UNIT 13  DESIGN OF GUIDEWAYS AND SPINDLE

Structure

13.1  Introduction

Objectives

13.2  Functions of Guideways

13.3  Types of Guideways

13.3.1  Guideways with Sliding Friction

13.3.2  Guideways with Rolling Friction

13.4  Design of Slideways

13.4.1  Design of Slideways for Wear Resistance

13.4.2  Design of Slideways for Stiffness

13.5  Functions of Spindle

13.6  Design of Spindle

13.6.1  Deflection of Spindle Axis due to Bending

13.6.2  Deflection of Spindle Axis due to Compliance of Spindle Supports

13.6.3  Optimum Spacing between Spindle Supports

13.6.4  Deflection due to Compliance of the Tapered Joint

13.6.5  Permissible Deflection and Design for Stiffness

13.7  Summary

13.8  Key Words

13.1 INTRODUCTION

Design of machine tool elements is critical in tool engineering. They must withstand against applied external load. The machine tool elements such as guideways, slideways and spindle unit are discussed in detail in the next section. Additionally, requirements, functions and types of guideways and spindle are also explained.

Objectives

After studying this unit, you should be able to understand

- various types and functions of guideways,
- the design of slideways,
- the design of spindle, and
- function of spindle.

13.2 FUNCTIONS OF GUIDEWAYS

The Guideway is one of the important elements of machine tool. The main function of the guideway is to make sure that the cutting tool or machine tool operative element moves along predetermined path. The machine tool operative element carries workpiece along with it. The motion is generally circular for boring mills, vertical lathe, etc. while it is straight line for lathe, drilling, boring machines, etc.

Requirements of guideways are:

(a)  Guideway should have high rigidity.

(b)  The surface of guideways must have greater accuracy and surface finish.
Guideways should have high accuracy of travel. It is possible only when the deviation of the actual path of travel of the operative element from the predetermined normal path is minimum.

Guideways should be durable. The durability depends upon the ability of guideways to retain the initial accuracy of manufacturing and travel.

The frictional forces acting on the guideway surface must be low to avoid wear.

There should be minimum possible variation of coefficient of friction.

Guideways should have good damping properties.

**SAQ 1**

(a) What are various functions of guideways?

(b) State requirements of guideways.

### 13.3 TYPES OF GUIDEWAYS

The guideways are mainly classified according to the nature of friction between contacting surfaces of the operative element:

(a) Guideways with sliding friction

(b) Guideways with rolling friction

#### 13.3.1 Guideways with Sliding Friction

The friction between the sliding surfaces is called as guideways with sliding friction. These guideways are also called as slideways. The slideways are further classified according to the lubrication at the interface of contacting surfaces. The friction between the sliding surfaces may be dry, semi-liquid, and liquid. When the lubrication is absent in between contacting surfaces, it is called as dry friction. Dry friction is rarely occurred in machine tools.

When two bodies slide with respect to each other having lubrication between them, the sliding body tends to rise or float due to hydrodynamic action of the lubricant film. The principle of slider is shown in Figure 13.1.

![Figure 13.1: Principle of a Slider](image)

The hydrodynamic force,
$F_h = C \cdot v_s \quad \ldots (13.1)$

Where $C$ is constant and depends upon wedge angle $\theta$, the geometry of sliding surfaces, viscosity of the lubricant and parameter of lubricant film.

$v_s$ is sliding velocity.

$W$ is weight of the sliding body.

The resultant normal force acting on sliding body,

$R = F_h - W$

From Eq. (13.1), it is clear that the hydrodynamic force increases with increase in sliding velocity. The sliding body rests on the stationary body when hydrodynamic force is less than the weight of the sliding body. Here, there are semi-liquid type friction conditions and under these conditions the two bodies are partially separated by the lubricant film and partially have metal to metal contact. The resultant normal force on sliding body starts to act upwards and the body floats as hydrodynamic force is greater than the sliding weight of the body. The sliding surfaces are completely separated by the lubricant film and liquid friction occurs at their interface. The slideways in which the sliding surfaces are separated by the permanent lubricant layer are known as hydrodynamic slideways. This permanent lubrication layer is due to hydrodynamic action. A permanent lubricant layer between the sliding surfaces can be obtained by pumping the liquid into the interface under pressure at low sliding speed. The sliding body is lifted by this permanent lubricant layer. Such slideways are called as hydrostatic slideways.

