

Bolt Torque For Polyethylene Flanged Joints TN-38/January 2010



**LAP-JOINT STYLE FLANGE ASSEMBLY
(Based on ASME B16.5)
24-Inch Dia. Polyethylene Flange Adapters, Metal Lap-Joint Flanges, and Bolt Set**

Foreword

This technical note was developed and published with the technical help and financial support of the members of the Plastics Pipe Institute. The members have shown their interest in quality products by assisting independent standard-making and user organizations in the development of standards, and also by developing reports on an industry-wide basis to help engineers, code officials, specifying groups, and users.

The Plastics Pipe Institute, Inc. has prepared this technical note as a service to the industry. The information in this note is offered in good faith and believed to be accurate at the time of its preparation, but is offered without any warranty, express or implied. Additional information may be needed in some areas, especially with regard to unusual or special applications. Consult the manufacturer or material supplier for more detailed information. A list of member manufacturers is available from PPI. PPI does not endorse the proprietary products or processes of any manufacturer, and assumes no responsibility for compliance with applicable laws and regulations.

PPI intends to revise this report from time to time, in response to comments and suggestions from users of this note. Please send suggestions for improvements to PPI. Information on other publications can be obtained by contacting PPI directly or visiting the web site.

The Plastics Pipe Institute, Inc.
469-499-1044
www.plasticpipe.org

DISCLAIMER

Due to the wide variation in service conditions, quality of installations, etc, no warranty or guarantee, expressed or implied, is offered nor given in conjunction with the use of this Plastic Pipe Institute Technical Note. All data and formulae and example values presented here-in should be independently verified and validated by the end-user, reader, designer, engineer, field installer, technician, etc, for a specific HDPE pipeline flange installation or project.

CAUTION

FLANGES IN NATURAL GAS PIPELINES

Note: In jurisdictional installations any metallic pipeline components must be protected from corrosion as prescribed in US CFR Title 49, Part 192, Subpart I, sections 451-491. Furthermore, Part 195 Subpart H, sections 551-589 applies to steel pipelines used in the transport of Hazardous Liquids.

Index	Page
i. Preface	5
ii. Introduction	6
iii. CHECKLIST : Bolt Torque Spec & Project Record	8
iv. TABLE #1 : Bolt Tightening Sequence	9
v. TABLE #2 : Example Torque Calculations (HDPE to HDPE)	10
vi. TABLE #3 : Example Torque Calculations (HDPE to Steel)	11
vii. APPENDIX A	12
Bolt Pre-Load	12
Bolt Diagram	13
Computational Model: Bolt Load & Bolt Torque	16
Hydraulic Operating Load / Gasket Seating Load	16
Minimum Required Tightness (MRT)	17
Design factor (scatter) (DF)	16
Assembly Required Tightness (ART)	16
Minimum Operating Bolt Load (MOBL)	16
Minimum Seating Force (MSF)	17
Design Seating Stress (DSS) EQ #4	17
Torque Per Bolt EQ #5	17
“K” nut-factor EQ #6	17
Table #4: Bolt Dia vs Torque vs Load vs Tensile Stress	18
General Considerations	18
Flange Adapter	18
Metal Lap-Joint Flange	19
Heavy Hex Nuts	19
Heavy Hex-Head Bolts and All-Thread Rod	20
Washers	20
Lubricant	20
Torque Wrench	21
Gasket (if specified)	21, 27
Corrosion Control	22
The Bolted Connection	22
Flange Face	22
Alignment	22
Torque Progression	23, 8
Residual Bolt Torque (RBT) & Mandatory Re-Torque	24
Safe Disassembly Procedures	24
Hydro-Testing and Leak-Closure Guideline	25
viii Appendix B: Wrench Sizes for Heavy Hex Bolts & Nuts	27
ix Appendix C: Gaskets	28
x Appendix D: Typical HDPE Compressive Stress-Strain Curve	32
xi Appendix E: Corrosion Control References & Specifications	33
xii GLOSSARY	34
xiii References	38

Preface:

Based on ASME B16.5 flange styles, the polyethylene **Lap-Joint Flange Assembly** is a three component device consisting of:

1. Polyethylene flange adapter (stub-end).
2. A loose, metal, Lap-Joint Flange (**LJF**).
3. The bolt set.

The metal Lap-Joint Flange (LJF) cross-section geometry may be a rectangular solid or a contoured cross-section. The rectangular cross-section typically is machined from metal plate; the contoured cross-section is typically cast using molten ductile-iron or stainless-steel.

The LJF is typically in flat-face contact with the polyethylene flange adapter hub, and has a radius on the contact side of the LJF ID which mates with the fillet radius of the matching polyethylene flange-adapter (stub-end). The LJF slips over the pipe; is not welded to the pipe; is loose until bolted; and is free to rotate into bolt-hole alignment with another flange. The bolt-load is transferred to the sealing face by the pressure of the LJF against the back-face of the HDPE hub.

Two methods are commonly used to seal polyethylene Lap Joint Flange assemblies between various combinations of pipe materials such as HDPE to HDPE; HDPE to Steel; HDPE to Ductile-Iron; HDPE to PVC; HDPE to Fiberglass.

The first method, (non gasketed), uses the specified HDPE seating torque initially applied to the HDPE flange adapters, followed by a mandatory re-torque applied 4-hours to 24-hours after completion of the initial torque application

The second method, (gasketed), uses a low gasket seating bolt torque, applied to a soft elastomeric gasket, for lower pressure applications (like landfill gas collection or use with torque-limited PVC or fiberglass flanges), followed by the mandatory re-torque 4 hours to 24-hours after the initial torque.

PPI strongly recommends that each flanged joint be independently analyzed by the project engineer for sealing capacity when subjected to all expected operating and installation loads.

By applying the higher initial seating torque to seat the un-marred HDPE faces, without gaskets, the final residual bolt torque (RBT) at the HDPE sealing stress is sufficient to contain flow-stream pressure under operating conditions.

As is discussed later, the mandatory re-torquing to the initial target torque after a 4 hour to 24 hour creep-relaxation period is done to compensate for possible bolt-creep, nut embedment, and, gasket compression-set (if gaskets are used).

- Consult the individual HDPE flange manufacturers for their recommended protocol.
- Flange-Adapter Manufacturers should verify their flange assemblies are performance rated when used with a specific style or manufacturer's LJF.
- LJF (lap-joint flange) manufacturers should verify the maximum allowable torque that can be applied to their product, and that their LJF's provide "disk" deformation in excess of the polyethylene flange-adapter's expected service life's visco-elastic creep deformation, at low residual compressive stress.

CAUTION: When bolting to fiberglass, cast iron, PVC pipe flanges, or PVC flanged valves, the "brittle" flange typically bolts to a special HDPE full-face flange adapter using lower bolt torque. Hence a soft gasket is frequently also used with "brittle" pipes. Over-tightening, misalignment, or uneven tightening can break brittle material flanges. Extreme care is advised. Refer to Appendix C, and consult with the sensitive, low-strain product manufacturer for its maximum torque limits, when bolting to "raised-face" HDPE flange adapters. When gaskets

are to be considered, review [Appendix C](#) very carefully, to perform calculations using the seating stress, blow-out resistance, crush resistance, and other performance values obtained from the gasket manufacturer. This Tech-Note does not provide guidance on gasket selection; consult with the gasket designer to discuss the parameters outlined in [Appendix C](#).

