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# Mechanical Details in Detail

This paper explores examples of typical details found in most movable bridge machinery installations. The current standard specifications, and shop practices are discussed for example keys, keyways, turned bolts, force fits, and other mechanical features that are encountered in the drafting and construction of movable bridges. Often these seemly minor details create major problems for the detailer, machine shop and, if not done properly, the owner. There are numerous interpretations of the industry, movable bridge and railroad standards that do not become evident until a project is well underway. The result is usually an unnecessary delay to sort out what the owner and engineer want vs. the common practice and capabilities of the shop doing the work. The objective of this paper is to highlight some of the issues and conflicts and illustrate some examples that should provide at least a template for avoiding delays in drawing approvals and shop work.

## Turned Bolts:

The current AASHTO standard gives the requirements for fasteners, turned bolts and nuts in the same paragraph, 6.7.15. It basically says that bolts should conform to ASTM A325 and that turned bolts have a shank 1/16 larger than the tread major diameter.

Turned bolts are not standard items. They are made to order for the job. The dimensions and finishes are shown on the shop drawings. A turned bolt can be made in several different ways. A standard bolt can be used and the treads cut off, A “blank” bolt can be made, or hex bar can be used. The most common approach is to use a blank bolt. This is a forged bolt with a head but no threads. The most common have a head sized in proportion to the blank and are fairly commonly available. Special blanks with non standard head sizes can be obtained but are usually made to order and have longer delivery times. Occasionally a requirement for fastener head sizes to be the same as the nut ends up in a job that uses turned bolts. This requirement is impractical when turned bolts are involved. Special blanks would have to be made with heads smaller than the standard for the body size. As mentioned above, this increases delivery times and cost. Furthermore it creates a bolt with less than normal head bearing area. While the lower area may not be a problem, the requirement should not be applied to turned bolts.

For particularly large diameter or long shanks, the blank size may need to be substantially larger than the final thread size to assure that the full length of the body cleans up in the machining process. AASHTO 6.7.15 states that the blank size is “usually” 1/8 inch larger than the thread size. As an example, for a one inch treaded sized turned bolt, the shank would finish to 1-1/6 in diameter per the specification. Therefore if the blank is 1-1/8 inch diameter, the machine stock on the shank would be only 1/32 inch per side. For a cap bolt in a pillow block or a bolt holding a rack to a rack support, the shank might be over 12 inches long. There would be a high probability of the blank being slightly bent or bowed or have a local scratch or blemish that would not machine completely out if only 0.03125 inches of stock is allowed. The designer must keep this in mind when reviewing drawings. A shop may opt to allow more stock on the blanks to avoid scraping blanks or sending them back if they are not perfect.

The other consideration is that larger blanks means larger heads. Both the designer and the fabricator/detailer need to check for wrench and head clearance and account for heads that in some cases will be substantially larger than the nut size. Few manufactured components are made with oversize heads in mind. Electric motors and standard manufactured gearboxes and brakes rarely have sub-drilled or under sized holes in them. Therefore to mount them with turned bolts, the shop must ream the production holes to a larger size. The result is the heads of the turned bolts will be much larger than what the manufacturer of the motor or brake intended. Sometimes the head is too big for a spot face or weld clearance or wrench access is limited. This issue is one that can cause delays late in the job and at the worst possible time.

As stated above, few manufactured items, electric motors in particular, are made with in place reaming of mounting holes and installation of turned bolts in mind. Motors all have standard frame sizes including mounting feet and hole sizes. Few, if any motors have feet that extend beyond the frame for reaming from above. Even if the support is fabricated without holes, the motor must be placed, holes transferred and then removed to drill and ream the holes separately. It is possible in some situations to ream from below. The process is time consuming and error prone.

Many industrial applications simply drill dowel pin holes through the base of the motor and support after alignment. The dowel holes can be drilled on an angle if necessary to prevent having to move the motor off the support after alignment.

AASHTO Table 6.7.8-1 gives the fits and finishes required for various mechanical elements. The fit for a turned bolt in a finished hole is LC6. AREMA Table 15-6-5 calls for the fit of turned bolt in a hole to be LT1. Assuming that most turned bolts will be between ½ and 3 inch in diameter, the LC6 total fit is as little as 0.0006 in clearance for the ½ inch bolt to as much as 0.006 inches clearance for the big 3 inch bolt. While the ANSI standard is scaled somewhat by diameter, there is no mention in any of the standards of any variations in the fit due to length. The considering the tolerances of the bolt, the hole, the depth of the hole, the possible limitations and constraints on the access for assembly, these are very difficult to install. Now look at the AREMA LT1 fit for the same sizes, ½ inch is 0.0002 INTERFERENCE to a maximum of 0.0015 inch clearance for a 3 inch diameter turned bolt! These types of fits are extremely costly and time consuming to make. The effort and time and cost must be weighed against the benefits.

