Fastener Head Strength

Defining a Common Metric and Why It is a Critical Characteristic
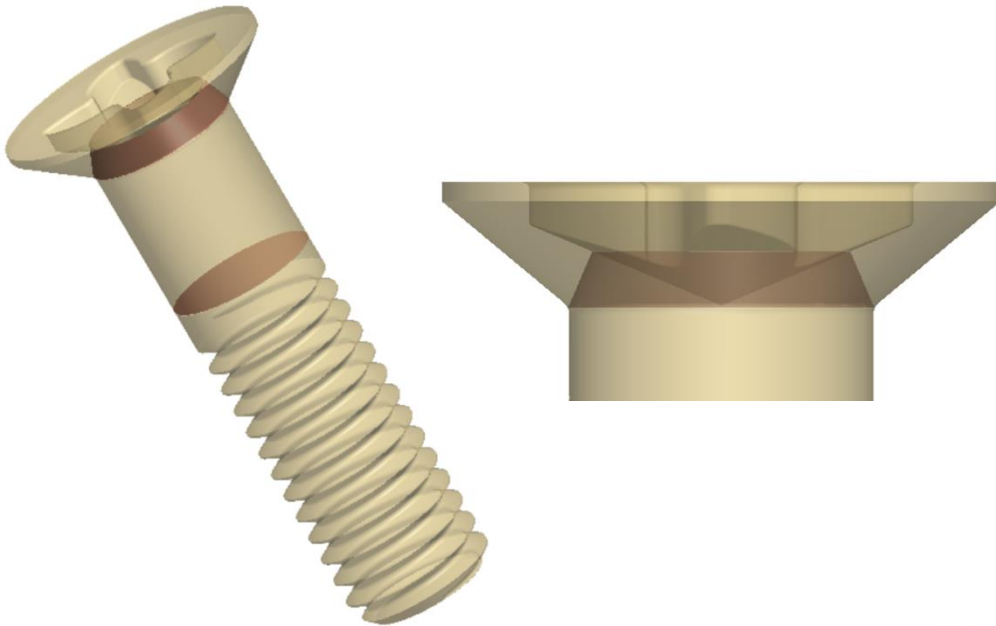
April 2018

High strength threaded fasteners are critical components in the assembly of virtually every form of transportation and industrial machinery. Industry accepted standards for the measurement and performance of these critical parts have existed for decades, but recent failures of threaded fasteners in service and qualification testing are bringing new focus to the critical area between the head and shank of the fastener. This head-shank juncture has been highly analyzed in large diameter (≥ ½"/12mm) high strength externally wrenched bolts (Hex Head, 12 Point, etc.) used in automotive and aerospace applications, but less attention has been paid to the many bolts and screws below this threshold.

Recent qualification failures of M5 (0.197") diameter fasteners made to a European standard have brought new focus on this issue. The aerospace industry has always been concerned with the weight of the airframe structure and the thin materials used in airframe construction necessitated the use of 100° countersunk flush head fasteners. These thin profile heads as well as other low profile designs intended to reduce weight in aerospace, automotive and industrial applications present unique challenges for the design of an effective torque transfer mechanism (internal recess or external head shape) while still assuring head to shank integrity.

Fastener Standards Development Organizations (SDO's) and company fastener standards engineers focus their attention on developing part standards that provide attributes (length, diameter, head diameter, head height, etc.) that can easily and accurately be measured to confirm conformance to form and fit requirements. Unfortunately, it is difficult to accurately measure the critical stress area between the top of the shank of the fastener and the bottom of the internal recess or lightening hole in an externally wrenched head without destroying the fastener to perform the measurement.

A consensus must be reached on how to calculate the Head Strength Ratio (HSR) to achieve the minimum acceptable tensile strength for the head to shank juncture and what method and measureable data should be used in the calculation. This white paper sets forth the current design limitations and a practical strategy to effectively assess the Head Strength Ratio (HSR) on current and future fastener designs.

