Asset Lifecycle – Guidance Note

Steel bolted flanged joint assemblies

MECHANICAL AND ELECTRICAL

Purpose

This Guidance Note aims to provide clarity on the fundamental principles of bolted flanged assemblies to ensure leak-free gasket joints and mitigate excess stress on components. This document outlines:

- Flanged assembly components
- Relevant standards
- Torque calculations and checks
- Potential issues
- Considerations during the fit-up and tightening of a Bolted Flanged Joint Assembly (BFJA)
- Recommended torque values

Limitations

This Guidance Note is limited to compliant metallic flanged assemblies, where flange patterns are generally in accordance with AS/NZS 4087 or EN 1092-1 (or 2). Torquing requirements for specialised components and assemblies such as magnetic flow meters, lugged valves and flanged joints for polyethylene pipe assemblies are not covered in this Guidance Note.

It is also imperative that BFJA's and connecting pipework are designed and constructed so that gasket faces are unconstrained upon assembly and within the tolerances outlined in the guidance note to achieve minimum seating pressure and constant compressive pressure. This should not create additional tensile stress on flanges, pipes, fittings or other components.

Overview

A BFJA is a common method for joining steel, ductile or cast-iron pipelines and spools to adjoining fittings including isolation valves, tees, control valves etc.

The correct assembly order and torque is required to keep flanges together, prevent gasket leakage, and mitigate potential damage and deformation of components forming part of the assembly.

Leaks at BFJAs present several ongoing challenges if not designed and installed correctly. These challenges include, but are not limited to the following:

- Potentially hazardous working conditions, compromising the safety of staff
- Maintenance liability to remediate leaks or replace deteriorated components
- Excessive cost, third party and reputational damage
- Supply interruptions for required isolations

References

Watercare

- **ME:** General Mechanical Construction Standard
- **MS:** Material Supply Standard

Other

- **ASME PCC-1: 2022 – Pressure Boundary Flange Joint Assembly.**
- **ASME BPC VIII - Division 1: Rules for construction of pressure vessels**
- **EN 1092-1: 2018**
- **BS EN 1092-2: 2023**
- **AS/NZS 4087: 2011**
- **AS 1110: 1995**
- **ISO 4014: 2011**
- **AS/NZS 1252.1: 2016**
- **AS 1111.1: 2015**
- **AS 1112.3: 2015**
- **AS1237.1:2002**
- **ISO 898: 2013**
- **Garlock Sealing Technologies**
- **Durlon Sealing Solution:** Installation training E-Book
- **Klinger**: C-4430 & C-6327 Product Information
- **James Walker**: Washers – critical element of successful bolting integrity
- **Lamons** – Gasket & fastener handbook

BFJAs aims to maintain a consistent compressive pressure between two flanges which allows gaskets to resist the internal operational pressure. In addition, gaskets generally require a minimum seating pressure to ensure the material accounts for all irregularities on the flange face when compressed. The minimum seating pressure varies depending on the gasket material.

The main objectives when designing and installing BFJAs are:

- ***Providing sufficient gasket pressure to seal the joint*** during installation and operation to prevent leakage.
- ***Preventing damage to the gasket*** resulting from over-compression due to excessive bolt tensile stress.
- ***Preventing damage to the bolts***, i.e. the specified bolt tensile stress should be well below the yield stress of the bolt.
- ***Preventing damage to flanges***, as excessive bolt stress can lead to flange rotation and permanent deformation eventuating in leakage.

Determining the appropriate parameters and specifying the correct materials plays an important part in ensuring long term leak-free BFJAs. Information should be carefully reviewed by a qualified design engineer during the design to confirm the appropriate assembly parameters and tolerances are specified for the application. This information generally includes:

- Pipe diameter, materials and standard: e.g. DN 800 mild steel cement lined (CLS)
- Flange pattern, type and dimensions: e.g. EN 1092, Type 11, raised face
- Bolt grade, size, and quantity: e.g. Grade 8.8, M36, 24 No.
- Operating and testing pressure: e.g. 16bar / 20bar
- Gasket type, thickness, minimum seating stress (y) and "m-value": e.g. Compressed Fibre, Klinger C4430, 3mm thick, y = 20 MPa, m = 2.2

The information parameters listed above can then be used to undertake the necessary calculations and checks before installation.

Components of a BFJA

The main components of a BFJA are:

- Flanges
- Fasteners (bolts and nuts)
- Washers; and
- Gaskets creating a seal between flange faces.

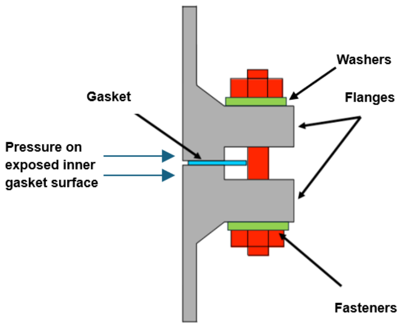


***Figure 1**: Basic arrangement of BFJA components.*

Flanges

Watercare's Material Supply Standard specifies acceptable flange standards and patterns as follows:

- AS/NZS 4087 (with weld necks manufactured to AS 2129) - Figure B7 for steel and B5 for ductile iron
- EN 1092-1 (-2 for ductile iron flanges) – Type 11 (Type 01 up to DN 250)

Note: EN 1092-1(2) flanges (including grade 8.8 bolts and fibre compressed gaskets) should be used for all water transmission assets. This includes smaller diameter pipework (less than DN 250) which may form part of the valve assemblies (e.g. bypass pipework).

All flanges must be raised face, have a minimum pressure rating of PN 16 to account for operational conditions, and have a minimum yield stress of 250MPa.

Flat face flanges may only be used on flat pump facings or specific design where a raised face may cause the connecting flange to break or deform the gasket in lower pressure systems.

Figure 2 below illustrates the general flange dimensions relevant to BFJA calculations.

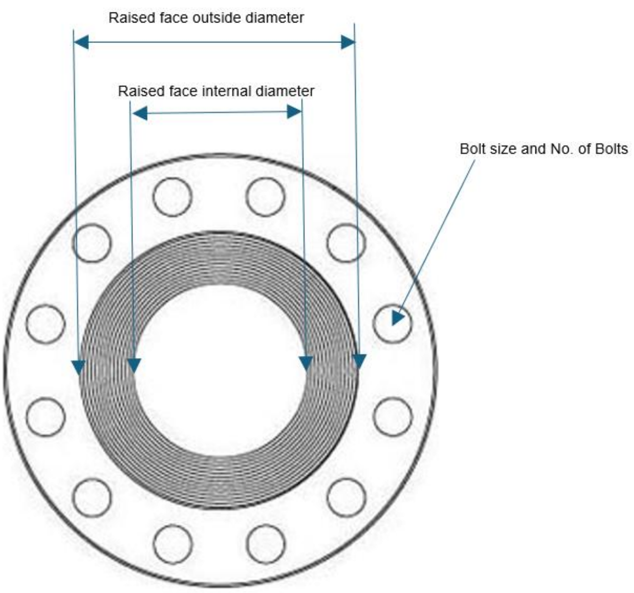


***Figure 2**: Flange dimensions used for BFJA calculations.*

Fasteners (bolts and nuts)

Bolt size

Bolts and nuts (fasteners) are used to create the necessary compressive force required to seal the BFJA. AS/NZS 4087 and EN 1092 specify the bolt diameter and thread pitch, and the number of bolts used for each flange size.

Figure 3 shows an extract from EN 1092:2018 where the bolt diameter and No. can be confirmed for the nominal diameter of the flange.

