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White-paper

THE DEVELOPMENT AND VALIDATION OF A NOVEL THREAD FORMING FASTENER FOR HIGH STRENGTH STEEL APPLICATIONS

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ABSTRACT

Automotive OEMs continue to lightweight their fleets to increase fuel economy and electric drive range to meet government requirements and consumer expectations. This push towards lightweighting has led to increasing use of advanced high strength steels in both car body and safety critical systems, such as seats. These steels present different challenges for traditional joining technologies, especially thread forming fasteners. Thread forming fasteners are externally threaded fasteners like machine screws that are driven into a pilot hole where they form their own threads and then tighten down to secure the target joint. Traditional thread forming fastener designs struggle to form threads into steels with tensile strengths greater than 600MPa. This paper utilizes simulation coupled with experimental validation to evaluate one of the most popular traditional thread forming fasteners that is currently used in low strength steel to determine why it fails to form threads in high strength steels. A new thread forming fastener design targeted for high strength steels is developed through simulation. The new fastener design is manufactured and evaluated across three different screw diameters, and six different high strength steels (>600MPa tensile strength) using the same heat treatments, materials, and platings as traditional thread forming fasteners. Validation tests for the new fastener include drive-to-failure to identify a target tightening torque and clamp load at the target tightening torque. Both tests are conducted across a hole size tolerance window. It is shown that the new thread forming fastener design performs well in steels up to 1200MPa without any special heat treatments but can benefit from specialized heat treatments in steels of tensile strength 1200MPa or higher.

Keywords: Fastening, AHSS, Mechanical Joining

1. INTRODUCTION

Thread forming fasteners offer an economical and efficient solution to fastening two pieces of material compared to traditional bolt and nut joints or machine screw joints. Bolt and nut joints require access to both sides of the joint as well as two components. Traditional machine screw joints do not require a separate nut but do require a tapped hole. Tapping operations can be expensive, especially if debris removal or cleaning to remove oils is necessary. Tapping costs increase in higher strength steels due to shorter tap life. Joints utilizing tapped holes or nuts can be more susceptible to loosening due to inherent tolerances designed into the joint [1; 2]. Thread forming fasteners form their own threads resulting in a tighter fit between the fastener and formed female thread than conventional machine screws. Thread forming fasteners require a correctly sized pilot hole in the lower material to be joined and a clearance hole in the top layer(s). The fastener will form its own threads and tighten into the lower material (mating material), offering a more cost effective and efficient method of joining.

Thread forming fasteners are installed to a predetermined tightening torque that comes from drive-to-failure testing. During drive-to-failure tests, torque is recorded as the screw is driven into the desired stackup until failure occurs. Failure modes can vary depending on the application, common modes include mating/female component thread stripping or fastener torsional failure. An example of a drive-to-failure curve is shown in Figure 1. The left side of Figure 1 shows a torque-angle curve for a single fastener installation. At Point 1, the torque is zero. At Point 2, the fastener has formed a full-sized thread into the mating material. The highest torque at Point 2 is called the drive or thread forming torque, this is the torque required to form threads into the mating material. Traditional threadforming fasteners often fail prior to moving beyond point 2. At Point 3, the bearing surface (bottom of the fastener head) touches the top surface of the mating materials.

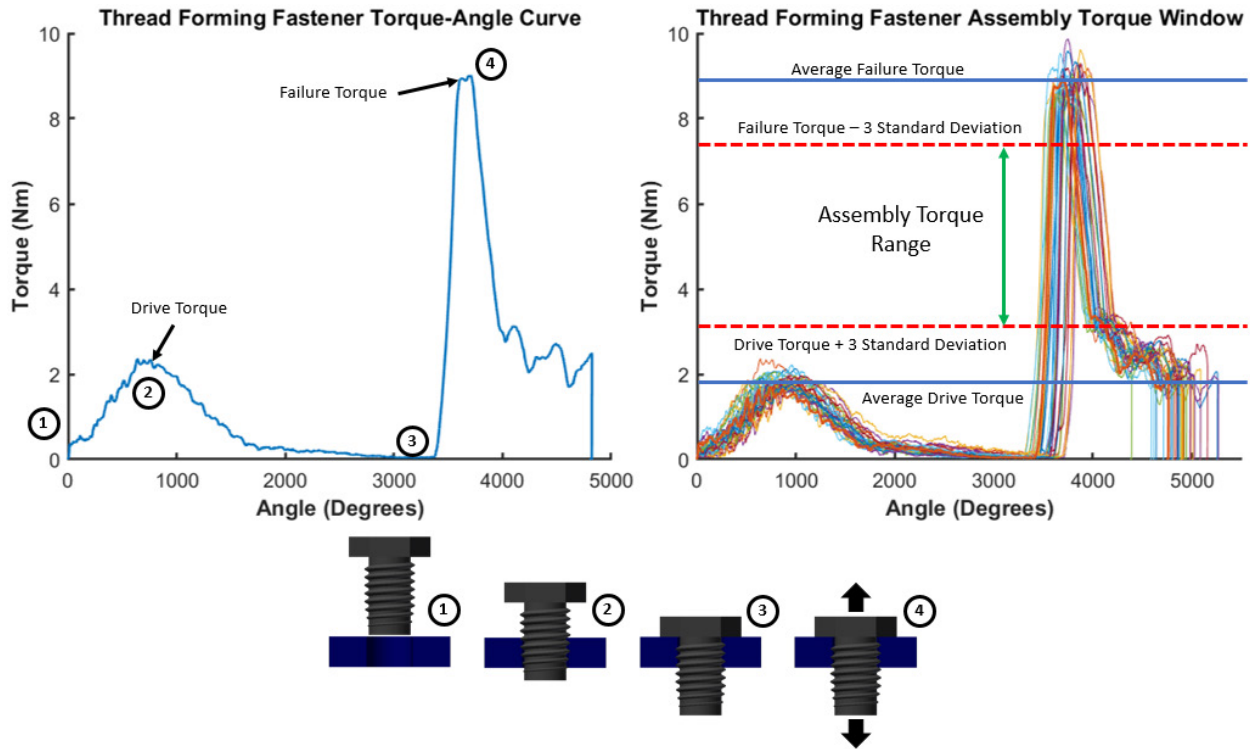


Figure 1: (left) Typical torque-angle curve for a thread forming fastener with fastener positions labeled, (right) assembly window based on multiple fastener installations

At this point the fastener starts to stretch, generating clamp load and holding the parts together. At Point 4, either the generated clamp load or resultant torque causes joint failure. To ensure a proper joint a tightening torque must be between the drive torque (Point 2) and failure torque (Point 4) for a simple torque-controlled driver. If the torque is too low, the fastener will not seat, too high and the joint could fail during installation. To account for variation, multiple repetitions are conducted as shown on the right side of Figure 1. The assembly window is the torque range between the upper limit of the drive torque and the lower limit of the failure torque. The limits are typically represented by “x” standard deviations from the mean, where “x” can vary based on the customer or application. In this research, 3 standard deviations are used to generate an assembly window. A value in the middle of the torque range is chosen, the larger the assembly window, the greater the chances of the fastener installing properly every time it is used. There are many causes of variation in fastening including tolerances on the fastener geometry, pilot hole geometry, mating material tolerances, fastener plating (friction coefficient), fastener material properties, work piece material and coating properties, and installation equipment. If the upper drive torque limit overlaps the lower failure torque limit, there is not a clean assembly window, and the joint should not be attempted in production. If these two limits are close to each other, the tolerances and variations of the parts can cause installation failure over the fastener’s usage in production.

Most thread forming fasteners have three basic requirements.

The fastener must have tapered lead threads, shown in Figure 2. These lead threads allow the fastener to gradually form threads into the mating material to reduce thread damage and decrease drive torque. The number of lead threads commonly varies from 2-5 depending on the application and fastener geometry.

