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Torque Tightening of Air Cooler Header Box Plugs

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ABSTRACT: *In a standard box header configuration, holes must be drilled and tapped in the plug sheet opposite each tube to allow maintenance of the tubes. A plug, normally a shoulder type plug with straight-threaded shanks and gasket, is threaded into each hole to seal under pressure, but allow access when required. Some operators specify two-part plugs with gasket compressor. The plugs are installed by means of a controlled torque wrench. Both air-cooled heat exchanger standards API 661 [1] and ISO 13706 [2] do not provide recommended tightening torque values for the plugs. Moreover, no reference is made to a recognized reference that makes it possible to determine the required torque. Practice shows that there is a need for a procedure that enables the engineer to determine in a prudent manner the tightening torque of the plugs based on current insights and proven practice. This article is intended to generate awareness and provide an impetus to tackle this problem.*

KEYWORDS: *plug sheet, shoulder plugs, two-part plugs, gasket compressor, tightening torque.*

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I. INTRODUCTION

It is crucial that the plug load required to seal the joint has been calculated using a reliable method and is known prior to installing the plug assembly. The principle of a plugged joint is based on the plug delivering sufficient joint compression and gasket seating stress to withstand maximum service pressure. For integrity a minimum level of operational gasket seating stress must be maintained throughout service, therefore the plug load /compression target should allow for a variety of uncertainties with regard to loading , tolerances of components and tools used. To establish the required torque, the plug load must first be determined. For this purpose, an algorithm has been developed that is derived from ASME BPVC Section VIII - Division 1, Appendix 2 [3]. This algorithm is included in section II. Once the normative plug load has been determined, it must be converted to the required tightening moment. There are several approaches for the conversion from plug load to tightening torque that will be included in section III.

The scope will be limited to the following plug sizes:

1"- 12 - UNF	**1 1/8" - 12 - UNF**	**1 3/8" - 12 - UNF**	**1 1/2" - 12 - UNF**

It should be noted that both API 661[1] and ISO 13706 [2] emphasize that plugs shall be shoulder type with straight-threaded shanks. This article will therefore concentrate on this type of plug.

II. ALGORITHM for PLUG LOAD

Equations and notations are consistent with ASME BPVC Section VIII - Division 1; Appendix 2 [3]

The required plug load for the operating condition:

$$\boxed{W_{m1} = H + H_p = \pi/4\, G^2 P + (2b \times \pi\, G\, m\, P) = \pi\, G\, P\, (G/4 + 2\, b\, m)}$$

The minimum required plug load for gasket seating condition:

$$\boxed{W_{m2} = \pi\, b\, G\, y}$$

The required plug load for the hydrostatic test condition:

$$\mathbf{W_{m1t} = \pi/4\, G^2 P_t + (2b \times \pi\, G\, m\, P_t) = \pi\, G\, P_t\, (G/4 + 2\, b\, m)}$$

Where:
b = effective gasket or joint- contact- surface seating width. (mm)
G = diameter at location of gasket load reaction. (mm)
H = total hydrostatic end force = $\pi/4\, G^2 P$ (N)
H_p = total joint- contact surface compression load = $2b \times \pi\, G\, m\, P$ (N)
m = gasket factor. (-)
P = internal design pressure. (MPa)
P_t = hydrostatic test pressure. (MPa)
W_{m1} = minimum required bolt load for the operating conditions. (N)
W_{m2} = minimum required bolt load for gasket seating. (N)
y = gasket or joint- contact- surface unit seating load. (MPa)
$F_{plug\ load}$ = max [W_{m1}; W_{m2} or W_{m1t}] = Assembly Preload (N)

III. CONVERSIONS from PLUG LOAD to TIGHTENING TORQUE

In the literature, codes and standards much attention has been paid to converting the tensile force in a bolt connection to a tightening torque. It is therefore obvious to make optimal use of this in the case of plugs. First of all an overview is given of the most common formulas used for bolted joints.