DN	Mating dimensions				
	Outside diameter	Diameter of bolt circle	Diameter of bolt hole	Bolting	
	D	K	L	Number	Size
	01, 02, 04, 05, 11, 1[illegible], 13, 21				
400	580	525	30	16	M27
450	640	585	30	20	M27
500	715	650	33	20	M30
600	840	770	36	20	M33
700	910	840	36	24	M33
800	1 025	950	39	24	M36
900	1 125	1 050	39	28	M36

***Figure 3**: Example from EN 1092-1 showing the bolt diameter for the respective flange / pipe diameter*

Once the bolt dimensions are confirmed it is important the correct bolt grade is selected for the application.

Bolt grade

Bolt grade (tensile rating) selection will be determined by the operating pressure, flange size and the gasket material to ensure the gasket can seal under the correct compressive force without yielding or causing damage to the flange.

The yield strength (stress) of the bolt subsequently also influences the torque value calculated.

Figure 4 below shows the stress / strain relationship of bolts and where plastic deformation is initiated as the stress exceeds the yield strength.

For this reason, the design utilization of bolts should be between 40% and 80% of the specified yield strength to prevent any permanent damage.

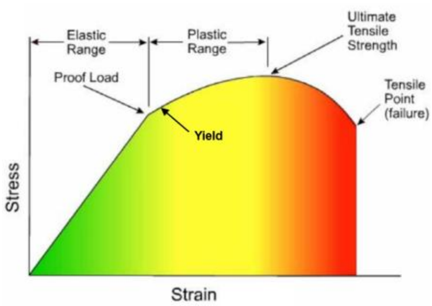


***Figure 4**: Bolt stress and strain relationship.*

For hot-dip galvanised bolts (e.g. grade 4.6), the whole number (4) signifies the tensile strength of the bolt (i.e. 400MPa for grade 4.6), while the decimal indicates the percentage where the bolt will begin to yield and permanently deform (e.g. 0.6 x 400MPa = 240MPa for grade 4.6 bolt).

Similarly, for stainless steel bolts (e.g. A4-50) the alphanumeric number denotes the type of stainless steel, i.e. A4 is 316 and A2 is 304. The hyphenated number specifies the tensile strength – refer to Table 1 below for further details on accepted bolt grades.

It should be noted that the yield strength for stainless steel bolts is comparatively lower when compared to galvanised steel bolts.

Table 1: Bolt grade and yield strength

Bolt Grade	Material	Tensile Strength	Yield Strength	Application
8.8	Galvanised steel	800 MPa	640 MPa	Transmission water and wastewater [1]
4.6	Galvanised steel	400 MPa	240 MPa	Network water and pressure wastewater
A4-50	Stainless steel 316	500 MPa	210 MPa	Network water and pressure wastewater

[1] **Note**: Grade 8.8 bolts shall only be used with compressed non-asbestos fibre (CNAF) gaskets

Bolt dimensions

Metric bolts are manufactured in accordance with the standards listed below which specify the physical dimension of each bolt size.

- AS 1111.1 and AS 1112.3 respectively for grade 4.6 or 316 stainless steel class 50.
  General purpose and small flange connections, bolts and nuts (network assets).
- AS/NZS 1252.1 grade 8.8 or 316 stainless steel class 70/80 or alternatively AS1110 (ISO4014) which is compliant with ISO898-1.
  Bolts and nuts for high strength structural engineering or large flanges (transmission assets).

Figure 5 below shows the physical characteristics of bolts, including the parameters used when calculating seating stress and torque.

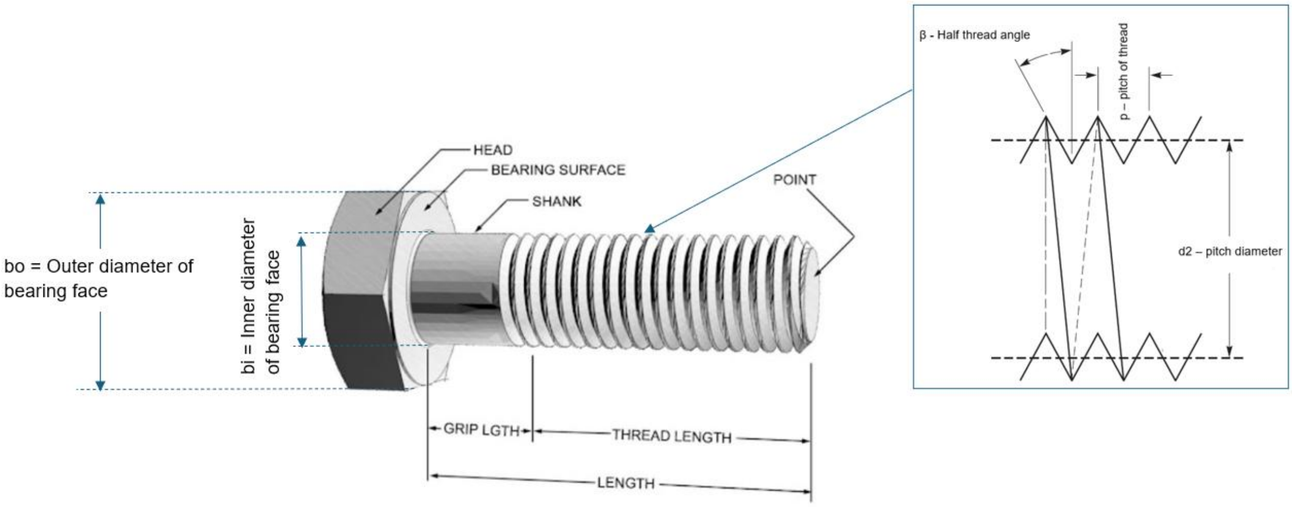


***Figure 5**: Physical characteristics of bolts*

Key dimensional parameters which should be captured when assessing bolts include:

- A_s: Tensile stress area of bolt (refer to Table 2)
- b_o: Outer diameter of bearing face
- b_i: inner diameter of bearing face
- D_e: effective bearing diameter $(b_o – b_i)/2$ (essentially the mean diameter of the bearing face)
- p: pitch of thread
- d_2: pitch diameter
- β: Half thread angle (assumed as 30° for all metric bolts)

Note: When selecting the bolt length, it should be long enough to extend one bolt diameter beyond the outer facing nut face to allow enough threads for nut restraint.

Table 2 shows resulting bearing areas based on dimensional parameters (tensile stress area is derived from d_2 pitch diameter), where the root area is compared to the minimum required bolt area to account for the internal pressure (W_{m1}) and initial seating pressure (W_{m2}) – refer to example calculation.

***Table 2**: Bolt tensile stress areas (reference Table H-1M of ASME PCC-1:2022)*

Bolt size	Thread pitch (p) (mm)	Root area (mm²)	Tensile stress area (As) (mm²)
M12	1.75	72	84
M14	2	100	115
M16	2	138	157
M20	2.5	217	245
M22	2.5	272	303
M24	3	313	353
M27	3	414	459
M30	3.5	503	561
M33	3.5	629	694
M36	4	738	817
M39	4	890	976
M42	4.5	1018	1121
M45	4.5	1195	1306
M48	5	1343	1473
M52	5	1615	1758

Washers

Approximately 50% of the torque applied when tightening a bolt is absorbed by friction between the mating surfaces fittings and the flange being forced together. Washers between bearing surfaces therefore play a crucial part in correctly tightening BFJAs where:

- Bolt tensile stress is translated over a larger area against the flange to ensure proportionate load distribution throughout the joint.
- The flange ring is protected from galling and nut embedment
- A firm, clean, and low friction bearing surface is provided for both the nut and bolt head

Washers shall be in accordance with AS1237.1-2002, and compatible with the bolt material. Table 3 below lists appropriate washer dimensions for grade 8.8 bolts.