Introduction:

Lap-Joint Flanges (LJF) have been used for decades. The typical polyethylene flange adapter with loose LJF is also known as a Van-Stone Flange joint. The HDPE flanged joint assembly is an engineered pressure containment connection subject to diverse forces. While simple in appearance, its design is complex due to the axial shear, radial dilation, disk-bending moments, residual interfacial sealing pressure, bolt-load versus bolt-torque, HDPE flange face creep-relaxation, LJF disc flexure, axial tension from thermal contraction of the pipe-line, some vibration, pressure-surge, pipe bending due to soil settlement, etc. The greatest contributors to flange leakage are insufficient torque, un-even torque, and flange misalignment. Written and correct bolt torque specifications and installation procedures will eliminate these problems. The flange assembly design, and written assembly specifications, are controlled by the pipeline design-engineer or project engineer-of-record.

The ideal flange-adapter joint should exhibit Compressibility, Resilience, and Creep-Resistance. The plastic flange-adapter face should be able to compress into any and all surface texture and imperfections of the mating flange. The plastic flange face should be sufficiently and elastically resilient to move with dynamic loadings to maintain seating stress. The flange-adapter face should exhibit sufficient creep-resistance so as not to permanently deform after bolt-up under varying load cycles of temperature and pressure.

The “memory” of pipe-grade HDPE makes it an ideal flange face sealing surface. It becomes its own “gasket flange”, and seals well when un-marred and torqued to meet or exceed the HDPE seating stress. When properly torqued with a flexible LJF, the HDPE flange-adapter becomes self-gasketing.

The LJF assembly is typically evaluated as a combined mechanical “spring” assembly. The torqued bolts are elastically stretched to initiate the sealing pre-load. The metal LJF (lap-joint flange) is elastically flexed (bent by the bolt-load) to maintain the pre-load and to transfer the load to the HDPE flange face. At small strains, the HDPE flange-face is elastically and visco-elastically deformed (axial compression and slight radial enlargement) so as to maintain pre-load sealing pressure on the flange-face surface. The HDPE flange face compressibility is the measure of its ability to deflect and conform to the mating flange face. This compressibility compensates for flange surface irregularities such as minor nicks, non-parallelism, metal corrosion, and variation in surface roughness or grooving depth. The HDPE flange face also exhibits Memory / Recovery / Resiliency which are measures of the elasticity of the HDPE material to recover shape and to maintain its deformation sealing pressure under varying loads across broad temperature ranges. Although the HDPE is a visco-elastic material that slightly creeps over time, at sufficient torque the flexure of the LJF and bolt stretch exceed the expected long-term compressive creep of the flange face, such that the residual sealing force exceeds the sum of the operating separation forces. In this way, the sealing pressure is maintained.

The combined “springs” of the stretched bolts, the flexed disc LJF, and the elastic component of the compressed flange-face, all serve to provide an elastic / visco-elastic, resilient “spring-seal” of the hydrostatically pressurized joint.

The key element to an effective sealing HDPE flanged joint, is to torque the bolts to a sufficiently high value to stretch the bolts, so that the LJF is flexurally distorted, and the HDPE flange-face sufficiently and continuously compressed. The joint is at equilibrium, with the compressive sealing force distributed across the sealing face and equal in magnitude to the pre-tension in the

bolts. The total bolt tension must be able to constrain the joint assembly against operating pressure, surge pressure, pipe-line axial thermal contraction, and pipe bending strain from soil settlement, and flange angular alignment; all with an applied safety factor.

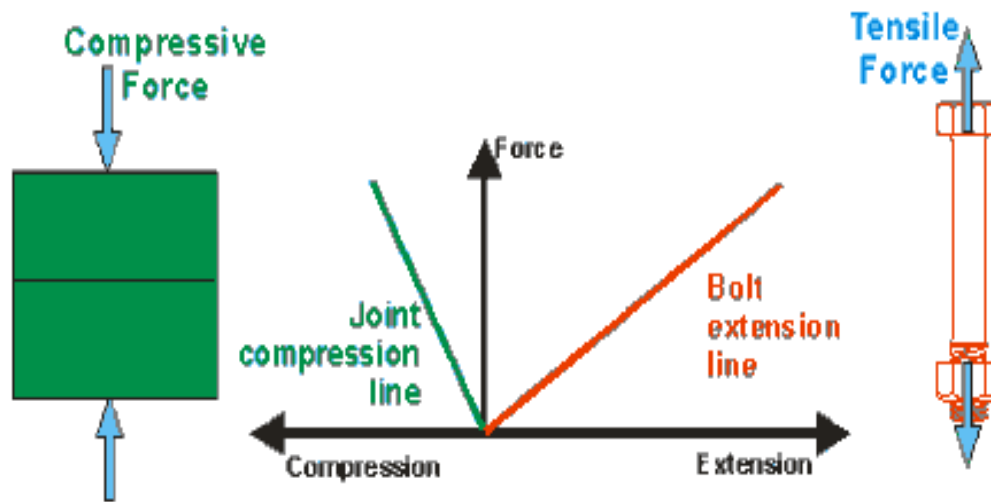


Figure 1

The total possible force required from bolting torque should equal and exceed the sum of applicable separation forces:

$$F_{\text{Total}} > \{ F_{\text{Press}} + F_{\text{Surge}} + F_{\text{Therm.Contr.}} + F_{\text{Pipe-Bend}} + F_{\text{Flng-Alignment}} + F_{\% \text{ Variance}} \}$$

Equation # 1

Caution: The component, $F_{\text{pipe-bend}}$ (forces from pipe beam-bending), in the above equation can sometimes exceed thermal contraction and hydraulic forces. HDPE flange joints are geometrically rigid assemblies, unlike the flexible HDPE pipe ring “hoop”. The rigid flanged joint cannot shed stress by ring deformation. Localized HDPE pipe beam-bending at a flanged joint due to soil settlement, water buoyancy or wave action, pipe “snaking” above ground, etc, must be managed so as to isolate the flange from beam-bending strain. External installation measures to protect PE flange joints from beam-bending strain are necessary. While additional torque can maintain the pressure seal, bending strain across the HDPE flange adapter should be limited to prevent flange adapter fracture.

NOTE:

Appendix “A” provides the method for calculation and determination of specified bolt torque at the required seating stress. Proceed to Appendix “A” to perform the required engineering calculations to determine the required target torque to be used in the Checklist following on the next page.

CHECKLIST and FLANGE TORQUE RECORD:

Project: _____ Flange Set Location : _____
Connecting HDPE Flange to _____ Flange.
Bolt Dia & Grade : _____ Nut Diameter & Grade : _____
Lap-Joint Flange Dia. & Pressure Rating : _____
Lubricant Used: _____ Flange Temp: _____
Torque Wrench ID #: _____ Calibration Date: _____
If Specified: Full-face Gasket Info: Material: _____ Thickness: _____
Deep-Well Socket / Heavy-Hex Nut Wrench Size Used : _____

Axis off-set: _____ Angular & Facial Gap: Top _____ Bottom _____ L _____ R _____

“Initial” Each Step Upon Completion:

- _____ 1. Visually examine and clean both flanges, bolts and nuts. Replace damaged units.
- _____ 2. Liberally Lubricate bolt threads & nut threads & flange surface under nut.
- _____ 3. If gasket is specified, insert full-face gasket. Do not use wrinkled or damaged gaskets.
- _____ 4. Number the bolt-holes in circumferential sequence stating at 12:00 position.
- _____ 5. Check Flange alignment, concentricity, angularity, and gap for acceptability.
- _____ 6. To firm the flanges squarely together, Hand Tighten, then pre-tighten all bolts in proper sequence to 10-20 foot-pounds torque, but do not exceed 20% of the TARGET TORQUE.
- _____ 7. Re-check any flange-adapter face gap and LJJ gap for uniformity. ****
- _____ 8. Use the appropriate criss-cross pattern tightening in numerical sequence for Rounds 1, 2, 3, and 4 (tightening all bolts once in sequence constitutes a “round”).