The Problem:

During recent reviews of qualification test data for Aerospace series - Screws, 100° countersunk head, six lobe recess, threaded to head, in titanium alloy TI-P64001,anodized, MoS2 coated - Classification: 900 MPa at ambient temperature)/ 350° C it was found that a batch of M5 diameter parts failed to meet the required tensile test limits with lower than required failure levels in the head to shank juncture. Through investigation it was found that the prior approval parts had previously been qualified by analogy with a different standard, Aerospace series - Screws, pan head, six lobe recess, coarse tolerance normal shank, medium length thread, in titanium alloy, anodized, MoS2 lubricated - 1100 MPa at ambient temperature)/315 °C which has a different head style (pan head), a different shank configuration (coarse tolerance normal shank) and higher material strength (1100 MPa). Additionally, because of the different head style and shank configuration the pan head part has a much different Head to Shank Juncture geometry than the 100° countersunk head that is threaded to the head. Given the differences between these designs and materials what method or data should be used to qualify similar or dis-similar parts by analogy or to evaluate the potential tensile strength capability of a given head to shank geometry?

A poorly designed head and recess configuration can result in fasteners prematurely failing to meet tensile strength requirements with lower than required failure levels in the head to shank juncture. This white paper discusses what method or data should be used to ensure that the strength around this area is maintained such that the fastener is capable of meeting the design tensile strength requirements.

Defining the Variables:

For the purpose of this exercise we will assume a basic level of understanding of the tensile strength of various materials and how it affects the mechanical properties that a specific design can achieve. The variables in the head to shank juncture area of a fastener that influence tensile strength capability can then be limited to the effective geometry of the head to shank interface while the tensile strength of the threaded portion of the fastener can be based on the effective tensile stress area of the thread itself . We must define the diameter to be used to calculate the effective tensile stress area of the portion of the fastener as it transitions from a threaded area to an unthreaded area. That "effective diameter" is different for a given thread geometry (a 10-32UNJF for example) as it transitions to an unthreaded shank (a bolt with a grip length defined by an unthreaded portion) as opposed to when it transitions from a fully threaded part (a screw that is threaded to the head) to the underside of the head of the fastener. In a bolt, the thread transitions to a full body diameter (0.190" for our 10-32 UNJF example) and the result is that the weakest tensile stress area occurs at the thread pitch diameter one or two threads below the thread to full body transition. The thread pitch diameter is often equivalent to the blank diameter (the diameter of the unthreaded fastener before the thread profile is rolled onto the base material) that the fastener manufacturer uses during the production process.
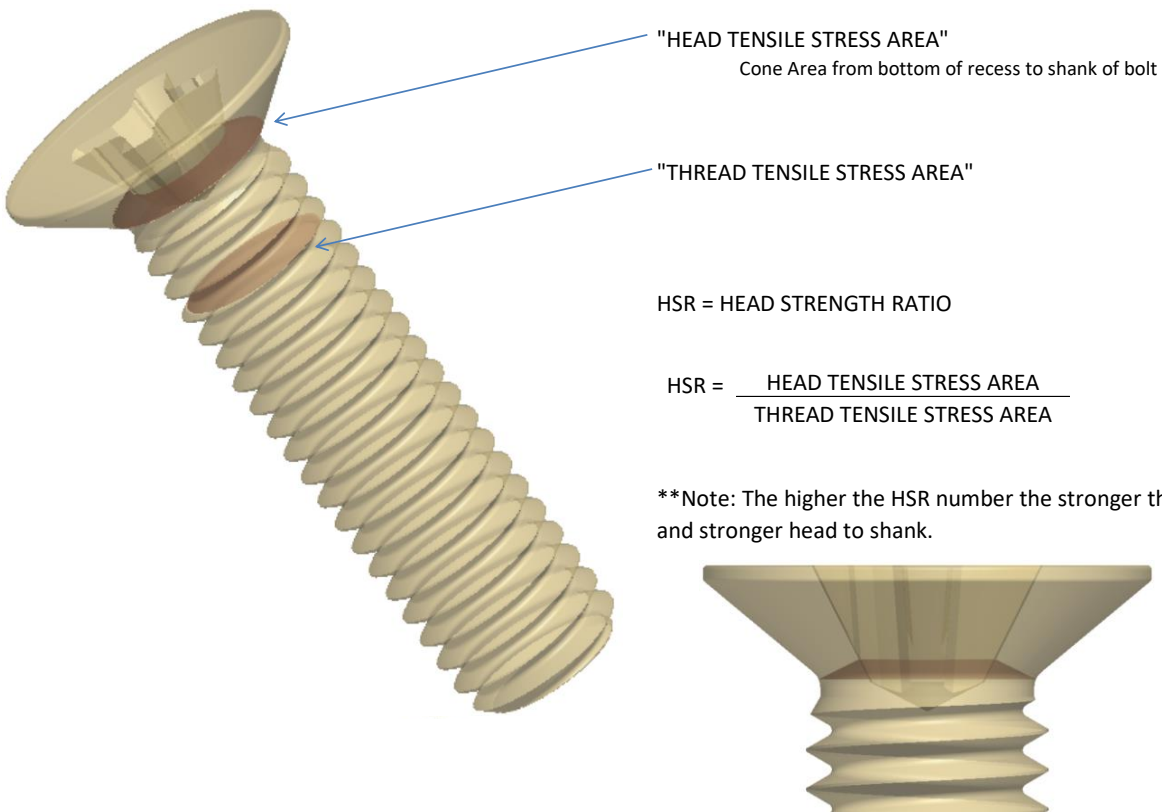
In the case of a fastener that is threaded to the head the minimum tensile stress area often occurs in the area between the last full thread and the bottom of the head