***Table 3**: Washer sizes for grade 8.8 bolts (AS 1237.1:2002, class C)*

Bolt size (mm)	Washer internal diameter (mm)	Washer outside diameter (mm)	Washer thickness (mm)
16	17.5	30	4
20	22	37	4
22	24	39	5
24	26	44	5
27	30	50	5
30	33	56	5
33	36	60	6
36	39	66	6
39	42	72	6
42	45	78	6
45	48	85	8
48	52	92	8
52	56	98	8

Gaskets

Gaskets are introduced to maintain a static seal between two flange faces which are acknowledged to have imperfect surfaces. Such minor imperfections might include unobservable scratches, dents or machining grooves.

Generally, it is recommended to use thin gaskets where possible as these have:

a) Lower gasket blow-out risk and seepage as a smaller surface area is exposed to the internal working pressure (refer Figure 1).
b) Better torque retention in fasteners, as thinner gaskets have lower creep relaxation characteristics.
c) Lower cost (less material being used for manufacture).

A thicker gasket may seal more flange imperfections but can lead to delayed issues due to higher creep relaxation resulting in the need to retorque the fasteners to maintain the required compressive force on the gasket over its life.

Note: ASME PCC-1 assumes that the gaskets being used undergo a reasonable amount (>15%) of relaxation during the initial stages of operation.

EN1092-1 Recommends that gasket thickness be:

- 2mm for DN ≤ 300mm; and
- 3mm for DN > 300mm

The appropriate gasket material should be selected based on the application and operating conditions.

***Table 4**: Gasket materials and acceptable applications.*

Application	Gasket material	Bolt grade	Approved gaskets
Water networks (up to and including DN 250)	EPDM Rubber	Grade 4.6 or A4-50	EPDM ("blue") (AS/NZS 4020 approved)
Water Transmission (DN 300 and larger)	Compressed Fibre (CNAF)	Grade 8.8	Klinger C-4430; Garlock Style 2500; James Walker (AS/NZS 4020 approved)
Wastewater pressure networks (up to and including DN 300)	Nitrile Insertion Rubber	Grade 4.6 or A4-50	Flexitallic SF 2400 and SF 2800
Wastewater pressure transmission (larger than DN 300)	Compressed Fibre (CNAF)	Grade 8.8	Klinger C-4430; Garlock Style 2500; James Walker

Note: Nitrile insertion rubber gaskets may be considered for wastewater transmission pipelines up to DN600, in circumstances where a compressed fibre gasket is unforgiving. For these applications Grade 4.6 or A4-50 bolts should be used with the appropriate torque value to prevent damaging the gasket.

Relevant Standards

The following standards should be used to specify the correct BFJA assembly, providing comprehensive guidance on assembly components to ensure the integrity of the BFJA and maintain a high level of leak-tightness:

- ASME PCC-1-2022: Guidelines for Pressure Boundary Bolted Flange Joint Assembly
- ASME Boiler Pressure Vessel Code VIII: Rules for construction of pressure vessels

The metric interpretation of these standards should be used in conjunction with relevant material standards such as EN 1092-1 and -2, AS/NZS 4087 for flanges, and AS 1111.1, AS 1112.3, AS/NZS 1252.1 and AS1110 (ISO4014) for fasteners.

Calculating bolt torque

Bolt torque values (known as "target torque") are calculated and used during installation of BFJAs to secure a leak-free joint whilst staying well within the material yielding limits of the assembly components, i.e. bolts, flanges and gaskets.

Several formulas including simple and complex are available to determine appropriate torque values. Examples of both are provided below, however it is widely accepted and noted in ASME PCC-1:2022 that the simple formula ("Nut factor calculation") is considered equally effective when compared to more complex formulas. The example below compares values of both methods to demonstrate the validity of the "Nut factor calculation".

The "Nut factor calculation" (Simple)

This method is used across many industries, including water, oil and gas. It is a simplified method which provides comparable results to other more complex formulas. The target torque is effectively calculated from the product of bolt load, bolt diameter and a nut factor which has been derived through extensive experimentation and analysis.

The nut factor formula is:

$$T = (K \times D \times F)/1000$$

Where:

- T = Target torque (Unit: Nm)
- K = the nut factor (0.16 for lubricated bolts. Although 0.2 can be used for non-lubricated bolts, lubrication is mandatory for all bolts)
- D = nominal diameter of bolt (Unit: mm)
- F = target bolt load (Unit: N)

Note: ASME PCC-1 adjusts the nut factor by adding 0.04 to account for ambient conditions. Thus, the nut factor is increased from 0.12 to 0.16 for lubricated bolts, accounting for site conditions.

The target bolt load can be determined from the formula:

$$F = A_S \times \sigma_y \times P_{\%}$$

Where:

- As = Tensile stress area of thread – refer to Table 2 (Unit: mm^2)
- σ_y = Minimum yield strength of bolt – refer to Table 1 (Unit: N/mm^2)

- $P_{\%}$ = Percentage utilization factor for bolt material yield strength. Default value is typically 50%, however this can generally vary between 40% - 80%, depending on the total sealing force required (Unit: %).

Mathematical Model (Complex)

Shown below is the more complete mathematical formula which can be compared to the "nut factor calculation". This formula considers the physical characteristics of the bolt, as well as the contact area between the bolt and washer face.

$$T = \frac{F}{2}\left(\frac{p}{\pi} + \frac{\mu_t \times d_2}{cos\beta} + D_e \times \mu_n\right)$$

Where:

- T = Target torque (Unit: Nm)
- F = target bolt load (Unit: N)
- p = pitch of threads (refer to Figure 5 and Table 2 for values)
- μ_t = coefficient of friction for the threads
- μ_n = coefficient of friction for the nut face or bolt head
- d_2: pitch diameter (refer to Figure 5)
- β: Half thread angle (30° for metric bolts)
- De: effective bearing diameter $(b_o - b_i)/2$ (refer to Figure 5)

Example calculation

Example information for bolted flanged joint assembly

- DN 800 Pipe, flanged Connection operating at 16 bar (1.6 MPa)
- Flange Pattern: EN 1092
- Bolt Size: M36 (Available from EN 1092-1:2018 – refer figure 3)
- No. of Bolts: 24 (Available from EN 1092-1:2018 – refer figure 3)
- Gasket Type and Thickness: Compressed fibre – 3mm
- Pipe outside diameter / Raised face inside diameter: 813mm (assumed to simplify the calculation as the raised face inside diameter is generally offset 2-3mm from the pipe outside diameter)
- Raised face outside diameter: 900mm

Nut factor calculation (Simple)

1) K = 0.16 for lubricated bolts
2) D = 36 mm (M36 bolt)
3) $F = A_S \times \sigma_y \times P_{\%}$ = (817mm^2) (640 N/mm^2) (50%) = 261,440 N

Once the target bolt force (F) has been calculated, the torque can now be derived.

4) $T = (K \times D \times F)/1000$ = [(0.16) x (36 mm) x (261,440 N)] / 1000 = 1,506 Nm

Note: It is important to appreciate the sensitivity of the nut factor on the formula above, and therefore **the importance of lubrication**. For example, for an M36 bolt, the difference between K = 0.16 (lubricated) and 0.2 (unlubricated) results in a 25% increase in torque – see Figure 6.

