Precision Machine Design

Topic 13

Design of joints and bearing rails

Purpose:

The characteristics of joints between a machine's components are strong functions of the components themselves, and the means by which they are joined. The latter is often thought of as being non-deterministic, but as this lecture illustrates, design and manufacturing methods can make joint design more deterministic.

Outline:

- Bolted joints
- Bearing rail design
- Adhesive joints

"Tis not a lip, or eye, we beauty call,
But the joint force and full result of all"

Alexander Pope
Bolted joints

- A necessary evil in most machines.

  - Bolts act as point forces that create localized deformations in the material.

  - The goal in precision bolted joint design is to minimize variation in deformation between the bolts.

  - This is accomplished by carefully designing the geometry of bolt placement and the tightening method used.

- Remember, most of the damping in a machine tool comes from the bolted joints (and sliding bearings):

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<th>Bed + Spindle</th>
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<th>Machine Spindle</th>
<th>Complete Spindle</th>
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Mechanics can be used to model bolted joints

- Many elements in a bolted joint act in series, which as a set acts in parallel with the stiffness of the parts' interface:

- As long as the applied load does not exceed the preload of the joint, the effective stiffness of the bolted joint will be:

\[ K = K_{\text{interface}} + \frac{1}{K_{\text{flange comp.}}} + \frac{1}{K_{\text{flange shear}}} + \frac{1}{K_{\text{bed shear}}} + \frac{1}{K_{\text{bolt}}} \]

- Including the effects of part stiffness (e.g., a bearing rail), the total stiffness of the system is:

\[ K_{\text{system}} = \frac{1}{K_{\text{part}}} + \frac{1}{K_{\text{Interface/bolt}}} + \frac{1}{K_{\text{Flange comp}}} + \frac{1}{K_{\text{Flange shear}}} + \frac{1}{K_{\text{Bed Shear}}} + \frac{1}{K_{\text{Bolt}}} \]
• The stiffness of a 45° cone of material under the bolt head is approximately:

\[
K_{\text{flange comp.}} = \int_0^t \frac{1}{\pi E_f \left( \frac{D_H}{2} + y \right)^2 - \frac{D_C^2}{4}} \, dy = \frac{\pi E_f D_C}{\log_e \left( \frac{D_C - D_H - 2t}{D_C + D_H} \right) \left( D_C + D_H + 2t \right) \left( D_C - D_H \right)}
\]

• The shear deflection of the flange of thickness \( t \) due to the bolt being in tension is found using energy methods:

\[
\tau = \frac{F}{2\pi R t}
\]

\[
\delta = \frac{\partial U}{\partial F} = \frac{F}{2\pi t G} \int_{R_B}^{2R_B} \frac{dR}{R} = \frac{F \log_e 2}{2\pi t G}
\]

\[
K_{\text{flange shear}} = \frac{\pi t E_f}{(1 + \eta) \log_e 2}
\]

• From geometric compatibility with the bolt head, this includes the effect of shear strain in the bolt head.
• Assume that the effective length of thread engagement is equal to $D_B$.

* The shear stiffness of the threaded region in the bed (neglecting the countersunk region) is:

\[
K_{\text{bed shear}} = \frac{\pi D_B E_t}{(1 + \eta) \log_2 2}
\]

• Assume the bolt is threaded in one bolt diameter.

• The threads start a distance $t_{cb}$ below the surface to avoid forming a crater lip upon tightening.

• Then the bolt stiffness is approximately (effective length = $t + DB/2 + t_{cb}$):

\[
K_{\text{Bolt}} = \frac{\pi E_B D_B^2}{4 (D_B / 2 + t + t_{cb})}
\]

• An example of the application of these formulas is given in conjunction with the discussion of bearing rail design.
Shear Resistance of Bolted Joints

- Bolted joints resist shear ONLY by clamping action and friction!
- NEVER rely on a bolt to take a shear load
- UNLESS the bolt is a shoulder bolt!

