific Stiffness N/mm^2	Coefficient of thermal Expansion $^0C^{-1}$	Thermal conductivity $Wm^{-1}k^{-1}$	Tensile strength N/mm^2
Cast Iron	117000	7.21	16000	12×10^{-6}	75	230
Mild steel	207000	7.93	26000	12×10^{-6}	80	460
Granite	39000	2.66	15000	8×10^{-6}	0.8	14.7
Ep. Con.	33000	2.5	14000	12×10^{-6}	0.5	25

Table 3.2 Comparison of structure materials [7]

Material	Merits	Demerits
C.I.	-Possible to cast it in complex and intricate shapes. -Easily machined, hand-scraped and lapped to high degree of accuracy. -Fairly good damping properties -Good anti-friction properties	-Comparatively lower strength -Time and cost taken to produce a finished casting -Technological constraints to produce cast structures i.e. minimum wall thickness -High shrinkage rates during curing -Need anti-corrosion treatment
Steel weldments	-Very high strength and stiffness -Less technological constraints in manufacturing -Structure of steel weldments have lower weights, less massive	-Poor damping compare to cast iron -Poor frictional properties -Long term dimension stability has not been verified to same degree of cast iron -Need stress relieving -Need for anti-corrosion treatment
Granite	-Damping properties are even better than cast iron -Good wear properties -Reputed to be very stable dimensionally	-More lead time in cut it to size, grind and lap it to shape -Drilling, machining is difficult as it is very hard
Epoxy Concrete	-Used for precision machine tool structures -It offers great design freedom -Outstanding damping properties,	

3.4.3 Design Criteria for Machine Tool Structure

Consider a simple machine tool bed with two side walls, which may be represented as a simply supported beam loaded by concentrated force P acting at its center, as shown in Fig. 3.1 below:

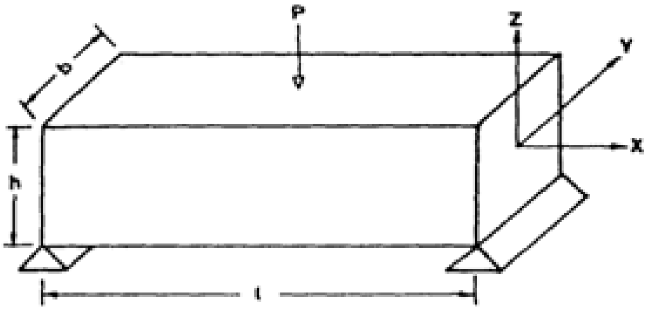


Fig. 3.1 simply supported beams centrally loaded

Maximum normal bending stress acting on the beam is given by the expression:

$$\sigma_{max} = \frac{M_{max} \cdot z_{max}}{I_y} \tag{3.1}$$

Where, $M_{max} = \frac{Pl}{4}$ = maximum bending moment

$z_{max} = \frac{h}{2}$ = distance of outermost fibre from the nuetral axis

$I_y = \frac{bh^3}{12}$ = moment of inertia of the beam section about the neutral axis

Upon substituting these values in Eq. 3.1, we get,

$$\sigma_{max} = \frac{\frac{Pl}{4} \cdot \frac{h}{2}}{\frac{bh^3}{12}} = \frac{3}{2} \frac{Pl}{bh^2}$$

If the permissible normal stress under tension for the beam material be denoted by [σ], then

$$[\sigma] = \frac{3}{2} \frac{Pl}{bh^2}$$

or $$V_\sigma = b \cdot h \cdot l = \frac{3}{2} \frac{P}{[\sigma]} \cdot \left(\frac{l^2}{h}\right)$$

where, V_σ is the minimum volume of metal required for sufficient strength of beam. For maximum deflection of simply supported beam, we know,

$$\delta_{max} = \frac{Pl^3}{48EI_y}$$

where E= Modulus of elasticity of the beam material, If the deflection of the beam is not to exceed a permissible value, denoted by [σ], then

$$[\delta] = \frac{Pl^3}{48EI_y} = \frac{Pl^3}{48E \cdot \frac{bh^3}{12}}$$

$$V_\delta = b \cdot h \cdot l = \frac{P}{4E[\delta]} \cdot \left(\frac{l^2}{h}\right)^2$$

where, V_δ is the minimum volume of metal required to ensure that deflection of the beam under load does not exceed the specified value.

The condition of optimum design is:

$$V_\sigma = V_\delta$$

$$\frac{3}{2}\frac{P}{[\sigma]} \cdot \left(\frac{l^2}{b}\right) = \frac{P}{4E[\delta]} \cdot \left(\frac{l^2}{h}\right)^2$$

$$\frac{l^2}{h} = \frac{6E[\delta]}{[\sigma]} \tag{3.2}$$

Equation 3.2 indicates that for every structure, there exists an optimum ratio, l^2/h depending upon:

a) Operation constraints, expressed in this case through $[\delta]$,
b) The material of the structure, expressed in this case through E and $[\sigma]$

For example, consider two beams of mild steel and cast iron, with mechanical properties as under:
For Mild steel……E = 2 x 10^5 N/mm^2, [σ] = 140 N/mm^2, [δ] = 0.002 mm
For Cast Iron…… E = 1.2 x 10^5 N/ mm^2, [σ] = 30N/mm^2, [δ] = 0.002 mm

For the steel beam,

$$\left(\frac{l^2}{h}\right)_{opt} = \frac{6 \times 2 \times 10^5 \times 0.002}{140} = 17.14$$

For the cast iron beam,

$$\left(\frac{l^2}{h}\right)_{opt} = \frac{6 \times 1.2 \times 10^5 \times 0.002}{30} = 48$$

The volume of the two beams with optimum l/b values will be in the ratio

$$\frac{V_{CI}}{V_{MS}} = \frac{\frac{3}{2}\cdot\frac{P}{3}\cdot 48}{\frac{3}{2}\cdot\frac{P}{14}\cdot 17.14} = \frac{\frac{P}{4\times 1.2\times 10^4\times 0.002}\times (48)^2}{\frac{P}{4\times 2\times 10^4\times 0.002}\times (17.14)^2} = 13.07$$

i.e. If the failure of the beams is determined by the normal stresses under tensile loading, the volume of the steel beam required to withstand the same load is 13.07 times less than that of the cast iron beam.

The variation of V_σ and V_δ for mild steel and cast iron beams with change of l^2/h is shown in Fig. 3.2. For identical beam length, the height of the steel section must be 48/17.14 = 2.80 times greater. Since the volume of the steel beam is 13.07 times less and height 2.80 times greater than that of the cast iron beam, the thickness of the mild steel beam will be 36.5 times less.

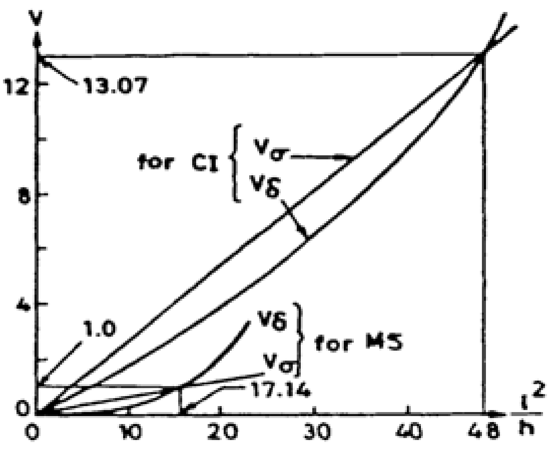


Fig. 3.2 Variation of V_σ and V_δ with L^2/b ratio for cast iron and mild steel

It is evident from Fig. 3.2 that for l^2/h values less than the optimum (Corresponding to the point of intersection of V_σ and V_δ curves), the structure should be designed for consideration of strength, while for l^2/h values exceeding the optimum value, the design should be guided by stiffness considerations. In practice, the l^2/h ratio for a majority of machine tools lies to the right of the point of intersection i.e. l^2/h is greater than the optimum value. Consequently, the stiffness and not the load-carrying capacity of a structure is the decisive factor, which determines its dimensions in most of the machine tools. So that the steel structure is lighter, deeper and thinner than a cast iron structure of equivalent strength is obvious. However, since structures are mostly designed from stiffness considerations, the actual economy of metal consumption by using steel instead of cast iron may be much less than 13.07 times, because the steel structure must be provided with stiffening ribs. This not only increases the weight of the steel structure but also adds to the labor cost [23].