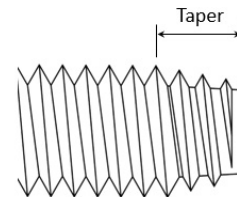


Figure 2: Tapered lead threads

A torque controlling feature, typically included on at least the tapered lead region of the fastener. The torque controlling feature functions to reduce drive torque in turn increasing the assembly window. A few different examples of common torque controlling features are shown in Figure 3. Each of these features serve the same purpose, they all focus thread forming loads to a smaller area of the thread and follow that area with a relief to reduce thread contact and drive torque. The thread forming point accomplishes this with lead threads that have a section of thread missing acting as a relief feature. The non-roundular body fasteners create forming stresses at the peaks of the thread while the valleys act as a relief. The part © 2022 by ASME

shown below is tri-roundular, its cross section is a triangle it has 3 high points with 3 relief areas; however other polygon shapes can be used [3-5]. Spiral lobe parts work like non-roundular except that the body of the fastener is round and enlarged lobes spiral down the threads acting as the same as the peaks on a non-roundular fastener. Multiple designs can also be combined when beneficial.

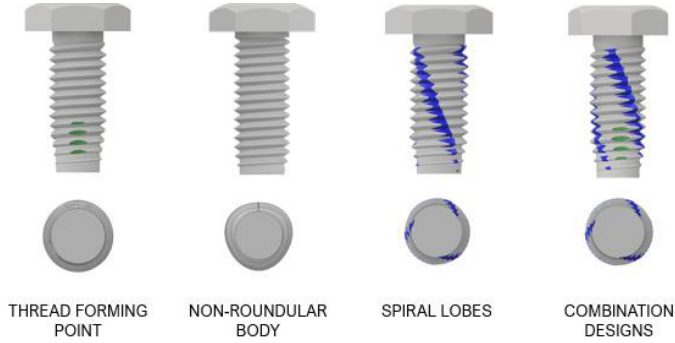


FIGURE 3: COMMON TORQUE CONTROLLING FEATURES USED ON THREAD FORMING FASTENERS

The fastener must be harder than the mating material. If the fastener is softer then the fastener threads will be destroyed before any threads are formed into the mating material. However, as the hardness of the mating material approaches the hardness of the fastener some thread damage and drive failure can be observed. This is a major problem with traditional thread forming fasteners, overcoming this problem is the focus of this research.

2. TRADITIONAL THREAD FORMING FASTENERS IN HIGH STRENGTH STEELS

To develop a better threadforming fastener for high strength steels it was important to first observe the failure mechanisms and then duplicate them in finite element simulation to allow for rapid development and digital testing of new concepts. To better understand the failure mechanisms and modes for traditional thread forming fasteners in high strength steels, a tri-roundular traditional thread forming fastener design with 2-3 lead threads was tested. The Traditional Design (TD) fastener used had approximately a 60-degree flank angle geometry, like a machine screw. Metric 5mm (M5) TD screws were manufactured from 10B21 cold-heading quality steel wire. The parts were carburized to a case hardness of 45 HRC minimum with a case depth of 0.1mm-0.22mm, the core hardness was 28-36 HRC. The fasteners were plated with a zinc-nickel electroplating that had a nominal friction coefficient of 0.13. The screws were driven into 3mm thick pieces of uncoated HSLA550 at 225rpm until failure. 4.6mm holes were punched in the steel using worn punches, the screws were driven into the punch exit side of the sheet. The punch exit side is notoriously difficult to drive screws through in the automotive industry due to the material break away and hardening from the punching process. The end load applied to the fastener was adjusted to ensure that the fastener drove until the head seated or until the fastener failed to progress any further into the sheet due to thread damage. Some of the screws managed to drive into the sheet and

strip out the HSLA mating material. However, some screws stalled during thread forming and refused to drive further into the sheet. Two of these screws are shown in Figure 4. The image on the left shows the lead threads of the screw are collapsed, removing the screw's ability to form threads into the HSLA resulting in the part stalling during installation. If the screw is forced through the sheet by adding significant end load, the entire length of thread can collapse, image on the right in Figure 4. The thread has more significant collapsing at the high points of the tri-roundular shape.



FIGURE 4: (LEFT) DAMAGED LEAD THREADS ON TRIROUNDULAR PART AFTER IT WAS DRIVEN IN HSLA550, THIS PART STALLED DUE TO THE THREAD DAMAGE. (RIGHT) FULL DAMAGED THREADS FROM FORCING THE TRIROUNDULAR SCREW TO DRIVE EVEN WITH LEAD THREAD DAMAGE (LARGE DOWNFORCE).

2.1 Simulation

The thread forming process has been modeled both analytically and empirically by multiple authors [6-9]. It has been more recently modeled using finite element analysis (FEA) as an adaptive meshing ALE simulation [10]. However, this paper takes a CEL approach like [11].

The thread forming process was simulated using an explicit dynamic solver in ABAQUS CAE. A simplified version of the previously mentioned experiment was modeled. An M5 round body diameter fastener was modeled as a deformable body with equivalent fastener class 10.9 properties. The model simplifications included using a round body design with exactly 60-degree thread flanks rather than a tri-roundular model. In addition, Class 10.9 properties were used instead of modeling a carburized part which has a hardness gradient. A round body fastener was used in place of the tri-roundular part used in experiment to simplify the model.

The steel sheet with a 4.6mm pilot hole was modeled as a eulerian body. The pilot hole was modeled as a void. The eulerian body was assigned a simple elastic-plastic material model for HSLA550. The model layout and dimensions are shown in Figure 5. The properties used are shown in Table 1. Simple material properties were used because the model was not

used to predict any torque or force values, only approximate thread deformation was of interest. Friction was modeled through general penalty tangential contact with a friction coefficient of 0.13, to match the plating used in experiment. The fastener's rotational velocity was set to 225rpm. The outer edges of the eulerian instance were fixed. The steel sheet (eulerian) was meshed with EC3D8R elements, the fastener was meshed with C3D8R elements.

TABLE 1: MATERIAL PROPERTIES USED IN SIMULATION

Material	Density (kg/m ³)	Young's Modulus (GPa)	Poisson's Ratio	Yield Stress (MPa)	Stress at fracture (MPa)
HSLA 550	7800	207	0.3	550	650
Fastener Class 10.9 Steel	7800	207	0.3	900	1000

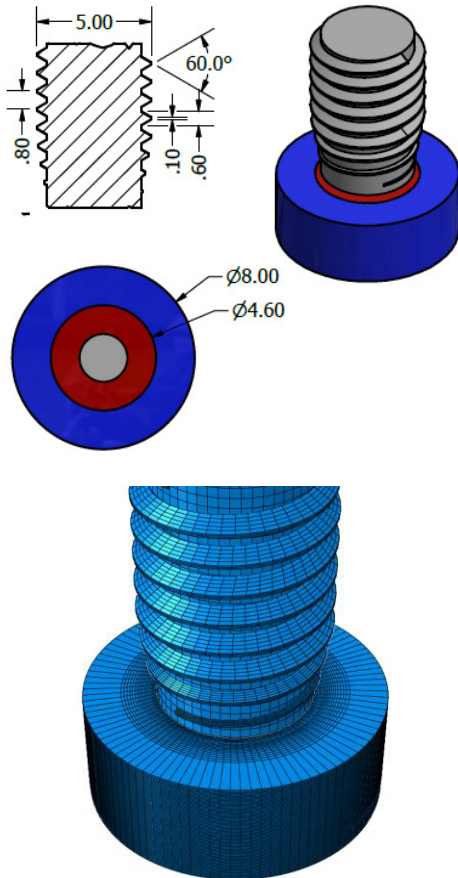


FIGURE 5: MODEL SETUP, THE BLUE COLORED MATERIAL IS HSLA550, RED IS VOID, GREY IS THE FASTENER WITH CLASS 10.9 PROPERTIES

The threadforming progression from the simulation is shown in Figure 6, this is a cross section view showing the steel as a blue material with the void as red. At first contact, image A

in Figure 6, the thread starts to compress as it encounters the steel. As threadforming progresses and the thread height increases along the tapered region the deformation transfers from compression to a combined compression and bending loading. At the end of thread forming (Figure 6 Image D), there is not a well-formed thread in the blue steel material and the fastener's threads are heavily damaged. Figure 7 shows the fastener before installation, after installation, and the tri-roundular part that deformed during experiment. The deformation looks similar between experiment and simulation. The experimental part is more deformed on the high points of its tri-roundular shape, whereas the simulation has even deformation because the part is round.

