requirements and then work backward through the structural system. Use guesstimates for sensor, bearing, and actuator limitations to help size structural components.

8. Locate the work volume at the center of mass and in the plane of support. Also try to make the natural frequencies of the various vibration modes (e.g., translational and rotational) close together. These steps will help to minimize cross coupling between modes.[89]

In addition to these general guidelines, one of the best ways to become proficient at structural design is to observe the world around you. For example, continually examine the configurations of existing machinery.[90,91] Developing an aptitude for laying out the general configuration of a machine (i.e., where to place axes with respect to one another, what range of travel is needed, etc.), can also be achieved by dreaming and taking courses in kinematics and dynamics of machinery.[92] It is also very important that the design engineer always question conventional wisdom and try to think of new and different ways of doing things, before a conventional approach is taken. Similar comments pertain to analysis methods. Proficiency at back-of-the-envelope calculations is a must in order to arrive at a suitable point from which finite element analysis can be used effectively. Finite element analysis itself is no panacea; it takes experience to know what types of elements and boundary conditions to use and how many elements are required to obtain a convergent solution.[93] In many cases, back-of-the-envelope calculations can be used to check parts of the finite element model to determine if the mesh is appropriate.

It should be apparent to the design engineer that the structure is not a stand-alone entity. The best machine design engineers integrate the designs of all components so that they work together, and the structure is the means by which all the components are brought together. For example, the job of the structure is not just to support the part and axes but to provide protection from (for) the surrounding environment, and provide regions in which seals and power, sensor, and coolant lines can be easily installed and maintained. More than one brilliant structural design has been thrown out because there was inadequate room for support systems. In many cases, especially for prototypes, it is also advisable to leave extra room in the structure to accommodate next-larger-size motors or bearings that may be used. This makes it easy to modify the prototype or the production machine to suit a customer's special requirements.

7.5 JOINT DESIGN[94]

Joint design is one of the most difficult aspects of machine design because there are so many variables that can affect the performance of a joint. In addition, complex geometries make modeling of joints extremely difficult, and finite element methods often can provide the only models (which still in many instances leave much to be desired). It is costly and time consuming to do finite element analysis of joints; hence, most joints in machine tools are designed using back-of-the-envelope calculations, a conservative nature, and rules of thumb. Once the joint is designed by these methods, then finite element methods can be used to help check the design.

This section will discuss issues in the design of permanent joints between parts and attempt to provide at least some rules of thumb to use when designing joints to achieve desired stiffness

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89 See, for example, H. Braddick, The Physics of Experimental Method, Chapman and Hall Ltd. London, 1963.

90 For example, the biannual International Machine Tool Show (IMTS) in Chicago (held the second week of September in even-numbered years) is one of the biggest and best places to see what's available.

91 See, for example, Huebner's Machine Tool Specs, Huebner Publishing Co., Solon, OH.

92 References include V. Faires, Kinematics, McGraw-Hill Book Co, New York, 1959; B. Paul, Kinematics and Dynamics of Planer Machinery, Prentice Hall, Englewood Cliffs, NJ, 1979; J. Phillips, Freedom in Machinery, Vol. 1, Introducing Screw Theory, Cambridge University Press, New York, 1984; and R. Paul, Robot Manipulators: Mathematics, Programming, and Control, MIT Press, Cambridge, MA 1981. Many computerized kinematic design packages are currently available.

93 See, for example, M. Weck and A. Heimann, "Analysis of Variants of Machine Tool Structures by Means of the Finite Element Method," Proc. 18th Int. Mach. Tool Des. and Res. Conf., Sept. 1977, pp. 553-559.

94 Joints are usually associated with assembly operations, and a good trade magazine to read is thus Assembly Engineering, Hitchcock Publishing Co., 25W550 Geneva Road, Wheaton, IL 60188. Also see Machine Design's annual Fastening, Joining, and Assembly reference issue. Other references include A. Blake, Design of Mechanical Joints, Marcel Dekker, New York, 1985, and the section on joint design in M. Kutz (ed.), Mechanical Engineer's Handbook, John Wiley & Sons, New York, 1986. Also see R. Connolly and R.H. Thornley, "The Significance of Joints on the Overall Deflection of Machine Tool Structures," 6th Int. Mach. Tool Des. and Res. Conf., Sept. 1965, pp. 139-156.

for machine tool applications.[95] Joints are defined here as being permanent in the sense that significant effort is needed to take them apart, as opposed to sliding joints with bearing interfaces (Chapters 8 and 9) and coupled joints for periodic mating of parts (Section 7.7). The most common types of permanent joints in machine tools include[96] bolted joints, pinned joints, bonded joints, and interference fit joints. Welded joints were discussed briefly in the preceding section and enough design text is available so they will not be discussed further.

7.5.1 Bolted Joints[97]

Bolts can be used to prevent two parts from separating or sliding relative to one another. For the former, the tensile forces across the joint are transferred through the bolt shaft. For the latter, sliding motion is resisted by frictional forces generated from the normal load on the joint produced by tightening of the bolt and the coefficient of friction between the joint's parts. Because more than one bolt is usually used at a joint, it would be virtually impossible to ensure a tight fit of the bolt shafts in the holes, so it is not even worth trying. Sufficient lateral stiffness is usually provided by bolt preload and joint friction. For better resistance to shock loads, parts can be bolted in place and then holes drilled, reamed, and pinned with hardened steel dowels or roll pins. In situ drilling and reaming of the holes through both parts while they are bolted together maintains hole alignment, so multiple pins can be used.

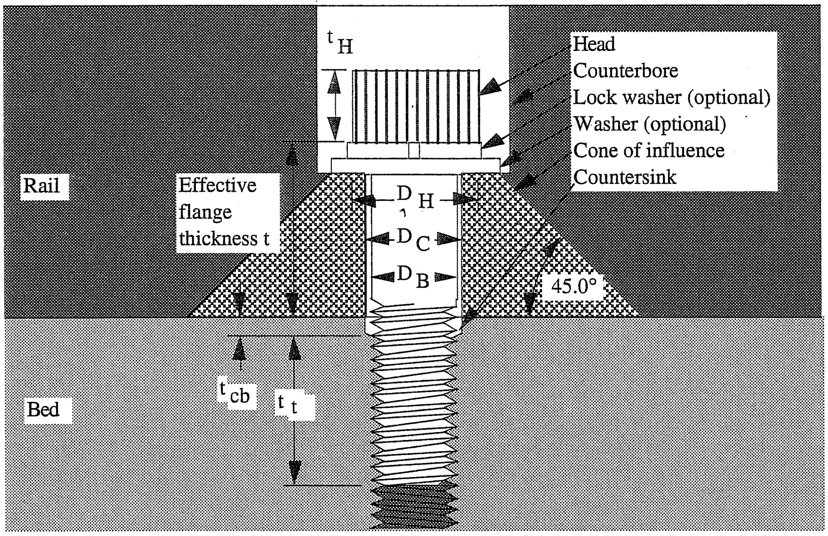

Figure 7.5.1 Typical components of a bolted assembly.

Figure 7.5.1 illustrates the cross section of a typical portion of a bolted joint. A common bolt configuration used to bolt bearing rails for T-slides is shown in Figure 7.5.2. Many rails have a double row of bolts. In general, the cantilevered length should not exceed the bedded length. Ideally, the bedded length should be about 1.5 times the cantilvered length, but sadly this often takes up too much room.

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95 Note that there is a virtual avalanche of literature associated with bolted joints. It is one of the oldest joining methods still commonly used. Most literature focuses on discussion of joint strength and gasket sealing ability. This section will focus on designing for stiffness because that is the primary concern of a machine tool designer.

96 See, for example, A. Blake, Design of Mechanical Joints, Marcel Deckker, New York, 1986, and M. Kutz (ed.), Mechanical Engineers' Handbook, John Wiley & Sons, New York, 1986.

97 For a more detailed discussion, see, for example, J. H. Bickford, An Introduction to the Design and Behavior of Bolted Joints, Marcel Deckker, New York, 1981, as well as A. Blake's book.

