Fatigue Analysis of Preloaded Bolted Joints

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This article presents Wöhler plots, or S-N curves, for use in the analysis of bolt fatigue of preloaded bolted joints. Preloaded bolts under cyclic loading have a high mean stress with a small alternating stress. This is combined with a large stress concentration at the thread root. The method of fatigue analysis presented uses S-N curves with a zero mean stress. The high mean stress experienced by the bolts is accounted for using a function to calculate a damage-equivalent stress. Notch sensitivity was considered to modify S-N curves for materials with chemical compositions encompassed within the material specifications for high strength bolts. This produced S-N curves for stress concentrations relevant to bolt threads. Curve fitting techniques were used to express these curves as a function of the equivalent stress and stress ratio. An estimate of residual stresses in bolts produced by thread rolling was made.The work provides a practical method of calculating fatigue life.

Keywords:*bolt fatigue, Wöhler plots, S-N curve, residual stress.*

1. INTRODUCTION

Preloaded bolted joints are a common feature in mechanical engineering. They are a convenient means by which to assemble components. There is no requu–irement for post assembly heat treatment to remove or reduce residual stresses, as is often the case for welded assemblies. Most importantly, they can be used to facilitate disassembly for maintenance and repair. The main disadvantage with preloaded bolted joints is that they tend to be bulky and take up a relatively large space envelope compared to the cross-sections of the structural elements or components they are used to connect. This can lead to designers keeping the joint size to an absolute minimum, particularly where the space envelope is critical. This often leads to the joint having to work at its maximum capacity. In many bolted joint applications, the main issue is the static strength of the joint. Providing the bolt preload is at least 1.5 time the design tension and the alternating load is less than 20% of the applied load fatigue should not be an issue, reference BS 7608 (1990)[1]. When bolted joints are highly loaded and/or subjected to high alternating loads the effect of fatigue of the joint needs to be considered.

The log-linear nature of Wöhler plots, or S-N curves, means that the calculated fatigue life is very sensitive to stress. It is well known that a small change in alternating stress can have a significant effect on the fatigue life of a component. Hence, it is important to determine applied stresses with a high degree of accuracy to be able to achieve a meaningful assessment of life. In order to produce an accurate prediction of actual stresses, margins of safety, safety factors, partial safety factors or reserve factors should not be applied to the loads or calculated stresses. The two most common methods of conducting stress analyses of bolted joints are Finite Element Analysis (FEA) and classical methods. A good example of an appropriate FEA can be found in the work conducted by Novoselac et. al. (2014) [2]. Classical methods of analysis need to consider a detailed analysis of stresses. The methods of detail analysis described by Welch (2018) [3] and (2022) [4] are particularly suited to this purpose. Note that the design analysis method that is also described by Welch (2018) [3] should not be used since it does not predict the true bolt stress, instead, it calculates a "bolt related load", which can be defined as "the load passing through the region of the joint controlled by the bolt".

This article considers the classical analysis of bolt fatigue within joints made using preloaded bolts.

2. CLASSICAL ANALYSIS OF BOLT LOADS

Figure 1 shows a typical bolted joint and the free body diagram for one mating component, with the bolt preloads considered as point loads.

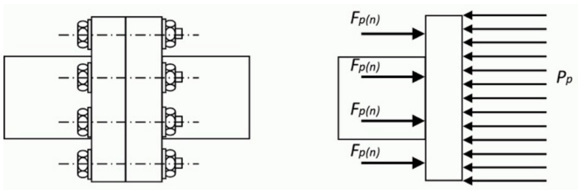

Figure 1. Preloaded Joint.

Figure 2 shows the same joint with an external axial load (tensile) and an external moment applied along with the free body diagram of one mating component.

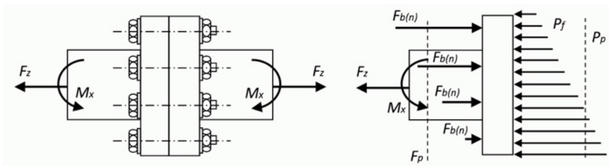

Figure 2. Bolted Joint with External loads Applied.

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The axial bolt load resulting from the combined loading of the preload, external axial load and external moment are given by equation (1).

$$F_{b_{(n)}} = F_p + F_z \cdot \frac{A_b}{A_i} + \frac{M_x}{I_{xx,i}} \cdot y_{(n)} \cdot A_b \tag{1}$$

where $F_{b(n)}$ is the axial load in bolt ‘n’.