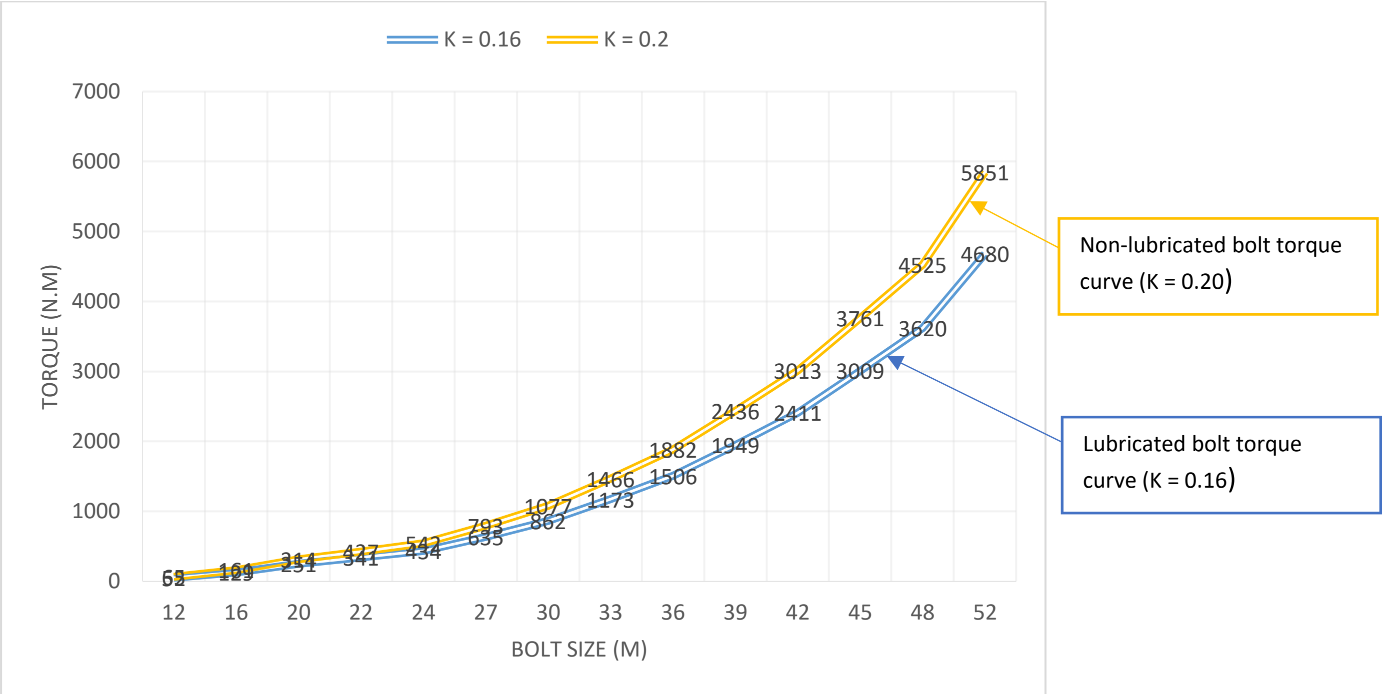


***Figure 6**: Various nut factors (K) impacting the torque value of a Grade 8.8 bolt at 50% utilization and highlighting the importance of bolt lubrication.*

Mathematical model (Complex)

$$T = \frac{F}{2}\left(\frac{p}{\pi} + \frac{\mu_t \times d_2}{cos\beta} + (D_e \times \mu_n)\right)$$

1) $F = A_S \times \sigma_y \times P_{\%}$ = (817mm^2) (640 N/mm^2) (0.5) = 261,440 N
2) p = 4mm (refer ***Table 2*** for values)
3) μ_t = assume 0.08 where graphite lubricant is used (reference: DIN 946)
4) μ_n = assume 0.15 where graphite lubricant is used (reference: DIN 946)
5) d_2 = d - 0.6495p = 36mm – 0.6495(4mm) = 33.4mm
6) β: Half thread angle (30° for metric bolts)
7) D_e = (54.4mm – 36mm)/2 = 45.2mm
8) $T = \frac{261{,}440N}{2}\left(\frac{4mm}{\pi} + \frac{(0.08)\times(33.4mm)}{cos30} + (45.2mm \times 0.15)\right)$ = 1456 Nm

Comparing the values from both these methods, the variance is only 3.4%. Figure 7 shows the comparison for a range of bolt sizes assuming a 50% utilization of the bolt load.

This suggest that the simple method provides a slightly more conservative torque value for all bolts sizes which is acceptable.

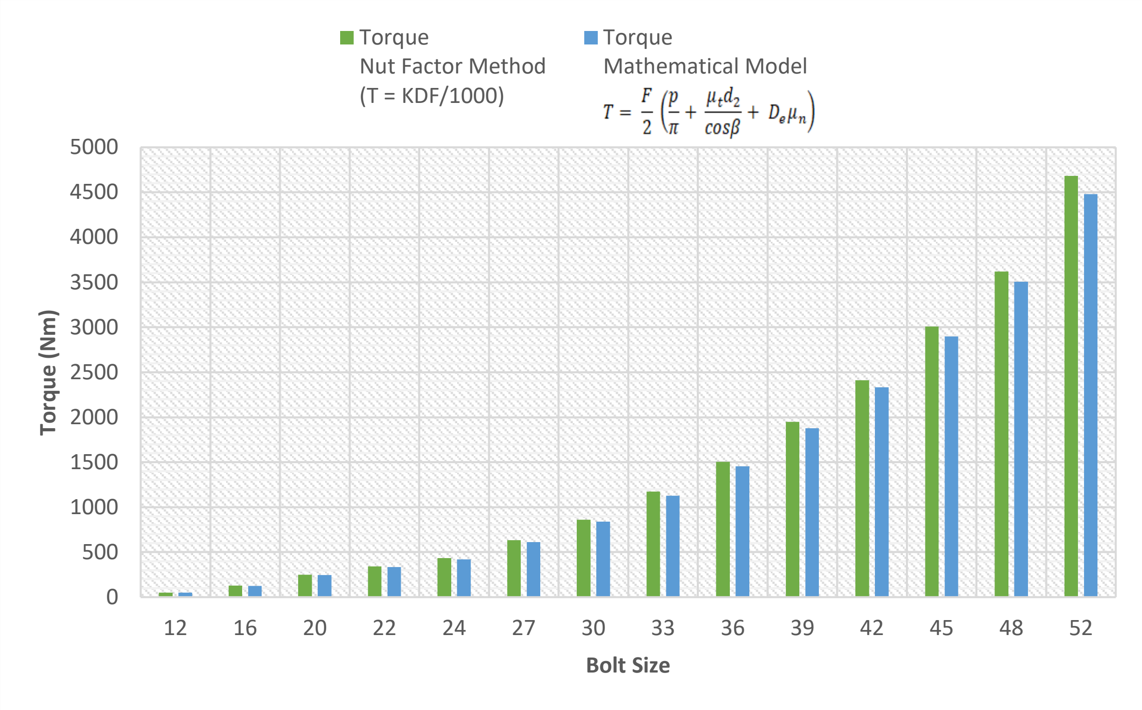


***Figure 7**: Comparison of torque values for bolt sizes based on 50% utilization for a grade 8.8 bolt.*

Bolt load distribution on flange

It is important to understand that the force, and resulting bolt stress experienced by each bolt is not equal to the stress exerted on the flange. Flange stress is relative to the area the bolt force is applied to.

Example

Assume a M36 bolt has a tensile stress area of 817mm^2, then the appropriately sized washer would have an equivalent contact area of 2,227mm^2 (ID = 39mm, OD = 66mm). The ratio of bolt to washer area is then 1:2.7. This means the stress translated over the washer contact area is 2.7 times less than that experienced by the bolt.