3.4.4 Manufacturing Considerations, Problems and Constraints

Another important factor for deciding the choice of material concerns the problems of manufacturing that are associated with the use of steel or cast iron structures:

- Wall thickness: For a given weight of the structure, high strength and stiffness can be achieved by using large overall dimensions and small wall thickness. Thus walls of minimum possible thickness should be employed. Generally, reduction of wall thickness in cast iron structures is restricted by process capability and depends upon the size of the casting in case of cast iron. These values are given in Table 3.3.

Size factor N is determined from the relationship,

$$N = \frac{2L + B + H}{4}$$

where, L, B, and H are length, breadth and height of the structure respectively in meters.

The wall thickness may also be determined from the following relationship:

$$\delta = 10\sqrt{\frac{2L + B + H}{3}}$$

Internal walls and ribs cool at a slower rate than external walls, and therefore, for them a minimum thickness equal to 0.8 that of external walls is permitted.

Table 3.3 Recommended min. wall thickness for C.I. structures

Size Factor N, (m)	Thickness of external walls, (mm)	Thickness of internal walls and ribs, (mm)
0.4	6	5
0.75	8	7
1.0	10	8
1.5	12	10
1.8	14	12
2.0	16	14
2.5	18	16
3.0	20	16
3.5	22	18
4.5	25	20

Welded structures made of steel can have much thinner walls as compared to cast structures as the technological constraints are much less. Steel structures in which the wall thickness is less than that of the cast structure by up to 50% are known as thick-walled structures. They are made of 10-12 mm thick plates and are easy to manufacture, but they are not particularly effective from point of view of economy of metal.

- Walls of different thickness can be welded more easily than cast structure, transition from one thickness to another (if $t_1/t_2 < 1.5$) is accomplished by means of a fillet radius, of proper value.

- Machining allowance for cast structures are generally larger than for weld steel structures, this is essential to remove the hardened skin of casting and also to account for casting defects, such as inclusions, scales, drops, etc., that result due to the falling of sand into the mould cavity.

- A welded structure can, if required, be easily repaired and improved. Any corrections in a cast structure are much more difficult. This property of steel structures is particularly useful in preparing a prototype.

3.4.5 Economic considerations

The final selection of material for structure will in most cases rest upon which of them provides for a lower cost of the structure. Correct selection can be made only on the basis of a comprehensive analysis of various factors, some of which are listed below:

- Economy of metal: Here it is important to remember that although the weight of the finished steel structure may be low, the actual metal consumption may be high; this is due to the fact that whereas holes in castings are obtained with the help of cores, those in welded structures have to be machined. This results not only in scrap but also in additional labor cost.
- Cost of pattern and welding fixtures.
- Cost of machining.

3.5 Basic Design Procedure for Machine Tool Structure

In order to design a particular machine tool structure, it is first essential to draw up its design diagram. Machine tool structures have, as a rule, highly complicated profiles. In designing the structure of a machine tool a number of requirements must be respected. These are the possibility of placing the whole range of work pieces into the machine, the necessary ranges of travel, sufficient room for chips, room for all mechanisms and for hydraulic, electric and other equipment, the possibility of easy assembly of the structure and of its parts and of subsequent dismantling, easy access for the operator wherever necessary, and the limitation of thermal distortions of the structure. Further, it is necessary to design all parts of the frame with such shapes and of such dimensions as to ensure suitable rigidity of the frame.

Forces occur during the machining operation giving rise to deformations, which disturb the accuracy of machining. Some of the forces do not depend on the intensity of the cutting process as for instance the weight forces of the moving parts of the machine. The influence on accuracy of the others, such as cutting forces, is related to the rate of machining. The relation between forces and deformations and their combined effect on the machining operation leads to requirements on the stiffness of the individual parts of the structure and of the structure as a whole.

According to various kinds of forces, which occur during the machining operation, various specifications of requirements on stiffness may be stated. These forces will be classified into four groups corresponding to four different criteria [23].

1. **Deformations caused by weight forces**

   During the movement of the individual parts of the structure the distribution of their weights and of the weight of the work piece varies. Consequently the deformations of the frame vary. The criterion is that any deviations arising do not disturb the prescribed geometric accuracy of the machine tool.

2. **Deformations caused by cutting forces**

   During the operation the cutting force varies and its point of application moves. In consequence, the deformations of the frame will vary causing deviations of the form of the machined surfaces. This effect may be limited by decreasing the cutting conditions and consequently the output of the operation. Cutting force depends upon the work piece material; machining parameters, wear of cutting tool etc. For a designer a knowledge about the nature and direction of the force and the point where it acts on the structure is often more important than a very precise knowledge of its magnitude.

3. **Forced vibrations**

   In the machine tool disturbing periodic forces occur. They are caused mainly by the unbalance of rotating parts and by errors of accuracy in some driving elements. They excite forced vibrations, which result in the waviness of machined surfaces. The criterion is to limit forced vibrations so as to achieve to the required surface quality.

4. **Self-excited vibrations**

   Under certain conditions, generally connected with the increase of the machining rate self-excited vibrations occur and these are energized by the cutting process. They cause unacceptable waviness of the machined surface and endanger the strength and life of the parts of the machine and of the tools. The criterion is that in the required range of operations and of cutting conditions self-excited vibrations shall not occur and the cutting process must be stable.

The individual criteria are almost independent of one another. Nevertheless, experience shows that criterion 4 prevails and if it is satisfied then criterion 2 and often also criterion 1 and 3 are more than fulfilled. The problem of stability of the frame against self-excited vibrations energized by the cutting process is not only the most important one but also the most difficult. All four criteria determine requirements on some resulting stiffness, static or dynamic, between the tool and the work piece. By analyzing this resulting stiffness, requirements on the individual parts of the frame may be derived.

Members of cutting machine tools are designed mainly on the basis of stiffness and stability. And thus deflection and deformation of all components along the line of action of forces should be a minimum. As already stated, machine tool structure can be broadly divided into three groups. In drawing the design diagrams for structures of each group the following guidelines may be useful:

Group 1: Structures like beds and columns with fully or partially closed thin box profiles or consisting of two walls connected by parallel and diagonal stiffeners may be analyzed as statically indeterminate thin-wall bars.

Group 2: Closed box type structures like housing of speed and feed boxes are designed for forces perpendicular to the walls, as the latter have sufficient stiffness in their own plane.

Group 3: Supporting structures like tables knees, etc. which are generally loaded normal to their base plane analyzed as plates.

Under general conditions of compound loading, most of the machines tools structures are analyzed as elements subjects to bending in two perpendicular planes and torsion. It was pointed out earlier also that the basic design requirement of machine tools is their stiffness. The common design strategy for machine tool structures can therefore be summed up as:

1. Design for bending stiffness,
2. Designing for torsional stiffness, and

3. Checking dimensions for sufficient strength due to bending and torsion.

The design of a structural member is determined by its use. There are three principal cases:

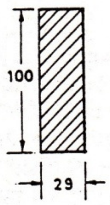
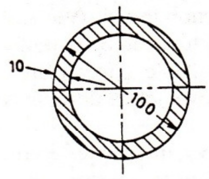
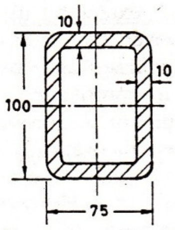
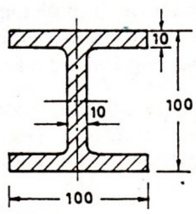
1. The structural member is to be designed with respect to stiffness for which shape is the criterion. From the viewpoint of strength the member may then be over dimensioned. Several members in machine tools fall in this group i.e. bed, column etc. for those parts which are designed on the basis of stiffness, the dynamic behavior is of special importance i.e. chatter in cutting machine tools.
2. The structural member is to be designed with respect to strength. Deformations must remain within allowable limits.
3. The structural member is to be designed with respect to both stiffness and strength.