Overview bolt tightening formulas

Source	**Formula for Tightening Torque**
Motosh [4]	$T_{in} = F_P\,(P/2\pi + \mu_t.\, r_t / \cos\beta + \mu_n.r_n)$
VDI 2230 [5]	$M_A = F_V\,(0.159P + 0.578.d_2.\mu_G + 0.5\, D_{Km}.\mu_K)$
BS EN ISO 16047 [6]	$T = F\,[0.5(P + 1.154.\pi.\mu_{th}.d_2) / (\pi - 1.154.\mu_{th}.P/d_2) + \mu_b(D_o + d_h)/4]$
EN 1591-1 [7]	$M_{t,norm} = F_{B0nom}\,[p_t/(2\pi) + \mu_t.d_t / (2.\cos\alpha) + \mu_n.d_n/2]$

Where:

Motosh	**VDI 2230**	**BS EN ISO 16047**	**EN 1591-1**	**Discription**
T_{in}	M_A	T	$M_{t,norm}$	Torque
F_P	F_V	F	F_{B0nom}	Preload or tension
P	P	P	p_t	Thread pitch
r_n	$0.5\, D_{Km}$	$(D_o + d_h)/4$	$d_n/2$	Effective radius of head contact
μ_n	μ_K	μ_b	μ_n	Coefficient of friction under head
μ_t	μ_G	μ_{th}	μ_t	Coefficient of friction in threads
r_t	$d_2/2$	$d_2/2$	$d_t/2$	Effective radius of thread contact (half the thread pitch diameter)
β			α	Half-angle of thread form (30° for UNF thread)

The most practical and commonly accepted equation for describing the tightening process is shown below and can be found in many authoritative sources on fastening.

$$T = F\,[\,P/2\pi + \mu_T R_T/\cos\alpha + \mu_B R_B\,]$$

Where:
T = Tightening torque (Nmm)
F = Tensile force generated in the bolt as a result of tightening (N)
P = Thread pitch (mm)
μ_T = Coefficient of friction between the internal and external threads (-)
R_T = Radius at which frictional force in the threads is assumed to act (mm)
α = Thread flank angle (30 ° for the common 60° thread form) (-)
μ_B = Coefficient of friction between bolt head and the bearing surface under the bolt head (°)
R_B = Radius at which frictional force between the fastener and bearing surface is assumed to act (mm)

This equations captures the major variables and effects of the tightening process. Although other equations encompass additional variables, these variables have relatively minor effects in most tightening situations.
Moreover it is common practice to mathematically separate the equation for T and show the three components of total torque as:

$T_P = F\ [\ P\ /\ 2\pi]$

$T_T = F\ [\mu_T R_T\ /\ \cos\alpha]$

$T_B = F\ [\mu_B R_B]$

Where:

$\mathbf{T_P}$ is the pitch torque, the component of torque that creates tensile force in the bolt through the wedging action of the threads.

$\mathbf{T_T}$ is the thread torque, the component of input torque that must overcome friction between the internal and external threads as they slide across each other.

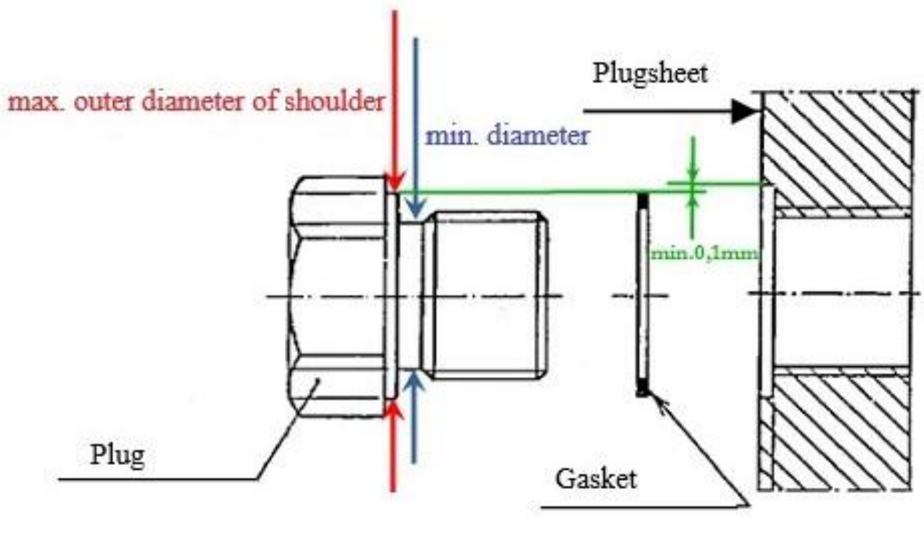
$\mathbf{T_B}$ is the bearing surface torque, the component of torque that must overcome friction between the fastener and bearing surface.

