THE HOBSON Update

Volume 25

From the desk of Peter Hobson

Warning: High Risk Factor

Jumping out of a plane without a parachute Vs. Buying your Zinc Plated Class 12.9 SHCS from a supplier who doesn't carry out Hydrogen Embrittlement Testing - About the same Risk!

Let me start this article by saying that all Hobson SOKO™ branded Class 12.9 ZP SHCS are all batch tested for Hydrogen Embrittlement. Why? …. *Because we care!*

I officially started in the fastener business in 1987, although, growing up in a fastener family, I have been involved my whole life.
I still remember an order I took in 1988, for M64 x 500mm Class 12.9 SHCS, and no they were *not* zinc plated. I imported them from a very reputable factory in Korea and they truly appeared to be of amazing quality when they arrived. They had a black shiny finish and were beautifully hot forged. To my surprise, I got a phone call from our client saying that they were "exploding" overnight! They were being installed in mining equipment out in a distant mining town in Western Australia where the temperatures got down to freezing overnight. They were all installed without a problem, however, in the morning, broken bolts littered the site. My stomach sank…I knew what the problem was…but they were not zinc plated??? Unfortunately the product was cleaned in a pickling process before blackening, and this is where the hydrogen was introduced into the steel.

In This Issue:

If you are having trouble with your ZP Class 12.9 SHCS failing, chances are they were failing due to Hydrogen Embrittlement. The potential liability and the severe headaches associated with these occurrences are tremendous.

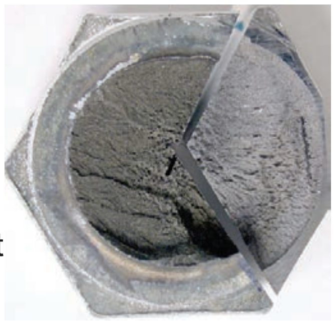
Perhaps the most high profile case of suspected Hydrogen Embrittlement failure is California's replacement San Francisco–Oakland Bay Bridge. The bridge uses shear keys, giant steel and concrete boxes sandwiched between the bridges' deck and supporting concrete pier, to counteract movement during an earthquake.

Hydrogen embrittlement in electroplated screws is still, to a great extent, a mystery. It seems to only occur sporadically, making it hard to detect and prevent. One fact seems to be agreed upon by most experts; the potential for hydrogen embrittlement in electroplated screws and bolts increases in direct proportion to the amount of carbon in the steel and the hardness of the parts.

- It is generally agreed that electroplated Class 4.6 and Grade 2 screws and bolts have virtually no tendency toward hydrogen embrittlement because they are made of low carbon steel and are not hardened at all.

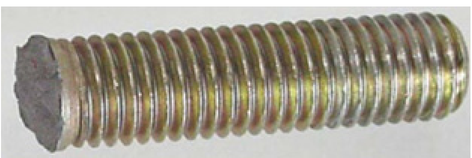
- Electroplated Class 8.8 and Grade 5 screws and bolts are very unlikely to have hydrogen embrittlement problems. They are made of medium carbon steel and are hardened to a maximum hardness of Rockwell C 34. Personally, I have never heard of a confirmed case of hydrogen embrittlement in these fasteners.

- Zinc electroplated Class 10.9 and Grade 8 screws and bolts are prone to hydrogen embrittlement. They are made from medium carbon alloy steel and are hardened up to Rockwell C 39. I cannot say it is common, but I have seen it.



- Class 12.9 socket head cap screws are hardened up to Rockwell C 45. Out of all of the types of standard screws and bolts, socket head cap screws have the highest possibility of hydrogen embrittlement failures when they are electroplated.
There continues to be a lot of confusion over what is and what is not a hydrogen embrittlement failure. It is simply this:

If a screw breaks between 1 and 48 hours after installation and the break is at the head to shank juncture or the thread run out it is probably a Hydrogen Embrittlement Failure (HEF). If, however, the parts break while they are being installed the problem is almost certainly something other than Hydrogen Embrittlement.



"The surface of the failure looks reletively smooth...However, under magnification, you see that the surface has a crystalline appearance with many sharp facets."