3.6 Profiles of Machine Tool Structures

During the operation of the machine tool, a majority of its structures are subjected to compound loading and their resultant deformation consists of torsion, bending and tension or compression. Under simple tensile or compressive loading, the strength and stiffness of an element depend only upon the area of cross-section. It is known from classical mechanics of elastic bodies that in the case of bending and torsion it is possible to decrease the requirement on material by a suitable choice of the form of the cross-section, by increasing the second moment of area at constant area of the cross-section i.e. at constant weight of the element. Another typical feature of machine tools is the rather small value of the length to width ratio of their parts. Shape and strength are interrelated and when this fact is disregarded, damage often occurs.

However, the deformation and stresses in elements subjected to torsion and bending depend additionally, upon the shape of the cross-section. A certain volume of metal can be distributed in different ways to give different values of inertia and sectional modulus. The shape that provides the maximum moment of inertia and sectional modulus will be considered best as it will ensure minimum values of stresses and deformation. The stiffness of four different commonly used sections of structures is compared with equal c/s area in Table 3.4**.**

Table 3.4 Comparison of stiffness of different sections having equal c/s area [23]

Section	*Area* mm^2	*Weight* kgf/m	*Relative value of permissible* — *Bending moment* kgf.cm — *Stress*	*Deflection*	*Torque* kgf.cm — *Stress*	*Angle of twist*
	29.0	22	1	1	1	1
	28.3	22	1.12	1.15	43	8.8
	29.5	22	1.4	1.6	38.5	31.4
	29.5	22	1.8	1.8	4.5	1.9

It is evident from the Table 3.4 that the box-type section has the highest torsional stiffness and in the overall assessment seems best suited both in terms of strength and stiffness. The additional advantage that goes in its favor is the ease of proper mating with other surfaces. Thus, in the case of bending and especially for torsion the optimum from the point of view of stiffness is that of a closed box cross-section, the bending stiffness of which is as advantageous as that of the I-section and its torsional stiffness approaches that of a circular section. All considerations combined point towards the overwhelming superiority of the box-type profile over others for machine tool structure.

In most of the cases the machine tool bed and other structures cannot be in the form of a closed-box profile. There must be apertures for bearings, openings for free flow of chips and other purposes. Thus the actual profiles of machine tool structures differ from a closed-box profile. The apertures and openings in the structure have an adverse effect upon its strength and stiffness.

3.7 Factors Affecting Stiffness of Machine Tool Structure and Methods to Improve It

In order to support the work piece and position it correctly with respect to the cutter under the influence of cutting forces it is necessary for the structure to have high static and dynamic stiffness values. Stiffness of the structure is related to its shape of cross-section, cuts and apertures in walls of structures cover plates, arrangement of ribs internally as well as externally etc.

3.7.1 Effect of aperture on torsional stiffness

In the most of the cases machine tools structures cannot be made of complete closed box type profile. There must be apertures, openings for free flow of chips and other purposes. Thus the actual machine tool profile is quite different from closed box profile. The apertures and openings in the structure have an adverse effect upon its strength and stiffness. The effect of aperture on the torsional stiffness of a box-type structure is shown in Fig. 3.3. It can be seen that a circular hole of diameter d affects a length of approximately twice the diameter, i.e. affected length $l_1 = 2d.$ An elongated aperture affects the stiffness even more. The reduction in the static and dynamic stiffness of a structure can be partially compensated by using suitable cover plates. Results using cover plates are compared in Table 3.5.

It is evident from Table 3.5 that the reduction in the bending stiffness due to apertures can be compensated to a large extent by using suitable cover plates. However, the effect on the torsional stiffness is significant and cover plates do not help much in improving it. For symmetrically placed apertures, the effects can be taken into account by multiplying the torsional stiffness with a reduction coefficient.

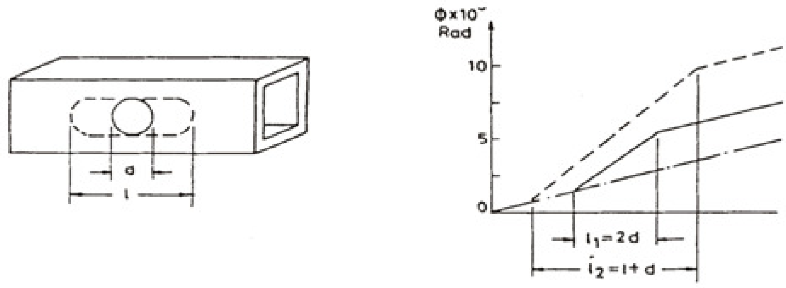
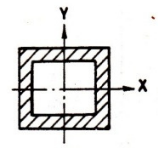
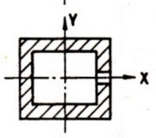
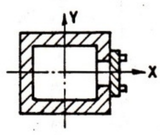
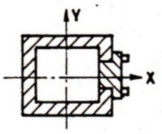


Fig. 3.3 Variation of angle of twist as a function of aperture shape and size [23]

Table 3.5 Effect of aperture and cover plate on stiffness of box type structures [23]

	Relative stiffness about			*Relative natural frequency of vibrations about*			*Relative damping of vibrations about*		
	X-X	*Y-Y*	*Z-Z*	*X-X*	*Y-Y*	*Z-Z*	*X-X*	*Y-Y*	*Z-Z*
	100	100	100	100	100	100	100	100	100
	85	85	28	90	87	68	75	89	95
	89	89	35	95	91	90	112	95	165
	91	91	41	97	92	92	112	95	185

3.7.2 Effect of stiffeners (ribs) on stiffness of structure

In a machine tool higher production rate together with good machining accuracy and surface finish can be achieved by aiming at a structural design that ensures a large stiffness to weight ratio. Accordingly, lightweight structures possessing large stiffness can be designed by employing box sections of large overall dimensions and very thin walls. Only limitation being there load carrying capacity in view of increased danger to warping and buckling. This problem, however, alleviated to a large extent by the use of suitable ribbing designs. The stiffness of structures can be improved by using ribs and stiffeners. However, it should be noted that the effect of the ribs and stiffeners depends to a large extent upon how they are arranged. Sometimes, an increase in rigidity due to partitions is negligible and does not compensate for the additional consumption of material and labor required for fabrication. Stiffness/weight ratio is an important factor in deciding the ribbing arrangements.

3.7.2.1 Effect of End Cover plate on stiffness of structure

Provision of an end cover plate reduces considerably, the deflections in y and z directions of a thin walled column in torsion Fig. 3.4, while in case of bending no significant improvement is observed. Thickness of end cover plate is varied and behavior of structure is observed and after analysis optimum thickness of end cover plate should be taken. Table 3.6 shows the behavior of column with varying thickness of end cover plate. It can be observed that thickness of end cover plate equal to the wall thickness is giving reasonably good result compare to thicker end cover plates [4].

