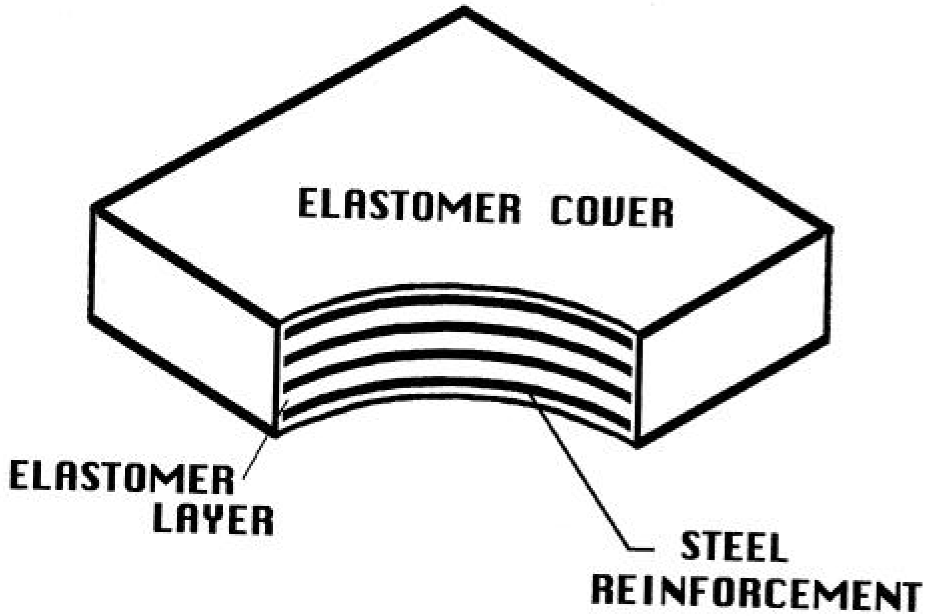


Figure II-2.2: Typical Steel Reinforced Elastomeric Bearing

The design of a steel reinforced elastomeric bearing requires an appropriate balance of compressive, shear and rotational stiffnesses. The shape factor affects the compressive and rotational stiffness, but it has no impact on the translational stiffness or deformation capacity.

A bearing must be designed so as to control the stress in the steel reinforcement and the strain in the elastomer. This is done by controlling the elastomer layer thickness and the shape factor of the bearing. Fatigue, stability, delamination, yield and rupture of the steel reinforcement, stiffness of the elastomer, and geometric constraints must all be satisfied.

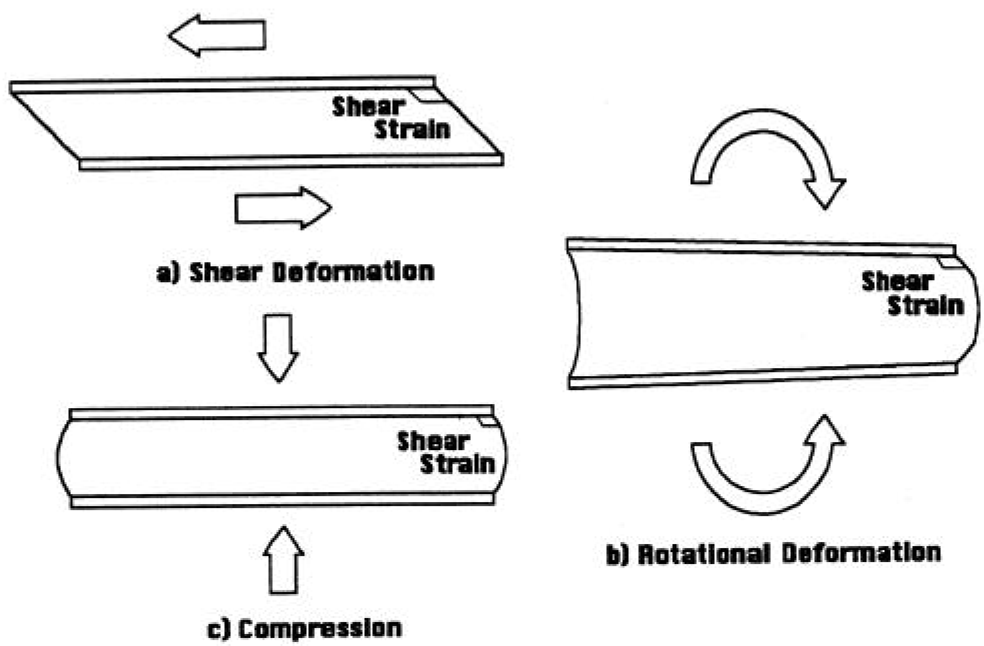


Figure II-2.3: Strains in a Steel Reinforced Elastomeric Bearing`

Large rotations and translations require taller bearings. Translations and rotations may occur about either horizontal axis of a steel reinforced elastomeric bearing, and this makes them suitable for bridges where the direction of movement is not precisely defined. Circular steel reinforced elastomeric bearings are particularly well suited for this purpose.

Steel reinforced elastomeric bearings become large if they are designed for loads greater than about 4500 kN (1000 kips). Uniform heating and curing during vulcanization of such a large mass of elastomer becomes difficult, because elastomers are poor heat conductors. Manufacturing constraints thus impose a practical upper limit on the size of most steel reinforced elastomeric bearings.

Design Requirements

The design of steel reinforced elastomeric bearings requires a balance between the stiffness required to support large compressive load and the flexibility needed to accommodate translation and rotation. The AASHTO LRFD Specifications provide these requirements. The balance is maintained by using a relatively flexible elastomer with a shear modulus, G, between 0.55 MPa and 1.25 MPa (80 and 180 psi) and an appropriate shape factor.

The height of the bearing is controlled by the movement requirements. The shear strains due to translation must be less than 0.5 mm/mm to prevent rollover and excess fatigue damage[8,11]. Therefore, Eq. 2-5b also applies to steel reinforced elastomeric bearings, and the total elastomer thickness, h_{rt}, must be greater than two times the design translation, D_S. Separation between the edge of the bearing and the structure must be avoided during rotation, since separation causes tensile stresses in the elastomer and the potential for delamination. Separation is prevented by the combined compression and rotation limits that require

$$\sigma_T \geq 1.0\ G\ S\left(\frac{\theta_{max}}{n}\right)\left(\frac{B}{h_{ri}}\right)^2 \qquad \text{(Eq. 2-7)}$$

where B is the horizontal plan dimension normal to the axis of rotation, q_{max} is the maximum service rotation about any axis, n is the number of elastomer layers, and h_{ri} is the thickness of an individual elastomer layer. Increased rotation capacity at a given load level may be achieved by an increase in h_{ri} or a reduction in S.

Delamination of the elastomer from the steel reinforcement is also an important consideration. This is controlled by limiting the maximum compressive stress due to combined loads on the elastomer to 11.0 MPa (16 ksi) for bearings subject to shear deformation and 12.0 MPa (1.75 ksi) for bearings fixed against shear deformation.

Steel reinforced elastomeric bearings are also subject to fatigue. The fatigue cracks occur at the interface between an elastomer layer and the steel reinforcement, and are caused by the local shear stresses which may arise from compression, rotation or shear loading. Fatigue damage during the lifetime of the bridge is controlled by limiting the average compressive stress on the bearing to a value that depends on the other loadings that are applied simultaneously. The fatigue design limits are

For bearings subjected to compression alone

$$\sigma_T \leq 2.00\ G\ S \leq 12.0\ \text{MPa}\ (1.75\ \text{ksi}) \qquad \text{(Eq. 2-8a)}$$

and

$$\sigma_L \leq 1.00\ G \qquad \text{(Eq. 2-8b)}$$

For bearings subjected to combined compression and shear deformation

$$\sigma_T \leq 1.66\ G\ S \leq 11.0\ \text{MPa}\ (1.60\ \text{ksi}) \qquad \text{(Eq. 2-9a)}$$

and

$$\sigma_L \leq 0.66\ G\ S \qquad \text{(Eq. 2-9b)}$$

where

σ_T = average compressive stress due to total service load = $\dfrac{P_T}{A}$

σ_L = average compressive stress due to live load = $\dfrac{P_L}{A}$

Steel reinforced elastomeric bearings must also satisfy uplift requirements. For rectangular bearings subjected to combined compression and rotation

$$\sigma_T \leq 2.25\ G\ S\left(1 - 0.167\left(\frac{\theta_{max}}{n}\right)\left(\frac{B}{h_{ri}}\right)^2\right) \qquad \text{(Eq. 2-10a)}$$

For rectangular bearings with combined translation, compression and rotation

$$\sigma_T \leq 1.875\ G\ S\left(1 - 0.20\left(\frac{\theta_{max}}{n}\right)\left(\frac{B}{h_{ri}}\right)^2\right) \qquad \text{(Eq. 2-10b)}$$

Elastomeric bearings may also buckle under compressive load and must satisfy stability limitations. Bearings which are susceptible to sidesway must satisfy

$$\sigma_T \leq \left(\frac{G}{\left(\dfrac{3.84(h_{rt}/L)}{S\sqrt{1+2.0\ L/W}} - \dfrac{2.67}{S(S+2.0)(1+L/4.0W)}\right)}\right) \qquad \text{(Eq. 2-11a)}$$

Bearings that are restrained against sidesway must satisfy

$$\sigma_T \le \left(\frac{G}{\left(\frac{1.92\left(h_{rt}/L\right)}{S\sqrt{1+2.0\ L/W}} - \frac{2.67}{S(S+2.0)(1+L/4.0W)} \right)} \right) \quad \text{(Eq. 2-11b)}$$

The buckling capacity depends upon the shear modulus, the total elastomer thickness h_{rt}, the base dimensions L and W, and the shape factor S. For the buckling equations, L is in the direction of buckling, and W is normal to it.