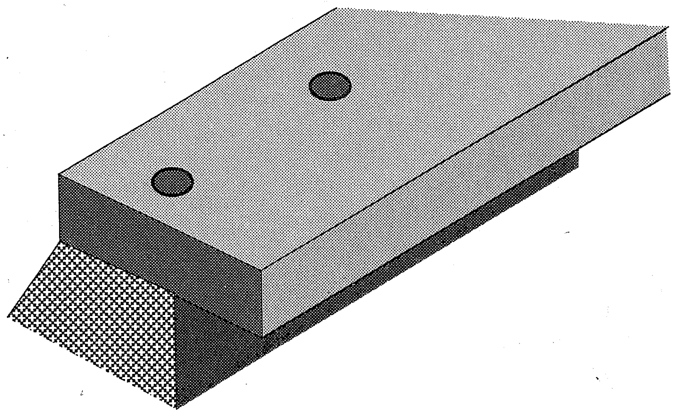

Figure 7.5.2 Counterbored and bolted bearing rail.

Some issues to consider in the design of bolted joints include:

- Tensile stiffness
- Compressive stiffness
- Lateral stiffness
- Preload
- Pinning
- Stability
- Bolting hardware

Tensile Stiffness of Bolted Joints

The region of material under the bolt head that effectively acts to resist compression by the bolt is equal to that of a 45° cone.[98] Hence the stiffness of the material under the bolt head is approximately

$$K_{\text{flange comp}} = \frac{1}{\displaystyle\int_0^t \frac{dy}{\pi E_f \left\{\left(\frac{D_H}{2} + y\right)^2 - \frac{D_C^2}{4}\right\}}} = \frac{\pi E_f D_C}{\log_e \dfrac{(D_C - D_H - 2t)(D_C + D_H)}{(D_C + D_H + 2t)(D_C - D_H)}} \tag{7.5.1}$$

The 1/logarithmic term approaches zero as D_C approaches D_H, hence the value of using a large-diameter head or a washer. Note that as the thickness goes to zero, the stiffness goes to infinity; however, one must bear in mind that the bending stiffness of the flange goes to zero along with the thickness. If the bolt is counterbored and rests on a massive flange (one or more bolt diameters thick), then locally around the bolt the stiffness of the flange will be due primarily to the ring of material in the flange shearing. Beyond two bolt radii, the shear stress is well diffused out into the material around the counterbore. For the bed which is more massive, the threads are assumed to diffuse the stress out so that inclusion of a compression term would only result in a second-order effect.

To find the deflection of the flange of thickness t due to the bolt being in tension, energy methods are used. The shear stress is

$$\tau = \frac{F}{2\pi R t} \tag{7.5.2}$$

The strain energy is given by Equation 2.3.19, where the differential volume element in the flange is $dV = 2\pi R t\,dR$. The deflection is given by

98 M. Spotts, Design of Machine Elements, Prentice-Hall, Englewood Cliffs, NJ, 1985, and J. Shigley and L. Mitchell, Mechanical Engineering Design, McGraw-Hill Book Co., New York, 1983. Both say that the region in compression is a cone with angle of 45° to the centerline of the bolt. Other references say the angle can be as small as 25°. In either case, one finds that the bolt itself is the most compliant spring in the system.

$$\delta = \frac{\partial U}{\partial F} = \frac{F}{2\pi t G}\int_{R_B}^{2R_B} \frac{dR}{R} = \frac{F \log_e 2}{2\pi t G} \tag{7.5.3}$$

The shear stiffness of the counterbored flange region is thus

$$K_{\text{flange shear}} = \frac{\pi t E_f}{(1+\eta)\log_e 2} \tag{7.5.4}$$

η – Poisson's ratio

From geometric compatibility with the bolt head, this includes the effect of shear strain in the bolt head.

If it is assumed that the effective length of thread engagement is equal to one bolt diameter, then the shear stiffness of the threaded region in the bed (neglecting the countersunk region) is

$$K_{\text{bed shear}} = \frac{\pi D_B E_t}{(1+\eta)\log_e 2} \tag{7.5.5}$$

Assuming that the bolt is threaded in one bolt diameter and the threads start a countersink (or counterbore) distance t_{cb} below the surface to avoid forming a crater lip upon tightening, then the bolt stiffness is approximately (effective length = $D_B/2 + t + t_{cb}$)

$$K_{\text{Bolt}} = \frac{\pi E_B D_B^2}{4\,(D_B/2 + t + t_{cb})} \tag{7.5.6}$$

For steel bolts in cast iron and steel, the stiffness of the flange structure (when the thickness is greater than at least one-half bolt diameter) is always stiffer than the bolt. A rule of thumb is to design to the flange to be at least one to two bolt diameters thick. It is also desirable to make the stress zone cones beneath the bolt heads overlap. This will help minimize straightness errors caused by bolt tightening but unfortunately, can lead to the use of a plethora of bolts.

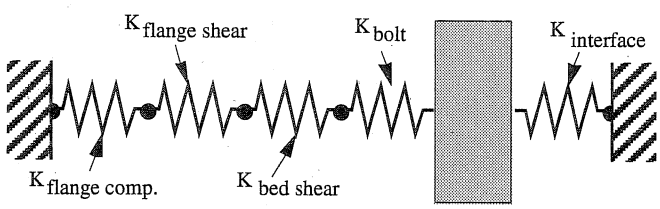

Figure 7.5.3 Spring model of a bolted joint.

As long as the applied load is less then the preload, then the total joint stiffness can be represented by the model shown in Figure 7.5.3. As long as the applied load does not exceed the preload of the joint, a simple force balance[99] shows that the effective stiffness of the bolted joint will be

$$K = K_{\text{interface}} + \cfrac{1}{\cfrac{1}{K_{\text{flange comp.}}} + \cfrac{1}{K_{\text{flange shear}}} + \cfrac{1}{K_{\text{bed shear}}} + \cfrac{1}{K_{\text{bolt}}}} \tag{7.5.7}$$

The joint stiffness decreases with the difference in preload and applied load (as the preload is unloaded) as discussed later. For a balanced design, the joint interface stiffness should equal at least twice the equivalent stiffness of the bolt system. Thus when fully loaded in tension, the preload will be sufficient to maintain the desired joint stiffness. When the flange is not really massive, one can approximate the stiffness of the flange using beam or plate theory (do not forget shear deformations in thick stubby sections) and then use the same type of solution method.[100] In order

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99 See Equations 8.2.1 and 8.2.2.

100 A. Blake, Design of Mechanical Joints, Marcel Deckker, New York, 1986, provides a detailed discussion on modeling of flanged joints.

to determine which are the dominant (least stiff) areas, imagine that the components are made of soft rubber and then apply loads and see how they deform.

Not all joints need this type of analysis, or in some cases the modeling becomes so complex that an experienced bolted joint design engineer is consulted to design the joint. The best way for a novice design engineer to gain experience about bolted joints is to do these types of calculations, observe all the machinery he/she can, and exploit every chance possible to fix or assemble machinery (e.g., his/her car).

Example

Consider a bearing rail shown in Figure 7.5.4. How does one decide how large and how many bolts to use? Ideally, the bearing rail would behave as if it were built into a wall, which would require an infinite number of bolts (too many). The more bolts that are used, the more costly the manufacturing operation, and the weaker the bearing rail gets because it becomes perforated with holes. In order to maintain a balanced design, the stiffness of the bearing rail and the bolted joint system should be the same. With respect to the number of bolts used, in order for the following analysis to be reasonably accurate, it is assumed that the bolt spacing L/γ is on the order of $^1/_2$ to $^1/_3$ of the bearing rail width. With this rule of thumb, there will usually end up being at least two bolts near each bearing pad that rides on the rail. The model of the cross section of the bearing rail is shown in Figure 7.5.4. For the purposes of a back-of-the-envelope determination of what size bolts to use, the following assumptions are made:

- The rail is not as stiff as if it were built into a wall, but on the other hand, it is stiffer than a simply supported section if the bolts behaved like a knife edge pushing down along the entire length of the rail. Thus a simply supported model should be conservative.
- The bolts do not provide knife-edge support; thus the stiffness will be less than modeled. This should be offset by the assumption above.
- When a bearing pad exerts a force on the bearing rail, neighboring sections provide support also, thus lending credence to the knife-edge assumption.
- The surface to which the rail is bolted supports the rail only at the rear point; however, because it actually provides some support due to the bowing of the rail, it will be assumed that the rear support point is infinitely stiff and the surface the rail is bolted to does not provide resistance to bowing.
- Bending and shear deformations must be considered.
- Previous analyses for the stiffness of bolts and flanges are valid.
- The bolts are counterbored, and bolt sizes are to be found for the flange thicknesses equal to 1.0, 1.5, and 2 bolt diameters, respectively. Other geometry assumptions are $D_C = D_B$ and $D_H = 1.5D_B$, $t_{cb} = D_B/4$, and the effective width of the rail between bolt centers is ℓ.
- The contact stiffness of the rail/bed interface is generally much greater than any of these terms; hence it is not included here but just as well should be when using a spreadsheet program.
- The rail and bolt are steel, and the base is cast iron, so $E_f = E$, $E_B = E$, and $E_t = E/2$.