or the thread to head transition zone. Here the effective tensile stress diameter may be the same as the thread pitch diameter (0.1658"min for our 10-32UNJF example) or it may be slightly smaller if this area is fillet rolled to relieve stress built up during the manufacturing process.

This is significant because the difference in the effective tensile stress area in the head to shank transition will be different for a full body bolt than for a threaded to the head screw due to this dimensional difference at the bottom of the head between 0.190" for the bolt and the smaller 0.1658" for the screw. The stress cone, or the effective cross sectional area between the shank to head transition and the bottom to the internal recess, is calculated using either the larger diameter of the bolt transition or the smaller diameter of the fully threaded transition.

The calculation of the Head Strength Ratio then becomes a comparison of the Stress Cone effective tensile area with the full shank diameter for a bolt or the thread pitch effective tensile area for a fully threaded screw. Ideally the Head Strength Ratio will always be 1 or greater so that a tensile failure of the fastener will always be at or above the minimum tensile strength of the thread pitch diameter. This assures that there is enough strength in the head to shank juncture to avoid "popping" the head off the fastener and failing the joint.

"HEAD TENSILE STRESS AREA"
  Cone Area from bottom of recess to shank of bolt

"THREAD TENSILE STRESS AREA"

HSR = HEAD STRENGTH RATIO

$$HSR = \frac{HEAD\ TENSILE\ STRESS\ AREA}{THREAD\ TENSILE\ STRESS\ AREA}$$

**Note: The higher the HSR number the stronger the bolt and stronger head to shank.

Fully Threaded Head Strength Ratio Diagram

The diagram above shows the tensile stress areas (shaded areas) in the thread profile and in the head to shank juncture area for a fully threaded fastener. Note in the head cross section that the stress area goes from the under head diameter that is approximately the thread pitch diameter to the nearest point of the bottom of the recess. In this case the recess is actually cruciform in the cross sectional area nearest to the head shank juncture. For the purpose of comparison with other recess fastener drive systems we will ignore the minute amount of added material between the wings of the recess and only use a circular area based on the outer diameter of the wings at the cross section.

Calculating the Head Strength Ratio:

Now that we have an agreed upon set of criteria that we will use for the calculations it is important to select whether the minimum or maximum of each dimensional variable should be used to determine the "worst case" scenario in which head to shank failure is most likely at levels below the thread pitch effective diameter tensile failure levels. We have already determined that for a fully threaded fastener the cross sectional area under the head is determined by the blank diameter and that for a fastener with a full body the larger body diameter will be used for the calculation. We should therefor use the minimum under head diameter as the base line for the calculation. Looking at the diagram below it is easy to see that the maximum recess depth should be used as the base line for the calculation since it yields the smallest cross sectional area between the blank diameter and the nearest intersection with the recess. Typically, fully threaded fasteners are not used in critical applications and in these circumstances it is often acceptable to allow a Head Strength Ratio that is less than 1.0 based on analysis of the application.
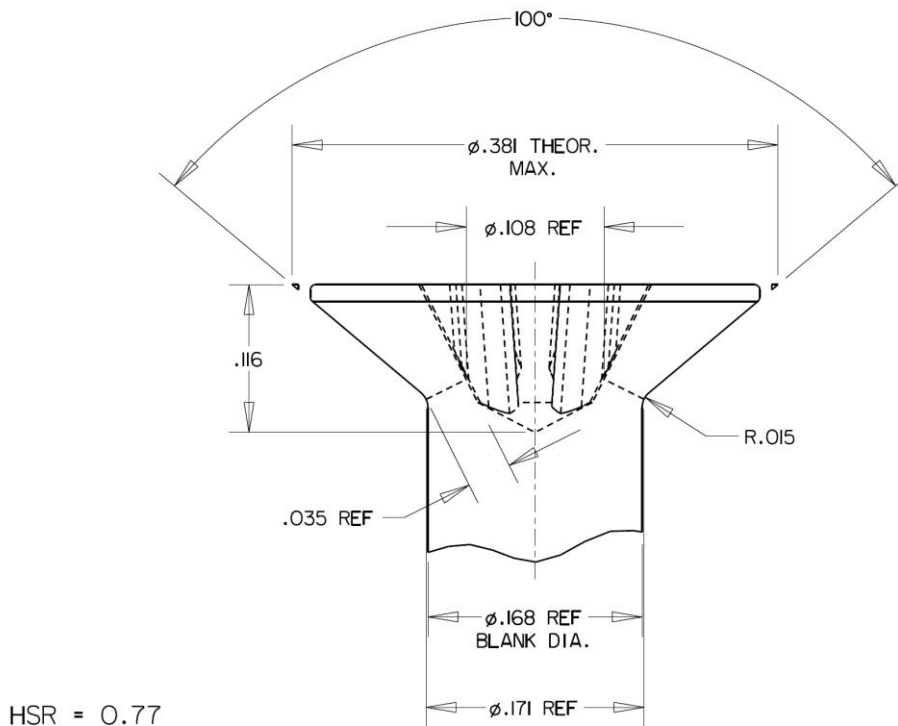
Range of total recess depth: 0.093 minimum to 0.116 maximum.

Influence of Recess Geometry on Head Strength Ratio:

The last factor to consider when analyzing the head strength in the head shank juncture area is the geometry of the recess drive system itself. Cruciform drive systems like Type 1 (PHILLIPS), NAS33781 (TORQ-SET), Type 1A (POZIDRIV), etc. have a defined cruciform shape for most of the recess depth from the top of the head to the bottom of the recess. In calculating the head strength ratio we typically will ignore this shape and use a basic circular cross section based on the outer wing diameter at the closest intersection to the under head transition radius for ease of calculation. Straight walled drive systems like Six Lobe (TORX) and others have a constant outer shape from the top of the fastener head to the tapered bottom of the recess. In these cases the distance from the head transition radius to the closest intersection with the recess shape is still used except that the nearest intersection is often a circular area in the bottom of the recess inside below the plane where the recess outer form has transitioned to a circular cross section.
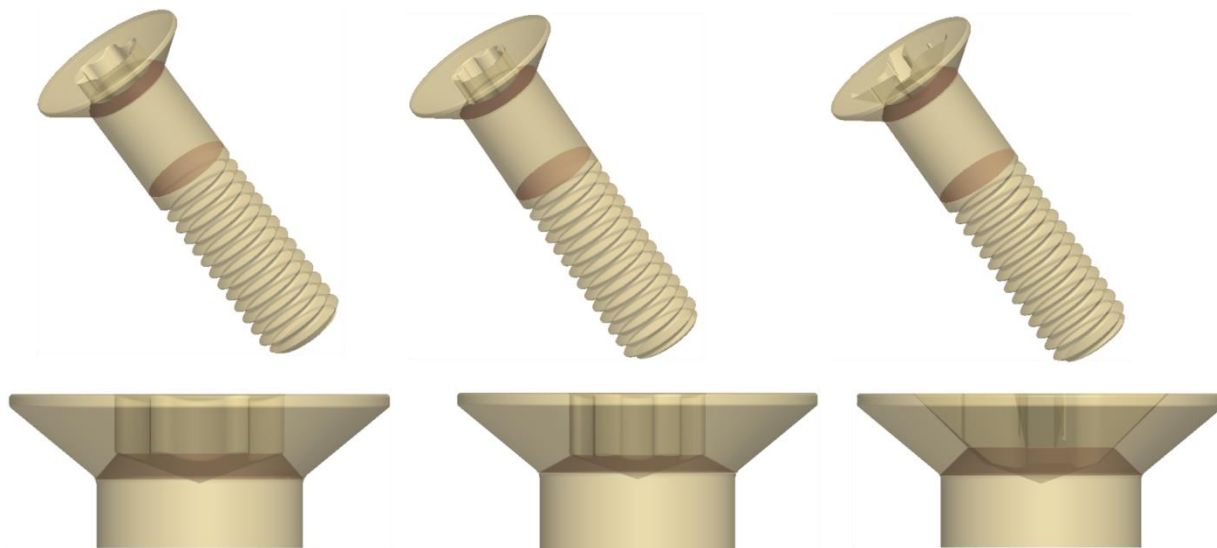
| NAS1800 (Six Lobe)	Enhanced Six Lobe	NAS33781
HSR: .88	HSR: .92	HSR: 1.01