Tensile stress develops in the steel reinforcement since it restrains the bulging of the elastomer. This tensile stress may control the thickness of the reinforcement. Therefore, the thickness of the steel reinforcement, h_s, must meet the following requirements. For total compressive stress,

$$h_s \ge \frac{3\, h_{rmax} \sigma_T}{F_y} \quad \text{(Eq. 2-12a)}$$

and, for live load only

$$h_s \ge \frac{2.0\ h_{rmax} \sigma_L}{(\Delta F)_{TH}} \quad \text{(Eq. 2-12b)}$$

where $(\Delta F)_{TH}$ is the constant amplitude fatigue threshold given in Table 6.6.1.2.5-3 of the AASHTO LRFD Specifications.

In general, elastomer layer thickness should be selected to satisfy all design requirements, but practical limitations of the bearing manufacturer should also been considered. The thickness should normally be a convenient dimension that the manufacturer will easily understand and can easily maintain during fabrication. Larger thicknesses are appropriate for larger plan dimensions, since manufacturers have increasing difficulty maintaining very thin layer thickness with large bearings.

If the bearing is to be used in a very cold climate the low temperature stiffness must be considered. Certification tests by the manufacturer are required if the elastomer is susceptible to these low temperature conditions which affect a small part of the United States. The AASHTO LRFD Specifications(10) contains a very conservative temperature zone map which shows regions requiring low temperature consideration. Bridge designers should use the written description(5,6) of the temperature zones to design for a more realistic temperature region.

Design Example

A design example is presented to illustrate the above design requirements. A steel reinforced elastomeric bearing is to be designed for the following service loads and translations.

Dead Load	2400 kN (540 kips)
Live Load	1200 kN (270 kips)
Longitudinal Translation	100 mm (4.0 in.)

Rotation 0.015 radians

The above bearing translation is in the longitudinal direction of the bridge with the bridge fixed against movement at the 5th support. The rotation is about the transverse axis. There are no design translations in the transverse direction, but restraint in this direction is provided only by the stiffness of the bearing. The steel girder has a bottom flange bearing width of 750 mm (30 in.). A schematic of the bridge is illustrated in Figure II-2.4.

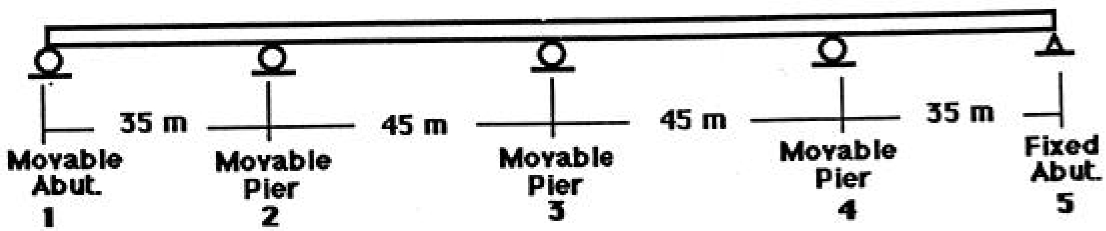


Figure II-2.4: Schematic of Example Bridge Restraint Conditions

These loads, translations and rotations are relatively large compared to those commonly considered acceptable for steel reinforced elastomeric bearings. However, examination of Figure I-2 of the *Steel Bridge Bearing Selection Guide* contained in Part I of this report suggests that a steel reinforced elastomeric bearing may be the most economical alternative. It will be shown that the bearing can indeed be designed for these requirements.

A typical elastomer with hardness in the range of 55 Shore A Durometer and a shear modulus in the range of 0.7 to 0.91 MPa (100 to 130 psi) is proposed. The total compressive load is 3600 kN (810 kips), and the 11.0 MPa (1.60 ksi) delamination stress limit of Eq. 2-9a requires a total plan area of at least

$$A > \frac{3600(1000)}{11} = 327\ 300\ \text{mm}^2$$

The bearing should be slightly narrower than the flange unless a stiff sole plate is used to insure uniform distribution of compressive stress and strain over the bearing area. The bearing should be as wide as practical to permit rotation about the transverse axis and to stabilize the girder during erection. Therefore a bearing width of 725 mm (29 in.) is an appropriate first estimate, and a 475 mm (19 in.) longitudinal dimension will assure that the delamination requirement is met. The longitudinal translation is 100 mm (4 in.), and so a total elastomer thickness of at least 200 mm (8 in.) is required to satisfy the rollover and excessive fatigue damage design requirements. A layer thickness of 15 mm (0.6 in.) is chosen in order to maintain an adequate shape factor. This leads to 14 layers with a total elastomer thickness of 210 mm (8.3 in.) and a preliminary shape factor of

$$S = \frac{725 \text{ x } 475}{2 \text{ x } 15 \text{ x } (725 + 475)} = 9.57$$

Prevention of uplift (Eq. 2-7) may also control the overall bearing dimensions. The base dimension, *B*, normal to the axis of rotation is 475 mm (19 in.), and the maximum compressive stress must satisfy

$$\sigma_T = \frac{P_T}{A} \geq 1.0\ GS\left(\frac{\theta_{max}}{n}\right)\left(\frac{B}{h_{ri}}\right)^2$$

$$\frac{3600(1000)}{475 \text{ x } 725} \geq 1.0 \text{ x } 0.91 \text{ x } 9.57\left(\frac{0.015}{14}\right)\left(\frac{475}{15}\right)^2$$

$$10.45 \text{ MPa} \geq 9.36 \text{ MPa} \qquad \text{ok}$$

G is taken as 0.91 MPa because the AASHTO LRFD Specifications require that, if the elastomer is defined by hardness rather than shear modulus, each calculation should use the least favorable value of *G* from the range that corresponds to the selected hardness.

Fatigue limits must also be checked. Since this bearing is subject to combined compression, shear deformation and rotation, Eqs. 2-9a, 2-9b and 2-10b will control.

$$\sigma_T = 10.45 \text{ MPa} < 1.66\ G\ S \leq 11.0 \text{ MPa}$$

$$< 1.66 \text{ x } 0.7 \text{ x } 9.57 \leq 11.0 \text{ MPa}$$

$$< 11.1 \text{ MPa} \leq 11.0 \text{ MPa}$$

$$10.45 \text{ MPa} < 11.0 \text{ MPa} \qquad \text{OK}$$

and

$$\sigma_L = \frac{1200 \text{ x } 1000}{475 \text{ x } 725} = 3.48 \text{ MPa} < 0.66\ G\ S$$

$$< 0.66 \text{ x } 0.7 \text{ x } 9.57 = 4.42 \text{ MPa} \qquad \text{ok}$$

Both are satisfied indicating that the bearing is acceptable for fatigue with combined shear and compression. The limit for combined shear, rotation and compression determined with Eq. 2-10b must also be checked, and

$$\sigma_T \leq 1.875\ G\ S\left(1 - 0.20\left(\frac{\theta_{max}}{n}\right)\left(\frac{B}{h_{ri}}\right)^2\right)$$

$$10.45 \text{ MPa} \leq 1.875 \text{ x } 0.7 \text{ x } 9.57\left(1 - 0.20\left(\frac{0.015}{14}\right)\left(\frac{475}{15}\right)^2\right) = 9.86 \text{ MPa} \qquad \text{NG}$$

This condition is not satisfied, because of the large rotation and the compressive load. However, this equation will be satisfied if the number of layers is increased to 20, and the total internal elastomer thickness is increased to 300 mm (12 in.).

Stability limits must also be checked. The bearing is free to sidesway in the transverse direction but is fixed against translation in the longitudinal direction. Thus, longitudinally Eq. 2-11b must be satisfied,

$$\sigma_T \le \left(\frac{G}{\left(\frac{1.92(h_{rt}/L)}{S\sqrt{1+2.0L/W}} - \frac{2.67}{S(S+2.0)(1+L/4.0W)} \right)} \right)$$

$$10.45 \text{ MPa} \le \left(\frac{0.7}{\frac{1.92(300/475)}{\left(9.57\sqrt{1+2.0(475)/725}\right)} - \frac{2.67}{9.57(11.57)(1+475/4.0(725))}} \right)$$

$$10.45 \text{ MPa} \le 11.17 \text{ MPa} \qquad \text{ok}$$

and transversely Eq. 2-11a must be satisfied,

$$\sigma_T \le \left(\frac{G}{\left(\frac{3.84(h_{rt}/L)}{S\sqrt{1+2.0L/W}} - \frac{2.67}{S(S+2.0)(1+L/4.0W)} \right)} \right)$$

$$10.45 \text{ MPa} \le \left(\frac{0.7}{\frac{3.84(300/725)}{\left(9.57\sqrt{1+2.0(725)/475}\right)} - \frac{2.67}{9.57(11.57)(1+725/4.0(475))}} \right)$$

$$10.45 \text{ MPa} \le 10.77 \text{ MPa} \qquad \text{ok}$$

Equations 2-12a and 2-12b must also be checked for reinforcement thickness. Assuming a steel with a 250 MPa (36 ksi) yield stress, the limit for total compressive stress is

$$h_s \ge \frac{3\, h_{rmax} \sigma_T}{F_y} = \frac{3 \text{ x } 15 \text{ x } 10.45}{250} = 1.88 \text{ mm}$$

The fatigue limit is less critical since the reinforcement has no holes or discontinuities, and can be treated as a plain member with a fatigue limit of 165 MPa (24 ksi).

$$h_s \ge \frac{2.0\, h_{rmax} \sigma_L}{(\Delta F)_{TH}} = \frac{2 \text{ x } 15 \text{ x } 3.48}{165} = 0.63 \text{ mm}$$

The required steel reinforcement thickness is approximately 2 mm (0.08 in.). It may also be desirable to use a thicker (say 3 mm) plate, since this may simplify manufacture and tolerance control, although it would also slightly increase the weight. Discussion with bearing manufacturers used by the bridge owner would help to establish the desirability of this final adjustment. Under these conditions, the finished bearing would be designed as shown in Figure II-2.5.