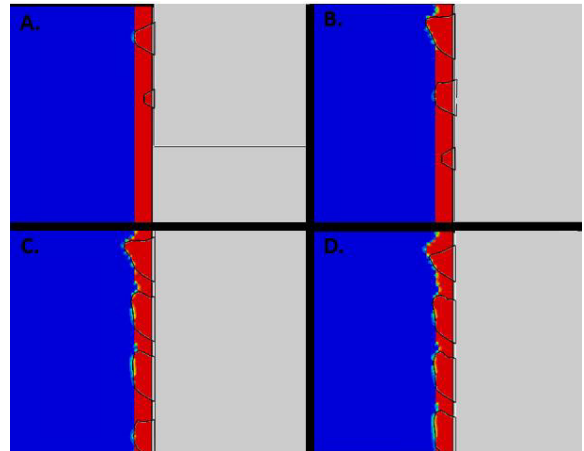


FIGURE 6: SIMULATED THREADFORMING PROCESS FOR A TRADITIONAL THREAD FORMING FASTENER IN 3MM THICK HSLA550

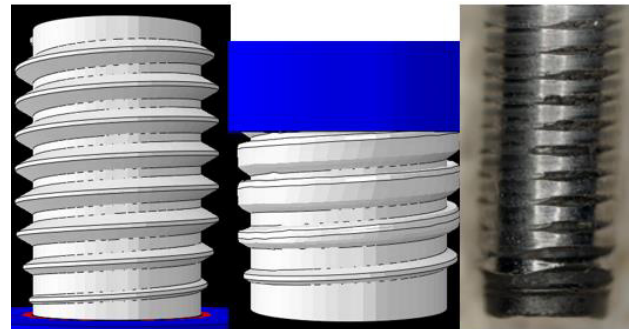


FIGURE 7: TRADITIONAL THREAD FORMING FASTENER SIMULATION BEFORE INSTALLATION, AFTER INSTALLATION, A DEFORMED TRI-ROUNDULAR PART FROM EXPERIMENT

3. DEVELOPMENT OF A NEW FASTENER DESIGN

From the experimental results and simulation shown in Section 1 of this paper, the primary deformation mechanism for traditional thread forming fasteners with 60-degree flank angles is thread bending followed by thread shearing or thread collapse. The following design targets were created for a new fastener design:

The fastener thread must not experience plastic deformation from bending loads in high strength steels
 The fastener must have a standard metric thread pitch and be serviceable with metric machine screws- this is a common requirement for serviceability in the automotive industry
 The fastener must not require special steel alloys, heat treatments or platings to function. This means it must use typical fastener alloys such as 10B21 or 4037 steel with heat treatments such as neutral hardening, case hardening/carburizing, or neutral hardening with induction hardened lead threads. 5
 The fastener must be manufacturable using traditional cold heading and rolling equipment.

It was expected that regardless of the design, without changing the fastener material or heat treatment that the resultant design would still deform during threadforming in high strength steels. As such a thread design was created that changed the loading of the fastener during the threadforming step. The resultant design is comprised of two sets of flank angles. The top flank angle is between 90 and 130 degrees while the lower flank angle is between 30 and 70 degrees. The large upper flank angle changes the loading of the thread away from shear and bending and into compression when it contacts the high strength steel during thread forming. This means that the fastener thread would crush inwards towards the axis of the fastener rather than bend. The lower flank angles have a smaller angle to prevent the thread from getting too large or requiring a special pitch spacing. The fastener uses 4-5 lead threads to reduce the depth of threads formed and resultant loading per rotation. This new fastener was named Rolok HS® and will be referred to as RHS for the remainder of this paper. RHS was simulated at the same conditions as TD, the results are shown in Figure 8 and Figure 9. The fastener uses the spiral lobes torque reducing feature (not modeled for simulation).

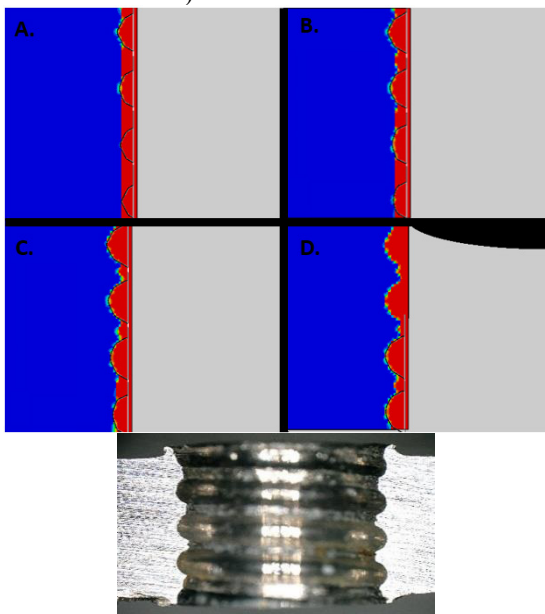


FIGURE 8: RHS SIMULATED THREADFORMING PROGRESSION IN 3MM THICK HSLA550 WITH A CROSS SECTION OF AN RHS FORMED THREAD



FIGURE 9: RHS SIMULATED THREAD SHAPE BEFORE AND AFTER THREADFORMING IN 3MM THICK HSLA550

4. EXPERIMENTAL METHODOLOGY

The testing progression for this research started with creating pilot holes in the steel mating materials. The holes were either punched or flow drilled at the upper and lower limits of the hole tolerance range, shown in Table 2. The tolerances were set based on common hole tolerances for thread forming fasteners held in the automotive industry. Smaller hole size tolerances will increase the cross-tolerance assembly window by restricting hole size-based changes in drive or failure torque. The M5 diameter was the first RHS part tested. It was initially tested in the same hole size as TD. From this testing it was discovered that RHS preferred a larger hole size than TD, the hole size was adjusted to be larger than TD on the tests for M6 and M8 sizes.

Some steels were flow drilled to increase the length of engagement. The holes were flow drilled using carbide flowdrill tools spun at 2000rpm in a Bridgeport knee mill. An example of a flow drilled hole is shown in Figure 10. For each material/fastener combination, holes were formed at the upper and lower tolerance limits, for some combinations, a hole size in the center of the tolerance was included.



FIGURE 10: FLOW DRILLED HOLES IN 1.1MM THICK GEN 3 1200MPa STEEL WITH AN M5 RHS SCREW DRIVEN THROUGH THE FLOW DRILLED HOLE

Once the holes were formed, drive-to-failure testing was conducted using a washer to simulate a clamped component. A minimum of 20 repetitions were conducted for each combination. This testing yielded the across tolerance assembly window. The midpoint of this window was selected as a tightening torque. Drive testing was done for all combinations. Clamp load testing was completed for some of the steel materials. Clamp load testing involved driving the fastener

through a load washer into the steel mating material to the tightening torque determined by drive testing. The clamping force was recorded from each installation, at least 8 repetitions were completed per combination.

When fasteners of the same size, heat treatment and plating were available in tri-roundular traditional design (TD) comparison testing was done between RHS to TD. Hole sizing for TD was based on the technology creator’s guidelines, the hole sizes were slightly increased to account for the high strength steels. The list of steels tested is shown in Table 3, the tensile strengths range from 420MPa (340LA) to 1200MPa. Different platings were used on each different diameter fastener to determine if RHS has sensitivity to any of the platings. The M6 and M8 fasteners were cold headed from 4037 alloy steel wire and neutral hardened to 33-39 HRC, the lead threads were induction hardened to 45HRC. This is a common heat treatment for threadforming fasteners 6mm and larger in the automotive industry. The M5 fasteners used in steels above 800 MPa were induction hardened. For steels less than 800MPa tensile strength, the fasteners were cold headed from 10B21 steel wire and carburized to 45 HRC with a case depth of 0.1-0.22mm.