The area and second moment of area for the joint, A_j and $I_{xx,j}$ respectively, should ideally reflect the effect of Rotscher’s pressure cone on the contact area of the joint. Hence, the total area of the joint A_j is given by $A_j = A_c + N_b \bullet A_b$where A_c is the effective, or true, contact area, A_b is the bolt tensile area and N_b is the number of bolts.

3. S-N CURVES FOR BOLTS

A characteristic of preloaded bolts under cyclic loading is that they have a high mean stress, due to the bolt preload, which is typically between 60% to 80% of the bolt proof load, with a relatively small alternating stress. This is combined with a large stress concentration at the thread root, in particular at the first thread of the thread engagement where bending loads in the thread flank accentuates the stress concentration. Any fatigue fai–lures usually occur at the root of the first engaged thread. The engaged threads experience bending which results in higher stresses at the root of the first engaged thread than in the threads that are not engaged but are still subjected to the same axial load. The engaged threads after the first thread experience reducing tensile loads, and hence a lower stress, than the first engaged thread as some of the tensile load is transferred from the bolt to the nut or internally threaded component.

Ideally, Wöhler plots, or S-N curves, produced using the same mean stress as experienced by the bolt should be used for the analysis. However, this is impractical since each load case, for each bolt, will have a different mean stress. The method of fatigue analysis being presented here is based on the use S-N curves with a zero mean stress. The high mean stress experienced by the bolts is accounted for by the use of a function that calculates a damage-equivalent stress which can be used with these S-N curves. The damage-equivalent stress is the alter–nating stress under fully reversal load conditions that pro–duces the same amount of damage as the combination of both the alternating stress and the (non-zero) mean stress.

A theoretical study of stress concentration at the root of bolt threads has been carried out by Lehnoff et. al. (2000) [5]. This work used FEA to determine the thread stress concentration factors at the first engaged thread for M8, M12, M16, M20 and M24 bolts. The results of this work are summarised in Table (1).

Table 1. Summary of elastic stress concentration factors.

Thread Size	Stress Concentration Factors		Mean elastic stress concen–tration factor
	Maximum metal condition	Minimum metal condition	
M8	4.33	4.80	4.565
M12	4.32	4.80	4.56
M16	4.67	5.12	4.895
M20	4.77	5.17	4.97
M24	4.82	5.22	5.02

A selection of fatigue S-N curves for materials with chemical compositions that are encompassed by the material specification for high strength bolts, Grade 8.8 and higher, can be found within ‘*Metallic Materials Properties Development and Standardization (MMPDS)*’, reference [6]. These material and available S-N curves are presented in Table 2 and the chemical compositions of the materials compared with the requirements of the bolt specification, given in BS EN ISO 868-1 (2009) [7], are presented in Table 3. Some fatigue tests have been carried out on Grade 10.9 M8 bolts by Marcelo et. al. (2011) [8]. The materials for these bolts were AISI 4135 and SCM 435H wire. The chemical compositions of the wires used by Marcelo et. al. (2011) [8] are also included in Table 3.

Table 2. Materials considered in the analyses.

Material	Condition: Tensile Strength	Stress Concentration	Product form
AISI 4130	Normalised *Ftu* 117ksi (807MPa)	Unnotched $K_t = 5.0$	Sheet 0.075 inch (1.905mm) thick
	Ftu 180ksi (1241MPa)	Unnotched $K_t = 4.0$	Sheet 0.075 inch (1.905mm) thick
AISI 4340	*Ftu* 125ksi (862MPa)	Unnotched $K_t = 3.3$	Rolled bar 1.125 inch (28.575mm) dia.
	Ftu 150ksi (1034MPa)	Unnotched $K_t = 3.3$	Rolled bar 1.12 inch (28.448mm) dia.
	Ftu 200ksi (1379MPa)	Unnotched $K_t = 3.3$	Rolled bar 1.125 inch (28.575mm) dia.

The approximation of S-N curves for the elastic stress concentrations given in Table 1 were made by considering notch sensitivity and the ratio of fatigue stress concentrations. A number of ‘donor’ S-N curves, taken from MMPDS-03 [6], were modified using the term:`

$$\sigma_{alt} = S_{alt,D} \frac{K_{f,D}}{K_f} \tag{2}$$

where σ_{alt} and K_f were the alternating stress and fatigue stress concentration factor of the approximated S-N curve and $S_{alt.D}$ and $K_{f.D}$ refer to the donor S-N curve used for the approximation. Considering the notch sensitivity of a material it was possible to say that:

$$q = \frac{K_{f-1}}{K_{t-1}} \tag{3}$$

where K_t was the elastic stress concentration factor and q was a constant for the material at a given fatigue life.

