FASTENING IN WIND TURBINES: NO small PART of the BIG picture

by John Hollowell

Wind turbines are awe-inspiring machines. It seems as if nothing could ever bring them down. A look from afar shows them as sleek and powerful. Close examination of the structure and the workings inside the nacelle reveal the thousands of components that all must come together and stay together for the great natural power to be harnessed.

Power generation from wind depends on bolts, studs, screws and nuts, from the structural bolting patterns at ground level all the way into the generator high in the air. An improperly designed joint, a substandard screw, or even a loose nut can bring the entire operation to a stop. It can happen suddenly or take many months, going unnoticed until the final failure occurs.

As the wind industry tries to get its footing in the United States, it seems like this is a good time to examine the critical nature of a component that has been dismissed as a "commodity" by too many buyers and engineers in the user community.

For many U.S. manufacturers, putting their components into turbines is something completely new. They are learning about metric components for the first time, this country having resisted metrication for decades now. It is the Europeans, for the most part, who have built the wind industry, and they rely on ISO standards. For fasteners, "grades" are now "property classes," and dimensions are in millimeters, not inches. There are other things that differ in the ISO standards, and they should be reviewed carefully. It is time to learn or re-learn fastening, standards and beyond.

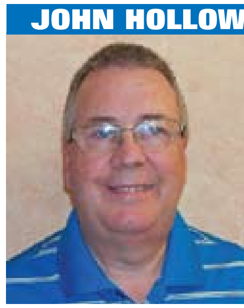
There are three basic elements to a sound joint: good joint design, properly assembly and maintenance of the joint, and consistent, high-quality fasteners. Proper joint design is science and art, and the engineer should keep in mind what has worked in the past for the application or similar ones. This is not simple with a wind turbine component because the history in these applications is short. I would believe that particular attention in wind turbine joints should be given to preventing fatigue, knowing that the turbine will be in a constant yet varying state of movement. That very condition with improperly preloaded joints would surely doom the joints to fatigue failure.

It may be useful to review the basic steps of bolted joint design. First, the basic geometry of the joint should be considered and the material of the clamped parts. Give thought to the number of axes in the joint. Make the initial selection of joint materials. Second, calculate the magnitude, direction and introductory point of the external forces acting upon the joint, and whether they are static or dynamic. Determine the acting temperature range of the joint and coefficients of thermal expansion of the mating parts. Next, select the material properties, size, drive style, and thread type of the fastener. Fasteners act like springs in storing energy to resist external forces that work to separate clamped parts. Long, slender screws are the best choice. A 1:4 or even 1:6 diameter-to-length ratio is desirable. Estimate the clamp load required to keep the joint functional. Develop a force diagram. Now calculate the mating material's capacity to resist embedment while keeping in mind other setting force losses. Determine safety factor desirable for the joint. Now, the most important step: TEST the joint in application under realistic conditions.

The second element of a sound joint, proper assembly and maintenance, also now

continued on page 40

JOHN HOLLOWELL



John Hollowell is the marketing and technical services manager for Kerr Lakeside Inc., a Euclid, Ohio manufacturer of socket screw products, cold-headed products and screw machine products. He has more than 23 years of experience in the fastener industry, with many hours of technical training by industry experts. John is a member of the ASTM Fastener Committee 16.02 and is the chairman of the Fastener Sub-Committee for NAAMS (North American Automotive Metric Standards) Stamping and Mechanical Components Groups. He has spent the last 16 years focused on high strength applications in various industries. John is part of a technical team at Kerr Lakeside that presents technical fastener training programs geared toward the specific needs of end user or distributor groups. Contact him at 800.316.5377 or at jdh@kerrlakeside.com.

Fastening in Wind Turbines continued from page 36

Method of Tightening	Accuracy Factor	Scatter %
To Yield Point Computer Controlled and Motorized	1	+/– 5–12%
Turn of the Nut (angle of rotation)	1	+/– 5–12%
Elongation Measurement of Calibrated Screw	1.2	+/– 10
Manual Torque Wrench with Experimental Tests	1.4–1.6	+/– 17–23%
Manual Torque Wrench w/o Experimental Tests	1.6–1.8	+/– 23–28%
Screw Driver with Preset Torque (friction, etc. estimated)	1.7–2.5	+/– 26–43%
Impulse Controlled Impact Wrench	2.5–4.0	+/– 43–60%
Hand Tightening	2.5–4.0	+/– 43–60%

Figure 1. Variability of preload as a result of using certain means of installation.

has new challenges in wind power applications. The forces involved in these applications may differ from those in the original design. Fastener maintenance on a wind turbine will present unusual challenges—working height, space restrictions, and awkwardness of tool use in various weather conditions. This must be dealt with at the design level. Assembly and maintenance of the bolted joint must be realistic. Keep in mind that preload or tension is the goal in maintaining the fasteners. Direct tensioning of the fasteners is certainly the best method to use. There are numerous devices on the market that can achieve these ends, and some fastener components, such as DTIs, even provide visual indication that desired tension has been achieved.

If preload cannot be achieved for a bolted joint due to space, height, or other considerations, then the designers and installers must turn to the use of torque to achieve tension. This is a less accurate method of achieving the desired preload but can be satisfactory if the variability in the method is understood and deemed acceptable. Torque and tension have a linear relationship, and it is examined most easily by a brief treatment of the short form torque equation and a review of a study published by the fastener industry outlining various means of tightening screws.