**** Note: Check LJJ gap around the flange circumference between each of these rounds, measured at every other bolt. If any gap is not reasonably uniform around the circumference, make the appropriate adjustments by selective bolt tightening before proceeding.

TARGET TORQUE (and 4 to 24-HOUR RE-TORQUE): _____ foot-pounds

_____ For 4-bolt, 8-bolt, 12-bolt Flanges

_____ For Large Flanges > 16 + Bolts

_____ Lubricate, Hand tight, Pre-tighten
_____ Round 1 – Tighten to _____ ft.lbs. (30%)
_____ Round 2 – Tighten to _____ ft.lbs. (60%)
_____ Round 3 – Tighten to _____ ft.lbs. (100%)
_____ Rotational (clockwise) Round

_____ Lubricate, hand tighten, Pre-tighten
_____ Round 1 – Tighten to _____ ft.lbs. (25%)
_____ Round 2 – Tighten to _____ ft.lbs. (50%)
_____ Round 3 – Tighten to _____ ft.lbs. (75%)
_____ Round 4 – Tighten to _____ ft.lbs. (100%)
_____ Rotational (clockwise) Round

_____ Rotational (clockwise) Round: 100% of Target Torque. Use rotational clockwise tightening sequence, starting with bolt #1, for one complete round, and continue until no further bolt or nut rotation occurs at 100% of the target torque value for each nut.

_____ 4- Hour Re-Torque & Inspection :
Re-torque to target torque value using one or two sequence-rounds, followed by one rotational round at the target torque value.

Documentation Recorded By : _____ Date: _____

Joint Technician/Mechanic: _____ Date: _____

Tightening Sequence:

Number the bolts in rotation around the Lap-Joint Flange circumference in a clockwise order, beginning with the first bolt at the top in the nominal 12:00 position, the second being the next bolt to the right, the third being the next bolt to the right, etc, until all bolts are numbered sequentially.

Following the table below, tighten the given bolt number to the desired torque value for the given round of tightening as specified on the Torque Record Checklist.

TABLE 1 [refer to ASME Document PCC-1 for Bolt Sequences]

NUMBER OF BOLTS	CRISS-CROSS PATTERN TIGHTENING SEQUENCE
4	1-3-2-4
8	1-5-3-7 >> 2-6-4-8
12	1-7-4-10 >> 2-8-5-11 >> 3-9-6-12
16	1-9-5-13 >> 3-11-7-15 >> 2-10-6-14 >> 4-12-8-16
20	1-11-6-16 >> 3-13-8-18 >> 5-10-15-20 >> 2-12-7-17 >> 4-14-9-19
24	1-13-7-19>> 4-16-10-22>> 2-14-8-20 >> 5-17-11-23 >> 3-15-9-21 >> 6-18-12-24
28	1-15-8-22 >> 4-18-11-25 >> 6-20-13-27 >> 2-16-9-23 >>>> 5-19-12-26 >> 7-21-14-28 >> 3-17-10-24
32	1-17-9-25 >> 5-21-13-29 >> 3-19-11-27 >> 7-23-15-31 >> 2-18-10-26 >>>> >> 6-22-14-30 >> 4-20-12-28 >> 8-24-16-32
36	1-2-3 >> 19-20-21 >> 10-11-12 >> 28-29-30 >> 4-5-6 >> 22-23-24 >>>> >> 13-14-15 >> 31-32-33 >> 7-8-9 >> 25-26-27 >> 16-17-18 >> 34-35-36
40	1-2-3-4 >> 21-22-23-24 >> 13-14-15-16 >> 33-34-35-36 >> 5-6-7-8 >>>> 25-26-27-28 >> 17-18-19-20 >> 37-38-39-40 >> 9-10-11-12 >> 29-30-31-32
44	1-2-3-4 >> 25-26-27-28 >> 13-14-15-16 >> 37-38-39-40 >>>> 5-6-7-8 >> 29-30-31-32 >> 17-18-19-20 >> 41-42-43-44 >>>> 9-10-11-12 >> 33-34-35-36 >> 21-22-23-24
48	1-2-3-4 >> 25-26-27-28 >> 13-14-15-16 >> 37-38-39-40 >>>> 5-6-7-8 >> 29-30-31-32 >> 17-18-19-20 >> 41-42-43-44 >>>> 9-10-11-12 >> 33-34-35-36 >> 21-22-23-24 >> 45-46-47-48
52	1-2-3-4 >> 29-30-31-32 >> 13-14-15-16 >> 41-42-43-44 >> 5-6-7-8 >>>> 33-34-35-36 >> 17-18-19-20 >> 45-46-47-48 >> 21-22-23-24 >> >> 49-50-51-52 >> 25-26-27-28 >> 9-10-11-12 >> 37-38-39-40

The criss-cross bolt tightening sequence and multi-round tightening are necessary to counteract the flange / bolt elastic interaction.

TABLE 2

EXAMPLES OF ESTIMATED BOLT TORQUE TO “SEAT” HDPE FLANGE FACES:

The **engineer of record** is usually responsible for establishing each flange joint criteria, and performing the required calculations to determine the initial and residual torque values.

These estimated values are based on non-plated bolts and studs, using a nut factor of **K=0.16** for lightly greased bolts and nuts. The calculations uses a HDPE flange face seating stress of **1200-psi** as a minimum and **1800-psi** as a maximum, and assumes the flanged joint is between two HDPE flange adapters (in which the contact area is largest), without a rubber gasket.

NOTE: For bolting to ductile-iron pipe, steel flanges or butterfly valves, the flange face contact area is about half, so bolt torque for that flange pair will be measurably less (refer to Table #3).

IPS Nominal Pipe Size	LJF Bolt Diameter	Initial Number of Bolts	Minimum Lubed Torque (Ft-Lbs)	Initial Maximum Lubed Torque (Ft-Lbs)	Flange OD/ ID (Inches)
2"	0.625	4	23	35	3.9 / 1.94
3"	0.625	4	33	50	5.0 / 2.86
4"	0.625	8	33	50	6.6 / 3.68
5"	0.75	8	44	66	7.5 / 4.40
6"	0.75	8	50	75	8.5 / 5.42
8"	0.75	8	80	120	10.63 / 6.76
10"	0.875	12	80	120	12.75 / 8.79
12"	0.875	12	105	160	15.00/ 10.43
14"	1.000	12	180	270	17.50 / 11.45
16"	1.000	16	180	270	20.00 / 13.09
18"	1.125	16	200	300	21.12 / 14.73
20"	1.125	20	200	300	23.50 / 16.36
22"	1.25	20	260	390	25.60 / 18.00
24"	1.25	20	290	435	28.00 / 19.64
26"	1.25	24	290-	435	30.00 / 21.27
28"	1.25	28	290	435	32.30 / 22.91
30"	1.25	28	325	488	34.30 / 24.54
32"	1.50	28	425	640	36.50 / 26.18
34"	1.50	32	425	640	38.50 / 27.82
36"	1.50	32	460	690	40.80 / 29.45
40"	1.50	36	460	690	46.00 / 35.29
42"	1.50	36	460	690	47.50 / 37.06
48"	1.50	44	460	690	54.00 / 43.43
54"	1.75	44	560	840	60.00 / 48.86

NOTE: Uniform bolt pre-load (torque), without large “scatter”, is as useful as the target pre-load. Within the limits of the HDPE flange adapter, gasket, or metal LJF, higher pre-load is desirable. The higher the pre-load safely achievable, the more closely the assembly will behave like the theoretical model and seal well. Higher pre-load means that a given internal pressure will result in the least possible change in contact sealing pressure. Be consistent (avoid changes) with materials and tools when following written assembly procedures.

