An Experimental Study of Bearing Surface Friction in Modified Nuts

Final Year Thesis
Submitted 28th May 2018

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This thesis is presented in partial fulfilment of the requirements for the Master of Professional Engineering (Mechanical) at the University of Western Australia.

Word Count: 6841

Acknowledgements

This project would not have been possible without the support and assistance provided by following people; and to them I give my thanks.

The entire Segnut Team,
Segnut has been very supportive of my efforts to conduct a thesis with them from the very beginning. Even when faced with a wall of confidentiality issues, we found a suitable thesis topic. Segnut has supplied the location, test equipment, machining, time, knowledge, and large quantity of coffee beans that were required to conduct this testing and write this thesis.

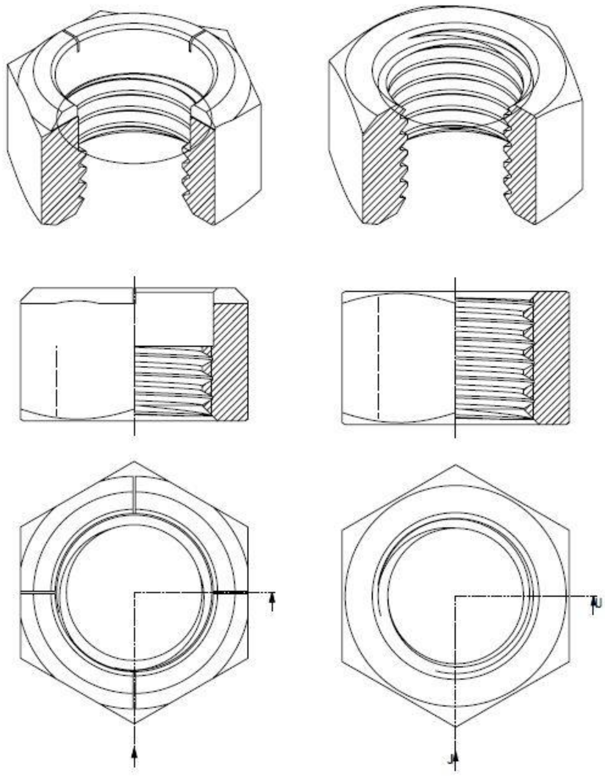
Specific thanks go out to Sam Turnbull for providing an abundance of advice from start to finish; ensuring I was on the right path and providing me with enough time to conduct the work required. I'd also like to thank Thomas and Dirk for helping out with the testing.

Westurn Engineering
Many thanks go to Marty, Daniel, and the whole team at Westurn for finding the time to machine my parts during their often very busy days, and for providing advice on all things machining.

Catherine Rose
For the continued support during all the late nights and long days. For the formatting and grammatical advice. And for brightening up every room you walk in.

Sue and Gerry Parry
For the encouragement you've provided me with throughout my education, and for always being there for me.

And to the mates who've kept me out of the Tav.



Abstract

Almost every complex engineering product uses threaded fasteners. Their major advantage over other joining methods is their ease of assembly and disassembly. SEGNUT Pty Ltd has an innovative self-named threaded nut which specialises in ease of disassembly. Their product prevents threaded fasteners from becoming seized from corrosion, thread damage, stripping or galling.

The most conventional method of tightening threaded fasteners remains the 'torque method', whereby the nut or bolt head is rotated until a predetermined torque is reached. Unless a direct tension indicating technology is used, there is no way to tell if the applied torque has produced the desired tension in the joint. The discrepancies in tension come from variations in friction at both the thread and bearing surface. For novel designs such as the Segnut, further analysis is required to investigate the parameters influencing friction.

This experimental study uses modified conventional nuts to investigate the bearing surface friction torque produced when the nut-washer interface average stress and distribution of stress is varied for nuts of different hardness' and surface characteristics. This study found that a reduced bearing surface outside diameter increased the variation in friction such that it is not recommended to use them in unlubricated conditions. Increasing hardness produced benefits in both the magnitude of friction and its consistency. And evening out the load distribution at the bearing surface had little effect on the friction.

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1. Introduction

1.1. Threaded Fastener Background

The screw thread began standardisation in Britain during the 1860's. Since then, many innovations have been produced to alleviate some of the shortcomings present in the traditional nut and bolt.

"Contrary to first impressions, the subject [Fasteners] is one of the most interesting in the entire field of mechanical design… the number of innovations in the fastener field over any period you might care to mention has been tremendous" (Richard G.Budynas, 2015).

The purpose of using threaded fasteners is to create a non-permanent joint. Non-permanent joints have applications in almost every industry involving complex engineered products. These non-permanent joints must maintain their clamping integrity whilst withstanding a variety of large, often oscillating load conditions; even when exposed to the vast array of this planets harshest environments. With the conclusion of each use, the fastener should still allow for disassembly. This, however, is not always the case.

The goal when tightening a threaded fastener is to develop a desired stretch (tension) in the bolt. This results in the desired compression in the joint, allowing it to take the design load conditions. Shear load is taken by the clamping friction between the joint surfaces and axial load is shared by both the bolt and joint depending on the relative stiffness' (Richard G.Budynas, 2015). Thus, the loading capacity of the joint is directly related to the bolt tension. The most conventional method of achieving this tension is via the 'torque method', whereby the nut (or bolt head) is rotated until a predetermined torque is reached. The issue with this method is that torque is measured and not tension, and thus the actual tension is unknown.

Large Discrepancies in tension come from variations in friction at both the thread and the nut – washer interface (bearing surface). Ideally, this friction would be low and very consistent, preventing adhesive wear (galling) and ensuring that the applied torque results in the desired tension.

There are four main challenges that most innovative products within the industry aim to solve;

- Self-Loosening
- Unknown Tension
- Thread Load Distribution
- Nut Seizure

It appears paradoxical that two common failure modes of a threaded fasteners are self-loosening and nut seizure (the inability to loosen). However, nut seizure effects many different industries and their maintenance practices around the world.

1.2. Nut Seizure

A universal definition is difficult to find; however a nut can be considered seized if the tools used for its assembly are insufficient for its disassembly. A nut can also be considered seized if the torque required for removal results in thread stripping or snapping of the bolt.

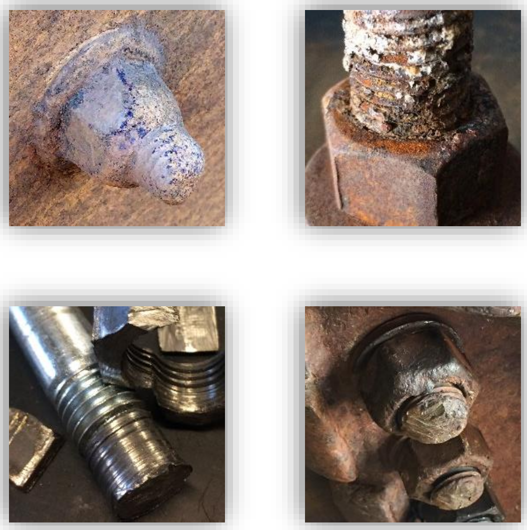


Figure 1 Seized Nuts

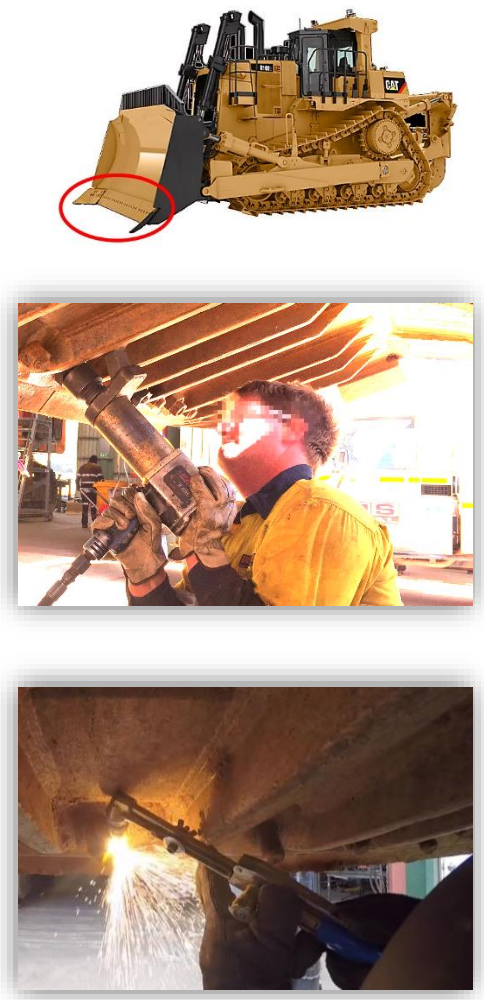


Figure 2 Cutting Edge Blade Replacement

An industry effected by seized nuts is that of the resources industry, shown above, where it's common practice in to remove nuts with an oxy cutter on ground engaging equipment. This increases downtime, reduces safety, and often damages the joint.

1.3. Segnut

The Segnut is a West Australian, Mandurah based innovation that was dreamt up on a long drive by inventor Brian Bradshaw. The Segnut avoids nut seizure by circumventing the thread. Removal of a Segnut does not involve rotation back up the thread. Instead, upon rotation of the outer sleeve, the inner segments move radially away from the thread, such that the nut pieces can then be slid off the bolt by hand.

1.4. Motivation

The Segnut has three inner threaded segments and an outer sleeve; as a result, it's significantly larger than a conventional nut. Any reduction in the bearing surface outside diameter of the Segnut would reduce its overall size. This would reduce weight, cost, and increase its ability to suit applications with tight packaging. This reduction however would not be supported by international fastener standards and thus requires thorough investigation and testing.

Figure 3 Segnut Images

2.0 Literature Review

2.1 Fastener Standards

The Segnut has various features that can be found on, or provide the same function as, a conventional nut. To increase product confidence, minimise risk, and reduce unnecessary design work, the relevant standards are consulted for these features. More specifically, these standards outline minimum and maximum quantities for a range of parameters, as well as (in most cases) the design principles used to develop those constraints.

For nuts, the relevant international standards are; ISO 898-2 (Mechanical Properties), ISO 4032 (Geometry) (same as AS1112.1 in Australia), ISO 16224 (Design Principles) and ISO 16047 (Torque – Tension Testing).

For features dissimilar to a conventional nut, the design is conducted from first principles, determining internal, product specific constraints. The bearing surface is a common feature between the Segnut and a conventional nut. However, the design principles for the bearing surface are not provided in ISO 16224 "Technical Aspects of Nut Design". Thus, there's no explanation as to how the minimum allowable bearing surface area was determined. Refer to figures 4 and 5.

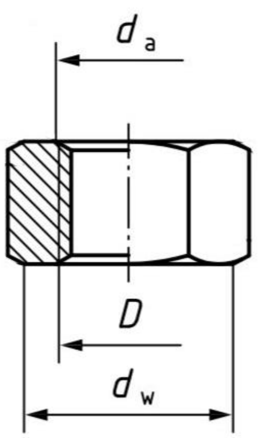


Figure 4 AS1112.1 Dimensions Side[1]

[1]Referring to AS1112.1, the maximum outer diameter and minimum internal diameter are specified as Dw and Da, respectively.

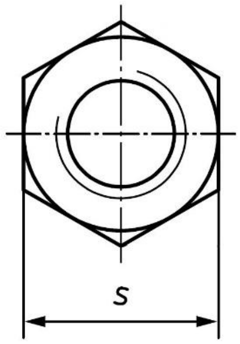


Figure 5 AS1112.1 Dimensions Top

It's reasonable to assume that the outside diameter of the bearing surface is largely influenced by the across flats "s" size of the nut. Referring to the image above, the nominal bearing surface outside diameter (OD) is the same size as the across flats. However, the across flats of a conventional nut is designed in conjunction with the 'effective nut height' to provide sufficient 'nut dilation resistance'. This nut dilation resistance ensures the desirable 'bolt breaking' failure mode, rather than 'bolt thread stripping' or, even less desirable, 'nut thread stripping'. Bolt breaking is the desired failure mode as this is quicker and easier to detect, enabling fast corrective action. Therefore, the nominal bearing surface OD is influenced by the required nut dilation stiffness.

The Segnut's dilation stiffness is provided by the outer sleeve geometry, whilst the bearing surface is defined by the inner segment geometry. Thus, the Segnut bearing surface is essentially decoupled from the required nut dilation resistance, enabling a degree of design freedom.

2.2 Bearing Surface, Thread Friction and K-Value

As previously mentioned, the most conventional method of tightening threaded fasteners is via the 'Torque Method', and this torque stretches (tensions) the bolt according to the thread and bearing surface friction. The combined effect of the thread and bearing surface friction is summarised in the K-Value; an experimentally derived dimensionless coefficient used to calculate Torque and Tension as shown in the equation below. Note, an increase in thread and bearing surface friction results in an increase in K-Value, and thus a reduced tension from the same applied torque. This is problematic if the calculated K-Value differs greatly from the actual K-Value, as this results in bolts at the wrong tension and consequential joint failures.

$$\boldsymbol{T = kFd}$$

$\boldsymbol{T}$: Torque (Nm)
$\boldsymbol{k}$: K-Value
$\boldsymbol{F}$: Tension (N)
$\boldsymbol{d}$: Thread Nominal Diameter (m)
(ISO 16047, 2005)

It should be noted that there are direct tensioning methods, resulting in significantly increased accuracy, however these are generally complicated and expensive for most applications - hydraulic tensioning is the most common.

There has been extensive research, both numerically and experimentally, into determining the effective radius of friction torque under nuts and bolt heads. This radius is a contentious issue as not only does the pressure distribution on the bearing surface depend on the thread load distribution, the sliding speed is also different as you move radially across the surface. To solve this issue, Nassar et al. (2005), created four different scenarios of contact pressure distribution and subsequent equations to determine the effective radius for each. The distributions were; uniform, sinusoidal, exponentially decreasing, and linearly decreasing. The standard (ISO 16027) method of determining effective radius is by assuming a uniform distribution and using the mean radius between the inside diameter (ID) and the OD, refer to the following equation. This was later found to be sufficiently accurate if “the ratio of the maximum to minimum bearing radius is relatively small” by Gong et al (2016) in their numerical study.

$$\boldsymbol{\mu_b = \frac{T_b}{0.5 D_b F}}$$

$$\boldsymbol{D_b = \frac{D_o + d_h}{2}}$$

$\boldsymbol{\mu_b}$: Bearing Surface Friction Coefficient
$\boldsymbol{T_b}$: Bearing Surface Friction Torque
$\boldsymbol{D_b}$: Effective Diameter
$\boldsymbol{F}$: Tension
$\boldsymbol{D_o}$: Bearing Surface OD
$\boldsymbol{d_h}$: Bearing Surface ID
(ISO 16047, 2005)

It has been found that statically analysing bolted joint connections significantly underestimates the stresses involved during tightening. During a numerical study (Paul Copeland, 2006), the von mises stress on the bearing surface was found to be 613Mpa for their dynamic model and only 211Mpa for their static model, stating that the large difference was due to the rotational shear forces produced in the dynamic model.

2.3 Adhesive Wear

There are five types of wear; Abrasive, Erosive, Adhesive, Corrosive, and Fatigue. Adhesive wear is of the most concern due to its common occurrence in the fastener industry and the drastic effect it has on the friction coefficient of surface contacts.

Adhesive wear is a very serious form of wear characterized by high wear rates and unstable friction coefficient. Sliding contacts can be rapidly destroyed and sliding motion may be prevented by very large coefficients of friction and seizure (Stachowiak, 2014). Adhesive wear is common in fastener applications and is often referred to as ‘galling’. Adhesion is not observed on objects casually placed together due to the Earth’s atmosphere and terrestrial organic matter providing layers of surface contaminant such as oxygen, water, and oil. The first experimental observations of adhesion were under high vacuum conditions, showing a completely different tribological behaviour of common materials. All metals apart from noble metals are covered by an oxide film when present in unreacted form in an oxidising atmosphere. This film can be only a few nanometres thick however it prevents true contact between metals and hinders adhesion. With metals in contact with metals, adhesion is found to be related to their cohesive binding energy (Buckley, 1982). This strong adhesion observed between metals can be explained by electron transfer between contacting surfaces. The electrons are not bound by a rigid structure and providing that the distance between two bodies in contact is sufficiently small, less than a nanometre, they can move from one body to another (Stachowiak, 2014). As an example, the coefficient of friction of clean (no oxide layer) iron surfaces is very high, up to $\mu = 3$. However, the simple theory of adhesion fails to predict such high values, and the phenomenon of 'asperity junction growth' is considered. Asperity junction growth is a process that assumes very high normal loads and involves frictional shear stress' that increase the asperity contact area, which enables a larger tangential force to be sustained. This tangential force and contact area will grow until the maximum shear stress (yield) of the material is reached. This increases the friction coefficient, thus causing a positive feedback loop that results in rapid friction increase and seizure.

Bond strength at the interface is, with some exceptions, stronger then the bond strength of the cohesively weaker of the two materials. The greatest adhesion occurs for a combination of like materials, such as iron on iron, however many other combinations of unlike metals also show quite high adhesions (Stachowiak, 2014). The ratio of adhesion force to contact force can be very high, around 20 in some cases, with the bonding process occurring almost instantaneously. It has been found experimentally that metals with hexagonal close packed structure show much less adhesion than other crystal structures. High hardness, elastic moduli and surface energy of the metal also suppress adhesion. Alloys and composite materials are usually superior to pure materials in terms of adhesive wear resistance.

2.4 Load Distribution

The thread load distribution in a nut is non-uniform; this could affect the load distribution on the bearing surface. In fact, there have been recent studies involving changing the bearing surface contact area to even out thread load distribution (Brutti, 2017). Brutti recently tested washers with an increased internal diameter and found that they reduced the stiffness of the first engaged threads, evening out the load distribution.

It's interesting to note that the distribution can change for tension vs compression loading conditions. Below are graphs of spring models developed in the 80's showing both tension and compression thread load distributions (Miller, 1983). The first graph represents the conventional nut-bolt loading conditions and shows the first threads taking most of the load. The second graph is the turnbuckle (tension) case and shows the first and last threads taking the most load.

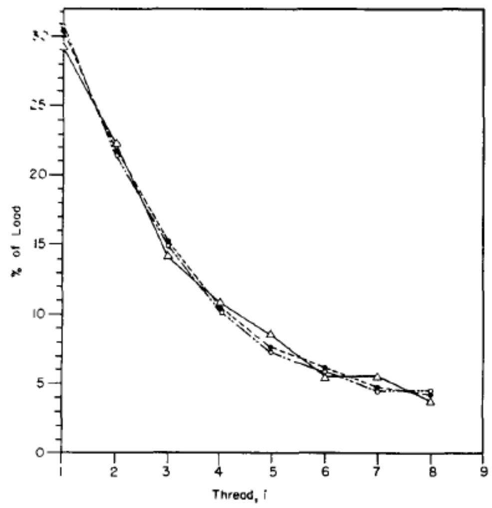


Figure 6 Compression Thread Distribution (Miller, 1983)

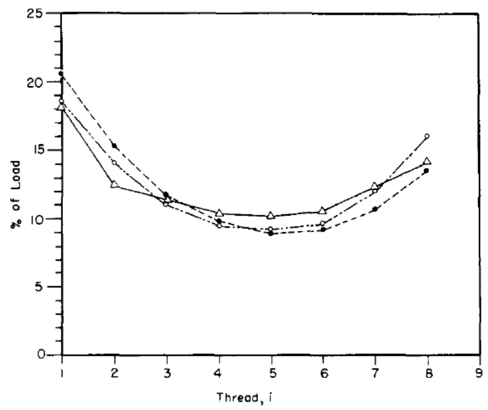


Figure 7 Tension Thread Distribution (Miller, 1983)

2.5 Coatings and Lubrication

It is conventional practice during many fastener installations to use an anti-seize lubricant on the thread of the bolt. It's application to the nut-washer interface is not as usual and therefore the unlubricated condition is a common occurring worst-case scenario. Silver Anti-Seize from Loctite is a commonly used lubricant that contains Graphite, Aluminium, petroleum hydrocarbons and Calcium Oxide (Loctite Silver Grade Anti-Seize Lubricant MSDS, 2003). It uses the mechanism of solid lubrication and should be used as a benchmark for any lubrication tests.

2.6 Speed, Temperature and Distance

The torque-tension relationship has been experimentally proven to be affected by tightening speed, as well as repeated tightening and loosening (S.A. Nassar, 2007). The effects of repeated tightening and loosening are more significant at low tightening speeds, less than 30 rpm.

It's also been found that the tribological properties of dry sliding surfaces are sensitive to surface temperature (Zhang Yongzhen, 2008). In a numerical study involving high tightening speeds and large rotation angles, the potential for localised melting was discovered (Nassar, 2008). This would have a large effect on the rate of surface wear.

2.7 Surface Finish, Roughness and Texturing

Many studies have been dedicated to the effects of surface texturing on hydrodynamic lubrication. It has been argued that, in cases of full or mixed lubrication, the dimples on the surfaces can act as micro-hydrodynamic bearings as well as traps for wear debris (Stachowiak, 2014). This could prove useful in maintaining lubrication on the nut-washer interface, as well as reducing friction change caused by wear debris. Textures can be produced by many different techniques such as milling, shot blasting, photochemical etching, or laser. The main problem with texturing is finding the optimum surface texture, as the common approach is a so-called 'exhaustive search', which is both expensive and time consuming.