Find the center of the bolt pattern, and compute the moments about it, using the products of bolt force, friction, and distance from the center point:

\[ F_{\text{max}} = \frac{\sum_{i=1}^{\text{number of bolts}} F_{\text{bolt}_i} r \mu}{L} \]
• The amount of force a bolt can generate, given friction in the threads, is determinable from basic physics of the force generated by a wedge\(^1\):

\[
\begin{align*}
\text{Thread angle } \alpha & \quad \text{Lead angle } \theta \\
\text{Force screw shaft thread} & \quad \text{applies to the nut thread}
\end{align*}
\]

Where:

• \( \Gamma \) is the applied torque
• \( \mu \) is the coefficient of friction
• \( r \) is the pitch radius of the bolt
• \( p \) is the lead of the thread (e.g., inches/rev)
• \( R \) is the bolt head radius
• \( \alpha \) is the thread angle
• \( \beta \) is \( 2R/p \)

• To raise a load (tighten a joint), the torque required is:

\[
\Gamma_{\text{required}} = \Gamma_{\text{desired}} \left( \frac{2\pi \mu r + p \cos \alpha}{2\pi r \cos \alpha - \mu p} \right) r + R \mu
\]

\( ^1 \) For a REALLY thorough examination of this subject, including off-axis moments created by the applied torque, see A. Slocum Precision Machine Design, published by SME, Dearborn, MI.
• To lower a load $F$, the torque required is:

$$\Gamma_{\text{required}} = F_{\text{desired}} \left( \left( \frac{2\pi r \mu - p \cos \alpha}{2\pi r \cos \alpha + \mu p} \right) r + R \mu \right)$$

• The screw will not back-drive when:

$$p < \frac{2\pi r \mu}{\cos \alpha}$$

• The efficiency $\eta$ is:

$$\eta = \frac{\cos \alpha (\pi \beta \cos \alpha - \mu)}{\pi \beta \cos \alpha (\cos \alpha + \pi \beta \mu)}$$
The torque applied to the bolt creates stresses

- The force that is generated creates tensile and torsional shear stresses.

- The thread root is a stress concentration area (a factor of 2-3), which is somewhat mitigated when the threads are rolled (as opposed to cut).

- Assuming a thread root diameter $r_{tr}$, the stresses are:

\[
\sigma_{\text{tensile}} = \frac{F_{\text{Tensile}}}{\pi r_{tr}^2}
\]

\[
\tau_{\text{shear}} = \frac{2\Gamma r_{tr}}{\pi r_{tr}^4}
\]

The equivalent (Von Mises) stress is:

\[
\sigma_{\text{tensile equivalent}} = \sqrt{\sigma_{\text{tensile}}^2 + 3\tau_{\text{shear}}^2}
\]
Some heuristic design rules for bolted joints:

• Never use a bolt diameter smaller than 6 mm on a structure: Small bolts are too easily twisted off!
  
  • Washers are rarely used with socket-head cap screws because stiffness would be reduced by the addition of an interface.
  
  • The purpose of a washer is to distribute the load on thin sections, or to keep a hex head bolt from chewing the surface as it is tightened!

• Ideally, the bedded portion of the rail should be at least as wide as the cantilevered portion.

• The flange should be at least one to two bolt diameters thick.
  
  • Make the stress zone cones beneath the bolt heads overlap.

• Make sure that the surfaces have been stoned with precision diamond ground stones just prior to assembly to remove raised lips around holes.
Bolt installation

• Make sure the threads are cleaned and lubricated.
• Incrementally tighten the bolts.
• Consider using vibration to stress relieve the joint.
• To lock a bolt in place, use one or more of the following:
  • Use a thread lubricant-locking agent.
  • A dab of epoxy on the bolt head.
  • A lock washer.
  • A "Nylock™" bolt.
  • Yield tighten the bolt (tighten to the yield point).
Joint interface stiffness:

- Attempts to analytically model the contact between hundreds of have never been successful.
  - There are just too many unknowns.
- The stiffness of a bare joint interface can only be accurately determined by experimental means.
- However, typically, the interface stiffness will be 5 times the bolt system stiffness.
- If more determinism is sought in joint interface stiffness, then an adhesive can be used.
  - Parting can be obtained by having one surface plated and Teflon impregnated.
    - Most mold releases use a wax, and the wax layer may be too compliant.
  - Using an adhesive can lesson damping from a bolted joint!
• **Empirical data must often be used to obtain joint stiffness values:** joint stiffness of 0.55% C ground steel parts. (After Yoshimura.)

![Graph showing equivalent values of spring stiffness per unit area](image)

• **Compressive stiffness at lower contact pressures** (after Dolby and Bell).

![Graph showing unit stiffness vs. pressure](image)
Heuristic rules for increasing bolted joint stiffness

• Bolts should not be spaced too far apart.
  • The joint pressure should be estimated as the total force provided by all bolts divided by the total joint area.