The short form torque equation is: $T_{in} = K \times D \times P$, where T_{in} is the input torque, K is the nut factor (loosely equivalent to a coefficient of friction), D is the stress area of the screw for inch series parts, or nominal diameter for metric parts, and P is the desired preload. It has been estimated that there are more than 250 factors that influence the value of K. Most fastener manufacturers publishing recommended installation torque charts have tested hundreds of their screws to find an average value for "K" in steel or another specified material and expect a scatter of plus or minus 20 percent.

A few of those 250-plus factors making up "K" include: thread fit and condition of the mating threads, finish on the fasteners and clamped parts, means of assembly and frequency and means of maintenance, and whether a fastener has been previously installed and in service.

Poor thread fit and condition not only hinders installation, it subtracts from clamp load achievement. Properly rolled threads on a screw, rerolled to smooth out nicks picked up in processing, assure a consistency in stretching the part to desired preload. Plating or coating screw threads or nut threads changes the relationship between torque and tension. For instance, zinc plating increases friction between the threads, while nickel reduces drag. Organic finishes can be modified to different friction coefficients, but care must be taken in choosing organics. The thickness of the coating needed to achieve the corrosion resistance desired can interfere in mating parts if that thickness is not accounted for in the screw design. Torque, when used as a means to achieve tension, has variability stemming from the tools, the operator, and even the conditions at the time of installation.

The inaccuracies of certain tightening methods make preload determination even more critical.

Other factors influencing clamp load in assembly or maintenance are tightening sequence, initial relaxation of the joint and improper use of mating fastener components. Fasteners lose five to 10 percent of initial clamp load in the first 24 hours after tightening. Screws in a joint should be tightened using two or three snugging passes, and then a final tightening to desired torque. It should be done in a pattern designed to spread the load equally among the screws and so as to not create prying action felt by any screw. Correct choice of mating fasteners should not be taken lightly. Screws and nuts should be strength matched. Having a nut the same strength as the screw ensures that the nut threads fail first because they are actually a bit stronger than the screw threads. A failed nut is more easily detected than a broken screw. Hardened washers should be used when needed under hardened screws and nuts to increase the effective bearing area of the joint. Split lock washers used to prevent loosening under the heads of hardened screws actually hinder the performance of the joint. Conical washers and disc springs can increase the effective length of a screw, allowing it to develop more preload to resist external forces.

A good view of a turbine's rotor and nacelle.

Reuse of fasteners previously installed can be covered in three words—DON'T DO IT! The relationship between torque and tension assumed during the original installation changes with subsequent reuse. The second time a screw is tightened by torque to the recommended value, the clamp load achieved is reduced. Fastener manufacturers' recommended installation torque data presume certain conditions and mating materials, which are usually spelled out in the chart. The values are not meant for anything other than new screws in "as received" condition.

Most often taken for granted is the third element in a sound joint: consistent, high quality fasteners. Making the correct choice for any application involves more than simply selecting the tensile strength of the fas-

continued on page 42

Wind Turbines

continued from page xx

teners. There are preferred methods of manufacturing for high strength screws. Threads should be rolled and have radiused roots; the fillet between head and body should also be radiused. The wire chosen should be free from harmful inclusions and properly prepared for the heading process. In extreme fatigue applications, screws can have reduced diameter bodies, effectively lengthening them. Thread can be rolled after heat treatment for greater strength.

Fastener specifications called out should be reviewed before inclusion on a drawing. Too often I hear from engineers that standards are all the same. While much has been done in the past few years to rationalize international standards, there remain differences. Dimensionally, ASME, ISO and JIS standards are very similar and have many overlapping tolerances. Generally they may be considered "functionally" equivalent with regard to dimensions. The same cannot be said in all cases for high strength fasteners when it comes to mechanical properties and permissible discontinuities. For instance, ASTM A574 and A574M, governing inch and metric socket head cap screws, do not permit laps and seams below the pitch diameter of the screw thread. ISO 6157-3, governing high strength socket cap screws for special applications, permits certain laps and seams below the pitch line. In effect, the standard allows a manufacturer to produce a screw designed for a tough application to have flaws in the smallest cross sectional area of the screw. Those thread laps and seams are a contributing factor to fastener and joint failure. And wind turbines certainly are a "special application."

Socket head cap screws and other high strength threaded products can be made without flaws in this critical area as proven over the last 100 years by many respected screw manufacturers. Proper raw material selection and maintenance of threading dies eliminate most issues. The best screw manufacturers continue to test production pieces every 30 minutes to ensure the parts being run are defect-free. Parts are taken from the machine, sectioned, placed in a mount, and then examined microscopically for the absence of thread laps. The cost difference for this benefit, critical to every part of a wind turbine that helps it work, is a matter of pennies per fastener, not dollars. What is the cost of downtime for a turbine, small or large? Couldn't we estimate it as hundreds, or thousands, or even hundreds of thousands of dollars?

Metric SHCS – one of many critical components inside turbine gearboxes and generators.

Finally, regardless of where or how a fastener is used on a wind turbine, let me implore: Know your supplier. There are many notable manufacturing companies across the world that test the worthiness of components from outside suppliers regularly and purchase only from those who are approved. Fasteners, if not a part of that list of "from approved sources only" components, should be a part of it. Keep a record of the manufacturers' markings stamped into the fasteners of your approved sources.

Fastening in wind turbines is no less critical than any other part of building these incredible machines. It should be treated as such. ■