Real surfaces are difficult to define; however, the surface roughness of components is critical as it determines the ability of surfaces to support load (Stachowiak, 2014). At least two parameters are needed to describe surface roughness, one describing a variation in height and the other describing how height varies in the plane of the surface (spatial surface characteristics). Parameters commonly describing Surface height characteristics are the roughness average (R_a) and the root mean square roughness (RMS or 'Rq'). The averaging effect from 'Rq' better describes the height of asperities. The second parameter describing surface roughness is the spatial characteristic. It is described by several statistical functions, one of which is the autocorrelation function (ACF). It has been found that at very high or very low values of 'Rq' only light loads can be supported, and that intermediate values allow for much higher loads. It is important that as manufactured surface conditions are preserved to prevent any deviation in results due to surface finish.

3.0 Experimental Design

3.1 Test Variables

Bearing Surface OD
As mentioned in the literature review, the fastener standards don't provide the design principles for the bearing surface area. However, for an M20 nut, product grade A, B, or C, AS1112.1 provides the minimum OD and maximum ID as 27.7 mm and 21.6 mm respectively (AS1112.1, 2015). This results in a minimum bearing surface area of 236 mm; which is coincidentally very close to the M20 coarse thread stress area of 245mm (ISO 898-2, 1992). Intuitively, this makes sense as this ensures that the average bearing surface stress is always less than or equal to the thread stress. This theory however is conservative when you factor in the hardness and, therefore, strength difference between nuts and bolts.

The proof stress for an M20 Class 8 nut is 920Mpa at 225.4kN, whereas for an equivalent 8.8 M20 bolt, the proof stress is 640Mpa at 147kN. This is to ensure bolt stripping failure occurs before nut stripping; thus the nut material is designed to be able to take a higher stress. Accordingly, we can determine the area that would result in the nut bearing surface reaching proof stress at the same tension as the bolt thread using the following formula;

$$A = \frac{F}{\sigma}$$

$$= \frac{147000}{920}$$

$$= 159.8\,mm$$

$$Bearing\ Surface\ OD \approx 26mm$$

Thus, the minimum theoretical bearing surface area would be 26mm. If this was reduced further, the bearing surface would yield before the bolt and increase the amount of non-rotational self-loosening that occurs after tensioning (Eccles, 2010); a concern which although outside the scope of this study should be kept in mind.

The first test variable is therefore reducing the bearing surface OD from 28.5mm to 26mm. This will increase the average stress and the potential for adhesive wear, thus potentially increasing friction force. It will also reduce the effective radius about which the friction force acts. As a result, the friction torque will comprise of a potentially increased force at a reduced radius. Thus, providing that severe adhesive wear is not present, the authors hypothesis was that reducing the bearing surface OD reduces the friction torque.

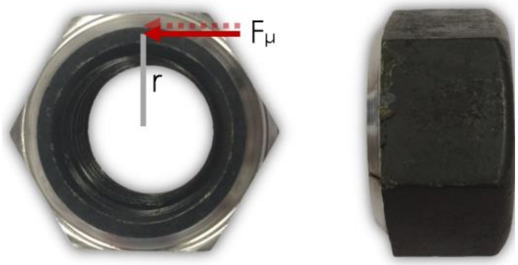


Figure 8 Reduced Bearing OD Test Sample

Hardness
Hardness testing is a convenient and non-destructive means of estimating the strength properties of materials (Richard G.Budynas, 2015). "Many experiments in the 1950's showed that adhesive wear is directly proportional to the distance traversed and the normal load, and inversely proportional to the hardness of the softer material" (Zeng, 2013). Hardness is therefore a desirable second test variable. Increasing the hardness should help offset any negative effects of increasing the stress from reducing the bearing surface OD. Nuts are conveniently classed in terms of hardness, thus both class 8 and 10 nuts will be tested.

Figure 9 Class 10 Test Sample

Surface Discontinuities

The Segnut bearing surface has three radial line surface discontinuities. These are a result of the bearing surface being comprised of three nut segments. To determine if this feature has any frictional effects, M20 sample nuts are wire cut to provide four 0.3mm line discontinuities.

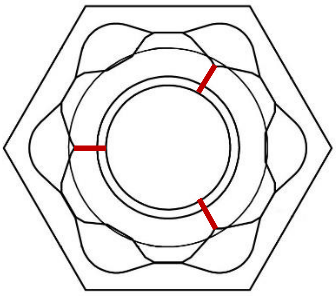


Figure 10 Segnut Surface Discontinuities

Figure 11 Surface Discontinuities Test Sample

Thread Cut-out

A conventional nut has an uneven thread load distribution, most of the load is taken in the first three threads. This load distribution causes a stress raiser in the first thread, negatively impacting the fatigue life of bolts, refer to the figure 12.

The cone angle compression theory is used to determine the stiffness of a joint under compression, refer to figure 13. The cone angle represents the cross-sectional area of material in compression, and for steel is assumed to be 30 degrees.

When this theory is applied to the uneven thread load distribution, an uneven bearing surface load distribution can be visualised, refer to figure 14. This is significant because the calculations made in the *Bearing Surface OD* section about both thread and bearing surface stress reaching proof stress at the same tension are based on average stress. This means that the uneven bearing surface load distribution could cause the innermost part to exceed proof stress, begin yielding, and initiate adhesive wear. This theory can be tested by removing the first three nut threads, ensuring that the 30-degree compression cone from the first engaged thread fully encompasses the bearing surface.

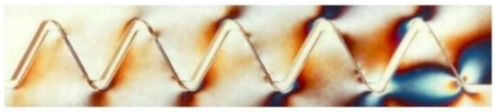


Figure 12 Uneven Thread Load Distribution (Stanley Engineered Fastening, 2018)

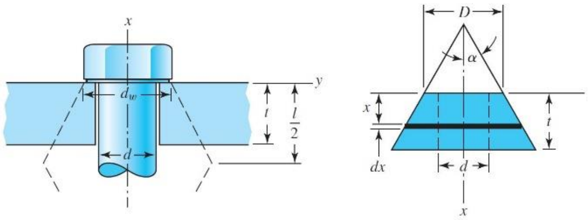


Figure 13 Compression Cone (Richard G.Budynas, 2015)

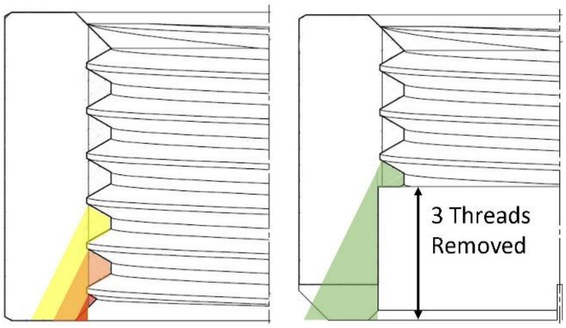


Figure 14 Bearing Surface Load Distribution

Figure 15 Thread Cut-out Sample

3.2 Objectives

1. Do the test variables correlate with an increase or decrease in friction torque?

2. Do the test variables change the consistency of friction?

3.3 Response Variable

The test response variable is the friction torque generated during tightening at 75% proof load of an 8.8 M20 Bolt (ISO 16047, 2005). This corresponds with approximately 110kN of tension.

3.4 Design of Experiment

To simultaneously test all four variables for their main effects as well as their interaction effects, a 2^4 full factorial test is implemented. This involves each variable having a high (+1) and a low level (-1), and every possible variable level combination being tested randomly. For four variables this corresponds to 16 tests, however one replication is introduced to obtain more information about the variance of results. This replication also allows for checking of the analysis assumptions to ensure that there is 'homogeneity of variance'; or in other words, that the response dispersion is uniform across the experimental space (Full Factorial Example, 2018).

Five centre points (0) are used to check the results for non-linearity as well as any potential time dependency. Unfortunately, the only test variable that is continuous and can have a true centre value is the Bearing Surface OD. Thus, this is the only variable for which nonlinearity can be tested. It is set to 27.7 mm to represent the minimum allowed by the standards. The other test variables were set to their lower levels.

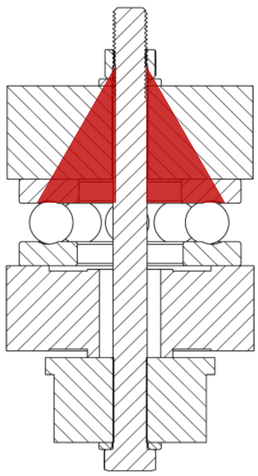
To summarise there are 32 randomised tests, with 5 equally spaced centre points, totalling 37 tests. A spare sample for each combination of factors was manufactured in case any tests needed to be repeated.

Variable	Levels		
	-1	0	1
Bearing Surface OD	28.5 mm	27.7 mm	26 mm
Hardness	Class 8	Class 8	Class 10
Surface Discontinuities	None	None	4 Lines
Thread Cut-out	None	None	3 Threads

3.5 Apparatus and Test

The testing apparatus is comprised of;

- Through-hole load cell to measure tension
- Strain gauges to measure bearing surface friction torque,
- Torque reaction arm to house strain gauges
- Thrust ball bearing to ensure all bearing surface torque was sent to the strain gauges
- Reaction stand to interface with strain gauges and mount torque converter
- Mating surface with 21mm hole as per fine series specified in ISO 16047,
- Mating Surface grooved and flame hardened to prevent washer spinning,
- 260mm 12.9 Bolt, enabled the same bolt to be reused,
- Plain Black M20 Washers
- Three Logitech USB webcams to record testing



The Ball Bearings are within the 30-degree compression cone; thus, any bending of the mating surface or unconventional load distribution will not occur from the addition of the bearing.

Figure 16 Test Joint Compression Cone

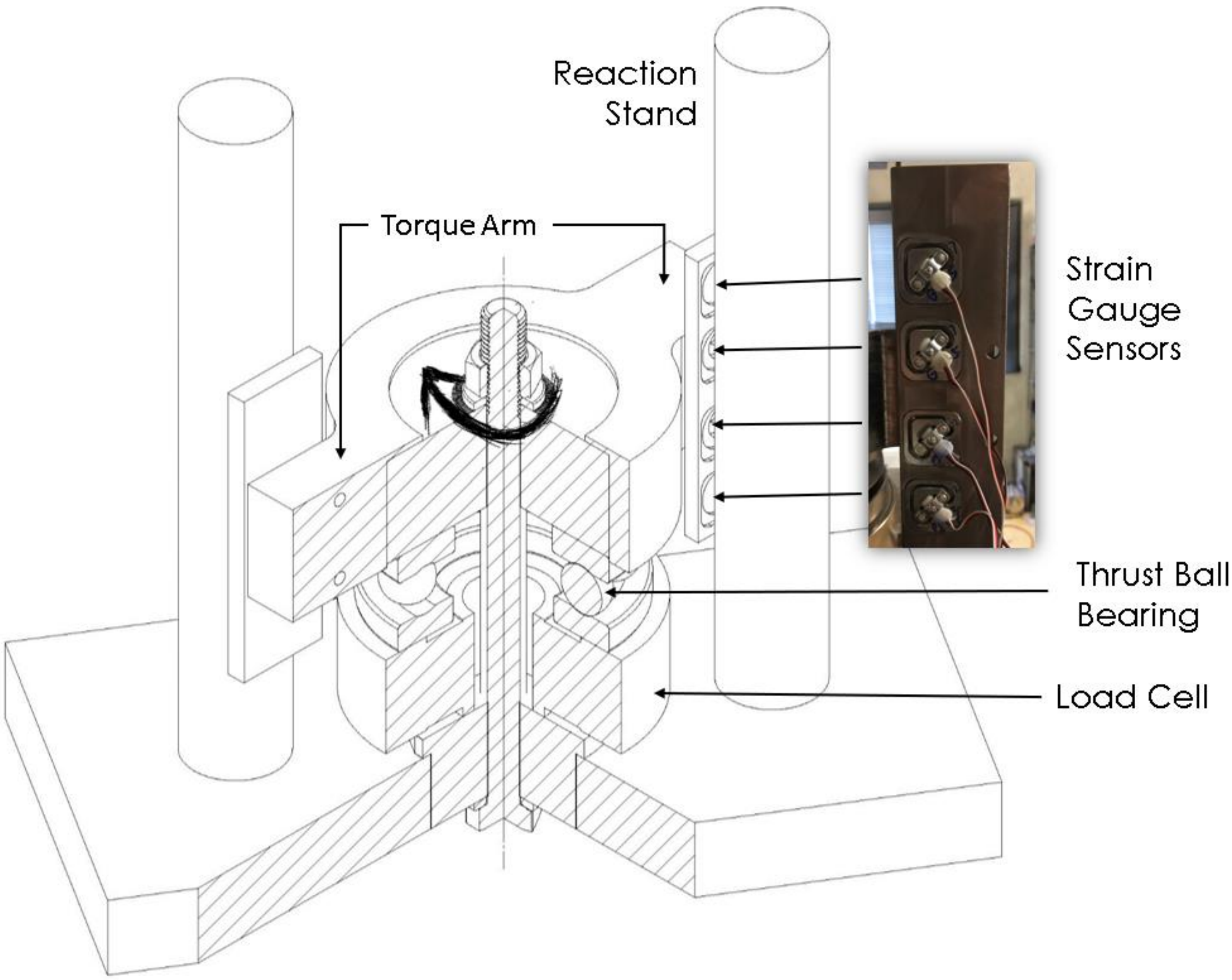


Figure 17 Test Apparatus Section View

The test procedure involved cleaning each sample and corresponding washer with acetone to ensure unlubricated conditions at the bearing surface. The nuts and washers were supplied with a light oil, however the coverage varied greatly between samples, therefore removal of this oil was appropriate. Good industry bolting practices always lubricate the bolt threads, usually with an anti-seize product, however the bearing surface is often still neglected. This is surprising considering that the bearing surface contributes more friction than the thread (Q. Zou, 2005). Thus, it is desirable to test these variables in the worst-case-scenario. To enable the reuse of the bolt between tests, anti-seize was applied to the bolt threads and great care was taken to ensure that this did not contaminate the bearing surface during assembly.

In order for all of the bearing surface torque to be transmitted to the strain gauges, the washer must be fixed rotationally relative to the torque arm. Preliminary test runs involved machining flats on each washer and a slot on the mating surface for the washer to sit in, these were successful however time consuming and could potentially affect the results due to modification of the washer. Instead, radial grooves were cut into the mating surface, which produces a similar principle to that of a Nordlock washer. However, for this to be effective the mating surface must be harder than the washer to ensure that the washer embeds into the grooves (and doesn't just flatten them). This was achieved by flame hardening the surface and applying a 'Cherry Red' hardening compound. The washer and mating surface were marked at the start of each test, and then checked for rotation whilst under full tension.

A 25:1 torque converter is used to aid in the hand tightening of all tested samples. The speed of tightening is not controlled during testing, and due to the strain gauge displays intermittently turning themselves off, the tightening was also non-continuous. Refer to Figure 18. This can potentially increase the friction and wear due to 'stick-slip', as static friction is larger than sliding friction.

Variable	Type
Input Torque	Measured
Tension	Measured
Bearing Surface Torque	Response
Grip Length	Controlled
Lubrication Condition	Controlled
Bearing Surface OD	Tested
Hardness	Tested
Surface Discontinuities	Tested
Thread Cut-out	Tested
Tightening Speed	Uncontrolled
Ambient Temperature	Measured
Contact Temperature	Uncontrolled
Test Duration	Uncontrolled

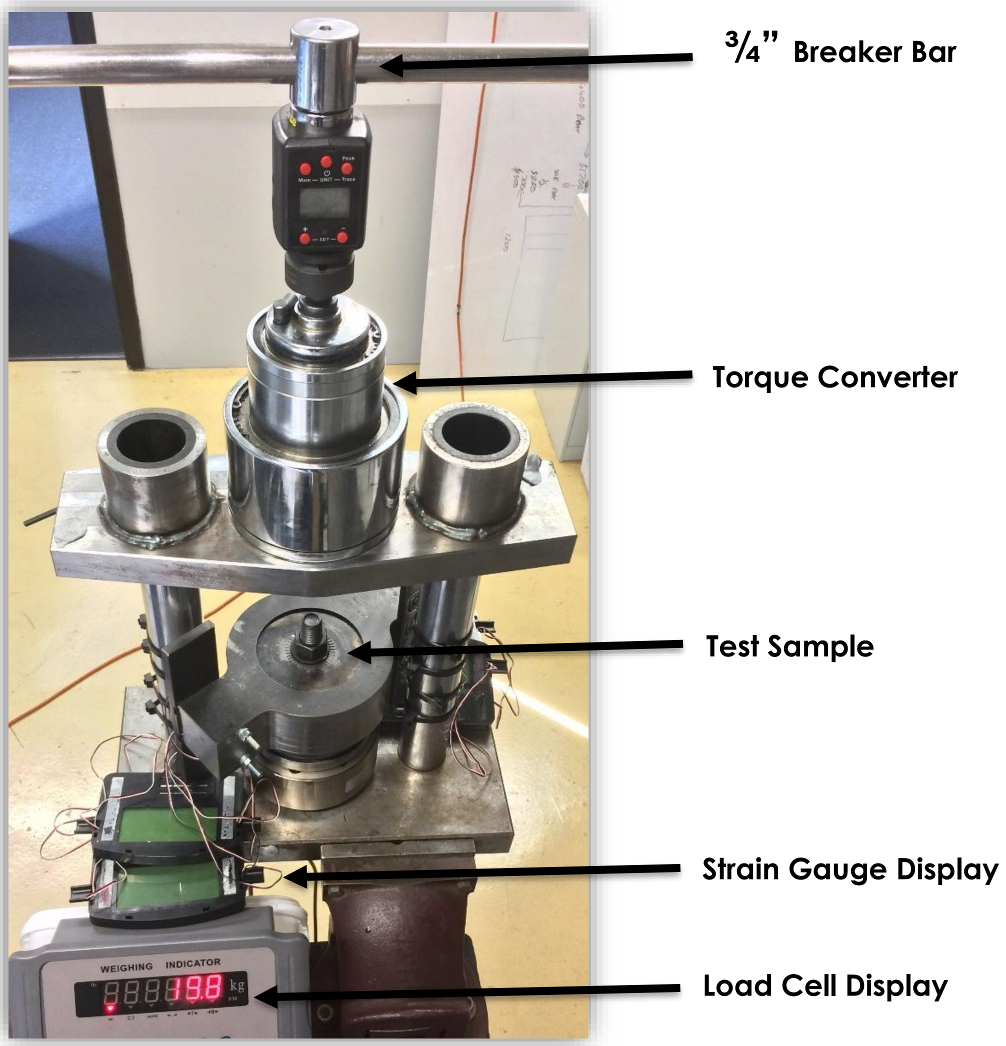


Figure 18 Full Apparatus Setup

3.6 Calibration

Two calibration tests were conducted to provide confidence in the test results; one for the load cell, and another for the strain gauges.

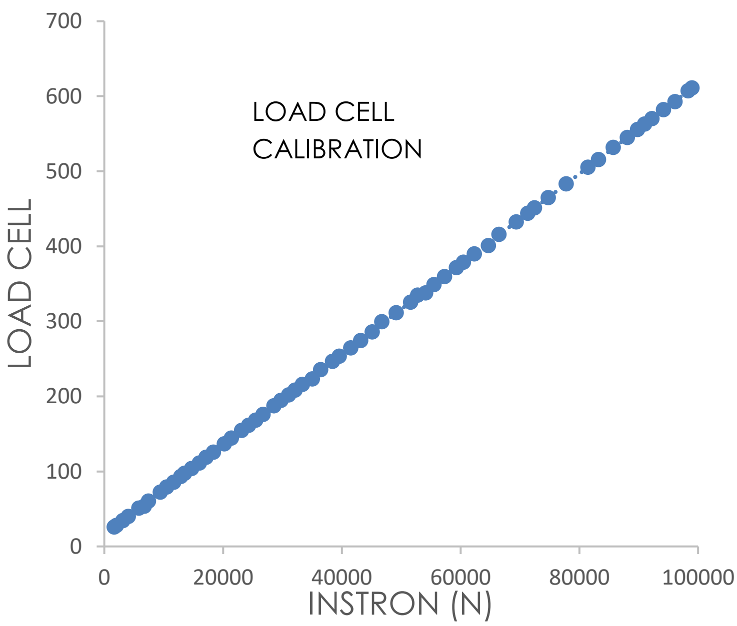
The load cell was taken to the UWA Civil and Mechanical Engineering building on 7th May to be tested on their 100kN Instron machine. The display was found to show loads that were significantly different to that applied, however the results showed great linearity, refer to the graph below. A linear fit was found, and the equation used to determine the 'actual' force being applied to the load cell.

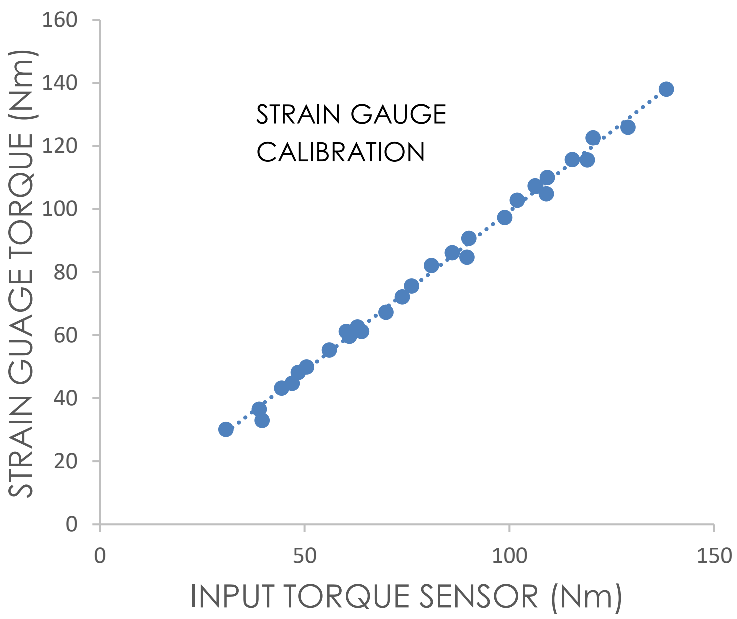
A Calibration plate was machined up such that a ¾" drive torque sensor could be directly fitted into the torque arm. This enabled a known torque to be applied to the strain gauges. Multiple increasing and decreasing torques were applied by hand and recorded on camera. The resulting data was linear and appeared to have minimal offset, refer to the graph below.