Train and supervise the bolting personnel. Tell the crew what is to be accomplished, why, and explain that good results are not automatically achieved. Skill and care are essential. Bolted Joint assembly is a technical skill that is not common in the construction and maintenance profession, being considered more like a specialty. There is no universally accepted testing, nor certification, of bolted-joint assembly mechanics. With no common training, certification, nor standards, it is no surprise there is +/- 25% variability in assembly torque. Specifications and instructions by the engineer, followed by trained mechanics, help to solve the dilemma.

(NOTE: Consult ASME Document PCC-1, Appendix A for training and certification of bolted joint assemblers)

TABLE 3

**Examples of Estimated Bolt Torque to “Seat” the HDPE Flange Face To
A Butterfly-Valve, Steel Pipe Flange, or Ductile Iron Flange.**

The engineer of record is usually responsible for establishing each flange joint criteria, and performing the required calculations to determine the initial and residual torque values.

These estimated liberally lubricated torque values assume the flanged joint connects one HDPE flange-adapter to a Butterfly-Valve or Steel Pipe flange of Schedule 40 ID, or a Ductile-Iron flange. For bolting to steel flanges or butterfly valves, the flange face contact area is just over half that of HDPE to HDPE flanges, so calculated bolt torque for this flange pair will be measurably less than the values listed in Table #2.

Dimensional flange data should be obtained for each case from the pipe flange suppliers, so as to be able to calculate the face contact area.

These estimated values are based on non-plated bolts and studs, using a K=0.16 for lightly greased bolts and nuts. These calculations use an HDPE material minimum and maximum compressive seating stress of **1200-psi to 1800-psi**.

IPS Nominal Pipe Size	LJF Bolt Dia. (inches)	Initial Minimum		Initial Maximum		HDPE Flange OD Steel Pipe ID (inches)
		Number of Bolts	Lubed Torque (Ft-Lbs)	Lubed Torque (Ft-Lbs)	Lubed Torque (Ft-Lbs)	
2"	0.625	4	22	32		3.90 / 2.067
3"	0.625	4	30	45		5.00 / 3.068
4"	0.625	8	30	45		6.60 / 4.026
5"	0.75	8	44	66		7.50 / 4.40
6"	0.75	8	44	66		8.50 / 6.06
8"	0.75	8	58	88		10.63 / 7.98
10"	0.875	12	58	88		12.75 / 10.02
12"	0.875	12	75	114		15.00 / 11.94
14"	1.000	12	140	210		17.50 / 13.13
16"	1.000	16	140	210		20.00 / 15.00
18"	1.125	16	140	210		21.12 / 16.88
20"	1.125	20	140	210		23.50 / 18.81
22"	1.25	20	160	240		25.60 / 21.25
24"	1.25	20	180	270		28.00 / 23.25
26"	1.25	24	180	270		30.00 / 25.25
28"	1.25	28	180	270		32.30 / 27.25
30"	1.25	28	180	270		34.30 / 29.25
32"	1.50	28	240	360		36.50 / 31.00
34"	1.50	32	240	360		38.50 / 33.00
36"	1.50	32	260	390		40.80 / 35.00
40"	1.50	36	310	465		46.00 / 39.00
42"	1.50	36	310	465		47.50 / 41.00
48"	1.50	44	310	465		54.00 / 47.00
54"	1.75	44	365	550		60.00 / 53.00

Train and supervise the bolting personnel. Tell the crew what is to be accomplished, why, and explain that good results are not automatically achieved. Skill and care are essential. Bolted Joint assembly is a technical skill that is not common in the construction and maintenance profession, being considered more like a specialty. There is no universally accepted testing, nor certification, of bolted-joint assembly mechanics. With no common training, certification, nor standards, it is no surprise there is +/- 25% variability in assembly torque. Specifications and instructions by the engineer, followed by trained mechanics, help to solve the dilemma. (Note: Consult ASME PCC-1, Appendix A)

APPENDIX A

Calculations, Considerations, and Guidelines

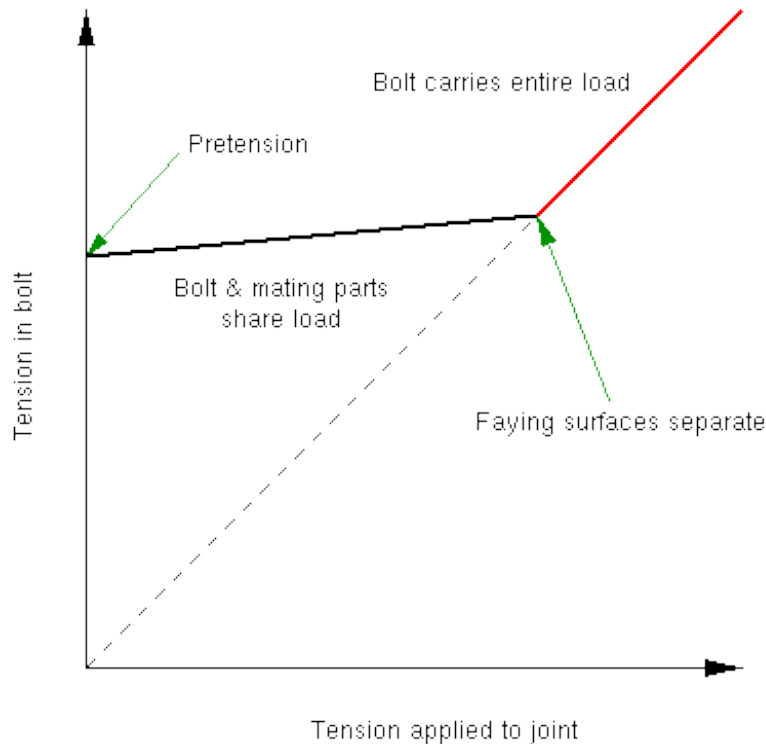


Figure 2

Bolt Pre-Load:

Within its proportional limit, the metal bolt tension is linearly related to the applied torque. A two dimensional graph plotting Total Applied Torque (y-axis) versus measured Bolt Load (x-axis) , displays a linear slope up-wards to the right. The bolt tension transmitted to the flange joint is directly dependent upon applied torque. The mechanical advantage of the torque-wrench lever and the helical threads enables one to stretch the length of the bolt between the head and the nut (this length is known and the *grip-length*), thus creating elongation (mechanical strain) resulting in tension stress in the bolt cross-section.

However, when using Torque-Control as the method for establishing flange assembly pre-load, one must understand there is a measurable variance between applied torque and theoretical bolt tension. Typically, only about 10% to 20 % of the applied torque is actually transmitted into bolt elongation. From tests, it is known that about 50% of the bolt torque is consumed by friction from the bolt-head contact face or the nut-face being rotated against its mating part. About 10% is used up in reversible twist of the bolt length. About another 30% is dissipated to overcome the friction in the bolt/nut threads. When more torque is needed to overcome friction, then less remains for bolt extension pre-load. Hence, small changes to reduce friction on the bolt-threads and under the rotating nut-face, will significantly increase the torque transmitted to bolt-extension pre-load. This is the reason a light duty grease or 30 weight motor oil should be sparingly applied to the bolt-thread and nut-thread before assembly. Metal or mineral filled lubricating greases are not usually used, because they may also enable nut loosening when subject to some vibration or repetitive pressure surge. The correct lubricant enables more bolt-

extension pre-load and bolt-torque retained thru residual friction at the final torque value. In ordinary practice, the bolt-head is usually held, and the nut usually rotated. It is good specification practice to specify which is to be held, and which is to be rotated, so as to minimize variability in bolt extension by applied torque.