• The joint stiffness is the product of the stiffness per unit area value obtained from the empirical data and the bolted area.
  • Assume that the bolt force acts over an area of four to five bolt diameters (the area of the Cone of Influence).

• The finer the surface finish and the tighter the preload:
  • The fewer and smaller the gaps between surface asperities on the surfaces, and the higher the stiffness.

• Where compressive stiffness is critical, surfaces should be ground, scraped, or lapped:
  • Surfaces should always be stoned with a precision ground flat stone just prior to assembly. This is critical when mounting linear guides!
  • Bolt holes should also be countersunk to minimize the forming of “crater lips” during hole drilling.

• Vibration annealing can help to seat the asperities.

• Bolting through machined bosses, and then grouting between the bolted regions with an adhesive can save machining costs.
  • This greatly increases joint stiffness, but can decrease the damping.
Design procedure to determine the required preload and bolt size for a joint:

- The stiffness of the joint should be greater than the stiffness of the parts that make up the joint.

- Determine the required joint interface pressure to achieve the required stiffness.
  - Assume that the bolt force acts over an area of four to five bolt diameters (the area of the Cone of Influence).

- The minimum total preload force should be the larger of:
  - The sum of the maximum tensile force on the joint and the required joint preload force or:
  - Four times the maximum tensile force.

- The product of the minimum force and the coefficient of friction for the joint should be:
  - 5-10 times greater than the maximum expected shear force on the joint.

- Space the bolts equal to the part thickness or a portion of the width of the bearing rail.

- On one extreme, size the bolts so the stresses in the bolts are below 25% of yield.
  - Compare this value to the bolt size found from a minimum stiffness criteria (as shown in the example).

- On another extreme, yield tighten the bolts.

- Check the bending, compression, and Poisson expansion of the rail to make sure they are within acceptable limits.
  - Do a wavelength FFT of the straightness to check for errors at a wavelength of the bolt spacing.
Stability of bolted joints

- Common bolted joints may suffer a loss of initial preload with applied cyclic stress.
  - Loss of preload seems to be due to the applied stress causing microslip between the surfaces.

- If the joint cannot be pinned with dowels, one may wish to specify vibration stress relief and re-tightening of bolts.
  - The final tightening can be done while the vibration stress relief process is being done.

- If the joint is to be subject to vibration loads or extreme stability is required:
  - A hardening thread lubricant should be used.
  - A film adhesive can be spread on the joint.
  - After tightening, a large glob of epoxy should be placed on the bolt head and seating surface.
• Ideally, to ensure stability in a machine that may have a high degree of vibration, one could:

  • The bearing rail should be aligned and the bolts incrementally tightened according to a specific pattern:
    • For example, from the inside out to prevent clamping in a bow.
  • The alignment should be checked after each tightening increment.
  • The assembly should be vibration stress relieved.
  • The alignment should be checked and the bolts re-tightened.
  • The alignment should be checked, and then the assembly vibration stress relieved while the bolts are re-tightened.
  • A final checking of the alignment should be performed.
  • Ideally, the rail would be pinned to the structure with dowels.
  • It can also be potted in place with an epoxy which helps increase joint stability and damping.
  • A loose fitting key that runs the length of the rail can be grouted or epoxied in place.
Bearing rail design

• Deterministic design is possible if the following are considered:
  • Bolted joint stiffness.
  • Bearing rail stiffness:
Bearing rail stiffness

- The rail is not as stiff as if it were ideally cantilevered, but it is stiffer than a simply supported beam.
  - A simply supported model should be conservative.
- The bolts do not provide knife edge support, thus the stiffness will be less than modeled.
- When a bearing pad exerts a force on the bearing rail, neighboring sections provide support also.
- The surface to which the rail is bolted only supports the rail at the rear point.
  - It actually provides some support due to the bowing of the rail.
  - Assume the rear support point is infinitely stiff and the surface beneath the rail doesn't resist bowing of the rail.
- Bending and shear deformations must be considered.
- For sliding contact bearings, force is applied at the center of the pad.
- For hydrostatic bearings, the deflection must be computed based on the pressure distribution.
Sliding contact bearing rail model:

- The system behaves like a beam attached to a pivot point so the equivalent stiffness at the point of force application is:

\[ K_{eq \text{ bolt system}} = \frac{K_a^2}{(a + b)^2} \]

\[ K_{eq, \text{ bolt system}} = CD_B \]

- The bending deflection is:

\[ \delta_{bend} = \frac{F(a + b)b^2}{3EI} \]

- The shear deflection is:

\[ \delta_{shear} = \frac{Fbh^2(a + b)(1 + \eta)}{5aEI} \]

- Assume that the bolt spacing \( \ell \) is a function of the diameter:

\[ \ell = D_B(M - 1) \]

  - This represents \( M \) bolt diameters minus the region occupied by the bolt.