Figure 19 Instron

Figure 20 Calibration Plate

Figure 21 Torque Sensor





4.0 Results and Discussion

4.1 Sample Visual Inspection

Upon visual inspection of the samples, it can be immediately distinguished that the outer diameter of the bearing surface suffers the most coating removal and adhesive wear (refer to Figure 22). This is at odds with the initial assumption that the peak stress is at the innermost part of the bearing surface and that this would cause any wear or coating removal to localise there. The resulting wear pattern however could be attributed to the 32% increase in sliding distance at the bearing surface OD relative to the bearing surface ID as adhesive wear has been shown to be directly proportional to distance traversed (Zeng, 2013)

Even some unmodified nuts showed galling, emphasising the large variations in friction associated with unlubricated sliding contact.

Figure 22 Standard Nut Tested

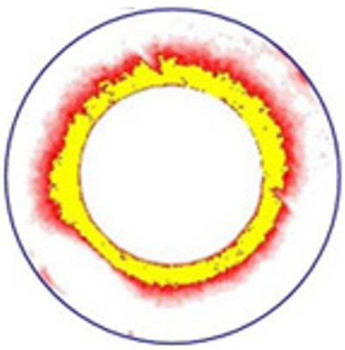


Figure 23 Assumed Pressure Distribution (Use of Washers and Flange Heads, 2018)

4.2 Data Validation

The run sequence and lag plots are used to determine if the results show any time dependency.

The red data points in the Run Sequence Plot represent the centre points, they show a slight downward time dependency, however the overall test results appear flat as desired. Both plots appear sufficiently noisy.

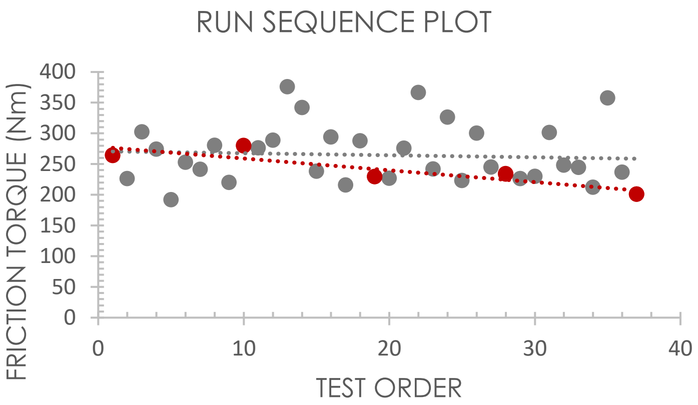
The Normal Distribution Plot shows some non-normality. Conducting an Anderson-Darling normality test, the P-Value is 0.0231, this is below 0.05 and thus the null hypothesis is rejected. Therefore, the data is not sufficiently normal. This will need to be addressed before a theoretical model can be constructed.

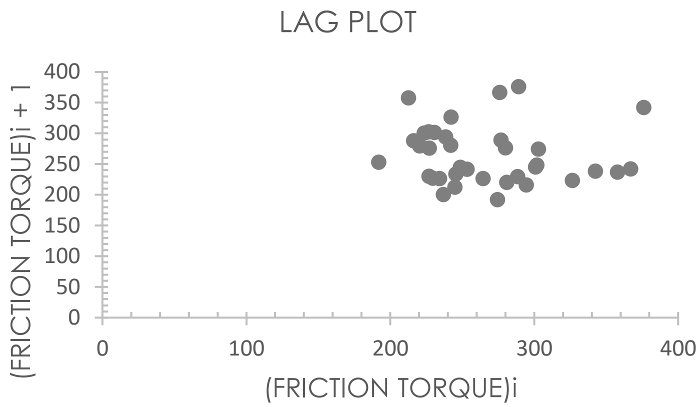
The Friction torque histogram appears to be right skewed. This is most likely due to the nature of galling; such that significantly higher friction is more likely to result than very low friction.

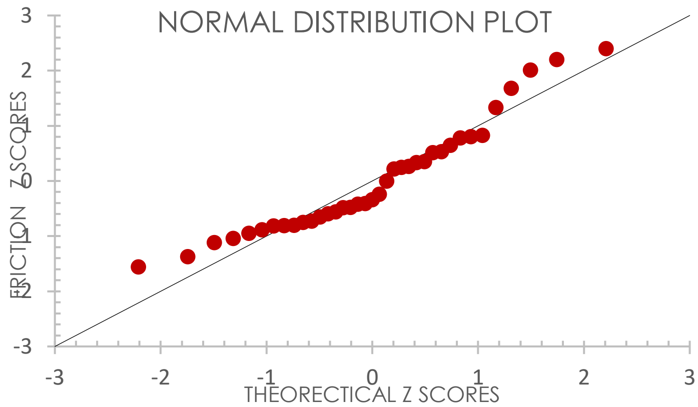
4.3 Mean Effects

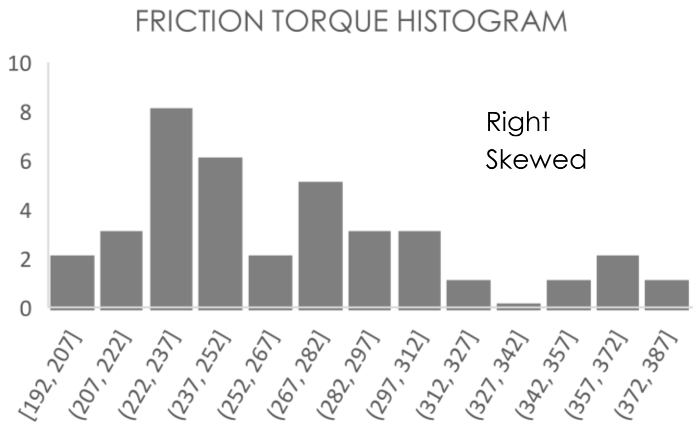
The mean effects plot compares all the tests that occurred with one variable in its low level against all of the other tests that occurred with the same variable at its high level. For example, the bearing surface OD mean effects plot, shown in red, averages all the 16 tests that were conducted with 28.5mm nuts and compares them with the other 16 tests that were conducted with 26mm nuts. This excludes any interaction effects and determines if the factor alone is significant enough to affect the average of all the tests.

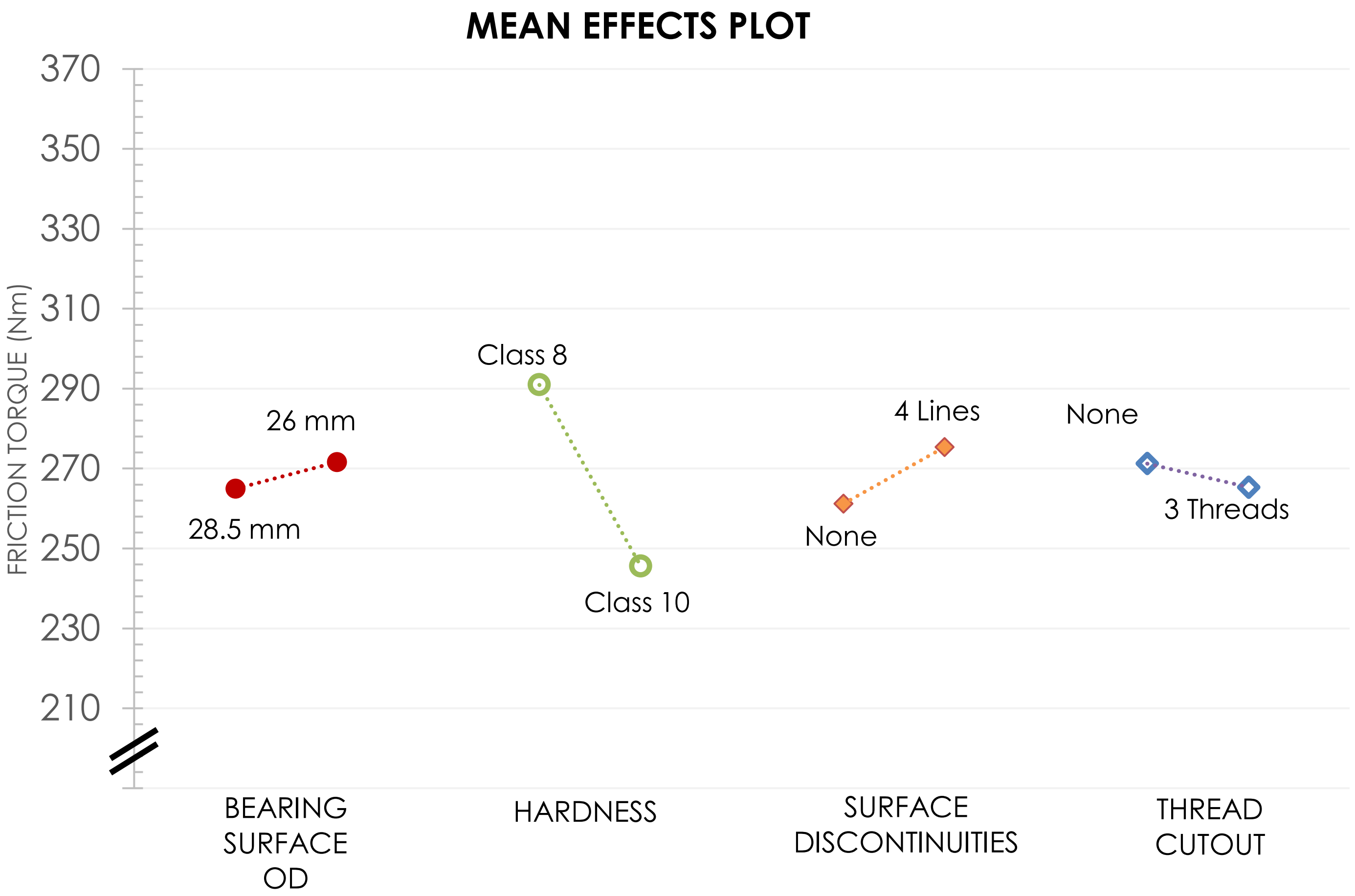
Referring to the plots, it should be noted that the graph's y axis is truncated and starts at 190Nm, which is slightly below the lowest test result, which was 192Nm. For context, the highest result was 376Nm, which is almost double the minimum.


RUN SEQUENCE PLOT
FRICTION TORQUE (Nm)
400
350
300
250
200
150
100
50
0
0
10
20
30
40
TEST ORDER

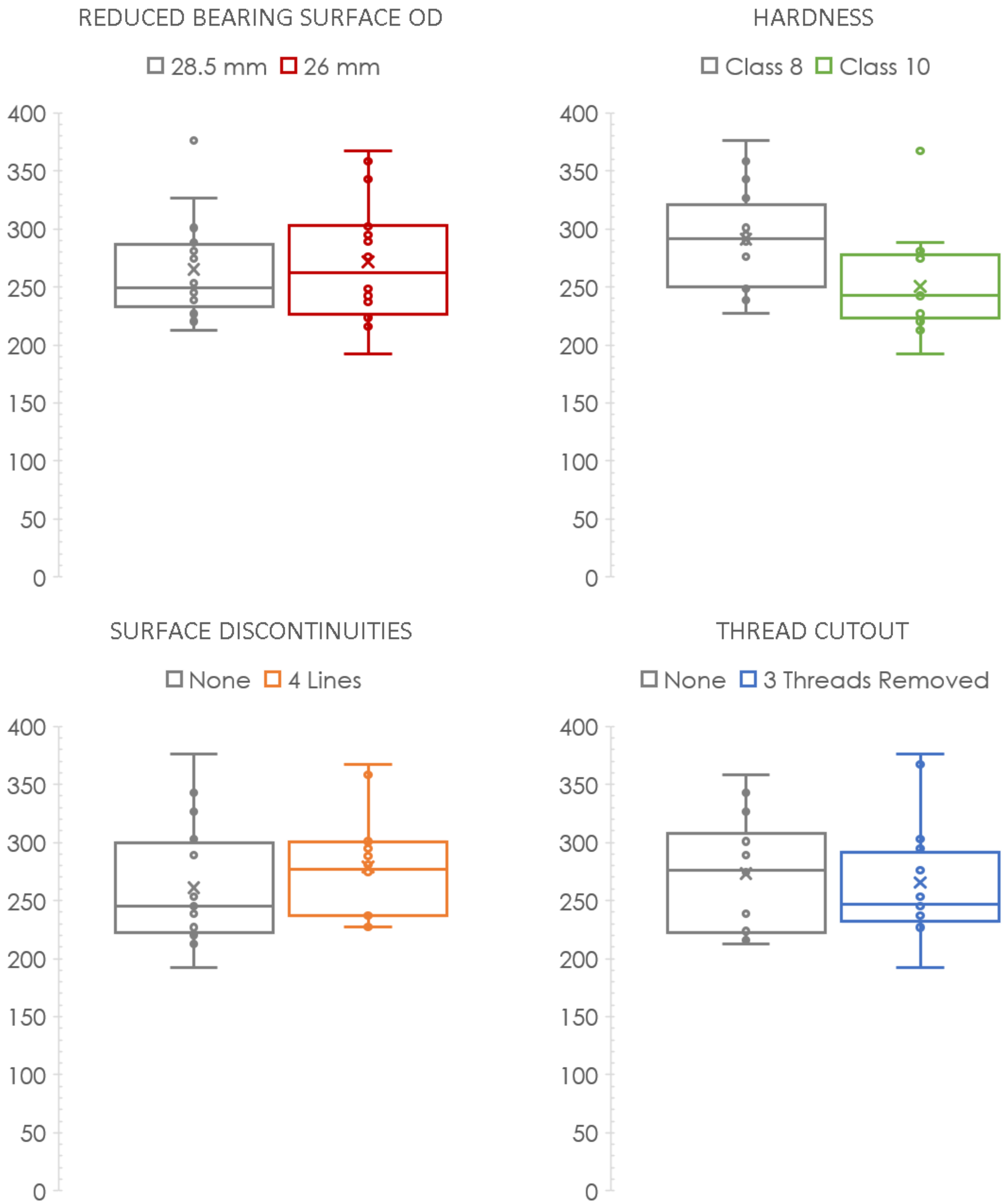

LAG PLOT
(FRICTION TORQUE)i + 1
400
350
300
250
200
150
100
50
0
0
100
200
300
400
(FRICTION TORQUE)i


NORMAL DISTRIBUTION PLOT
FRICTION Z SCORES
3
2
1
0
-1
-2
-3
-3
-2
-1
0
1
2
3
THEORECTICAL Z SCORES

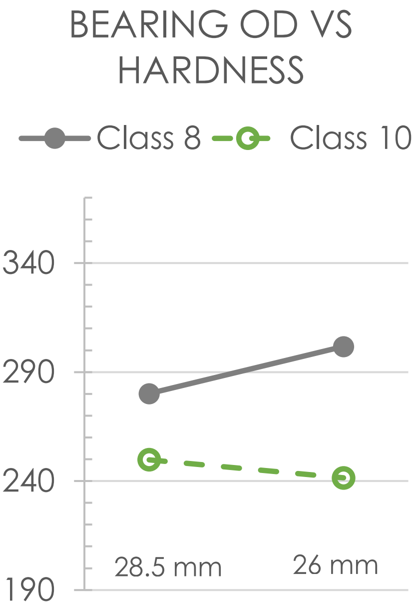

FRICTION TORQUE HISTOGRAM
10
8
6
4
2
0
Right
Skewed
[192, 207]
(207, 222]
(222, 237]
(237, 252]
(252, 267]
(267, 282]
(282, 297]
(297, 312]
(312, 327]
(327, 342]
(342, 357]
(357, 372]
(372, 387]

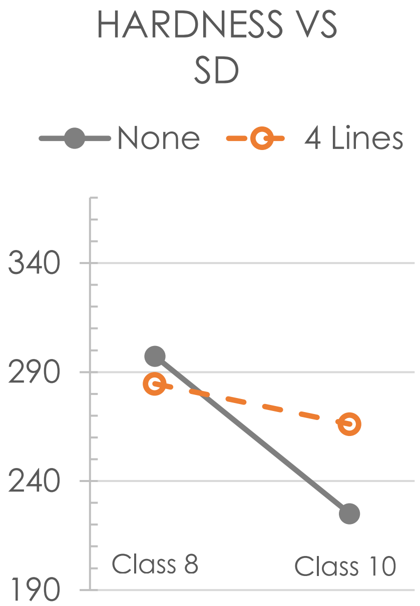

MEAN EFFECTS PLOT
FRICTION TORQUE (Nm)
370
350
330
310
290
270
250
230
210
26 mm
28.5 mm
Class 8
Class 10
4 Lines
None
None
3 Threads
BEARING SURFACE OD
HARDNESS
SURFACE DISCONTINUITIES
THREAD CUTOUT

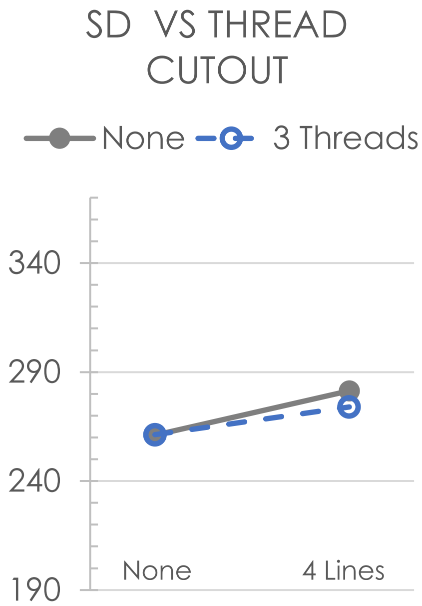
BOX PLOTS

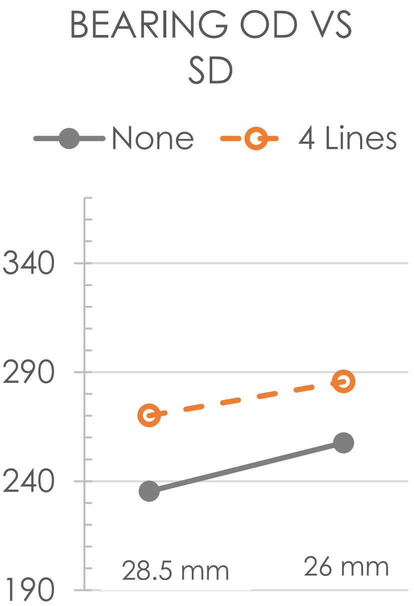

REDUCED BEARING SURFACE OD
28.5 mm
26 mm
HARDNESS
Class 8
Class 10
SURFACE DISCONTINUITIES
None
4 Lines
THREAD CUTOUT
None
3 Threads Removed
400
350
300
250
200
150
100
50
0

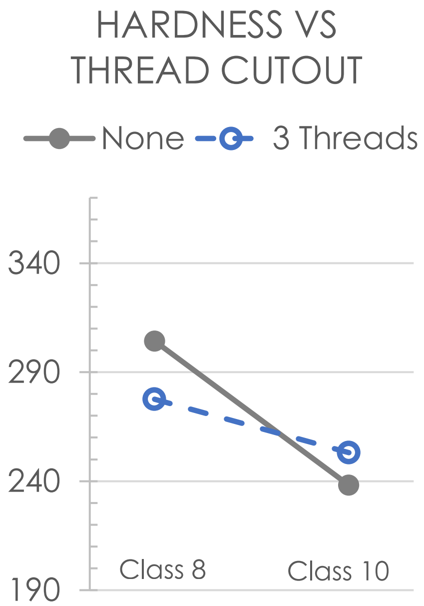
INTERACTION EFFECT PLOTS

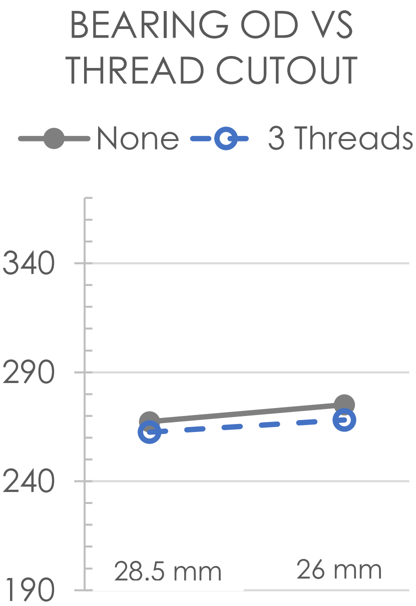












The Bearing Surface OD does not appear to significantly change the bearing surface friction torque, only increasing the torque by a minor amount. The hardness however appears to have the biggest effect, resulting in a reduction in friction for increased hardness, as per the literature. The surface discontinuities plot shows a small increase in friction torque, and the thread cut-out showed an insignificant reduction.

The box and whisker plots of the mean effects display any changes in friction consistency. The Bearing Surface OD box plot shows a significant increase in spread and thus a reduction in friction consistency. This is confirmed by an undesirable 23% increase in the standard deviation for the reduced bearing surface. The hardness box plot shows a decrease in spread however there appears to be a significant data point not contained within the box and whiskers. The surface discontinuities box plot shows a slight decrease in spread, interestingly this appears to have come from a lack of low friction results. The Thread cut-out box plot shows a decrease in quantile size but an increase in whisker size. The standard deviation however remained the same.

