A Method for the Stress and Fatigue Analysis of Bolted Joint Connections: together with Programmed Solution

Tonye K. Jack

Abstract*: *Often the weakest link in integral engineering equipment, bolted joint connections require proper attention and detailed analysis at the design stage for a fail safe operation in service. The analysis is often lengthy with several variables under consideration. A step-by-step guide, together with all required equations for evaluating a typical bolted joint connection is given. A computer programmed solution in Microsoft Excel ™ for such analysis is shown through a worked example.

Index Terms*: *Bolt and nut connection, bolted joint analysis, bolt fatigue, joint stresses, bolt preload.

I. INTRODUCTION

The literature on the analysis of bolt and nut connections is vast. It is also the subject of certain Country or Engineering institutional codes – such as Engineering Sciences Data Unit Guide (ESDU), American Society of Mechanical Engineers (ASME), British Standards (BS), German VDI, and others [1]-[6]. The major interest in all the standards is the question: Will the joint or connection fail in service? Failure in service here will imply a dynamic condition; as for example, effect of excitations on, a flowing oil and gas pipeline bolted flanged connection, the hinged end-cover of a pressurized heat exchanger vessel, hold-down foundation bolts of an engine, aged effect through environmental degradation and corrosion on the bolted connections of bridge frameworks or the chassis of a motor vehicle or marine vessel, street or other lamp posts, truss linkages of telecommunication masts, general equipment vibration effects resulting in bolt loosening. But, static failure is also a source for worry. Field experience shows that it is not unusual for bolts to break during tightening. What level of tightening or torque then is adequate? Can the applied tightening torque provide the required preload or initial tension to keep the joint together and avoid separation of the connection while in service?

II. ENGINEERING MATHEMATICAL RELATIONSHIPS

A. Sectional Area Relationships

Tensile Stress Area:

$$A_s = \frac{\pi D^2}{4} \qquad (1)$$

Cross-sectional Area of body:

$$A_b = \frac{\pi D_s^2}{4} \qquad (2)$$

Thread Lengths:

For L<125 mm, $L_t = 2D + 6$ (3)

For 125 mm<L<200 mm, $L_t = 2D + 12$ (4)

For L>200 mm, $L_t = 2D + 25$ (5)

Nominal body Length:

$$L_{bn} = L - L_t \qquad (6)$$

Effective body Length:

$$L_{beff} = L_{bn} + \left(B/2\right) \qquad (7)$$

Effective thread Length:

$$L_{teff} = L_g - L_{beff} + \left(N/2\right) \qquad (8)$$

Thread stripping through wear is a common bolt and nut connection failure concern. Thus, the length of engagement, *N*, as indicated in Fig. (1) is an important designer concern.

B. Stiffness Calculation Relationships

The principle upon which the stiffness is obtained in a bolt and nut connection is based on the deformation of an axially loaded bar guided by Hooke's law. The deformation, δ, for such cases is defined by (9):

$$\delta = \frac{FL}{AE} \qquad (9)$$

Hooke's Law:

$$k = \frac{F}{\delta} = \frac{AE}{L} \qquad (10)$$

Using a bolt and nut to tighten an engineering member, stretches the bolt and induces tension load in the bolt and compression in the clamped members. A bolt, from Hooke's *Elastic* law consideration is viewed as a series combination of a head/shank body section, and a threaded section. The stiffness of the body and threaded sections are thus obtained separately, and the effective total bolt stiffness, calculated through a series combination.

Stiffness of body section:

$$k_{be} = \frac{A_b E}{L_{beff}} \qquad (11)$$

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Stiffness of threaded section:

$$k_{te} = \frac{A_s E}{L_{teff}} \tag{12}$$

Stiffness of Clamped Member:

$$k_m = \frac{\pi E_m}{4L}\left\{\left[D_w + \left(L/10\right)\right]^2 - D^2\right\} \tag{13}$$

Total Stiffness of bolt:

$$k_b = \frac{(k_{be} k_{te})}{(k_{be} + k_{te})} \tag{14}$$

Total Joint Stiffness:

$$k_t = k_b + k_m \tag{15}$$

Stiffness Parameter of bolt:

$$Y_b = \frac{k_b}{k_t} \tag{16}$$

Stiffness Parameter of member:

$$Y_m = \frac{k_m}{k_t} \tag{17}$$

Resilience of bolt

$$R_k = \frac{1}{k_b} \tag{18}$$

Resilience of clamped member

$$R_m = \frac{1}{k_m} \tag{19}$$

C. Load Evaluation Relationships

Proposed Preload:

A bolt is often employed to connect two or more parts together. Preload or pretension determines the strength of the joint connection [7]. Recommended values for preload as suggested in the open literature vary. Shigley [8], [9] suggests a preload of 90 percent of the proof strength. Baumann [10] recommends a preload value of 70% - 80% of the static tensile strength. This for a safer condition is best taken as the yield strength of the bolt material. This is also backed by Brenner's [11] argument, that a bolt torque tightened to induce a preload does not develop as high a tensile strength as one statically loaded, the reason being the additional torsion component due to the torque. Brenner [11] further states that, because of the straight line relationship between increasing tensile load and bolt stretch up to the yield point of the bolt, a bolt torque tightened within its yield strength will be capable of developing the full rated tensile strength when subjected to additional load in excess of the preload. Thus,

$$P_L = y\sigma_{yp} A_s \tag{20}$$

Where, y, is the preload-yield factor.

Make-up Torque:

ESDU [2] provides a relation for make-up torque defined by equation (21),

$$T = P_L\left\{\left[\frac{(D+D_s)}{2}\right]\left(\frac{\mu_a}{2}\right) + \frac{p}{2\pi} + \frac{\left(D\mu_t/2\right)}{\cos\theta}\right\} \tag{21}$$

Where, p is the metric thread pitch.

Friction has a dominating influence in threaded joints, and exact values of the friction coefficients are not known [2]. While, μ_a can vary from 0.05-0.4, μ_t can vary from 0.05-0.25 [12]. For design estimation purposes, figures ranging from values between 0.15-0.2 can be assumed for both friction coefficients. [13]

Approximate Make-up Torque:

$$T_{approx.} = K.(P_L).D \tag{22}$$

The torque coefficient, K, varies for different bolt sizes. A table of values for different bolt sizes is available in [13]. The average value is $K \approx 0.2$.

The torque values obtained in (21) is compared with that obtained by (22).

III. ANALYSIS DURING TIGHTENING

A first step to analysis is the evaluation of the static state of the bolt during tightening to answer such questions as: Will the applied preload shear the bolt? In plain language, will the bolt break? Will the applied static preload sustain the connection under dynamic condition, i.e., operation in service? What is the static working stress? The reliability of the joint connection as judged through an index of reliability or factor of safety under static condition when considered alongside the bolt material type selected? The following relations help answer these questions:

Tension Load or tensile stress induced in bolt body (with tightening torque):

$$\sigma_{bt} = \frac{P_L}{A_b} \tag{23}$$

Torsional Shear Stress:

$$\tau = \frac{16T}{\pi D^3} \tag{24}$$

Principal Stress or Working Stress in bolt

$$\sigma_1 = \frac{\sigma_{bt}}{2} + \sqrt{\left(\frac{\sigma_{bt}}{2}\right)^2 + \tau^2} \tag{25}$$

IV. ANALYSIS AFTER TIGHTENING – EVALUATION IN SERVICE

Tension load in bolt:

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$$F_p = (Y_p F_{ext}) + P_L \tag{26}$$

Maximum tensile stress in bolt in service

$$\sigma_{ts} = \frac{F_p}{A_b} \tag{27}$$

Compressive load induced in member

$$F_{cl} = (Y_m F_{ext}) - P_L \tag{28}$$

Minimum Preload to prevent loss in compression

$$P_{L\min} = Y_m F_{ext} \tag{29}$$

Joint Adequacy for Fatigue loading:

Equivalent Stress in Shank section

$$\sigma_{eqv.} = \sigma_{ts} + K_s \left({}^{\sigma_{yp}}\!/_{\sigma_e} \right) \sigma_b \tag{30}$$

Equivalent Stress in threaded section

$$\sigma_{eqv.} = \sigma_{ts} + K_t \left({}^{\sigma_{yp}}\!/_{\sigma_e} \right) \sigma_b \tag{31}$$