These design equations appear relatively cumbersome because several features must be checked and the behavior of steel reinforced elastomeric bearings is governed by relatively unusual principles of mechanics. The different requirements also interact, so design may involve some trial and error. However, they can easily be programmed into a spreadsheet, in which case the design becomes very simple. An example spreadsheet is given in Appendix B.

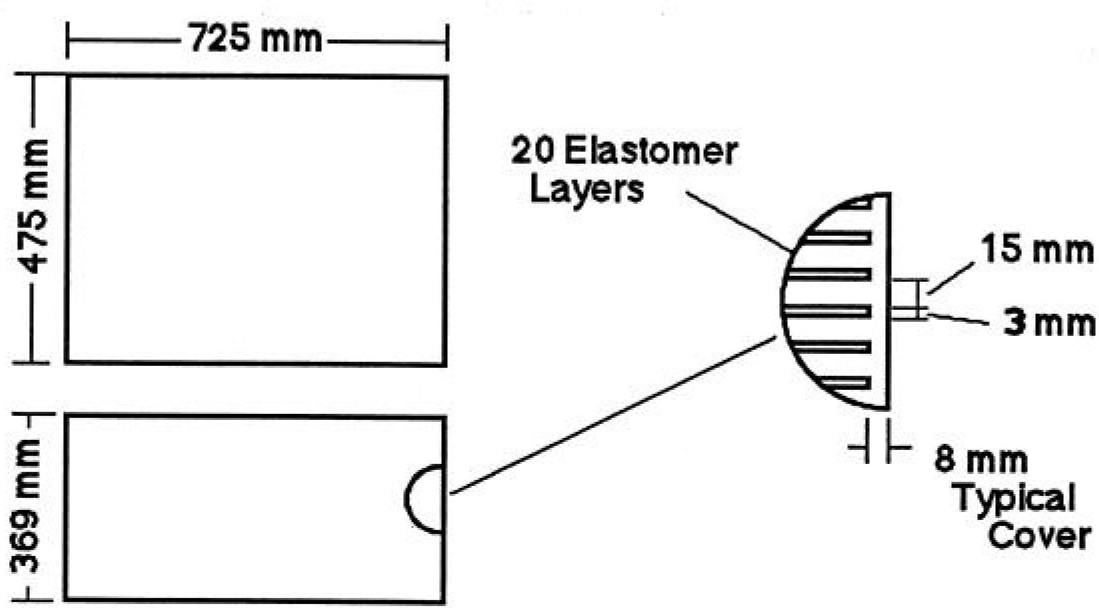


Figure II-2.5: Final Design of a Steel Reinforced Elastomeric Bearing

Summary

Many engineers incorrectly assume that steel reinforced elastomeric bearings are unsuitable for steel bridges because of the relatively large translations and rotations of the bridge. If proper design, materials, manufacturing and construction requirements are used, steel reinforced elastomeric bearings are very versatile. They may support loads as large as 4500 kN (1000 kips) and accommodate translations up to 150 mm (6 in.). Rotations of 2 or 3 degrees are achievable. Steel reinforced elastomeric bearings have an advantage over pot and spherical (HLMR) bearings where the rotations are large and their orientation is uncertain. Over-rotation of HLMR bearings causes metal to metal contact and possible permanent damage. An elastomeric bearing, by contrast, can accept a small number of short-term over-rotations with a low probability of damage.

The economy of the elastomeric bearing depends on both the load and displacement. In the 450 to 2200 kN (100 to 500 kips) range with moderate displacement and rotation requirements, a steel reinforced elastomeric bearing is likely to be less expensive than other alternatives. At higher loads or displacements, elastomeric bearings may still be the most economical alternative. However, the most economical alternative may be a combination of steel reinforced elastomeric bearings with other components such as a PTFE sliding surface to accommodate translations larger than 100 mm (4 in.).

POT BEARINGS

Elements and Behavior

The basic elements of a pot bearing are a shallow cylinder, or pot, an elastomeric pad, a set of sealing rings and a piston as shown in Figure II-2.6. Masonry plates and base plates are common, because they allow attachment of the bearing and increase the support area on the pier or abutment. Pot bearings are fixed against all translation unless they are used with a PTFE sliding surface.

The pot and piston are almost always made from structural carbon steel, although stainless steel and aluminum have occasionally been used if corrosion control is a concern. A variety of types of sealing ring have been used. Most sealing rings are either a single brass ring of circular cross-section, or a set of two or three flat brass rings. The circular rings have traditionally been brazed into a closed circle, whereas the flat ones are usually bent from a strip and the ends are not joined. Brass rings are placed in a recess on the top of the elastomeric pad. PTFE rings have been tried, but have been abandoned because of their poor performance. Other proprietary sealing ring systems have been used.

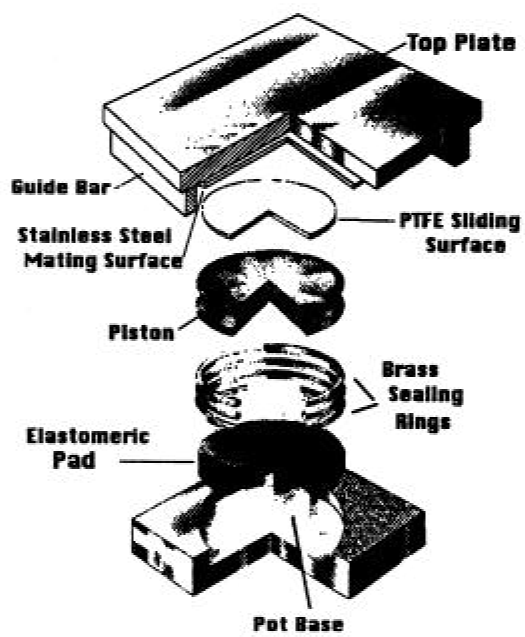


Figure II-2.6: Components of a Typical Pot Bearing

Compression

Vertical load is carried through the piston of the bearing and is resisted by compressive stress in the elastomeric pad. The pad is deformable but almost incompressible and is often idealized as behaving hydrostatically. In practice the elastomer has some shear stiffness and so this idealization is not completely satisfied. Experiments(12) have shown that pot bearings typically have a large reserve of strength against vertical load.

Deformation of the pot wall is a concern, since this deformation changes the clearances between the pot and the piston and may lead to binding of the bearing or to elastomer leakage. Two effects influence the displacements of the pot wall. First, compression in the elastomeric pad causes outward pressure on the pot wall, and this induces tension in the baseplate and outward bending of the pot wall. Second, the compressive stress on the bottom of the pot causes elastic deformation(13,14) of the concrete under the bearing. This deformation leads to downward dishing of the baseplate under the compressive load, and the baseplate deformation causes the pot wall to rotate inward. The bending stresses associated with this rotation of the pot wall are largest at the inside corner of the pot, and must be considered in the bearing design. Failures of pot bearings that were constructed by welding a ring to a flat baseplate have occurred because the weld, located at the critical location, was not designed to account for this load.

Rotation

Pot bearings are often regarded as suitable for use when bridge bearing rotations are large. Rotation may occur about any axis and is accommodated by deformation of the elastomeric pad. Large cyclic rotations can be very damaging to pot bearings in a relatively small number of cycles due to abrasion and wear of the sealing rings and elastomeric pad. However, pot bearings can sustain many cycles of very small rotations with little or no damage.

During rotation, the elastomeric pad compresses on one side and expands on the other, so the elastomer is in contact with the pot wall and slips against it. This causes elastomer abrasion and sometimes contributes to elastomer leakage. Lubrication is often used to minimize this abrasion, but experiments(14,15) show that the lubricant becomes less effective over time. Silicone grease, graphite powder and PTFE sheets have all been used as lubricants and, of these, the silicone grease has proven to be the most effective.

Inadequate clearances represent a second potential problem during rotation of pot bearings. These may cause binding of the bearing, and may induce large moments into the support or superstructure. However, these problems can be controlled by proper design. Figure II-2.7 illustrates typical clearances required in the design of the bearing.