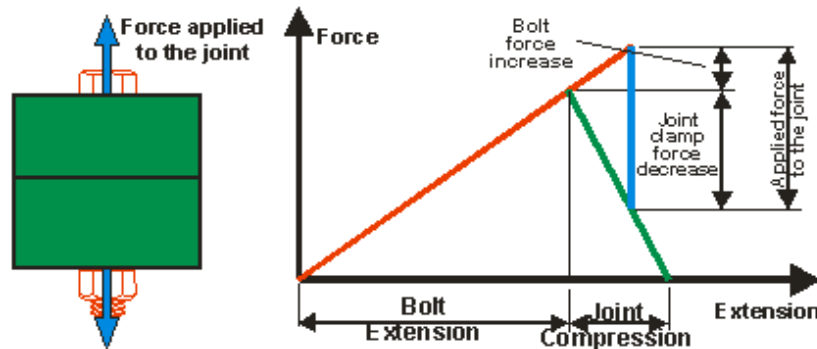


Figure 3

Bolting Basics : The Bolting Diagram

At zero pressure and no axial forces present in the pipeline, there is equilibrium between the elastic tension in the bolt and the compression in the HDPE flange. When the line is pressurized or is subjected to thermal contraction, the resulting axial force is applied across the joint and ultimately ends up being resisted by tension in the bolts. As the bolt elongates, part of the preload due to bolt torquing is reduced and the compressive stress on the mating flange faces (sealing surface) decreases. Because the HDPE flange was initially compressed, it elastically recovers and continues applying stress to the sealing surface. In this manner the HDPE flange is acting just like a gasket.

As the applied external tensile load is further increased, the bolts stretch more, thus relieving and further decreasing the compression at the sealing interface. If the flange face compression is relieved beyond the sealing force, the flange probably will leak. This decrease can only go so far, or the compression will ultimately go to zero, and there will be a gap between the sealing surfaces. The point of sealing surface separation is known as the “decompression point”. For pressurized pipelines, the external tension forces only need to decrease the pre-load down to a level near the operational working pressure, such that the working pressure exceeds the sealing pressure, and the water radially escapes / leaks.

From the diagram, it is obvious that the bolt-tension must be sufficiently high to endure external force loadings (pressure, surge, thermal contraction, beam-bending due to soil settlement, etc...), such that under all cases, the sealing pressure exceeds by a safety factor, the sum of the pipeline operating pressure plus surge pressure. Leaks will originate when the initially applied torque is not sufficient to pre-load the bolts to overcome external forces. Out of many possible contributing variables, low torque is usually the predominant, but not the only possible culprit when leaks appear.

Additionally, HDPE is a ductile, malleable material. Malleability is the ability of a material to exhibit large deformation or plastic response when being subjected to compressive force. Based on its compressive stress-strain curve, it has a compressive strength at a 2% offset strain of approximately 1600-psi, a compressive strength of approximately 2000-psi at a 3.5% offset strain, and a compressive strength of approximately 4000-psi at 6% offset-strain.

Hence, based on the sealing surface area and seal pressure, the “reverse computed” maximum bolt-load should impose less than 6% flange face compressive strain to maintain long-term, elastic, recoverable compression of the HDPE flange faces.



Figure 4 Cast Stainless Steel, Lap-Joint Flange, 6-inch IPS

The metal, Lap-Joint Flange (LJF) is an elastic, resilient, flexible “plate-spring” engineered to work with HDPE flange adapters. When the bolts are torqued, the LJF flexes and applies a uniform compression to the flange adapters. Generally it is desirable to torque the bolts so that the average HDPE flange-adapter face thickness compression is in the 2% to 5% range. At this low level of strain, the HDPE flange face is elastically and recoverably compressed, such that when subjected to thermal pipe contraction, or vibration, or bolt stretch, the HDPE flange face recovers slightly so as to maintain the required minimum level of (pre-load) interfacial sealing pressure.

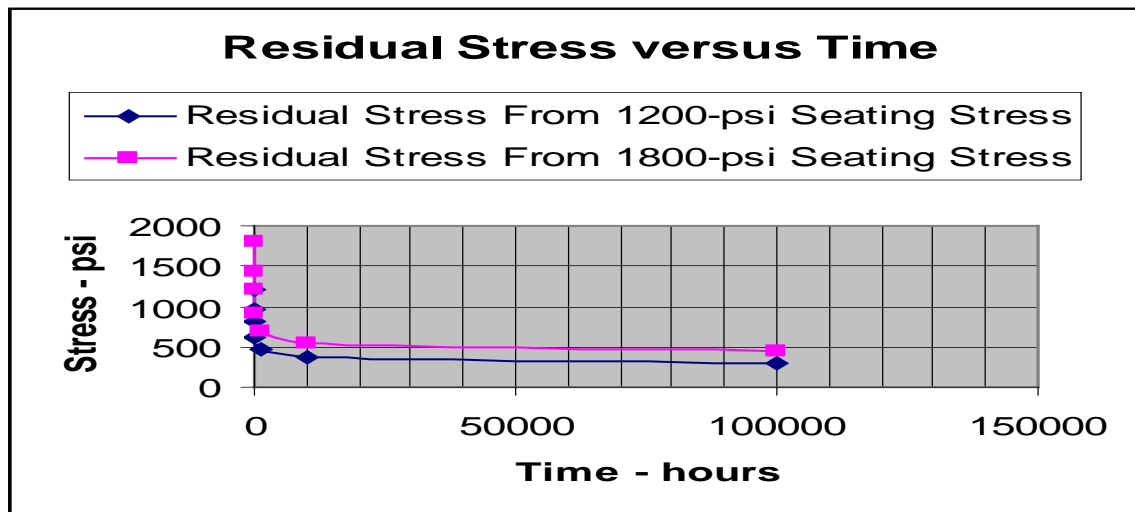


Figure 5: Residual Seating Stress versus Time, at 73° F

As shown in the Figure-5 (above), HDPE exhibits a low level of stress relaxation over a long time (creep) at 73°F, such that the residual compressive stress diminishes to an asymptotic value of approximately 35% of the initial interfacial stress. For example, at 1800-psi HDPE initial compressive seating-stress at 73°F, the initial bolt-torque will provide a long-term residual compressive interfacial sealing pressure of approximately 630-psi. This provides a residual sealing pressure sufficient to seal against 200-psi working pressure plus a 100% surge over-pressure. In the past, metal Lap-Joint Flanges have been initially torqued, followed by a 24-hour waiting period, followed by a re-torquing to the same initial value to compensate for slight compressive creep. For this assembly technique, the total compressive HDPE flange-face strain is the sum of the first compression strain plus the second compression strain. This is labor expensive and time consuming. By properly torquing to a higher initial value that immediately produces the same or greater total compressive flange-face strain, the time delay is eliminated, and the same or greater residual sealing stress is provided.

The residual sealing stress can be converted into bolt-torque for all flange sizes through simple mathematical formulae, as will be discussed.