A hydrogen embrittlement failure is the worst kind of problem a fastener supplier can have because they are delayed failures and a lot of time has been spent in installing the fasteners. The failures are dramatic.

My first experience with HEF was a crane rail bolt, basically a M20 Class 10.9 ZP bent bolt with a very sharp tight bend. It was this tight bend which had been cold worked that was the main contributing factor. In this case, the items were never installed, but one was dropped and it basically shattered. This was a very dramatic illustration, one that is not often seen, but due to the very tight un-radiused bend, the issue was greatly exaggerated.

Most customers ask for zinc plating to improve the corrosion resistance of the socket head cap screws in their application. The standard finish on socket head cap screws is black oxide which provides virtually no salt spray corrosion resistance. The objective of better corrosion resistance can be met by having the socket head cap screws zinc phosphated and oiled instead of zinc electroplated. In most cases the thin zinc plating added to socket head cap screws will only provide about 12 to 24 hours of salt spray corrosion resistance. A good phosphate and oil coating will provide a minimum of 72 hours of salt spray resistance. Phosphate and oil has been a favourite finish in the automotive industry for many years because of its low cost, relative to its good corrosion resistant.
These performance differences are documented in the General Motors finish specifications. GM 4342 M, Code 20K24 requires a minimum salt spray resistance of 24 hours for .0002" zinc and clear chromate, where as, GM 4435 M requires 72 hours of salt spray resistance for zinc phosphate and oil.

Phosphate and oil is what is known as a conversion finish and not a plating. The surface of the steel is made porous and penetrated by the phosphate which holds the oil very tenaciously. Because it is not a plating, it cannot seal hydrogen in the steel. Therefore, phosphate and oiled socket head cap screws do not suffer from hydrogen embrittlement failures, and unlike electroplated zinc plating, phosphate and oil does not build up in the threads of screws and bolts.

Therefore, zinc phosphate and oil does not cause thread fit and size problems.
Electroplated zinc is less slippery than black oxide and phosphate and oil. This means that it requires more torque to properly tighten a zinc plated socket head cap screw than it does the same screw with a black oxide or phosphate and oil finish.

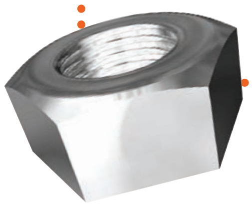
In more technical terms, it can be stated that the co-efficient of friction of black oxide and phosphate and oil is significantly lower than that of electroplated zinc. Where consistent bolt tightening (bolt tension) is critical, phosphate and oil is superior to electro-plated zinc.

I have learnt it is almost impossible to convert a market, so Hobson do offer ZP Class12.9 SOKO™ SHCS, however we batch Hydrogen Embrittlement test them to give our customers an extra level of confidence.

Source: IFI Technical note

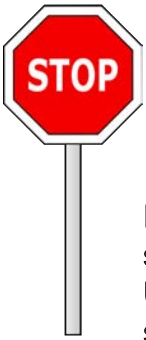
SOKO™

Hobson now stocks Class 12 Hex Nuts to DIN934 to suit our SOKO™ range. Sizes available are M10 to M64.
NATA reports are available on all items!



WALL OF FAME...
20 YEARS @ HOBSON!

(Left to Right) Bill, Mr. Lan, Naeem, Mark, Peter, Steven, Frankie. - December 2013



HOW TO STOP THREAD GALLING ON STAINLESS FASTENERS:

I have done a previous article on this subject for an earlier volume of the Hobson Update, however it is still a topic that our sales staff are asked about, so I thought it may be useful to cover it again...

The customer's complaint is that during installation the bolts are twisting off and/or the bolt's threads are seizing to the nut's thread. The frustration of the supplier is that all required inspections of the fasteners indicate that they are acceptable, but the fact remains that they are not working.
According to the Industrial Fastener Institute's 6th Edition Standards Book (page B-28), this problem is called "thread galling".