4.4 Interaction Effects

The interaction effects plots ideally show any specific effects that occur between two variables. Parallel lines show no interaction. This leaves the three graphs; Bearing OD vs Hardness, Hardness vs Surface Discontinuities, and Hardness vs Thread Cut-out as the only interactions which demonstrate any significant effects. Therefore, not only is Hardness the most significant mean effect but it is also involved in all three of the interaction effects.

The Bearing Surface OD vs Hardness plot is quite interesting as it illustrates that reducing the OD has two different effects depending on hardness. The 'soft' class 8 nuts increase in friction torque when their bearing surface is reduced; conversely the harder class 10 nuts slightly reduce in friction torque. The reduction in torque can be explained by the reduced radius in the absence of adhesive wear. Whereas the increase in torque can be due to the increased susceptibility to adhesive wear, and the resulting drastic increase in friction.

The Hardness vs Surface Discontinuities plot shows that the friction reducing effects of increasing hardness are decreased by the presence of surface discontinuities. This was an unexpected result and should be investigated further.

The Hardness vs Thread Cut-out plot showed that the thread cut-outs slightly reduced the friction torque for the 'soft' class 8 nuts but had less of an effect on the harder class 10 nuts. The evening out of the stress distribution should have helped reduce the peak stress and thus the initiation of galling, which explains the result for class 8 nuts. The class 10 nuts are most likely less effected by the peak stress due to their increased strength and thus less effected by the thread Cut-out. However, it should be noted that the nut wear patterns cause this theory to be questioned as the 'peak stress' location was not the location of wear for any of the samples.

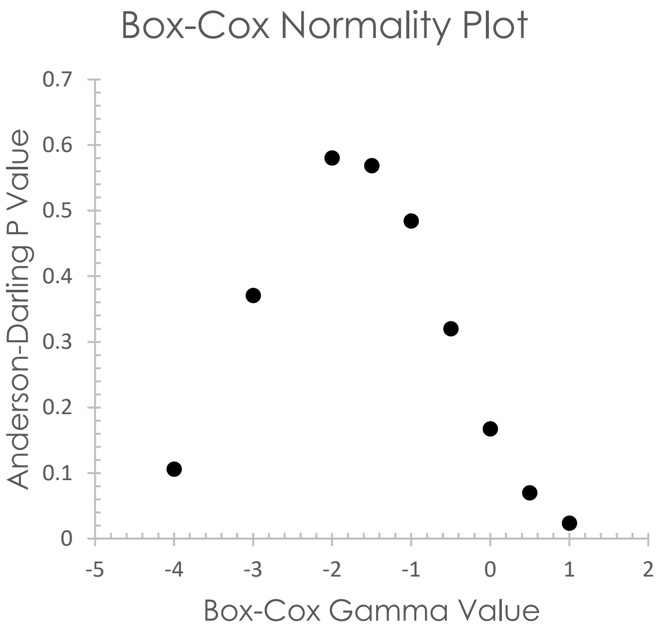
4.5 Box – Cox Transformation

As mentioned previously, the normal probability distribution shows that the data contains significant non-normality and the histogram shows some skewness. An analysis of variance and the construction of a theoretical model requires that the data sufficiently represents a normal distribution (Box-Cox Normality Plot, 2018). To test the response variable's normality, an Anderson-Darling Normality Test is conducted. The resulting P-Value is 0.0231, which is less than 0.05 and thus the null hypothesis that the data is normal is rejected.

To obtain a normal distribution, the response variable data must be transformed. This is done using the Box-Cox transformation;

$$T(Y) = \frac{(Y^{\gamma} - 1)}{\gamma}$$

The data is transformed for various γ values ranging between -5 and 5, and each is tested for normality. The resulting P-Values are plotted in a Box – Cox Normality Plot to determine the optimum value for gamma. Referring to the normality plot, (-1.5) or (-2) values of gamma appear to result in the highest normality. The data was transformed using (-1.5). The resulting P-Value was 0.569, which is greater than 0.05, and thus the null hypothesis for normality could be accepted.



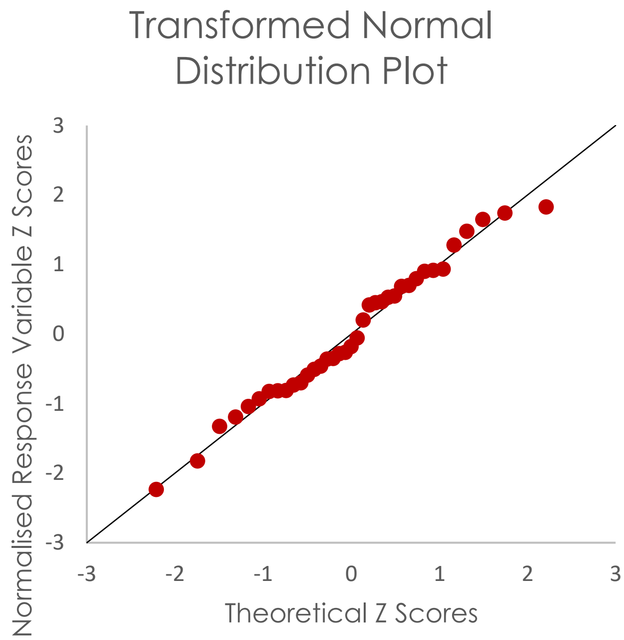
4.6 Analysis of Variance

An analysis of variance, also referred to as an ANOVA, was conducted using a free trial of IBM SPSS. Conducting an ANOVA allows the statistical significance of each mean and interaction effect to be quantified. It also conducts the multilinear regression required to develop a theoretical model. There are four main types of ANOVA, one-way, two-way, MANOVA, and Factorial ANOVA. Since this test has more than two independent variables and only one dependent variable, a Factorial ANOVA is conducted.

The Factorial ANOVA contains multiple assumptions (Statistics Solutions, 2018);

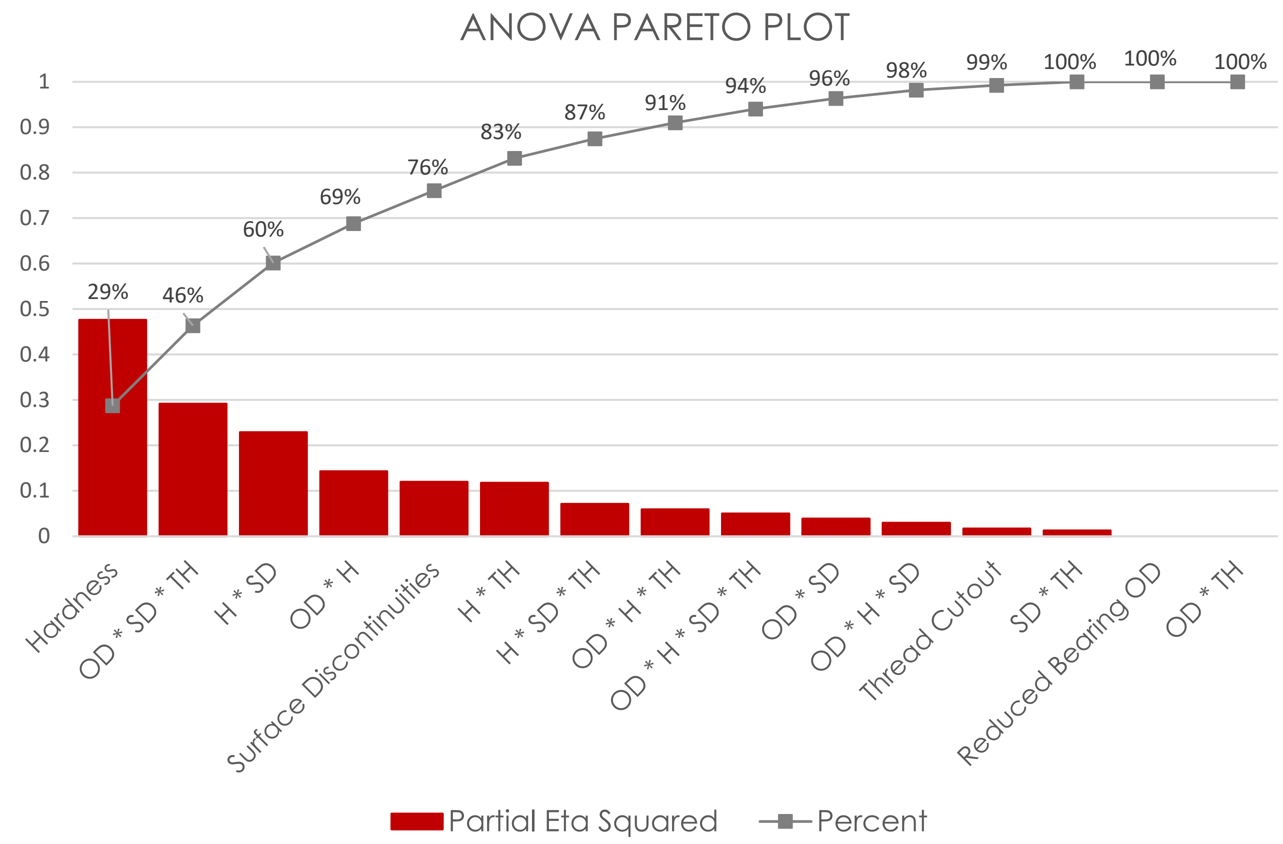
- Continuous Dependent Variable,
- Normality
- Homoscedasticity
- No Multicollinearity

This testing measures the reaction torque continuously and has been normalised using a Box-Cox transformation. Multicollinearity will not be a problem since each variable is very independent. Homoscedasticity however, also referred to as homogeneity of variance, assumes that the variance of error is constant between variables.



The opposite is heteroscedasticity, which is the presence of conditional variance, whereby the variance of a random variable is driven by the values of one or other variables.

The normalised data was tested for heteroscedasticity using the SPSS software. The data failed Levene's Test and White's Test, but passed the Breusch-Pagan Test and the F Test. Further transformations are required to reduce the heteroscedasticity of the data and this was found to be outside the scope of the project. Thus, the ANOVA may not be able to build a theoretical model but can still provide insight for investigating the significance of each effect. The ANOVA outputs the Partial Eta Squared for each effect. This is the proportion of variance accounted for by that effect (SPSS Tutorials, 2018).



OD – Bearing Surface OD **H – Hardness** **SD – Surface Discontinuities** **TH – Thread Cutout**

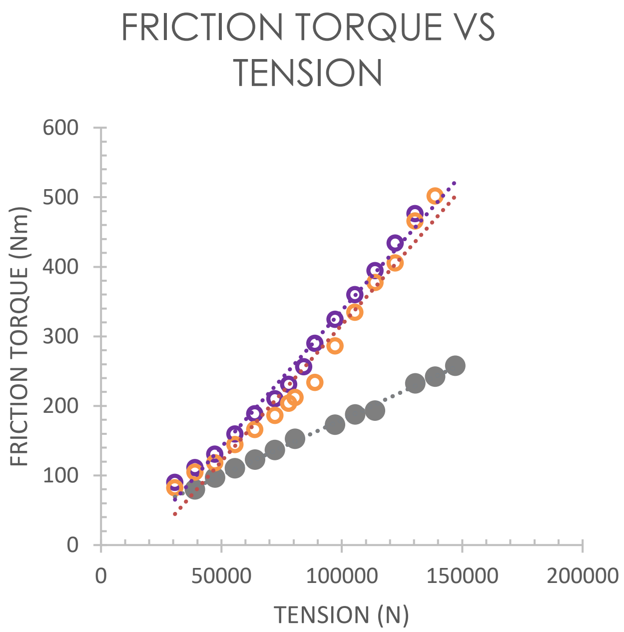
The ANOVA Pareto plot displays the most significant effects. As expected, the Hardness has the largest influence on the bearing surface friction torque. Unexpected however, is that a three-way interaction is the second largest influence – and it doesn't involve hardness. This was not foreseen as only two-way interactions were graphed. The previously discussed two-way effects are also at the significant end of the pareto plot.

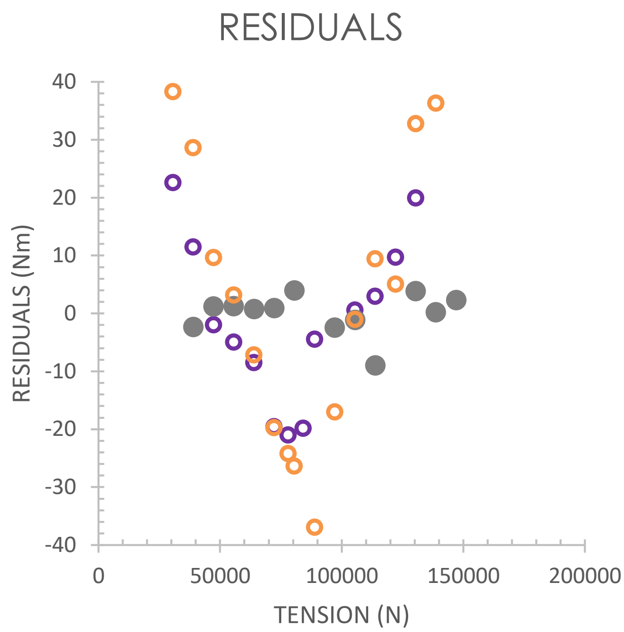
The Thread Cut-out and Bearing Surface OD isolated effects can be safely ignored, whilst the Surface Discontinuities might require further investigation, contributing approximately 7%.

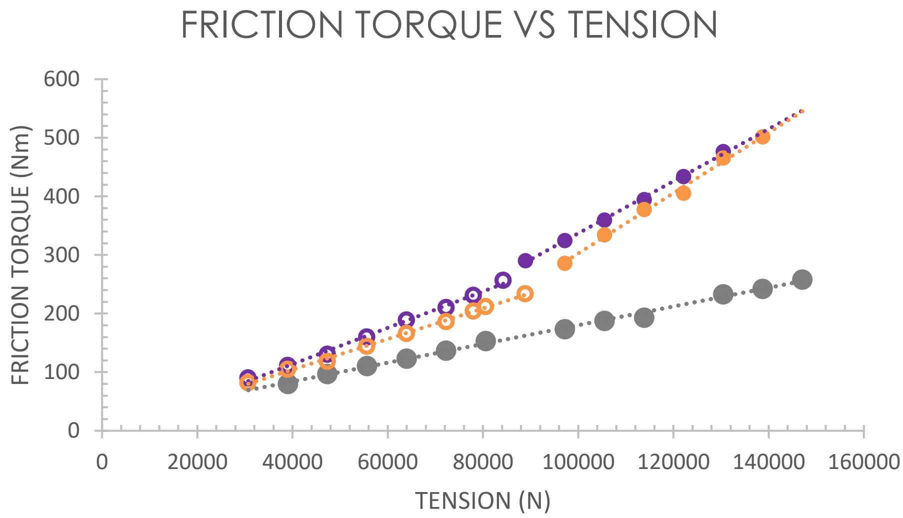
4.7 High vs Low Friction

It should be noted that the factorial analysis is concerned with one specific data point from each test, 75% proof as per ISO 16047. However, it's of interest to investigate the entire tightening process. The following graph shows the torque-tension from the beginning to M20 8.8 proof load (147kN) for the highest and lowest friction torque tests.

Interestingly, the lowest friction test appears to follow a linear relationship while the two highest friction tests both appear to be nonlinear and potentially contain two linear functions with an inflection point around 80-90kN. This inflection point is assumed to be where galling and adhesive wear began to effect the friction. It should be noted that these friction coefficients were already significantly higher before the inflection point. This contradicts the idea that most of friction variation came from the onset of galling and adhesive wear.

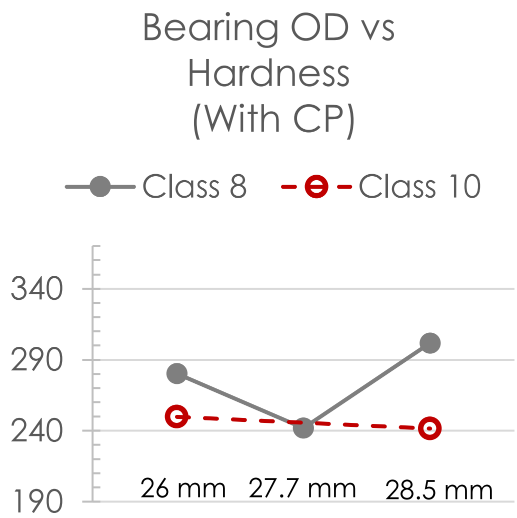






4.8 Centre Points

The centre points were used previously to ensure that no extraneous effects had caused time dependency in the data. However, the centre points can also be used to determine if any nonlinearity exists in the variables. Unfortunately, the centre points only represented a true centre point for the Bearing Surface OD, and thus it is the only factor that can be tested. Referring to the graph on the right, the Bearing Surface OD appears to have some nonlinearity, with the centrepoint having the lowest friction torque.



4.9 Summary

The objectives were to determine if and how each variable affects bearing surface friction torque.

Bearing Surface OD: Reduced consistency of friction torque, negligible change in magnitude, however a potential nonlinear relationship exists.

Hardness: Significant reduction of friction torque as well as an increased consistency, both positive outcomes. It was the most significant factor.

Surface Discontinuities: Slight increase in friction torque and an increase in consistency. It has an interaction effect with hardness and should be considered in future tests.

Thread Cut-out: Negligible change in friction torque and consistency. Found insignificant in the ANOVA.

5.0 Recommendations

5.1 Practical Applications

It's recommended that any nuts manufactured with a reduced bearing surface must not be installed unlubricated. The increased scatter in friction torque will result in undesirable scatter in tension and an increased chance of joint failure.

5.2 Design Insight

The hardness of the bearing surface should be increased where possible, both reducing friction and scatter. The bearing surface OD should not be reduced if the application is likely to result in unlubricated installation. This is significant because it means that potentially other avenues need to be investigated for reducing the Segnut's size.

The surface discontinuities slightly increase the friction at the bearing surface and should be minimised where possible. The even load distribution on the bearing surface provided by the thread cut-out did not affect the friction torque. Any pursuits for even thread load distribution should not be conducted with bearing surface friction reduction goals in mind.

5.3 Field of Nut Tribology

The wear pattern on the nuts differed significantly from the theoretical stress distribution. This could imply that the theoretical stress distribution is incorrect, or that the distribution of stress is an insignificant factor in bearing surface wear.

The plots of friction torque vs tension show the onset of galling and the rapid increase in friction; however, they also show that the plots start with a high friction. This begs the question whether galling is occurring between the asperities from the very beginning and the friction increases sharply when the wear becomes bulk deformation. Or whether it's the high initial friction which causes coating and oxidation layer removal, resulting in metal on metal adhesive contact, and then the resulting increase in friction. Either way, these nonlinear characteristics were not present in the low friction results that didn't show visible signs of galling.

5.4 Future Work

Reduced bearing surface OD tests should be conducted under varying lubricated and coated conditions. Surface discontinuities should also be tested as the minor negative effects could be negated by the reduction in overall friction.

If a specially coated nut with a reduced bearing surface OD resulted in reduced friction and increased consistency, when compared to a conventional nut, that product would much more viable. Also, a coating solution is much safer as applying lubrication can easily be forgotten.

Future tests should also involve varying the properties of the washer, this testing used the same washer throughout.