13.3.2 Guideways with Rolling Friction

These are also called as anti friction ways. The anti friction slideways may be classified according to the shape of the rolling element as:

(a) Roller type anti friction ways using cylindrical rollers.
(b) Ball type anti friction ways using spherical balls.

SAQ 2

(a) Explain various types of guideways.
(b) Explain with figure the principle of slider.

13.4 DESIGN OF SLIDEWAYS

Slideways are designed for wear resistance and stiffness.

13.4.1 Design of Slideways for Wear Resistance

The wear resistance of slideways is mainly dependent upon maximum pressure acting on the mating surfaces. This condition may be given as

$p_{max} \leq P_{mp} \quad \ldots (13.2)$

where $p_n$ = maximum pressure acting on the mating surface, and

$P_{mp}$ = permissible value of the maximum pressure.

It is seen during the subsequent analysis that slideway designed for maximum pressure is quite complicated. Sometimes, this design is replaced by a simple procedure based upon the average pressure acting on the mating surfaces. The condition is that:

$P_a \leq P_{ap} \quad \ldots (13.3)$

where $p_n$ = average pressure acting on the mating surface, and
The permissible value of the average pressure, $p_{ap}$, is required for the design of slideways for wear resistance.

Hence from Eqs. (13.2) and (13.3), the design of slideways for wear resistance requires that:

(a) $p_m$ and $p_a$ to be known,
(b) $p_{mp}$ and $p_{ap}$ to be known, and
(c) The values of $p_{mp}$ and $p_{ap}$ are given for different operating conditions of slideways on the basis of experience. For determining $p_m$ and $p_a$, the first and foremost task is to determine the forces acting on the mating surfaces.

Forces acting on the mating surfaces in combination of $V$ and flat slideways.

The combination of $V$ and flat slideways is commonly used in lathe machines. The schematic diagram of slideways and the forces acting on the system for the case of orthogonal cutting are illustrated in Figure 13.2.

The forces acting on $V$ and flat slideways are:

(a) Cutting force component $F_z$ (in the direction of the velocity vector) and $F_y$ (radial),
(b) Weight of carriage $W$, and
(c) Unknown forces $F_1$, $F_2$ and $F_3$ acting on the mating surfaces.

The unknown forces are calculated from following equilibrium conditions:

Sum of components of forces acting along $Y$-axis = 0

\[ \sum Y = 0 \]
\[ F_1 \sin \lambda - F_2 \sin \gamma + F_y = 0 \]  \[ \ldots (13.4) \]

Sum of components of forces acting along $Z$-axis = 0

\[ \sum Z = 0 \]
\[ F_1 \cos \lambda + F_2 \cos \gamma + F_3 - W - F_z = 0 \]  \[ \ldots (13.5) \]

Moment of all forces about $X$-axis = 0

\[ \sum M_x = 0 \]
Design of Guideways and Spindle

\[ F_1 \cos \lambda \cdot \frac{b}{2} + F_2 \cos \gamma \cdot \frac{b}{2} - F_z \cdot \frac{d}{2} - F_y \cdot h - F_3 \cdot \frac{b}{2} = 0 \]

\[ \therefore F_3 = F_1 \cos \lambda + F_2 \cos \gamma - F_z \frac{d}{b} - F_y \frac{2h}{b} \quad \ldots (13.6) \]

Substituting value of \( F_3 \) in Eq. (13.5), we get,

\[ F_1 \cos \lambda + F_2 \cos \gamma + F_1 \cos \lambda + F_2 \cos \gamma - F_z \frac{d}{b} - F_y \frac{2h}{b} - F_z - W = 0 \]

or

\[ 2 (F_1 \cos \lambda + F_2 \cos \gamma) = F_z \left(1 + \frac{d}{b}\right) + F_y \frac{2h}{b} + \frac{W}{2} \]

\[ \therefore F_1 \cos \lambda + F_2 \cos \gamma = \frac{F_z (d + b)}{2b} + F_y \frac{h}{b} + \frac{W}{2} \quad \ldots (13.7) \]