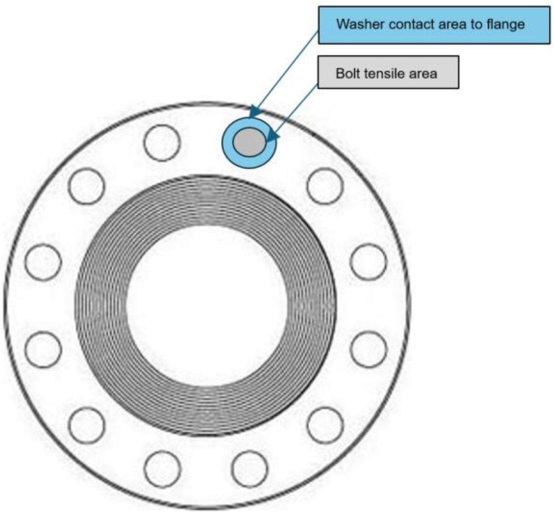


***Figure 8**: Bolt stress area and washer contact area.*

Therefore, if the bolt load is assumed at 50% utilization, the bolt stress would be 640MPa x 50% = 320MPa, and the stress translated through the washer to flange ring will equate to 320MPa / 2.7 = 119MPa. This is still well below the yield strength of the flange material - generally specified by industry standard as 250MPa minimum.

Checking bolt cross-sectional area and flange loading

ASME BPVC VIII Division 1, Appendix 2 sets out rules for bolted flange connections with ring type gaskets. This should be checked to confirm that the minimum combined bolt area is provided to seal the gasket without exceeding the allowable bolt stress. The bolts should provide enough pre-tension to:

1) Counteract the internal working pressure (W_{m1})
2) Provide a compressive force required to seat the gasket during initial assembly before the system is pressurised (W_{m2})

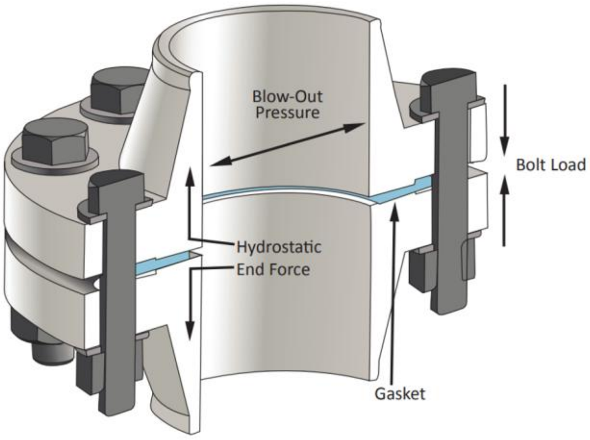


***Figure 9**: Axial forces operating on gasket (reference: Lamons)*

Counteracting internal working pressure (W_{m1})

The force required to counteract the internal working pressure which attempts to pry open the flange under pressure conditions is comprised of two forces:

1) A compressive force “H” to counter the distributed pressure imposed on the flange in operation – this is also referred to as the hydrostatic end force.
2) A compressive force “Hp” needed to seal the gasket in operation, i.e. the compressed area of the gasket under operating pressure conditions

Thus, the counteracting pressure force can be determined using the formula below:

$W_{m1} = H + H_p$ (reference ASME BPVC VIII)

Where:

$H = \frac{\pi}{4} G^2 P$ (Unit: N)

This force is the internal pressure (P) times the flow area but assumes the internal working pressure will partially pry open the raised flange face, causing the pressurized area to be wider than the internal diameter. The pressure thrust diameter to be used is a diameter “G” which can generally extends 2-3mm beyond the pipes’ outside diameter (i.e. up to the point before the raised face of the flange).

$H_p = 2.b \times \pi \times G \times m \times P$ (Unit: N)

The force H_P applies to a ring with a width equal to the full gasket sealing width (2b), and a diameter G, therefore a perimeter π x G. The formula for H_P includes a gasket factor m, which reflects an experimental ratio of the required contact pressure to the contained pressure P.

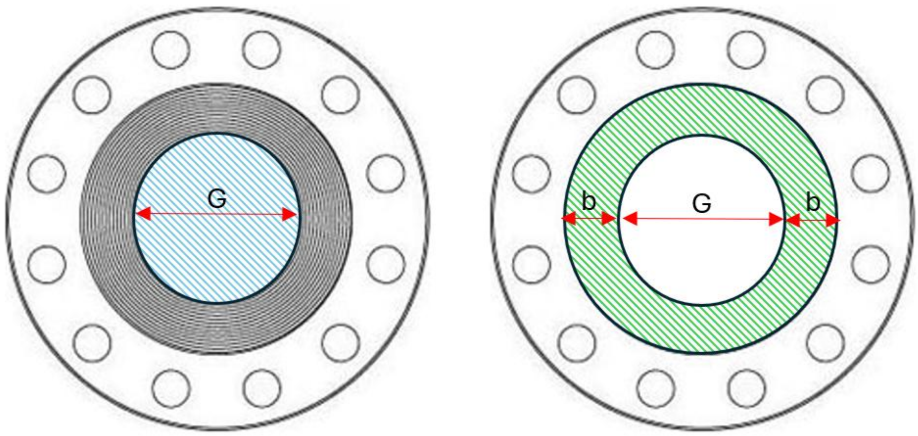


Figure 10: Illustration of dimensions for operational forces

Using the example information for bolted flanged joint assembly

- DN 800 Pipe, flanged Connection operating at 16 bar (1.6 MPa)
- Flange Pattern: EN 1092
- Bolt Size: M36 (Available from EN 1092-1:2018 – refer figure 3)
- No. of Bolts: 24 (Available from EN 1092-1:2018 – refer figure 3)
- Gasket Type and Thickness: Compressed Fibre – 3mm (y = 20 MPa, information received from manufacturer)
- Pipe outside diameter / Raised face inside diameter: 813mm
  Raised face outside diameter: 900mm

$$H = \frac{\pi}{4}\,(813mm^2)\,(1.6\frac{N}{mm^2}) = 830{,}598\,N$$

$$H_p = 2(43.5\,mm) \times \pi(813mm)(2.2)\left(1.6\frac{N}{mm^2}\right) = 782{,}172\,N$$

$$W_{m1} = 830{,}598\,N + 782{,}172\,N = 1{,}612{,}770\,N$$

***Table 5**: Gasket factors (reference: Table 2-5.1 ASME BPVC VIII)*

Gasket material	Gasket Factor (m)	Min. Design Seating Stress "y" (MPa)
EPDM (Less than 75A Shore hardness)	0.5	0
EPDM (75A or more Shore hardness	1	1.4
Compressed Fibre		
3-ply	2.25	15
2-ply	2.50	20
1-ply	2.75	26

Note: these values should be confirmed with the gasket manufacturer.

The minimum initial bolt force to seat the gasket (W_{m2})

Before a joint can be fully tightened, it is necessary to seat the gasket properly on the-contact surface by applying a minimum initial force. This force is a function of the gasket material and the effective gasket area to be seated without the presence of internal working pressure. The minimum initial bolt force is denoted as W_{m2} and can be calculated using the formula below

$W_{m2} = \pi \times b \times G \times y$ (reference ASME BPVC VIII)

Where:

1) b = gasket width (outside diameter of raised face minus pipe outside diameter) (Unit: mm)
2) G = Gasket inner diameter (can be assumed as the pipe outside diameter) (Unit: mm)
3) y = gasket or joint-contact-surface unit seating load (Unit: N/mm^2 or MPa)

Note: The "y" value is not the final sealing stress – this should be confirmed with the manufacturer, also refer to Table 3 above.