Table 3.6 Effect of end cover thickness on torsional and bending stiffness

Case	Computed deflection (average) at load point, μ_m			
	Torsion		Bending	
	Y - comp	X - comp	Y - comp	X - comp
No cover	101.74	122.17	0.177	57.11
End cover				
t/2 thick	6.08	14.35	0.116	57.12
t thick	6.38	13.78	0.115	57.12
1.5 t	6.82	13.60	0.115	57.10
2.0 t	6.92	13.50	0.116	57.05
2.5 t	6.97	13.45	0.116	57.02
3.0 t	7.01	13.41	0.117	56.93
3.5 t	7.04	13.38	0.117	56.87
4.0 t	7.06	13.38	0.117	57.80

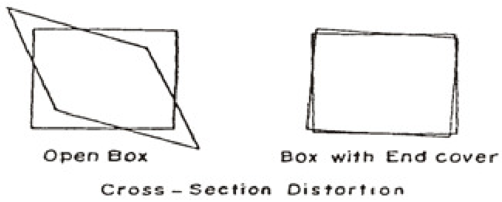


Fig. 3.4 Comparison of cross section deformation with and without end covers plate

3.7.2.2 Effect of ribs arrangement in closed box structure

For the purpose of direct comparison, different ribbing arrangement in case of closed box-structure is shown in Table 3.7. It is evident from Table 3.7 that only stiffeners used as shown in arrangements 5 and 6 provide significant improvement in the bending and torsional stiffness of box-type structures [23]. But the most effective arrangement of ribs is ‘diamond shaped ribs’, which is not shown in above table. The results of above table can be realized with graphical plots, Fig. 3.5 (a) and (b) [4].

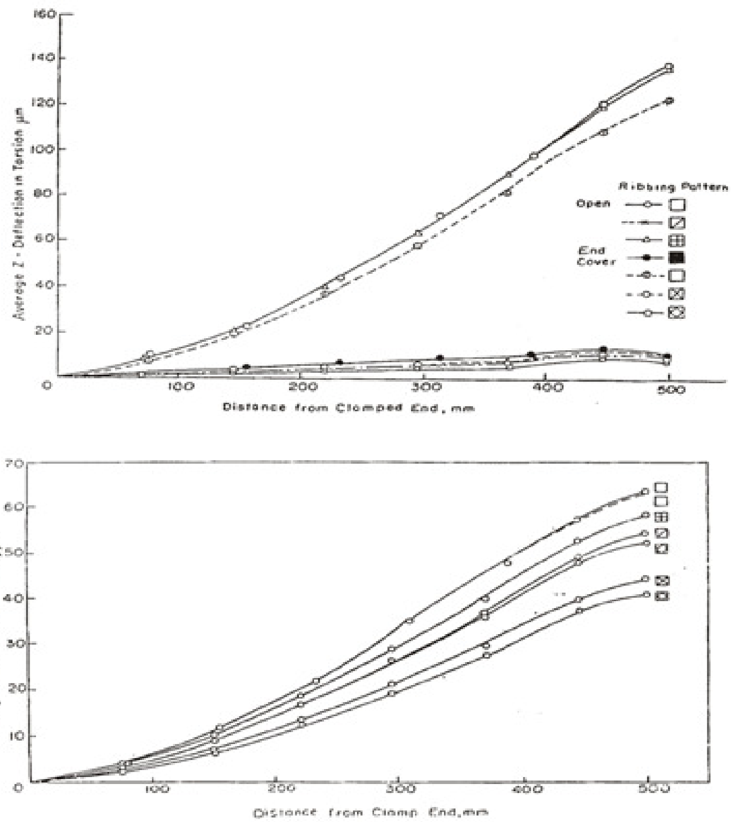
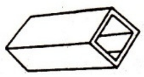
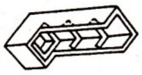
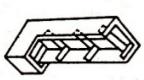
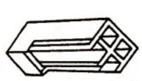
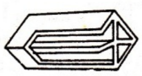
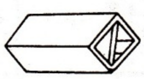


Fig. 3.5 (a) Stiffness variation with different ribbing arrangement with and without end covers (b) Stiffness variation as a function of different ribbing arrangement

Table 3.7 Effect of stiffener on bending and torsional stiffness of box-type structures

Stiffener arrangement		*Relative stiffness under*		*Relative weight*	*Relative stiffness per unit weight under*	
		Bending	*Torsion*		*Bending*	*Torsion*
1		1.0	1.0	1.0	1.0	1.0
2		1.10	1.63	1.1	1.0	1.48
3		1.08	2.04	1.14	0.95	1.79
4		1.17	2.16	1.38	0.85	1.56
5		1.78	3.69	1.49	1.20	3.07
6		1.55	2.94	1.26	1.23	2.39

3.7.2.3 Effect of Vertical Stiffeners

Fig. 3.6 shows the columns having internal and external vertical stiffeners, which have been analyzed. Each side of stiffener cross section is kept equal to wall-thickness of the column. In both cases, vertical internal and external stiffeners, depth of the stiffeners, denoted by ‘a’, is varied in order to analyze effect on performance.

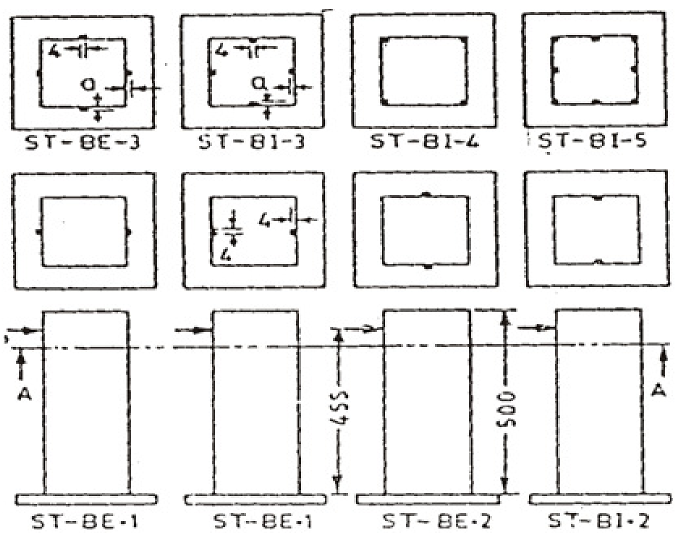


Fig. 3.6 Columns with vertical internal and external stiffeners

Table 3.8 Effect of external and internal vertical stiffeners

Structure	a_{mm}		Bending		Torsion	
		Per cent weight	Per cent rigidity	Stiff/Wt. ratio	Per cent rigidity	Stiff/Wt. ratio
ST -BI.1	-----	101.24	138.0	1.363	182.0	1.798
ST -BI.2	-----	101.24	132.0	1.304	178.0	1.761
ST -BI.3.1	4	102.60	155.1	1.512	219.8	2.142
ST -BI.3.2	6	104.24	170.1	1.632	235.3	2.257
ST -BI.3.3	8	105.65	183.6	1.738	249.3	2.360
ST -BI.4	-----	102.60	141.1	1.375	176.2	1.719
ST -BI.5	-----	105.65	179.9	1.703	254.7	2.411
ST -BE.1	-----	101.24	138.3	1.366	179.9	1.777
ST -BE.2	-----	101.24	133.0	1.314	174.0	1.719
ST -BE.3.1	4	102.60	155.6	1.520	212.7	2.073
ST -BE.3.2	6	104.24	171.4	1.644	227.4	2.182
ST -BE.3.3	8	105.65	185.0	1.751	234.0	2.271

It can be observed from Table 3.8 that for similar arrangements the external vertical stiffeners are more effective in bending loads while internal vertical stiffeners are more effective in torsional loads though the difference is very small.

3.7.2.4 Effect of Horizontal stiffeners

Fig. 3.8 shows the positions of internal and external horizontal stiffeners. Depth ‘a’ of stiffeners is varied in steps as shown in Table 3.9, which also shows results of combination of horizontal and vertical stiffeners, to analyze its effect on performance of structure. Large improvement in torsional rigidity by these stiffeners is due to their resistance to rotation of the column. But under bending loads these stiffeners are less effective. Columns having internal horizontal stiffeners are stiffer than those having external horizontal stiffeners. As earlier said, these stiffeners have very high torsional rigidity, while under bending loads they are less effective.

