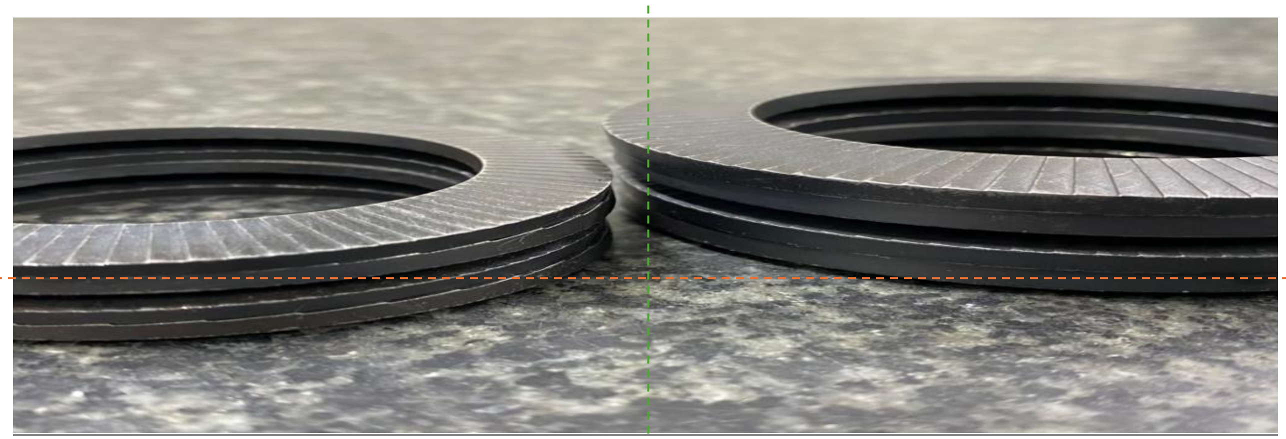
HOW TO USE CONICAL WEDGE LOCK WASHER PAIRS

Introduction

The Conical Wedge Lock washers contained in this package are to be used in pairs. They are manufactured from Spring Steel, hardened to Rockwell C of 46 to 48, with a Black Oxidization surface treatment which provides both anti-corrosion properties and reduced friction, reducing Torque, when properly lubricated. They have 3 distinct physical features:

1. They are shaped in a dome, so that a force must be applied to flatten them, just like a spring.
2. Each has a mating surface with wedge shaped CAMS into which the other locks.
3. The outer surface has smaller radial GRIPS that will embed themselves into the respective assembly surface.

These physical properties are specifically designed to prevent loosening of any static bolted connection they are applied to, and deals with both **spontaneous** loosening and **non-rotational** loosening.

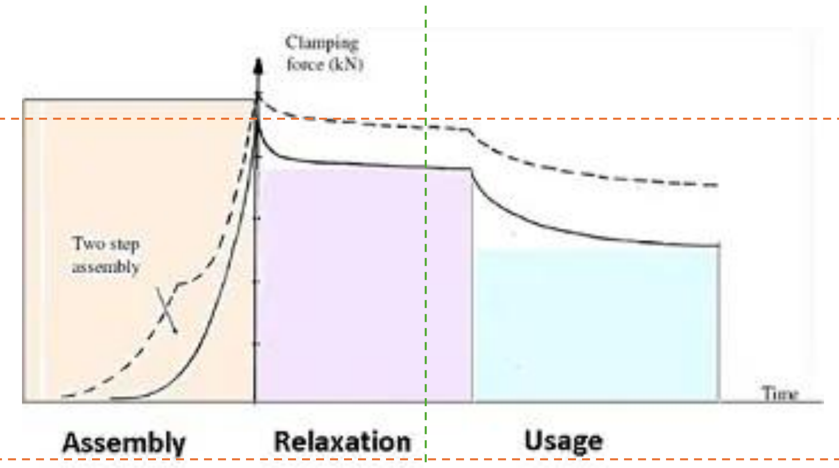


Spontaneous loosening of a bolted assembly takes place when the fastened bolt/nut rotates "spontaneously", and often the slightest rotation of only a few degrees is sufficient for a bolted connection **to lose all its preload**.