Neither of the specifications take into account the length of engagement of the turned bolts or the number of turned bolts involved in a particular connection. A bascule rack segment mounted in a rack support has historically been mounted with many turned bolts. The bolts often extend through the support plates and the entire width of the rack. The bolt lengths are quite long and on the order of 12-16 inches. There are usually bolts the entire length of the segment sometimes in a staggered pattern totaling 50 – 100 bolts in the connection. The active body fit of the turned bolt is only the ends of each bolt where it goes through the support plates and the first inch or so of rack material. Is it necessary to hold the LC6 or LT1 fit the entire length of the bolt? No. Is the bolt drawing going to be approved if it has different tolerances in the middle? Maybe. With 50 bolts in one connection with about 0.001 inch of "play" in a 12 inch long hole the statistical likelihood of the rack being able to move is nearly impossible. The likelihood of getting three or four of the bolts stuck is highly probable. Is the cost and time required to meet the spec in this application really necessary? No.

Given the hole clearance discussion above, hopefully it is obvious that turned bolts should never, ever be used as cap screws where the bolt has to be rotated from the head to go into a threaded hole with a LC6 or LT1 tolerance.

Often turned bolts get treated as structural bolts by misapplication of general standard specifications. Typical specs that are applied are; thread stick through maximums, torque values, thread length, threads in the grip, coatings, head stamp requirements, washer requirements, and ro-cap tests. None of these specifications should be applied to turned bolts. Turned bolts are typically used in shear connections for mounting machinery elements. Shims are almost always used in the grip. It is impossible to make one turned bolt fit every combination of shims if there are certain maximum thread stick through requirements or conversely a prohibition of threads in at least a portion of the grip. It is impossible to make a turned bolt meet the thread length dimensions of an A325 bolt and still have enough thread to account for most shim combinations and double nuts. If turned bolts are required to be tensioned or torqued, then the design engineer needs to provide the values in the contract documents and the method to be used to accomplish the desired tension. Typical turned bolts with larger bodies and heads don't fit in a standard Skidmore testing machine. Most shops don't account for making additional bolts for testing and ro-caps. Tension values for 7/8 inch diameter structural bolts connecting two relatively thin, blasted and primed plates do not correlate to a 1-1/2 diameter, 10 inch long turned bolt going through the 3 inch thick machined base of a reducer, 1-1/2 inch of stainless shims, and a 2 inch thick machined base plate. These are not friction connections. How can you tension any bolt with double nuts? What torque goes on the second nut? Either spell it out or allow all turned bolts to be "snug tight".

## Keys and Keyways

Keys and keyways have been in use for securing hubs on shafts for thousands of years. You would think that by now we would have this perfected. It's not. First of all the key must be drawn correctly on the detail drawings. For a square or rectangular key, the depth of the keyway in the shaft and hub is not measured at the center of the key, it is measured at the edge. This can be somewhat confusing since we normally think of half of the key in the shaft and half in the hub. With the curve of the shaft, it is natural to think of there being ½ inch of a 1 inch high key in the hub at the center of the key and ½ inch in the shaft at the center. In fact this is not the case due to the curve of the shaft. Looking at the example detail, you can see that the key should be half in the shaft and half in the hub at the sides of the key not the center. This is logical because the sides are where the load is and therefore an equal distribution of the forces from the shaft to hub is achieved with this arrangement. The generally accepted standard for sizing and tolerancing keys and keyways is ANSI B17.1. Unfortunately, AASHTO does not directly reference this standard for fits and sizes, only for corner radiuses. Actually, ANSI B17.1 does not require use of corner radiuses and chamfers, it simply provides a suggested table for them "when used…..as a guide". AREMA states that "Details of keys and keyways shall conform to ANSI B17.1 except for fit…." Of course the whole point of ANSI B17.1 is to establish tolerances and fits. As mentioned above, it does not require filleted keyways or chamfered keys so it these details are required, please make it clear. For simplicity in machining and measuring the depth of keyseats and keyways, the dimension from the bottom of the keyway to the back of the shaft and from the opposite side of the bore to the bottom of the keyway is used on detail drawings to establish the depth. These dimensions are given in ANSI B17.1 and the formulas for calculating them. This is the simplest and most direct way of measuring the depth of a keyway. The curvature of the bore prevents using a depth mic.

Now that we have the dimensional details, we must address the tolerances and the fits. Neither AASHTO nor AREMA use the ANSI standard for the fit of the keys. AREMA references the fit and finish table 15-6-5 which simply states FN2. It does not differentiate between the height and width of the key. AASHTO uses table 6.7.8-1 that specifies FN2 fit on the sides and LC4 fit top and bottom. Neither of these specs is consistent with ANSI B17.1 or practical. In fact, if the AREMA specification is taken to the extreme and keys are set with FN2 fits all the way around, the hubs on some products may be over stressed simply from the keys. No coupling manufactures that I'm aware of require any interference on the height of the key. Even with the AASHTO standard of LC4 the bottom line is that use of the ANSI table is much more complete and practical.