TABLE 2: INSTALLATION DRIVE SPEEDS FOR EACH FASTENER DIAMETER

Fastener Diameter-Thread Pitch	Drive Speed (RPM)	Plating Specification	Hole Size Tolerance RHS (mm)	Hole Size Tolerance TD (mm)
M5-0.8	225	GMW16730: Zinc-Nickel Electroplate	4.53-4.67	4.53-4.67
M6-1.0	150	GMW3359 Type C: Zinc Rich Dip Spin	5.55-5.69	5.44-5.58
M8-1.25	150	Ford S439: Zinc Rich Dip Spin	7.40-7.60	7.30-7.50

TABLE 3: STEELS INCLUDED IN THIS STUDY AND WHICH DIAMETER FASTENERS WERE TESTED

Steel Alloy	Thickness (mm)	Fastener Diameters Tested
MP800	3	M6, M8
980XG3	1.5	M5, M6, M8
340LA	3	M6, M8
DP800	1.5	M6, M8
DP980	2	M5
Gen 3 1200	1.1	M5
DP 1200	1.2	M5
HSLA550	3	M5, M6, M8

5. RESULTS AND DISCUSSION

5.1 Result Graph Explanation

Each fastener was tested across its hole size range to determine the assembly window for the given material, fastener, and hole tolerance combination. For each drive test, a torque-angle curve was saved and later analyzed. The drive torque and failure torque values were extracted for each test from the torque-angle curves, the average drive torque, average failure torque, and standard deviations were recorded. In the torque-angle curves shown in Figure 11, the dark blue lines show the lower failure torque limit for each hole size, the light blue line shows the upper drive torque limit for each hole size. These drive and failure torque limits are shown as bars in the lower graph in Figure 11.

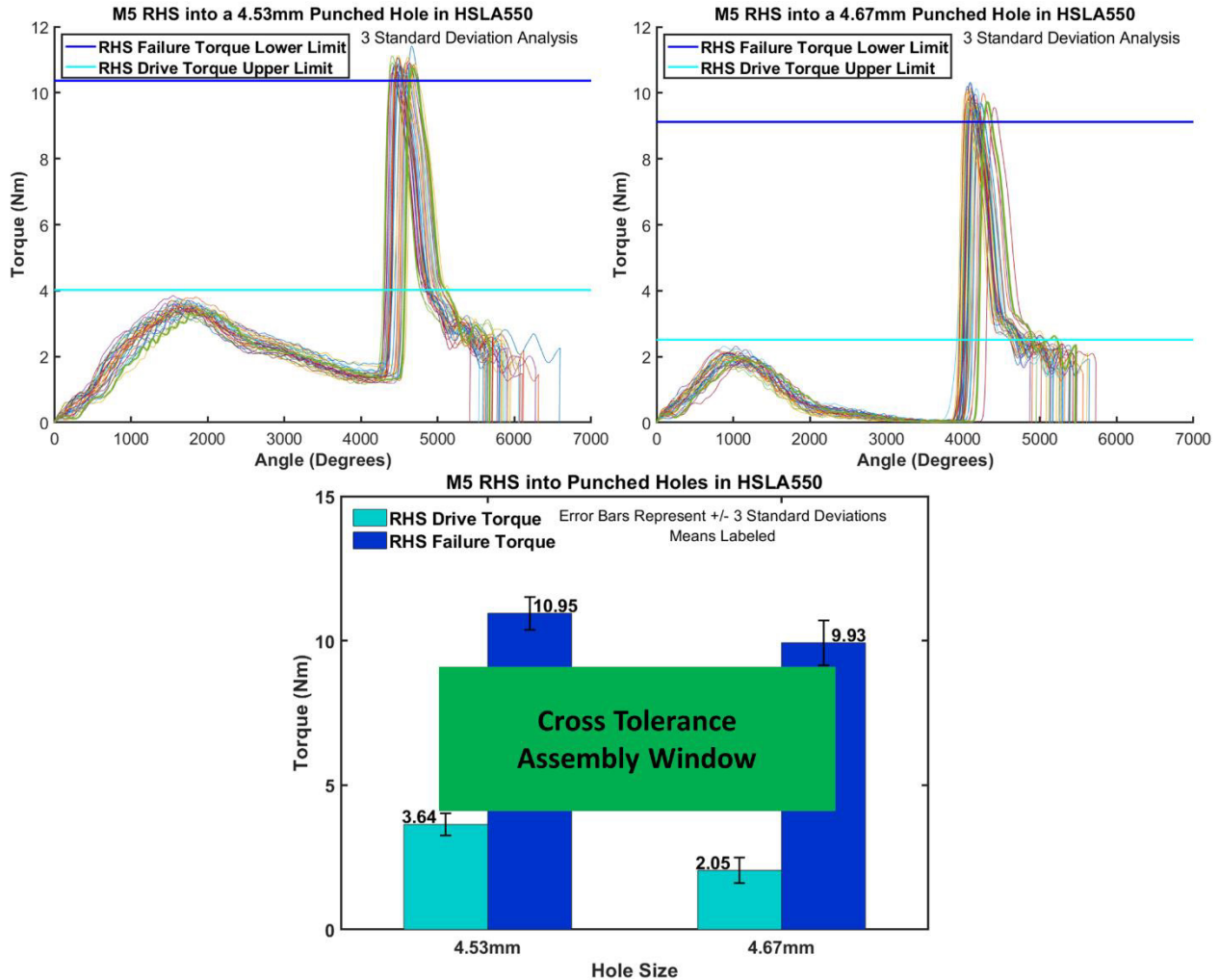


FIGURE 11: (TOP) TORQUE-ANGLE CURVES FROM DRIVE TESTING M5 RHS ACROSS ITS TOLERANCE RANGE IN 3MM THICK HSLA550, (BOTTOM) SIMPLIFIED GRAPHICAL REPRESENTATION OF A CROSS-TOLERANCE ASSEMBLY WINDOW

The cross-tolerance assembly window is the torque range between the highest drive torque limit and lowest failure torque limit, depicted in Figure 11. To reduce the number of graphs and simplify the data, the cross-tolerance assembly window for each parameter set will be shown in bar graph form.

5.2 High Strength Low Alloy 550 Steel (HSLA550/G80)

The cross-tolerance assembly windows for HSLA550 are shown in FIGURE 12, new punches were used to create these holes. Comparing the M5, M6, and M8 screws, increasing the fastener diameter increased both the stripping torque and drive torque. The larger diameter fasteners have greater thread height which displaces more material resulting in increasing thread friction and higher drive torque. The larger fasteners also have larger head diameters increasing under-head friction which coupled with increased thread friction results in higher failure torques, even though they may have less threads engaged. RHS for the M8, M6 and M5 fastener sizes has an assembly window size of 16.2, 7.7, and 5.2Nm, respectively. These are good size assembly windows considering they cover the entire

hole size tolerance. M5TD has a smaller window of 1.4Nm, this may still work but is could be problematic due to variation in parts between batches from heat treatment, geometry, and plating. A good assembly window for each fastener diameter and could be used in production without any issues. Since new punches were used in this testing, none of the TD fasteners failed to drive into the HSLA plates. However, the large variation in stripping torque suggests that the fasteners experienced thread damage during installation. To investigate the effect of punch wear on the ability of TD screws to drive into the HSLA550, holes were punched using worn punches for the M5 fasteners. To create the worn punches, the edges of a new punch were ground down, simulating a significantly worn punch, shown in FIGURE 13. The screws were driven through the punch exit side. The resultant assembly windows are shown in FIGURE 13. The data shown for TD does not include parts that failed to drive into the HSLA, these were removed because the parts never seated, as such the joint could not be taken to failure. TD screws were driven until there were 30 successful drives. On the 4.53mm hole size, 9/39 parts failed to seat for TD. On the 4.67mm hole size at

the upper end of the tolerance, 3/33 parts failed to drive. For RHS, all 30 parts succeeded in seating in both hole sizes and the threads did not experience any significant damage. This results in much lower variation in both drive torque and failure torque for RHS compared to TD. RHS does have a higher drive torque than TD due to its larger thread geometry which increases contact friction. Comparing to the clean punch data in FIGURE 12, the drive torque and failure torque both increased and the standard deviation increased in the damaged punch testing. This

is caused by the increase blow-out/fall away from the creating a hole with a damaged punched. There was less material in contact with the screw which reduced the torque values, at the same time the punching process was less repeatable causing increased torque variance. However, RHS still has an acceptable assembly window even with the damaged holes. The error bars for TD overlap, showing that there is no available assembly window.