If the apex angle of the V is 90°, and assume that present angle \( \gamma \) may change to \( \gamma = 90 - \lambda \), the solution of simultaneous algebraic Eqs. (13.4) and (13.7) gives :

\[ F_1 = \frac{F_z (d + b)}{2b} \cos \lambda + F_y \frac{h}{b} \cos \lambda - F_y \sin \lambda + \frac{W}{2} \cos \lambda \quad \ldots (13.8) \]

\[ F_2 = \frac{F_z (d + b)}{2b} \sin \lambda + F_y \frac{h}{b} \sin \lambda + F_y \cos \lambda + \frac{W}{2} \sin \lambda \quad \ldots (13.9) \]

Substituting the values of \( F_1 \) and \( F_2 \) in Eq. (13.6) we get,

\[ F_3 = \frac{F_z (b - d)}{2b} - F_y \cdot \frac{h}{b} + \frac{W}{2} \quad \ldots (13.10) \]

Eq. (13.10) represents the forces acting on the mating surfaces in combination of two flat slideways.

The schematic diagram of the slideways and the forces acting on the system under orthogonal cutting conditions are shown in Figure (13.3).

![Figure 13.3 : Forces Acting on Combination of Two Flat Slideways](image)

The forces acting on combination of two flat slideways are :

(a) Cutting force components, i.e. axial \( F_x \), radial \( F_y \), and \( F_z \) in the direction of velocity vector.

(b) Weight of carriage, \( W \).

(c) Unknown forces \( F_1, F_2 \) and \( F_3 \) acting on the mating surfaces.

(d) Frictional forces \( \mu F_1, \mu F_2, \mu F_3 \), where \( \mu \) is the coefficient of friction between the sliding surfaces.

The unknown forces \( F_1, F_2 \) and \( F_3 \) are calculated from following equilibrium conditions :
\[ \sum X = 0 \]
\[ F_x + \mu (F_1 + F_2 + F_3) - R = 0 \]  \( \ldots \) (13.11)
\[ \sum y = 0 \]
\[ F_2 - F_y = 0 \]  \( \ldots \) (13.12)
\[ F_2 = F_y \]
\[ \sum z = 0 \]
\[ F_1 + F_3 - F_z - W = 0 \]  \( \ldots \) (13.13)
\[ \sum M_x = 0 \]
\[ W \frac{b}{2} + F_z y_p - F_y h - F_3 b = 0 \]  \( \ldots \) (13.14)

From Eq. (13.14)
\[ F_3 = \frac{F_z y_p - F_y h}{b} + \frac{W}{2} \]

On substituting the value of \( F_3 \) in Eq. (13.13)
\[ F_1 = F_z + \frac{W}{2} - \frac{F_z y_p - F_y h}{b} \]  \( \ldots \) (13.15)

The pulling force, \( R \) is calculated from Eq. (13.11) on substituting the values of \( F_1, F_2 \) and \( F_3 \).
\[ R = F_x + \mu (F_z + F_y + W) \]  \( \ldots \) (13.16)

**Determination of Average Pressure**

The average pressure can be determined as :
\[ p_{F_1} = \frac{F_1}{vL} \]
\[ p_{F_2} = \frac{F_2}{wL} \]
\[ p_{F_3} = \frac{F_3}{uL} \]

where \( L \) = length of the carriage, and
\( v, w, u \) = the width of slideway faces on which forces \( F_1, F_2 \) and \( F_3 \) are acting respectively.

**Determination of Maximum Pressure**

It is necessary to establish the points of action of the resultant normal forces \( F_1, F_2 \) and \( F_3 \) on the respective faces for calculating the maximum pressure. The distance between the point of action of normal force \( F_1 \) on the flat slideway I and the center of the carriage is denoted by \( x_A \). This is shown in Figure 13.3. The distance between force \( F_2 \) acting on the vertical face of flat slideway II and the center of the carriage is denoted by \( x_B \), and the distance between force \( F_3 \) acting on horizontal face of flat slideway II and the center of the carriage is denoted by \( x_C \). For determining \( x_A, x_B \) and \( x_C \), we have two equilibrium conditions :
\[ \sum M_{x_A} = 0 \]
\[ F_y h + F_x x_p - F_1 x_A - F_3 x_c + Rz_Q = 0 \]  \( \ldots \) (13.17)
And \[ \sum M_z = 0 \]

\[ F_x y_p + F_y x_p - R y_Q - F_2 x_B + \mu F_2 \frac{(l + u)}{2} + \mu F_3 l = 0 \quad \ldots (13.18) \]