Hence it was possible to write:

$$q\frac{K_{f-1}}{K_{t-1}} = \frac{K_{f,D}-1}{K_{t,D}-1}$$

Rearranging:

$$K_f = \left(\frac{K_{f,D}-1}{K_{t,D}-1}\right) \cdot \left(K_t - 1\right) + 1 \tag{4}$$

Table 3. Chemical composition of materials.

Bolt Grade	Material									
		C	P	S max	B max	Cr	Mn	Mo	Ni	Si
	AISI 4130	0.28 to 0.33	0.035 max	0.04 max		0.80 to 1.10	0.40 to 0.60	0.15 to 0.25		0.15 to 0.35
Grade 8.8	Carbon steel with additives	0.15 to 0.40	0.035 max	0.035 max	0.003					
	Carbon steel	0.25 to 0.55	0.035 max	0.035 max						
	AISI 4340	0.37 to 0.43	0.035 max	0.04 max		0.70 to 0.43	0.65 max	0.20 to 0.30	1.85 max	0.25 max
Grade 9.8	Carbon steel with additives	0.15 to 0.35	0.035 max	0.035 max	0.003					
	Carbon steel	0.25 to 0.55	0.035 max	0.035 max						
	AISI 4135	0.36	0.022	0.010		0.97	0.81	0.17		0.26
	Standard AISI 4135	0.34	0.018	0.008		0.94	0.79	0.16		0.23
	SCM 435H	0.35	0.032	0.011		0.98	0.76	0.15		0.19
	Standard SCM 435H	0.35	0.012	0.006		0.99	0.74	0.17		0.20
Grade 10.9	Carbon steel with additives	0.15 to 0.35	0.035 max	0.035 max	0.003					
	Carbon steel with additives	0.20 to 0.55	0.035 max	0.035						
	Carbon steel	0.25 to 0.55	0.035 max	0.035	0.003					
	Alloy steel	0.20 to 0.55	0.035 max	0.035						
Grade 12.9	Alloy steel	0.28 to 0.50	0.035 max	0.035	0.003					

The fatigue stress concentration factor for the donor curve was calculated from:

$$K_{f,D} = \frac{S_{alt,0}}{S_{alt,D}} \tag{5}$$

where $S_{alt.0}$ was the alternating stress of the unnotched S-N curve for the material.

Equations (2) to (5) were applied using the materials and stress concentrations presented in Table 2. The resulting S-N curves for AISI 4340, with a tensile strength of 1379MPa showed what appeared to be an anomaly. The fatigue strengths of these curves were less than the fatigue strengths of the same material speci–fication tempered to give the lower tensile strengths of 862MPa and 1034MPa. However, this anomaly can be explained by reference to work by de Souza et. al. (2021) [9]. This work reports that heat treated speci–mens of AISI 4340 exhibited high tensile residual stres–ses at the surface after quenching. Subsequent tempe–ring at 300°C and 400°C showed a small reduction in the residual stresses. Tempering at 500°C and 650°C resulted in compressive residual stresses. These higher tempering temperatures, and hence compressive residual stresses at the surface, result in improved fatigue performance. A conclusion that can be drawn from the work by de Souza et. al. (2021) [9] and the S-N curves produced using equations (2) to (5) is that; in general, an increase in tensile strength results in improved fatigue properties however, if a carbon steel is at or close to its maximum achievable tensile strength its fatigue performance may be impaired. Based on this conclusion, it was decided not to consider the approximated S-N curves for AISI 4340, heat treated to give a tensile strength of 1379MPa or the S-N curves for AISI 4130, heat treated to give a tensile strength of 1241MPa as appropriate for use in the fatigue analysis of preloaded bolted joints.

The work by Marcelo et. al. (2011) [8] considered Grade 10.9 bolts manufactured from AISI 4135 and SCM 435H wire. Both of these materials are at the lower end of the carbon content range allowable under BS EN ISO 868-1 (2009) [7]. AISI 4130 is of similar carbon content to AISI 4135 and SCM 435H, just slightly higher, therefore the S-N curves for bolt Grades 8.8, 9.8 and 10.9 were approximated using the S-N curves for normalised AISI 4130. These S-N curves were factored by the ratio of minimum tensile strength for the bolt grade to the tensile strength associated with normalised AISI 4130 (807MPa).

