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MECHANICAL DETAILS IN DETAIL

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Introduction

This paper explores examples of typical details found in most movable bridge machinery installations. The current standard specifications, and shop practices are discussed for 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 movable bridge, railroad, and industry 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. Most have a head sized in proportion to the blank and are readily available. Special blanks with non standard head sizes can be obtained but are 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 large diameters 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 1-1/16 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 head interference 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 extended 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. Either way, the process is time consuming and error prone.

Many industrial applications use standard bolts in clearance holes and 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. Considering the relatively low torque of the motors, this approach should be given consideration if access is limited. Hard fast adherence to the turned bolt requirement may not produce the best 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. 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.

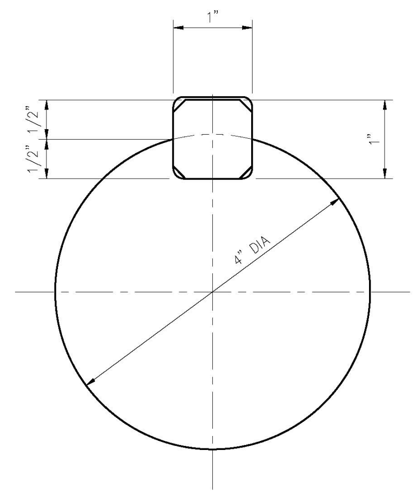
Given the hole clearance discussion above, hopefully it is obvious that turned bolts should NEVER be used as cap screws where the bolt has to be rotated from the head to go into a threaded hole with a LC6 or LT1 tolerance.

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, 1/2 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".

Turned bolts are essential to guaranteeing the alignment and security of the large elements of heavy movable structures. Particularly given the lifespan of these drive systems. The comments and suggestions above are intended to assist the engineer, manufacturer, millwright, and ultimately owner in getting the best service from quality installations. It is in the best interest of all involved that the full scope of the tasks be clear from the onset and rational application of the specifications be considered for the specific application. As is the case with many specifications, the current standards do not address every detail or every application. Clearly the ASTM standard for A325 is for standard structural bolts and cannot be applied to custom made turned bolts in any way except possibly similar material properties. Blind adherence to the spec when reviewing shop drawings or installations is not always the best approach.

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 tolerances of 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 if 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



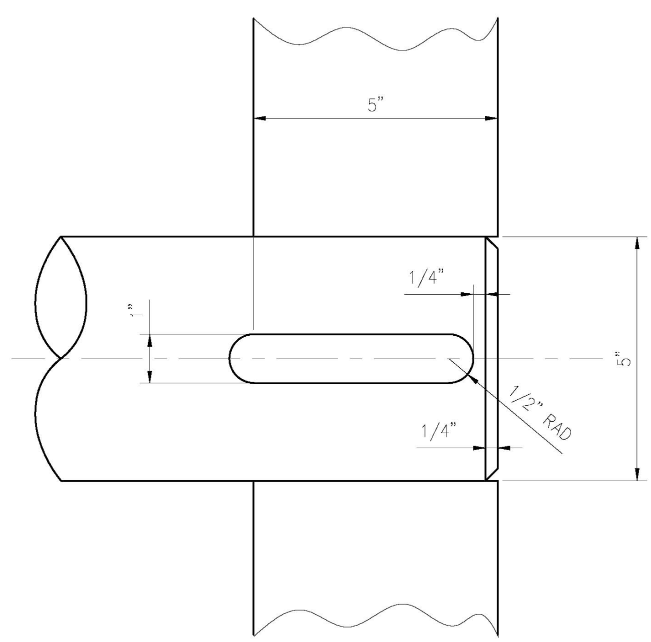
EXAMPLE KEY DETAIL

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 manufacturers that I'm aware of require any interference on the height of the key or the side fits. The issues with interference fits on the sides of the keys become glaringly evident during assembly of the components. Both AASHTO and AREMA require that the hub and shaft have FN2 interference fits. Therefore the bore in which the keyway is cut must be heated to assemble the parts. The behavior of the keyway during heating is not always predictable. It is also very difficult to measure the width of a keyway inside a five inch bore when it is 250 degrees or more. Cooling the key is not very practical as the amount of shrinkage is fairly small for small section parts. Also the key will warm up fast once installed in the keyway. With closed end keyways, the key must be installed in the shaft first. This may require heating the shaft and/or cooling the key. As an example, a FN2 fit on a one inch wide key is 0.002 inches interference. The temperature difference required to shrink a one inch key, 0.002 inches is 300 deg. F. Therefore, to simply get enough clearance to install the key in the keyseat, would require the use of liquid nitrogen. Even then the key would be metal to metal to just barely loose. A long key in an awkward location would likely get stuck and quickly warm to the point where the key will be destroyed and the keyseat damaged in the removal process. After making a new key and eventually successfully freezing it into the shaft, we have to go back to the issue of the key and keyway in the hub. If we heat the hub to a temperature difference of 250 deg, we would have enough clearance to install the hub on the shaft but not enough for the key to go into the keyway. The result is that the hub would get stuck on the key and then warm up and be permanently stuck in the wrong location. The bottom line is that use of the ANSI table which allows clearances on both the sides and top of all keys is much more complete and practical. There are thousands of industrial applications including huge conveyor systems, steel mill rolls, mining applications, cranes etc. all over the world that run for thousands of hours with keys made and installed to ANSI standards. There are also hundreds of movable bridges built using the previous standards which allowed clearances that operate every day without issue.