Cyclic rotation may also be damaging to the sealing rings of pot bearings. Flat brass rings are more susceptible to ring fracture and elastomer leakage, while circular brass rings are susceptible to severe wear. Contamination of the pot by dirt or debris increases the potential for wear and damage to both the elastomeric pad and the sealing rings. A rough surface finish on the inside of the pot and piston produced by metalization or a rough machined surface produces results similar to those caused by contamination. A smooth finish results in less wear and abrasion. Bearings with a smooth finish, no internal metalization, and a dust seal appear to offer substantial benefits.

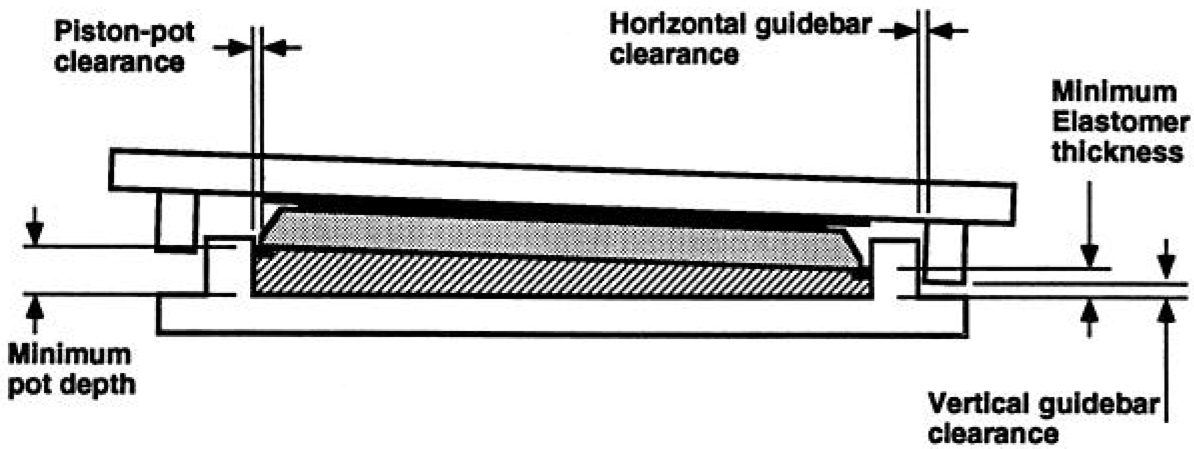


Figure II-2.7: Tolerances and Clearances for a Typical Pot Bearing.

Pot bearings have traditionally been designed so that the maximum compressive strain in the elastomer due to rotation is 15 percent. For 0.02 radians of rotation, the ratio *D/t* of the elastomeric pad must then be 15 at most. Tests have been performed on pot bearings with *D/t* ratios as large as 22 and as small as 12. Increasing the pad thickness accommodates higher rotations but increases the required depth, and therefore the cost of the pot.

Lateral load

Lateral loads on the bearing must also be accounted for in design. Lateral load is transferred from the piston to the pot by contact between the rim of the piston and the wall of the pot. The contact stresses can be high because the piston rim may be relatively thin to avoid binding when the piston rotates and the rim slides against the pot. The pot wall must transfer the load down into the baseplate and this is done by a combination of shear stresses in the part of the wall oriented parallel to the direction of the load and cantilever bending of the part in contact with the piston. The loads are then transferred into the substructure through friction under the base of the bearing and shear in the anchor bolts. Lateral loads may also contribute to increased wear of the elastomeric pad and greater potential for wear and fracture of the sealing rings. The damage observed in tests suggest that lateral loads should be carried through an independent mechanism wherever possible.

Design Requirements

The components of a pot bearing that need to be designed are the elastomeric pad, the metal pot and piston and the concrete or grout support. The sealing rings are perhaps the most critical element of all, but they are not amenable to calculation because no adequate mechanical model for their behavior has yet been proposed. In the absence of such a model, there is little choice but to use a type of sealing ring that has performed adequately in the past. As a result, closed circular brass rings and sets of two or three flat brass rings are permitted. The sealing rings of circular cross section must have a diameter no less than the larger of $0.0175D_p$ and 8 mm (0.375 in.), and sealing rings with a rectangular cross-section must have a width greater than at least $0.02D_p$ and 6 mm (0.25 in.) and a thickness of at least 0.2 times the width, where D_p is the internal diameter of the pot.

Elastomeric Pad

Pot bearings are designed for a compressive stress of 25 MPa (3.5 ksi) on the elastomeric pad under total service load. This controls the diameter of the pot and the pad. The pad thickness is controlled by the permissible compressive strain. The required thickness is

$$t_r \geq 3.33\, \theta_u\, D_p \qquad \text{(Eq. 2-13)}$$

where t_r is the pad thickness, θ_u is the design rotation angle of the piston, and D_p is the internal diameter of the pot. This limits the compressive strain in the elastomeric pad due to rotation to 15 percent. The strain may be larger under the sealing ring recess, since the effective thickness of the pad is reduced there. Therefore, the recess for the sealing rings should be shallow relative to the total thickness of the elastomeric pad in order to prevent damage to the thinner elastomer layer below the rings.

The pad should be made of an elastomer with a hardness in the range of 55 to 65 Shore A Durometer, and should provide a snug fit into the pot. The elastomer should be lubricated, preferably with silicone grease, and the pot should be sealed against dust and moisture.

Pot Walls and Base

The pot walls must be strong enough to withstand the large internal hydrostatic pressure in the elastomeric pad. This is ensured if

$$t_w \geq \frac{\sigma_u\, D_p}{2\phi_t F_y} \qquad \text{(Eq. 2-14)}$$

where t_w is the pot wall thickness, σ_u is the factored average compressive stress or hydrostatic pressure in the elastomer, D_p is the internal diameter of the pot, and F_y is the yield stress of the steel. The term ϕ_t is the resistance factor for tension (0.9). Using the normal 25 MPa (3.5 ksi) service stress with a load factor of 2 and a 345 MPa (50 ksi) yield stress for the steel leads to $t_w \geq 0.08D_p$.

The pot wall must be deep enough to assure that the piston does not lift out of the pot under any load or rotation. This results in a clearance requirement as illustrated in Figure II-2.7, and it is best satisfied as a performance requirement based on the design requirements and the geometry of the bearing.

If the bearing is subjected to lateral load, the analysis becomes more complicated. The wall thickness must be a minimum of

$$t_w \geq \sqrt{\frac{62\, H_T \theta}{F_y}} \qquad \text{(Eq. 2-15)}$$

where H_T is the service lateral load (kN), and θ is the service rotation angle (radians) about the axis normal to the direction of load. The wall thickness of the pot is controlled by the larger of the thicknesses produced by Eqs. 2-14 and 2-15. It should be noted that a version of Eq. 2-14 is included in the current pot bearing section of the AASHTO LRFD Specifications and it will control the wall thickness for pot bearings with lateral loads less than approximately 10 percent of the maximum compressive load. However, Eq. 2-15 is rational[14] and will likely be included in future Interim revisions to the AASHTO LRFD Specifications, since it controls the wall thickness when larger lateral loads are present [a Customary U.S. Units version of Eq. 2-15 would use a constant of 40 in place of 62].

The base must be thick enough to resist the moments from the cantilever bending of the wall and so should have a thickness at least equal to that required by Eq. 2-15. In addition, the base thickness should be no less than the larger of $0.06D_p$ and 19 mm (0.75 in.) for a base bearing directly against concrete or grout, and no less than $0.04D_p$ and 12.5 mm (0.5 in.) for a pot bearing base resting on load distribution plates.

In order to minimize the wear on the sealing rings and damage to the elastomeric pad, the inside of the pot walls should be machined to a fine surface finish [e.g., 1.5 micrometers (64 microinches) or better] and should not be metalized. The pot wall should not be metalized because the rough surface damages the piston, sealing rings and elastomeric pad. Corrosion protection should be provided by other means such as lubrication and sealing.

Piston

The piston must have adequate clearance between the rim of the piston and the wall of pot as illustrated in Figure II-2.7 to permit rotation of the bearing without elastomer leakage. This also results in a clearance requirement (illustrated in Figure II-2.7) which is best satisfied as a performance requirement based on the design requirements and the geometry of the bearing. However, a minimum clearance of 0.5 mm is required. Equation 14.7.4.7-2 of the 1994 AASHTO LRFD Specification is an approximate equation for determining the required clearance as a function of rotation and pot diameter. This equation is conservative for most practical cases, but it may also be deficient under some circumstances and is not repeated here.

The piston must be stiff enough not to deform significantly under load. As a minimum the piston thickness must satisfy

$$t_{pist} \geq 0.06\ D_p \qquad \text{(Eq. 2-16)}$$

The piston rim also must be thick enough to carry the contact stresses caused by lateral load, when the lateral load is transferred to the pot through the piston. The rim thickness must satisfy

$$t_{rim} \geq \frac{2.5\ H_T}{D_p\ F_y} \qquad \text{(Eq. 2-17)}$$

Eq. 2-17 is presently not included in the AASHTO LRFD Specifications, but it is likely(14) to be included in the future Interims to the specification. The diameter and shape of the rim should be selected so as to prevent binding of the piston in the pot when it undergoes its maximum rotation.

Concrete Bearing Stresses and Masonry Plate Design

A masonry plate is often supplied below the bearing, although in Europe many pot bearings have been installed without one. However, as discussed in Section 3, the use of a masonry plate may be desirable because it simplifies bearing removal and replacement. The masonry plate must be designed by normal bearing strength base plate design methods. These methods are also used for a wide range of other bridge components and as a result are not summarized here.