"Thread galling seems to be the most prevalent with fasteners made of stainless steel, aluminium, titanium, and other alloys which self-generate an oxide surface film for corrosion protection. During fastener tightening, as pressure builds between the contacting and sliding thread surfaces, protective oxides are broken, possibly wiped off and interface metal high points shear or lock together. This cumulative clogging-shearing-locking action causes increasing adhesion. In the extreme, galling leads to seizing - the actual freezing together of the threads. If tightening is continued, the fastener can be twisted off or its threads ripped out."

Carpenter Technologies, the fastener industry's largest supplier of stainless steel raw material, refers to this type of galling in their technical guide as "cold welding." Anyone who has seen a bolt and nut with this problem understands the graphic nature of this description.

Effective solutions are as follows :

→ Slowing down the installation RPM speed will frequently reduce, or sometimes completely solve, the problem. As the installation RPM increases, the heat generated during tightening increases. As the heat increases, so does the tendency for the occurrence of thread galling.

→ Lubricating the internal and/or external threads frequently eliminates thread galling. Effective lubricants mostly contain large percentages of molybdenum disulphide (moly) or graphite. You must be aware of the end use of the fasteners before settling on a lubricant. Stainless steel is frequently used in food related applications which may make some lubricants unacceptable.

→ What some think is an urban myth, using different stainless alloy grades for the bolt and the nut is actually effective in reducing galling. The reason is not they are different alloys, but rather the components having different hardnesses. If one of the components is 316 and the other is 304 they are less likely to gall than if they are both of the same alloy grade. This is because the different alloys harden at different rates.

→ Another factor affecting thread galling in stainless steel fastener applications is thread roughness. The rougher the thread flanks, the greater the likelihood galling will occur. In an application where the bolt is galling with the internal thread, the bolt is usually presumed to be at fault, because it is the breaking component. Generally, it is the internal thread that is causing the problem instead of the bolt. This is because most bolt threads are smoother than most nut threads. Bolt threads are generally rolled, therefore, their thread flanks are relatively smooth. Internal threads are always cut, producing rougher thread flanks than those of the bolts they are mating with. The reason galling problems are inconsistent is probably due largely to the inconsistencies in the tapping operation. Rougher than normal internal threads may be the result of the use of dull taps or the tapping may have been done at an inappropriately high RPM.

→ It is interesting that our BUMAX™ 88 and 109 high tensile product is renowned for displaying less tendencies to gall.

This is because, firstly it is harder and secondly, the threads are very well formed with all product being manufactured in Europe.

Source: IFI Technical Bulletin

THE HOBSON QUIZ

A farmer challenges an engineer, a physicist, and a mathematician to fence off the largest amount of area using the least amount of fence...

The engineer made his fence in a circle and said it was the most efficient.

The physicist made a long line and said that the length was infinite. Then he said that fencing half of the Earth was the best.

The mathematician laughed at the others and with his design, beat the others.

What did he do?

The FIRST person to correctly solve the riddle will win a $100 gift card!

email your answer to:
emma.hobson@hobson.com.au

Let's Torque

Over the years I written a number of articles in the Hobson Update on Torque and its relationship to tension. Hopefully all our readers now know that it is the correct tension in the assembly that is the important issue, the correct torque does not always relate to the correct tension. The article usually ends up singing the praises of the Squirter Washer™, an individual bolt assembly load cell, however in this article I am going to concentrate on recommended tightening torques. I think it is important to understand a little of the theory to get a better comprehension of the topic. Obviously the readership of the Hobson Update is not just engineers, so I will attempt to keep the article informative to the general reader, while perhaps even provoking some thought in engineers. Before I start, I need to make it very clear that values derived from the below formula are approximate tightening values based upon variables that may or may not be present in individual applications.
The formula we use is:

T[i] = WKD

T = Torque (Nm {Newton Meters} or Ibf.ft {Pounds force Foot})
W = Force in the bolt (kN or Ibf)
K = Torque Factor (A non-dimensional indication of lubricity, refer table)
D = Nominal diameter of the bolt (mm or inch)

[i] Analysis shows that approximately 50% of tightening torque applied is required to overcome the friction between the nut turning against the joint surface another 40% to overcome the friction between the nut and bolt mating threads with only the remaining 10% used to develop tension in the bolt. It should be remembered that recommended tightening torque figures published are based upon first tightening of single assemblies in isolation and multi fastener joints may require individual analysis to determine a tightening sequence and values. Retightening of a single assembly also may dramatically alter the preload achieved depending upon lubrication.
It is important to release that the correct torque for any application can only be determined by testing the actual fasteners in application.