- Flanged Ductile Iron Fittings (ASME B16.42; AWWA C110 & C153) are joined to HDPE pipe (AWWA C906) by means of the HDPE flange-adapter using the metal LJF. The HDPE flange-adapter provides the sealing surface. The metal LJF evenly distributes the compressive load from the bolts through the HDPE flange-face onto the sealing surface. The Ductile Iron Flange Fittings have the same ASME B16.5 Class-150 bolt-hole circle and number of bolts, as the HDPE metal LJF. The bolts should be alternately and evenly torqued, in four incremental stages, to impose about 2% to 5% compression of the HDPE flange face thickness. (Refer to Check- Sheet, Bolt Sequence: pgs. 9, 10, 11, 12)

For traditional metal-to-metal, highly rigid, standard steel, or ductile-iron flanges, elastic gaskets are required to seal the small metal strain generated by thermal expansion and contraction. The elastic gaskets preserve the rigid metal flange sealing stress by allowing them to thermally “move”.

High Density Polyethylene is a compressively elastic material at small strains of less than 8%. It elastically displaces; It does not volumetrically compress. The hardness of HDPE is about 65 Shore D, slightly harder than some rubber, or Teflon gaskets. The thick face of the HDPE

flange adapter enables the user to compress the flange face, through bolt torque, such that the flange face is elastically compressed. For example, a 5% squeeze on a 1" thick flange face is about 0.050". For a 3.5" thick flange-face, the elastic compression strain is 0.175-inches. This strain is the approximate thickness of a traditional elastic, resilient, reinforced rubber gasket. As the HDPE thermally strains, the flange face compression compensates for the thermal strain and maintains an elastic sealing stress greater than the operating pressure. Hence un-damaged HDPE flanges remain sealed without gaskets. The HDPE flange face is a compliant sealing material at less than 8% compressive strain.

NOTE: Some users specify sealing gaskets, based on their experience with metal flanges, but gaskets are not necessarily required for HDPE flanges at temperatures less than 140° F when the LJF is properly aligned, torqued, and the flange-adapter face is un-damaged. However, when the pipeline designer actually specifies gaskets, full-face gaskets are recommended, not the smaller ring-gaskets, as the full-face gasket bolt-holes provide for proper centering and alignment during flange-joint assembly. Gasket diameters should match the HDPE flange face OD & ID.

The contractor should comply with the torque recommendations of the specifying engineer. Alternately, the compression can be calculated by measuring the flange face thickness, torquing the bolts evenly, and re-measuring the flange face thickness, and then computing the %-compression by dividing the original flange face width by the squeezed flange face width, subtracting 1.0, then multiplying by 100 to give the % compression. Once this is done, identical flange sets can be torqued to the same value which gave the original percentage compression to seal effectively.

Because the HDPE pipe has a thicker wall and smaller ID than the Ductile Iron or Steel flanged fittings (larger ID), the metal to plastic flange face contact area is less than that of two HDPE flange-adapters being bolted face-to-face. Detailed dimensional flange data can be obtained from the HDPE flange adapter and metal flange manufacturers. To achieve the required seating stress over a smaller contact area, a measurably lower total bolt load is specified. This calculates as an obviously lower torque, proportional to the reduced sealing contact area. This compensation will still impose the HDPE flange adapter to metal flange compressive strain in the nominal 2-5% range. For this and other reasons, the bolt torque must be calculated for each installation and for each flange pair with differing sealing surface area. Refer to the torque examples presented in Tables #2 and #3, on pages 10 and 11.

COMPUTATIONAL MODEL for BOLT LOAD and BOLT TORQUE:

The Total Bolt Load is governed by the larger of either the sum total of external loads (Eq #1), or the gasket seating load.

$$F_{\text{Total}} > \{ F_{\text{Press}} + F_{\text{Surge}} + F_{\text{Therm.Contr.}} + F_{\text{Pipe-Bend}} + F_{\text{Flng-Alignment}} + F_{\% \text{ Variance}} \}$$

Equation # 1

Looking at the first two components of Equation #1, the hydraulic thrust force working to separate the flanges is equal to the working pressure rating (WPR) plus expected or allowable surge pressure, multiplied by the pipe's ID bore area. Assuming a WPR of 1.0 and an allowable surge of 1.0 x WPR, the sum is 2.0 x WPR. This is the Minimum Required Tightness (MRT). The MRT is multiplied by a Design Factor (DF) of about 1.75 to assure long term sealing and cover the last three variables noted in Equation #1 (thermal contraction should be

covered separately). The design-factor also covers variability in applied torque, elastic interaction between adjacent bolts (bolt-cross-talk), flatness of flange faces and LJF's, flange angular alignment, variability in the tightness factor, etc... The DF "tightness factor" is a measure of the inter-active scatter in the clamp force between bolts as a result of torque method used, calculated as the ratio of the max-tension to min-tension. This design factor (DF) for compression of the deformable HDPE is equivalent to the gasket "m" factor (the maintenance or multiplier factor). This larger value is defined as the Assembly Required Tightness (ART).

$$\mathbf{ART = 2.0 WPR \times 1.75 DF = 3.5 WPR} \qquad \mathbf{Eq \# 2}$$

To compensate for long-term stress relaxation, that force is then divided by approximately 0.35 (relaxation to 35% of the initial stress) in the HDPE polymer flange. (3.50 WPR / 0.35 ~-> 10 WPR) This provides the much higher initial bolting/ sealing force, which will diminish over time, down to the required residual long term force of: 3.5 WPR.

Thus, for polyethylene flanges, to seal against fluid pressure, the short-term, immediate (prior to stress relaxation), initial Minimum Operating Bolt Load is determined by :

$$\mathbf{MOBL = 10 \times WPR \times (Area \ of \ HDPE \ Pipe \ ID)} \qquad \mathbf{Eq.\# 3}$$

Alternately, the Minimum Seating Force (MSF) can be computed using the Design Seating Stress (DSS) gasket factor "y", which is the compressive stress required to "seat" and deform the gasket material into the imperfections and irregularities of the mating flange joint surface, to establish a no-leak-path seal. Even if the bolt load is sufficient to provide hydraulic sealing, the fluid can still drip-leak if the HDPE flange face is not seated to conform into all seal-surface imperfections.

The HDPE Design Seating Stress, at 2% to 5% strain, at less than 100°F, is in the range of 1200-psi to 1800-psi. The flange-face Minimum Seating Force (MSF) is calculated as the product of the initial design seating stress times the contact surface area. (NOTE: When seating against a ductile-iron or steel flange, the contact area is less than when bolting HDPE flanges to HDPE flanges. This reduced flange face contact area dramatically lowers the required torque.

$$\mathbf{MSF = 1800\text{-psi} \times Area \ of \ Interfacial \ Contact} \qquad \mathbf{Eq \# 4}$$

Note: Usually, for HDPE flanges, the Minimum Seating Force, MSF, slightly dominates over the Minimum Operating Bolt Load (hydro-dynamic pressure separation force), but both must be checked.

The required tensile Force per bolt, "F_b", is calculated by dividing THE LARGER of the Minimum Seating Force (MSF) , or, the Minimum Operating Bolt Load (MOBL), by the number of bolts, "n".

$$\mathbf{F_b = MSF / n} \quad \mathbf{--or--} \quad \mathbf{F_b = MOBL / n} \qquad \mathbf{Eq \# 5}$$

The applied Torque-per-bolt is calculated from the required Force-per-bolt, F_b:

$$\mathbf{T_b = (K d F_b) / 12} \qquad \mathbf{Eq \# 6}$$

Where : T_b = Torque per bolt, in foot-pounds.