An additional equation may be written by assuming that the moment of reactive forces \( F_1 \) and \( F_3 \) about the \( Y \)-axis is proportional to the width of the slideway face, i.e.

\[ \frac{F_1 x_A}{F_2 x_c} = \frac{v}{u} \quad \ldots (13.19) \]

On solving Eqs. (13.17), (13.18) and (13.19), we get, the values of \( x_A \), \( x_B \) and \( x_c \).

The ratio of \( x_A / L \), \( x_B / L \) and \( x_c / L \) calculates the shape of the pressure distribution diagram and the maximum pressure on a particular face of the slideway. The procedure for determining the maximum pressure on flat slideway I is being subjected to the normal force \( F_1 \) which is described below. The most general case of pressure distribution along the length of contact \( L \) corresponds to a trapezoid as shown in Figure 13.4.

![Figure 13.4: Trapezoidal Pressure Distribution along Slideway Length](image)

Force \( F_1 \) acts at the center of gravity of trapezoid. The distance \( y_c \) at the center of gravity from the larger arm of the trapezoid can be determined as:

\[ y_c = \frac{p_{\text{min}} L}{2} + \frac{(p_{\text{max}} - p_{\text{min}})}{2} \frac{L}{3} \frac{L}{L} \]

\[ p_{\text{min}} L + \left( \frac{p_{\text{max}} - p_{\text{min}}}{2} \right) \frac{L}{L} \]

\[ y_c = \frac{p_{\text{max}} + 2p_{\text{min}}}{p_{\text{max}} + p_{\text{min}}} \]

As a result,

\[ x_A = \frac{L}{2} - y_c \]

\[ x_A = \frac{L}{6} \frac{p_{\text{max}} + p_{\text{min}}}{p_{\text{max}} + p_{\text{min}}} \]

\[ \ldots (13.21) \]

Now \[ \frac{x_A}{L} < \frac{1}{6} \]

In this case the pressure distribution diagram represents a trapezoid. This is explained with the help of following example.
Let \( \frac{x_A}{L} < \frac{1}{10} \)

From Eq. (13.18), we get,
\[
P_{\text{max}} - P_{\text{min}} = \frac{6}{10} (P_{\text{max}} + P_{\text{min}})
\]
\[
P_{\text{min}} = \frac{0.4}{1.6} P_{\text{max}}
\]

From above equation, it is clear that \( P_{\text{min}} \) is a positive, non-zero value. Hence pressure distribution diagram must be a trapezoid.

\[
\therefore P_{\text{max}} + P_{\text{min}} = 2P_{\text{av}} \quad \ldots (13.22)
\]

From Eq. (13.16), we get,
\[
P_{\text{max}} - P_{\text{min}} = \frac{12x_A}{L} P_{\text{av}} \quad \ldots (13.23)
\]

On solving Eqs. (13.22) and (13.23), we get
\[
P_{\text{max}} = P_{\text{av}} \left(1 + \frac{6x_A}{L}\right)
\]

### 13.4.2 Design of Slideways for Stiffness

Stiffness is one of the important parameters for designing slideways. The design of slideways for stiffness requires that the deflection of the cutting edge due to contact deformation of the slideway should not exceed certain permissible value. This value is specified from considerations of required machining accuracy. For example, in lathe machine, the vertical deflection of the tool or its horizontal deflection in the direction of feed motion has little effect on the dimensional accuracy of the machined workpiece. However, diametrical error of 2 \( C \) takes place due to a deflection of the cutting edge by \( C \) in the radial direction. Hence the lathe slideways are designed for stiffness with a view to restrict their radial deflection. The appropriate direction for stiffness design in various machine tools can be selected in each particular case from the above mentioned principle.