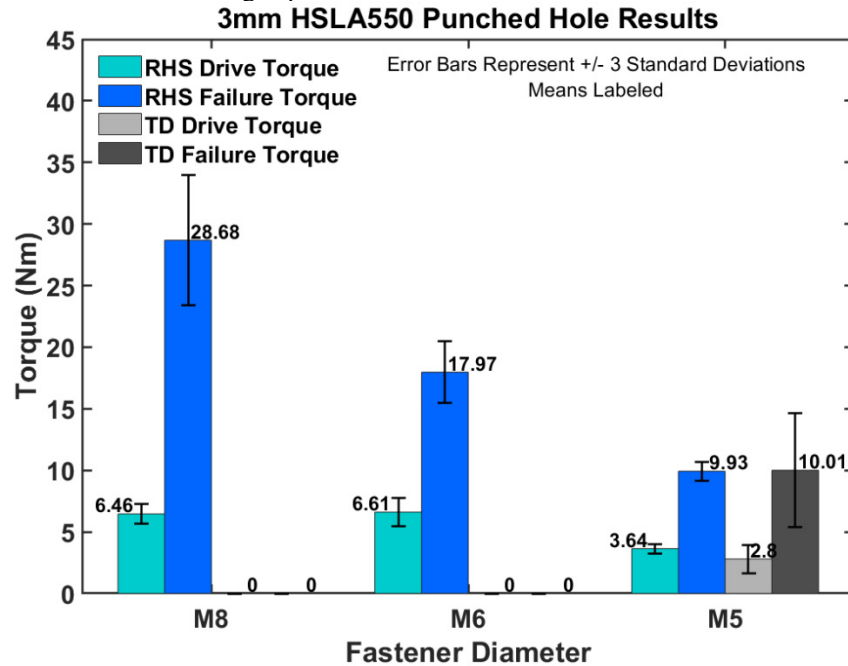


FIGURE 12: CROSS-TOLERANCE DRIVE TESTING RESULTS FOR 3MM HSLA550 WITH PUNCHED HOLES FROM NEW PUNCHES

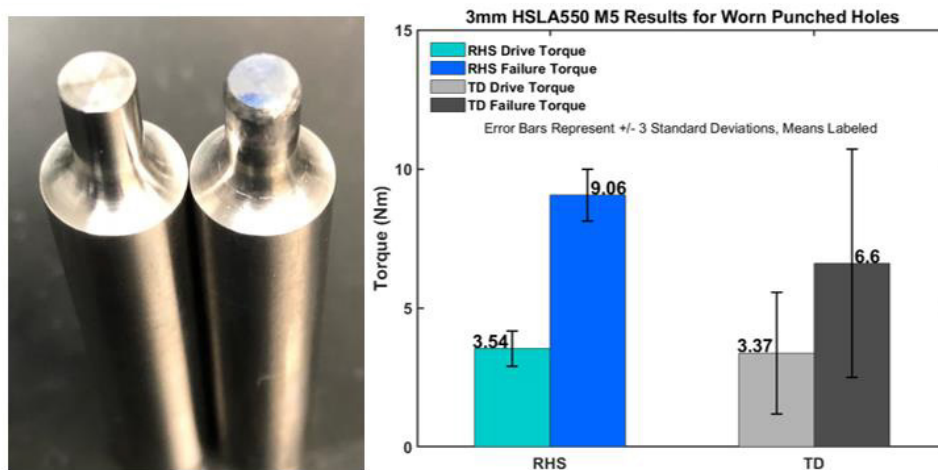


FIGURE 13: NEW M5 PUNCH NEXT TO AN ARTIFICIALLY WORN M5 PUNCH

5.3 1.5mm Dual Phase 800 (DP800)

The assembly window results for 1.5mm DP800 are shown in FIGURE 14, clamp load results are shown in FIGURE 15. For the M8 size, both the TD and RHS designs had the same drive torque, but the stripping torque and stripping torque variation

was much higher for the RHS design. The M8 RHS had an assembly window size of 6.84Nm, TD had an assembly window of 1.98Nm. RHS experienced large variation on its 7.4mm hole causing a lower failure torque value, however, even with this large variation, its assembly window is three times larger than

TD. Cross section of M8 TD and RHS driven into their smaller hole sizes are shown in FIGURE 16. We see that both screw's threads look normal well below the head. However, close to the head of the fastener both screws experience deformation that occurred when the joints were taken to failure. The threads on TD bent and collapsed, like what was seen during the M5 testing in HSLA550. The RHS thread is wider and will not easily bend, as such it sheared. This explains the difference between the strip torque of the two fasteners, if the body threads on both screws fail, it will take more force to shear RHS's thread than it will to bend TD's. However, it should be noted that M6 and M8 parts have pitches that are quite large for use in steels that are only 1.5mm thick, which is likely why there was not any thread

damage observed on the TD fastener. An M4 or M5 screw would have been a better fit for this thickness. Clamp load testing was executed for M8 RHS and TD as well as M6 RHS, the results are shown in FIGURE 15. The M8 and M6 RHS both generated around an average of 3.5kN clamp load, while the M8 TD had an average of around 2.8kN. The RHS parts had a larger assembly window and as such were tightened to a higher torque. However, due to the wide thread flanks on RHS, the increase in clamp load is not proportional to the increase in tightening torque. However, the larger assembly window for RHS would make it more practical in high volume manufacturing, especially given it will result in the same clamp load as TD, if not higher depending on the hole size.