This rotational loosening is exacerbated when any part of the bolted assembly experiences slackening. This is a collective term for the various causes **of non-rotational loosening**.

The Conical Wedge Lock Washer Pair contained in this packet address both the rotational and non-rotational causes for loss of preload in a well-designed bolted connection, firstly (as the Safety Washer does) using CAM geometry and contact GRIPS to deal with spontaneous bolt loosening **from shocks, vibrations, and dynamic loading** and secondly, (which is where our improvement focuses) it addresses slackening due to **settlement, creep, differential thermal expansion, relaxation and yielding**.

Settlement (also called Embedding) takes place on the contact surfaces between the various assembly parts and is easily observed by indentations. It is caused by localised plastic deformation. Surfaces may appear to be smooth but when sufficiently magnified, jagged peaks are evident (these are the same structures that Shot Peening targets as they are also the source of fatigue crack propagation). They are called asperities.



A drop of preload directly following the tightening procedure is mainly attributed **to settlement and elastic recovery**. Unlike with settlement or creep, the **clamp length does not change**, which makes it more difficult to detect. Subsequent loss of preload follows a power function with time. Retightening of bolts can reduce the preload loss due to relaxation, but the preload loss directly following the tightening procedure prevails, especially when tightening is done at a fast rate.

The actual contact area between opposing asperities can be substantially less than the apparent area. It is these asperity contacts that deform and flatten. The loading to cause this is substantially less than the load needed to plastically deform the entire apparent contact area. **Stress-Relaxation is the decrease of preload, experienced over time, with no visible physical change to the bolt**. This is a form of creep and occurs when high stresses present in a bolt are relieved over time, most notably, at elevated temperatures.

Creep (sometimes called cold flow) is a property of materials that results in **progressive deformation** and is a result of long-term exposure **to high levels of stress** that are still below the yield strength of the material. Creep is more severe in materials that are subjected to heat for long periods and generally increases as they near their melting point. The rate of creep deformation is a function of the material's inherent mechanical properties, exposure time, exposure temperature and the applied load.

When **slackening occurs** for any reason, the typical wedge geometry will NOT maintain preload and keep the connection secure. This is why we have taken our years of experience in **Conical Disc Spring design** and manufacture and applied these insights to the issue of slackening as a cause of preload loss. A good understanding and control of the torque vs clamp load relationship is essential to minimise the probability of loss of preload as we must firstly work within the mechanical constraints and realities of the assembly.

Conditions arise frequently where these phenomena that cause slackening materialise, here ae a few examples:

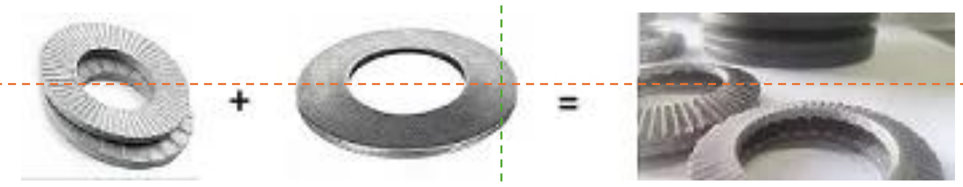
1) Whenever a gasket or similar seal is part of the assembly.
2) Where the clamp load has a Powder Coating or thick Paint surface protection finish
3) Where the clamping is made up of multiple components
4) Where clamped parts are softer metals, or polymers
5) The assembly is subjected to thermal differentials and expands and contracts
6) Where lower property class bolts of 4.6, 8.6 are used, instead of 8.8, 10.9 and 12.9
7) Where field conditions are beyond the control of operations and installation lacks discipline.