In a bed using two flat slideways as shown in Figure 13.5, the radial deflection takes place due to the contact deformation \( C_B \) of the vertical face of the slideway and rotation of the saddle due to unequal contact deformation \( C_A \) and \( C_C \) of the horizontal flat faces.

Hence radial deflection is given by
\[
\frac{C_A - C_B}{C_C} \cdot h
\]

Hence total radial deflection is given by
\[
C_{FP} = C_B + \frac{C_A - C_C}{b} \cdot h \quad \ldots (13.24)
\]

In the lathe bed using a combination of flat and \( V \) slideways, the contact deformation of the flat slideways is shown by lowering \( C_C \). The faces of the \( V \)-slideways suffer a deformation of \( C_A \) and \( C_B \). This will result in:

(a) Vertical lowering of the \( V \)-slideway by
\[
C_v = C_B \sin \lambda + C_A \cos \lambda
\]

(b) Horizontal displacement of the apex of the \( V \)-slideway by
\[
C_h = C_B \cos \lambda - C_A \sin \lambda
\]
This nature of deformation of flat and V-slideways results in:

(a) radial deflection of the cutting edge by $C_h$ and
(b) rotation of the saddle due to unequal vertical lowering of the flat and V slideways which leads to radial deflection of the cutting edge by $\frac{C_v - C_e}{b} \cdot h$

The total radial deflection then becomes

$$C_{Fv} = C_h + \frac{C_v - C_e}{b} \cdot h$$

The contact deformation is assumed proportional to the average pressure for the purpose of stiffness design. The coefficient of proportionality $d$ is known as contact compliance. This coefficient must be determined for each pair of slideway materials. However for approximate calculations, an average value of $d = 1 \times 10^{-6}$ mm$^2$/N may be used.

Eqs. (13.24) and (13.25) can be rewritten as follows:

$$C_{FF} = d \ p_B + d \ \frac{p_A - p_C}{b} \cdot h$$

$$C_{Fv} = d \ (p_B \cos \lambda - p_A \sin \lambda) + \frac{dh}{b} \ (p_B \sin \lambda + p_A \cos \lambda - p_C)$$

After calculating the total radial deflection of the cutting edge, the design for stiffness is carried out in accordance with $C_i = C_{ip}$.

**SAQ 1**

(a) What are principle parameters in designing slideways?

(b) Design the slideways for machine tool.

---

**13.5 FUNCTIONS OF SPINDLE**

The spindle is one of the most important elements of the machine tool.

The functions of the spindle of machine tool are as follows:
(a) It clamps the workpiece or cutting tool in such a way that the workpiece or cutting tool is reliably held in position during the machining operation.

(b) It imparts rotary motion or rotary cum translatory motion to the cutting tool or workpiece.

(c) It is used for centering the cutting tool in drilling machine, milling machine, etc. while it centers the workpiece in lathes, turrets, boring machine, etc.

Requirements of spindle are:

(a) The spindle should rotate with high degree of accuracy. The accuracy of rotation is calculated by the axial and radial run out of the spindle nose. The radial and axial run out of the spindle nose should not exceed certain permissible values. These values depend upon the required machining accuracy. The rotational accuracy is influenced mainly by the stiffness and accuracy of the spindle bearings especially by the bearing which is located at the front end.

(b) The spindle unit must have high dynamic stiffness and damping.

(c) The spindle bearing should be selected in such a way that the initial accuracy of the unit should be maintained during the service life of the machine tool.

(d) Spindle unit should have fixture which provides quick and reliable centering and clamping of the cutting tool or workpiece.

(e) The spindle unit must have high static stiffness. Maximum accuracy is influenced by the bending, axial as well as torsional stiffness.

(f) The wear resistance of mating surface should be as high as possible.

(g) Deformation of the spindle due to heat transmitted to it by workpiece, cutting tool, bearings etc. should be as low as possible. Otherwise it will affect the accuracy of the machining accuracy.

13.6 DESIGN OF SPINDLE

Figure 13.6 shows schematic diagram of spindle. A spindle represents a shaft with

(a) length ‘a’ which is acted upon by driving force $F_2$, and

(b) cantilever of length ‘m’ acted upon by external force $F_1$.