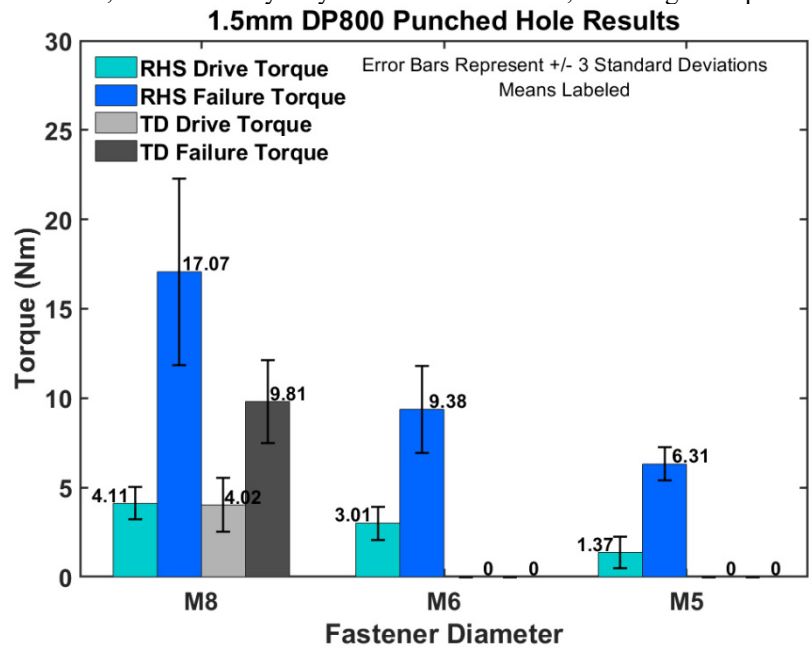


FIGURE 14: CROSS-TOLERANCE DRIVE TESTING RESULTS FOR 1.5MM DP800 WITH PUNCHED HOLES

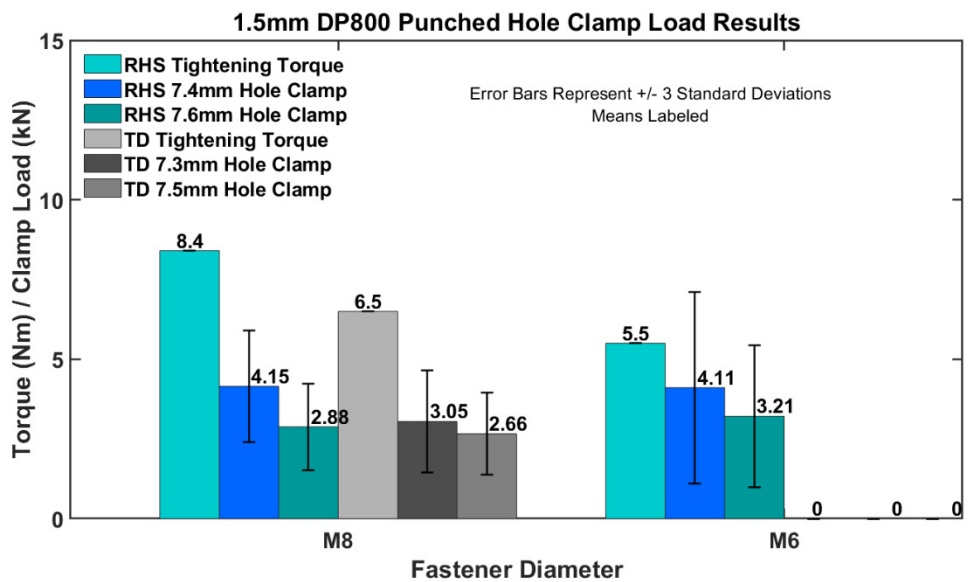


FIGURE 15: CLAMP LOAD COMPARISON BETWEEN RHS AND TD FOR M8 DIAMETER IN 1.5MM PUNCHED DP800

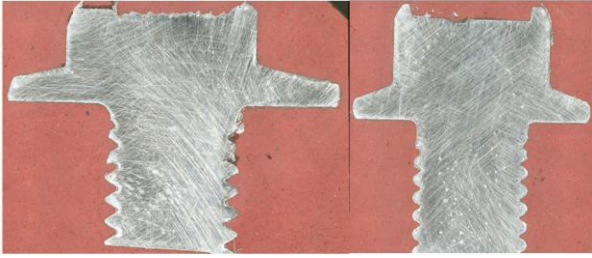


FIGURE 16: (LEFT) M8 TD AND (RIGHT) M8 RHS SCREW AFTER DRIVING TO FAILURE IN 1.5MM DP800

The MP800 is double the thickness of the DP800 but has the same tensile strength. Both the drive and failure torque for both fasteners approximately doubled, giving RHS a much larger advantage over TD. The assembly window for the M8 RHS is large at 18.95Nm, while the TD screw did not perform as well with an assembly window of only 3.55Nm. The clamp load values (FIGURE 18) also increased due to the increased available tightening torque from the larger torque windows. RHS was able to use a much higher tightening torque with its large assembly window, even with its flank angles suppressing some of the clamp load gains from increased torque, it produced almost 2.5x the clamp load of TD in the center of its assembly window

5.4 3mm Multi-Phase 800 (MP800)

The assembly windows for 3mm thick MP800 steel are shown in

FIGURE 17.

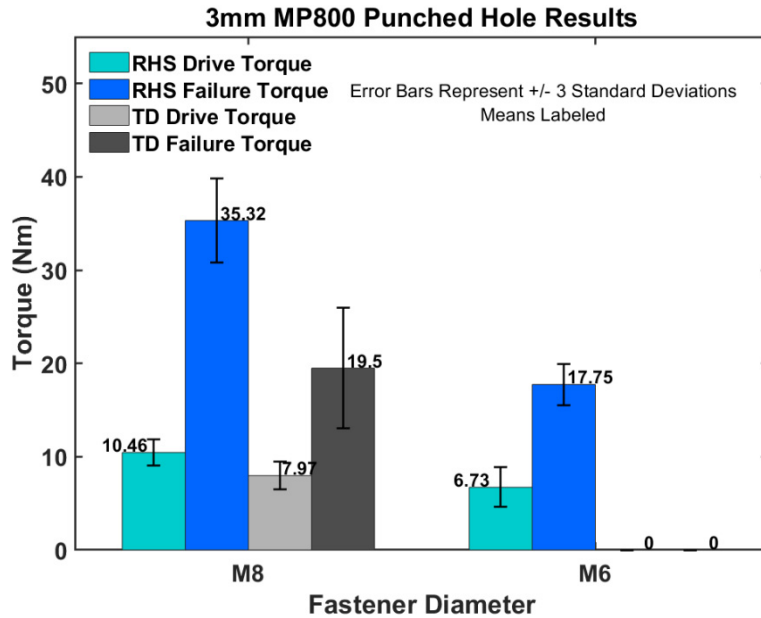


FIGURE 17: CROSS-TOLERANCE DRIVE TESTING RESULTS FOR 3MM MP800 WITH PUNCHED HOLES

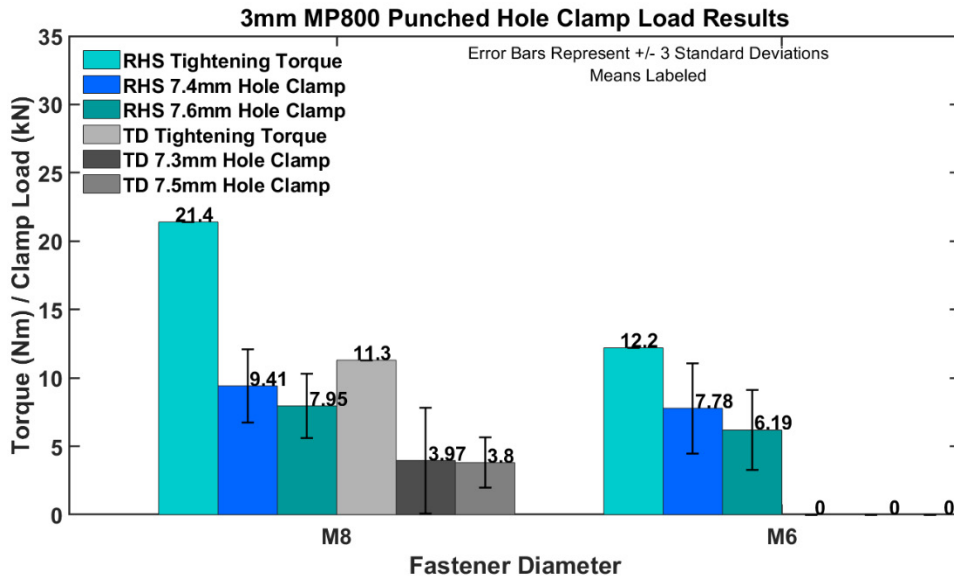


FIGURE 18: CLAMP LOAD DATA FOR 3MM MP800 WITH PUNCHED HOLES

5.5 1.5mm 980 Generation 3 AHSS (980XG3)

The punched hole results for 1.5mm 980XG3 are shown in FIGURE 19. In this testing, RHS has a useable assembly window for all 3 fastener sizes tested. TD was only tested as an M8 but its drive torque and failure torque error bars overlap, showing that there is not a useable drive window. While it was originally noted that TD could start to struggle in steels greater than 600MPa, as seen in the M5 testing in 3mm HSLA550. There are two reasons that significant TD failure was not seen until this material. First, the M6 and M8 parts were induction hardened, the entire tip of the screw and the lead threads are 45 RC minimum. In the 3mm HSLA testing, the parts were case hardened, only the outside shell of the lead threads was 45 RC minimum so they could not withstand as much contact with the

high strength steels as induction parts can. Second, when the thickness of the material is not much greater than the pitch of the part, the screw can somewhat wobble its way through the hole reducing the amount of thread forming that it does. There is still some degree of thread forming but it is less significant than it would be in thicker steels. Even the induction hardened TD screws cannot be used in 1.5mm 980XG3, there is no assembly window.