In all these scenarios, the geometry of the CAM alone will be insufficient to prevent loosening, and the extra Spring effect, provided by the Conical Shape is required to further reduce the likelihood of preload being lost.

The Evolution of the Conical Wedge Lock Washer

We have explained the sources and cause of slackening in a static bolted connection and why, when slackening occurs for any reason, the typical wedge geometry will not maintain preload and keep the connection secure. This is why we have taken our years of experience in Conical Disc Spring design and manufacture and applied these insights to the issue of slackening as a cause of preload loss.

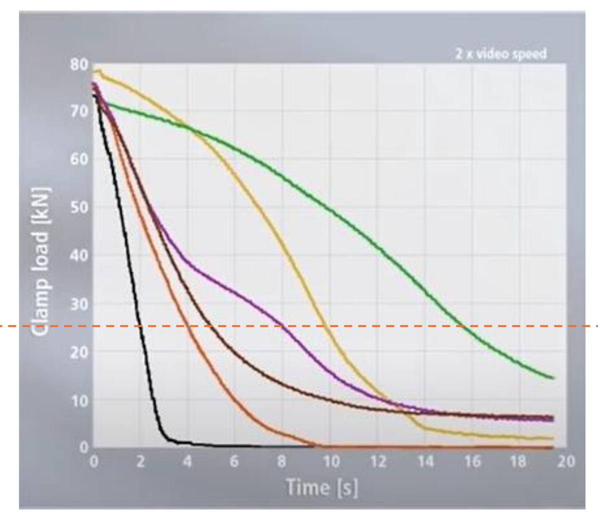
When we combine the Nordloc Safety Washer's Geometry with a Conical Disc Spring

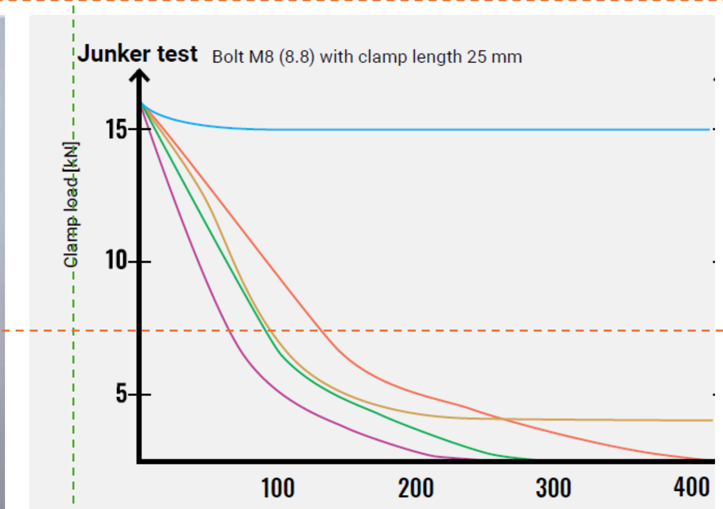


When the fasener is tightened the CAMS lock and he GRIPS on the outer surfaces embed themselves into the fastener nut and the clamped part, leaving impressions on both. Because the gradient of of the wedges is greaer than the size of the fasteners pitch, any rotational movement results in tightening of the wedges and an increase in preload, thus securing the assembly.