## Comparison of HSR for Bolts with Unthreaded Shank Beneath the Head

Fasteners with unthreaded shanks are typically used in higher stress structural applications and in these situations an HSR of 1 or greater is recommended. Ideally the maximum recess depth will be shallow enough to provide a Head Strength Ratio greater than 1 especially if the fastener is in an application that is deemed critical or semi-critical. While the configurations shown above show an HSR below or near 1 at maximum recess depth, if the parts are manufactured with the recess depth closer to the minimum allowed in the part standard the Head Strength Ratio will increase as the recess depth approaches this minimum limit.

While we have used 100° countersunk aerospace fasteners for our discussion the Head Strength Ratio should be evaluated for any other head configuration (pan head, fillister head, button head, etc.) during the initial design evaluation. While protruding heads like these will often have a higher HSR the move to lighter weight and tighter tolerances has begun to require lower protruding head height designs and the recess depth may not always be considered when head height reductions are done to accommodate tighter clearances.
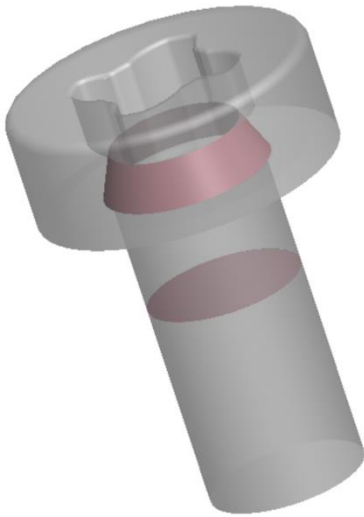
Fillister Head
HSR: 1.4
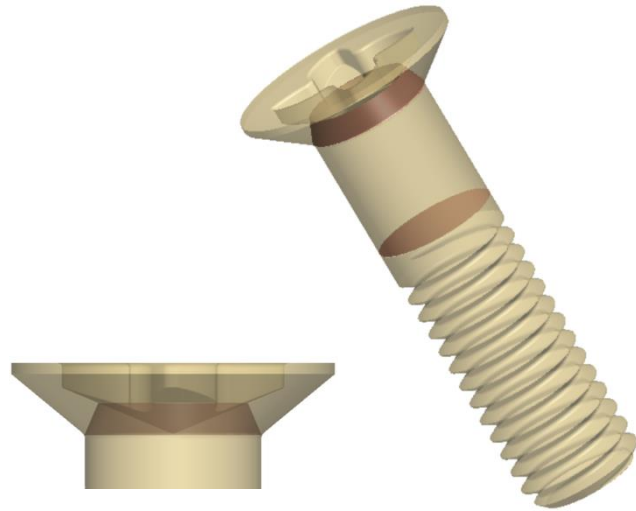
AS6305/EN4609
HSR: 1.11

Higher head strength is achieved with protruding head configurations where the bottom of the recess has significant distance from the head to shank juncture as in the Fillister head shown above. Increases in head strength can also be achieved in thin head designs if a shallow recess can be used (AS6305 MORTORQ spiral drive, NAS33750 Dovetail slot, etc.), however in a bolt application where significant torque is required the drive system should have a very high contact area between the driver and recess to accommodate the high torque required to achieve the needed clamp load.

Conclusions:

Recent failures of some fastener designs in tensile testing have highlighted the need for a better understanding of the dynamics of the head to shank juncture area. By developing a set of defined characteristics that can be evaluated to judge the Head Strength Ratio the design engineer can assure that there is sufficient strength in the head to shank area to at least equal the tensile capability of the threaded portion of the fastener.