The clamp load results for 1.5mm 980XG3 are shown in FIGURE 20, TD was not tested in clamp load since there was not an acceptable assembly window available. The clamp load results are like the DP800 results since in both cases, the male thread of RHS fails which drives the clamp load limit.

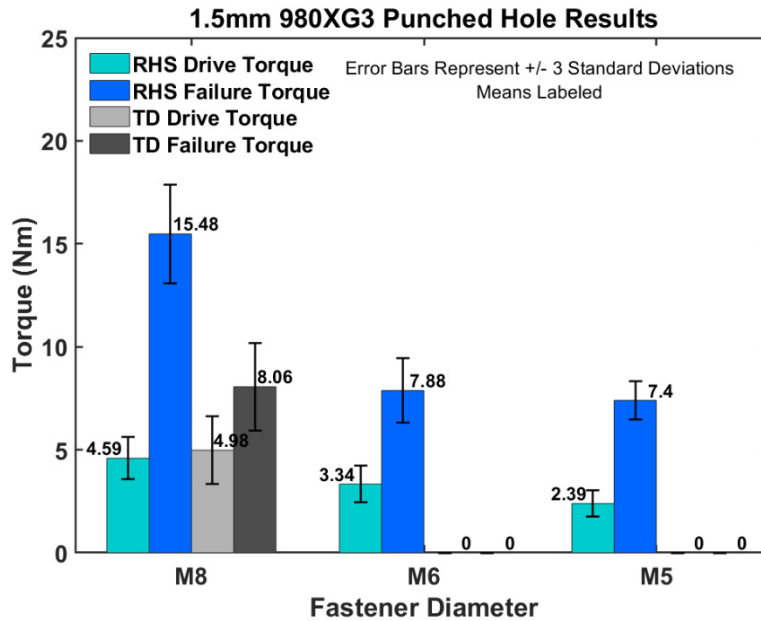


FIGURE 19: CROSS-TOLERANCE DRIVE TESTING RESULTS FOR 1.5MM 980XG3 WITH PUNCHED HOLES

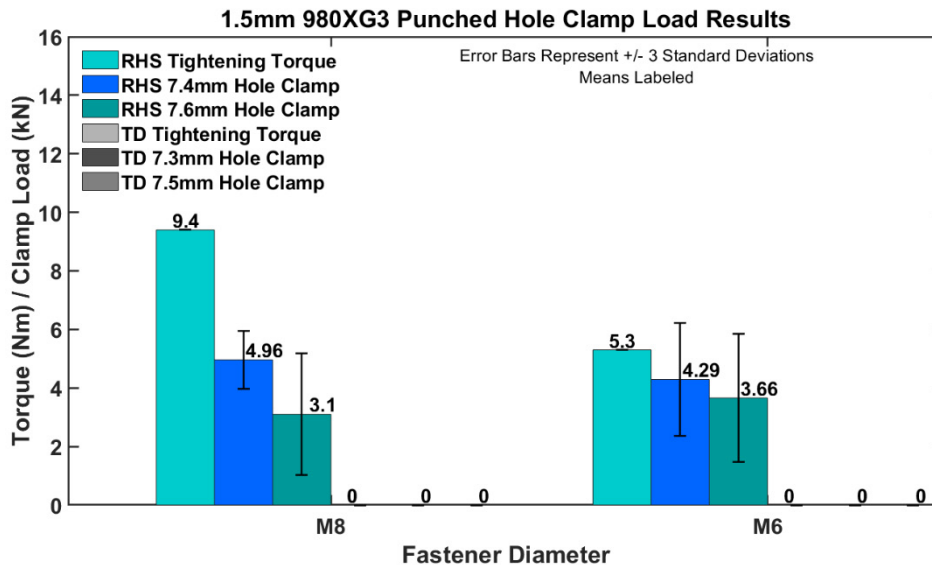


FIGURE 20: CLAMP LOAD RESULTS FOR 1.5MM 980XG3 WITH PUNCHED HOLES

5.6 3mm Hot Rolled Low Alloy 340 (HRLA340)

3mm thick HR340LA is the weakest steel tested in this research, its tensile strength is under the 600MPa limit where TD starts to struggle (when case hardened). The results for this material are shown in FIGURE 21 and FIGURE 22. In all the high strength steels shown in this report, RHS significantly beat out TD with much larger assembly windows and often time equal or higher clamp load. However, in HR340LA, TD does not suffer any thread damage nor does RHS. At the M8 size, the drive torques are similar, TD is slightly lower but also has a lower stripping torque. The assembly windows are near identical, as such, both fasteners were tightened to 12Nm in the clamp load testing, allowing for an even comparison. In the higher strength

steels, TD often could not reach the tightening torques that RHS used without risking joint failure. When both M8 designs were tightened to 12Nm, TD had near even performance on both of its hole sizes with clamp load right around 5.3kN. RHS had a larger difference between its two results, in its smaller hole size of 7.4mm, it had much higher clamp load with an average of 7.3kN. In its larger 7.6mm hole size, its clamp load dropped to 4.5kN. This is caused by the dual flank angle design. When the hole size is larger, more of the bolt stretching forces are concentrated on the wide upper flank angle, which is less efficient at transferring torque into clamp load. When the hole size is smaller, the smaller flank angle is loaded which is more efficient at transferring torque to clamp load.