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Appendix

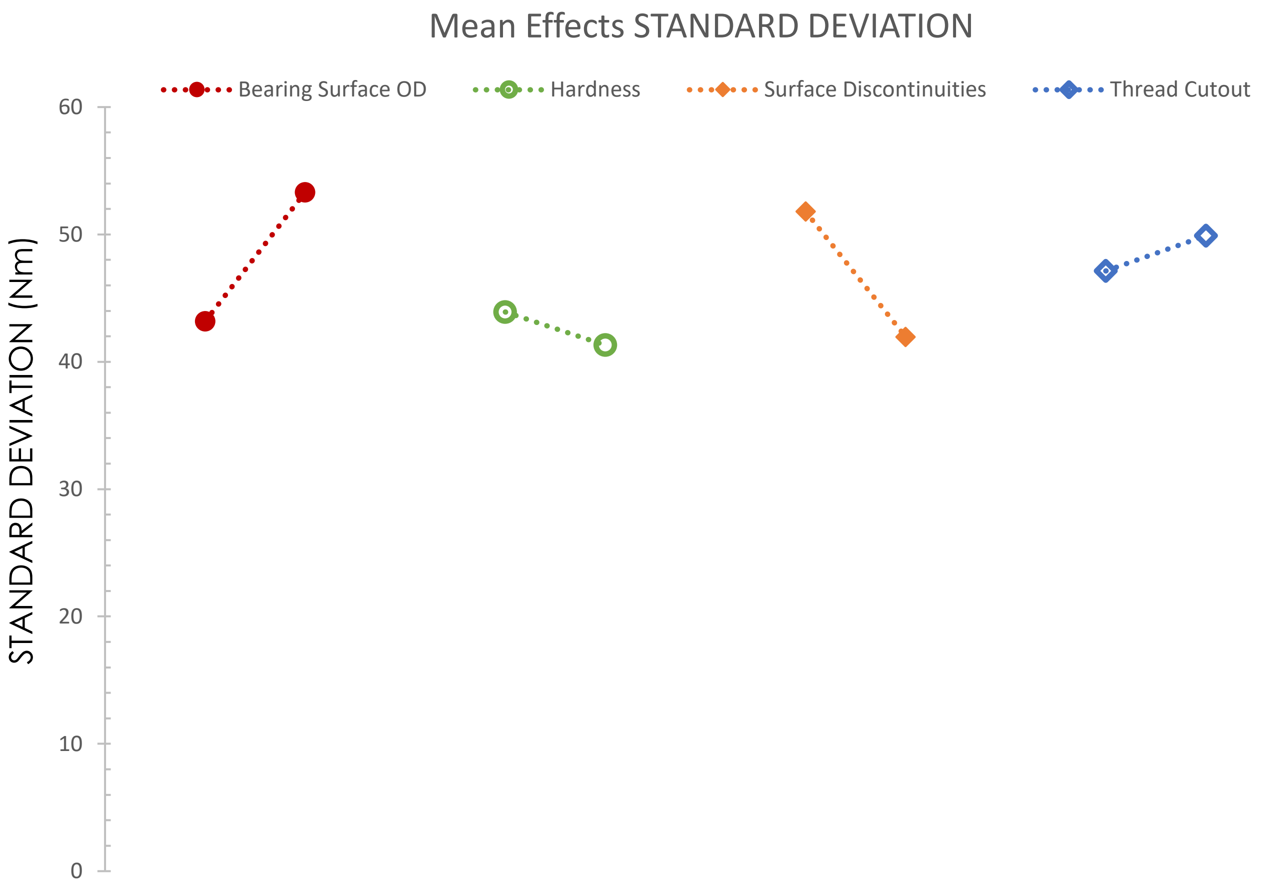
YATES ORDER		Bearing OD	Hardness	Discontinuities	Thread Cutout			RESPONSE VARIABLE		
Run Order	Yates Order	A	B	C	D		Random No.	Friction Torque	Bearing OD	µ
24	1	-1	-1	-1	-1	Standard	0.68470469	326.607636	28.5	0.2375328
12	2	1	-1	-1	-1	A	0.18358824	289.1746576	26	0.2213777
34	3	-1	1	-1	-1	B	0.92531387	212.7408274	28.5	0.1547206
23	4	1	1	-1	-1	AB	0.66304152	242.3344386	26	0.1855192
11	5	-1	-1	1	-1	C	0.16273059	276.8276543	28.5	0.2013292
31	6	1	-1	1	-1	AC	0.86528833	301.9136293	26	0.2311301
4	7	-1	1	1	-1	BC	0.03759748	274.5738362	28.5	0.1996901
25	8	1	1	1	-1	ABC	0.70032712	223.5199573	26	0.1711158
6	9	-1	-1	-1	1	D	0.10131468	253.4075447	28.5	0.1842964
3	10	1	-1	-1	1	AD	0.0288127	302.9915423	26	0.2319552
33	11	-1	1	-1	1	BD	0.89874499	244.8822329	28.5	0.1780962
5	12	1	1	-1	1	ABD	0.10086407	192.0644964	26	0.147035
7	13	-1	-1	1	1	CD	0.12684785	241.9424702	28.5	0.1759582
21	14	1	-1	1	1	ACD	0.61245103	276.0437175	26	0.2113253
30	15	-1	1	1	1	BCD	0.86443543	230.5753878	28.5	0.1676912
36	16	1	1	1	1	ABCD	0.94447341	236.9448736	26	0.1813932
15	17	-1	-1	-1	-1	Standard	0.43788284	238.5127471	28.5	0.1734638
14	18	1	-1	-1	-1	A	0.41207827	342.5803466	26	0.2622625
9	19	-1	1	-1	-1	B	0.15972037	220.3842104	28.5	0.1602794
17	20	1	1	-1	-1	AB	0.44716447	216.1705505	26	0.1654894
26	21	-1	-1	1	-1	C	0.71489612	300.8357163	28.5	0.2187896
35	22	1	-1	1	-1	AC	0.93110406	357.9651048	26	0.2740403
18	23	-1	1	1	-1	BC	0.51936234	288.3907209	28.5	0.2097387
20	24	1	1	1	-1	ABC	0.58663138	227.1456646	26	0.1738914
13	25	-1	-1	-1	1	D	0.2283461	376.0936414	28.5	0.2735226
32	26	1	-1	-1	1	AD	0.89560576	248.7039244	26	0.1903953
27	27	-1	1	-1	1	BD	0.74319758	245.4701855	28.5	0.1785238
2	28	1	1	-1	1	ABD	0.02608175	226.7536963	26	0.1735913
29	29	-1	-1	1	1	CD	0.80411866	226.9496804	28.5	0.1650543
16	30	1	-1	1	1	ACD	0.44200534	294.5642225	26	0.2255037
8	31	-1	1	1	1	BCD	0.15612492	280.8453299	28.5	0.2042511
22	32	1	1	1	1	ABCD	0.61395479	366.9803771	26	0.2809419
1		0	-1	-1	-1			264.3826588	27.7	0.1954048
10		0	-1	-1	-1			280.0613932	27.7	0.2069929
19		0	-1	-1	-1			229.5954669	27.7	0.1696936
28		0	-1	-1	-1			234.2010951	27.7	0.1730976
37		0	-1	-1	-1			200.7857924	27.7	0.1484004

Response Variable	
Mean	264.700471
Standard Erroı	7.644656582
Median	248.7039244
Standard Devi	46.50063062
Sample Variar	2162.308648
Kurtosis	0.058819066
Skewness	0.812840947
Range	184.029145
Minimum	192.0644964
Maximum	376.0936414
Sum	9793.917427
Count	37
Confidence Le\	15.50408215

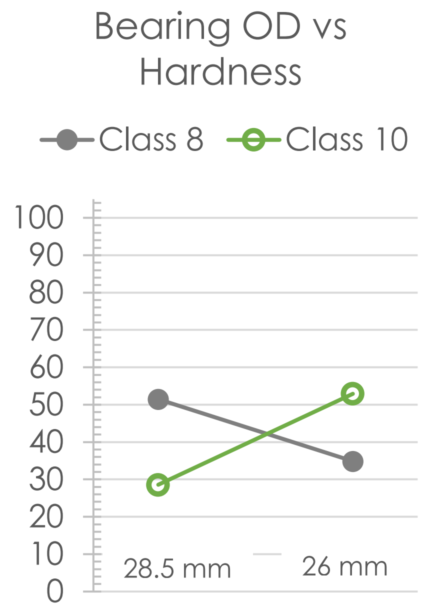
MEAN EFFECTS	AVERAGE	SD
Bearing OD Negative	264.94	43.1736
Bearing OD Positive	271.616	53.3129
Hardness Negative	290.945	43.909
Hardness Positive	245.611	41.3028
Surface Discontinuities Negative	261.18	51.7944
Surface Discontinuities Postive	286.031	41.9618
Thread Cutout Negative	271.23	47.1313
Thread Cutout Posistive	265.326	49.9003
Centre Points Average	241.805	
Centre Points SD	31.0756	

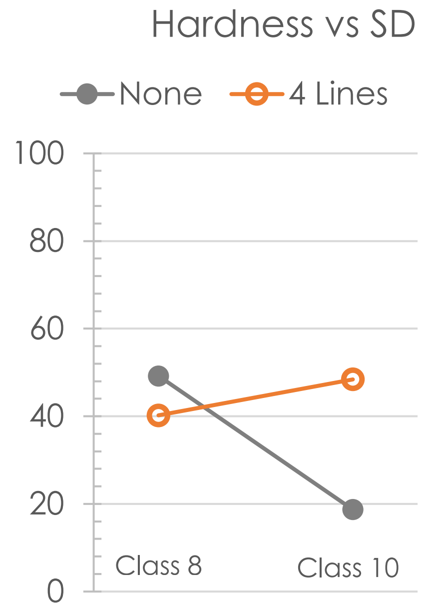
INTERACTION EFFECTS

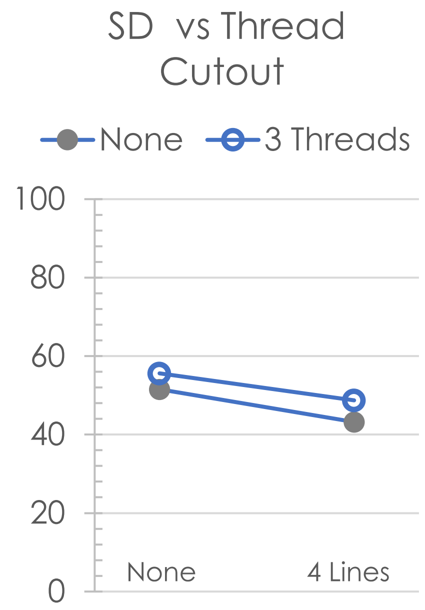
Negative	Positive	AVERAGE	SD	Negative	Positive	AVERAGE	SD	Negative	Positive	AVERAGE	SD
OD,H		280.147	51.4661	H,SD		297.259	49.2228	SD,TH		261.063	51.5417
OD	H	249.733	28.5806	H	SD	284.63	40.2171	SD	TH	261.296	55.6056
H	OD	301.742	34.8738	SD	H	225.1	18.7163	TH	SD	281.397	43.2117
	OD, H	241.489	52.9137		H,SD	266.122	48.4111		H,SD	273.986	48.7278
OD,SD		235.344	102.951	H,TH		304.302	38.1203				
OD	SD	270.052	26.1876	H	TH	277.587	47.6492				
SD	OD	257.597	50.0771	TH	H	238.158	28.4048				
	OD,SD	285.635	55.9789		H,TH	253.065	52.1702				
OD,TH		267.359	40.066								
OD	TH	262.521	48.7395								
TH	OD	275.101	55.8617								
	OD,TH	268.131	54.2428								

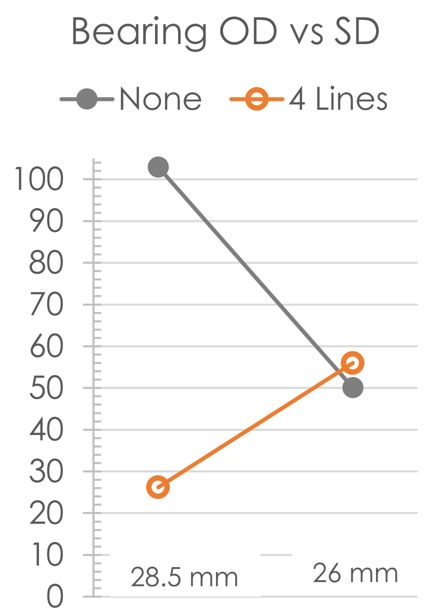


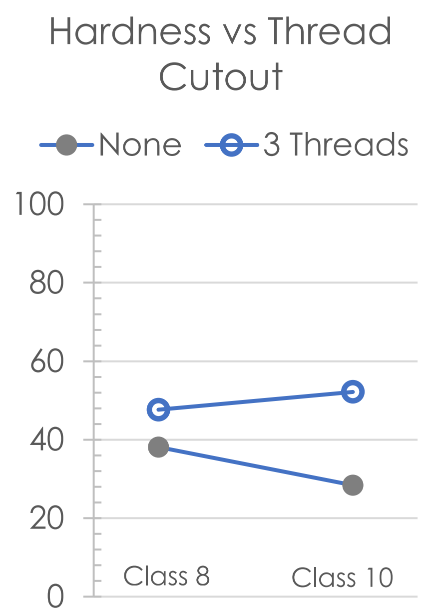
Interaction effects **STANDARD DEVIATIONS**

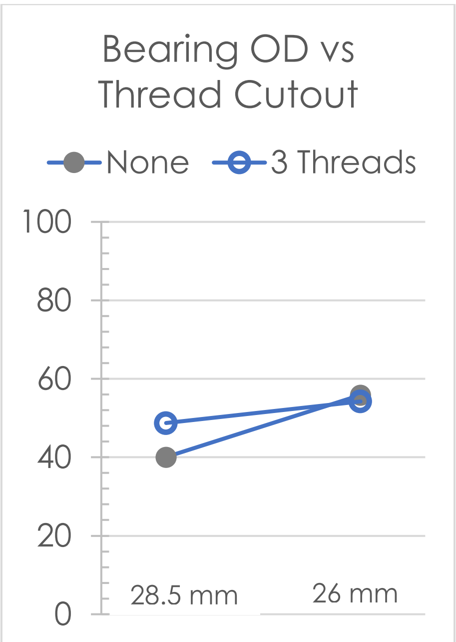




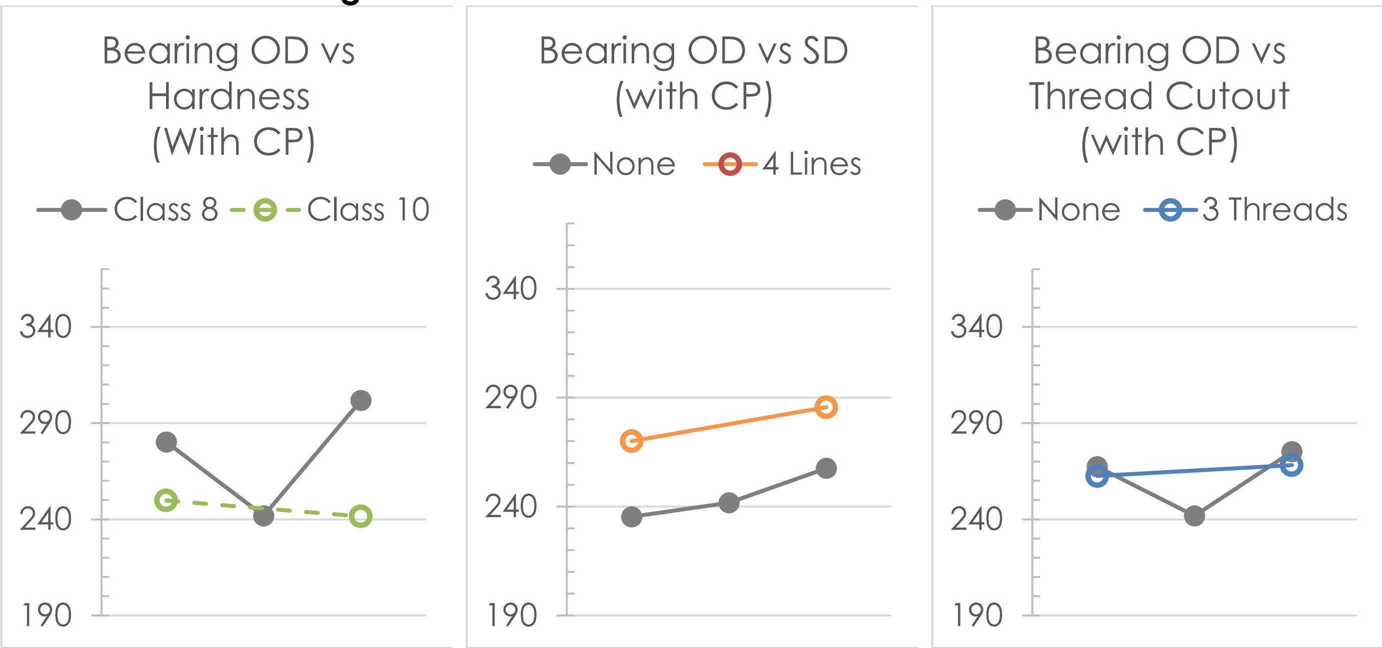








Bearing Surface OD Interactions Effects with Centre Points



Levene's Test of Equality of Error Variances[a,b]

		Levene Statistic	df1	df2	Sig.
NormalisedResponse	Based on Mean	9.072E+22	15	16	.000
	Based on Median	9.072E+22	15	16	.000
	Based on Median and with adjusted df	9.072E+22	15	10.000	.000
	Based on trimmed mean	3.024E+22	15	16	.000

Tests the null hypothesis that the error variance of the dependent variable is equal across groups.

a. Dependent variable: NormalisedResponse

b. Design: Intercept + ReducedBearingOD + Hardness + SurfaceDiscontinuities + ThreadCutout + ReducedBearingOD * Hardness + ReducedBearingOD * SurfaceDiscontinuities + ReducedBearingOD * ThreadCutout + Hardness * SurfaceDiscontinuities + Hardness * ThreadCutout + SurfaceDiscontinuities * ThreadCutout + ReducedBearingOD * Hardness * SurfaceDiscontinuities + ReducedBearingOD * Hardness * ThreadCutout + ReducedBearingOD * SurfaceDiscontinuities * ThreadCutout + Hardness * SurfaceDiscontinuities * ThreadCutout + ReducedBearingOD * Hardness * SurfaceDiscontinuities * ThreadCutout

Tests for Heteroskedasticity

White Test for Heteroskedasticity[a,b,c]

Chi-Square	df	Sig.
32.000	15	.006

a. Dependent variable: NormalisedResponse

b. Tests the null hypothesis that the variance of the errors does not depend on the values of the independent variables.

c. Design: Intercept + ReducedBearingOD + Hardness + SurfaceDiscontinuities + ThreadCutout + ReducedBearingOD * Hardness + ReducedBearingOD * SurfaceDiscontinuities + ReducedBearingOD * ThreadCutout + Hardness * SurfaceDiscontinuities + Hardness * ThreadCutout + SurfaceDiscontinuities * ThreadCutout + ReducedBearingOD * Hardness * SurfaceDiscontinuities + ReducedBearingOD * Hardness * ThreadCutout + ReducedBearingOD * SurfaceDiscontinuities * ThreadCutout + Hardness * SurfaceDiscontinuities * ThreadCutout + ReducedBearingOD * Hardness * SurfaceDiscontinuities * ThreadCutout

Modified Breusch-Pagan Test for Heteroskedasticity[a,b,c]

Chi-Square	df	Sig.
1.499	1	.221

a. Dependent variable: NormalisedResponse

b. Tests the null hypothesis that the variance of the errors does not depend on the values of the independent variables.

c. Predicted values from design: Intercept + ReducedBearingOD + Hardness + SurfaceDiscontinuities + ThreadCutout + ReducedBearingOD * Hardness + ReducedBearingOD * SurfaceDiscontinuities + ReducedBearingOD * ThreadCutout + Hardness * SurfaceDiscontinuities + Hardness * ThreadCutout + SurfaceDiscontinuities * ThreadCutout + ReducedBearingOD * Hardness * SurfaceDiscontinuities + ReducedBearingOD * Hardness * ThreadCutout + ReducedBearingOD * SurfaceDiscontinuities * ThreadCutout + Hardness * SurfaceDiscontinuities * ThreadCutout + ReducedBearingOD * Hardness * SurfaceDiscontinuities * ThreadCutout

Breusch-Pagan Test for Heteroskedasticity[a,b,c]

Chi-Square	df	Sig.
1.138	1	.286

a. Dependent variable: NormalisedResponse

b. Tests the null hypothesis that the variance of the errors does not depend on the values of the independent variables.

c. Predicted values from design: Intercept + ReducedBearingOD + Hardness + SurfaceDiscontinuities + ThreadCutout + ReducedBearingOD * Hardness + ReducedBearingOD * SurfaceDiscontinuities + ReducedBearingOD * ThreadCutout + Hardness * SurfaceDiscontinuities + Hardness * ThreadCutout + SurfaceDiscontinuities * ThreadCutout + ReducedBearingOD * Hardness * SurfaceDiscontinuities + ReducedBearingOD * Hardness * ThreadCutout + ReducedBearingOD * SurfaceDiscontinuities * ThreadCutout + Hardness * SurfaceDiscontinuities * ThreadCutout + ReducedBearingOD * Hardness * SurfaceDiscontinuities * ThreadCutout

F Test for Heteroskedasticity[a,b,c]

F	df1	df2	Sig.
1.475	1	30	.234

a. Dependent variable: NormalisedResponse

b. Tests the null hypothesis that the variance of the errors does not depend on the values of the independent variables.

c. Predicted values from design: Intercept + ReducedBearingOD + Hardness + SurfaceDiscontinuities + ThreadCutout + ReducedBearingOD * Hardness + ReducedBearingOD * SurfaceDiscontinuities + ReducedBearingOD * ThreadCutout + Hardness * SurfaceDiscontinuities + Hardness * ThreadCutout + SurfaceDiscontinuities * ThreadCutout + ReducedBearingOD * Hardness * SurfaceDiscontinuities + ReducedBearingOD * Hardness * ThreadCutout + ReducedBearingOD * SurfaceDiscontinuities * ThreadCutout + Hardness * SurfaceDiscontinuities * ThreadCutout + ReducedBearingOD * Hardness * SurfaceDiscontinuities * ThreadCutout