Stress concentration factors, K_s, K_t, are available from Peterson [14]. Area reduction or transitions such as bolt head fillet or chamfer, the start of the first thread and fillets in the plane of the nut are points of stress concentration [9].
Endurance or fatigue limit is derived experimentally. When a value is not available, the practice is to make an estimate as a percentage of the yield strength [15].
Factor of safety for bending/fatigue:
Fatigue assessment is based on the Soderberg criterion and the calculation of a safety factor or reliability index based on fatigue is:

$$n_f = \frac{\sigma_{yp}}{\sigma_{eqv.}} \tag{32}$$

Angular Turn of the Nut:

$$C_L \frac{360.P_L}{k_t p} \tag{33}$$

Moment of Inertia through bolt axis:

$$I_y = \left(\frac{\pi D_s^2}{4} \right) y_{NA} \tag{34}$$

Maximum Bending Stress:

$$\sigma_b = \frac{M y_{NA}}{I_E} \tag{35}$$

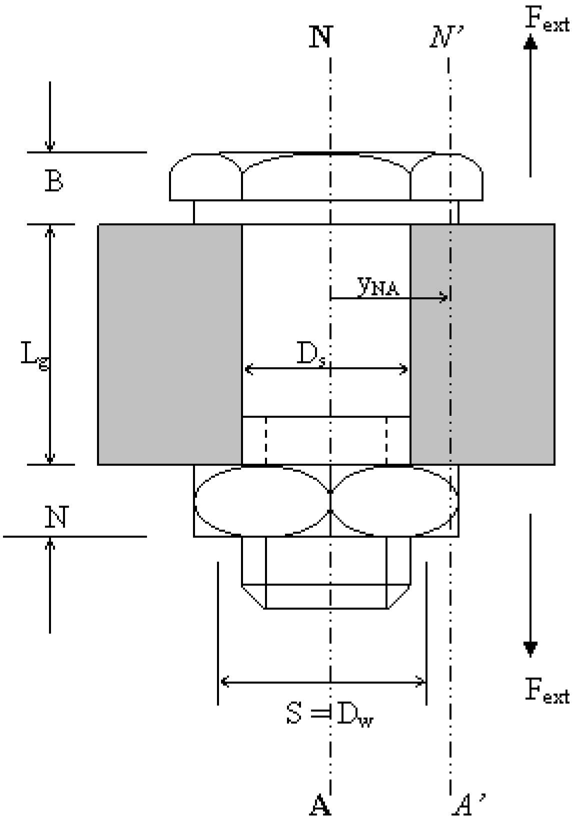

Fig.(1): - Basic Mathematical model used to formulate relationships

NA = Neutral Axis
N'A' = Transfer Axis
Note:

$$y_{NA} = \frac{D_s}{2} + \frac{(D_w - D_s)}{2}$$

V. NUMERICAL EXAMPLE

If the bolt in Fig. (1) is an ISO M24 bolt subjected to combined bending and tensile separating forces:

a) Conduct a detailed stress evaluation analysis of the joint connection?

The bolt and Nut details are:
Bolt Diameter, D_s=24.84 mm
Height of Bolt Head, B=15.9 mm
Nut Thickness, N=18.84 mm
Nut Width across flat, S=36 mm
The basic bolt length, L=120 mm
Clamping length = 100 mm
E_{bolt}=E_{member}=2.07 x 1010 Pa
Take, σ_e=2.75 x 108 Pa
σ_{yp}=6.90 x 108 Pa
Separating Load, F_{ext} = 10,000 N
Bending Moment, M=130 Nm

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Input Specification

Dimensional Input Data

Nominal Diameter D (mm)	24.0000
Body Diameter Ds (mm)	24.8400
Height of Head B (mm)	15.9000
Threads pitch (mm)	3.0000
Nut Thickness N (mm)	18.8400
Nut Seat Diameter Dw (mm)	36.0000
Nut width across flat (mm)	36.0000
Basic Body length L: For L<125mm	120.0000
Half Thread Flank angle in DEGREES	30.0000
Grip Length Lg (mm)	100.0000

Loading Consideration

YIELD STRENGTH (Pa)	6.90E+08
MODULUS OF ELASTICITY of bolt material:E (Pa)	2.07E+10
MODULUS OF ELASTICITYof clamped member:(Pa)	2.07E+10
FATIGUE ENDURANCE LIMIT	2.75E+08
PRELOAD-YIELD FACTOR	0.75
FRICTION COEFFICIENT :	
between angular bearing face	0.200
between thread mating face	0.200
Stress concentration factor - head	1.5
Stress concentration factor- thread	3.0
EXTERNAL LOAD Fa (N)	10000.00
BENDING MOMENT, M (N.m)	130.000