Continuing from the example above

$W_{m2} = \pi\ x\ (43.5mm)\ x\ (813mm)\ x\ (20N/mm^2)$ = 2,222,080 N

which greater than W_{m1}

It can be noted that the initial bolt force to seat the gasket (W_{m2}) exceeds that of the operating conditions (W_{m1}). This is typical on lower-pressure designs (when compared to the oil and gas industry) where the material requires a higher seating load. In this case the initial load to seat the gasket (W_{m2}) governs.

Comparing the required bolt area to the actual bolt area available

The minimum required bolt area (A_m) is determined by taking the larger value of W_{m1} and W_{m2} (in this case W_{m2}) and dividing it by the allowable bolt stress (S_a) at atmospheric temperature.

Thus

$$A_m = \frac{W_{m2}}{S_a}$$

Where Sa = $\sigma_y \times P_{\%} = (640\ N/mm^2)\ x\ (50\%) = 320\ N/mm^2$

Thus A_m = 2,222,080N / 320 N/mm^2 = 6,944 mm2

Where the available bolt area (refer Table 2) equal No. bolts x **Root bolt area** = (24) x (738mm^2) = 17,712mm^2 which is greater than 6944 mm^2 so this is ok.

Long term gasket pressure

Considering the force required to counteract internal working pressure, the residual clamp force should still be sufficient to keep the gasket seated and sealed.

The long-term gasket pressure refers to the force applied during operation after assembly and compensating for the hydrostatic end force trying to pry the BFJA open. This means the total bolt force minus the hydrostatic end force still needs to exceed the minimum sealing force acting on the gasket area, which will always be greater than the initial gasket seating force (W_{m2}).

Table 6 below summarises the gasket preferences and minimum sealing pressure.

Table 6*: General gasket applications and properties for PN16 applications (PN20 test). (reference: adapted from Garlock)*

Gasket material	Thickness	Application	Minimum Sealing Pressure (N/mm^2)	Maximum Sealing Pressure (N/mm^2)
Nitrile Insertion Rubber. Hardness range 60-75 IRHD units.	3mm	Wastewater (up to 1200mm)	4	6
EPDM Rubber. Hardness range 60-75 IRHD units. (AS/NZS 4020 approved)	2 - 3mm	Water networks (up to DN 250 diameter)	4	6
Compressed Fibre	3mm	Greater than 300mm	33	100

Note: Most EPDM or Nitrile Insertion Rubber gaskets will seal at approximate 1.5 MPa pressure.

Continuing from the example.

The **total bolt load available** = As x Bolt Stress (Sa) x No. Bolts

= 817mm^2 x 320 N/mm^2 x 24 bolts = 6,274,560 N **(A)**

Hydrostatic end force (H) $= \frac{\pi}{4} \times (813mm^2) \times (1.6\frac{N}{mm^2}) = 830{,}598\ N$ **(B)**

Thus A – B = 5,443,962 N

And the long-term pressure on the gasket will be 5,443,962 N / 117,049 mm^2 (gasket area) = 47 N/mm^2 (MPa) which is greater than 33 N/mm^2 (MPa).

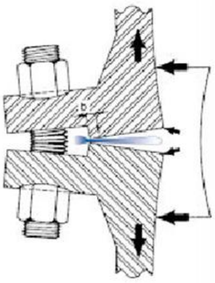
Therefore, enough loading will be applied to keep the assembly sealed and the gasket seated.

Note: Re-torquing may still be required following the initial tightening of the BJFA – refer to section below on *Tightening sequence and steps* for various gasket materials.

What can go wrong

Table 7 below shows some of the common issues encountered with BFJAs. These issues should be avoided when following the principles of assembly and applying the correct installation parameters.

***Table 7**: BFJAs Issues, symptoms and possible causes.*

Issue	Description	Symptoms	Causes	Illustration
Over compression of gaskets	• Gasket is cut almost entirely around the OD of the raised face • Gasket protrudes into the bore of the pipe	• BFJA leak • Excessive relaxation of gasket post tightening	• Severe over-tightening • Lack of bolt lubrication • Incorrectly sized / grade bolts (oversized)	
Contamination of gaskets	• The gasket leaks and/or blows out • Axial splitting	• BFJA leaks	• Gaskets are coated or comes into contact with bolt lubricants	
Under compressed gaskets	• Gasket is egg shaped and/or buckled near the bolts	• BFJA leaks • Gasket not seated properly	• Not tightening in sequence and flange deformation • Gasket shifted on flange • Minimum seating pressure not achieved	
Flange rotation	• Visible rotation of flanges as assembly is tightened to target torque	• Crushed gasket (may be visible) • Accelerated deterioration and leaks • Flange and bolt deformation visible	• Overtightening of bolts • Incorrectly sized bolts (oversized) Reduced gasket contact area • Misalignment of BFJA • Lack of bolt lubrication	
Nut or washer embedment of flanges	• Embedding of nut, bolt head or washer on flange ring surface	• Embedment marks on flange surface • Signs of galvanic corrosion around nut or washer • Flange coating damaged	• Overtightening of bolts • Not tightening in sequence and flange deformation • Lack of bolt lubrication	
Galling of nuts and bolts	• Threads of bolts and nuts shows visible wear	• Seizing of fasteners which lock up	• Lack of proper lubrication • Non-compatible lubricant (such as spray film lubricant) • Overtightening of bolts	

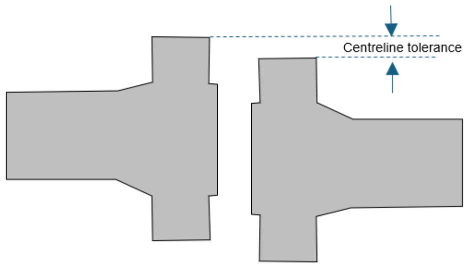
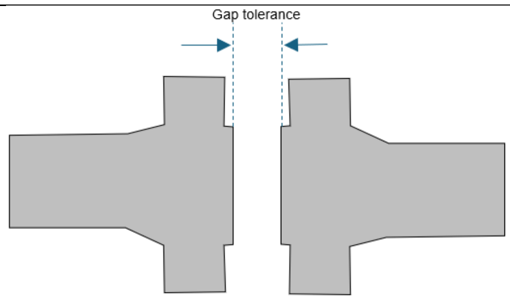
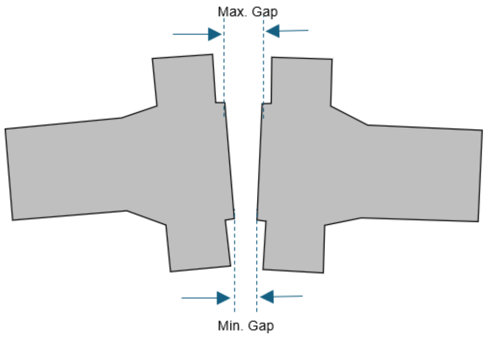
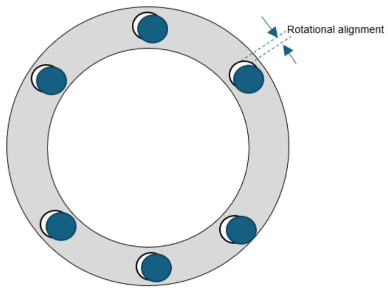
Considerations during fit-up and tightening of BFJAs

Preparation and the handling of equipment and materials is just as important as the physical assembly and tightening of the BFJA. What follows are key consideration which should form part of every contractor's Quality Assurance (QA) plan.