Although the minimum tensile strength requirements for Grade 12.9 bolts is achievable with AISI 4130 the work by de Souza et. al. (2021) [9] leads to the conclusion that this would result in poor fatigue performance. It was considered that it would be more appropriate to approximate the S-N curves for Grade 12.9 bolts by factoring the S-N curves for AISI 4340, heat treated to give a tensile strength of 1034MPa.

Curve fitting techniques were applied to each of the approximated S-N curves to allow them to be expressed as a function in the form of equation (6):

$$\log\left(N_{life}\right) = C_1 - C_2 \cdot \log\left(\frac{\sigma_{alt}}{F_{tu}} - C_3\right) \tag{6}$$

The constants, C_1, C_2 and C_3, were evaluated using the minimum tensile strength, Ftu, for each bolt grade. The results of equation (6) were then plotted for each bolt grade. The values of constants are presented in Table 4 and the resulting plots are shown in Figures 3 to 6.

Table 4. Summary of elastic stress concentration factors.

	AISI 4130 Normalised ($Ftu = 807MPa$)			AISI 4340 $Ftu = 1034MPa$		
	Grade 8.8 d $\leq$ 16mm, $Ftu = 800MPa$ d $>$ 16mm, $Ftu = 860MPa$ Grade 9.8, $Ftu = 900MPa$ Grade 10.9, $Ftu = 1040MPa$			Grade 12.9 $Ftu = 1200MPa$		
K_t	C_1	C_2	C_3	C_1	C_2	C_3
4.56	1.82	4.71	0.0	3.25	1.83	0.135
4.89	1.79	4.60	0.0	3.25	1.81	0.127
4.97	1.78	4.58	0.0	3.28	1.76	0.126
5.02	1.78	4.56	0.0	3.24	1.81	0.124

A study of bolt geometries suggests that the stress concentrations considered here could be applied to a range of bolt sizes as given in Table 5.

Table 5. Applicable thread sizes.

K_t	Coarse Thread Bolt Sizes
4.56	Up to and including M14
4.89	M16 and M18
4.97	M20
5.02	M22 to M36