The spindle is basically designed for bending stiffness which requires that maximum deflection of spindle nose should not exceed a prespecified value, i.e.

$\frac{d_{\text{max}}}{d_{\text{per}}} \leq \ldots \ (13.28)$

The total deflection of spindle nose consists of deflection $d_1$ of the spindle axis due to bending forces $F_1$ and $F_2$ and deflection $d_2$ of the spindle axis due to compliance of the spindle supports. When the spindle has tapered hole in which a center or cutting tool is mounted, the total deflection of the center or cutting tool consists of deflections $d_1$, $d_2$ and $d_3$ of the center or cutting tool due to compliance of the tapered joint.
13.6.1 Deflection of Spindle Axis due to Bending

To calculate the deflection of the spindle nose due to bending, one must establish a proper design diagram. The following guidelines may be used in this regard.

(a) If the spindle is supported on a single anti-friction bearing at each end, it may be represented as a simply supported beam, and

(b) If the spindle is supported in a sleeve bearing, the supported journal is analyzed as a beam on an elastic foundation; for the purpose of the design diagram the sleeve bearing is replaced by a simple hinged support and a reactive moment $M_r$, acting at the middle of the sleeve bearing.

The reactive moment is given as:

$$M_r = C \cdot M$$

where $M = $ bending moment at the support, and

$C = $ constant $= 0$ for small loads and $0.3$ to $0.35$ for heavy load.

![Diagram](image)

Figure 13.7 : Effect of Various Force on Spindle

Figure 13.7(a) shows schematic diagram of spindle. Figure 13.7(b) depicts the design diagram of the spindle and figure 13.7(c) illustrates deflected axis of the spindle.

Consider the spindle shown in Figure 13.7(a). By replacing the rear ball bearing by a hinge and the front sleeve bearing by a hinge and reactive moment $M_r$, the spindle can be reduced to the design diagram as shown in Figure 13.7(b). The deflection at the free end of the beam (spindle nose) can be determined by Macaulay’s method and is found out to be

$$d_1 = \frac{1}{3EI_a} \left[ F_1 m^2 (a + m) - 0.5 F_2 b m \left(1 - \frac{k}{a}\right) - M_r a m \right] \quad \ldots (13.29)$$

where $E$ is Young’s modulus of the spindle material.

$I_a$ is average moment of inertia of the spindle section.

The deflection of the beam is shown in Figure 13.7(c).

13.6.2 Deflection of Spindle Axis due to Compliance of Spindle Supports

Let $\delta_E$ and $\delta_G$ represent the displacement of the rear and front support respectively. Owing to compliance support, the spindle deflects are shown in Figure 13.8. From similarity of triangles $OHH'$ and $OGG'$
\[
\frac{d_2}{m + x} = \frac{\delta_G}{x}
\]

\[
\therefore \quad d_2 = \left(1 + \frac{m}{x}\right)\delta_G 
\]...

(13.30)

Figure 13.8 : Deflection of the Spindle due to Compliance of Support

From similarity of triangles \(OEE'\) and \(OGG'\), we get

\[
\frac{\delta_G}{x} = \frac{\delta_E}{a - x}
\]

\[
\therefore \quad x = \frac{a\delta_G}{\delta_E + \delta_G}
\]

On substituting these values of \(x\) in Eq. (13.30), \(d_2\) changes to,

\[
\therefore \quad d_2 = \left(1 + \frac{m}{a}\right)\delta_G + \delta_E \frac{m}{a}
\]

(13.31)

Hence it is clear from above equation that displacement \(\delta_G\) of the front bearing has greater influence upon deflection \(d_2\) of spindle nose than displacement \(\delta_E\) of the rear bearing.

Displacement \(\delta_E = \frac{R_E}{S_E}\)

and

\(\delta_G = \frac{R_G}{S_G}\)

Where \(R_E\) and \(R_G\) are the support reactions at \(E\) and \(G\) respectively.
\(S_E\) and \(S_G\) are stiffness at \(E\) and \(G\) respectively.