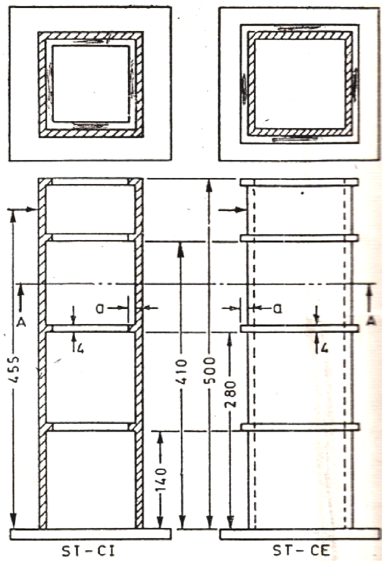


Fig. 3.7 Columns with horizontal stiffeners

Table 3.9 Effect of Combination of both horizontal and vertical stiffeners

Structure	a_{mm}		Bending		Torsion	
		Per cent weight	Per cent rigidity	Stiff/Wt. rigidity	Per cent ratio	Stiff/Wt. ratio
ST - CI.1	4	103.30	143.2	1.387	279.3	2.704
ST- CI.2	6	104.88	148.2	1.413	316.8	3.020
ST - CI.3	8	106.42	156.3	1.469	350.2	3.291
ST - CE.1	4	103.66	139.9	1.349	274.1	2.644
ST - CE.2	6	105.56	148.2	1.404	307.1	2.910
ST - CE.3	8	107.50	155.4	1.445	337.0	3.135
ST - DI.1	-----	106.01	175.8	1.659	336.2	3.172
ST - DI.2	-----	108.80	201.2	1.849	373.8	3.436
ST - DE.1	-----	106.40	177.3	1.667	322.4	3.030

Generally, combination of both horizontal and vertical stiffeners is provided in column to improve stiffness of the column in both bending and torsional loading. Columns ST-BI.3.1 and ST-BI.5 having vertical stiffeners, as shown in Fig. 3.6, are combined with the column ST-CI.1 having horizontal stiffeners to get the stiffener arrangements with ST-DI.1 and ST-DI.2 respectively, in Table 3.8, while in arrangement ST-DE.1, the stiffeners of columns ST-BE.3.1 and ST.CE.1 are combined. It can be seen from Table 3.8 that ST-DI.2 has the highest stiffness/weight ratio for both bending and torsional loading [5].

3.7.2.5 Effect of Fastening bolts and External Vertical stiffeners at bottom

The stiffness of structures can also be improved by providing a proper arrangement of fastening bolts. The effect of bolt arrangement and stiffening ribs on the bending and torsional stiffness of a vertical column is depicted in Fig. 3.8. It is evident from the Fig. 3.8 that by arranging the fastening bolts uniformly the stiffness can be improved by 10-20%. By additionally providing flange stiffeners, at bottom, the column stiffness can be increased by almost 50%. Rigidity of the machine tool as a whole depends upon the rigidity with which various units are clamped. It should be kept in mind those joints between various structural elements e.g. joints between the head stock and tail stock to a lathe with bed, the base plate of a drilling machine with its column etc. should be made as rigid as possible [23].

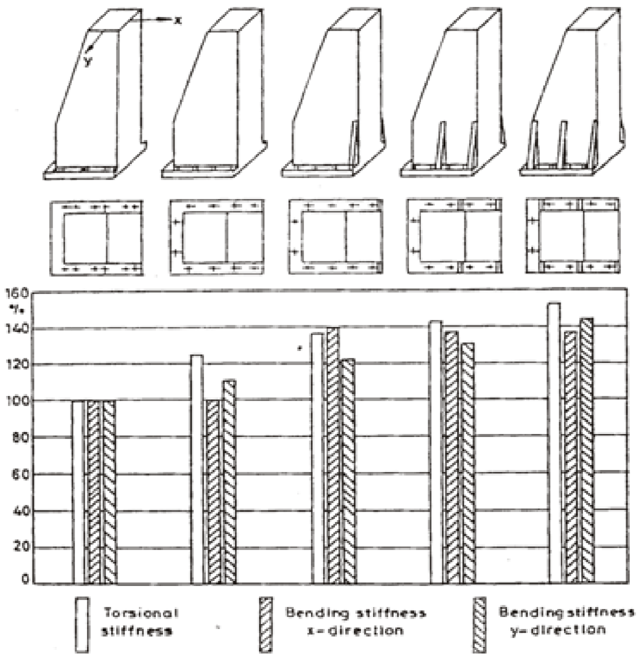


Fig. 3.8 Effect of bolt arrangement and external bottom stiffeners

Fig. 3.9 shows the columns having different arrangements of flange type stiffeners, which have been analyzed. These columns have higher stiffness because these types of stiffeners have the effect of reducing the effective length of the column thereby reducing the deflections. For the present case the effective length of column is reduced by 150 from total height of 500 mm. ST-A.M is the column, which is assumed to be clamped at a distance of 150 mm (height of the stiffeners) from the bottom. This column will have the highest possible stiffness that can be achieved by this type of stiffening arrangement.

Table 3.10 shows the effectiveness of different ribs positions in bending and torsional loadings. Base width of stiffeners is then varied, the parameter shown by 'a' in Fig. 3.9, keeping the thickness of the stiffeners equal to wall thickness of column in all cases.

Table 3.10 Effects of External Vertical Bottom Stiffeners

Structure	a_{mm}		Bending		Torsion	
		Per cent weight	Per cent rigidity	Stiff/Wt. ratio	Per cent rigidity	Stiff/Wt. ratio
ST - UR	-----	100.0	100.0	1.000	100.0	1.000
ST-A.M.	-----	-----	247.0	-----	223.8	-----
ST - A.1	60	103.8	124.8	1.203	146.0	1.408
ST - A.2	60	103.8	124.6	1.200	150.1	1.449
ST - A.3	60	108.6	132.6	1.221	153.9	1.416
ST - A.4.1	20	107.6	115.4	1.073	131.9	1.226
ST - A.4.2	40	111.4	185.9	1.669	171.1	1.536
ST - A.4.3	60	115.5	189.8	1.643	176.2	1.526
ST - A.4.4	80	119.4	193.1	1.617	177.2	1.484

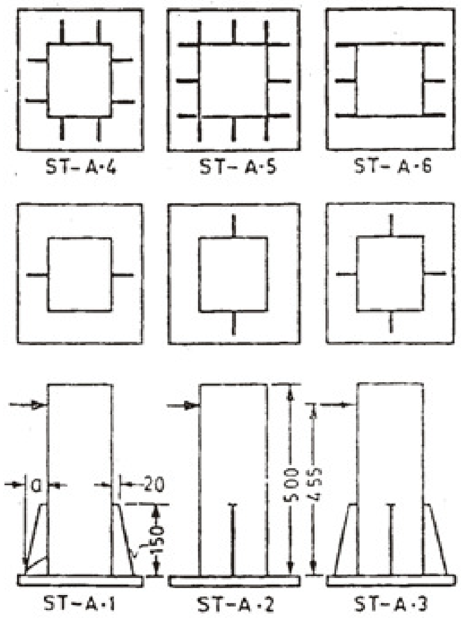


Fig. 3.9 External vertical bottom stiffeners

From the Table 3.10, it can be seen that for stiffening arrangement ST-A.4, a = 40, has highest stiffness in bending. This is because this stiffening arrangement reduces the distortion of column wall. Increasing the base width of the stiffeners increase the rigidity but increasing it beyond a certain limit is less effective as indicated by falling stiffness to weight ratio, shown in Fig. 3.10, below. The corner stiffeners of ST-A.5 becomes less effective under bending loads, but under torsional load all the stiffeners are effective in reducing the column rotation thereby increasing the rigidity of column[5].