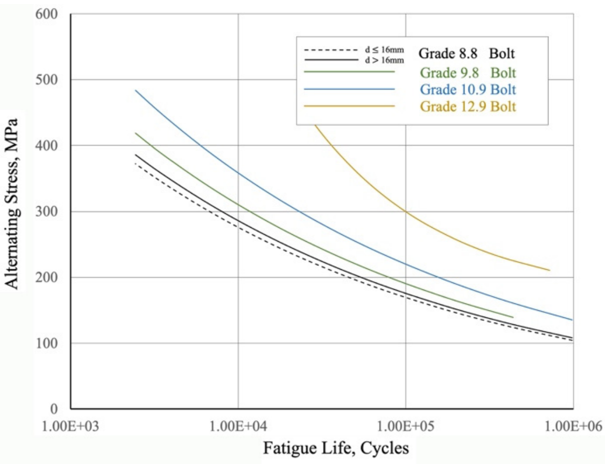

Figure 3. S-N curves for stress concentration $K_t = 4.56$

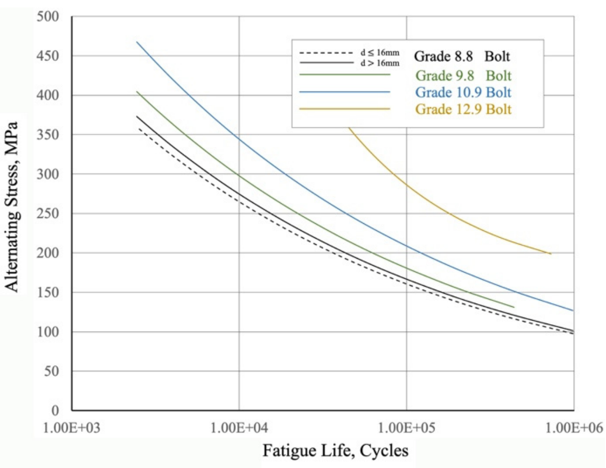

Figure 4. S-N curves for stress concentration $K_t = 4.89$

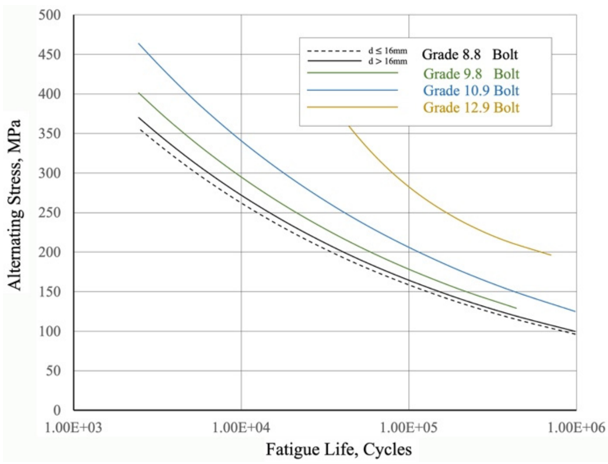

Figure 5. S-N curves for stress concentration $K_t = 4.97$

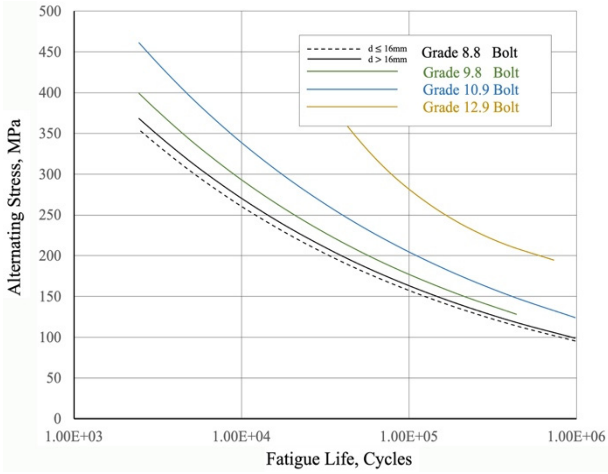

Figure 6. S-N curves for stress concentration $K_t = 5.02$

4. DAMAGE-EQUIVALENT STRESS

Equation (6) is applicable to S-N curves with a zero mean stress hence, for the fatigue analysis of bolts with a high mean stress the value for σ_{alt} used in the equation needs to represent a damage-equivalent stress. There are several methods of calculating a damage-equivalent stress, most notably, Goodman or Goodman-Haig diagram, Gerber, Soderberg and Smith-Watson-Topper. There is also a more recent damage-equivalent function proposed by Welch (2022) [10].

Again, since the calculated fatigue life is very sensitive to stress, the method of calculating the fatigue damage-equivalent stress needs to introduce the minimum of error. The high mean stresses associated with preloaded bolts results in 'corrections' for mean stress having to be made over a large increment. The method of determining the damage-equivalent stress has to be able to deal with these large increments. This requirement virtually rules out the use of both the Goodman and Soderberg methods. Hence, Smith-Watson-Topper is probably the most reliable of the established methods of calculating a damage-equivalent stress for preloaded bolts, and can be expressed as:

$$\sigma_{equ} = \sigma_{alt} \cdot \left(1 + \frac{\sigma_{mean}}{\sigma_{alt}}\right)^{\frac{1}{2}} \quad (7)$$

However, the more recent method proposed by Welch (2022) [10] appears to provide more consistent and improved accuracy over the Smith-Watson-Topper method. This method can be expressed as:

$$\sigma_{equ} = \sigma_{alt} \cdot \left(1 + \frac{a_2 \cdot (1+R)^{a_1}}{(K_t + 1)^{a_3}} \right) \tag{8}$$

where: a_1, a_2 and a_3 are given by:

$$a_1 = b_1 + b_2 \cdot K_t \cdot \left(\frac{Fty}{E} \right)^{b_3} \tag{9}$$

$$a_2 = b_4 + b_5 \cdot \left(\frac{Fty}{E} \right)^{b_6} \tag{10}$$

$$a_3 = b_7 + b_8 \cdot \left(\frac{Fty}{E} \right)^{b_9} \tag{11}$$

with:

$b_1 = 1.854$
$b_2 = 4.224 \times 10^{6}$
$b_3 = 3.260$

$b_4 = -1.015$
$b_5 = 38.120$
$b_6 = 0.635$

$b_7 = 1.038$
$b_8 = -2.032 \times 10^{-6}$
$b_9 = -2.485$

The stress ratio R is given by:

$$R = \frac{\sigma_{\min}}{\sigma_{\max}} \tag{12}$$

5. RESIDUAL STRESSES IN BOLTS

The method of manufacture for bolts is not prescribed in BS EN ISO 868-1 (2009) [7]. Most, if not all, bolt manufactures use thread rolling procedures to form bolt threads. These procedures result in high compressive residual stresses at the thread root, which is beneficial to fatigue performance. Thread rolling can be carried out either before or after heat treatment. Thread rolling after heat treatment requires higher operational loads than thread rolling before heat treatment. This results in higher power requirements and hence more energy con–sumption. Higher operational loads also lead to inc–reased wear and hence shorter working life for the thread rolling dies. Consequently, for most manufac–tures, thread rolling will presumably be carried out before heat treatment in order to minimise manufacturing costs.