- The moment of inertia for the rail section is then \( I = \ell h^3/12 \).
• The total stiffness for $N$ segments of the rail (e.g., $N$ segments under a bearing pad) is thus:

$$K_{\text{Rail}} = \frac{1}{K_{\text{Rail bend & shear}}} + \frac{N_{\text{Segments}}}{K_{\text{Joint}}} + \frac{1}{K_{\text{Flange comp}}} + \frac{1}{K_{\text{Flange shear}}} + \frac{1}{K_{\text{Bed Shear}}} + \frac{1}{K_{\text{Bolt}}}$$

• Bolt diameter is not a sensitive parameter for stiffness when bolt spacing is made a function of the bolt diameter.

• The length of bolt and the cone are expressed as bolt lengths:

  • Bolt joint stiffness becomes linearly dependent on the bolt diameter.

• As a result, bolt diameter cancels out.
The interesting parameters are:

- The ratio of joint interface stiffness to joint stiffness.
- The total length of rail over which one wants to know the stiffness.
- Ideally, for a balanced design, the interface stiffness should be equal to the bolt and rail stiffness.
- If the interface stiffness is too high:
  - The displacement caused by the preload will be very small.
  - Manufacturing defects may cause loss of preload around a bolt.
- Realistically, the interface stiffness may be set to as high as five times the bolt system stiffness.
• For example, these two bolted systems have the same stiffness:

![Diagram 1](image1)

![Diagram 2](image2)

• The design goal is to achieve high stiffness and minimize manufacturing cost, while maintaining accuracy.

• For maximum accuracy, it is desirable for the cones of influence under the bolt heads to overlap.
  
  • This minimizes bending of the rail on an elastic foundation as described below.

• Thus even though a thicker flange has slightly lower stiffness, it allows for the use of less bolts.

• For example, with a flange thickness of $2D_B$, the bolt spacing should not be more than $4D_B$
• Lateral deformations due to the Poisson effect should also be prevented, particularly if the rail is also a guide rail:

Bulge (exaggerated) caused by overtightened bolt

• These effects are minimized by:
  • Maximizing the ratio of rail width to bolt diameter.
  • Minimizing the bolt spacing.
  • Minimizing the bolt torque.

• Poisson expansion is a leading cause of rotary bearing failure, when bolted flanges are used to preload bearing assemblies.

• Use more small bolts to minimize deformation.

• Remember St. Venant’s principle: 3 characteristic dimensions (bolt head diameters) away, effects are no longer felt!
• In some cases, where joint loosening would be disastrous (e.g., in spindle components), yield tightening may be employed.

• Yield tightened joints can use a torque limit or a displacement limit.

• Torque limits (with a wrench) can vary due to friction changes.

• Displacement criteria can be more deterministic (tighten until a fixed torque, and then apply 1/Nth of a turn).

• Be careful to measure the straightness of the system:

• Do an FFT (straightness error vs. wavelength) and see if there is a peak at the bolt stiffness.

• If there is a peak, then the bolts are too tight.
Example: Spreadsheet for bolt joint torque

Bolted rail design
Enter numbers in bold

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Rail deformation due to bolt preload

- Even when a "perfect" rail is bolted to a "perfect" bed:
  - Significant (micron to submicron depending on bolt and rail size and preload) deformations can result.
- Bolt preload causes various types of deformation:
  - Bending.
  - Compression.
  - Poisson expansion.
- These errors can be corrected by lapping after bolting.
- The use of a large number of bolts can be used to deform a machine into a desired shape (e.g., to straighten an axis).
- If the rails were finished while bolted and then transferred to another machine:
  - Desirable to be able to determine the deformation that could result from a variation in the bolt force.
• 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.

• Analysis methods exist (see text) for estimating the amount of deformation.
• For the previous bolted rail spreadsheet example, the rail deformation is about 0.15 µm (6 µin.).

• Recall the Poisson effect:
  • Compressive deformation is accompanied by a lateral expansion:
    • On the order of 0.3 times the compressive deformation (for most metals), depending on the rail geometry.

• For large rails, 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.
  • Manufacturers' bolt-tightening recommendations should be carefully followed.

• Beware of mounting to surfaces made from materials with a low modulus of elasticity.

