Bolting of flange joints – Part 2

In part one of this article we discussed (published in Valve World September 2020) various properties related to bolt strength as well as several challenges to developing and maintaining a leak-tight clamping load. In part two we use the long form torque equation to evaluate the required bolt load necessary to develop a given gasket sealing stress and resist the full design hydraulic pressure.

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We then review the challenges discussed in part one and enumerate the additional bolt load required for each of these. With the total bolt load found, we then check the state of stress in the M16 25CrMo4 bolts and compare the results to both the proof load and yield strength to judge whether the original choice of bolt material is sufficient to maintain the targeted gasket stress.

Case 1: targeted operating gasket stress and hydraulic load (base case)

Presume a PN16/ 16 bar, PN1092-1 welding neck flange pair with nominal bore of 125mm, 8x - M16 25CrMo4 Bolts (with a true stress area of 156,67mm²), an internal pressure of 1N/mm² (10 bar) and a targeted gasket stress of 20MPa. Gasket dimensions are 192mm and 141mm outer and inner diameter, respectively. With no further consideration to design temperature or other affects, the target torque M_t is evaluated as follows:

$$M_t = [(p_t/2\,\pi) + (u_t d_t/2\,\cos a) + (u_n\, d_n/2)]\, F_{Bo}/B_{no} \quad \text{Eq. 1}$$

In the case of M16 stud bolts and the published coefficient data from a commonly used lubricant, we fill in the following values.

p_t = pitch of the bolt thread = 2mm

u_t = friction coefficient on threads = 0,13

d_t = mean contact diameter of thread = 14,7mm
m_t = friction coefficient of threads = 0,08
a = half thread angle = 30 degrees
m_n = friction coefficient under nut spot face
d_n = mean contact diameter on nut spot face = 23,8mm; based on average permissible value
F_{B0} = the total bolt load as evaluated below = 282383N
B_{no} = number of bolts = 8
F_{B0}, the total bolt load, is equal to the sum of the load to produce the targeted gasket stress (F_G) and the load to resist the hydraulic load (F_H). F_G and F_H are evaluated as follows:

F_G = 20MPa (π/4) [(192mm)² – (141mm)²] = 266768 N Eq. 2
F_H = 1N/mm² (π/4) (141mm)² = 15615 N = 15600 N Eq. 3
F_{B0} then becomes, 266768 N + 15600 N = 282368 N Eq. 4

Substituting the total bolt load 282368 N into equation 1 the targeted value of torque for Case 1 becomes 84 Nm.
A check of the bolt stress is made using the *true stress* area of the 16mm bolts and is found to be:

$S_B = F_{G+H}/(B_{no})\ (A_s)$ = (266768N + 15615N)/(8 bolts)(156,67mm²) = 225 N/mm². Eq. 5

Considering only the targeted gasket and hydraulic load we find the bolt stress is be well below its 21°C yield value of 440 N/mm² value. We now begin to evaluate the considerations given in part one.

Case 2: temperature

First we include the effect on bolting strength at a design temperature of 200°C. From Table A in Part 1 we find a reduction in bolt strength of 9,4%. This reduces the stress on a given bolt by 0,094 x 225 N/mm² = 21 N/mm². The loss of bolt force, on a per bolt basis, is found by multiplying bolt stress times bolt true stress area (21N/mm²) (156,67mm²) and is found to be 3290N. Totaling the loss of 8 bolts the compensation becomes (8)(3290N) = 26320N.

Case 3: gasket creep and relaxation

For our hypothetical example, we'll assume that testing verifies a creep relaxation value of 10%, of its original value. We now multiply F_{G+H} by 10% and get an additional 28238N of force required to compensate for the creep relaxation loss of the gasket.

Case 4: cyclic condition

It's recommended that the gasket manufacture be consulted for the best available values. For our example, we'll presume a reduction in gasket stress of 5%. The increase on bolt load for a cyclic condition becomes F_{G+H} times 5% = 14119N.

Case 5: equipment misalignment

This is one of the more difficult adjustments to make as there's no practical method to evaluate the loss of bolt load to correcting misaligned flanges. ASME PCC-1, Appendix E provides useful guidance on limits of the various types of misalignments; centerline high/low, parallelism, rotational two-hole, and excessive gap[1]. Correcting the misaligned flanges is always the best option. However, to demonstrate the potential effect of misalignment we'll consider the case where the misalignment is presumed to raise the necessary bolt preload by 20%. An additional load of (0,2 x F_{G+H}) = 56477N is now added to the total bolt load.

Case 6: embedment consideration

The value of bolt load lost to embedment, like so many conditions effecting bolt load, will have a range of applicable values and is never really known to a high degree of precision. A value as high as 10% is not uncommon[1]. In our instance we'll presume prior tightening has reduced the effect to a value of 5%. This is also applied to the total bolt load of F_{G+H}. The 5% compensation results in an additional bolt-up load adjustment of 14119N.

Tally of compensations

In Table 2 below we now tally the load compensation for our five additional considerations, with Case 1.
The total force at bolt-up now becomes 370828N. This is approximately 30% higher than the initial targeted bolt load for the gasket and hydraulic load. This underscores the importance of including these types of considerations in a check of the required bolt strength as well as including these considerations under actual service conditions. We now recheck the state of stress for the bolt material at the total compensated loading as:

$S_B = F_{Total}\ /\ (B_{no})(A_s)$ = 370828N/ (8) (156.67N/mm²) = 296 N/mm².

This is significantly higher than the initial 225 N/mm² value. The bolts are now at ~50% percent of their yield strength and ~75% of their proof strength. Within the precision of these considerations we verify that the 25CrMo4 bolting material is still a good choice. This of course presumes that the upper strength limit of the flanges and gasket is not exceeded. Still, the designer is cautioned that this is nevertheless a relatively high bolt load and depending on cyclic conditions, fatigue affects might also be considered.
In part one of this article we introduced basic bolting properties and common adjustments needed to compensate for affects that will reduce the bolt-up gasket stress. In this article, part two, we enumerate examples of what force adjustment may needed to compensate for these effects. With the total bolt load known, we then perform a final check on the state of stress for the initial choice of bolt. In a similar manner the designer or plant engineer can assess the effects of their particular considerations in attaining and maintaining a successful in-service bolt load.

Reference

1. Bickford, J. H., 1995, An Introduction to the Design and Behavior of Bolted Joints, 3rd ed., Marcel Dekker, New York.

Table 2, Force Tally of Bolt Load Considerations

Consideration	Force (N)
Gasket Stress and Hydraulic Load	282385
Design Temperature	+26320
Creep and Relaxation	+28238
Cyclic Loading	+14119
Misalignment	+56477
Embedment	+14119
Total	370828