Output Data

Thread Pitch (mm)	3.000		
Tensile stress Area As (m^2)	4.52E-04	Cross sectional Area of Body Ab (m^2)	4.85E-04
Thread Length Lt: for L <125 mm	54	Nominal body Length: For L<125 mm	66.0000
Effective Body length: For L<125 mm	73.95	Effective Thread Length:ForL< 125 mm	43.42
STIFFNESS CALCULATIONS			
Stiffness of body section: Kbe	1.36E+08	Stiffness of Threaded section: Kte	2.16E+08
Total Stiffness of bolt: Kb	8.33E+07	Clamping Stiffness: Kc	2.50E+08
Total Joint stiffness: K	3.34E+08		
Stiffness Parameter for Bolt Yb:	0.250	Stiffness Parameter for member Ym	0.750
PRELOAD EVALUATION			
PROPOSED PRELOAD (N)	234111.48		
MAKE-UP TORQUE in (N. m)	1472.74	APPROXIMATE MAKE UP TORQUE (N.m)	1123.74
EFFECTIVE MAKE-UP TORQUE (N. m)	736.37	APPROX. EFFECTIVE MAKE-UP (N.m)	561.87
BOLTED JOINT ANALYSIS DURING TIGHTENING			
Tension induced in bolt body (N/m^2)	4.83E+08	Torsional shear stress (N/m^2)	2.71E+08
Maximum Principal stress (N/m^2)	6.05E+08	Factor of Safety during tightening	1.14
BOLTED JOINT ANALYSIS AFTER TIGHTENING : preload plus externally applied load			
Tension load induced in bolt (N) :	236607.39	Max. tensile Stress in bolt in service (N/m^2)	4.88E+08
Compressive load induced -member (N)	-226607.39	Factor of Safety in service :	1.41
Min. Preload to prevent loss of compression (N)	7504.09	Minimum Tightening Torque: (N- m)	47.21
Angular turn of the nut	337.36		
JOINT BENDING/FATIGUE CONSIDERATION:			
Moment of Inertia through bolt axis (m^4)	1.87E-08	Second moment of area (m^4)	1.57E-07
Equivalent Moment of inertia (m^4)	1.76E-07	Distance from N.A to transfer axis (m)	0.0180
Maximum Bending Stress (N/m^2)	1.33E+07		
EQUIVALENT STRESS in shank section (N/m^2)	5.38E+08	EQUIVALENT STRESS in threaded section (N/m^2)	5.88E+08
Factor of Safety for Bending Fatigue Condition:			
in shank section	1.28	in threaded section	1.17

Fig.(2) – Microsoft Excel Programmed Solution for Example with a Preload-Yield Factor = 0.75

Input Specification

Dimensional Input Data

Nominal Diameter D (mm)	24.0000
Body Diameter Ds (mm)	24.8400
Height of Head B (mm)	15.9000
Threads pitch (mm)	3.0000
Nut Thickness N (mm)	18.8400
Nut Seat Diameter Dw (mm)	36.0000
Nut width across flat (mm)	36.0000
Basic Body length L: For L<125mm	120.0000
Half Thread Flank angle in DEGREES	30.0000
Grip Length Lg (mm)	100.0000

Loading Consideration

YIELD STRENGTH (Pa)	6.90E+08
MODULUS OF ELASTICITY of bolt material:E (Pa)	2.07E+10
MODULUS OF ELASTICITYof clamped member:(Pa)	2.07E+10
FATIGUE ENDURANCE LIMIT	2.75E+08
PRELOAD-YIELD FACTOR	0.50
FRICTION COEFFICIENT :	
between angular bearing face	0.200
between thread mating face	0.200
Stress concentration factor - head	1.5
Stress concentration factor- thread	3.0
EXTERNAL LOAD Fa (N)	10000.00
BENDING MOMENT, M (N.m)	130.000