• Remember, a Fourier transform of the straightness as a function of wavelength can help spot bolting problems!

Bulge (exaggerated) caused by overtightened bolt
• Bolt torques and thread efficiencies are never exactly the same.

• Varying vertical and horizontal straightness errors in the rail will often be present.
  • The period will often be equal to that of the bolt spacing.
  • Always do a Fourier transform of the straightness error.
  • Plot the results as error amplitude verses wavelength.

• When designing and manufacturing a precision machine:
  • Bolt preload, thread friction, and tightening methods should all be chosen with care.
Modular "T" slide bearing configuration:

- Bearing pad
- Bearing rail
- Keeper rail
- Inboard bolt (not needed if top bearing pad extends the width of the rail)
- Ballscrew
- Magnetically encoded scale
- Carriage (Saddle)
- Tapered gibs
- Bed
Bolt torque summary:

- The theory will often yield a recommended torque level which is far below the allowable torque for a particular bolt:
  - The theory represents the minimum required torque.
  - The higher the torque, the better as far as machine stability is concerned.
  - Too high torques can deform components (e.g., cause rail straightness errors).
- It is best to experiment:
  - See what the component deformations are at different torque levels.
  - Use as high a torque level as possible.
  - Do not forget to stone surfaces before they are bolted together, in order to ensure that there are no burrs.
Bonded joints:

- Bonded joints can be strong and fatigue resistant.

- Adhesives fill small voids between mating parts and thus increase the following characteristics of the joint (Courtesy of Loctite Corp.):
  
  - Strength
  - Stiffness
  - Damping (>welded, <bolted)
  - Heat transfer

- Even when two ground surfaces are to be bolted together:
  
  - It can be wise to specify the use of a few drops of low viscosity adhesive that will flow and fill the small voids.
  - Sometimes used on very high precision instruments and tools
  - Not used on machine tools
  - A sliding fit joint held by an adhesive can make a joint more robust than a press fit joint:
    
    - More dimensionally stable.
    - More accurate.
    - Longer lasting.
  - Bearing races can also be fixed to bores in this manner to avoid altering the preload by press fitting.
• A potted (grouted) joint uses adhesive (low shrinkage epoxy\textsuperscript{2}) to lock in place a joint.

• Mechanical means hold parts in alignment and support the static weight of the structure.

• In some cases:
  • The mechanical means are often there just for initial alignment.
    • Kinematic couplings can be used.
  • The grouting carries large applied loads.
  • Allows for kinematic assembly for easy adjustment and then potting for added rigidity.

• In other cases:
  • The mechanical joint supports the static load.
  • The potting compound resists microslip of the mechanical joint and greatly increases joint stiffness and damping.

\textsuperscript{2} Low shrinkage potting epoxies are available from 1) Andrew Devitt, Devitt Machinery Co., Twin Oaks Center, Suite G, 4009 Market Street, Aston, PA 19014, (610) 494-2900; 2) Loren Power, ITW Philadelphia Resins, 10132 Spiritknoll Ln., Cincinnati, OH 45252-1937, (513) 385-4577
• Examples include:
  • Grouting machine tool beds to concrete floors.
  • Potting major machine tool components to the machine tool structure.
  • Potting sensors in sensor mounts.
• Replication: a polymer is poured or injected around a master shape that has been coated with mold release.
• When the polymer cures, it has the shape of the master which can then be removed and used again.
• Polymer shrinkage is very low (typically 0.25%) shrinkage, BUT it can be significant: Example:
  • 1 meter tall column 0.5 m wide at the base.
  • Wedge of polymer 2 mm thick at one side and 3 mm thick at the other side, and 0.25% shrinkage.
  • Differential shrinkage across the wedge will be 0.001m x 0.0025 = 2.5 microns.
  • Abbe error at the top of the column will be 5 microns!
• Hardening of many polymer resins is an exothermic reaction.
  • Minimize the amount of polymer used.
  • Maximize the stiffness and thermal diffusivity of the master and the part.
• This is required to prevent:
  • The heat of polymerization from heating the structure and deforming it.
  • Then the polymer hardening to the thermally deformed shape.
• Unlike bolted assemblies, once the polymer cures to form the desired shape realignment is no longer possible.
• Replicated parts and assemblies can be as dimensionally stable as the rest of the machine.
• In machine tools, mounting surfaces for linear bearing rails can be replicated.
• In some cases for sliding or hydrostatic bearings the replicated surface functions as the way surface.