Comparative Junker Tests

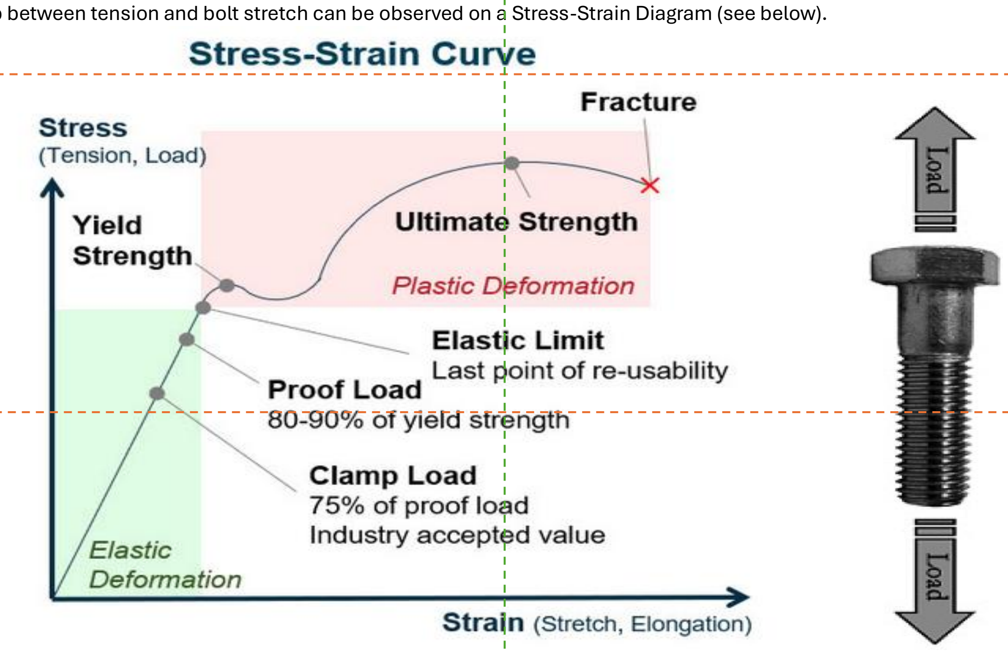
What typically happens when different solutions are applied to a bolted connection? Without exception, whether a nut and washer, spring washer, double nut, Nylon insert, Helical Spring washer, preload is lost spontaneously.





The Stress-Strain Curve

The relationship between tension and bolt stretch can be observed on a Stress-Strain Diagram (see below).



Most parts of a bolted assembly operate within the elastic region and revert to shape after the load is released, but this is only if the induced stresses **have not exceeded the underlying material's yield strength**. A bolt assembly that is properly tensioned should be working in the elastic range (in green), if the load applied causes the fastener to exceed its yield point, it enters the plastic deformation range (in pink).

Bolts have a **defined Proof Load** which is an applied tensile load that the fastener must support without permanent deformation. The Yield Point is the point at which permanent deformation begins. At this point, the metal is no longer able to return to its original shape if the load is removed. Continuing with the application of more load, gets us to the maximum stress load. Beyond this point the fastener continues to "neck" and elongate further, with a reduction in stress.

We have used the ISO 898-1 standard for the calculations we provide on the various class of bolt most commonly used in bolted connections. We use **a Clamp Load that is 75% - 80% of the defined Proof Load**, and we will show you how to verify whether any combination of applied torque and preload for any particular class of bolt falls within the mechanical limits that the standard sets. Everything revolves around how much stress a load will induce in the bolt, and what deformation of the bolt can be tolerated.

Understanding the effect of k-factor

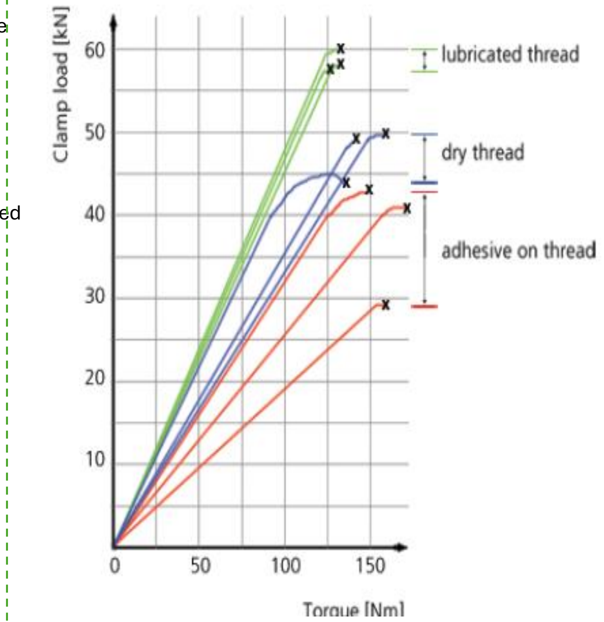
The best way to think of the K_{est} / Nut Factor is as a **measure of as anything that increases or decreases the friction** within the bolted assembly and the threads of the nut. There are three contribution factors:

1) the friction between the threads of the bolt and the threads of the nut – called the Thread Friction - μ_{th} or μ_s (s as in shank)

2) the friction of the nut against the surface against which it rotates and bonds – called the under-head friction, - μ_h or μ_w and

3) friction causes by variations in geometry and the profile of the threads – geometric friction – sometimes frictional drag associated with the thread locking adhesive is included here

   This 3rd contribution is only generally only apparent when the thread is damaged (nicked) or intentionally through design



We calculate what Torque must be applied to a bolted connection using:

$$T = K \times d \times F$$

Where:

d = the nominal diameter of the bolt thread
F = the Load that must be generated from applying the Torque, and
K = the Nut Factor (also called the K factor, torque coefficient, friction factor)

Fortunately, the K-Factor can be looked up in tables that have been empirically worked out for you. But as a rule of thumb, you can use the following estimations, where the Bolt Condition is:

- Non-plated and dry – K = 0.2 to 0.3
- Zinc-plated and clean. But dry – K = 0.17 to 0.22
- Black Oxided and lubricated– K = 0.13 to 0.18
- Properly lubricated – K = 0.10 to 0.14

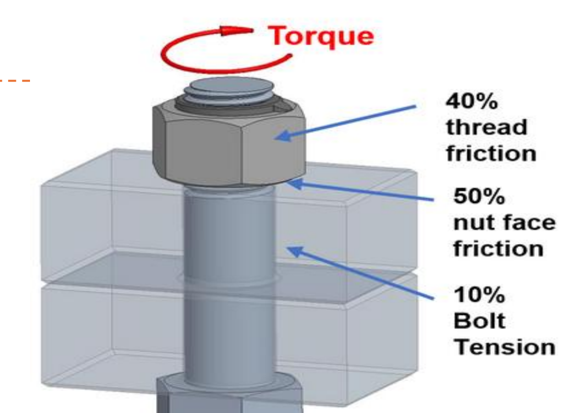
Bolt and Nut Coarse Screw Threads
JIS B 0205 (ISO 724) thread standard

Coefficient of Friction K										
Between Threads, μ_s	Between Bearing Surfaces, μ_W									
	0.08	0.10	0.12	0.15	0.20	0.25	0.30	0.35	0.40	0.45
0.08	0.117	0.130	0.143	0.163	0.195	0.228	0.261	0.293	0.326	0.359
0.10	0.127	0.140	0.153	0.173	0.206	0.239	0.271	0.304	0.337	0.369
0.12	0.138	0.151	0.164	0.184	0.216	0.249	0.282	0.314	0.347	0.380
0.15	0.153	0.167	0.180	0.199	0.232	0.265	0.297	0.330	0.363	0.396
0.20	0.180	0.193	0.206	0.226	0.258	0.291	0.324	0.356	0.389	0.422
0.25	0.206	0.219	0.232	0.252	0.284	0.317	0.350	0.383	0.415	0.448
0.30	0.232	0.245	0.258	0.278	0.311	0.343	0.376	0.409	0.442	0.474
0.35	0.258	0.271	0.284	0.304	0.337	0.370	0.402	0.435	0.468	0.500
0.40	0.285	0.298	0.311	0.330	0.363	0.396	0.428	0.461	0.494	0.527
0.45	0.311	0.324	0.337	0.357	0.389	0.422	0.455	0.487	0.520	0.553