At equilibrium,

\[
\sum \! M_E = 0
\]

\[
\therefore \quad R_G a - F_2 k + M_r - F_1 (m + a) = 0
\]

\[
\therefore \quad R_G = \frac{F_2 k - M_r + F_1 (m + a)}{a}
\]

Similarly

\[
\sum \! M_G = 0
\]

\[
\therefore \quad R_E a - F_2 b - M_r + F_1 m = 0
\]
\[ R_E = \frac{(F_2 b + M_r - F_1 m)}{a} \]

\[ \text{Deflection } d_2 = \frac{F_2 k - M_r + F_1 (m + a)}{a \cdot S_B} \left(1 + \frac{m}{a}\right) + \frac{F_2 b + M_r - F_1 m}{a \cdot S_A} \left(\frac{m}{a}\right) \ldots (13.32) \]

Hence total deflection \( d \) (shown in Figure 13.9) is obtained as

\[ d = d_1 + d_2 \]

**Figure 13.9 : Total Deflection of Spindle Axis**

### 13.6.3 Optimum Spacing between Spindle Supports

The ratio ‘\( \tau \)’ is an important parameter in spindle design.

Where, \( \tau = \frac{a}{m} \). The optimum value of this ratio is the one that makes sure that minimum total deflection ‘\( d \)’ can be determined from differentiating it partially with respect to \( \tau \).

For minimum deflection \( d \), \[ \frac{dd}{d\tau} = 0. \]

The point of the minimum of the \( d_1 + d_2 \) curve, gives the optimum value of ratio \( a/m \) which generally lie between 3 and 5. The value of \( \tau_{\text{opt}} \) depends upon ratio of stiffness of the front and rear bearings, \( \eta = \frac{S_E}{S_G} \) and factor \( J = \frac{S_G I_m}{S_C I_a} \).

Where \( S_C = \frac{3E I_m}{m^3} = \) bending stiffness of the cantilever.

\( I_m = \) average moment of inertia of the spindle over cantilever.

\( I_a = \) average moment of inertia of the spindle over the supported length.

An opposite constraint on maximum span stems from the requirement that for normal functioning of the spindle driving gear, the stiffness of the span should not be less than 245-260 N/µm. This constraint is expressed through the following relationship:

\[ a \leq \frac{D_o^{4/3}}{i^{1/3}} \]

where \( D_o = \) average diameter of the supported length of the spindle,

\( i = 0.05 \) for normal accuracy machine tools, and

\( = 0.1 \) for precision machine tools.
When the spindle is supported on hydrostatic journal bearings, the maximum deflection at the middle of the span should satisfy the condition:

\[ d_{a \text{ max}} \leq 10^{-4} a \]

And the maximum span length \( a_{\text{ max}} \) should be limited by the above constraint. This constraint is based upon the requirement that the maximum misalignment due to deflection of the journals should not exceed one third of the bearing gap.

### 13.6.4 Deflection due to Compliance of the Tapered Joint

The spindle ends of most machine tools have a tapered hole for accommodating a center (in lathe) or cutting tool shanks (in milling and drilling machine). The deflection of center or shank at a distance \( d \) from the spindle axis where the force \( F \) is acting is given by equation:

\[ d_3 = \Delta + \phi d \]

where \( \Delta \) = displacement of the shank or center at the edge of taper due to contact compliance, and \( \phi \) = angle of slope of the shank or at the edge of taper.

If manufacturing the errors of the taper are ignored, \( \Delta \) and \( \phi \) can be calculated from following equations:

\[ \Delta = \frac{4 \psi D C_1}{\pi D} \left( \psi d C_2 + C_3 \right), \mu m \]

\[ \phi = \frac{4 F \psi^2 C_1}{\pi D} \left( 2 \psi d C_4 + C_2 \right) \]

where \( C_1 \) = coefficients of contact compliance

\( C_2, C_3, C_4 \) = coefficients that account for the diameter variation along the length of taper.

\[ \psi = \left( \frac{1}{2.3 C_1 D^4} \right)^{1/4}, \text{ cm}^{-1} \]

\( D \) and \( d \) are expressed in cm.

Generally displacement \( \Delta \) due to contact compliance can be ignored in comparison with the displacement due to bending of the shank or center.