$K = \frac{F_s}{\delta_a} = \frac{F_b(a+b)^2}{a^2\delta_b}$

$K = K_{eq.}\frac{(a+b)^2}{a^2}$

$K_{eq} = \frac{a^2 K}{(a+b)^2}$

Identical (in principle) to the "dog clamp"

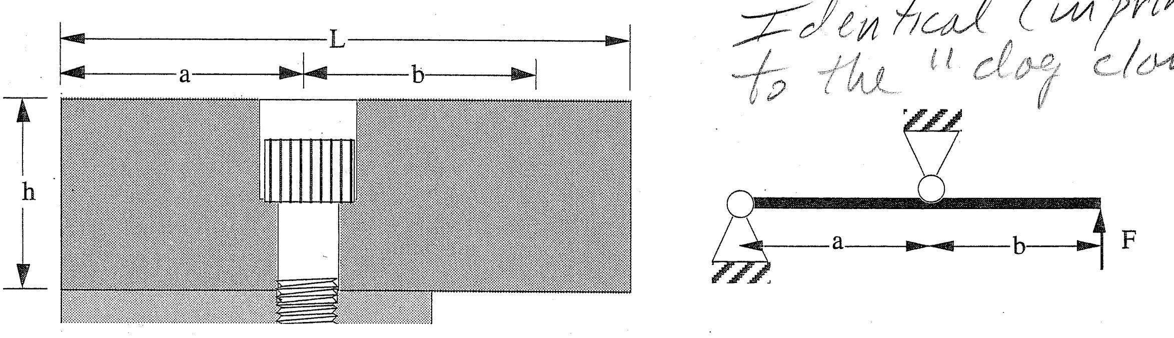
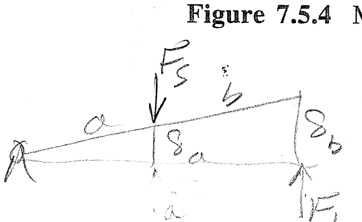
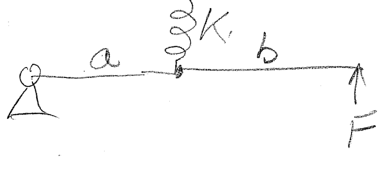


Figure 7.5.4 Model of a bolted rail.

$F_b(a+b) = F_s a$

$\frac{\delta_a}{a} = \frac{\delta_b}{a+b}$





Equations 7.5.1 - 7.5.7 are first evaluated. The results are shown in Figure 7.5.5. The next step is to find the equivalent stiffness at the end of the rail. The system behaves like a spring attached to a fulcrum so the equivalent stiffness is just

$$K_{\text{eq bolt system}} = \frac{Ka^2}{(a+b)^2} \tag{7.5.8}$$

For the remainder of this example, it is assumed that

$$K_{\text{eq. bolt system}} = CD_B \tag{7.5.9}$$

The loading function for the bearing rail is as shown in Figure 7.5.4 is

$$q = F \langle x\rangle_{-1} - F\left(\frac{a+b}{a}\right)\langle x - b\rangle_{-1} \tag{7.5.10}$$

The shear is

$$V = - F \langle x\rangle^0 + F\left(\frac{a+b}{a}\right)\langle x - b\rangle^0 + C_1 \tag{7.5.11}$$

The shear is zero at x = 0, hence C_1 is zero. The moment is

$$M = F \langle x\rangle^1 - F\left(\frac{a+b}{a}\right)\langle x - b\rangle^1\ C_2 \tag{7.5.12}$$

The moment is zero at x = 0, hence C_2 is zero. The slope due to bending is thus

$$\alpha = \frac{F}{EI}\left\{\frac{x^2}{2} - \frac{a+b}{a}\frac{\langle x - b\rangle^2}{2} + C_3\right\} \tag{7.5.13}$$

The deflection due to bending is thus

$$\delta = \frac{F}{EI}\left\{\frac{x^3}{6} - \frac{a+b}{a}\frac{\langle x - b\rangle^3}{6} + C_3 x + C_4\right\} \tag{7.5.14}$$

The deflection is zero at b and a + b; the bending deflection is thus found to be

$$\delta_{\text{bend}} = F\frac{(a+b)\,b^2}{3EI} \tag{7.5.15}$$

Energy methods are used to find the shear deflection. The shear stress for a rectangular beam is given by Equation 2.3.17 ($\tau = V[(h/2)^2 - y^2]/2I$). The strain energy is

$$\begin{aligned} U &= \int \frac{\tau^2}{2G}\,dV = \ell \int \frac{V^2}{8GI^2}\left[\left(\frac{h}{2}\right)^2 - y^2\right]^2 dydx \\ &= \frac{1}{8GI^2}\int_0^{a+b} V^2 \int_{-h/2}^{h/2}\left[\left(\frac{h}{2}\right)^2 - y^2\right]^2 dydx = \frac{\ell h^5}{240GI^2}\int_0^{a+b} V^2\,dx \end{aligned} \tag{7.5.16}$$

Because of the singularity functions, the integral must be done in parts from 0 to b and from b to (a + b). Before integrating, recall that the deflection is given by $\delta_{\text{shear}} = dU/dF$ and the shear was a function of F; hence

$$
\begin{aligned}
\delta_{\text{shear}} &= \frac{\ell h^5}{120GI^2}\int_0^{a+b} V\frac{dV}{dF}\,dx \\
&= \frac{F\ell h^5}{120GI^2}\left[\int_0^b (-1)(-1)\,dx + \int_b^{a+b}\left(-1+\frac{a+b}{a}\right)\left(-1+\frac{a+b}{a}\right)dx\right] \\
&= \frac{F\ell h^5 b\,(a+b)}{120GI^2 a}
\end{aligned}
\tag{7.5.17}
$$

Recall from Equation 7.3.4 that $G=0.5E/(1+\eta)$, and with the beam's moment of inertia $I = \ell h^3/12$, the shear deflection is

$$
\delta_{\text{shear}} = \frac{Fbh^2\,(a+b)\,(1+\eta)}{5aEI}
\tag{7.5.18}
$$

1 not I

The total deflection is thus

$$
\delta_{\text{bean}} = \frac{Fb}{EI}\left[\frac{b(a+b)}{3} + \frac{h^2\,(I+\eta)\,(a+b)}{5a}\right]
\tag{7.5.19}
$$

Substituting $I = \ell h^3/12$ and $\ell = L/\gamma - D_B$, the stiffness of the beam is found to be

$$
K_{\text{beam}} = \frac{5Ea\,[L/\gamma - D_B]\,h^3}{4b\,(a+b)\,[5ab + 3h^2\,(1+\eta)]}
\tag{7.5.20}
$$

Assume the following notation:

$$
A = \frac{5Eah^3}{4b\,(a+b)\,(5ab + 3h^2\,(1+\eta))} \qquad B = \frac{L}{\gamma}
\tag{7.5.21}
$$

The inverse sum of Equations 7.5.20 and 7.5.9 must equal a portion χ of K_{desired} (remember Equation 7.5.7). The factor N indicates how many bolts there are per section of rail that the desired stiffness is required. This results in a quadratic in D_B:

$$
\frac{K_{\text{desired}}}{\chi} = \frac{1}{\dfrac{1}{NK_{\text{beam}}} + \dfrac{1}{K_{\text{eq. bolt system}}}}
\tag{7.5.22a}
$$

$$
D_B^2\,\frac{\chi NCA}{K_{\text{desired}}} + D_B\left[-\frac{\chi NCAB}{K_{\text{desired}}} - NA + C\right] + NAB = 0
\tag{7.5.22b}
$$

where the coefficients of the powers 2, 1, and 0 of D_B are a_{term}, b_{term}, and c_{term}, respectively. The bolt diameter is then found using the quadratic formula. For a balanced design, the interface stiffness should be equal to the bolt and rail stiffness. By making the interface stiffness much larger, the bolt diameter will decrease. However, if the interface stiffness is too high, the displacement caused by the preload will be so small that manufacturing defects may cause loss of preload around a bolt. This will result in a soft spot on the rail. To ensure stiffness is maintained in the presence of maximum tension, the bolts should be torqued to make the interface stiffness equal to $2K_{\text{desired}}(1 - 1/\chi)$. Typically, χ should be 2 but this may yield unrealistic bolt diameters, so a value as high as 4 is not unreasonable. If unreasonable bolt diameters are still obtained, perhaps a longer section of rail should be considered (e.g., N = 2).