If the ANSI tables are used, there still must be some reason and logic applied to the use and application of the tables. It is clear from the description of the standard that the intent is to use as rolled bar stock material for the keys. As such, the tolerance tables are divided into classes depending on the type of bar stock used. AASHTO requires the use of forged material for keys. Therefore, bar stock cannot be used and keys must be machined all over from forged bars. AASHTO also requires that the keys be full feathered (ends rounded) to fit the entire keyway. This is a good practice that eliminates pockets for debris and corrosion to collect. This also defines the sides and top of the keys. Since the keys are going to be machined precisely, the tighter class 2 fit tables in ANSI can be used. These fits still allow a small amount of interference on the sides of the keys. But the the tolerance also allows some clearance such that a key can be made that will assemble without the use of heat or cold. Also important is that the tolerance for the keyseats and keyways is generous enough to allow reasonable clearances on top of the keys. This makes the process of cutting the keyways much less precise in depth and much more economical. The width of a keyway cut in a hub is governed by the width of the tool. The tool width can be precisely ground and measured before the cutting starts to assure a good width tolerance on the finished keyway. The depth of the keyway is determined by the feed of the tool by the machine. This process is not as precise and more importantly, the depth is difficult to measure with the part on the machine. The result is that the operator has to “sneak up on” the depth to meet a tight depth tolerance such as that required by current AASHTO and AREMA standards. The hubs are cut and then removed from the machine, measured across the bore, then put back in the machine and set up again and cut some more then the process is repeated until the tolerance is met. Remember, you can always cut more metal but you can’t put it back. The benefit of having a tight tolerance on the top of a key/keyway is simply not worth the cost, effort, and potential errors required to meet it. There is one other area in the ANSI charts that must be applied carefully. As mentioned above, the chart is based on stock key material. Therefore the chart tolerance for the top and bottom and side to side tolerance for a square key is the same due to the fact that the key could be installed either way. In bridge applications with fully machined keys and rounded ends, the sides are clearly defined. Therefore the need to tightly tolearance the top and bottom of the key is not necessary. The charts clearly give ample tolerance to the keyways and seats to allow for the key to be much more generous tolerances top/bottom. It is suggested that for bridge applications, the top/bottom tolerance for all keys be that shown in the table for rectangular keys not the square key tolerances. Since the rectangular key, like all keys for bridges can only be installed one way, the tolerance for top/bottom of the key is +/- 0.005 inches.

Another AASHTO required detail that is of questionable benefit is a “preference” for closed end keyways. This detail causes the machining and inspection process for cutting keyseats in shafts to be more costly and error prone. A closed end keyway requires the machinist to plunge into the shaft before cutting longitudinally along the keyway. This process creates more stress on the cutting tool, limits the types of tools used, and if not done properly is much easier to create a keyway that is out of tolerance in width due to tool deflection and push-off. With a keyway that is open on the end, the machinist can use a tool that only cuts on the sides, can be set at proper depth before entering the material and is only deflected in the longitudinal direction. Since the keyway can be observed and measured from the end, the depth can be easily measured to prevent cutting too deep. Similarly, when the keyway is finished and inspected, the depth and width can be easily checked on the open end. With a closed end keyway, it is much more difficult to get a micrometer into the keyway and around the shaft. Sometimes gage blocks or other tricks must be used. This introduces opportunity for error as the inspector must do calculations to obtain the final number and the size of the gage block could be read incorrectly or the additional parts could introduce error in the readings if the blocks are not fully seated and securely held to the shaft. In addition to the potential problem with manufacture, a closed end keyway reduces the effective length of the key. Since the key has to be rounded on both ends, the bearing area of the key is reduced by a length equivalent to half the thickness of the key plus the amount the keyway is set back from the end of the shaft. As an example, a 1 inch wide key used on a 5 inch long hub would have a shear area of 1 x 5 = 5 square inches if the keyway was allowed to extend to the end of the shaft. If the keyway is closed, the area would be reduced by ½ + 1/4+1/4 = 1x1= 1 square inch of lost shear area. This is based on the loss of length due to the rounded end (1/2 inch) plus the set back of the keyway from the chamfer on the shaft (1/4 in) plus the size of the chamfer (1/4 in) So the net result is that in this example, the capacity of the key is reduced by 20% just because closed end keyways were required. It is understandable that the standards do not want the keys to fall out. However, there are other ways of making sure the keys do not come out which



EXAMPLE CLOSED END KEYWAY DETAIL

are much less costly and do not reduce the overall strength of the key. Other benefits of an open ended keyway are that the fit of the key in the shaft and hub can be easily inspected even with the hubs still on the shafts. Also any distress to the key or keyway would be immediately evident by simply looking at the end of the shaft. Closed end keyways hide a multitude of potential variations due to the fact that the fits are difficult to observe and inspect. Why pay more for something that reduces the strength, and is difficult to inspect?