F_b = desired tensile clamp Force-per-bolt, in pounds.
 d = nominal diameter of the bolt (major diameter, OD), inches.
 K = Nut Factor: for friction, material, lube, coatings, etc.

This mathematical relationship is based on the provision that clean, heavy-nuts on clean SAE J429 Grade 2, or Grade 5, bolts (with rolled threads) are used.

The following nut-factor K values apply:

Dry (no lube & no plating) mid-size steel bolts	K = 0.20
Non-plated "black" finish –or- stainless steel ...	K = 0.30
Lightly rusted bolts and nuts	K = 0.30
Zinc Plated	K = 0.25
Cadmium Plated	K = 0.20
Oil or Grease Lubricated.....	K = 0.15 to 0.18
Copper or Moly based Grease / Paste	K = 0.13

Note : Thirty additional K nut-factors and ranges of K values can be found in Reference #1, pages 231 and 232.

TABLE 4 : Illustration of the Relationship between:

Bolt Diameter :vs: Approx. Torque :vs: Approx. Load :vs: Tensile Stress

This table is meant to illustrate the relationship between bolt diameter, torque, load and bolt-stress. These are not maximum torques nor maximum bolt tensile stresses. In some applications, bolts may be torqued to 80% of their yield strengths, far above values shown here.

<u>Nominal Bolt Diameter</u>	<u>Threads Per Inch</u>	<u>Root-Area of Thread sq.-in.</u>	<u>Initial Torque Foot-Pounds</u>	<u>Approximate Tension Load in Pounds per Bolt.</u>	<u>Approx. Bolt Stress - psi</u>
5/8"	11	0.202	40	4,000 lbs.	19,800-psi
3/4"	10	0.302	100	9,200 lbs.	30,500-psi
7/8"	9	0.419	140	11,000 lbs	26,250-psi
1"	8	0.551	240	16,200 lbs	29,400-psi
1-1/8"	8	0.728	260	16,000 lbs	22,000-psi
1-1/4"	8	0.929	380	21,200 lbs	22,800-psi
1-1/2"	8	1.405	600	31,600 lbs	22,500-psi
1-3/4"	8	1.980	700	27,700 lbs	14,000-psi
1-7/8"	8	2.304	800	27,650 lbs	12,000-psi
2"	8	2.652	900	32,500 lbs.	12,200-psi

Note : The bolts are elastically stressed at a fraction of their yield stress.

GENERAL CONSIDERATIONS:

THE POLYETHYLENE FLANGE ADAPTER:

The HDPE flange adapter is typically made using PE3408 or PE4710 pipe grade polyethylene resin with an ASTM D3350 material property cell-classification of 345464C, or better. The flange adapter has a nominal flange face OD (hub OD) equal to the diameter of the standard raised-face Class 150 dimension for raised-face steel flanges (ASME B16.5 & B16.47). The flange face thickness typically is at least 1.25 times the nominal pipe wall thickness, and usually not thicker than 1.5 times the pipe wall thickness. The radius from the flange's back-face to pipe OD is usually a minimum of 3/8" for 2" to 12" IPS; and 1/2" or larger for 14" to 54" IPS sizes. The fusion-end wall-thickness is nominally 10% thicker than the pipe-wall to which it will be fused, to compensate partially for pipe toe-in (in addition to "facing back" most of the pipe toe-in); and to provide the potential for 100% pipe-wall fusion-contact. The HDPE flange adapter is normally rated at the same working pressure rating (WPR) as the nominal wall thickness pipe to which it will be fused. The neck of the flange adapter is sufficiently long so as to fit in a fusion machine and provide for fusion joining twice. Stub-end flanges may require a "holder" for use in the fusion machine. This technical note applies equally to serrated face and flat face HDPE flange adapters, when the minimum seating stress is met or exceeded, and when used with or without gaskets. Testing of the self-gasketing properties of HDPE was done at temperatures less than 100°F. Flange adapter manufacturers have internal standards regarding flange face flatness, parallelism between the seal face and the LJF back-face, seal face angular alignment to the theoretical bore centerline, etc. If questions arise regarding such technical issues, then the flange adapter manufacturer should be consulted.

THE METAL LAP-JOINT FLANGE:

Lap-Joint Flanges can be cut from carbon steel plate, radiused, and drilled to the required bolt-hole pattern. The metallic, contoured cross-section LJF is cast from ductile-iron or stainless steel. The cast Ductile-Iron is normally ASTM A536 Grade 65-45-12. The cast Stainless Steel is normally ASTM 351 Grade CF8M (#316 stainless). The LJF OD, bolt-hole diameter, and bolt-circle dimensions conform to the ANSI /ASME B16.5 Class 150 dimensional patterns and specifications for diameters 3/4" thru 24" nominal pipe sizes (IPS & DIPS), ASME / ANSI B16.47 Ser. A - CL 150 for diameters 26" to 54", and, B16.1- CL 125; AWWA C207 - CL 150 / B, D & E flanges. The surface finish may be Plain, Painted, Hot-Dipped Galvanized, Aqua-Armor™ coated, or otherwise corrosion protected. The LJF thickness should be sufficient in stiffness to provide the high initial seating stress, and flexible enough to deform to provide a long-term working pressure rating in excess of the operating plus surge pressures of the HDPE pipe system, with a reserve safety factor. The LJF ID must have a chamfer or radius that approximately matches the crotch radius of the HDPE flange adapter.

(Note: Aqua-Armor is a registered trademark of S & B Technical, Inc. Fort Worth, Texas)

THE HEAVY HEX NUTS:

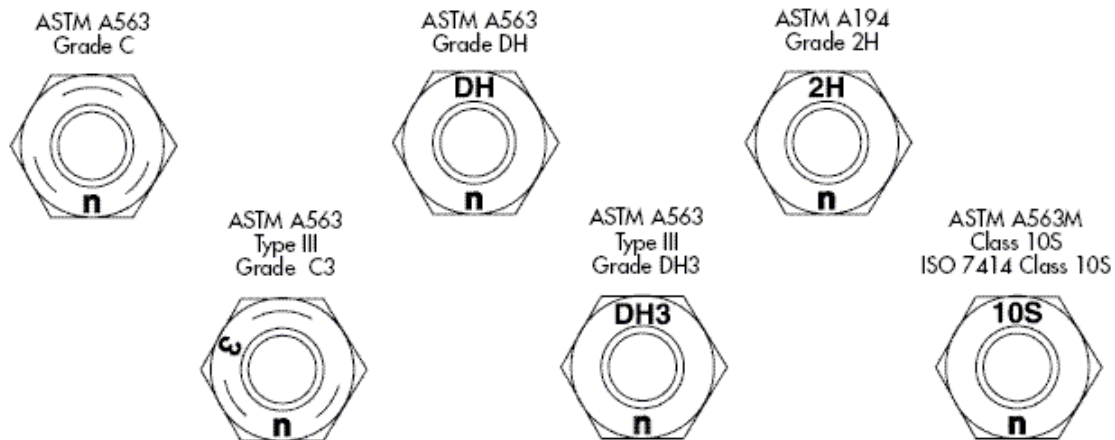
Nut strength is designated by proof strength or proof stress. Nuts should be selected such that the proof stress is equal to or greater than the tensile strength of the mating bolt or stud. When properly selected for compatibility, bolts and studs usually yield well before the nuts deform. Typically the carbon steel nuts are at least 100-ksi min-proof stress heavy-hex-nuts (not finish nuts) per ASTM A563 Grade A (standard specification for carbon and alloy steel nuts). The nuts are designed to be slightly softer than their matched grade of bolt. At full torque, the first few threads of the nut take most of the load, and thus yield into the mating bolt threads. After one or more uses, the nut thread will not match the bolt thread due to distortional flow of the nut metal, such that the nuts should be replaced when re-connecting a critical connection.