Bolt and Nut Fine-Screw Threads
JIS B 0207 thread standard (ISO 724)

Coefficient of Friction K										
Between Threads, μ_s	Between Bearing Surfaces, μ_W									
	0.08	0.10	0.12	0.15	0.20	0.25	0.30	0.35	0.40	0.45
0.08	0.106	0.118	0.130	0.148	0.177	0.207	0.237	0.267	0.296	0.326
0.10	0.117	0.129	0.141	0.158	0.188	0.218	0.248	0.278	0.307	0.337
0.12	0.128	0.140	0.151	0.169	0.199	0.229	0.259	0.288	0.318	0.348
0.15	0.144	0.156	0.168	0.186	0.215	0.245	0.275	0.305	0.334	0.364
0.20	0.171	0.183	0.195	0.213	0.242	0.272	0.302	0.332	0.361	0.391
0.25	0.198	0.210	0.222	0.240	0.270	0.299	0.329	0.359	0.389	0.418
0.30	0.225	0.237	0.249	0.267	0.297	0.326	0.356	0.386	0.416	0.445
0.35	0.252	0.264	0.276	0.294	0.324	0.353	0.383	0.413	0.443	0.472
0.40	0.279	0.291	0.303	0.321	0.351	0.381	0.410	0.440	0.470	0.500
0.45	0.306	0.318	0.330	0.348	0.378	0.408	0.437	0.467	0.497	0.527

A **low** nut factor gives a **higher preload and clamping force** but puts the bolt **closer to yield** while a **high** nut factor gives a **lower preload and clamping force** but the capacity of the joint to resist external tensile loads has been reduced



When tightening a bolt, the same amount of torque applied to the assembly will result in more of the energy being used on overcoming friction rather than applying load to the bolt!
All things being equal, the same Torque applied to a DRY bolt will end up with LESS preload than the same amount of Torque being applied to a well lubricated bolt!
Of course, a well lubricated connection, will more easily loosen spontaneously! Below is a summarised chart of the ISO 898-1 mechanical properties for bolts which are most widely used in fastening. Throughout our calculations and testng we have adhered to the most common convention of bolt design
Proof load is defined as the maximum tensile force that can be applied to a bolt that will not result in plastic deformation. A material must remain in its elastic region when loaded up to its proof load typically between 85-95% of the yield strength.

Clamp L:oad (preload) - For a reasonable factor of safety, use 75% to 80% of the Bolt rating. Acceptable Pre-Load load is typically 75% of Proof load but we have used 80%.

Mechanical Property as per ISO 898-1		Property Class					
		4.6	6.8	8.8<=M16	8.8>M16	10.9	12.9
Tensile Strength, N/mm^2	nominal	400	600	900	900	1040	1200
Rm	min	420	600	800	830	1000	1220
Yield Strength, N/mm^2	nominal	320	480	640	640	900	1080
Rm	min		480	640	660	940	1100
Stress Under Proof Load Rp0,2, N/mm²	nominal	225	440	580	600	830	970
Vickers Hardness,	max	220	250	320	335	380	435
HV	min	120	190	2250	255	320	385
Brinell Hardness,	max	209	238	304	315	360	415
HB	min	114	181	238	240	305	365
Rockwell C Hardness,	max				34	39	44
HRC	min				22	32	39

Please turn over to read the Torque and Target preload tables for different lubrication and bolt class combinations

K-factor of 0.153 would be if something like **Graphite Paste** is used
K-factor of 0.173 if a **lubricating oil** is used
K-factor of 0.258 is if no lubrication is used and the bolted **connection is dry.**

Match Engineering
Anti-loosening Conical Wedge Lock Washer Pairs for Bolted Connections.
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Torque and target preload tables for different lubrication and bolt class combinations
Mechanical Properties as per ISO 898-1