Keys on shaft ends are most often used in coupling connections. The hubs of the shafts are usually only a small distance apart. Therefore the keys can't go anywhere even if they did back out. In fact the typical coupling gap is no more than 3/8 inch. If a 1in wide key backed out 3/8in, the capacity of the key would still be greater than if the end had been rounded and the keyway closed. Furthermore, if the key did get loose and extend into the gap, it could be easily seen during inspection. The inspector could identify the key as loose and corrective measures could be taken. If the key is captured in the shaft, the looseness would not be detected until a failure occurs. If the keyway is on the end of a shaft that does not have a coupling, then a keeper plate could be installed very inexpensively that would prevent the keys from moving without reducing the capacity or substantially increasing the cost. Regardless of whether the key is for a coupling or other item the simple task of peening over the ends of the keyway will contain the key in all situations. The other option is to use set screws as described in AASHTO.

The other major issue with closed end keyways comes into effect when two keys are needed in one hub. As per the standards, these are cut 120 deg apart. Typically the hubs are installed with an FN2 fit on the shaft. In order to make sure the keyways in the hub line up with the shaft, the keyways are cut in the shaft first. Then one keyway is cut in the hub. Then the hub is held up to the end of the shaft and a key is installed half in the shaft and half in the hub to align the one keyway with one of the keyways in the shaft. Then the second keyway in the shaft is used to scribe the location of the second keyway to be cut in the hub. Obviously this process is much more difficult if the keyways in the shaft are closed end. Once the second keyway is cut in the hub, then the hub is heated or the shaft frozen for assembly. Due to the very tight fits of the keys in the keyways, and the uncertainty of the angular location of the keyways in the shaft relative to the keyways in the hub, it is virtually impossible to assemble a hub on a shaft with both keys in position as is required with closed end keyways. If double keyways are used, at least one must be open ended for installation of the key after assembly. Even if the tolerances and positions were perfect, the temperature effect as the bore is heated would change the relative position of the keyways such that one key would hang up as the hub is installed. One key must be left out of the shaft during installation. Therefore at least one keyway must be open. If we are going to leave one keyway open, why not make them all open?

The current standards have created the "perfect storm" for hub/shaft/key connections. The interactions of the requirements are probably well intentioned to give the best possible connection. However, the result of piling all the requirements together creates a situation that unnecessarily drives up costs, increases the opportunity for errors and delays, and reduces the capacity of the connection. The requirement for an FN2 fit of hubs on shafts is pivotal to the conflicts. If all hubs were slip fits, tight keys and closed end shafts would make sense. If the hubs are truly fitted with FN2 fits then keys are not really necessary and the fit and location become irrelevant. A certain level of redundancy is understandable though. The fit of the hub to the shaft is by far the most important requirement. Ironically the torque capacity of the fit is not addressed at all in the AASHTO standard. By calculation and demonstration, an FN2 fit torque capacity will exceed the capacity of the standard key for a given shaft size. With a good interference hub fit, the key becomes the secondary member for transmitting torque. Therefore a rational approach to the details of the key would be in order. The ANSI chart for class 2 fits represents a good compromise between tight fit and constructability. Also the standard has been in use since 1967, reaffirmed in 1989, and proven throughout industry. Still the side fit of the key is by no means sloppy. Recall that the class 2 fit only allows a max of 0.002 inches clear on a 1 inch key. Given the alignment tolerances of the hub to the shaft, possible angular misalignments of the two keyways, and variations of the key, the as assembled key is going to be quite tight. The likelihood of the key coming out of an open ended keyway, properly peened on the end is negligible. Sure, if the fit were loose on the hub, the key were taking torque, and the key were loose, the closed end would keep the key in place. But a keeper plate, the other hub of the coupling, a set screw or peening would do the same thing and give 20% more capacity. The FN2 fit of the hub on the shaft is the most important component of combination. It is highly recommended that this AASHTO and AREMA spec be followed for all connections transmitting primary drive torques.

Due to the unique nature of heavy movable structures drive systems, every application is different. However, most all contain certain details such as turned bolts, keys, and hub fits. It is to everyone's benefit to have a clear understanding of the necessary details involved in making these seemingly simple connections. Sooner or later the details will have to be worked out. Sooner, preferably before bid time, is always better.