Design Example

Design a movable bearing for the following conditions:

Dead Load	2670 kN (600 kips)
Live Load	1110 kN (250 kips)
Lateral Load	330 kN (75 kips)
Rotation	± 0.02 radians

The design rotation falls near the boundary that separates the use of Figures I-2 and I-3 of the *Steel Bridge Bearing Selection Guide* in Part I of this document. Those figures suggests that a pot bearing or a spherical bearing would be viable alternatives. However, Table I-A indicates that the pot bearing has a lower initial cost. Therefore, a movable pot bearing is designed.

Use AASHTO M270 Grade 345W (ASTM A709M Grade 345W) structural weathering steel. A PTFE pad is to be recessed into the top of the piston. The concrete piercap is 1050 mm (3.5 ft) wide; f'_c is 28 MPa (4 ksi).

The diameter of the pot and the elastomeric pad are determined by the maximum stress, 25 MPa (3.5 ksi), permitted on the pad at the maximum load.

$$A \geq \frac{P_D + P_L}{25} = \frac{3780 \text{ x } 1000}{25} = 151 \text{ x } 10^3 \text{ mm}^2$$

or $D_p \geq 439$ mm (use 450 mm). The thickness of the pad is determined by the strain in the elastomeric pad. Eq. 2-13 requires

$$t_r \geq 3.33\, \theta_u\, D_p = 3.33 \text{ x } 0.02 \text{ x } 450$$

$$= 30 \text{ mm (use 30 mm)}$$

The sealing rings are selected to be 3 flat brass rings of width, b_{ring}, and thickness, t_{ring}, where

$$b_{ring} \geq \max(0.02D_p, 6 \text{ mm}) = \max(0.02 \text{ x } 450, 6)$$

$$= 9 \text{ mm (use 9 mm)}$$

$$t_{ring} \geq 0.2\, b_{ring} = 1.8 \text{ mm (use 2 mm)}$$

The total thickness of the three rings is 6 mm (¼ in.). This is less than 1/3 the total thickness of the pad, which is the limit commonly employed to control the concentration in elastomer strain at this location

The piston should have a minimum thickness of $t_{pist} \geq 0.06\, D_p = 0.06 \text{ x } 450 = 27$ mm (use 27 mm). The minimum thickness of the rim, t_{rim}, is

$$t_{rim} \geq \frac{2.5\ H_T}{D_p F_y} = \frac{2.5 \text{ x } 330\ 000}{450 \text{ x } 345} = 5.3 \text{ mm (use 6 mm)}$$

The PTFE must be designed and recessed as required by PTFE design criteria, and the minimum piston thickness will need to consider the loss of thickness produced by the recess.

The pot wall thickness is controlled by the larger of Eqs. 2-14 and 2-15. Vertical load alone, Eq. 2-14, requires

$$t_w \geq \frac{\sigma_u D_p}{2\ \phi_t\ F_y} = \frac{\dfrac{2 \text{ x } 3\ 780\ 000}{\pi\ (225)^2}(450)}{2 \text{ x } 0.9 \text{ x } 345} = 34.4 \text{ mm}$$

and for horizontal load

$$t_w \geq \sqrt{\frac{62\ H_T\ \theta}{F_y}} = \sqrt{\frac{62 \text{ x } 330\ 000 \text{ x } 0.02}{345}} = 34.4 \text{ mm}$$

The pot base thickness is determined as follows

$t_{base} \geq 0.06 \text{ x } 450$ and $t_{base} \geq t_w$

$t_{base} \geq 27 \text{ mm} < 34.4 \text{ mm}$ (use 35 mm)

Thus, the 35 mm thickness controls both the pot base and wall thickness. Masonry plates are selected by the normal concepts for steel bearing on concrete. Figure II-2.8 illustrates the final design for this example.

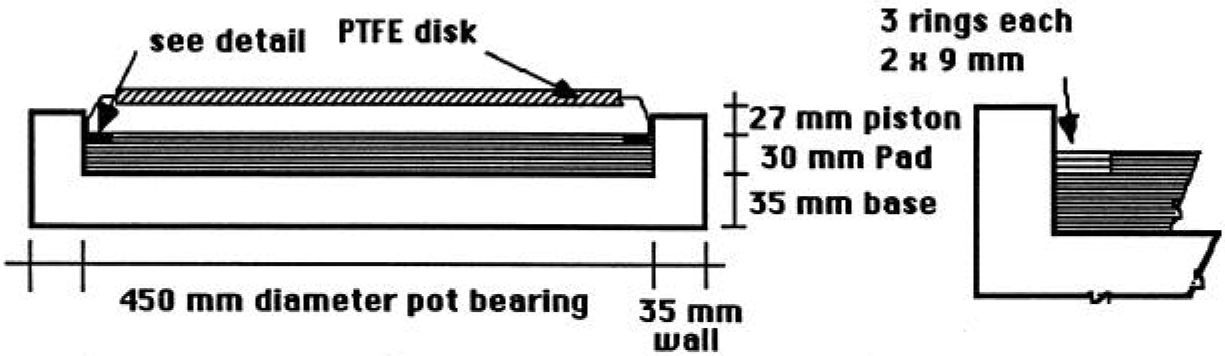


Figure II-2.8: Final Pot Bearing Design

SLIDING SURFACES

General

Lubricated bronze and PTFE (polytetrafluorethylene) sliding surfaces(14) are commonly used as components of bridge bearings. Sliding surfaces develop a frictional force that acts on the superstructure, substructure and the bearing. As a result, friction is an important design consideration. The friction force, *F*, can be estimated by

$$F = \mu N \qquad \text{(Eq. 2-18)}$$

where *m* is the coefficient of friction and *N* is the normal force on the sliding surface. While both lubricated bronze and PTFE are used for sliding surfaces, there are many differences in their behavior, and as a result they are discussed separately.

Lubricated Bronze Sliding Surfaces

Flat lubricated bronze sliding surfaces are used to accommodate very large translations. Cylindrical surfaces as shown in Figure II-2.9 (or spherical surfaces) may be used to accommodate rotation about one (or two) axes. The magnitude of the translation and rotation are limited only by the dimensions of the sliding surface. The displacement may be multidirectional unless guideways or geometric constraints (such as spherical or cylindrical geometry) are provided. The load capacity can be very large since it is limited only by the surface area. The mating surface should be significantly harder than the bronze surface and have a comparable surface finish. The mating surface is normally structural steel and is often supplied by the fabricator.

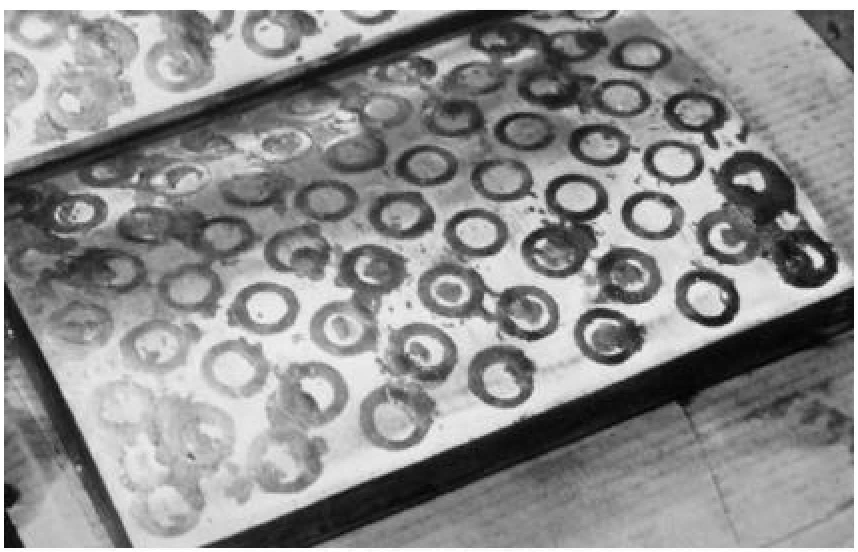


Figure II-2.9: Lubricated Bronze Sliding Cylindrical Surface

Lubricated bronze bearings use a regularly spaced pattern of recesses for lubricant as shown in Figure II-2.9. The recesses are usually in the order of 13 mm (½ in.) deep. Individual bearing manufacturers regard the recess pattern and the lubricant compound as proprietary, but the patterns used by most manufacturers are similar. The recesses are formed by casting the bronze in a mold and then machining to the proper geometry and surface finish. The bronze surface is cut to a fairly smooth but not highly polished finish. The lubricant is placed into the recesses under pressure and projects above the bronze approximately 1.5 mm (1/16 in.). The mating surface grips the lubricant in its asperities and spreads it over the bronze surface as movement occurs. The surface lubrication dissipates with time and movement, and eventually direct contact is developed between the bronze and the mating surface. After this, further movement causes the harder mating surface to abrade the bronze surface.