As an observation, BS EN ISO 868-1 (2009) [7] quotes minimum tempering temperatures of 340ºC, 380ºC and 425ºC for various bolt grades and material chemical compositions. The work by de Souza et. al. (2021) [9] would suggest that quench hardening and then tempering at these minimum temperatures would induce a tensile residual stress component. However, thread rolling before heat treatment would induce a large compressive residual stress that would not be completely removed/relaxed by the post rolling heat treatment.

Compressive residual stresses will act to reduce the maximum and minimum stresses at the thread root. This will reduce the mean stress at the thread root, hence improving the fatigue performance of the bolt, but will have no effect on the stress range. In static analyses bolt stresses are usually calculated using the bolt nominal tensile stress area. The nominal stress area for a range of bolt sizes is given in a number of standards, including BS EN ISO 868-1 (2009) [7]. The area is calculated using the mean of the pitch diameter and the minor diameter. The S-N curves presented here are applicable to stresses based on the nett section therefore, the maximum, minimum and mean stresses are given by equation (13), (14) and (15) respectively:

$$\sigma_{\max} = \frac{F_{b.\max(n)}}{A_{core}} + \sigma_{res} \tag{13}$$

$$\sigma_{\min} = \frac{F_{b,\min(n)}}{A_{core}} + \sigma_{res} \tag{14}$$

$$\sigma_{mean} = \frac{F_{b,mean(n)}}{A_{core}} + \sigma_{res} \tag{15}$$

where $F_{b.max(n)}$, $F_{b.min(n)}$ and $F_{b.mean(n)}$ are the maximum, minimum and mean bolt loads on bolt ‘n’ and A_{core} is the core area of the bolt. In equations (13) to (15) the residual stress σ_{res} is positive for tensile stress and ne–gative for compressive stress. The core area is given by:

$$A_{core} = \frac{\pi \cdot D_s^2}{4} \tag{16}$$

where D_s is the minor diameter of the bolt thread.

An estimate of the residual stresses in the bolts used in the work by Marcelo et. al. (2011) [8] was made by iteration. Each bolt was considered in turn. A value of residual stress was assumed which was then used in equations (13) and (14), along with the maximum and minimum bolt loads given in reference [8], to calculate the maximum and minimum bolt stresses. These stresses were then used with equations (12) and (8) to calculate the damage-equivalent stress for the bolt. This stress was then used as the alternating stress σ_{alt} in equation (6) to calculate a predicted life for the bolt. This calculation process was repeated, using a range assumed residual stresses, until the calculated predicted life for the bolt matched the fatigue life of 10^6 cycles, as given in reference [8].

2.1 Thread Rolling After Heat Treatment

Reference [8] considered one set of bolts manufactured from SCM 435H wire, quench hardened and then tempered at 550ºC to give a tensile strength of 1154MPa. The bolt threads were then rolled after heat treatment. The residual stress for this set of bolts was estimated to be -740MPa.

The process of thread rolling has been simulated by Furukawa and Hagiwara (2014) [11] using 3D elastic-plastic Finite Element Methods. The heat treatment process, quench hardening and tempering, was not si–mulated within the model hence, the results are representative of thread rolling after heat treatment. The simulation used the material properties and geometry for Grade 8.8 M10 x 1.25 bolts. A compressive residual stress at an inner layer 30mm from the thread root was calculated to be -830MPa. This is of a similar mag–nitude to the residual stress estimated here for the M8 x 1.25 bolts considered by Marcelo et. al. (2011) [8].

The magnitude of the residual stress in a Grade 8.8 bolt (minimum tensile stress of 800MPa) as calculated by Furukawa and Hagiwara (2014) [11] was greater than the magnitude of residual stress, as estimated in this paper, for the bolts with a tensile strength of 1154MPa as tested by Marcelo et. al. (2011) [8]. This was consistent with what had been observed in the results of the work by de Souza et. al. (2021) [9].

2.2 Thread Rolling Before Heat Treatment

Reference [8] also considered two sets of bolts that were thread rolled before heat treatment and had a finale tensile strength (after heat treatment) close to the minimum requirements for Grade 10.9 bolts. One set was manufactured from AISI 4135 wire and the other from SCM 435H wire. Both sets were quench hardened and then tempered at 550ºC to give tensile strengths of 1025MPa for the bolts from AISI 4135 and 1051MPa for the bolts from SCM 435H. The residual stresses for these two set of bolts were estimated to be -679MPa for AISI 4135 and -649MPa for SCM 435H. The residual stress for a Grade 10.9 bolt with the minimum tensile stress specified in BS EN ISO 868-1 (2009) [7] (1040MPa) was then estimated to be -660MPa.