[ii] Sometimes technically incorrectly known as Foot . Pounds (Ft.Lbs)
[iii] Known as Pre-Load, (W) is an interesting "constant". The North American practice is to use 75% of the yield strength of the bolt multiplied by the tensile stress area of the bolt. If one refers to Ajax Fasteners technical publications, 65% of the proof stress and sometimes 65% of yield is used. In fact, in most Hobson publications we use 65% as well, however I tend to now use 75% as most good technical publications are now from the USA. It is interesting to note, just the variation this difference makes in the calculation will illustrate just how "approximate" recommended tightening torques are.

[iv] As distinct from friction factor, although there is some relationship between the two, they are definitely not the same thing)

The K factor is another source of variation. The table below gives a few common coatings and K factors that the Industrial Fastener Institute publish, however for critical applications these factors should always be individually determined.

FINISH TYPE	K FACTOR
Black Oxide	0.20
Cadmium Plated	0.19
Geomet 321	0.22
HDG (clean & dry)	0.23
Inorganic Zinc	0.17
Light Oiled	0.20
Moly	0.18
Phosphate & Oil	0.18
Plain	0.20
Xylan 1424	0.10
Zinc Plating (cr+6)	0.22

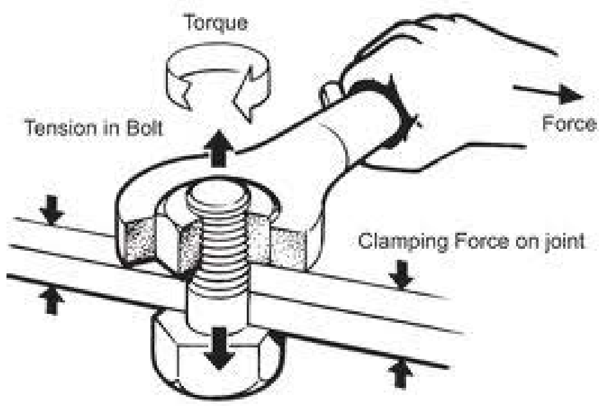
Let's do an example:

M10-1.5 Class 10.9 plain

T[i] = WKD

W = *0.75 (75%) x 900Mp (Yield) x 58mm2 (Stress Area) = 39,150N*
K = *0.20*
D = *10*
T *= 39,150 x 0.2 x 10*
= 78,300Nmm / 1000
= 78.30Nm

If 65% of yield was used, T = 67.86Nm. If one refers to the Ajax Fasteners handbook Issue 99 page 60, one will see a figure of 63Nm. This figure was calculated using 65% of the proof stress (830Mpa) rather than the yield stress (900Mpa).



So you can see, even in this simple formula, large variations (25%) can occur with just differing practices. Once the K factor variations are included, the variations are very significant. So you are probably thinking at this point, what is the use of this formula? ...It has its place; however there are *several* methods suitable for the real world.

Method # 1: Electronic Torque-turn tension for establishing tightening values

This technique involves using sophisticated and expensive force measuring load cells in the joint. Torque and tension values are recorded on a chart creating a torque-tension curve showing how much tension is created in the joint as progressively more torque is applied to the fastener. Turning and tension are recorded on another chart and hence a “turn of nut” table is created that can be used on the shop floor.

The disadvantages of this system is the cost of the equipment, however the following yield torque method is a good alternative. It is not as good, but it is a LOT BETTER than using recommended tightening torque tables.