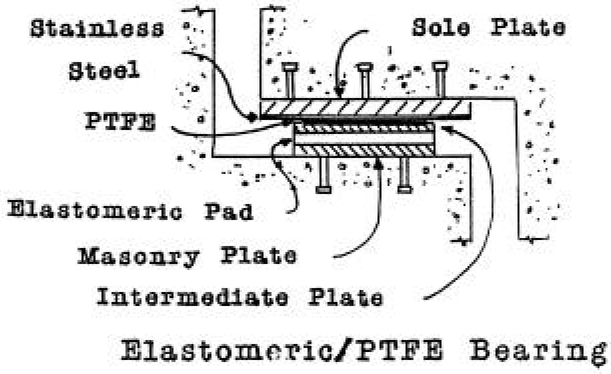
According to the AASHTO LRFD Specifications, the coefficient of friction is typically in the order of 0.07 under initial lubricated conditions, and it increases to approximately 0.1 after the bronze starts to erode. Coefficients of friction in the order of 0.4 must be expected for unlubricated bronze or for lubricated bronze bearings after the lubrication has completely dissipated.

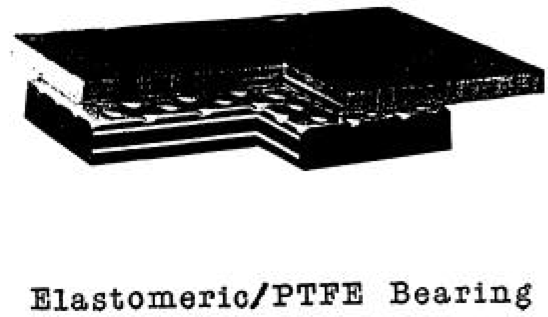
Bronze bearings are economical and do not require the high degree of quality control required for PTFE sliding surfaces. They do not require a highly polished mating surface, nor do they require tight geometric constraints since the material is thicker than typical PTFE and significant wear is expected. However, the frictional resistance may be considerably larger than that achievable with PTFE surfaces.

PTFE Sliding Surfaces

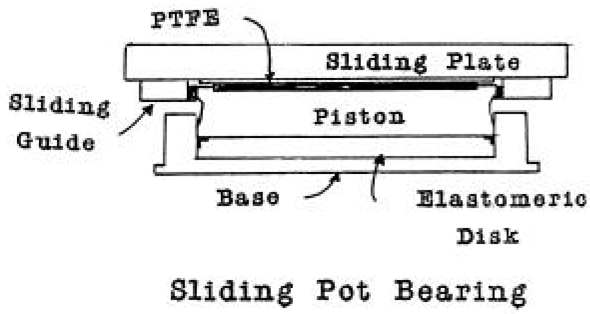
PTFE (polytetrafluorethylene) sliding surfaces as shown in Figure II-2.10 are also used to accommodate large translations and rotations when combined with spherical or cylindrical bearings. These surfaces have similarities with lubricated bronze sliding surfaces, but they may develop substantially smaller friction forces. PTFE sliding surfaces require greater care in design and greater quality control in construction and installation. PTFE is used with mating surfaces made of very smooth stainless steel (for all flat sliding surfaces and many curved surfaces) or anodized aluminum (for some spherical or cylindrical surfaces). The stainless steel surface must be larger than the PTFE surface so that the full movement can be achieved without exposing the PTFE and, whenever possible, the stainless steel is placed on top of the PTFE to prevent contamination with dust or dirt. PTFE sliding surfaces are used in combination with a wide range of other bearing systems.

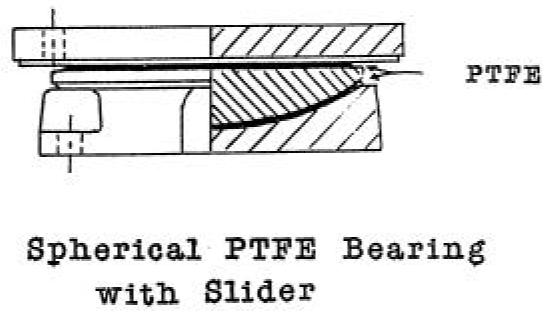
The low frictional resistance(15,17,18,19) of PTFE is its most important characteristic. The coefficient of friction decreases with increasing contact compressive stress between the PTFE and the mating surface. Friction is smaller for static or slowly applied translations than it is for moderate dynamic translations, and it is larger for the first cycle of movement than for later ones. At much higher sliding speeds such as are found in seismic isolation bearings, the friction is considerably higher. The coefficient of friction of PTFE increases at very low temperatures and if the mating surface is rough or contaminated with dust or dirt. The friction is significantly reduced if the interface is lubricated, and it is increased if the PTFE contains filler such as fiberglass. Dimpled PTFE (as shown in Figure II-2.11) is sometimes used to prevent the lubricant from seeping out under cyclic translations.





Elastomeric/PTFE Bearing **Elastomeric/PTFE Bearing**





Sliding Pot Bearing **Spherical PTFE Bearing with Slider**

Figure II-2.10: Typical PTFE Sliding Surfaces

PTFE may creep (or cold flow) laterally when subjected to high compressive stress, and shorten the life of the bearing. The reduction in PTFE thickness may also allow hard contact between metal components. Thus, while the compressive stress should be high to reduce friction, it must also be limited to control creep. PTFE is frequently recessed for one half its thickness to control creep and permit larger compressive stress. Filled PTFE is reinforced with fiberglass or carbon fibers, and it is sometimes used to resist the creep or cold flow.

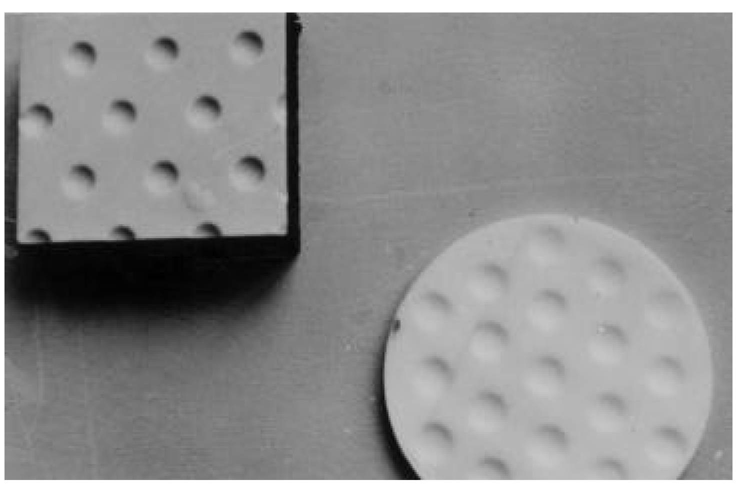


Figure II-2.11: Dimpled PTFE

PTFE is sometimes woven into a fabric or mat and used as a sliding surface in bridge bearings as shown in Figure II-2.12. The woven mat is often placed over a gridlike metal substrate to control creep without increasing the friction. In some cases, the woven mat is reinforced with strands of material that are woven and interlocked into the strands of PTFE, but the reinforcement should not come to the surface. It is recommended that the bridge engineer require certification tests for all types of PTFE to ensure that they meet the design requirements.

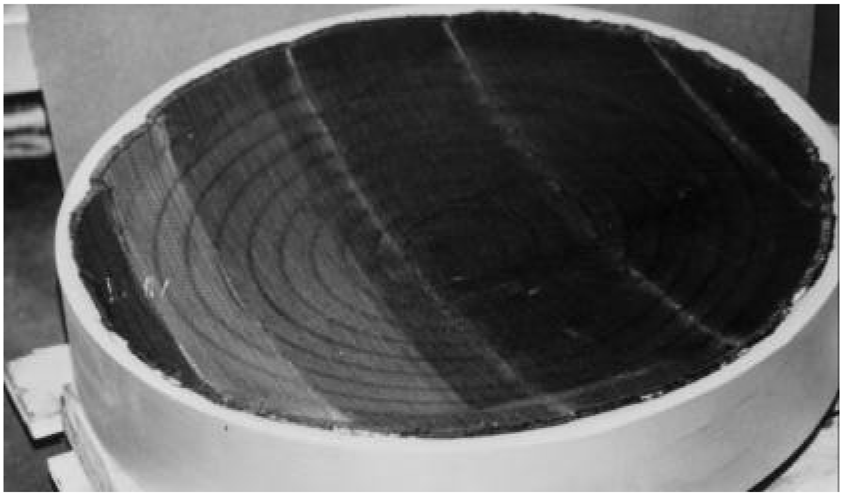


Figure II-2.12: Woven PTFE Sliding Surface

PTFE wears under service conditions and it may require replacement after a period of time. Low temperatures, fast sliding speeds, rough mating surfaces, lack of lubrication, and contamination of the sliding interface increase the wear rate. Relatively thin layers of from 1.5 to 3 mm (1/16 to 1/8 in.) are commonly used in the United States, but engineers in other countries often use thicker PTFE layers 4.5 to 6 mm (3/16 to 1/4 in.) to account for recess thickness and accommodate the potential for wear.

Design Requirements

The coefficient of friction, m, is the most critical design requirement for sliding surfaces. The design coefficient of friction is taken as 0.1 for self-lubricating bronze components and up to 0.4 for other types of bronze sliding surfaces, unless better experimental data is available. The design coefficients of friction are smaller with PTFE sliding surfaces, but m varies widely for different types of PTFE. Table II-B provides the design coefficient of friction values to be used in the absence of better experimental data. Dimpled lubricated, unfilled sheet, woven and filled sheet PTFE are all recognized by the AASHTO LRFD Specification, but all types of PTFE must be made of virgin material.