	Meters and Newtons			Inches and pounds		
$K_{desired}$	5.26E+08	5.26E+08	5.26E+08	3.00E+06	3.00E+06	3.00E+06
χ	3.00	3.00	3.00	3.00	3.00	3.00
E	2.07E+11	2.07E+11	2.07E+11	3.00E+07	3.00E+07	3.00E+07
N	1	1	1	1	1	1
Rail width/Bolt spacing	1.0	1.0	1.0	1.0	1.0	1.0
Flange thickness	t = 2Db	t = 1.5Db	t = Db	t = 2Db	t = 1.5Db	t = Db
K flange comp/E*D_B	2.530	2.714	3.075	2.530	2.714	3.075
K flange shear/E*D_B	6.973	5.230	3.486	6.973	5.230	3.486
K thread shear/E*D_B	3.486	3.486	3.486	3.486	3.486	3.486
K bolt/E*D_B	0.286	0.349	0.449	0.286	0.349	0.449
C*(a/(a+b))^2	1.29E+10	1.51E+10	1.79E+10	1.87E+06	2.19E+06	2.59E+06
Rail width L	0.150	0.150	0.150	5.91	5.91	5.91
a	0.065	0.065	0.065	2.56	2.56	2.56
b	0.060	0.060	0.060	2.36	2.36	2.36
h	0.050	0.050	0.050	1.97	1.97	1.97
Bolt spacing	0.150	0.150	0.150	5.905	5.905	5.905
D_B (20% thread allowance)	0.019	0.016	0.014	0.741	0.634	0.533

Figure 7.5.5 Spreadsheet results for bolt sizing example.

Typically, L and h are set by a standard rail size. As mentioned before, γ is typically 2 or 3. The bolt diameter D_B is shown for three different flange thicknesses in the spreadsheet output of Figure 7.5.5. An experienced machine design engineer will tell you that these bolt diameters look about right; thus it is possible for a novice to make reasonable back-of-the-envelope calculations to determine bolt sizes. One still needs to make sure that the bolts can be tightened by an amount that will create the necessary joint preload and associated lateral stiffness without causing the rail to deform. Lateral deformations due to the Poisson effect should also be prevented, if the rail is also a guiderail, by maximizing the ratio of rail width to bolt diameter, minimizing the bolt spacing, and minimizing the bolt torque. Also note that if $K_{desired}$ was very high, more than the system could reasonably achieve, an unreal bolt diameter would be obtained. This is why a spreadsheet is nice because it allows you to play with the numbers.

What about the rest of the system? Figure 7.5.6 shows bearing rails that can be subject to balanced or unbalanced preload forces. Which design is best? Would making the bearing rails integral with the casting make for a better design? What must be done to accommodate integral bearing rails? How would the model above have to modified if the keeper rail was used for a hydrostatic bearing application? (See the end of Section 8.7.)

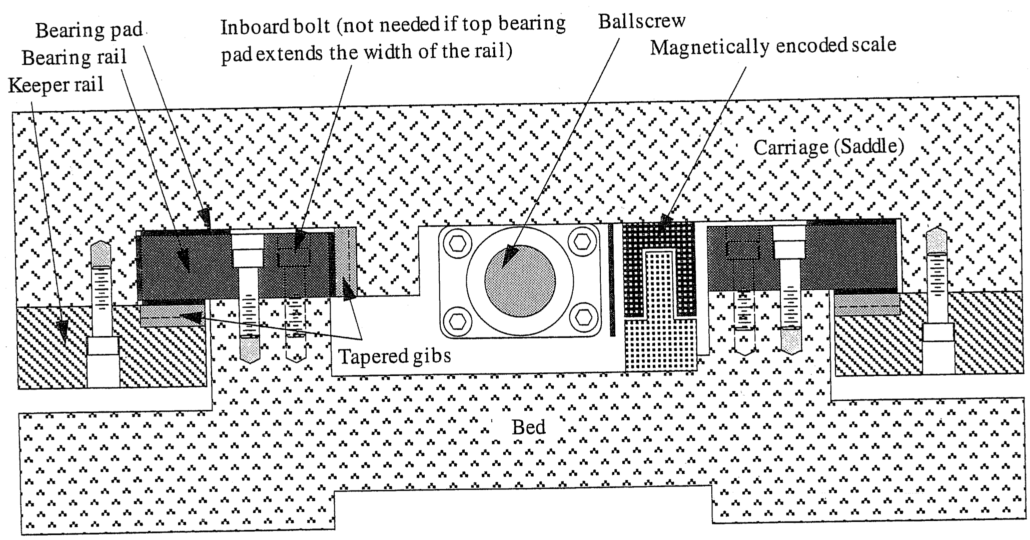


Figure 7.5.6 T-slide bearing configuration.

Compressive Stiffness of Bolted Joints

Ideally, in compression a bolted joint would behave as if the assembly were made from a solid piece of material and calculation of the stiffness would be straightforward; however, due to surface finish variability and imperfections, this is not the case. The finer the surface finish and the tighter the preload, the fewer and smaller the gaps between surface asperities on the surfaces. Hence where compressive stiffness is critical, surfaces should be ground, scraped, or lapped. Vibration annealing can help to seat the asperities. Grouting the joint with an adhesive can fill both large and micro voids and greatly increase joint stiffness as well as increase the damping. The joint may even be designed so it has only three height adjustable contact points (e.g., tapered gibs, tapered horizontal screw threads, etc.). After the entire structure is aligned, the joints can then be grouted with an appropriate material (e.g., cement or epoxy).

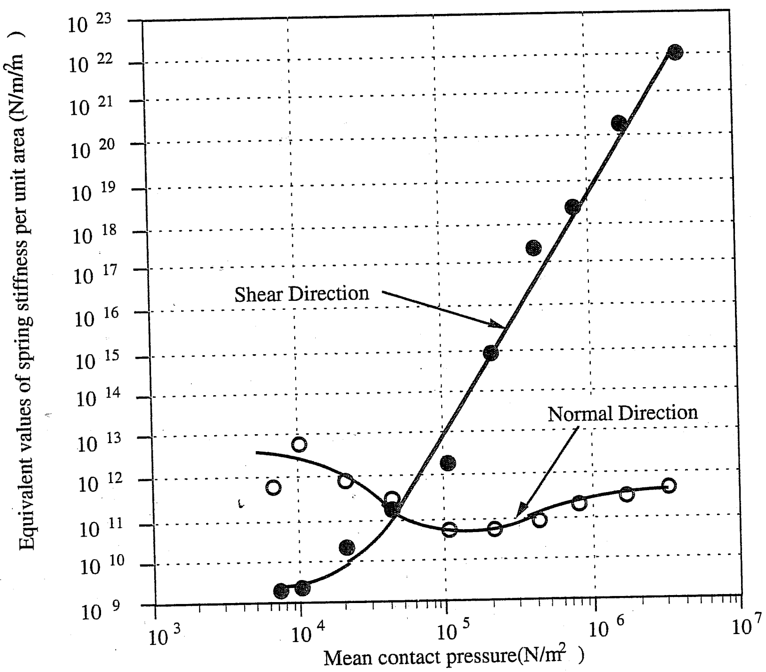


Figure 7.5.7 Joint stiffness of 0.55%C ground steel parts. (After Yoshimura.)