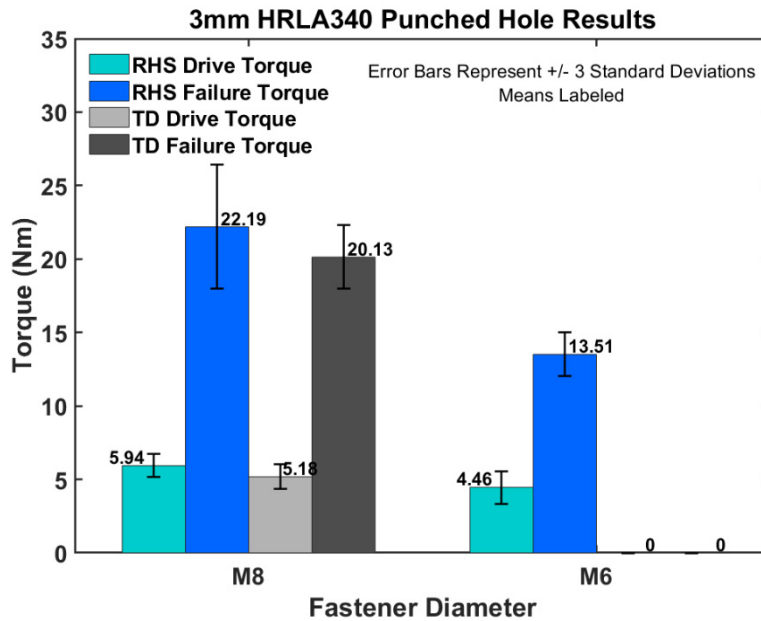


FIGURE 21: CROSS-TOLERANCE DRIVE TESTING RESULTS FOR 3MM HRLA340 WITH PUNCHED HOLES

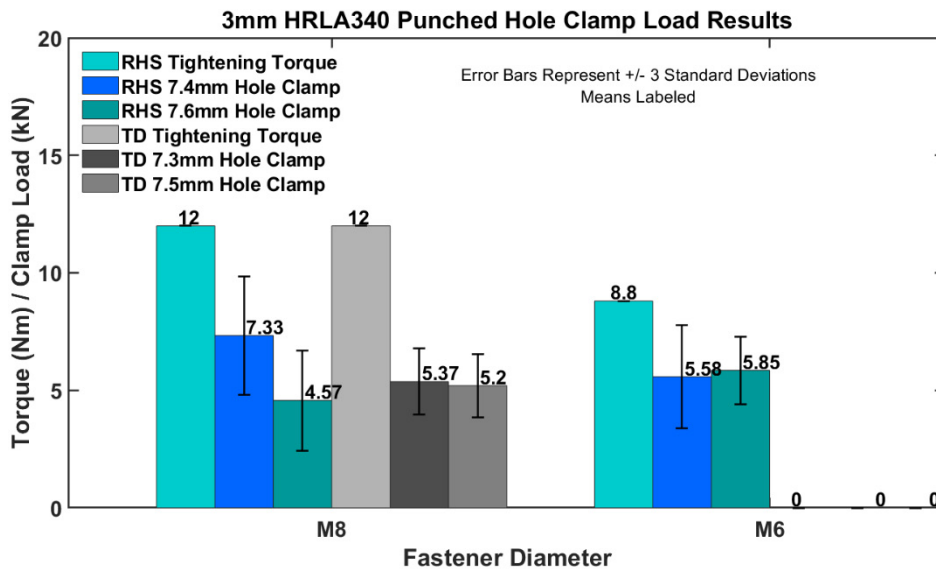


FIGURE 22: CLAMP LOAD RESULTS FOR 3MM HRLA340 WITH PUNCHED HOLES

5.7 1.2mm DP1200 and 1.1mm Flow Drilled Gen3-1200

In the testing shown so far in this report, the fasteners have all had standard heat treatments and materials. There was no significant difference in the performance of RHS between DP800 and 980XG3 driven by the deformation of the body threads on RHS during tightening. In this section we will compare an induction hardened M5 RHS that uses 4037 steel to an austempered M5 RHS that used the same steel. The austempering process produces a bainitic microstructure that is less susceptible to hydrogen embrittlement than the martensitic microstructure from an induction process. This allows the manufacturer to safely harden the entire part using austempering rather than just the lead threads. As such, the austempered parts have a hardness of 45RC minimum across the entire fastener, while the induction hardened parts have a tip hardness of 45RC but the body threads are softer at 33-39HRC. With harder body threads it will be harder to shear the RHS thread during tightening. Two steels were examined, 1.2mm DP1200 with punched holes and 1.1mm Gen 3 1200 with flow drilled holes. The flow drilled holes increase the number of threads engaged

but do require a longer secondary process to create the holes compared to punching. The 1200MPa steel is stronger than the body threads on the induction hardened part. This results in low stripping torque, shown in FIGURE 23. The flow drilled parts have greater thread engagement which results in higher drive and stripping torques compared to the punched DP1200. The austempered parts increased the stripping torque for both materials by increasing the fasteners ability to resist male thread deformation during the tightening stage. The standard deviation was high for the austempered parts, likely caused by uncertain failure modes. Since the 1200MPa steel has a hardness between 35-40RC, it is almost as hard as the fastener, even with austempered threads, it is possible that the failure mode alternated between female thread strip and male thread failure causing high standard deviation. In the future, a higher carbon content steel could be austempered to increase the hardness of the part well above the hardness of the 1200MPa steels. Increasing the stripping torque shifted the assembly window upward allowing for higher tightening torques to be used. This results in 2kN of increased clamp load, shown in FIGURE 24.

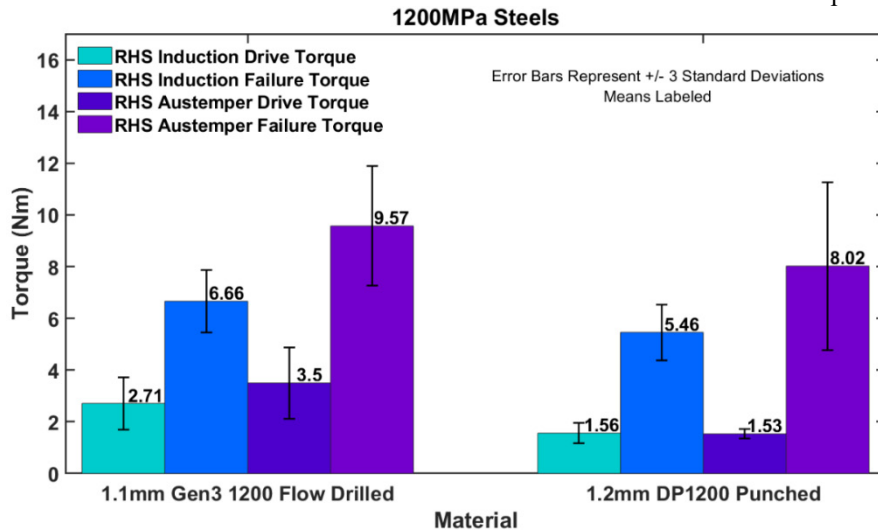


FIGURE 23: 1200 MPA STEEL TEST RESULTS COMPARING INDUCTION HARDENED RHS TO AUSTEMPERED RHS

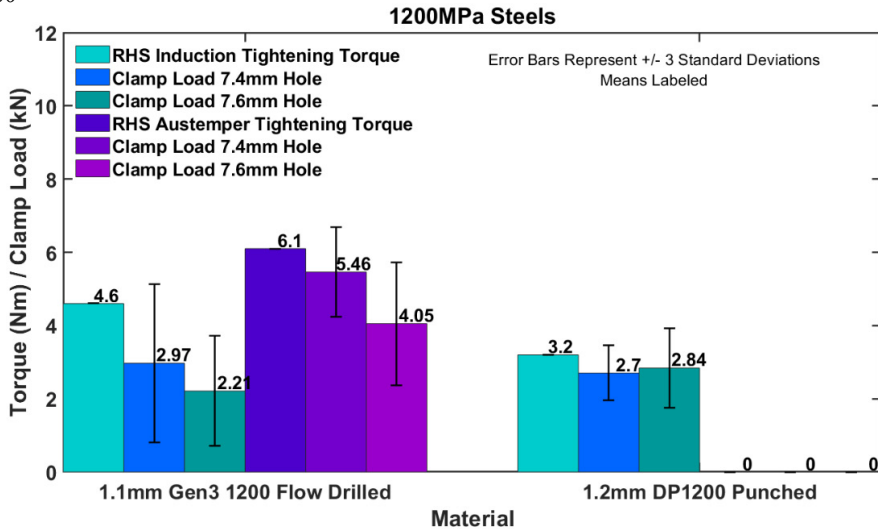


FIGURE 24: CLAMP LOAD RESULTS FOR 1200MPA STEELS COMPARING INDUCTION HARDENED AND AUSTEMPERED RHS

6. CONCLUSIONS

This research utilized simulation and experiment to determine that the failure mechanism of traditional thread rolling screws in high strength steels is thread bending that leads to thread collapse. The simulated model used for traditional thread rolling screws was used to develop a new thread geometry that would not bend or collapse when thread forming in high strength steels. The new fastener was named Rolok HS® (RHS). It was manufactured in 3 different diameters and tested in 6 different high strength steels ranging from 450-1200MPa tensile strength. RHS showed useable cross-tolerance assembly windows that were larger than a tri-roundular traditional thread rolling design (TD) in every high strength steel comparison conducted. RHS also maintained equal or better clamp load compared to TD. RHS has raised the thread forming application limit to 1200MPa. Future work will look at validating special heat treatments and steel alloys on RHS to extend its useable thread forming range to 1500MPa without risking hydrogen embrittlement.

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