We can easily see that thread torque and bearing surface torque are directly proportional to their respective coefficients of friction. We then conclude with an equation for torque in terms of its three components:

$T_I = T_P + T_T + T_B$

Since there is no physical contact between the plug head and the plug sheet, there is no head friction. However, for the shoulder type plug there is physical contact between the shoulder and the gasket causing friction when tightening the plug. Therefore, the term T_B can be used as such as a contribution to the tightening torque .

The image and figure below shows a typical shoulder plug configuration for an air cooled heat exchanger.



Now the expression for the plug tightening torque becomes:

$T = T_P + T_T + T_B = F [P / 2\pi] + F [\mu_T R_T / \cos\alpha] + F [0.5 \mu_B G]$
With:
$1/2\pi = 0.16$, $\mu_T = 0.13$ (friction coefficient for Molykote 1000 anti-seize lubricant), $R_T = 0.5 d_2$, G = the mean gasket diameter and $\mu_B = 0.15$ (face friction coefficient for solid flat metallic gaskets). The tightening torque formula becomes:

$$\mathbf{T = F [0.16 P + 0.075 (d_2 + G)]}$$

Elaborated case
Data

Plug Size	1½ " 12-UNF
Design Pressure (MPa)	13
Test Pressure (MPa)	20.38
Design Temperature (^{0} C)	170
Plug Material	UNS S31803 (Duplex)
Yield Strength Plug Material @ Design resp. room temperature (MPa)	362.8 / 450
Header Material	UNS S31803 (Duplex)
Yield Strength Header Material @ Design resp. room temperature (MPa)	362.8 / 450
Gasket Ring Material (Solid Flat Metal)	Monel 400
Gasket Factor, m (-)	6.0
Gasket Seating Stress , y (MPa)	150
OD Gasket (mm)	45.9
ID Gasket (mm)	39.3
Gasket Thickness (mm)	1.5
Engaged Thread Length (mm)	32
Plug Tensile Stress Area (mm^2)	1020.0
Plug Root Area (mm^2)	981.36
Lubricant	Molykote 1000

Determination of plug loads
Effective Gasket Seating Width, b
$b = b_o$, when $b_o \leq 1/4$ in. (6 mm); $b = 2.52\sqrt{b_o}$, when bo > 1/4 in. (6 mm)
N = (OD Gasket - ID Gasket)/2 = (45.9 – 39.3)/2 = 3.3 mm
$b_o = N/2 = 3.3/2 = 1.65$ mm ; $b = b_o = 1.65$mm
G = (OD gasket – 2.b) = (45.9 – 2 x 1.65) = 42.6 mm
$W_{m1} = \pi G P (G/4 + 2 b m) = \pi \times 42.6 \times 13 (42.6/4 + 2\times 1.65 \times 6) = 52977$ N
$W_{m2} = \pi b G y = \pi \times 1.65 \times 42.6 \times 150 = 33123$ N
$W_{m1t} = \pi G P_t (G/4 + 2 b m) = \pi \times 42.6 \times 20.38 (42.6/4 + 2 \times 1.65 \times 6) = 83052$ N
Associated plug stress based on root area of the plug: 83052/981.36 = 84.6 MPa
Ratio of yield strength plug material / plug stress = 362.8/84.6 = 4.2 (@ 170°C
Tightening torque formula: $T = F [0.16 P + 0.075 (d_2 + G)]$
With $F = W_{m1t} = 83052$ N = max $[W_{m1;} W_{m2;} W_{m1t}]$, P = 25.4/12 = 2.1167 mm, d_2 = 36.72586 mm according ASME B1.1 - Table 7) and G = 42.6 mm we arrive at:
T = 83052 [0.16 x 2.1167 + 0.075 (36.72586 + 42.6)] = 522240 Nmm ≈522 Nm
Contributions of components to the total torque T

$T_P = F [P / 2\pi]$	$T_T = F [\mu_T R_T / \cos\alpha]$	$T_B = F [0.5 \mu_B G]$
5.4%	43.8%	50.8%

The percentages can be divided as follows: 5.4% due to inclination of the thread helix , 43.8% due to friction between threads and 50.8% due to friction under the shoulder of the plug.
In order to ensure that thread stripping will not occur, verification of the thread engagement length should be verified against the applicable design code.