Output Data

Thread Pitch (mm)	3.000		
Tensile stress Area As (m^2)	4.52E-04	Cross sectional Area of Body Ab (m^2)	4.85E-04
Thread Length Lt: for L <125 mm	54	Nominal body Length: For L<125 mm	66.0000
Effective Body length: For L<125 mm	73.95	Effective Thread Length:ForL< 125 mm	43.42
STIFFNESS CALCULATIONS			
Stiffness of body section: Kbe	1.36E+08	Stiffness of Threaded section: Kte	2.16E+08
Total Stiffness of bolt: Kb	8.33E+07	Clamping Stiffness: Kc	2.50E+08
Total Joint stiffness: K	3.34E+08		
Stiffness Parameter for Bolt Yb:	0.250	Stiffness Parameter for member Ym	0.750
PRELOAD EVALUATION			
PROPOSED PRELOAD (N)	156074.32		
MAKE-UP TORQUE in (N. m)	981.82	APPROXIMATE MAKE UP TORQUE (N.m)	749.16
EFFECTIVE MAKE-UP TORQUE (N. m)	490.91	APPROX. EFFECTIVE MAKE-UP (N.m)	374.58
BOLTED JOINT ANALYSIS DURING TIGHTENING			
Tension induced in bolt body (N/m^2)	3.22E+08	Torsional shear stress (N/m^2)	1.81E+08
Maximum Principal stress (N/m^2)	4.03E+08	Factor of Safety during tightening	1.71
BOLTED JOINT ANALYSIS AFTER TIGHTENING : preload plus externally applied load			
Tension load induced in bolt (N) :	158570.23	Max. tensile Stress in bolt in service (N/m^2)	3.27E+08
Compressive load induced -member (N)	-148570.23	Factor of Safety in service :	2.11
Min. Preload to prevent loss of compression (N)	7504.09	Minimum Tightening Torque: (N- m)	47.21
Angular turn of the nut	224.91		
JOINT BENDING/FATIGUE CONSIDERATION:			
Moment of Inertia through bolt axis (m^4)	1.87E-08	Second moment of area (m^4)	1.57E-07
Equivalent Moment of inertia (m^4)	1.76E-07	Distance from N.A to transfer axis (m)	0.0180
Maximum Bending Stress (N/m^2)	1.33E+07		
EQUIVALENT STRESS in shank section (N/m^2)	3.77E+08	EQUIVALENT STRESS in threaded section (N/m^2)	4.27E+08
Factor of Safety for Bending Fatigue Condition:			
in shank section	1.83	in threaded section	1.61

Fig. (3) - Microsoft Excel Programmed Solution for Example with a Preload-Yield Factor = 0.50

Input Specification

Dimensional Input Data

Nominal Diameter D (mm)	24.0000
Body Diameter Ds (mm)	24.8400
Height of Head B (mm)	15.9000
Threads pitch (mm)	3.0000
Nut Thickness N (mm)	18.8400
Nut Seat Diameter Dw (mm)	36.0000
Nut width across flat (mm)	36.0000
Basic Body length L: For L<125mm	120.0000
Half Thread Flank angle in DEGREES	30.0000
Grip Length Lg (mm)	100.0000

Loading Consideration

YIELD STRENGTH (Pa)	6.90E+08
MODULUS OF ELASTICITY of bolt material:E (Pa)	2.07E+10
MODULUS OF ELASTICITYof clamped member:(Pa)	2.07E+10
FATIGUE ENDURANCE LIMIT	2.75E+08
PRELOAD-YIELD FACTOR	0.70
FRICTION COEFFICIENT :	
between angular bearing face	0.200
between thread mating face	0.200
Stress concentration factor - head	1.5
Stress concentration factor- thread	3.0
EXTERNAL LOAD Fa (N)	10000.00
BENDING MOMENT, M (N.m)	130.000