In order to estimate joint stiffness, one can consult experimental data as shown in Figures 7.5.7 through 7.5.9. Figure 7.5.7 shows joint stiffness as function of the contact pressure between ground steel specimens[101] up to relatively high contact pressures. Figure 7.5.8 shows the effect of contact pressure on the damping properties of the joint. Figure 7.5.9 shows a joint's compressive stiffness as a function of contact pressure[102] for lower contact pressures. Often lower contact pressures are typically found in components that must be assembled with minimal preload to prevent deformation. One reason to space bolts not too far apart from each other or the boundaries of the contact surface (e.g., not farther than the rail thickness) is so that the joint pressure can be estimated as the total force provided by all bolts divided by the total joint area. The joint stiffness would thus be the product of the stiffness per unit area value obtained from either Figure 7.5.7 or 7.5.9 and the joint area.

101 From M. Yoshimura, "Computer Aided Design Improvement of Machine Tool Structure Incorporating Joint Dynamics Data," Ann. CIRP, Vol. 28, 1979, pp. 241-246. The test specimens were made of 0.55% carbon steel and the surfaces were ground and coated with a light machine oil.

102 From M. Dolbey and R. Bell, "The Contact Stiffness of Joints at Low Apparent Interface Pressures," Ann. CIRP, Vol. 19, pp. 67-79.

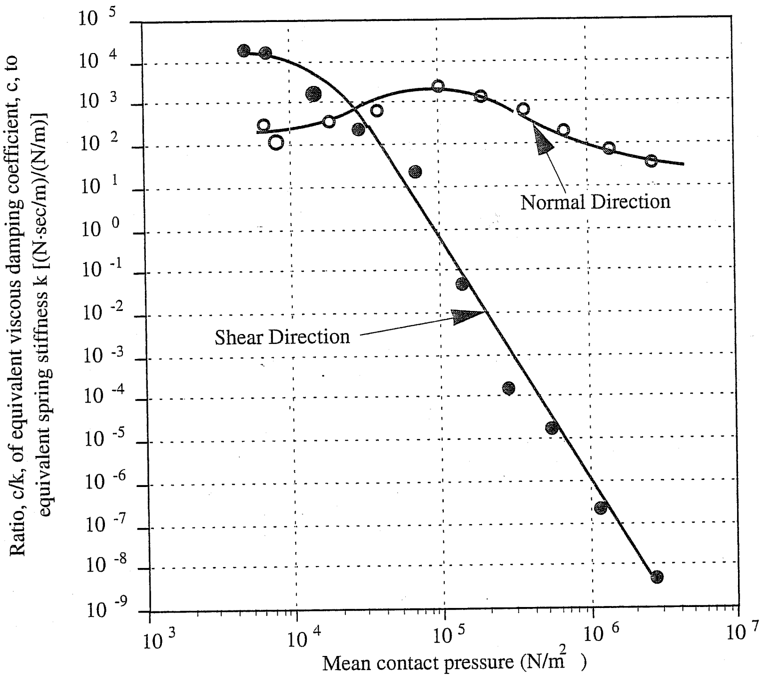


Figure 7.5.8 Joint damping of 0.55%C ground steel parts. (After Yoshimura.)

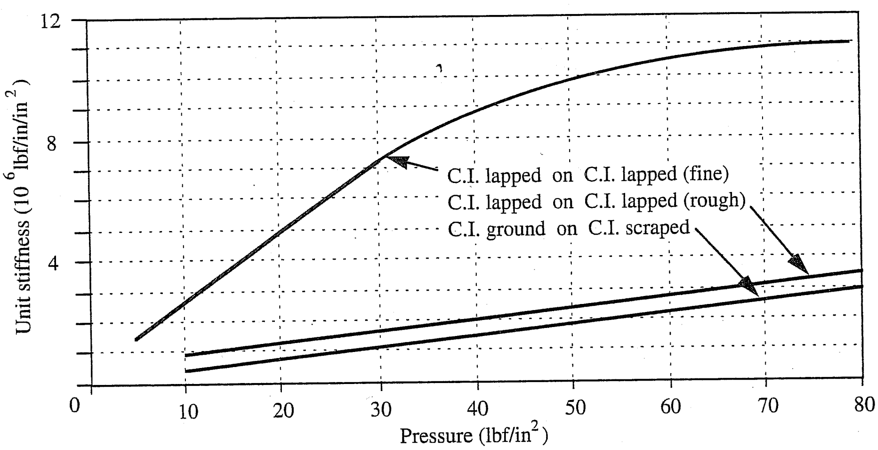


Figure 7.5.9 Compressive stiffness at lower contact pressures (after Dolby and Bell).

Lateral Stiffness of Bolted Joints

The lateral stiffness of a bolted joint is also a function of the surface finish and the preload on the joint. The higher the preload, the greater the lateral force that can be resisted by the coefficient of friction. Lateral stiffness seems to be a function of how the surface asperites interlock when the joint is preloaded, and how they subsequently bend and shear when lateral forces are applied to the joint. Attempts at modeling the asperites as a random distribution of peaks and valleys have yet to prove satisfactory, and empirical data is best used, such as shown in Figures 7.5.7 or 7.5.9. If very high lateral stiffness is required from a bolted joint, it may be appropriate to key or pin the joint after the structure has been aligned. For the former, a loose-fitting key is grouted or epoxied in place as a final manufacturing procedure. For the latter, the dowel pin holes are drilled and reamed into the bed using the finally positioned rail as a template.

Preload[103]

When a torque Γ is applied to a bolt with lead ℓ ($1/\ell$ threads per unit length), the axial force generated is

103 See J. Bickford, "Preload: A Partially Solved Mystery," Mach. Des., May 21, 1987.

$$F = \frac{2\pi\Gamma_{e}}{\ell} \tag{7.5.23}$$

For lubricated threads, the efficiency e can vary from 0.2 to 0.9, depending on the surface finish and accuracy of thread mating.[104] In addition to the efficiency of the threads, one must consider the friction μ under the bolt head. Hex socket cap screws are most often used in machine tools, and the bolt head diameter is typically 1.5 times the bolt diameter. Equating the work in to the work out, the axial bolt force as a function of torque is

$$F = \frac{4\pi\Gamma}{\frac{2\ell}{e} + 3\pi D_B \mu} \tag{7.5.24}$$

One should assume that the coefficient of friction will be 0.3 normally, and 0.1 if the bolt is lubricated and vibration stress relieved during final tightening.

The tensile stress in the bolt shaft is $\sigma = 4F/\pi D_B^2$, and the shear stress in the threads is $\tau = F/\pi D_B L$ where L is the thread engagement length. Whether the bolt or threads fail depends on the relative strength of the bolt and thread material, and the thread engagement length L. Note that if L is greater than one or two thread diameters, then due to imperfections in thread geometry all the threads cannot engage anyway. Thus it does not make sense to specify tapping holes for more than two bolt diameters of usable thread. If a bolt is to be removed from the part periodically, and the bolt must carry a high load and/or preload, then one should consider specifying hardened-steel coiled-thread inserts. In addition, female threads should never be specified in a material that is harder than about RC30, because the sharp thread corners are prime stress concentrators, which can lead to failure.

When many bolts are used on a joint, uniform bolt preload is necessary to minimize distortion of the part and maintain consistency of performance from machine to machine. Special bolts have been developed that have an indicator on the head that lets the bolt tightener know when the proper bolt tension has been achieved. For inexpensive common bolts, it is often desirable to minimize the coefficient of friction and its variability. This can be done in the following manner:

- The tapped threads should start $^1/_4$ bolt diameter below the surface to avoid raising the surface and causing a crater lip to form should the bolts be overtightened.
- Inspect bolts to make sure that the thread forms have a good surface finish and are well formed. If necessary, buff the threads. Avoid the use of cheap bargain basement bolts with poorly formed threads.
- Thoroughly clean and degrease the male and female threads, counterbore seat, washers, and the underside of the bolt head. Thoroughly lubricate the parts and make sure that the bolt threads easily into the hole.
- Make sure that the surface finish of the counterbore seat and the underside of the bolt head are smooth and flat. For critical assemblies where all bolt preload forces must be the same, place a superfinished steel washer in the counterbore (finished side up) and a lubricated superfinished brass washer (finished side facing the superfinished steel washer) on top of it, then insert the bolt. The washers will act as a thrust bearing to minimize friction at the head.
- For permanent joints or joints subject to vibration, use a thread lubricant that hardens after several hours. Under shock and vibration loads, bolt preload may be reduced by a factor of 2 after only a few hours of service if a locking mechanism is not used.
- Tighten bolts in an incremental sequence. Vibration anneal the assembly, and then retighten the bolts. Repeat this process until the bolts do not require further retightening.
- If possible, pin the assembly.
- Perform one final retightening of the bolts while the structure is being vibration stress relieved. The dither action provided by the vibration anneal process can help to overcome stick-slip friction in the bolt threads and head-to-counterbore interface.