K-factor approach

The torque applied to tighten a bolt or plug and the load subsequently induced into it, are largely dependent upon the friction present between the rotating surfaces on the fastener. Changes in the friction conditions over time will affect the torque to rotate the bolt and make this approach to checking the tightness of bolts problematic.

The torque coefficient is sometimes used in threaded fastener torque calculations. The torque coefficient K is a factor used to represent the fastener friction conditions and is used in, and defined by, the equation: T = F d K , where F is the clamp force provided by a tightening torque T, applied to a threaded fastener of diameter d and K the torque coefficient. The torque coefficient K is often referred to as a nut factor and usually ranges between 0.1 and 0.35. The primary advantage of the torque coefficient approach relative to the full torque-tension equation, which uses the coefficient of friction, is that it is much simpler to apply. The primary disadvantage is that it is applicable to a particular type and pitch of fastener and strictly to a particular thread size. In spite of these issues, it is very widely used to determine the torque to be applied to a fastener. A key assumption in the approach is that the friction in the tightening direction is the same as in the un-tightening direction. A series of tests were conducted on a range of bolt sizes, types and lubrication conditions to assess the efficacy of the method under different circumstances.
Setting F = W_{m1t} = 83052 N, d = nominal plug diameter = 38.1 mm and K = 0.2 (typical and most often used value) we will arrive at: T = 83052 x 38.1 x 0.2 = 632856 Nmm ≈ 633 Nm

Alternative approach based on gasket stress

Gasket stress is a term commonly used to describe the unit load on its surface. It is one of the most important parameters of a bolted joint because it directly impacts the ability of the gasket to seal. A hard metal gasket require a much higher stress than a soft gasket. Shoulder type plugs are mainly used in combination with flat ring solid metal gaskets thus require a quite high gasket seating stress to ensure sealing performance.

Target gasket stress (Sg_T) is the load that allows the gasket, as well as the entire joint, to operate at optimal performance and sealability. Additionally, the installation stress creates a preload in the joint that compensates for overall bolted joint relaxation after installation and during operation for the service life of the joint (with consideration given to joint integrity). ASME PCC-1 [8] recommends that the target stress should be as high as possible; “The target gasket stress should be selected to be towards the upper end of the acceptable gasket stress range, as this will give the most amount of buffer against joint leakage.”

The simplest method of selecting the target gasket stress is to calculate the available compressive stress at the maximum bolt stress. As long as the available gasket stress at maximum bolt stress is below the maximum gasket stress (or crush strength of the gasket) and above the minimum recommended gasket stress for operating conditions, that can be the target stress. Clause O-3 of ASME PCC-1[8] outline this simple approach and will be further elaborated for the subject case.

Step # 1 Determining the Appropriate Bolt Stress

The appropriate bolt stress for a range of typical joint configurations may be determined using the following equation:
$Sb_{sel} = Sg_T \text{ x } A_g/A_b$
Where:
Sb_{sel} = selected assembly bolt stress, MPa
Sg_T = target assembly gasket stress, MPa
A_g = gasket area $[\pi/4\ (G_{O.D}{}^{2} - G_{I.D.}{}^{2})]$, mm^2
A_b = bolt root area, mm^2
$G_{I.D.}$, $G_{O.D}$ = gasket sealing element inner/outer diameter, mm
$A_g = [\pi/4(45.9^2 - 39.3^2)] = 441.645$ mm^2
$A_b = 981.36$ mm^2
$Sg_T = 500$ MPa

$Sb_{sel} = Sg_T \text{ x } A_g/A_b = 500 \text{ x } 441.645/981.36 = 225$ MPa

Corresponds to 50% yield strength of plug material.