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HEXAGON FULL NUTS DIN 934 / ISO 4032 Target Preload and Required Torque by Bolt Size and Property Class for Coarse Pitch at defined G_F (70%) for different lubrication scenarios (k-factor). Graphite Grease, WD40 & Dry

Bolt Size	S [mm]	M [mm]	Pitch Course [mm]	Area [mm²]	G_F = 70.00% 70.00% factor	6.8 Preload [kN]	6.8 Torque {Nm}	8.8 Preload [kN]	8.8 Torque {Nm}	10.9 Preload [kN]	10.9 Torque {Nm}	12.9 Preload [kN]	12.9 Torque {Nm}
M80	64	100	6	4311	0.153 0.206 0.258	1328	16255 21885 27410	1811	22162 29839 37371	2505	30658 41278 51697	2927	35829 48240 60417
M85	68	120	6	4912	0.153 0.206 0.258	1513	19677 26493 33180	2063	26829 36123 45242	2854	37115 49972 62586	3335	43374 58399 73141
M90	72	130	6	5553	0.153 0.206 0.258	1710	23547 31703 39706	2332	32116 43241 54156	3226	44426 59816 74915	3771	51920 69905 87551
M95	76	135	6	6233	0.153 0.206 0.258	1920	27907 37574 47059	2618	38051 51232 64165	3621	52637 70871 88761	4232	61515 82824 103731
M100	80	145	6	6952	0.153 0.206 0.258	2141	32757 44105 55238	2920	44673 60148 75331	4039	61798 83206 104209	4720	72222 97240 121786
M105	84	140	6	7711	0.153 0.206 0.258	2375	38154 51371 64339	3239	52028 70051 87734	4480	71973 96905 121366	5236	84113 113250 141838
M110	88	155	6	8509	0.153 0.206 0.258	2621	44111 59392 74384	3574	60147 80982 101424	4944	83203 112024 140302	5778	97237 130920 163968
M115			8	9346	0.153 0.206 0.258			3925	69066 92990 116464	5430	95541 128637 161108	6346	111656 150334 188283
M120			8	10222	0.153 0.206 0.258			4293	78823 106128 132918	5939	109040 146812 183871	6941	127431 171574 214884
M125	100	180	6	9346	0.153 0.206 0.258			4524	86525 116498 145906	6259	119694 161156 201837	7314	139884 188341 235883
M130			8	10222	0.153 0.206 0.258			4919	97831 131720 164970	6804	135334 182214 228210	7952	158161 212949 266703
M135			6	10772	0.153 0.206 0.258			5497	113530 152858 191443	7604	157050 211453 264830	8886	183542 247122 309503
M140	112	200	8	11711	0.153 0.206 0.258			5757	123324 166043 207957	7964	170595 229690 287671	9308	199371 268434 336194
M145	120	210	6	13087	0.153 0.206 0.258			6381	141565 190604 238717	8827	195829 263666 330222	10316	228861 308139 385922
M150	128	230	8	13708	0.153 0.206 0.258			6662	152893 205856 257819	9216	211503 284768 356652	10770	247178 332802 416811
M155			8	17456	0.153 0.206 0.258			7332	173867 234095 293187	10142	240515 323831 405575	11853	281084 378454 473986
M160	120	210	6	18173	0.153 0.206 0.258			7633	186849 251574 315078	10559	258472 348008 435855	12340	302071 406710 509375
M170	128	230	8	20640	0.153 0.206 0.258			8669	225476 303581 380214	11992	311907 419953 525960	14015	364520 490791 614680

G_F = Ratio of yield point. When tightening according to guidelines and with no deviation, this is the pre-stress achieved expressed as % of yield point.

Checklist before tightening.

1. Have you decided on what lubrication you will use for your bolted connection?
2. Have you worked out what Torque you should apply to achieve your desired preload?
3. The maximum preload you apply should be around 80% of the bolt's proof load.
4. Are you using the Wedge Lock Washers as a Pair so ha hey can work together?