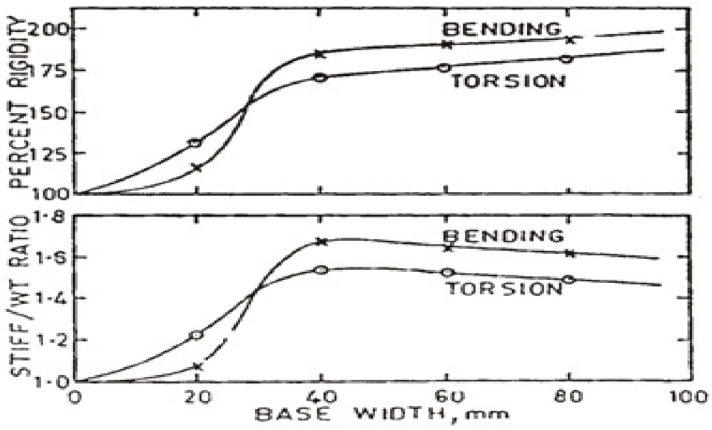


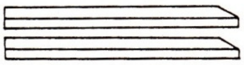
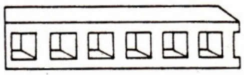
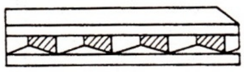
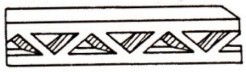
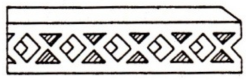
Fig. 3.10 Effect of stiffener base width on stiffness/weight ratio and rigidity

3.7.2.6 Improving Stiffness of Open Structures

The stiffness of open structures, such as lathe beds where two plates of structures, top and bottom, connected by ribs, also get affected by arrangement of ribs. The torsional rigidity of open structures has been compared under different stiffener arrangements and the results are shown in Table 3.11, below. The results of table indicate that only arrangements 4 and 5 are effective in terms of stiffness-to-weight ratio of the structure. Arrangements 4 consisting of two parallel shears, which are connected by diagonal ribs, is commonly used in machine tool beds.

Finally, stiffness to weight ratio is important factor in deciding the ribbing arrangements. Higher values of stiffness to weight ratio is desirable. Because ribs increase stiffness of structure but it also increase weight on other hand. So to arrive at final arrangement of ribs, one has to see value of stiffness to weight ratio. Again increasing no. of ribs means adding material and so adding cost, finally. Apart from economical reasons, a high stiffness to weight ratio will increase the values of natural frequencies of the structure, which may then become free of any resonant vibrations in the operating range thus improving dynamic conditions [23].

Table 3.11 Effect of stiffener arrangement on torsional stiffeners of open structures

Stiffener arrångement		*Relative torsional stiffness*	*Relative weight*	*Relative torsional stiffness per unit weight*
1		1.0	1.0	1.0
2		1.34	1.34	1.0
3		1.43	1.34	1.07
4		2,48	1.38	1.80
5		3.73	1.66	2.25

3.7.3 The significance of joints and their orientation upon the overall stiffness of structure

It is well established that one of the great obstacles to a complete understanding of the static and dynamic behavior of machine tool structure is the inability to take the effects of the joints fully into account. Contribution of various parts of structure in overall flexibility of machine tool, and actual machine tool behavior, as a whole, differs. This is because during analysis of individual structural elements, joint properties of machine tool are not specified; however, it is of much importance. Because resultant force will transfer from one part to another through contact area, which is nothing but a joint, either “fixed” or “sliding”.

In design of machine tool structure, designer is often faced with a decision as to on which plane the joint should be provided between two elements. Again joints are present as an essential part of the functional requirements in the operational movements, and also enable the manufacture and machining of the elements. In large machine tools, joints are also required to assist in the transportation of the finished machine.

Joints in machine tools may be of two basic forms, depending upon the relative movement, which takes place between the joint interfaces:

1. Joints, which connect structural parts without any intended motion, e.g., the joints between the headstock and bed of a lathe, these are called “fixed joints”
2. Joints, which connect parts, which are to have intended relative motion to one another, e.g. joints between the saddle and bed of a lathe, these are called “sliding joints”.

It is sometimes necessary for elements to be jointed together and possess both the qualities of fixed and sliding joints, as in the case of the joint between the tailstock and bed of the lathe.

As the joints form a link or a number of links in the chain of elements closing the flow of the cutting forces, they should possess a stiffness matching that of the other structural elements; even having other structural elements of a high stiffness would not help if there were only one flexible joint in the chain of elements, i.e. springs in series.

Following properties of joints require attention:

a) The static and dynamic stiffness of joint faces loaded in a plane normal to the joint surface;
b) The significance of joints on the overall deflection of the structure
c) The damping effect of joints

Research into the overall stiffness of structure has shown that the joints usually incorporate a high percentage of the overall deflection. According to some results, deflection due to joints is of the order of 85 to 90% of the total structural static deflection in a machine tool.

Investigations to study the dynamic characteristics of joints have shown that damping may be obtained in a joint but only at the expense of stiffness. From the metal removal viewpoint, damping in a machine tool is advantageous. Joints do introduce frictional damping which is greater in value than internal material damping. The relative displacement between sliding elements have to be limited especially when they are situated in series with other elements. In this case, an increase of frictional damping within joints, at the expense of decreasing their static stiffness, is hardly justified.

With regards to above facts, the most efficient method to achieve both stiffness and damping would be to design the joints for maximum stiffness and to introduce damping by external means such as vibration absorbers.

3.8 Evaluation of Machine Tool Structure

The analysis of static rigidity and dynamic characteristics of machine tool structures is one of the most important factors in designing high-precision and high-efficiency machines. In present work, an attempt is made to explore the meaning of analysis, both static and dynamic.

In order to evaluate operational efficiency and accuracy of machine tools at the design stage, and finally to optimize the machine structures with respect to the performance, structural analysis using computer programming/software must be conducted. The extent to which the behavior of the various elements of the machine contributes to its overall performance is by no means fully understood. However, the results of research together with the experience of the user form the basis for the design procedures currently available. The aim of this chapter is to present systematic procedures for the analysis of the machine tool structure – a prerequisite of the sound systematic design. It is useful to know how the actual design will work in actual condition.

To achieve perfection in machine tool structure design, one has to check the design alternates for both the conditions, i.e. statically, which considers only time independent parameters, and dynamically, which takes care of dynamic behavior of structure, during actual machining. So in short, we arrive at,

a) Static analysis, and
b) Dynamic analysis of machine tool structure.

3.8.1 Static Analysis

Forces of an essentially static nature result from the static component of the cutting force, the weight of the various machine elements and thermal stresses. The latter source of stressing is not considered here although the methods to be discussed are adaptable to the thermal problem.

Thus the static problem is defined as that of obtaining a measure of the deformed shape of the structure under the action of both the cutting force and the distributed gravity force. Although the changing distribution of the gravity force, due to the movement of the machine carriages and the variable magnitude and direction of the cutting force add to the complexity of the problem this is by far outweighed by simplifications arising from the linear behavior of the stressed structure.