Tests of Between-Subjects Effects

Dependent Variable: NormalisedResponse

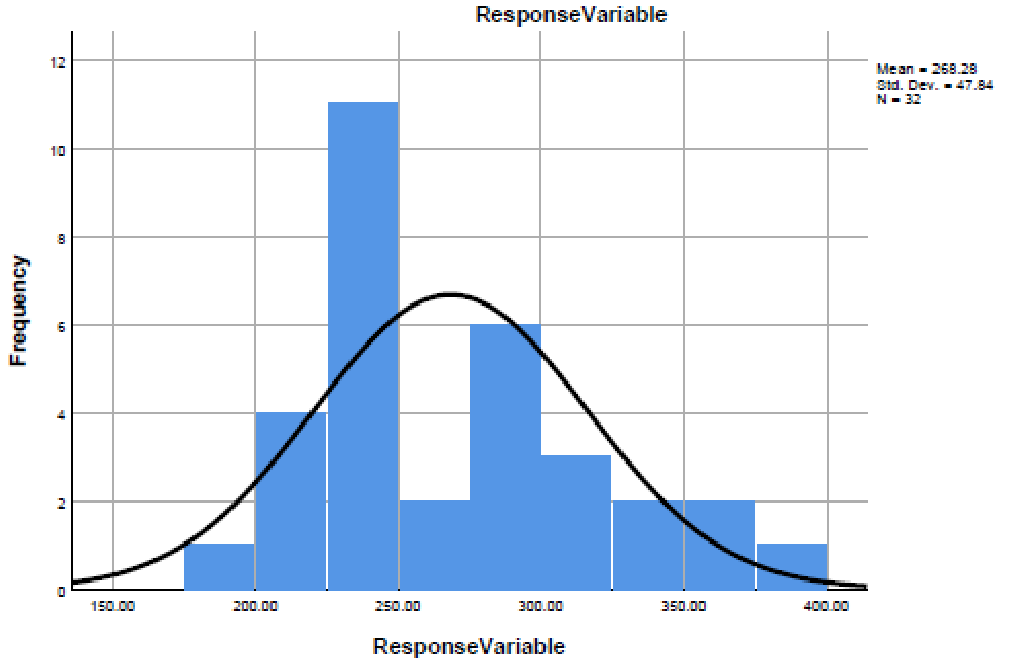
Source	Type III Sum of Squares	df	Mean Square	F	Sig.
Corrected Model	3.340E-8[a]	15	2.227E-9	2.455	.042
Intercept	14.215	1	14.215	1.567E+10	.000
ReducedBearingOD	.000	1	.000	.000	1.000
Hardness	1.319E-8	1	1.319E-8	14.537	.002
SurfaceDiscontinuities	1.982E-9	1	1.982E-9	2.185	.159
ThreadCutout	2.557E-10	1	2.557E-10	.282	.603
ReducedBearingOD * Hardness	2.429E-9	1	2.429E-9	2.678	.121
ReducedBearingOD * SurfaceDiscontinuities	5.909E-10	1	5.909E-10	.651	.431
ReducedBearingOD * ThreadCutout	.000	1	.000	.000	1.000
Hardness * SurfaceDiscontinuities	4.307E-9	1	4.307E-9	4.748	.045
Hardness * ThreadCutout	1.938E-9	1	1.938E-9	2.137	.163
SurfaceDiscontinuities * ThreadCutout	1.891E-10	1	1.891E-10	.209	.654
ReducedBearingOD * Hardness * SurfaceDiscontinuities	4.447E-10	1	4.447E-10	.490	.494
ReducedBearingOD * Hardness * ThreadCutout	2.025E-10	1	2.025E-10	.223	.643
ReducedBearingOD * SurfaceDiscontinuities * ThreadCutout	5.979E-9	1	5.979E-9	6.592	.021
Hardness * SurfaceDiscontinuities * ThreadCutout	1.105E-9	1	1.105E-9	1.219	.286
ReducedBearingOD * Hardness * SurfaceDiscontinuities * ThreadCutout	7.576E-10	1	7.576E-10	.835	.374
Error	1.451E-8	16	9.071E-10		
Total	14.215	32			
Corrected Total	4.792E-8	31			

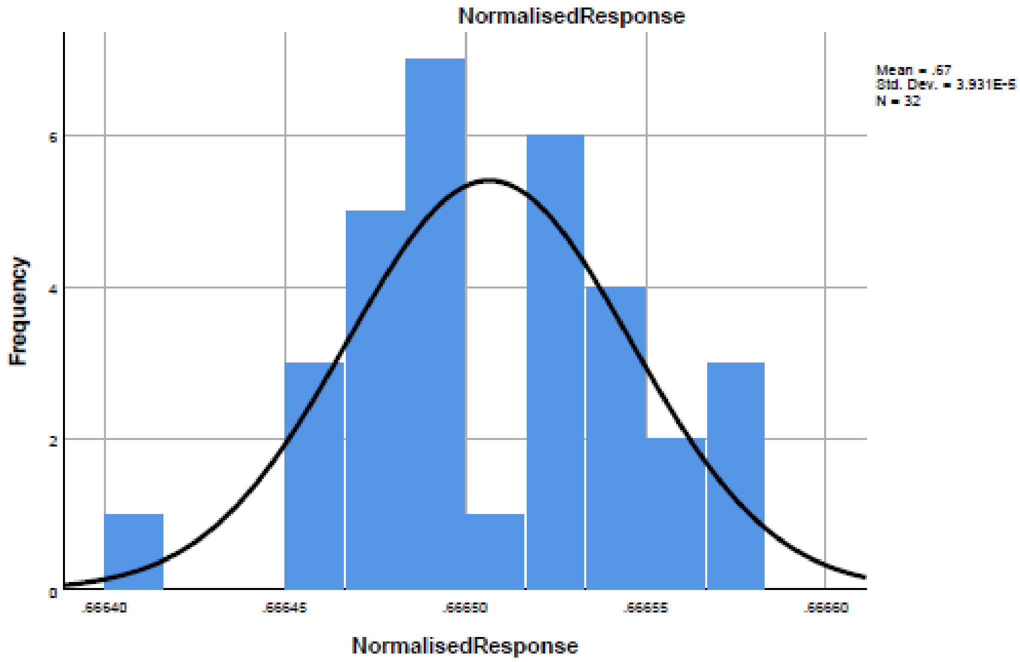
Tests of Between-Subjects Effects

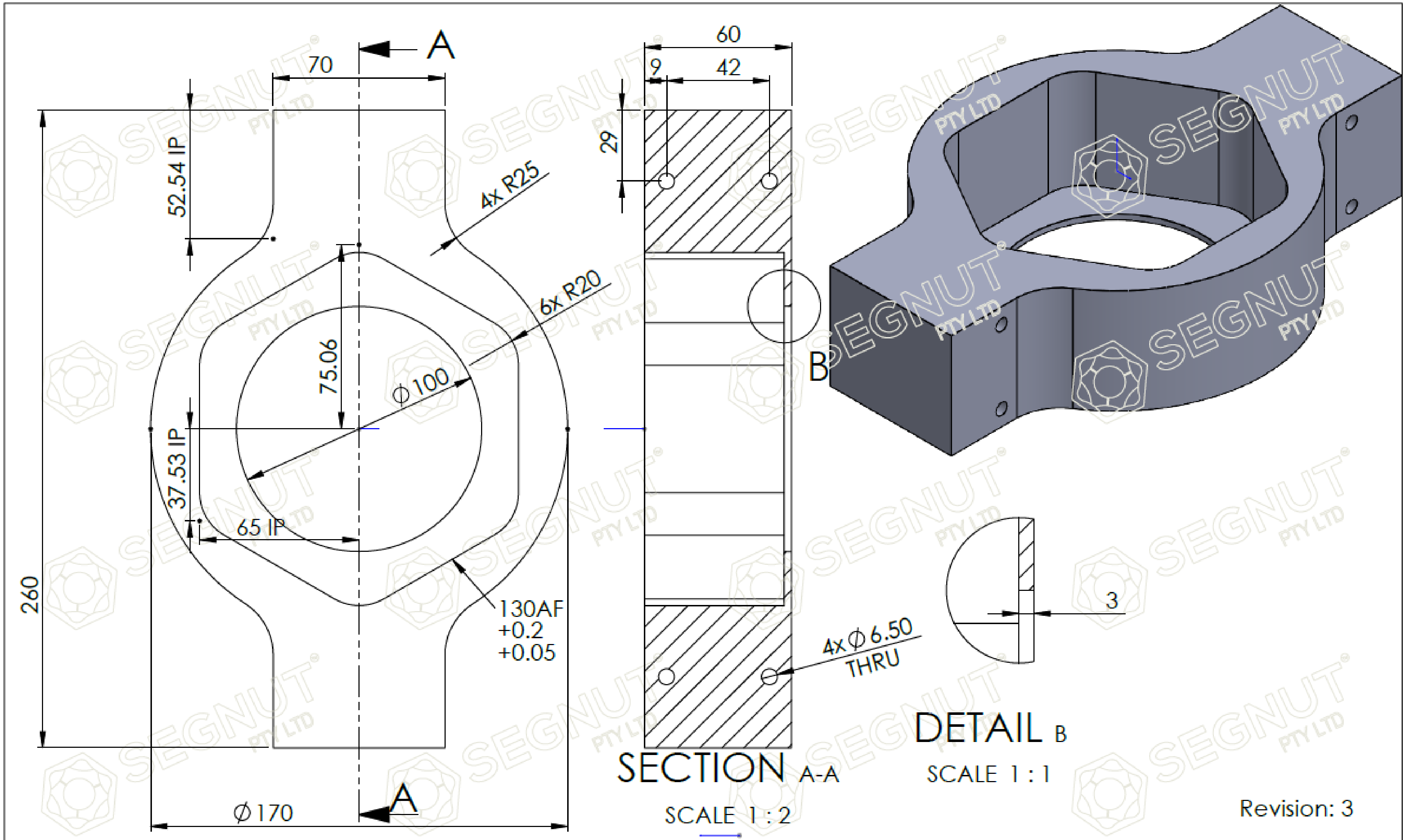
Dependent Variable: NormalisedResponse

Source	Partial Eta Squared
Corrected Model	.697
Intercept	1.000
ReducedBearingOD	.000
Hardness	.476
SurfaceDiscontinuities	.120
ThreadCutout	.017
ReducedBearingOD * Hardness	.143
ReducedBearingOD * SurfaceDiscontinuities	.039
ReducedBearingOD * ThreadCutout	.000
Hardness * SurfaceDiscontinuities	.229
Hardness * ThreadCutout	.118
SurfaceDiscontinuities * ThreadCutout	.013
ReducedBearingOD * Hardness * SurfaceDiscontinuities	.030
ReducedBearingOD * Hardness * ThreadCutout	.014
ReducedBearingOD * SurfaceDiscontinuities * ThreadCutout	.292
Hardness * SurfaceDiscontinuities * ThreadCutout	.071
ReducedBearingOD * Hardness * SurfaceDiscontinuities * ThreadCutout	.050
Error	
Total	
Corrected Total	

a. R Squared = .697 (Adjusted R Squared = .413)


ResponseVariable
Mean = 268.28
Std. Dev. = 47.84
N = 32
Frequency
0
2
4
6
8
10
12
150.00
200.00
250.00
300.00
350.00
400.00
ResponseVariable


NormalisedResponse
Mean = .67
Std. Dev. = 3.931E-5
N = 32
Frequency
0
2
4
6
.66640
.66645
.66650
.66655
.66660
NormalisedResponse



Revision: 3

NOTES:

STOCK SIZE	260x170
MATERIAL	Plain Carbon Steel
Heat Treatment	XXX
HRc	
FINISH	XXX

UNLESS OTHERWISE SPECIFIED:
Dimensions are in mm
Break all sharp edges 0.3 x 45° or R0.3
TOLERANCES:
ANGULAR: ± 0.5°
0= ±0.5
0.0 = ± 0.15
0.00 = ± 0.05
SURFACE FINISH

DRAWN	AKP	DATE Monday, April 16, 2018 2:53:55 PM
CHECKED	SWT	14/09/2017
Part Nº		

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DRW Nº: Rev3 Torque Reaction Outer

Description:

DO NOT SCALE DRAWING | Rev A | WEIGHT: kg | SHEET 1 OF 1 | A4

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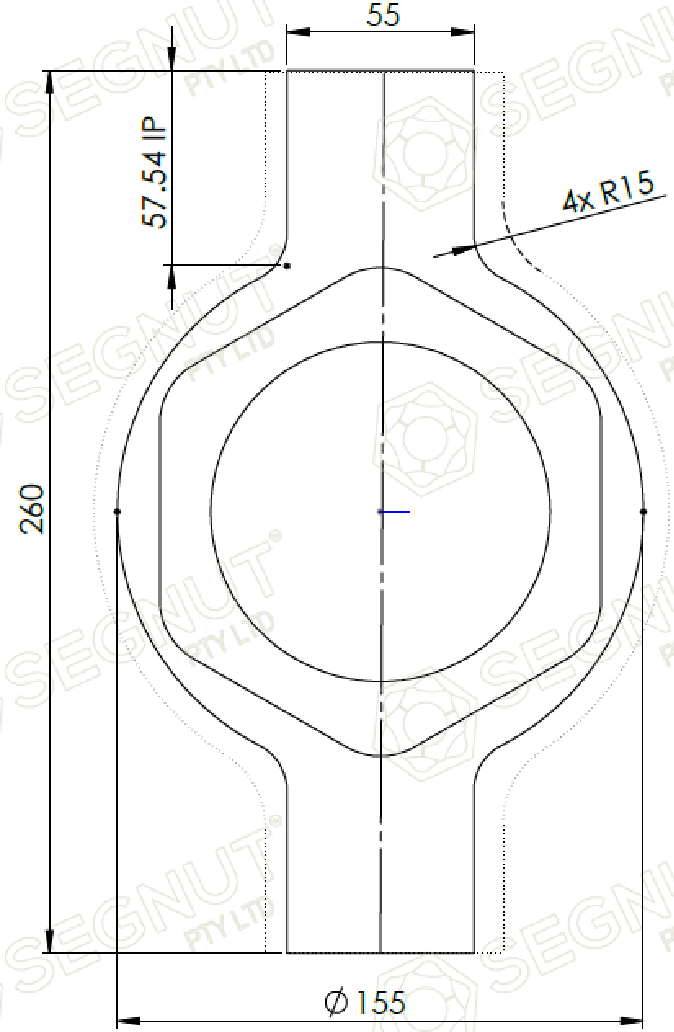

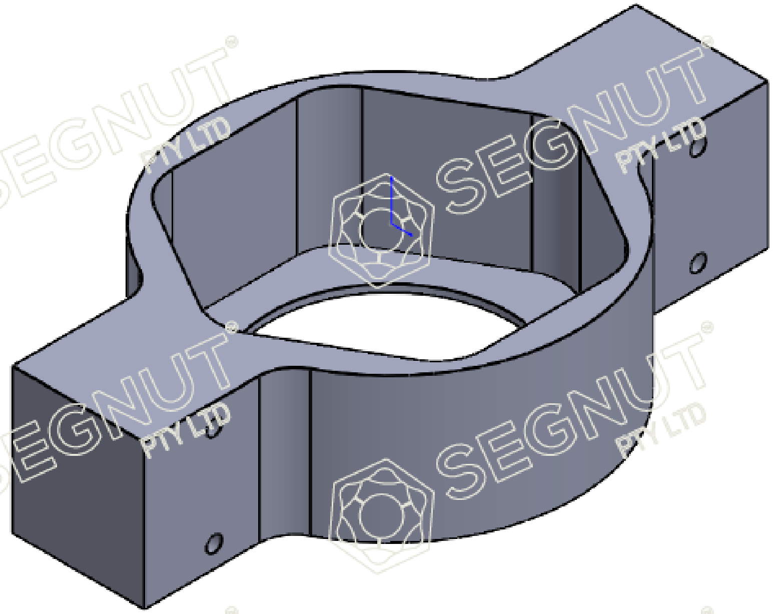

Revision: 4
Outer Profile Material Removal

NOTES:	
STOCK SIZE	260x170
MATERIAL	Plain Carbon Steel
Heat Treatment	XXX
HRc	
FINISH	XXX

UNLESS OTHERWISE SPECIFIED:
Dimensions are in mm
Break all sharp edges
0.3 x 45° or R0.3
TOLERANCES:
ANGULAR: ± 0.5°
0= ±0.5
0.0 = ± 0.15
0.00 = ± 0.05
SURFACE FINISH

DRAWN	AKP	DATE Monday, April 16, 2018 2:53:55 PM
CHECKED	SWT	30/04/2017
Part N°		

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DRW №: Rev4 Torque Reaction Outer
Description: Profile Material Removal
DO NOT SCALE DRAWING | Rev A | WEIGHT: kg | SHEET 1 OF 1 | A4

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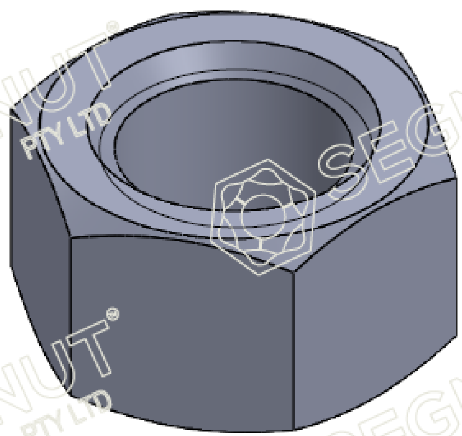
Configuration	Nut	Quantity
1	Class 8	3
	Class 10	3
2	Class 8	3
	Class 10	3
3	Class 8	3
	Class 10	3
4	Class 8	3
	Class 10	3
5	Class 8	3
	Class 10	3
6	Class 8	3
	Class 10	3
7	Class 8	3
	Class 10	3
8	Class 8	6
Total		48

All M20 Nuts are provided



NOTES:

STOCK SIZE	XXX
MATERIAL	XXX
Heat Treatment	XXX
HRc	
FINISH	XXX

UNLESS OTHERWISE SPECIFIED:
Dimensions are in mm
Break all sharp edges
0.3 x 45° or R0.3
TOLERANCES:
ANGULAR: ± 0.5°
0= ±0.5
0.0 = ± 0.15
0.00 = ± 0.05
SURFACE FINISH

DRAWN	AKP	DATE: Tuesday, April 17, 2018 11:00:46 AM
CHECKED	SWT	17/09/2017
Part N°		

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DRW №: M20 Nut for machining
Description:
DO NOT SCALE DRAWING | Rev A | WEIGHT: kg | SHEET 1 OF 9 | A4

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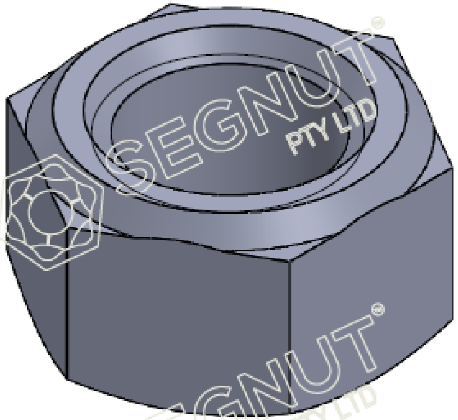
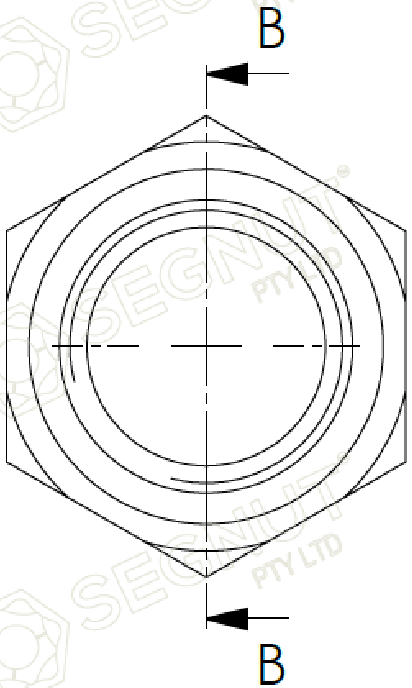
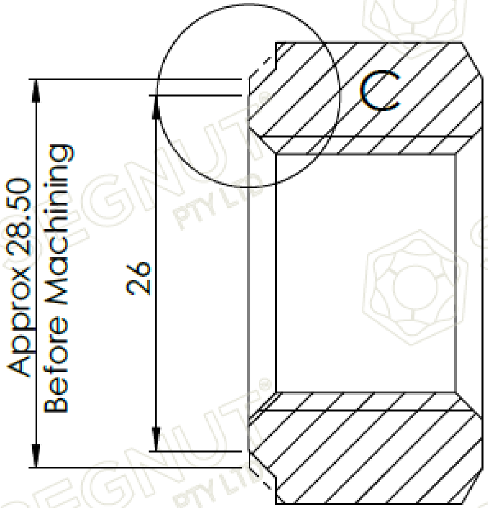
Configuration **1**

Reduced Bearing Surface OD

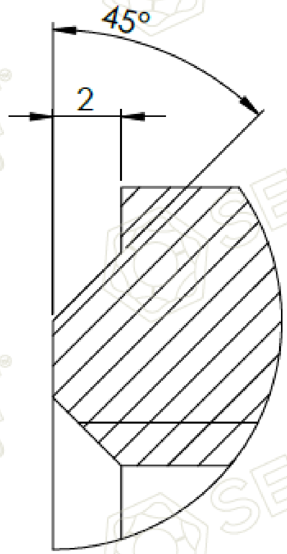
Nut	Quantity
Class 8	x3
Class 10	x3







SECTION B-B



DETAIL C

SCALE 5 : 1

NOTES:

STOCK SIZE XXX	UNLESS OTHERWISE SPECIFIED: Dimensions are in mm Break all sharp edges 0.3 x 45° or R0.3 TOLERANCES: ANGULAR: ± 0.5° 0= ±0.5 0.0 = ± 0.15 0.00 = ± 0.05 SURFACE FINISH	DRAWN AKP	DATE Tuesday, April 17, 2018 11:00:46 AM
MATERIAL XXX		CHECKED SWT	17/09/2017
Heat Treatment XXX		Part N°	
HRc		DRW №:	M20 Nut for machining
FINISH XXX		Description:	

REV | DESCRIPTION | DATE | CHK | APP

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DO NOT SCALE DRAWING | Rev A | WEIGHT: kg | SHEET 2 OF 9 | A4