Type of PTFE	Pressure (MPa)	3.5	7	14	³ 21
	Temperature (°C)	m	m	m	m
Dimpled Lubricated	20	0.04	0.03	0.025	0.02
	-10	0.06	0.045	0.04	0.03
	-45	0.10	0.075	0.06	0.05
Unfilled	20	0.08	0.07	0.05	0.03
	-10	0.20	0.18	0.13	0.10
	-45	0.20	0.18	0.13	0.10
Filled	20	0.24	0.17	0.09	0.06
	-10	0.44	0.32	0.25	0.20
	-45	0.65	0.55	0.45	0.35
Woven	20	0.08	0.07	0.06	0.045
	-10	0.20	0.18	0.13	0.10
	-45	0.20	0.18	0.13	0.10

Table II-B: Design Coefficients of Friction for PTFE

The mating surface for a flat PTFE sliding surface should be Type 304 stainless steel with a #8 mirror finish, and anodized aluminum may be used with some curved sliding surfaces. A slightly rougher #3 finish may be desirable with the woven material. The coefficient of friction data provided in Table II-B are design values that are based on laboratory experiments. They are larger than the average values recorded in the experiments in order to allow for the differences between laboratory and field conditions. Note that Table II-B is different than the table presently used in AASHTO LRFD Specifications, but it is likely(14) to be included in future Interim revisions to the Specifications.

The mating surface for lubricated bronze bearings should be steel, and it should be machined to have a surface finish of 3 micro meters (125 micro inches) rms or better.

The contact stress between the sliding and the mating surface must be checked as an average stress on the projected contact area for both lubricated bronze and PTFE. In addition, eccentricity and edge loading must also be considered for PTFE, where the contact stress at the edge is computed by taking into account the maximum moment and eccentricity using a linear distribution of stress across the PTFE. The average contact stress must be limited to 21 MPa (3 ksi) for most commonly used lubricated bronze. The stress limits for PTFE are controlled by creep and cold flow of PTFE as illustrated in Table II-C. This table is slightly different than the table presently used in AASHTO LRFD Specifications, but it is likely(14) to be included in future changes.

Material	Average Contact Stress (MPa)		Edge Contact Stress (MPa)	
	Dead Load	All Loads	Dead Load	All Loads
Unconfined PTFE:				
Unfilled sheets	14	20	18	25
Filled sheets—these figures are for maximum filler content	28	40	35	55
Confined sheet PTFE	30	40	35	55
Woven PTFE over a metallic substrate	30	40	35	55
Reinforced woven PTFE over a metallic substrate	35	50	40	65

Table II-C. Permissible Contact Stress for PTFE

Attachment and confinement of the PTFE are also design considerations. Sheet PTFE should preferably be confined in a recess in a rigid metal backing plate for one half its thickness. Recessed PTFE must normally be thicker, since half of its thickness is recessed into the steel backing. Woven PTFE is normally attached to a metallic substrate by mechanical interlocking which can resist a shear force no less than 0.10 times the applied compressive force. Sheet PTFE which is not confined must be bonded to a metal surface or an elastomeric layer with a Shore A Durometer hardness of at least 90.

Design Example

As a design example, consider a bearing with the following design loads and movements.

Dead Load	2400 kN (540 kips)
Live Load	1200 kN (270 kips)
Longitudinal translation	±200 mm (±8.0 in.)
Rotation	0.005 radians

The above bearing translations are in the longitudinal direction. The rotation is about the transverse axis. There are no design translations in the transverse direction; translation in this direction is restrained by the stiffness of the bearing. The steel girder has a bottom flange bearing width of 750 mm (30 in.).

Examination of Figure I-1 of the *Steel Bridge Bearing Selection Guide* contained in Part I of this report illustrates that a CDP or a steel reinforced elastomeric bearing with a PTFE sliding surface is a logical alternative. PTFE sliding surfaces are not able to accommodate rotation without some other bearing component, and CDP have limited rotational capacity. Therefore, the very small rotation combined with the relatively large compressive load suggest that the steel reinforced elastomeric bearing combined with a PTFE sliding surface is the most viable. The loads on this bearing are identical to those used for the steel reinforced elastomeric bearing example except that the rotation is now smaller. The steel reinforced elastomeric bearing was 475 mm (19 in.) long by 725 mm (29 in.) wide with a layer thickness of 15 mm (0.6 in.) and a shape factor of 9.57. This same elastomeric bearing will be able to support all of the loads based on calculations described earlier, if it is shown that the steel reinforced elastomeric bearing can tolerate the rotation and the horizontal translation is accommodated by a PTFE sliding surface.

The average compressive stress under maximum loading as determined previously is 10.45 MPa (1.5 ksi). The elastomeric bearing can tolerate the rotation if it satisfies

$$\sigma_T = 10.45\text{ MPa} \le 1.875(0.7)9.57\left(1 - 0.20\left(\frac{0.005}{n}\right)\left(\frac{475}{15}\right)^2\right)$$

if *n* is equal to 7, then the condition becomes

$$\sigma_T \le 1.875(0.7)9.57\left(1 - 0.20\left(\frac{0.005}{7}\right)\left(\frac{475}{15}\right)^2\right) = 10.76\text{ MPa}$$

Therefore, 7 layers will be adequate. The total elastomer thickness would be 105 mm (4.1 in.), and the shear deformation limit indicates that this elastomeric bearing could tolerate a maximum translation of 52 mm (2.0 in.). Thus, a PTFE slider must be used to accommodate at least 148 mm (6.0 in.) of translation and preferably the entire 200 mm (8 in.).

The PTFE could be attached in several ways. Two of these options are illustrated in Figure II-2.13. A steel plate could be vulcanized to the top of the steel reinforced elastomeric bearing, and the PTFE could be recessed into the plate. Under these conditions, the actual contact area for the PTFE would be smaller than the plan area of the steel reinforced elastomeric bearing and the coefficient of friction achieved with the PTFE would be smaller than if the PTFE covered the entire area. It will be necessary to ensure that the top plate is stiff and strong enough to accommodate the load transfer. As an alternative, the PTFE could be directly bonded to the top cover layer of elastomeric bearing. The top cover layer of the elastomer must be very hard (90 Durometer) for this arrangement. This second option is likely to produce a somewhat more simple and economical bearing and attachment detail, but it will result in slightly larger coefficient of friction and slightly inferior overall behavior.

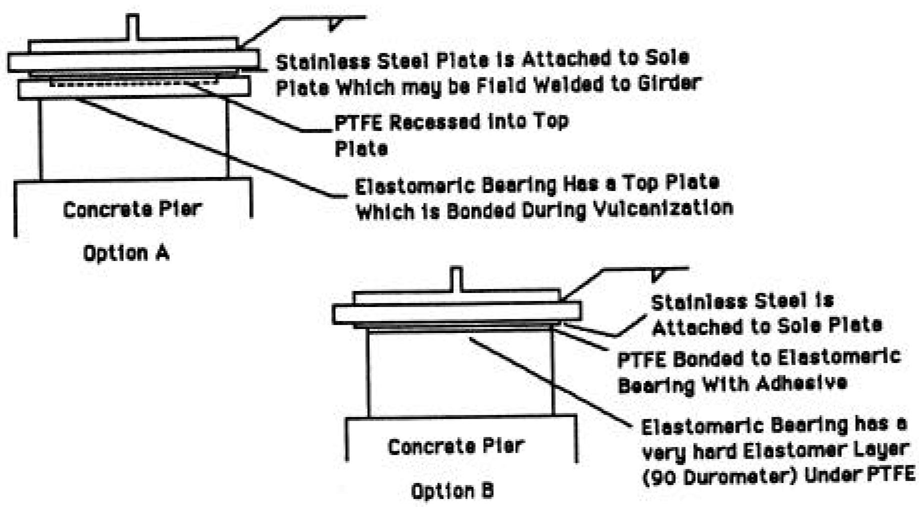


Figure II-2.13: Two Options for the Attachment of A PTFE Sliding Surface to a Steel Reinforced Elastomeric Bearing

The second option is selected here and it is to be used with flat, dry sheet unfilled PTFE. By interpolation of the data in Table II-B, this application achieves a coefficient of friction of approximately 0.06 at room temperature. Larger friction must be expected at very low temperature. The average contact stress under full loading is 10.45 MPa (1.5 ksi) and under dead loading 6.97 MPa (1.0 ksi), and Table II-C shows that these are well below the limits for unconfined PTFE for control of creep and cold flow. The maximum force transferred by the sliding surface at room temperature should be in the order of

$$F = 0.06\ (3600) = 216 \text{ kN}$$

If this force is unacceptable to the structure or substructure, another type of PTFE could be employed or the alternative attachment procedure could be used. The edge contact stress must also be checked. It should be recognized that part of the movement will be taken up by deformation of the steel reinforced elastomeric bearing, and part by sliding action. The deflection of the elastomeric bearing can be estimated as

$$\Delta_s = \frac{F \text{ x } h_{rt}}{G \text{ x } W \text{ x } L} = \frac{216\ 000 \text{ x } 105}{0.7 \text{ x } 725 \text{ x } 475} = 94 \text{ mm}$$