The preload force, area, and surface finish determine the interface pressure between the parts of a joint, which affects the stiffness across the joint as discussed above. High bolt preloads are often necessary for sealing pressurized joints with gaskets, or where high alternating stresses must be made a small proportion of the total stress state in the bolt. For machine tool joints where the

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104 See Section 10.8.3.1 for a detailed discussion of the calculation of the efficiency based on thread geometry and the coefficient of friction.

bolts are not subject to high alternating stresses, the preload should be low and more bolts used to achieve the total desired force and joint interface pressure. This will also help to distribute the force over the joint, thereby minimizing deformation of the joint. To minimize warping, compression, or Poisson expansion of the parts that make up the joint, a rule of thumb is to make the bolt spacing on the order of the thickness of the part or flange.

Remember, the tighter the bolt, the larger the local deformation (dishing) around the bolt, which can deform critical components such as bearing rails; however, the looser the bolt, the lower the stiffness of the joint. Note that bolt diameter can be increased to keep stresses low (<10-25% of yield) to help maintain long-term dimensional stability of the system,[105] and larger bolt diameters increase the stiffness of the bolts. The following guidelines can be used to determine the required preload and bolt size for a joint.

- From experimental data (e.g., Figures 7.5.7-7.5.9) determine the required joint interface pressure to achieve the required stiffness, and the required force on the joint area to achieve the interface pressure. The stiffness of the joint should be greater than the stiffness of the parts that make up the joint, as discussed in the context of Equation 7.5.22. Assume that the bolt force acts over an area of four to five bolt diameters (the area of the cone of influence).
- Determine the maximum tensile force exerted on the joint. This force would lower the preload on the joint and thus decrease joint stiffness. The minimum total preload force that must be exerted by the bolts will be the larger of the sum of the maximum tensile force and the required joint preload force or four times[106] the maximum tensile force.
- Make sure that the product of the minimum force and the coefficient of friction for the joint are at least a factor of 5-10 greater than the maximum expected shear force on the joint. If the joint is to also be pinned, then this may not be necessary.
- Space the bolts according to one of the rules of thumb given above (i.e., bolt spacing equal to the part thickness or a portion of the width of the bearing rail).
- Size the bolts so that the stresses in the bolts are below 25% of yield, and compare this value to the bolt size found from a minimum stiffness criteria (as shown in the example). Check the bending, compression, and Poisson expansion of the rail (see next section) to make sure that they are within acceptable limits.

	Meters and Newtons		Inches and pounds	
D_B	0.016	0.016	0.625	0.625
Area	0.0002	0.0002	0.307	0.307
$4D_B$ area	0.0030	0.0030	4.602	4.602
Lead	0.002	0.002	0.08	0.08
Thread friction	0.1	0.3	0.1	0.3
Efficiency	0.28	0.11	0.29	0.12
Kdesired	5.26E+08	5.26E+08	3.00E+06	3.00E+06
Joint pressure (from Figure 7.5.7)	4.00E+04	4.00E+04	6	6
Joint pressure (from Figure 7.5.9)	1.38E+05	1.38E+05	20	20
Force required (Ground on scraped cast iron)	416	416	92	92
4xAlternating force on carriage	40000	40000	4494	4494
Number of bolts under carriage	8	8	8	8
Total force per bolt	2916	2916	654	654
Torque required	6.9	18.8	60.6	164.9
Shear stress	8.52E+06	2.33E+07	1263	3440
Tensile stress	1.45E+07	1.45E+07	2131	2131
Mises equivalent stress	2.07E+07	4.29E+07	3054	6328

Figure 7.5.10 Bolt loads for the example of Figure 7.5.5 ($\chi = 3.00$).

As an example, consider the bolted rail designed in Figure 7.5.5. Figure 7.5.10 summarizes the selection of the torque levels to be used. As shown, the primary design consideration is the alternating force level, as only a very low bolt torque is required to attain the desired stiffness. Another reason for using a minimum bolt torque would be to ensure that the bolts do not loosen during the life of the machine.

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105 To maximize fatigue strength and consistence of the preload force applied, it has been proposed that bolts should be tightened to the point where they begin to yield. For joints that seal or only require micron accuracy, this method is useful; however, data on long-term dimensional stability of yield-tightened joints is not yet available. For more information on the yield tightening process, see J. Monaghan and B. Duff, "The Effects of External Loading on a Yield Tightened Joint," Int. J. Mach. Tools Manuf., Vol. 27, No. 4, 1987.

106 This will help to maximize the fatigue life of the bolt. Note that one should check company policy regarding design of fatigue-loaded bolted joints, as a value different than 4 may be used.

Rail Deformation Due to Preload

Even when a "perfect" rail is bolted to a "perfect" bed, significant deformations can result in the form of bending, compression, and Poisson expansion. These deformations can be seen in the way that light is reflected off a precision-bolted bearing rail. Unfortunately, it is not always possible to finish the rails after they are bolted in place, so it is necessary to estimate the amount of deformation so that it can be minimized beforehand. Also, if the rails were finished while bolted and then transferred to another machine, it would be desirable to be able to determine the amount of deformation that could result from a variation in the bolt force.

Two cases will be considered here. The first case is where the counterbore diameters are small with respect to the bearing rail width b_{rail}, and thus the bearing rail can be modeled as a beam on an elastic foundation, as shown in Figure 7.5.11. The second case is where the bolt holes take up a significant part of the beam cross section and thus the other model shown in Figure 7.5.11 is used. Both cases assume that the beams (rails) are preloaded, so in effect the foundation (bed) can exert downward as well as upward forces on the beams. For the first case, the relative lateral deflection is found from Roark:[107]

$$\delta = y_A \left\{ \text{Cosh}\frac{\beta\ell}{2}\cos\frac{\beta\ell}{2} - 1 \right\} + \frac{M_A \,\text{Sinh}\frac{\beta\ell}{2}\sin\frac{\beta\ell}{2}}{2EI\beta^2} \tag{7.5.25}$$

where

$$y_A = \frac{-F}{4EI\beta^3}\left[\frac{C_2 C_{a1} + C_4 C_{a3}}{C_{14}}\right] \qquad M_A = \frac{F}{2\beta}\left[\frac{C_2 C_{a3} - C_4 C_{a1}}{C_{14}}\right] \tag{7.5.26}$$

and

$$\beta = \left(\frac{b_{bed}k}{4EI}\right)^{1/4} \qquad I = \frac{b_{rail}\, h_{rail}^3}{12}$$

$$C_2 = \cosh\beta\ell \sin\beta\ell + \sinh\beta\ell \cos\beta\ell \qquad C_4 = \cosh\beta\ell \sin\beta\ell - \sinh\beta\ell \cos\beta\ell \tag{7.5.27}$$

$$C_{a1} = \cosh\frac{\beta\ell}{2}\cos\frac{\beta\ell}{2} \qquad C_{a3} = \sinh\frac{\beta\ell}{2}\sin\frac{\beta\ell}{2}$$