Hence,

\[ d_3 = \frac{4 F \psi^2 C_1}{\pi D} \left( 2 \psi d C_4 + C_2 \right) d, \mu m \]

\[ \ldots (13.33) \]

### 13.6.5 Permissible Deflection and Design for Stiffness

The deflection \( d_1, d_2 \) and \( d_3 \) are calculated from equations (13.29), (13.32) and (13.33) respectively. The total deflection of the spindle nose can be determined as the sum of three deflections. The design of stiffness can be carried out in accordance with equation (13.28). The permissible deflection of the spindle nose \( d_{\text{perm}} \) depends upon the machining accuracy required for the machine tool. Generally, it should be less than one-third of the maximum permissible tolerance on radial run out of the spindle nose. The machining accuracy depends not only upon the radial stiffness but also upon its axial and torsional stiffness. The axial displacement of the spindle unit consists of the axial deformation of the spindle and the deformation of the spindle thrust bearing. The torsional stiffness of machine tool spindles significantly influences the machining accuracy in metal removal operations, such as gear and thread cutting in which the feed and primary cutting motions are kinematically linked. The torsional deformation of the spindle unit consists of the
Design of Guideways and Spindle

deflection of the spindle and deformation of drive components. As stated earlier, spindles are designed for stiffness, primarily radial. However in heavily loaded spindles the stiffness design must be proved by a strength check against fatigue failure. The strength check requires that

\[ f \geq f_{\text{min}} \]

where \( f \) = factor of safety against fatigue failure.

\( f_{\text{min}} \) = minimum value of safety factor

\( = 1.3 \) to \( 1.5 \)

If spindles are subjected to combined bending and torsion, the factor of safety \( f \) is determined from the equation,

\[ f = \frac{(1 - \eta^4) d_e^3 \sigma_e}{10 \sqrt{(kM_b)^2 + (bT)^2}} \]  

where \( \eta = \frac{d_i}{d_e} \) = ratio of the internal diameter to the external
d\( d_i \) = internal diameter of the spindle, mm
d\( d_e \) = external diameter of the spindle, mm
\( \sigma_e \) = Endurance limit of the spindle material, N/mm\(^2\)
\( M_b \) = mean value of the bending moment acting on the spindle, N.mm
\( T \) = mean value of the torque acting on the spindle, N.mm
\( k \) = coefficient that accounts for variation of bending moment and stress concentration.
\( b \) = coefficient that accounts for variation of torque and stress concentration.

Coefficient \( k \) can be determined from the following expression:

\[ k = U_a (1 + Q) \]

where \( U_a \) = dynamic stress concentration coefficient for normal stresses

\( = 1.7 \) to \( 2.0 \)

\( Q = \frac{M_{ba}}{M_b} \) = ratio of amplitude of the bending moment to its average value.

Coefficient \( b \) can be calculated from the following formula:

\[ b = \frac{\sigma_e}{\sigma_y} + U_\tau Q_t \]  

where \( \sigma_y \) = yield stress of the spindle material.

\( U_\tau \) = dynamic stress concentration coefficient for shearing stress.

\( \sigma_y \) = yield stress of the spindle material.

\( = 1.7 \) to \( 2.0 \)

\( Q_t = \frac{T}{T} \) = ratio of amplitude of torque to its average value.

The values of \( Q \) and \( Q_t \) generally depend upon the machining conditions. For example, in a super-finishing operation there is virtually no variation of the bending moment and torque, i.e. \( M_{ba} = T_a = 0 \) and therefore \( Q = Q_t = 0 \).

SAQ 2

(a) What are functions and requirements of spindle?
13.7 SUMMARY

The guideway is used to ensure moving of the cutting tool or machine tool operative element along predetermined path. Slideways and anti friction slideways are two types of guideways. Slideway is basically designed for wear resistance and stiffness. The pressure acting on mating surfaces create wear. Hence it is essential to resist wear. The design for stiffness specifies that the deflection of cutting edge due to contact deformation of slideways should not exceed permissible value, which is required to maintain the accuracy. Spindle is generally used for centering and clamping the workpiece, and imparting rotary motion. Spindle is basically designed for bending stiffness.

13.8 KEY WORDS

Guideways : Its function is to ensure that the cutting tool or machine tool operative element moves along predetermined path.