If only the static deformation of the structure is of interest it is generally adequate to obtain a measure of the three orthogonal displacements of each selected node. Whilst a knowledge of the relations of the nodes would lead to a more accurate construction of the deformed shape this is usually unnecessary. However, the formulation of the static characteristics represents the first requirement of more general dynamic analysis and where this is required the inclusion of node rotations will lead to a more accurate assessment of the frequencies and modes of natural vibrations. This is especially critical for the higher modes involving significant rotations of the elements. [3]

3.8.2 Dynamic Analysis

The dynamic analysis was carried out to determine the natural frequencies, the mode shapes and resistance to chatter. The dynamic behavior of the machine may be expressed in terms of its deformation resulting from the action of harmonic forces. These forces arise from two basically different sources. Firstly there exist the externally applied dynamic forces resulting from the intermittent nature of certain cutting processes and unbalanced rotating motors, shafts, gears, etc. secondly, self-induced forces are created by the dynamic motions of the structure itself. [3]

These forces include:

a) A force dependent upon the dynamic variation in the cutting condition of the tool relative to the work. (e.g. chip thickness variation)
b) The inertial force distributed throughout the entire structure caused by its acceleration during dynamic motion.
c) A distributed damping force.

The damping capacity of a machine tool is an important feature of its dynamic performance and any model which excludes damping is of only limited value. It is not possible at present to define the distribution of the damping force, prior to manufacture, with sufficient accuracy for it to be of practical value.

Still it is not easier every time to predict the dynamic characteristics at design stage. The following problems exist for evaluating characteristics of machine tool structures at design stage:

1. A machine tool structure has an infinite degree of freedom in dynamics. Therefore, an infinite number of natural modes exist in a machine structure. How to efficiently select natural modes necessary for evaluation of the characteristics among an infinite number of natural modes is very important, but in many cases thorough consideration is not given to this problem. When commercially available structural analysis programs of the finite element method, such as Pro/Mechanica, NASTRAN etc. are used, large amounts of information at many natural modes are output. In some cases, it is difficult to judge whether or not information at computed natural modes is sufficient to evaluate the characteristics.

2. At the design stage, evaluative parameters having errors which are difficult to exclude must be distinguished from parameters which can be obtained with sufficient accuracy. In vibrational characteristics of machine structures, accurate values of damping ratios denoting damping capacity are difficult to obtain at the design stage.

In such a case, damping ratios must be analyzed based on the consideration of the existence of errors.

3. Often in practical machine tool structures, directional orientations of the excited force change greatly, or the waveform of the excited force in time domain is unknown. For example, in machine tools, the directional orientations of the cutting force, etc. may change greatly as a result of the machining conditions and operational states. The evaluative method must be able to cope with such practical situations.

One of the most broadly applicable approaches to computer-aided machine dynamic analysis and design is use of a maximal set of Cartesian generalized coordinates to locate and orient bodies in space. Connections or constraints between bodies are then imposed to represent kinematics of the system. Constraints include kinematics relations that describe interconnections and driving constraints that describe specified inputs to the system. The resulting equations of motion of constrained multi body dynamic systems comprise a mixed system of differential and algebraic equations (DAE). Constraint stabilization method is used to solve set of DAE.

3.8.3 Relation between Static and Dynamic Analysis

It will be explained that if between two particular points of the structure with particular directions (P) of the force and (x) of the displacement, at one time the static stiffness and at other times the dynamic stiffness are considered, the significance of the static stiffness of individual element is different for each of both stiffness. Moreover, the dynamic stiffness being a function of frequency the significance of the static stiffness of the individual elements of the structure for the resulting stiffness must vary with the frequency.

Nevertheless, especially if the resulting stiffness as previously defined between tool and work piece, k and k', are concerned, experience shows that from the dynamic point of view the basic modes of natural vibrations are more important then the higher modes. It is mainly in the higher modes that the significant springs are very different from the significant springs resulting from the static considerations. In the basic modal shapes the roll of the individual springs would be approximately the same as in the static deformation shape, if the damping was regularly distributed in the frame.

Therefore, it may be concluded that if the stiffness of those springs which contribute mostly to the static resulting stiffness, K or K', respectively, are increased then the dynamic stiffness corresponding in point of action and directions of the force with K or K', respectively, will also increase. It may be that some of the springs which are significant statically are non-important dynamically because of a high damping in the corresponding part of the frame, so that their stiffening from the dynamic point of view is unnecessary and some effort would be wasted if as basis for the improvement of the structure the static instead of the dynamic analysis were used.

3.9 Analysis of Machine Tool Structure

In earlier times, when other efficient methods of analysis were not explored and used, some other techniques were used. Analysis of the prototype was done by preparing one scale down model of the prototype and the performance of the machine tool was approximated. It is known as "Model analysis". The method is discussed in some details:

3.9.1 Use of Models for Analysis of Machine Tool Structures

Reduced scale models are the most direct form of analog and since they had proved valuable in other fields, it seemed reasonable to try it for machine tool structures. Although the overall cutter-work piece flexibility is the property required for the prediction of machine tool performance it is evident that a geometrically similar model can only predict this property correctly if the detail behavior of the model is also similar. For this reason the investigation has involved a very detailed examination of a complete prototype machine tool and a model [1].

Basic Considerations and Preliminary experiments:

1) Choice of Model Material:

   Reduced scale models were decided to make of thermoplastic for two reasons:

   a) It was hoped that thermoplastic models would be easy to construct and alter as required.
   b) The mechanical properties of thermoplastics are related to those of cast iron,

and that of steel in such a way that the resonant frequencies of models would come comfortably inside the bandwidth of readily available test equipment.

"Perspex" was chosen from among the various possible thermoplastics because:

1) It was readily available in suitable forms
2) Methods of fabricating it are highly developed
3) Information was available about its mechanical properties, especially dynamic.

2) Tests on simple beam Models:

Experimental work started with vibration tests on calculable beam models whose important behavior was found to be as predicted within the limits of experimental error. The technique of using inductive proximity pick-ups for displacement measurement was evolved at this stage.

3) Tests on an actual Machine Tool Structural Element and a Model thereof:

The next step taken was to test the validity of a model of a complex beam like structure by comparing it with a prototype.

In an experiment, a ¼ scale Perspex model and cast iron prototype was very good both in respect of resonant frequency and mode shape.

The results obtained with testing models separately differ from the behavior of complete machine tool. From this it became clear that the only really useful sort of model would be a complete one in which the detail load distributions on individual elements would be automatically similar provided that the model was truly similar to the prototype. Unfortunately a difficulty arises when trying to make a scale model of a complete machine tool. The reason for this is that almost no information is available about the mechanical properties of joints in machine tool. Hence it was decided to try to solve the basic problem of predicting overall flexibility with model in two stages:

a) Use a model to represent the metallic structures only in the fist instance and keep the effects of the joints to a minimum by eliminating sliding clearances.
b) Introduce the effect of the joints later by some means

Although, the analysis using scale models give faithful results, the major drawback of the technique is inability to check design alterations for finding the optimum design. Again the technique requires ample time to construct very similar model of prototype with required accuracy.

3.9.2 Mathematical Model approach for Analysis of structures

The limitations of physical model techniques, in other branches of engineering led to the development of mathematical models representing a variety of mechanical structures. The development of the analytical approach has accelerated rapidly since the introduction of large digital and analogue computer installations, especially in the structural and aircraft industries and numerous examples of the computer analysis of many structures are now to be found in the technical journals of these industries.

In developing a mathematical model careful consideration must be given to the problem as a whole. Firstly the information required from the solution of the model must be defined and secondly the availability of the fundamental information required by the model should be considered both quantitatively and qualitatively. A detailed discussion of all the available method is beyond the scope of this chapter.

Firstly, the machine tool structure is considered too complicated geometrically to use classical formulations involving the expression of elasticity and mass as continuous functions of the structural dimensions and subsequently calling for the solution of partial differential equations. Hence, the method adopted will be based upon a "lumped" representation of the elasticity and mass. Thus the structure is approximated to by a model composed of a number of interconnected weightless springs (beams) and concentrated masses. A simple portal frame and its lumped parameter model are shown in Fig. 3.11 below [2].