Alignment of BFJAs

When fitting up the BFJA, accurate alignment is necessary to ensure uniform loading of the gasket and prevent unintended stress and strain transferring to the flanged elements. It is also imperative that BFJA's and connecting pipework is designed and constructed so that gasket faces are unconstrained upon assembly and within the tolerances outlined. Four key alignment considerations are shown in Table 8 below.

***Table 8**: Flange joint alignment aspects and tolerance.*

Alignment aspect	Tolerance	Illustration
Centreline tolerance	1.5mm	
Gap between flanges	2 x the gasket thickness	
Parallelism	Max. gap – Min. gap ≤ 0.8mm	
Alignment of bolt holes (rotational alignment)	3mm	

Handling of gaskets

The correct handling and storage of gaskets will help prevent deformities and contamination which may affect the quality of the seal over the operational life of the asset. Its important that the contractor takes the following precautions as part of a Quality Assurance (QA) plan:

- Store gaskets in a flat position, and avoid bends or folds
- Store gaskets away from moisture
- Avoid contamination (this includes contamination with bolt lubricants)
- Never reuse gaskets or use damaged gaskets

Lubricating bolts correctly

An approved lubricant is mandatory for BFJAs. As previously outlined, this reduces the coefficient of friction between the bolt and nut which results in a lower target torque to achieve the required force. It also helps improve consistency of achieving the required force from the bolt which is transferred to the BFJA, prevents galling, and aids in the disassembly of the fasteners when required.

Lubricants must be chemically compatible with the bolt material to prevent adverse torquing conditions and possible reactions with the bolt material (e.g. galvanic corrosion).

- Acceptable lubricants include Molykote 1000 and Loctite 771
- Lanotec is recommended for cathodically insulated joints.

Before applying the lubricant to the bolt and nut threads, the nuts must run freely by hand past where they will come to rest after full torque. If nuts will not turn freely by hand, check for damage and make necessary corrections/replacements to avoid thread galling.

The lubricant should be applied after the bolts are inserted through the flange bolt holes (before commencing with torquing) to avoid possible gasket contamination. Once inserted, lubricant should be applied liberally and completely to the nut contact faces and to the threads on both ends of the bolts past where the nuts will come to rest after tightening.

No lubricant is permitted to make contact with the gasket on either side of the assembly.

***Table 9**: Lubricant application to bolts*

Insufficient lubricant applied	Lubricant applied correctly
 X	 ✓

Note: its good practice when tightening bolts to have the nut face with engraved data showing outward for any future inspections.

Tightening sequence and steps

Tightening sequence or patterns forms an important part of the assembly to evenly align the BFJA and seat the gasket. This also helps mitigate potential misalignment aspects (as listed in Table 8). AMSE PCC-1 and manufacturers cover a range of acceptable tightening patterns which is generally in a variant of a star or quadrant pattern. The Contractor should confirm and apply an acceptable tightening pattern to ensure all bolts achieve the target torque without unintentional stress or damage to the BFJAs elements.

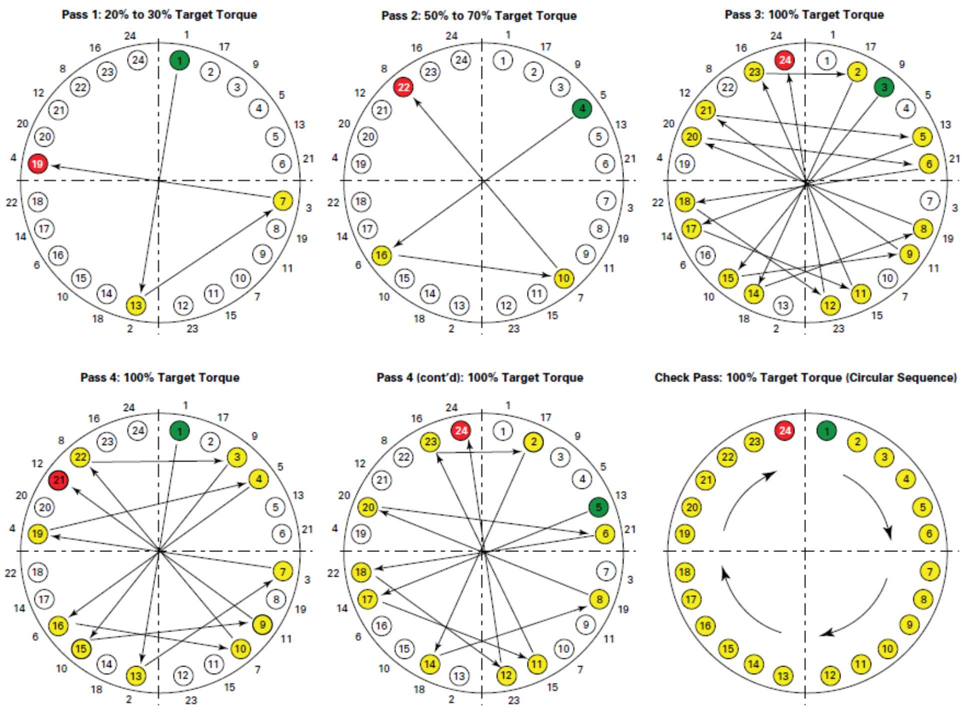


***Figure 11**: Example of Star Pattern tightening sequence where outer number indicate sequence of tightening (reference ASME PCC-1:2022)*

ASME PCC-1: 2022 outlines the recommended tightening procedure to help ensure:

- The gasket is initially uniformly seated across the seating area,
- The flanges are tightened in increments to prevent misalignment and uneven torquing
- The gasket is fully tightened following any short-term creep relaxation

The recommended steps are listed below, in accordance with the sequence outlined in Figure 11:

Step 1: Hand tighten the assembly to between 15 Nm and 30 Nm (but not exceeding 20% of the target torque “T” value). This ensure that the flanges are fitted up snug and uniformly against the gasket.

Step 2: Tighten to 20% - 30% of the target torque “T”, while monitoring the flange gap around the circumference of the assembly. Ensure that that the gap is kept continuously uniform. Adjust appropriately if required.

Step 3: Tighten to 50% - 70% of the target torque "T", while monitoring the flange gap around the circumference of the assembly. Ensure that that the gap is kept continuously uniform. Adjust appropriately if required.

Step 4: Tighten to 100% of the target torque "T", while monitoring the flange gap around the circumference of the assembly. Ensure that that the gap is kept continuously uniform. Adjust appropriately if required.

Step 5: Continue tightening the bolts, but in a circular clockwise pattern until no further rotation occurs at 100% target torque.

Step 6: Wait 4-hours following the final torquing (Step 5) and repeat Step 5 to account for any short-term creep relaxation or embedment losses. This may also be required following pressure testing where the test pressure is higher than the operating pressure.

Lifting and moving BFJAs

When BFJAs are handled and fitted up, this can be done by either aligning horizontally or vertically. It is important that regardless of the orientation, all parts of the assembly are well supported whether in a static position or when transferring to the final position for tying in. Lack of proper support during lifting, moving or storage may subject components of the assembly to unnecessary stress.

***Figure 12**: BFJA assembled vertically*

Recommended torque values

Considering the principles covered in all sections of this Guidance Note, the tables below provide the recommended torque values for steel and ductile iron BFJAs.

Note: torque values to achieve a suitable force in the following tables will vary for the same DN pipe and identical gasket material / thickness based on the flange standard, No. and diameter of bolts and contact surface area.

For example, DN100 EPDM gasket: EN 1092 flanges require 39 Nm applied to 8 No. M16 bolts whereas AS/NZS 4087 flanges require 48 Nm applied to 4 No. M16 bolts.