$$C_{14} = \sinh^2\beta\ell + \sin^2\beta\ell$$

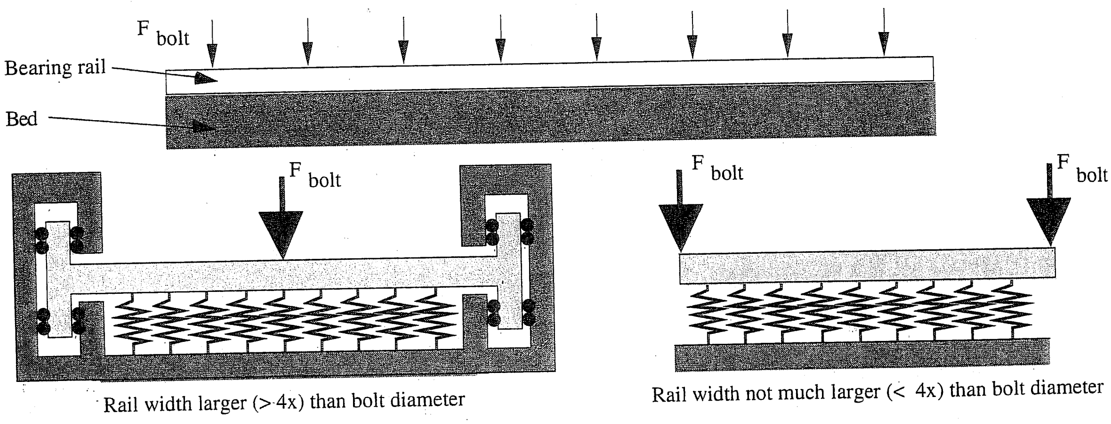

Figure 7.5.11 Model of rail deformation due to bolt preload. When the counterbore diameter is much less than the rail width, the model of a guided beam on an elastic foundation of modulus k is used. When the counterbore diameter is a significant portion of rail width, the simply supported beam model on the right is used.

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107 R. J. Roark and W. C. Young, Formulas for Stress and Strain, 5th edition, McGraw-Hill Book Co., New York, 1975, p. 134.

In order to estimate the foundation modulus k (stiffness per unit area, hence it has units of N/m^3), consider the following. The bolt pulls down on the rail and up on the bed. If the bed's moment of inertia is much greater (e.g., >10) than that of the rail, then bending effects of the bed can be ignored. For a precision machine this should be an accurate assumption. The deeper the section of the bed, the stiffer it is in bending, but the more compliant it is in compression. If the bolt extended to the bottom of the bed, then it would be easier to make an assessment of k. Assume that the depth of the section is that which makes the bed moment of inertia 10 times greater than that of the rail. Assuming that the bed width is b_{bed} and the rail width is b_{rail}, the effective depth h_{bed} of the bed is

$$h_{bed} = h_{rail}\left(\frac{10\ b_{rail}}{b_{bed}}\right)^{1/3} \tag{7.5.28}$$

The foundation modulus k for the bed is thus

$$k\ \frac{E}{h_{rail}\left(\frac{10\ b_{rail}}{b_{bed}}\right)^{1/3}} \tag{7.5.29}$$

In many instances where a "standard" type of bed is used, simple experiments could be performed to evaluate k. This value could then be used with the rail deflection equations to scale new designs. For the second case, modeled in Figure 7.5.11, Timoshenko[108] gives the relative deflection between the ends and the middle to be

$$\delta = \frac{2F\beta\left[\cosh\beta\ell + \cos\beta\ell - 2\cosh\frac{\beta\ell}{2}\cos\frac{\beta\ell}{2}\right]}{b_{bed}\ k\ (\sinh\beta\ell + \sin\beta\ell)} \tag{7.5.30}$$

This equation would be used, for example, when bolting down modular recirculating rolling element linear bearing rails (e.g., linear guides). Note that these solutions do not consider shear deformations. An analysis that took into account shear deformations would be too complex for general application; however, a conservative engineering estimate is to assume that the shear deformations are on the order of the bending deformations, because in general this type of rail is fairly stocky and the bolt spacing is usually equal to an amount less than the rail width. For the previous example, the foundation modulus (k) is 2.7529E + 11N/m^3, and 2 × Equation 7.5.25 (to account for shear deformations) is 0.152 μm (6 μin.). For other geometries and loads, deflections obtained from the theory presented above correlates reasonably well with experimental data provided by Levina.[109]

Even if the subgrade were infinitely rigid, the bolts would still compress the metal in the rails. The result would be low regions near the bolt heads and high regions between them. This is referred to here as compressive deformation of the rail. To estimate the compressive deformation, use the area of the rail encompassed by three to five bolt diameters.[110] In addition, recall the Poisson effect given by Equations 7.3.3. Hence the compressive deformation is accompanied by a lateral expansion that is on the order of 0.3 times the compressive deformation (for most metals), depending on the rail geometry. For large rails such as shown in Figure 7.5.1, the Poisson expansion is usually negligible because it is diffused into the surrounding metal. For thin rails, such as those used by many types of modular linear bearings, the effect can be more prominent, which is why manufacturers' bolt-tightening recommendations should be carefully followed.

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[108] S. Timoshenko Strength of Materials, Part II, 3rd ed., Robert E. Krieger Publishing Co., Melbourne, FL, p. 17. Note that "Timoshenko's k" is the stiffness per unit width whereas Roark's k (and k given by Equation 7.5.29) is the stiffness per unit area, and hence Equation 7.5.30 has the term "$b_{bed}k$" where k is defined by Equation 7.5.29.

[109] See Z. M. Levina, "Research on the Static Stiffness of Joints in Machine Tools", Proc. of the 8th Int. Mach. Tool Des. and Res. Conf., Sept. 1967, pp. 737-758.

[110] It would be nice to see detailed graphs of compressive and Poisson deformations caused by bolts in various-size rails. Such graphs would probably have to be generated using finite element methods.

Since bolt torques and thread efficiencies are never exactly the same, varying vertical and horizontal straightness errors in the rail will often be present with a period equal to that of the bolt spacing. Bolt preload, thread friction, and tightening methods should all be chosen with care when designing and manufacturing a precision machine.

Pinned Joints

Bolts cannot transmit shear loads because their clearance holes are too inaccurate to allow for a tight fit in a joint that may have many bolts. If a bolted joint were subject to a high shock load or sustained vibration, it is possible that the joint may slip. A tight-fitting solid pin of length L_{pin} in a hole bored through the joint can prevent this from occurring, as well as greatly increase the lateral stiffness of a bolted joint. Far away from the joint interface at the pin tip, the area A_{part} of the part that bears the shear load is equal to the cross section of the part. At the joint interface the pin's cross-sectional area A_{pin} supports all the shear load. Assuming a symmetric joint with linearly varying area, the shear stress as a function of position in the joint is

$$t = \frac{F}{A_{part} - \left(\frac{A_{part} - A_{pin}}{L_{pin}/2}\right) y} \tag{7.5.31}$$

Steel pins are almost always used, regardless of whether the parts are made from steel, cast iron, or aluminum. A conservative assumption is thus to assume that the shear modulus of the pins and parts are equal. Thus the stiffness of the region from the top of the pin to the bottom of the pin is

$$K = \frac{F}{\delta} = \frac{F}{2\int_0^{L_{pin}/2} \frac{\tau}{G}\, dy} = \frac{G\,(A_{part} - A_{pin})}{L_{pin}\, \mathrm{Log}_e\,(A_{part} / A_{pin})} \tag{7.5.32}$$

Note that in the limit as $A_{part} \Rightarrow A_{pin}$, $K \Rightarrow GA_{part}/L_{pin}$.

Because the desired amount of lateral stiffness can often be obtained with bolt preload, often only two pins are used to maintain alignment should the joint be subject to vibration. If the joint is subject to shock loads and it is not desired for the joint to give, more pins may be needed to prevent deformation around the pins. After drilling and reaming a hole through both parts (after they are bolted and aligned), the pins can be pressed in. If desired, the first part can be reamed slightly larger, so the pin is a tight press fit in one part and a light press fit in the other part.[111] In this manner, if the parts are separated, the pins always reside in the same part. Care must be taken to make the pin hole 25% deeper than the pin to allow room to compress air in the bottom of the blind holes, or to grind a small flat on one side of the pin to allow the air to escape. Ideally, the hole should be a through hole, so the pin can be knocked out if required at a later date.

Solid steel pins (dowels) provide the greatest shear stiffness; however, they require the holes to be reamed, which is an extra manufacturing step. A roll pin is a round, hollow, hardened steel pin with an axial slit in one side. When pressed into a drilled hole that is a few percent smaller than the pin, the slit width is forced to decrease, thereby allowing the roll pin to fit the hole perfectly. Roll pins are well suited to general manufacturing applications or where very high shear stiffness is not needed (one just wants to locate and hold the parts for assembly or to resist light service loads). It is also less expensive, for example, to roll pin a part (e.g., a handle) to a shaft, then cut a keyway. Also, a roll pin will never loosen the way a setscrew can.