Similarly, reference [8] considered two sets of bolts that were thread rolled before heat treatment and had a finale tensile strength close to the minimum requirements for Grade 12.9 bolts. Again, one set was manufactured from AISI 4135 wire and the other from SCM 435H wire. Both sets were quench hardened and then tempered at 490ºC to give tensile strengths of 1211MPa for the bolts from AISI 4135 and 1233MPa for the bolts from SCM 435H. The residual stresses were estimated to be -459MPa for AISI 4135 and -465MPa for SCM 435H. The residual stress for a Grade 12.9 bolt with the minimum tensile stress specified in BS EN ISO 868-1 (2009) [7] (1220MPa) was then estimated to be -460MPa.

Note that although the work by Marcelo et. al. (2011) [8] was primarily for Grade 10.9 bolts statistically, 33% of Grade 10.9 bolts could possibly achieve a tensile strength of 1211MPa or more. Similarly, 24% could possibly achieve a tensile strength of 1233MPa or more. Hence, the tensile strengths of 1211MPa and 1233MPa referred to above both fall within the range of Grade 10.9 bolts.

Finally, reference [8] considered two sets of bolts that were thread rolled before heat treatment and had a finale tensile strength a little above the minimum requirements for Grade 10.9 bolts. Again, one set was manufactured from AISI 4135 wire and the other from SCM 435H wire. Both sets were quench hardened and then tempered at 520ºC to give tensile strengths of 1119MPa for the bolts from AISI 4135 and 1070MPa for the bolts from SCM 435H. The residual stresses were estimated to be -476MPa for AISI 4135 and -619MPa for SCM 435H.

The estimated residual stresses were plotted against tensile strength and are shown in Figure 7.

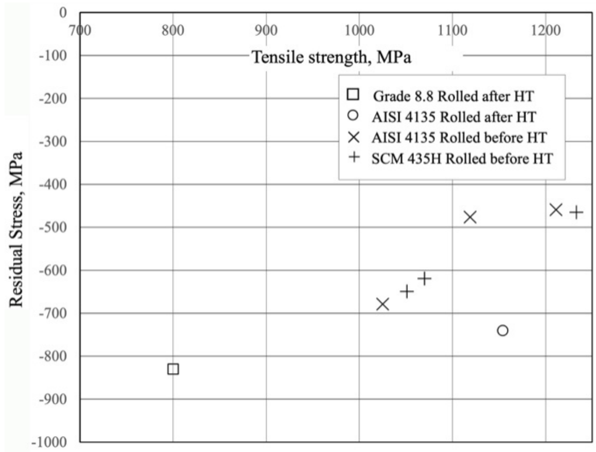

Figure 7. Residual stress vs Tensile strength

The of results shown in Figure 7 indicate that extra–polating to find residual stresses outside a range of tensile strength of, say, 1000MPa to 1250MPa would produce an unreliable result and should be avoided.

6. CONCLUSIONS AND RECOMMENDATIONS

In order to produce an accurate prediction of fatigue life, margins of safety, safety factors, partial safety factors or reserve factors must not be applied to the loads or calculated stresses used for the fatigue analysis.

It is more meaningful to calculate margins of safety, reserve factors or other means of expressing reliability based on predicted life rather than applied loads or stresses.

It was observed that, in general an increase in tensile strength results in improved fatigue properties. How–ever, if a carbon steel is at or close to its maximum achievable tensile strength its fatigue performance may be impaired.

S-N curves for Grade 8.8, Grade 9.8, Grade 10.9 and Grade 12.9 have been produced for stress concentrations of 4.56, 4,89, 4.97 and 5.05 and presented in the form of an equation. The stress concentration of 4.56 is relevant for bolt sizes up to M14. The stress concentration of 4.89 is relevant to M16 and M18 bolts and the stress concentration of 4.97 is relevant to M20 bolts. Similarly, the stress concentration of 5.02 is relevant to bolt sizes within the range M22 to M36.

The residual stress in Grade 12.9 bolts can be assu–med to be approximately -460MPa. This assump–tion is based on estimated residual stresses in bolts ma–nufac–tured from AISI 4135 and SCM 435H that had tensile strengths of 1211MPa and 1233MPa respec–tively.