***Table 10**: Recommended torque values using **EPDM or Nitrile Insertion Rubber Gaskets** for **EN 1092-1(2)** Flanges with **Grade 4.6 or A4-50 Bolts** (network water and wastewater up to DN 600[(a)])*

Pipe DN	No. Bolts	Bolt Size (mm)	Bolt grade	Target Torque "T" (Nm)	Gasket Thickness (mm)	Water Networks	Wastewater Pressure
100	8	16	4.6 or A4-50	39	2	EPDM	Nitrile Rubber
150	8	20	4.6 or A4-50	75	2	EPDM	Nitrile Rubber
200	12	20	4.6 or A4-50	75	2	EPDM	Nitrile Rubber
250	12	24	4.6 or A4-50	130	2	EPDM	Nitrile Rubber
300	12	24	4.6 or A4-50	130	2	N/A	Nitrile Rubber
350	16	24	4.6 or A4-50	130	3	N/A	Nitrile Rubber[(a)]
400	16	27	4.6 or A4-50	190	3	N/A	Nitrile Rubber[(a)]
450	20	27	4.6 or A4-50	190	3	N/A	Nitrile Rubber[(a)]
500	20	30	4.6 or A4-50	259	3	N/A	Nitrile Rubber[(a)]
600	20	33	4.6 or A4-50	352	3	N/A	Nitrile Rubber[(a)]
700	24	33	N/A	352	3	N/A	N/A
800	24	36	N/A	452	3	N/A	N/A
900	28	36	N/A	452	3	N/A	N/A
1000	28	39	N/A	585	3	N/A	N/A
1200	32	45	N/A	903	3	N/A	N/A
1400	36	45	N/A	903	3	N/A	N/A
1600	40	52	N/A	1404	3	N/A	N/A
1800	44	52	N/A	1404	3	N/A	N/A

(a) **Note**: Nitrile rubber gaskets should only be used for transmission wastewater where it's demonstrated that compressed non-asbestos fibre (CNAF) gaskets are not suitable. Where Nitrile rubber gaskets are required, bolts shall be grade 4.6 or A4-50, using the associated torque values.

***Table11**: Recommended torque values using **EPDM or Nitrile Insertion Rubber Gaskets** for **AS/NZS 4087** Flanges with **Grade 4.6 or A4-50 Bolts** (network water and wastewater up to DN 600[(a)])*

Pipe DN	No. Bolts	Bolt Size (mm)	Bolt grade	Target Torque "T" (Nm)	Gasket Thickness (mm)	Water Networks	Wastewater Pressure
100	4	16	4.6 or A4-50	48	2	EPDM	Nitrile Rubber
150	8	16	4.6 or A4-50	48	2	EPDM	Nitrile Rubber
200	8	16	4.6 or A4-50	48	2	EPDM	Nitrile Rubber
250	8	20	4.6 or A4-50	94	2	EPDM	Nitrile Rubber
300	12	20	4.6 or A4-50	94	2	N/A	Nitrile Rubber
350	12	24	4.6 or A4-50	163	3	N/A	Nitrile Rubber[(a)]
400	12	24	4.6 or A4-50	163	3	N/A	Nitrile Rubber[(a)]
450	12	24	4.6 or A4-50	163	3	N/A	Nitrile Rubber[(a)]
500	16	24	4.6 or A4-50	163	3	N/A	Nitrile Rubber[(a)]
600	16	27	4.6 or A4-50	238	3	N/A	Nitrile Rubber[(a)]
700	20	27	N/A	238	3	N/A	N/A
800	20	33	N/A	440	3	N/A	N/A
900	24	33	N/A	440	3	N/A	N/A
1000	24	33	N/A	440	3	N/A	N/A

[(a)] **Note**: Nitrile rubber gaskets should only be used where it's demonstrated that compressed non-asbestos fibre (CNAF) gaskets are not suitable. Where nitrile rubber gaskets are required, bolts shall be grade 4.6 or A4-50, using the associated torque values.

Table 12: Recommended torque values using **Compressed non-asbestos fibre (CNAF) Gaskets** for **EN 1092-1(2)** Flanges with **Grade 8.8 Bolts** (Transmission assets only)

Pipe DN	No. Bolts	Bolt Size (mm)	Bolt Grade	Target Torque "T" (Nm)	Gasket Thickness (mm)	Gasket Type	Application	
100	8	16	8.8	129	2	CNAF	Water Transmission	N/A
150	8	20	8.8	251	2	CNAF	Water Transmission	N/A
200	12	20	8.8	251	2	CNAF	Water Transmission	N/A
250	12	24	8.8	434	2	CNAF	Water Transmission	N/A
300	12	24	8.8	434	2	CNAF	Water Transmission	N/A
350	16	24	8.8	434	3	CNAF	Water Transmission	Wastewater
400	16	27	8.8	635	3	CNAF	Water Transmission	Wastewater
450	20	27	8.8	635	3	CNAF	Water Transmission	Wastewater
500	20	30	8.8	862	3	CNAF	Water Transmission	Wastewater
600	20	33	8.8	1173	3	CNAF	Water Transmission	Wastewater
700	24	33	8.8	1173	3	CNAF	Water Transmission	Wastewater
800	24	36	8.8	1506	3	CNAF	Water Transmission	Wastewater
900	28	36	8.8	1506	3	CNAF	Water Transmission	Wastewater
1000	28	39	8.8	1949	3	CNAF	Water Transmission	Wastewater
1200	32	45	8.8	3009	3	CNAF	Water Transmission	Wastewater
1400	36	45	8.8	3009	3	CNAF	Water Transmission	Wastewater
1600	40	52	8.8	4680	3	CNAF	Water Transmission	Wastewater
1800	44	52	8.8	4680	3	CNAF	Water Transmission	Wastewater

***Table13**:* Recommended torque values using **Compressed Fibre Gaskets** for **AS/NZS 4087** Flanges with Grade 8.8 Bolts (Transmission wastewater)

Pipe DN	No. Bolts	Bolt Size (mm)	Bolt Grade	Target Torque "T" (Nm)	Gasket Thickness (mm)	Gasket Type	Application
100	4	16	8.8	-	2	-	N/A
150	8	16	8.8	-	2	-	N/A
200	8	16	8.8	-	2	-	N/A
250	8	20	8.8	-	2	-	N/A
300	12	20	8.8	-	2	-	N/A
350	12	24	8.8	564	3	CNAF	Wastewater
400	12	24	8.8	564	3	CNAF	Wastewater
450	12	24	8.8	564	3	CNAF	Wastewater
500	16	24	8.8	564	3	CNAF	Wastewater
600	16	27	8.8	825	3	CNAF	Wastewater
700	20	27	8.8	825	3	CNAF	Wastewater
800	20	33	8.8	1407	3	CNAF	Wastewater
900	24	33	8.8	1407	3	CNAF	Wastewater
1000	24	33	8.8	1407	3	CNAF	Wastewater

Legend for tables above

Application	Gasket type	Bolts grade	Legend
Water networks (up to and including DN 250)	EPDM (AS/NZS 4020)	4.6 or A4-50	
Water transmission (larger than DN 250, but also includes small transmission bypass pipework)	CNAF	8.8	
Wastewater pressure networks (up to and including DN300)	Nitrile rubber	4.6 or A4-50	
Wastewater transmission (larger than DN 300)	CNAF	8.8	

Review and validation of torque values

The torque values have been calculated using industry recognised formulas listed in ASME PCC-1. In addition, the values calculated have been verified by comparing values recommended by gasket manufacturers and pipeline installation manuals.