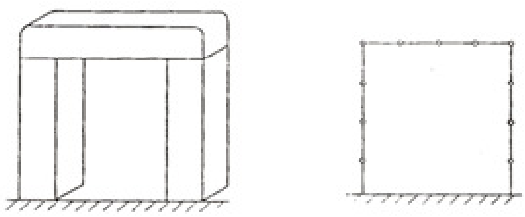


Fig. 3.11 Lumped model of portal frame

The dynamic behavior of such a model may now be obtained from the solution of the set of ordinary differential equations yielding a number of natural frequencies and corresponding normal mode shapes. The latter will be defined by the amplitudes of displacements of all the mass points during the vibration of the structure at a natural frequency. If the structure is sufficiently simple it is possible to estimate some of its modal shapes and so use this information to calculate the potential and kinetic energies and hence deduce the natural frequency corresponding to the assumed mode shape. This approach might be applicable to the calculation of some of the lower natural frequencies of certain machines but it is of very limited practical value.

Fortunately the solution of the equations describing the behavior of the "lumped" model presents no difficulty if a sufficiently large computer is available. Available computer routines require the formulation of problems of this type in matrix notation. The systematic character of matrix methods also permits large amounts of information to be handled by routine procedures.

3.9.3 The Lumped Constant Model

By far the most important step in the lumped constants method is that of transforming the actual structure into a representative lumped model. This is by no means a straight forward task and attention must be given to a number of considerations.

Firstly the structure should be replaced by a line diagram describing its overall topology. Thus each section of the machine is represented by a single line drawn through its principal elastic axis and the whole structure is composed of the interconnection of such lines. Typical examples are shown in Fig. 3.12**,** next page. Next consideration concerns the number of further subdivisions and their locations. This will be dictated by the following four factors:

a) Since the analysis will yield the deflections at discrete points resulting from static or dynamic deformation, these points (nodes) should be sufficient in number and of such a distribution to enable a practically useful deformation shape to be determined.
b) Where, two elements are physically joined, or where a change in the cross section of an element or the directional orientation of its principal elastic axis occurs, at least

one additional node is required. In the case of a clamped or bolted joint it is desirable to place a node at either side so that the joint stiffness characteristics, if known, can be included with ease.

c) For the subsequent dynamic analysis the number of nodes must be sufficient to represent reasonably the actual mass distribution when this is replaced by concentrated masses at these nodes.

d) From the computational point of view the minimum number of nodes consistent with the latter requirements should be selected.

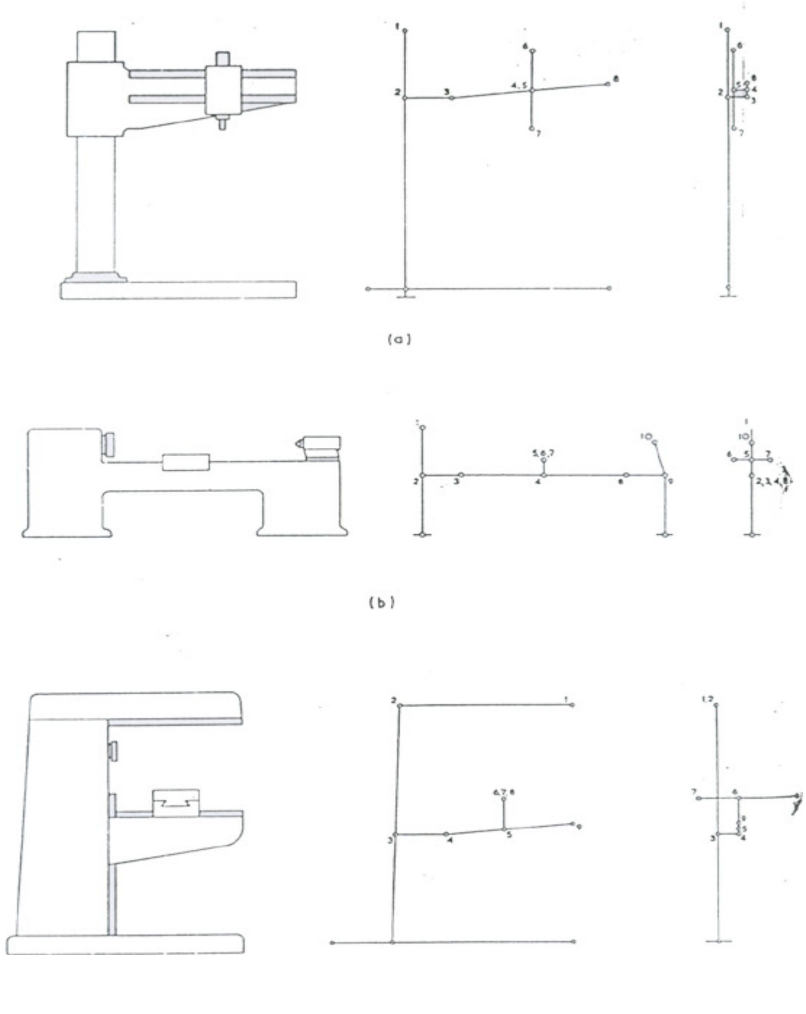


Fig. 3.12 Lumped models of a) Radial Arm Drill

b) Center lathe

c) Horizontal milling machine

It is not possible to give a rigid set of rules which will satisfy all of these requirements and to some extent experienced judgment must be used. If only the static deformation of the structure is of interest it is adequate to obtain a measure of the three orthogonal displacements of each selected node. Whilst a knowledge of the rotations of the nodes would lead to a more accurate construction of the deformed shape this is usually unnecessary. However, the formulation of the static characteristics represents the first requirement of the more general dynamic analysis and where this is required the inclusion of node rotations will lead to a more accurate assessment of the frequencies and modes of natural vibrations. This is especially critical for the higher modes involving significant rotations of the elements. Therefore, the static analyses to be carried out will account for the application of forces and moments at all the selected nodes and the resulting rectilinear deflections and rotations of such points.

3.9.4 FEA: An Efficient, Quick and Reliable Technique of Analysis

As explained, the earliest accepted method of analysis prior to manufacture of the actual machine is model analysis. This method involves manufacture of a scale model of proposed design. Since, such a model is reasonably faithful representation of the actual design, a considerable amount of useful information is obtained from subsequent testing. However, with this method it is not possible to check different design variations in structure, because it requires every time modified model to be prepared. Thus, physical prototyping is an expensive, time-consuming way to do this, and the usual alternative – traditional numerical analysis – depends on highly trained specialists to get accurate results.

The limitations of physical model techniques have led to the development of mathematical models representing a variety of mechanical structures. In all the mathematical models, the first step is to subdivide the real structure into simple elements such as beams and polygonal plates, these elements being interconnected at specified nodal or station points only. Then relationships between the forces and displacements at these nodal points are then derived in the form of element matrices, the manipulation of these element matrices giving the required information about the whole structure. As in this approach, whole structure is divided into finite elements, it is known as 'Finite

Element Analysis'. The FEA is a very useful tool in engineering today and same has proved to be an important technique in machine tool structural analysis. Furthermore, FEA can also be used to determine portions of an object which are more massive than required, and therefore can be reduced in amount of material. This can be a useful tool in streamlining and perfecting a product where weight, size, or material is of critical concern.

Delivering higher quality products with tighter schedules compels companies to adopt computer aided engineering (CAE) software as an intrinsic part of their design process, i.e. Creo 1.0, I-Deas, ANSYS, Nastran etc., which are quicker, reliable and efficient in performing even more complex and odd cases of real world problems. Thus, Computer is an invaluable tool for a designer in his task for evaluating alternative designs to arrive at the optimum design and also predicting the static, dynamic and thermal behavior of the machine before arriving at the final design.

In next chapter, after studying design aspects of machine tool structures, analysis of a Turning centre is done for its major structural parts, i.e. Head, Saddle and Bed and then according to the results design optimization is suggested where required. For analysis of such a complex structures Pro-Mechanica, simulation module of Creo 1.0, is preferred over other simulation software.