This 94 mm (3.7 in.) deflection exceeds the allowable shear deformation of the steel reinforced elastomeric bearing acting alone, and is cause for rejecting the proposed system. The force and the elastomer deformation may be reduced by using a PTFE with a lower coefficient of friction. A dimpled lubricated PTFE will have a coefficient of friction less than 0.03. This will produce a maximum friction force of 108 kN., and the elastomeric bearing deformation will be below acceptable limits. As an alternative, a stiffer elastomer could also be employed. It can be shown that, in order to satisfy deflection limits of Eq. 2-5b

$$\mu \leq \frac{0.5\, G}{\sigma_T}$$

The maximum resisting moment about the transverse axis, M_x, of the elastomeric bearing can be estimated at the maximum service rotation, q_x, by the equation

$$M_X = (0.5\ E_C I)\theta_X / h_{rt} \qquad \text{(Eq. 2-19)}$$

where I is the moment of inertia of plan shape of bearing, and E_c is the effective modulus of the elastomeric bearing in compression. The values of I, q_x and h_{rt} are reasonably clear. The value of E_c can be estimated by the equation[(8)].

$$E_c = 3G\left(1 + 1.3\ S^2\right) = 3 \text{ x } 0.91 \text{ x } \left(1 + 1.3(9.57)^2\right)$$
$$= 328 \text{ MPa}$$

It should be noted the E_c is sometimes conservatively approximated as $6GS^2$ in these stiffness calculations. Thus,

$$M_x = \left(0.5\ E_c I\right)\theta_x / h_{rt}$$
$$= 0.5 \text{ x } 328 \text{ x } (1/12) \text{ x } 725(475)^3 \frac{0.005}{105}$$
$$= 50.57 \text{ x } 10^6 \text{ N-mm}$$

The maximum contact stress on the edge is then

$$\sigma = 10.45 + \left(\frac{50.57 \text{ x } 10^6}{(1/6) \text{ x } 725(475)^2} \right) = 12.3 \text{ MPa}$$

This is well below the allowable stress due to edge loading listed in Table II-C. Similar calculations would be used to account for the moment about any other axis.

The mating surface should be Type 304 stainless steel with a #8 mirror finish or better. The stainless steel should be long enough that the full 200 mm (8 in.) translation can be accommodated in each direction without exposing the PTFE, although a significant part of the total translation will be accomplished by elastomer deformation. In addition there should be adequate freeboard, say 50 mm (2 in.) at each end, to cover uncertainties. Thus, the total length of the stainless steel mating surface should be 975 mm (39 in.), and the stainless steel should be centered over the initial zero movement position of the bearing.

Summary

Lubricated bronze and PTFE sliding surfaces can support a wide range of compressive loads and accommodate large translations if they are properly designed. Movements in excess 1000 mm (39 in.) are possible. These bearings can accommodate rotation only if machined into a curved (spherical or cylindrical) surface or if combined with another bearing component such as a steel reinforced elastomeric bearing or pot bearing. A lubricated bronze sliding surface is a relatively robust system and is less sensitive to abnormalities than a PTFE sliding surface, but it induces larger forces into the structure and substructure. Less care is required in the design and manufacture of lubricated bronze sliding surfaces than in comparable PTFE sliding surfaces, but they are less versatile in achieving many design objectives. A PTFE sliding surface is frequently used in conjunction with other bearing systems. For example, PTFE is often used as a sliding surface in conjunction with steel reinforced elastomeric bearings as shown in Figure II-2.10 to accommodate large translations. PTFE sliding surfaces may also be used on top of a pot bearing to allow lateral displacement.

BEARINGS WITH CURVED SLIDING SURFACES

General Behavior

Bearings with curved sliding surfaces include spherical and cylindrical bearings, and they are special cases of lubricated bronze or PTFE sliding surfaces. Figure II-2.9 illustrates a cylindrical bearing of lubricated bronze, and Figure II-2.12 shows a spherical bearing with a woven PTFE sliding surface. They are used primarily for sustaining large rotations about one or more axes. The rotation occurs about the center of radius of the curved surface, and the maximum rotation is limited by the geometry and clearances of the bearing.

Spherical and cylindrical bearings may develop horizontal resistance by virtue of the geometry. This lateral load capacity is limited and large lateral loads require an external restraining system. Spherical and cylindrical bearings rotate about their center of radius, and they are fixed against translation. The center of rotation of the bearing and the neutral axis of the beam seldom coincide, and this eccentricity introduces additional translation and girder end moment which must be considered in the design. An additional flat sliding surface as shown in Figure II-2.14 must be added if the bearing is to accommodate displacements or to reduce the girder end moment. The moment, *M*, on the end of the girder can be estimated by

$$M = \mu N\ d \qquad \text{(Eq. 2-20)}$$

where *m* is the coefficient of friction, *N* is the normal load on the projected area of the bearing, and the moment arm, *d*, is the distance between the center of radius of the bearing and the center of rotation of the girder. This additional moment must be considered in the design of the bearings, the superstructure, and the substructure. However, the end moment cannot be used to develop continuity or restraint for the piers or the girders, since it will change with time.

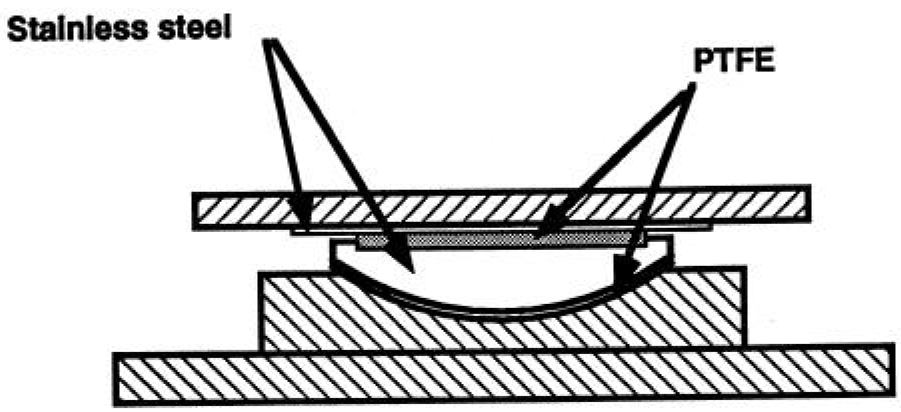


Figure II-2.14: Flat Sliding Surface Used in Conjunction with a Curved Sliding Surface

The inside and outside radii of spherical and cylindrical bearings must be accurately controlled and machined to assure good performance. When using PTFE, a small tolerance between the two radii and a smooth surface finish is required to prevent wear, creep or cold flow damage due to nonuniform contact and to ensure a low coefficient of friction. A realistic estimate of the thickness of the PTFE under load is also necessary. Tolerances for lubricated bronze bearings are less critical because some wear of the bronze is expected. However, the tolerances must be tight enough to prevent fracture of the bronze due to point or line contact on the steel mating surface.

Design Requirements

The design of bearings with curved sliding surfaces uses many of the parameters required to design flat sliding surfaces. The coefficient of friction, *m*, and allowable contact stresses are the same. The moment transferred by the curved surface about its center of rotation is given by the friction force multiplied by the lever arm. For curved sliding bearings with a companion flat sliding surface

Error! AutoText entry not defined. $= \mu\ P\ R$ (Eq. 2-21a)

and for curved sliding bearings without a companion flat sliding surface

$$M_u = 2\ \mu\ P\ R \qquad \text{(Eq. 2-21b)}$$

The allowable contact stresses are applied over the projected area of the curved surface, so that for cylindrical bearings

$$\sigma_T = \frac{P}{B\,W} \qquad \text{(Eq. 2-22a)}$$

and for spherical bearings

$$\sigma_T = \frac{4\,P}{\pi\,D^2} \qquad \text{(Eq. 2-22b)}$$

where s_T is the maximum total stress due to maximum loading on the projected area, *D* is the diameter of the projection of the loaded surface, *W* is the length of the cylinder and *B* is the plan dimension of the cylindrical surface perpendicular to its axis.

The lateral load capacity inherent in a curved bearing without an added restraint can be determined from

$$H_u \le 2\,R\,W\left(\sigma_{PTFE}\right)\sin\left(\phi - \beta - \theta_u\right)\sin\beta \qquad \text{(Eq. 2-23a)}$$

for a cylindrical sliding surface where

$$\beta = \tan^{-1}\left(\frac{H_u}{P_D}\right) \qquad \text{(Eq. 2-23b)}$$

and

$$\phi = \sin^{-1}\left(\frac{D}{2\,R}\right) \qquad \text{(Eq. 2-23c)}$$

where *D* is the projected length of the sliding surface perpendicular to the rotation axis, *R* is the radius of the curved sliding surface, H_u is the maximum factored horizontal load, P_D is the factored compressive dead load, b is the resultant angle of the applied loads, f is the corresponding subtended angle of the curved sliding surface, s_{PTFE} is the maximum average contact stress permitted on the PTFE for all loads from Table II-C and *W* is the length of the cylindrical sliding surface. For spherical bearings

$$H_u \le \pi\,R^2\left(\sigma_{PTFE}\right)\sin^2\left(\phi - \beta - \theta_u\right)\sin\beta \qquad \text{(Eq. 2-23d)}$$

It should be noted that these equations are different than those presently appearing the AASHTO LRFD Specifications, but those equations are in error and will be corrected in future Interims to the Specifications.

Summary

Cylindrical and spherical bearings tend to be relatively costly bearings which become a practical choice primarily when the gravity load or the required rotation is large. They are able to support loads up to several thousand tonnes and may accommodate rotations of more than 5 degrees if the bearing is properly designed and constructed. They are likely to be more expensive than a pot bearing, but they can be designed to tolerate larger rotations than pot bearings. As with pot bearings, translational displacements require the addition of a flat sliding surface.