The residual stress in Grade 10.9 bolts can be assumed to be approximately -660MPa. This was the average of the results for bolts manufactured from

AISI 4135 and SCM 435H that had tensile strengths of 1025MPa and 1051MPa respectively.

It was considered unreliable to extrapolate the results to determine a residual stress for Grade 8.8 and Grade 9.8 bolts. It was concluded that the residual stresses in these bolt grades would be compressive and with a higher magnitude than the residual stresses for Grade 10.9 bolts. It was recommended that the residual stresses in Grade 8.8 and Grade 9.8 should be assumed to be -680MPa. This was the residual stress associated with the bolts having lowest tensile considered, i.e. 1025MPa.

The maximum, minimum and mean stresses of the bolts should be calculated using the core area of the bolts, not the tensile area. The core area is defined by the minimum diameter of the bolt. These calculated stresses should also include the effect of the residual stresses.

The maximum and minimum stresses, which should include the effects of residual stresses, should be used to calculate the stress ratio and the damage-equivalent stress for each bolt. It is recommended that the damage-equivalent stress is calculated using the method proposed by Welch (2022) [10].

The damage-equivalent stress for each bolt can be used, along with the minimum tensile strength for the bolt grade, to calculate the bolt life. Bolt life can be calculated by means of a function that describes an appropriate S-N curve.

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NOMENCLATURE

A_b	Tensile area of each bolt
A_c	True contact area of the joint
A_{core}	Core area of a bolt
A_j	Total area of joint (Contact surface plus bolts)
a_1, a_2, a_3	Constants
b_1, b_2, b_3	Constants
b_4, b_5, b_6	Constants
b_7, b_8, b_9	Constants
C_1, C_2, C_3	Constants
D_s	Minor diameter of bolt thread (root diameter)
E	Young's modulus of elasticity
$F_{b(n)}$	Bolt load in bolt 'n'
$F_{b,max(n)}$	Maximum bolt load onbolt 'n'
$F_{b,mean(n)}$	Meanbolt load onbolt 'n'
$F_{b,min(n)}$	Minimum bolt load onbolt 'n'
F_p	Preload in each bolt
Ftu	Material ultimate tensile strength
Fty	Material yield/proof stress
F_z	External axial load in direction of 'z' axis
$I_{xx,j}$	Second Moment of Area of joint about 'x' axis
K_f	Fatigue stress concentration factor
$K_{f,D}$	Fatigue stress concentration factor of a 'donor' S-N curve
K_t	Elastic stress concentration factor
$K_{t,D}$	Elastic stress concentration factor of a 'donor' S-N curve
M_x	External moment acting about 'x' axis
N_b	Number of bolts in joint
N_{life}	Fatigue life – Number of cycles
P_f	Contact pressure at faying surface
P_p	Pressure at faying surface, preload pressure
q	Material notch sensitivity factor
R	Stress ratio
$S_{alt,D}$	Alternating stress of a 'donor' S-N curve
$S_{alt,0}$	Alternating stress of an unnotched S-N curve
$y_{(n)}$	Coordinate of bolt 'n' (from neutral axis of joint)

Greek symbols (Times New Roman 10 pt, bold, italic)

σ_{alt}	Alternating stress
σ_{equ}	Damage-Equivalent stress
σ_{max}	Maximum stress

σ_{mean}	Mean stress
σ_{min}	Minimum stress
σ_{res}	Residual stress

АНАЛИЗА ЗАМОРА ПРЕДНАПРЕГНУТИХ ВИЈЧАНИХ СПОЈЕВА

М. Велч

Овај чланак представља Велерове дијаграме, или S-N криве, за употребу у анализи замора вијака преднапрегнутих вијчаних спојева. Преднапрегнути вијци под цикличним оптерећењем имају велики средњи напон са малим наизменичним напоном. Ово је комбиновано са великом концентрацијом напрезања у корену навоја. Приказана метода анализе замора користи S-N криве са нултим средњим напрезањем.
Висок средњи напон који доживљавају завртњи се узима у обзир коришћењем функције за изра–чунавање напона еквивалентног оштећењу. Сматрало се да осетљивост на зарез модификује S-N криве за материјале са хемијским саставом обухваћеним спецификацијама материјала за вијке високе чврстоће. Ово је произвело S-N криве за концентрације напона релевантне за навоје вијака. Технике уклапања криве су коришћене да се ове криве изразе као функција еквивалентног напона и односа напона. Извршена је процена заосталих напона у вијцима произведеним ваљањем навоја. Рад даје практичну методу прорачуна заморног века.