Method # 2: Yield Torque method for establishing tightening torque values

- Assemble say 10 application joints exactly as they are to be done in production.
- With a calibrated torque wrench, drive the fasteners in the joint to yield and record the “yield torque” value. The “yield torque” is the value at which the indicated torque does NOT increase as the wrench continues to rotate.
- Take an average of the 10 figures above. Multiply this figure by 75% to establish the tightening torque. 75% of the yield point is the accepted figure used in North America. Although 65% is still used by some engineers in Australia.

So, hopefully many of you are now asking, well this is all well and good, but what do I do when I get a call from our client saying *“Your crappy bolts are breaking on installation….. I want new ones…. yesterday!!!”.* Common …. Right ??

My suggestion is to tell them that 90% of all fastener failures at installation are related to torqueing issues rather than anything being wrong with the fastener. This of course will not work…. You will get a few more expletives about rubbish bolts! I suggest the following; it really is the only way to resolve these issues.

1. Go to the clients premises and ask to see some assemblies that have the subject parts installed.
2. Using a permanent marker, draw a line on the corner of the bolt or nut and extend it onto the surface application.
3. Loosen the subject bolt or nut until there is no tension in the fastener.
4. Draw a line on the outside of the socket that corresponds with the line on the corner of the bolt or nut.
5. Affix the socket on a calibrated torque wrench that has a greater capacity than the determined tightening torque value.
6. Rotate the bolt or nut with the torque wrench until the line on the outside of the socket lines up with the line marked on the surface of the application indicating the original seated position. The value indicated on the torque wrench at the point where the marks line up is very close to the value that was originally applied to the bolt or nut when tightened by the customer. I strongly recommend this is done on several assemblies to get a better sample of what has been occurring on site.

In the vast majority of cases this method will reveal that the installer is tightening the suspect bolts at a torque value significantly greater than what they intended. Tightening bolts to torque values beyond the bolts ultimate strength during the assembly process is the most common source of broken bolts during installation.

Method # 3: Analysis of Gasketed Joints

Gasketed joints are complex and there is no easy method of understanduing their behaviour. Gaskets are installed at joint interfaces with the sole purpose of deforming to prevent leakage. The real problem is determining the recommended tightening torques in fasteners that secure gasketed joints. If it is too low, the joint will leak and if too high, he gasket will crush or warp and allow leakage. Frequently the most desirable level of preload can only be achieved through tightening and retightening although, Hobson Engineering recommends consultation with gasket suppliers as many have developed programs based upon field results to supply recommended tightening torques for use with their gaskets.

Schnorr HDS / HS Load washers are belleville shaped washers used to maintain tension on a static joint. Such as those found in flanges around gaskets.
HDS / HS washers provide the optimum compensation for setting losses in a bolted joint or losses caused by thermal expansion and contraction (prevalent in high temperature gaskets). The washer's spring characteristics allows the washer enlarge or compress which maintains the original clamp load and prevents leakage.
HS washers provide the same forces as HDS washers but with a smaller OD for confined spaces.

HOW LONG SHOULD HEX COUPLERS BE?

Hex couplers are used to join two threaded components together, in most cases one will be threaded rod. As a general conservative rule, thread engagement should be at least the diameter of the fastener. Due to the depth of a coupler, it is not always easy
to determine how far the thread has engaged, so couplers are made longer to allow some adjustment.

The Industrial Fasteners Institute is the accepted reference standard for Couplers, namely IFI-128:2000. This standard states the length of hexagonal couplers to be 3 x their diameter. So for example a 1"unc coupler is 3" long.

It is very important to ensure the Property Class (Grade) of the coupler matches that of the fasteners it is used to join. For this reason, Hobson stock Class 5, Class 8 and 316 (A4-50) stainless steel hex couplers. Our HDG class 8 are stamped and NATA test reports are available on line.

Hobson Newsfeed :

Peter turns 50!

Thomas Jefferson once said: *"In matters of style, swim with the current; in matters of principle, stand like a rock."*
At Hobson Engineering, not too much happens by sheer luck, so we are fortunate to work for a leader who doesn't create followers, but rather, more leaders.
Congratulations to Peter on his recent 50th birthday celebration.