Stability of Bolted Joints

Most common joints that rely on mechanical contact between parts (i.e., not bonded or welded joints) may suffer a loss of initial preload with applied cyclic stress. This loss of preload seems to be due to the applied stress, causing microslip between the surfaces. For maximum dimensional stability, the design engineer may wish to specify vibrational stress relief and retightening of bolts, with the final tightening done while the vibration stress relief process is being done. In addition, if the joint is to be subject to vibrational loads or extreme stability is required, a hardening thread lubricant should be used.

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111 One can calculate the desired press fit, or use standard tables provided in Machinery's Handbook, Industrial Press, New York. For large sections where there is obviously no danger of splitting the section, it is easier to do the latter.

For example, when a bearing rail is bolted in place, the bolts and threads should be prepared as discussed above, and then assembly should proceed in the following manner:

- The bearing rail should be aligned and the bolts incrementally tightened according to a specific pattern (e.g., from the inside out to prevent clamping in a bow) until the desired bolt torques are reached. The alignment should be checked after each tightening increment.
- The assembly should be vibration stress relieved.
- The alignment should be checked and the bolts retightened.
- The alignment should be checked, and then the assembly vibration stress relieved while the bolts are retightened.
- A final checking of the alignment should be performed.
- Ideally, the rail would be pinned to the structure with dowels and also potted in place with an epoxy, which helps increase joint stability and damping. As an alternative to pinning, a loose-fitting key that runs the length of the rail can be grouted or epoxied in place as the final assembly procedure.

The vibratory procedure accelerates the wear-in period of the permanent joints and helps prevent the machine from changing after the customer gets it. When properly implemented along with control of the maximum stress levels caused by the bolting process, the stability of a joint bolted by this process will probably be as good as one can get it. Most machine tools do not require the use of such an elaborate bolting procedure; however, as nanometer performance is sought, this type of procedure may become more common.

Bolting hardware[112]

For the machine design engineer, there are two basic types of bolts used to hold together structures: hex head bolts and socket head cap screws. The former are tightened with an open-end wrench, and the latter are tightened with a hex key (Allen) wrench. Handbooks give all necessary geometrical dimensions, such as head, body, and thread dimensions and required wrench clearances. There are also a number of other head configurations available, including decorative and tamperproof types. One need only consult a catalog of fastener companies to be overwhelmed with all the various types of fasteners that are available.

From an ergonomic viewpoint, it is wise never to specify the use of a bolt with a diameter smaller than 4-6 mm (No. 8-1/4 in.) and on heavy equipment 10 mm (3/8 in.) to avoid having the bolt head twisted off by overzealous bolt tighteners that may be tightening other larger bolts in the neighborhood. The best way for a design engineer to develop an intuition for "what bolt is right" is to take apart old machinery and fix old cars (e.g., rebuild an engine or change the rear end in an old truck). Remember, education teaches best those who teach themselves.[113]

Most machine tool applications use threads cut into one of the parts, but some applications require the use of nuts. As with bolts, there are many different types of nuts that are available to provide many different functions through special features such as:

- Special shapes to minimize the stress state in the threads to maximize fatigue life.
- Nonround threads and inserts to provide a locking effect.
- Special geometries (e.g., convex, serrated) on the seating face to enhance clamping action of the bolt on the parts.

For machine tool applications, a plain nut or a nut with a nylon insert to prevent loosening is most often used. The specific choice is usually a matter of company standards.

There are two basic types of washers used with bolts: flat washers, which prevent the surface from being marred by the bolt head and enhance load distribution on thin parts, and lock washers, which help maintain constant preload and prevent loosening of a bolt subject to vibration. In counterbored holes, flat washers are not used because the counterbore is only slightly larger than the bolt head. Split lockwashers are sometimes used with counterbored bolts; however, often a hardening thread lubricant is used instead. A split lockwasher is compressed by the bolt

112 The next three sections discuss bolts, nuts, and washers. For illustrative examples of different types, consult Machinery's Handbook, Industrial Press, New York, or a Thomas Register of American Manufacturers, Thomas Publishing Co., New York, which often shows informative pictures in the company listings.

113 Translated for students in contemporary educational institutions, take off those damn headphones and observe the world around you. Signs, bridge girders, construction machinery, all have dozens of fascinating bolted joints to be looked at and analyzed in your head.

and thus can help maintain preload, even in the presence of bolt length changing, due to creep of highly stressed thread.

7.5.2 Adhesive Joints[114]

Adhesive joints are fast becoming the most preferred type of joint in assemblies ranging from consumer products to airplanes. Bonded joints can be strong and fatigue resistant, and the process by which they are made is far more automatable than that used for almost any other joint. Virtually anything can be bonded to anything to perform in any type of condition desired. As shown in Figure 7.5.12, adhesives fill small voids between mating parts and thus can dramatically increase strength, stiffness, damping, and heat transfer characteristics of the joint. Even when two ground surfaces are to be bolted together, one can specify the use of a few drops of low viscosity adhesive that will flow and fill the small voids.[115] Some manufacturers have found that a sliding fit joint held by an adhesive (e.g., Loctite™ Retaining Compound) makes a more dimensionally stable, accurate, and long-lasting joint than does a press-fit joint. This is particularly true for precision mechanisms (e.g., index tables and computer disk drives). Bearing races can also be fixed to bores in this manner to avoid altering the preload by press fitting.

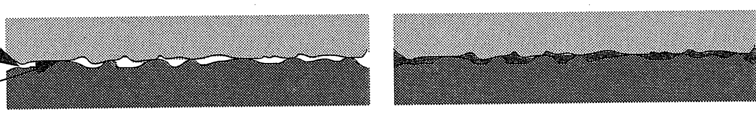

Figure 7.5.12 Most joints typically make contact over 30% of the surface area. An appropriate adhesive can fill gaps between mating parts, thereby increasing strength, stiffness, and damping. [From Guide to Adhesive Selection and Use (LT-1063), courtesy of Loctite Corp.]

As with bolted joints, there is a plethora of information available on adhesive joints; thus only adhesive processes for machine tools will be discussed here, such as bonding, potting, and replication. In all cases, joint preparation is of prime importance, and the user should be careful to follow the manufacturer's recommendations.

Bonding

Bonding quite simply entails sticking one piece of material to another using an adhesive. The goal is to design the best geometry for the joint and choose the correct adhesive. Volumes of text exist on this subject, and adhesive manufacturers are usually happy to provide design assistance. As with other joints on machine tools, bonded joints are usually designed for stiffness, not strength. If the layer is thin enough, then the effective stiffness of the adhesive layer will often be much greater than that of the part itself. Because they act as continuous media for forming a joint, adhesives used by themselves or in conjunction with bolts are a must in every design engineer's tool kit.[116] A typical adhesive polymer (of the epoxy family) for machine tool use will have properties similar to the ones shown in Tables 8.2.2 and 8.2.3.

Bonding does not always mean that an isotropically rigid joint must be formed. For example, in Section 1.6 the coordinate measuring machine described had an aluminum frame and steel measuring scales. In this case it was necessary to prevent the aluminum frame from stretching the scales when the temperature the machine was used at was different than that at which it was assembled, or to prevent damage during shipping. To do this, one end of the scales was pinned to the aluminum frame, and the rest of the scale was bonded along its length using an adhesive that had excellent peel resistance, but low shear resistance. This type of joint can be used when thermal property mismatch between materials or temperature gradients can be expected to cause differential thermal expansion between components, and one wants to hold the two components together with constant spacing while still allowing the components to slide relative to

114 For a more detailed discussion of adhesive joints, see, for example, A. Blake, Design of Mechanical Joints, Marcel Dekker, New York, 1985, and M. Kutz (ed.), Mechanical Engineers' Handbook, John Wiley & Sons, New York, 1986.

115 A quantitative study of how much adhesives help stiffness and damping as a function of surface finish and joint preload was not available at the time of this writing.

116 See, for example, M. Chowdhary, M. Sadek, and S. Tobias, "The Dynamic Characteristics of Epoxy Resin Bonded Machine Tool Structures," Proc. 15th Int. Mach. Tool Des. Res. Conf., 1975.

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