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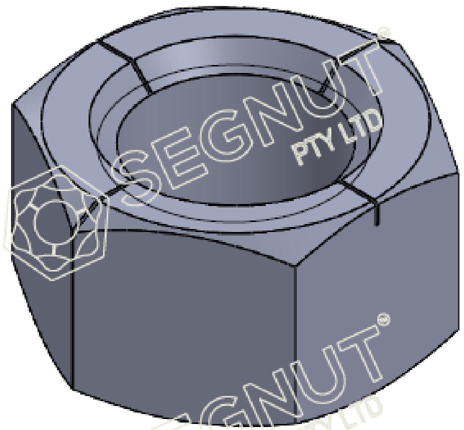
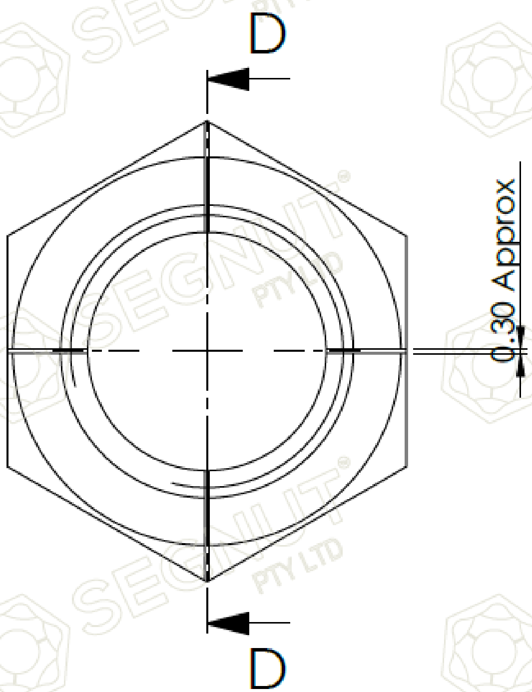
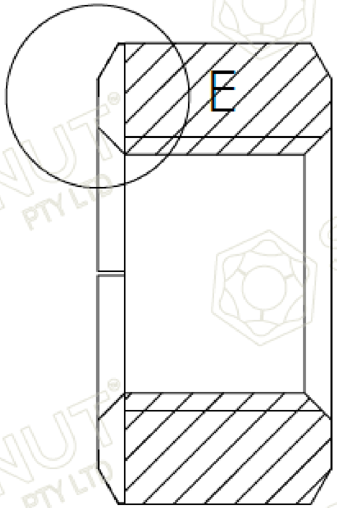


Configuration **2**

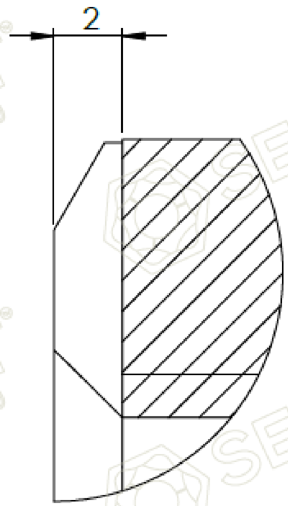
4x Grooves

Nut	Quantity
Class 8	x3
Class 10	x3





SECTION D-D



DETAIL E

SCALE 5 : 1

NOTES:

STOCK SIZE	XXX
MATERIAL	XXX
Heat Treatment	XXX
HRc	
FINISH	XXX

UNLESS OTHERWISE SPECIFIED:
Dimensions are in mm
Break all sharp edges
0.3 x 45° or R0.3
TOLERANCES:
ANGULAR: ± 0.5°
0= ±0.5
0.0 = ± 0.15
0.00 = ± 0.05
SURFACE FINISH

DRAWN	AKP	DATE Tuesday, April 17, 2018 11:00:46 AM
CHECKED	SWT	17/09/2017
Part N°		

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DRW №: M20 Nut for machining

Description:

DO NOT SCALE DRAWING | Rev A | WEIGHT: kg | SHEET 3 OF 9 | A4

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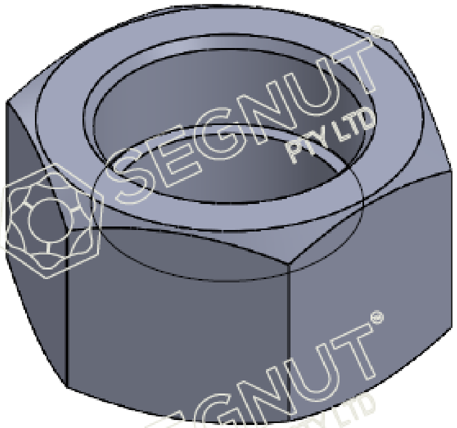
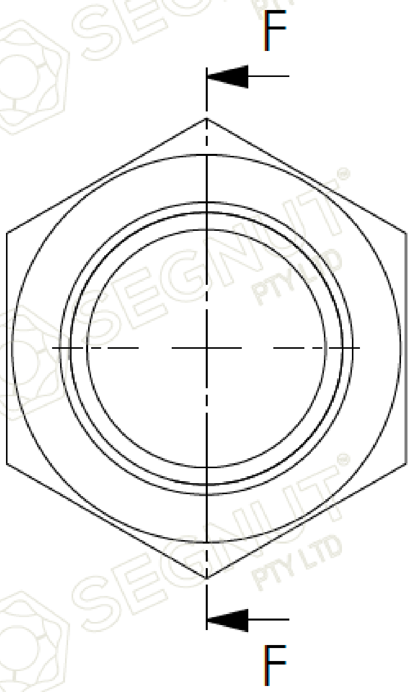
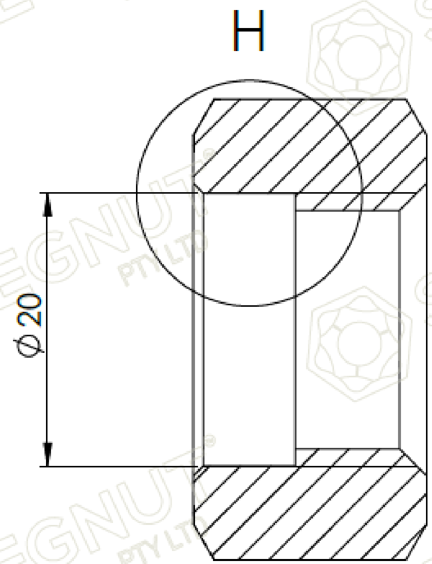
Configuration **3**

Thread Cutout

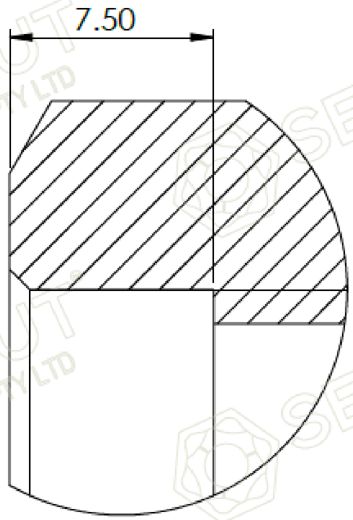
Nut	Quantity
Class 8	x3
Class 10	x3







SECTION F-F



DETAIL H

SCALE 4 : 1

NOTES:

STOCK SIZE XXX
MATERIAL XXX
Heat Treatment XXX
HRc
FINISH XXX

UNLESS OTHERWISE SPECIFIED:
Dimensions are in mm
Break all sharp edges 0.3 x 45° or R0.3
TOLERANCES:
ANGULAR: ± 0.5°
0= ±0.5
0.0 = ± 0.15
0.00 = ± 0.05
SURFACE FINISH

DRAWN AKP | DATE Tuesday, April 17, 2018 11:00:46 AM
CHECKED SWT | 17/09/2017
Part N°

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DRW Nº: M20 Nut for machining
Description:
DO NOT SCALE DRAWING | Rev A | WEIGHT: kg | SHEET 4 OF 9 | A4

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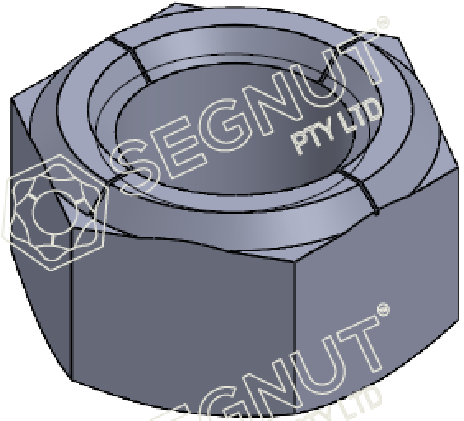
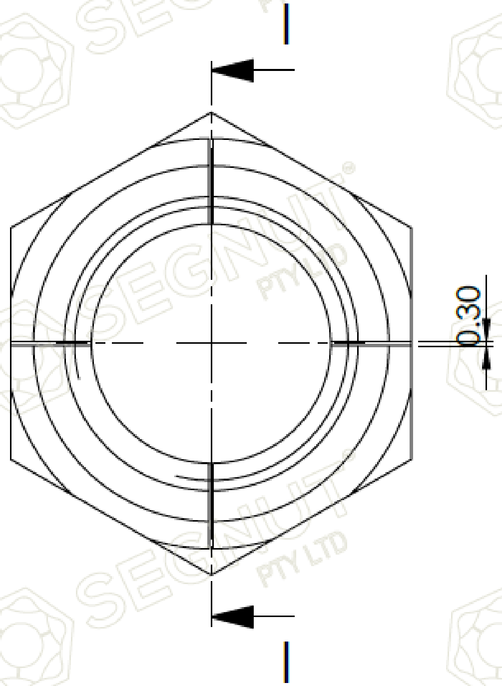
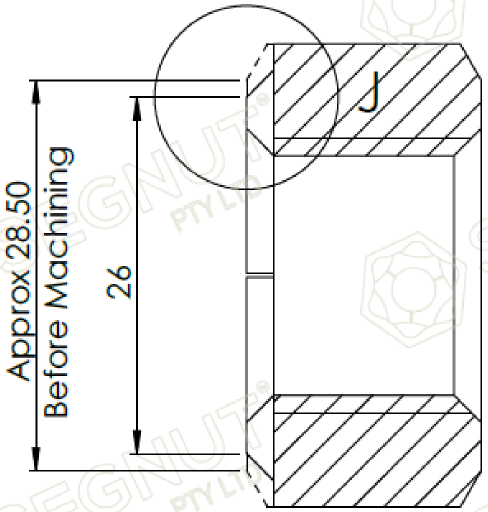
Configuration **4**

Reduced Bearing Surface OD
and 4x Grooves

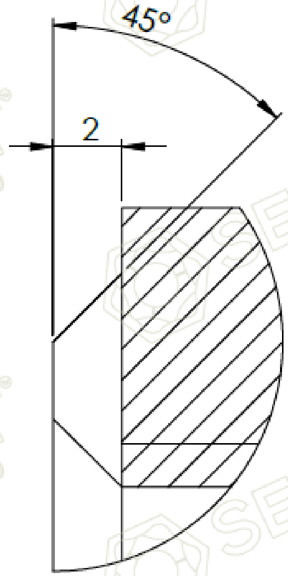
Nut	Quantity
Class 8	x3
Class 10	x3







SECTION I-I



DETAIL J

SCALE 5 : 1

NOTES:

STOCK SIZE	UNLESS OTHERWISE SPECIFIED:	DRAWN	DATE
XXX	Dimensions are in mm	AKP	Tuesday, April 17, 2018 11:00:46 AM
MATERIAL	Break all sharp edges	CHECKED	
XXX	0.3 x 45° or R0.3	SWT	17/09/2017
Heat Treatment	TOLERANCES:	Part N°	
XXX	ANGULAR: ± 0.5°		
HRc	0= ±0.5		
	0.0 = ± 0.15		
	0.00 = ± 0.05		
FINISH	SURFACE FINISH		
XXX			

REV | DESCRIPTION | DATE | CHK | APP.

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DRW №: M20 Nut for machining

Description:

DO NOT SCALE DRAWING | Rev A | WEIGHT: kg | SHEET 5 OF 9 | A4

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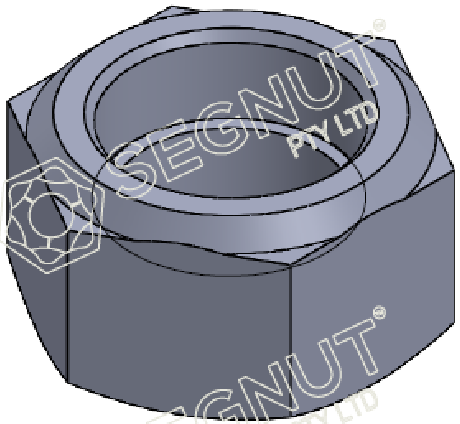
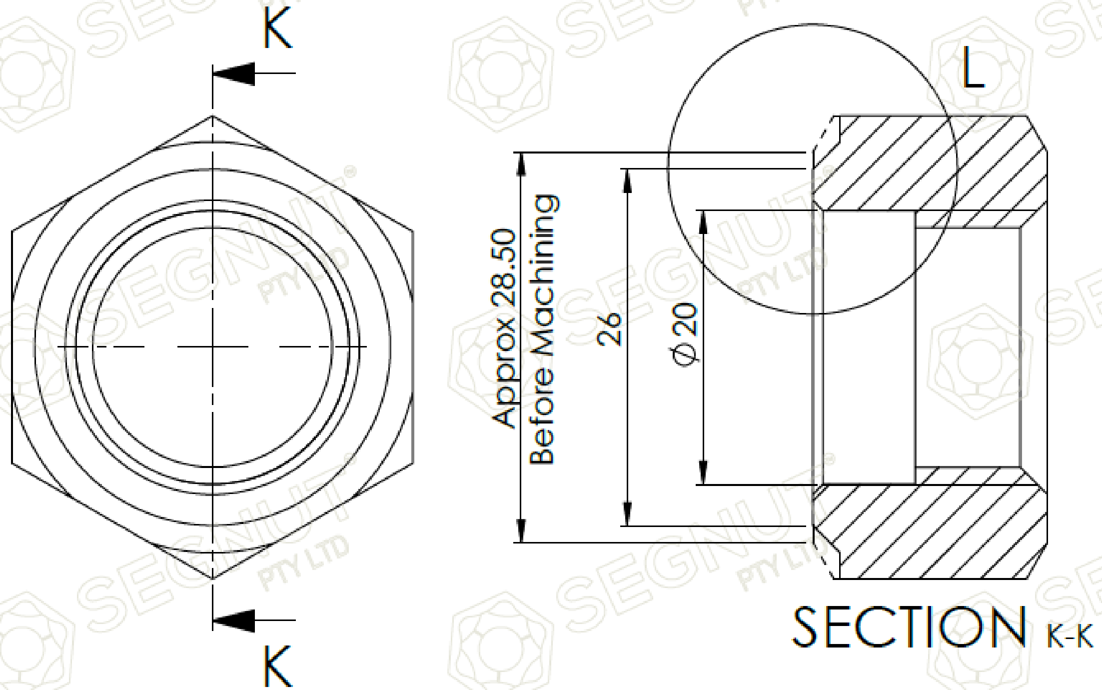


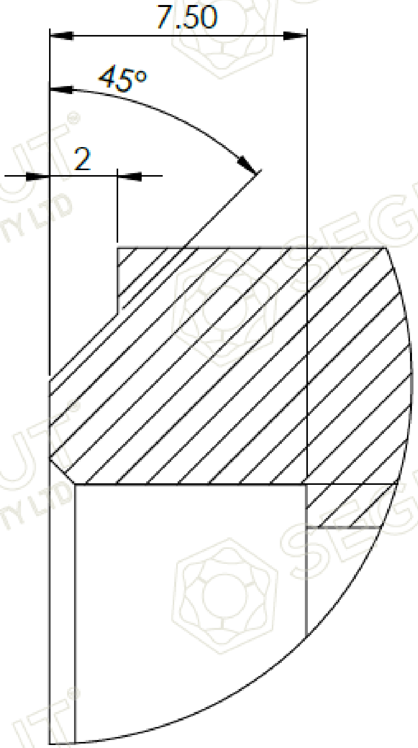
Configuration **5**

Reduced Bearing OD and
Thread Cutout

Nut	Quantity
Class 8	x3
Class 10	x3



SECTION K-K



DETAIL L

SCALE 5 : 1

NOTES:

STOCK SIZE
XXX
MATERIAL
XXX
Heat Treatment
XXX
HRc
FINISH
XXX

UNLESS OTHERWISE SPECIFIED:
Dimensions are in mm
Break all sharp edges
0.3 x 45° or R0.3
TOLERANCES:
ANGULAR: ± 0.5°
0= ±0.5
0.0 = ± 0.15
0.00 = ± 0.05
SURFACE FINISH

DRAWN	AKP	DATE	Tuesday, April 17, 2018 11:00:46 AM
CHECKED	SWT		17/09/2017
Part N°			

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DRW №: M20 Nut for machining
Description:
DO NOT SCALE DRAWING | Rev A | WEIGHT: kg | SHEET 6 OF 9 | A4

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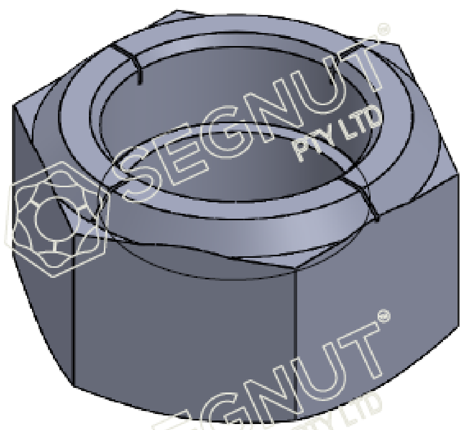
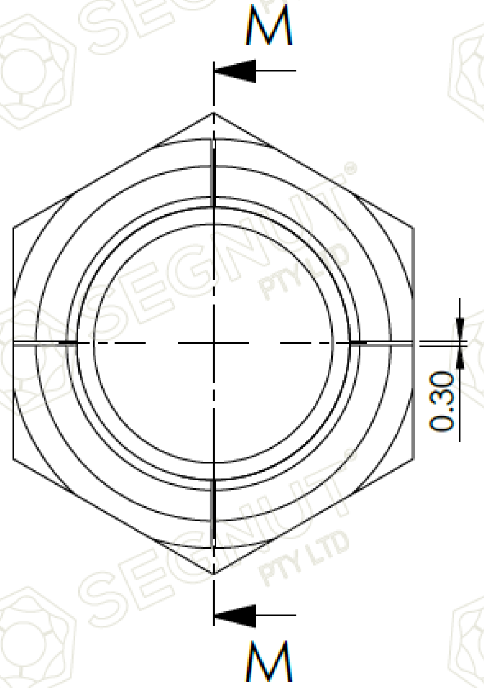
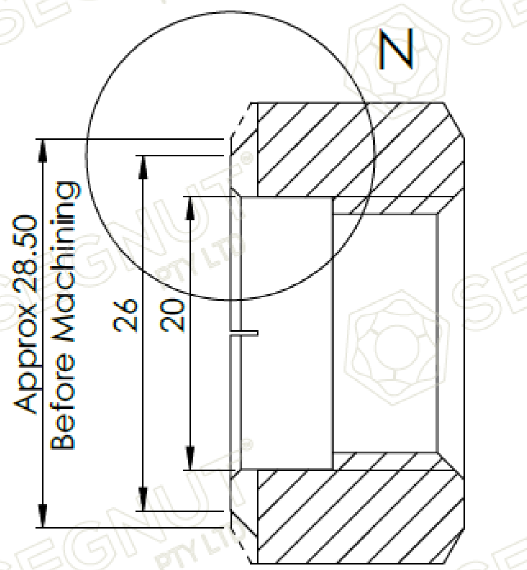
Nut	Quantity
Class 8	x3
Class 10	x3

Configuration **6**

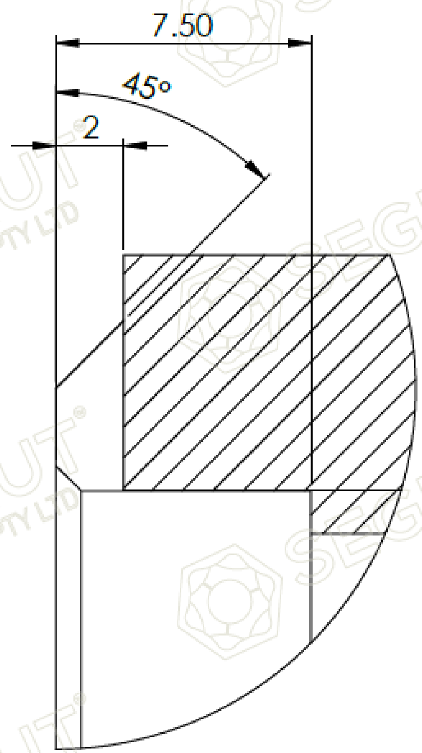
Reduced Bearing Surface OD,
4x Grooves, and
Thread Cutout







SECTION M-M



DETAIL N

SCALE 5 : 1

NOTES:

STOCK SIZE XXX	UNLESS OTHERWISE SPECIFIED: Dimensions are in mm	DRAWN AKP	DATE Tuesday, April 17, 2018 11:00:46 AM	SEGNUT PTY LTD			
MATERIAL XXX	Break all sharp edges 0.3 x 45° or R0.3	CHECKED SWT	17/09/2017	THE INFORMATION CONTAINED IN THIS DRAWING IS THE SOLE PROPERTY OF SEGNUT PTY LTD . ANY REPRODUCTION IN PART OR AS A WHOLE WITHOUT THE WRITTEN PERMISSION OF SEGNUT PTY LTD ISPROHIBITED.			
Heat Treatment XXX	TOLERANCES: ANGULAR: ± 0.5°	Part N°					
HRc	0= ±0.5 0.0 = ± 0.15 0.00 = ± 0.05	DRW №:	M20 Nut for machining				
FINISH XXX	SURFACE FINISH	Description:					
		DO NOT SCALE DRAWING	Rev A	WEIGHT: kg	SHEET 7 OF 9	A4	