Step # 2 Determining the Assembly Torque
Assembly torque: $T_b = Sb_{sel} \times K \times A_b \times \phi_b / 1\,000$ (Nm)

Where:
K = nut factor (-)
ϕ_b = bolt diameter (mm)

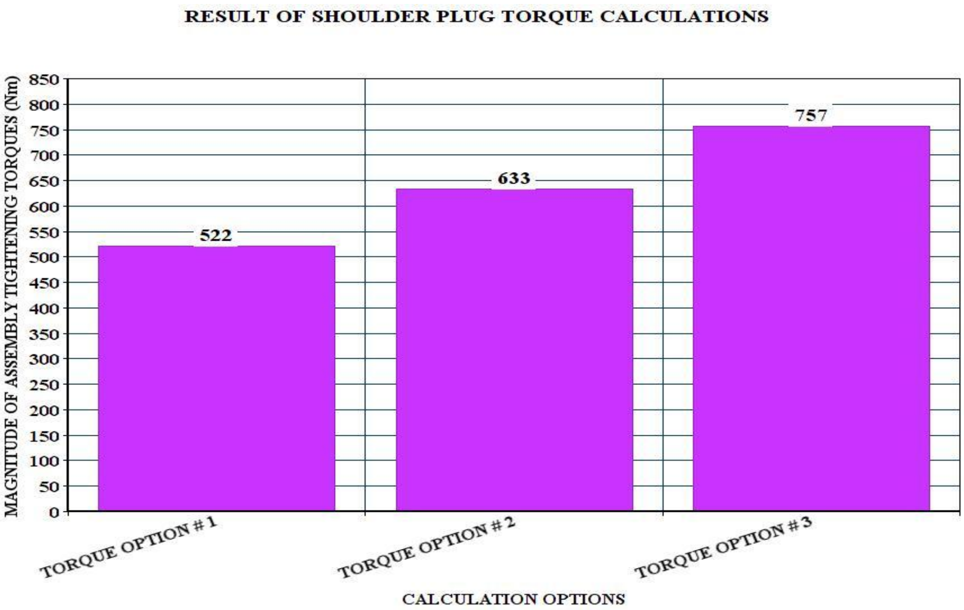
For most applications the nut factor K can be set at 0.2.

$T_b = 225 \times 0.2 \times 441.645 \times 38.1 / 1000 = 757$ Nm

IV. GRAPHICAL PRESENTATION of TIGHTENING TORQUE RESULTS

In the graphical representation of the calculation results, a distinction is made between the following approaches:

APPROACH	Motosh VDI 2230 ISO 16047 EN 1591-1	K-Factor	Gasket Stress ASME PCC-1
GRAPH IDENTITY	TORQUE OPTION #1	TORQUE OPTION #2	TORQUE OPTION #3



The coefficient of total friction μ_{tot} can be determined according the following approximate formula:

$\mu_{tot} = [\ (T/F) - (P/2\pi)] / (0.577\ d_2 + 0.5G)$

$\mu_{tot} = [(522240/83052) - (2.1167/2\pi)] / (0.577 \times 36.72586 + 0.5 \times 42.6) = 0.14$ for option # 1

$\mu_{tot} = [(632856/83052) - (2.1167/2\pi)] / (0.577 \times 36.72586 + 0.5 \times 42.6) = 0.17$ for option # 2

$\mu_{tot} = [(757000/83052) - (2.1167/2\pi)] / (0.577 \times 36.72586 + 0.5 \times 42.6) = 0.20$ for option # 3

V. CONCLUSION

The applied approaches for the considered case show a significant spread in calculation results. This corresponds with experience gained with manufacturers for such equipment. In the first approach, the friction component appears to play a dominant role in determining the tightening moment. This is also the case with the second approach, where the K-factor is in fact one of the total friction-dependent parameter. In the third approach, both the target gasket stress and the K - factor play a prominent role in determining the assembly tightening torque.

VI. CLOSING REMARK

Expertise, experience and engineering judgement together form the key for a prudent choice of the required assembly torque for the plugs. However, I would argue for encouraging code - and standard committees to draw up coherent rules for determining the required tightening moments for air cooled heat exchanger header box plugs. Hopefully, this article will motivate code committee members enough to take this issue on board.

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