Output Data

Thread Pitch (mm)	3.000		
Tensile stress Area As (m^2)	4.52E-04	Cross sectional Area of Body Ab (m^2)	4.85E-04
Thread Length Lt: for L <125 mm	54	Nominal body Length: For L<125 mm	66.0000
Effective Body length: For L<125 mm	73.95	Effective Thread Length:ForL< 125 mm	43.42
STIFFNESS CALCULATIONS			
Stiffness of body section: Kbe	1.36E+08	Stiffness of Threaded section: Kte	2.16E+08
Total Stiffness of bolt: Kb	8.33E+07	Clamping Stiffness: Kc	2.50E+08
Total Joint stiffness: K	3.34E+08		
Stiffness Parameter for Bolt Yb:	0.250	Stiffness Parameter for member Ym	0.750
PRELOAD EVALUATION			
PROPOSED PRELOAD (N)	218504.05		
MAKE-UP TORQUE in (N. m)	1374.55	APPROXIMATE MAKE UP TORQUE (N.m)	1048.82
EFFECTIVE MAKE-UP TORQUE (N. m)	687.28	APPROX. EFFECTIVE MAKE-UP (N.m)	524.41
BOLTED JOINT ANALYSIS DURING TIGHTENING			
Tension induced in bolt body (N/m^2)	4.51E+08	Torsional shear stress (N/m^2)	2.53E+08
Maximum Principal stress (N/m^2)	5.64E+08	Factor of Safety during tightening	1.22
BOLTED JOINT ANALYSIS AFTER TIGHTENING : preload plus externally applied load			
Tension load induced in bolt (N) :	220999.96	Max. tensile Stress in bolt in service (N/m^2)	4.56E+08
Compressive load induced -member (N)	-210999.96	Factor of Safety in service :	1.51
Min. Preload to prevent loss of compression (N)	7504.09	Minimum Tightening Torque: (N- m)	47.21
Angular turn of the nut	314.87		
JOINT BENDING/FATIGUE CONSIDERATION:			
Moment of Inertia through bolt axis (m^4)	1.87E-08	Second moment of area (m^4)	1.57E-07
Equivalent Moment of inertia (m^4)	1.76E-07	Distance from N.A to transfer axis (m)	0.0180
Maximum Bending Stress (N/m^2)	1.33E+07		
EQUIVALENT STRESS in shank section (N/m^2)	5.06E+08	EQUIVALENT STRESS in threaded section (N/m^2)	5.56E+08
Factor of Safety for Bending Fatigue Condition:			
in shank section	1.36	in threaded section	1.24

Fig. (4) - Microsoft Excel Programmed Solution for Example with a Preload-Yield Factor = 0.70

VI. CONCLUSION

Figs. (2) and (3) and (4) shows that changing the preload value through altering the preload-to-yield factor, significantly influences the degree of reliability of the joint. The question of an adequate preload is thus best answered by conducting a what-if type analysis with the program. This is also in line with Aaronson's [1] design check for static load condition:

Preload ≤ 70% of load at yield point

Design check for Fatigue loading condition:

Bolt working load or equivalent stress ≤ load at yield point.

Nomenclature

A_s Tensile Stress Area, mm^2
A_b Cross-Sectional area of body, mm^2
B Height of Head, mm
C_L Angular Turn of the Nut, mm
D Nominal Bolt Diameter, mm
D_s Body Diameter, mm
E Elastic Modulus, N/mm^2
F_{ext} External Load, N
$I_{E.}$ Equivalent Moment of Area, mm^4
k_b Stiffness of Bolt, N/mm
k_m Stiffness of Clamped member, N/mm
K_t Stress Concentration factor for Thread
K_s Stress Concentration factor for Shank section
L_g Clamping or Grip Length, mm
M Bending Moment, Nm
n_f Factor of Safety
N Nut Thickness, mm
p Pitch of threads, mm
P_L Nominal Preload, N
P_{Lmin} Minimum Preload, N
R_b Resilience of bolt, mm/N

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R_m Resilience of clamped member, mm/N
S Nut Width across flats, mm
T Tightening Torque, Nm
y Preload-Yield factor
y_{NA} Distance from Neutral axis, mm
Y_b Stiffness Parameter of Bolt
Y_m Stiffness Parameter of Clamped Member

Greek Letters

θ Half thread flank angle, degrees
σ_{yp} Yield Strength, N/mm^2
σ_e Fatigue Strength, N/mm^2
$\sigma_{eq.}$ Equivalent Stress, N/mm^2
σ_b Bending Stress, N/mm^2
σ_t Tension Induced in Bolt, N/mm^2
τ_t Torsional Shear Stress, N/mm^2
σ_1 Maximum Principal Stress, N/mm^2
σ_c Compressive Load Induced, N/mm^2
μ_a Friction coefficient between angular bearing faces
μ_t Friction coefficient between thread mating surfaces

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