REV	DESCRIPTION	DATE	CHK	APP
-	-	-	-	-

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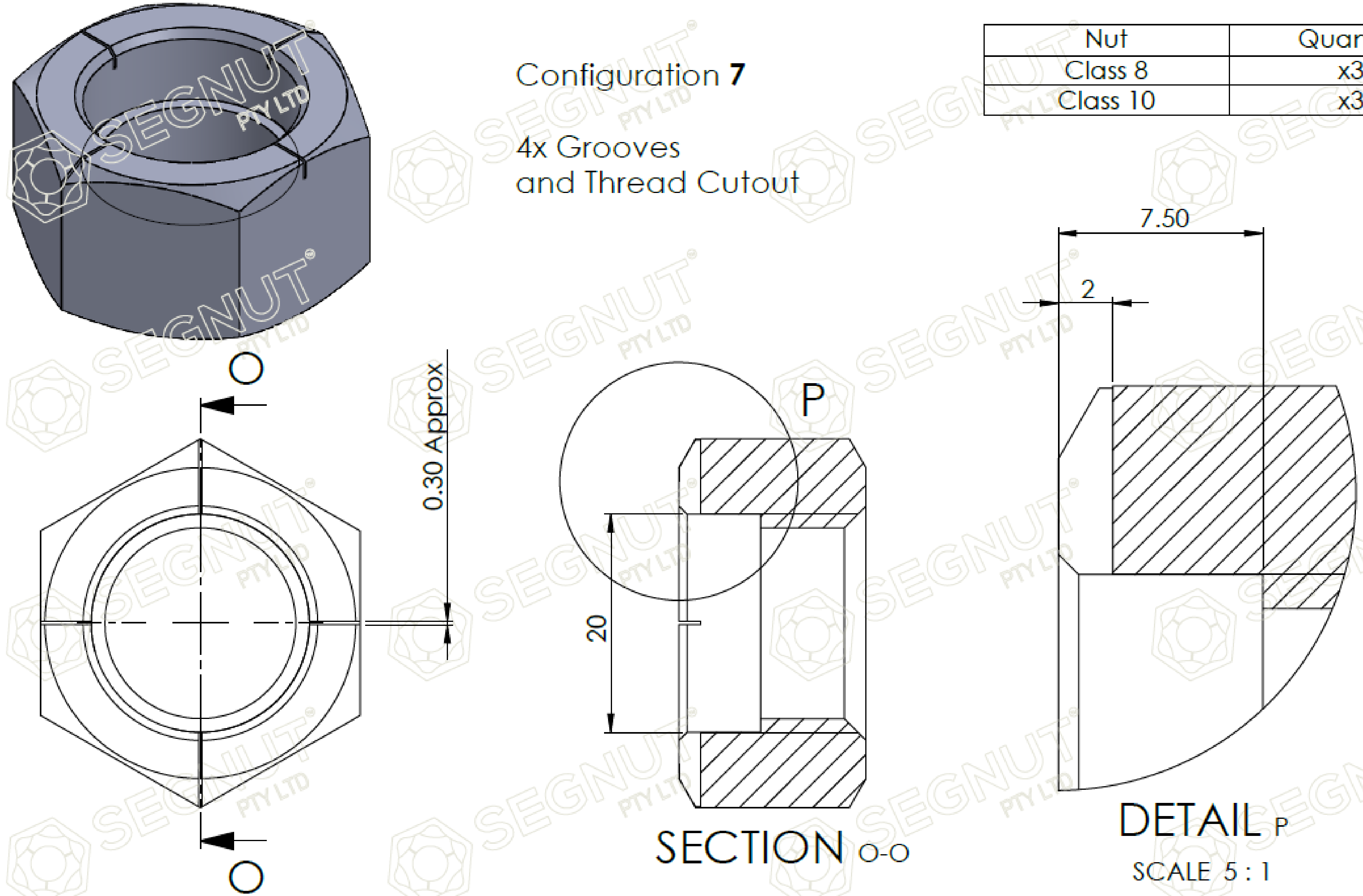
Configuration **7**

4x Grooves
and Thread Cutout

Nut	Quantity
Class 8	x3
Class 10	x3



NOTES:

STOCK SIZE XXX	UNLESS OTHERWISE SPECIFIED: Dimensions are in mm Break all sharp edges 0.3 x 45° or R0.3 TOLERANCES: ANGULAR: ± 0.5° 0= ±0.5 0.0 = ± 0.15 0.00 = ± 0.05 SURFACE FINISH	DRAWN AKP / DATE Tuesday, April 17, 2018 11:00:46 AM	SEGNUT PTY LTD
MATERIAL XXX		CHECKED SWT 17/09/2017	THE INFORMATION CONTAINED IN THIS DRAWING IS THE SOLE PROPERTY OF SEGNUT PTY LTD. ANY REPRODUCTION IN PART OR AS A WHOLE WITHOUT THE WRITTEN PERMISSION OF SEGNUT PTY LTD IS PROHIBITED.
Heat Treatment XXX		Part N°	
HRc		DRW №: M20 Nut for machining	
FINISH XXX		Description:	
		DO NOT SCALE DRAWING / Rev A / WEIGHT: kg	SHEET 8 OF 9 / A4

REV / DESCRIPTION / DATE / CHK / APP

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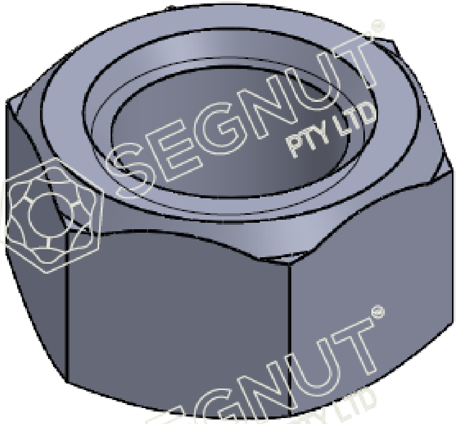
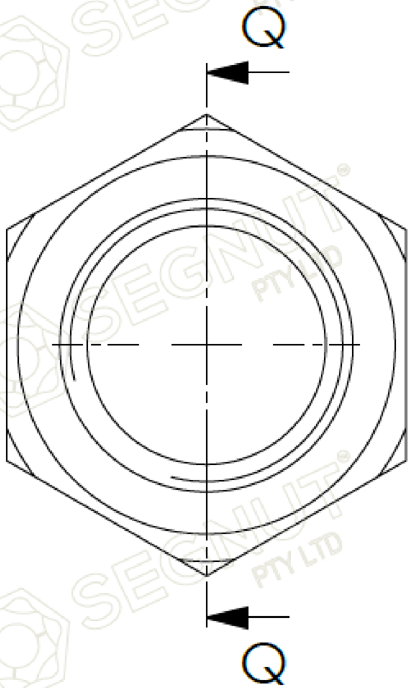
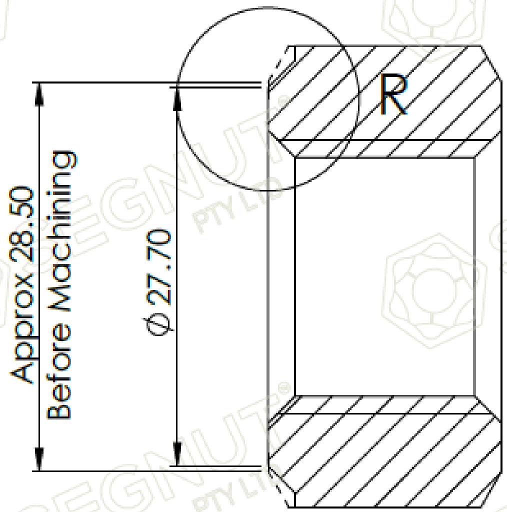


Configuration **8**

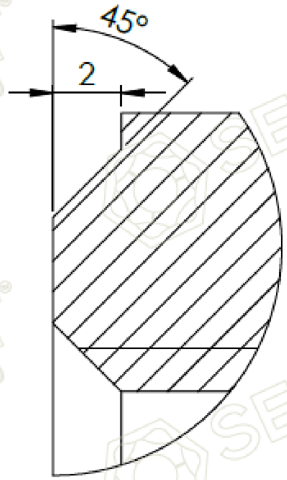
Reduced Bearing Surface OD to min allowed by AS1112.1, 27.7mm OD

Nut	Quantity
Class 8	x6





SECTION Q-Q



DETAIL R

SCALE 5 : 1

NOTES:

STOCK SIZE XXX
MATERIAL XXX
Heat Treatment XXX
HRc
FINISH XXX

UNLESS OTHERWISE SPECIFIED:
Dimensions are in mm
Break all sharp edges
0.3 x 45° or R0.3
TOLERANCES:
ANGULAR: ± 0.5°
0= ±0.5
0.0 = ± 0.15
0.00 = ± 0.05
SURFACE FINISH

DRAWN	AKP	DATE	Tuesday, April 17, 2018 11:00:46 AM
CHECKED	SWT		17/09/2017
Part N°			

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DRW Nº: M20 Nut for machining

Description:

DO NOT SCALE DRAWING | Rev A | WEIGHT: kg | SHEET 9 OF 9 | A4

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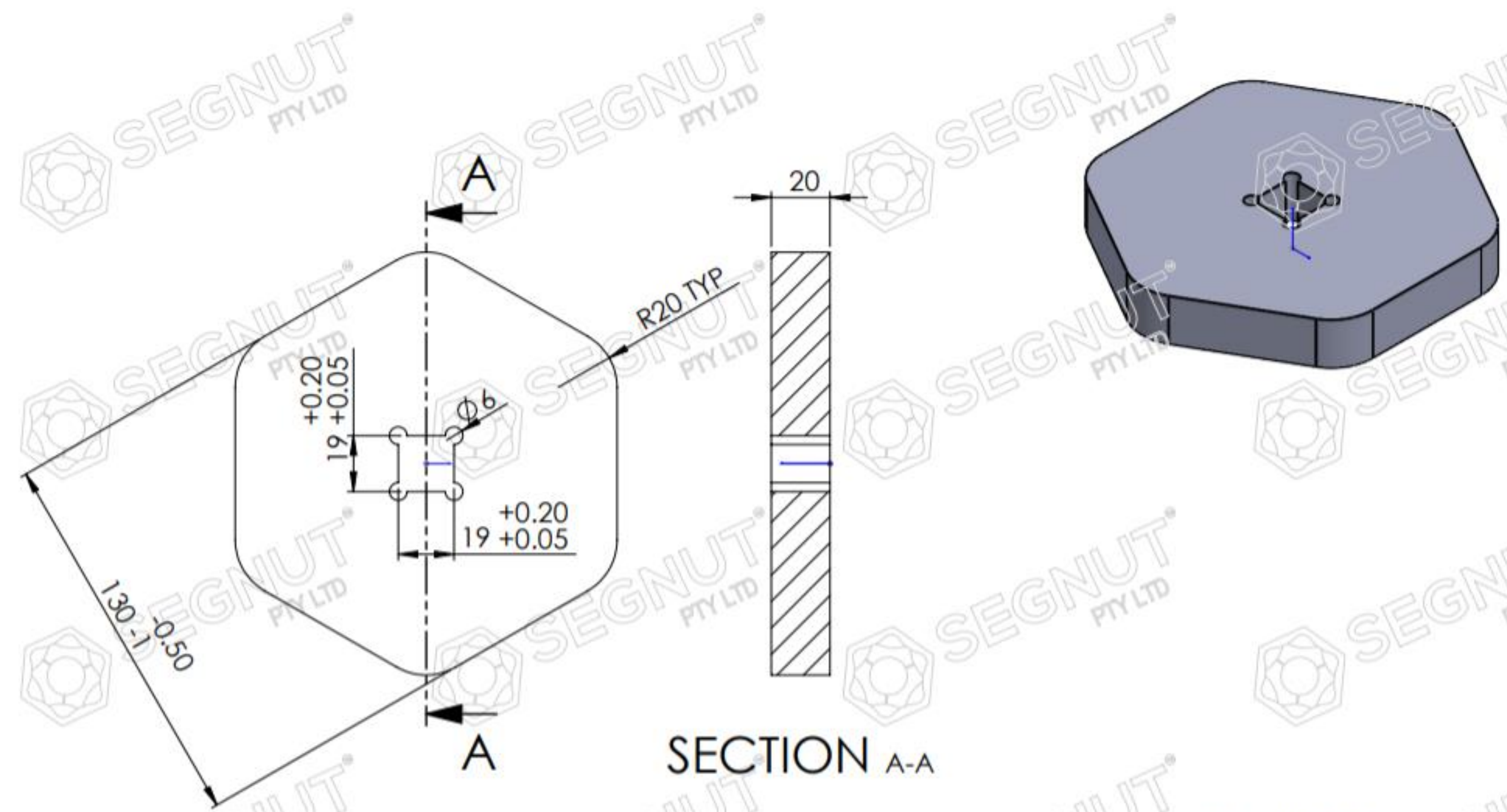

SECTION A-A

Quantitiy: 1

NOTES:

STOCK SIZE 145x145x20	UNLESS OTHERWISE SPECIFIED: Dimensions are in mm	DRAWN AKP	DATE Tuesday, May 8, 2018 3:14:53 PM
MATERIAL Plain Carbon Steel	Break all sharp edges 0.3 x 45° or R0.3	CHECKED SWT	08/05/2018
Heat Treatment XXX	TOLERANCES: ANGULAR: ± 0.5°	Part N°	
HRc	0= ±0.5	DRW No:	Calibration Plate
	0.0 = ± 0.15	Description:	
FINISH XXX	0.00 = ± 0.05 SURFACE FINISH	DO NOT SCALE DRAWING Rev A	WEIGHT: kg SHEET 1 OF 1 A4

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REV | DESCRIPTION | DATE | CHK | APP

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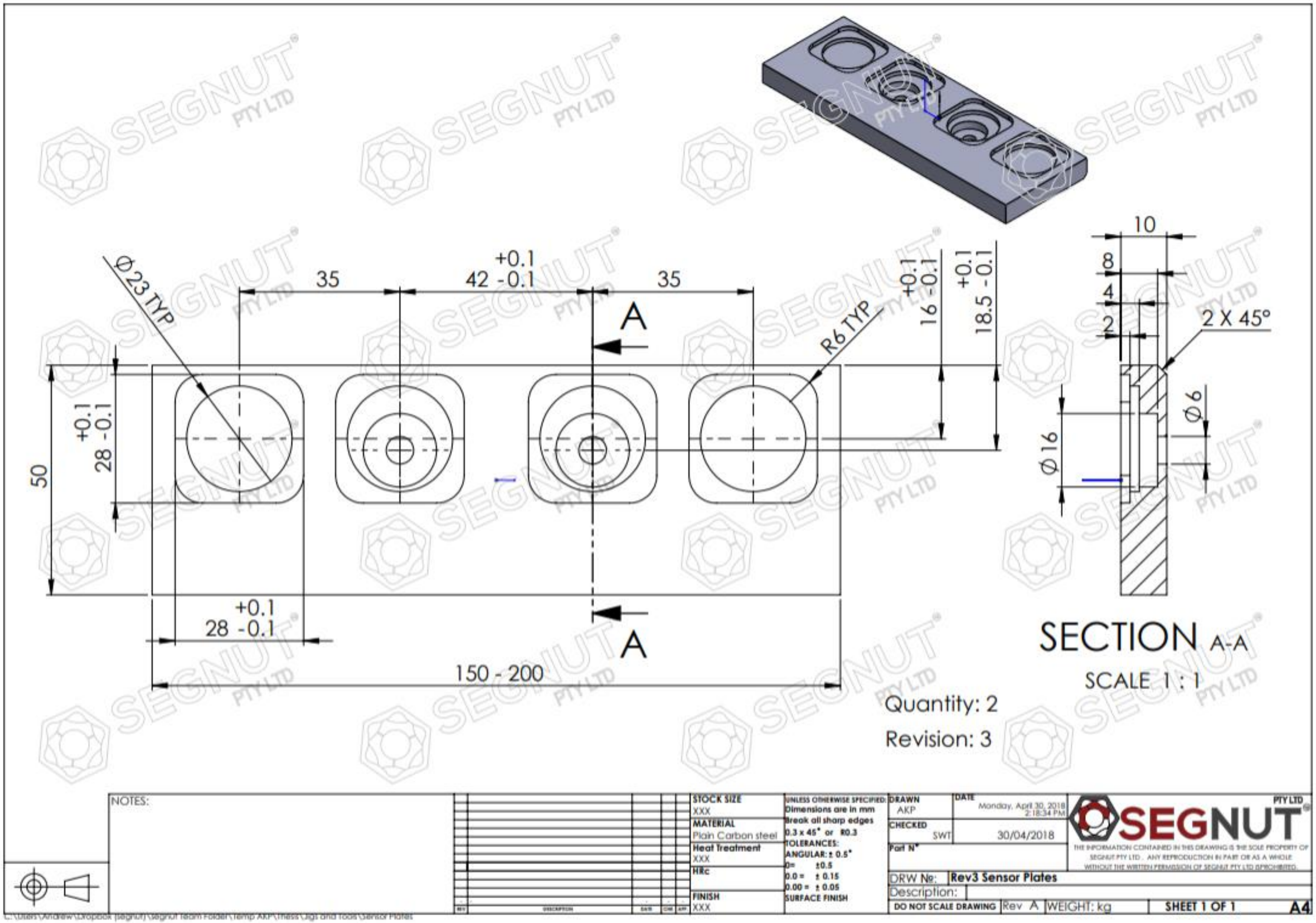

Ø23 TYP
35
+0.1
42 -0.1
35
A
R6 TYP
+0.1
16 -0.1
+0.1
18.5 -0.1
10
8
4
2
2 X 45°
Ø6
Ø16
50
+0.1
28 -0.1
+0.1
28 -0.1
A
150 - 200
SECTION A-A
SCALE 1 : 1
Quantity: 2
Revision: 3
NOTES:
STOCK SIZE
XXX
MATERIAL
Plain Carbon steel
Heat Treatment
XXX
HRc
FINISH
XXX
UNLESS OTHERWISE SPECIFIED:
Dimensions are in mm
Break all sharp edges
0.3 x 45° or R0.3
TOLERANCES:
ANGULAR: ± 0.5°
0= ±0.5
0.0 = ± 0.15
0.00 = ± 0.05
SURFACE FINISH
DRAWN
AKP
DATE
Monday, April 30, 2018 2:18:34 PM
CHECKED
SWT
30/04/2018
Part N°
SEGNUT PTY LTD
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DRW No: Rev3 Sensor Plates
Description:
DO NOT SCALE DRAWING
Rev A
WEIGHT: kg
SHEET 1 OF 1
A4

HAIYAN YUXING NUTS CO.,LTD.

CHANGQIAN TOWN,HAIYAN COUNTY ZHEJIANG ,314304 CHINA

QUALITY CERTIFICATE COUNTRY OF ORIGIN-CHINA

CUSTOMER:BRIGHTON-BEST INTERNATIONAL,INC. SIZE:M20-2.50
GOODS: HEX NUT, CLASS 10 ISO 4032/ISO 8673, PLAIN (METRIC)
MARKED H10
ORDERNO.:A05765 DATE:AUG.08,2017
PART NO.:316019 INV NO.: 00169993
LOT NO.: HY17526M20P10 LOT SIZE:1.60 MPCS
MATERIAL TYPE: SWRCH45K HEAT NO.:J11704877

CHARACTERISTIC	SEPCIFICAT	STANDARD (MM)	RESULT	ACCEPT
WIDTH ACROSS FLATS SAMPLE SIZE N=32	ISO4032	MAX-MIN 30-29.16	MAX-MIN 29.65-29.58	OK
WIDTH ACROSS CORNER SAMPLE SIZE N=32	ISO4032	MIN 32.95	MAX-MIN 33.28-33.20	OK
HEIGHT SAMPLE SIZE N=32	ISO4032	MAX-MIN 18-16.9	MAX-MIN 17.35-17.28	OK
THREAD 'GO"SAMPLE SIZE N=32	ISO 724	6H	Ok	OK
THREAD "NO GO" SAMPLE SIZE N=32	ISO 724	6H	Ok	OK
PROOF LOAD SAMPLE SIZE N=4	ISO 898-2	MIN 259700N	259700N	OK
HARDENESS SAMPLE SIZE N=8	ISO 898-2	HV5 MAX 272-353	330-310	OK
CHEMICAL ANALYSIS	C 0.45	Mn 0.68 / Si 0.15	P 0.016	S 0.005

THIS CERTIFICATE CONFIRMING QUALIFICATION TO ISO 898-2-2012/EN10204-3.1

FACTORY INSPECTOR: Huang Weiming DIRECTOR: Shen Jiahua

沈家华

Bureau Veritas AIRS Pty Ltd
30 Tulloch Way
Canning Vale WA 6155
T 08 6250 2800 F 08 6250 2801
E ronnie.james@au.bureauveritas.com
ABN 86 000 928 816

Accredited for compliance with ISO/IEC 17025
Accreditation number 1267
Site Number 13379

Certificate of Analysis

Bureau Veritas Report Number: MT6589
Date: 24/11/2014

<u>VICKERS HARDNESS TESTING (NUTS)</u>

PRODUCT CODE:	AS/NZS4291.2 Class 8
TEST SPECIFICATION:	AS 1817.1-2003
BUREAU VERITAS PROCEDURE:	P-3000-ME-0029
TEST PIECE FORM:	Cross Section
TEST SURFACE PREPARATION	Ground Flat
HARDNESS SCALE:	HV30
INDENTER:	Diamond

Test No	Test Location	Hardness Results HV30
1	Face	233, 236, 236
2	Face	235, 256, 251
3	Face	248, 246, 246
4	Face	293, 291, 294
6	Face	248, 259, 256
12	Face	224, 220, 226
13	Face	246, 241, 261
14	Face	226, 236, 230
15	Face	242, 233, 241

Compliance statement:
The results from the following tests comply with the requirements of ***AS/NZS4291.2 Class 8*** *200-302 HV30*

Test No	Test Location	Hardness Results HV30
8	Face	310, 305, 306
10	Face	307, 322, 300
11	Face	300, 315, 310

Compliance statement:

The results from the following tests comply with the requirements of ***AS/NZS4291.2 Class 8*** *233-353 HV30*

Test No	Test Location	Hardness Results HV30
5	Face	330, 322, 310
7	Face	333, 334, 326
9	Face	341, 337, 343

Compliance statement:

The results from the following tests comply with the requirements of ***AS/NZS4291.2 Class 10*** *272-353 HV30*

Ronnie James
Materials Test Lead
DESIGNATED SIGNATORY

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