Corrosion proof nuts are available with coatings. Flange assembly corrosion proofing (nuts & bolts) may also be applied after assembly.

Grade Identification Markings : http://www.americanfastener.com/technical/grade_markings_steel.asp
<http://www.labsafety.com/refinfo/ezfacts/ezf269.htm>

Figure 6

Nut Identification and Grade Markings



THE HEAVY HEX-HEAD BOLTS and ALL-THREAD ROD:

Typically the carbon steel Heavy Hex Head Bolts or all-thread rod should possess at least a 55-ksi min yield strength. The Heavy Hex Head Bolts may be SAE Standard J429 Grade 2 or Grade 5 ; The Heavy Hex Head bolts may be ASTM A325 Type 1 (or 2,3), ASTM A449, or stronger. The All-Threaded Rod may be ASTM F1554 Grade 55, ASTM A36, or stronger. Heavy Hex-Head Bolt dimensions are normally in compliance with ASME B18.2.1. Heavy hex nuts are used for bridge across the 1/8" clearance between the bolt and flange hole. The heavy hex head maximizes the bearing-load surface area under the head so as act like a "washer"; and matches the same wrench size for the heavy hex nuts. Corrosion proof materials and coatings are available.

It is recommended that the all-thread rod be cut at least one to two "rod diameters" longer than the minimum overall length, so that the all-thread stud length is sufficiently long to provide ease in assembly, and protrude at least one full thread beyond the face of the two nuts on each side of the flange assembly when made-up at final torque. Studs are normally of equal length on a single flange. Refer to ASTM F704: Standard Practice for Selecting Bolting Lengths for Piping System Flanged Joints. _

Note: For pressurized connections, one cannot simply substitute any all-thread rod for a headed bolt without a significant loss in stud-strength, unless the grade of all-thread rod specified meets the minimum 55-ksi yield strength.

Bolt Identification Markings: http://www.americanfastener.com/technical/grade_markings_steel.asp
http://www.akrongear.com/bolt_head_identification.htm

WASHERS:

The Lap-Joint Flange's bolt-hole has an ID about the same diameter as that of the standard washer, thus precluding the absolute need for washers. Washers are typically used with 100-psi WPR, or higher WPR LJF's when the LJF's are powder-coated or galvanized, as washers prevent galling of the LJF top-plate by the nut rotation. The contact face of the nut should be lightly greased, just like the threaded bolt or stud. Heavy-Hex nuts have an enlarged contact surface approximating the diameter of washers.

Note : The common flat washer is made of soft metal. When used with a high strength bolt, it will be virtually impossible to achieve, and then maintain, the desired pre-load in the flanged connection due to cold-flow of the soft metallic washers. Only SAE, through-hardened (not case-hardened) heat-treated washers are normally considered. SAE washers give the smallest ID and acceptable OD.

If the application is critical, if there is frequent thermal cycling, or if the flange cannot be accessed again for re-torque if required, then conical (Bellville) washers may be considered as a helpful aid to torque retention. They act as a very stiff spring to lessen the effects of potential torque loss.

LUBRICANT:

PPI members recommend applying a thin layer of light grease or oil on the threads of the bolts and nuts, the nut face, and around the bolt-hole, as well as using the correct nut-factor (k) inserted into the torque calculation (Equation #6).

(Refer also to the example torque calculation beginning on Page 11).

Silver / copper / moly / metal-paste-lubricants are not as strongly recommended, as they lower the friction required to engage the nut, and may even enable reverse nut rotation (self-loosening) when subject to pipeline flow-stream vibration.

TORQUE WRENCH:

The Torque Wrench calibration should be recent (within in the last 4 months). The working capacity of the torque wrench is normally broad enough such that the required torque is in the middle 60% of its torque range. Beam type torque wrenches or adjustable torque wrenches are acceptable. The adjustable torque wrench is set to the specific torque value. When the torque value is reached or exceeded, the adjustable torque wrench releases and further handle rotation does not add torque to the bolt. Precision wrenches are required to do a proper job of uniform torque control during flange bolt-up.

Note: The least accurate torque control method is a hammer-wrench. The next is the air-wrench, then the torque-wrench with extension. The proper size torque-wrench is the most widely accepted. Better yet is hydraulic-wrench torque control. And the best yet are the micrometer or ultrasonic bolt-stretch measurement, or hydraulic bolt-extensioner.

THE GASKET:

IF gaskets are to be used, the gasket material should be chemically and thermally compatible with the internal fluid and the external environment. It should be of the appropriate thickness, hardness, style, and should be recommended by the gasket material manufacturer for use with polyethylene pipe flanges. Upon seating, a gasket must be capable of overcoming minor alignment and flange imperfections such as:

non-parallel flanges *distortion troughs / grooves* *surface waviness* *surface scorings*

(When these imperfections are minor or limited, self-gasketing HDPE flanges are fully capable of sealing into and across such imperfections. Gaskets are not usually required for properly torqued self-gasketing HDPE flange assemblies, where the face of the mating flange and/ or HDPE flange adapter is un-damaged by partial width gouging across the width of the flange face.)

When gaskets harder than HDPE are used, the hard gasket seating stress may be in excess of the HDPE seating stress. When higher seating stress gaskets are used with mating metal flanges, the HDPE may seat on its side of the gasket but may not be able to seat the harder gasket into the metal on the other metal flange face. Hence, there is a limit on the gasket seating stress when other material gaskets are used.

(Note: IF the engineer-of-record specifies a sheet rubber gasket, it usually will be in the 60 to 75 Durometer - Shore A hardness, internally fabric-or-fiber reinforced to avoid radial extrusion from the flow-stream internal hydraulic pressure, and may be 1/8" or 3/16" thickness per the judgment of the specifying engineer. Obviously, the gasket material should be compatible with the flow-stream liquid across the expected temperature range and service life. The most widely accepted gasket design is the full-face gasket with bolt-holes, so as to hang, center, and avoid wrinkling of the gasket during installation. No lubricants , "dopes", nor sealants should be applied to the flange faces, nor gasket. If gaskets are to be used, the gasket designer should provide the specifying engineer the following information and explain how it was used in gasket selection: Material ; Thickness ; Hardness, Density ; Internal Reinforcement Type; P x T Factor : test data at 1/6" thickness (which reduces 25% to 30% for 1/8" thickness and reduces 40% or more for 3/16" thickness; ASTM F36 Compressibility; ASTM F36 Recovery; ASTM F38 Creep Relaxation : (graph or plot of compression–set versus time and sealing stress @ temp.); ASTM F152 Tensile Strength; Gasket Factors : "m", "y", and , if available : Gb, a, Gs ; and the ASTM F 104 Line Callout, or, ASTM D 2000 Line Callout)

Refer to: APPENDIX "C".

CORROSION CONTROL:

The HDPE flange adapters are corrosion resistant. The Stainless Steel LJF's are corrosion resistant. The Ductile-Iron LJF's are corrosion resistant by virtue of its oxidation layer forming over time. However, DI LJF's are not corrosion-proof, and may be painted, zinc dipped, FBE coated, Aqua-Armored™, or otherwise coated to enhance longevity. Corrosion resistant steel bolts / nuts can be specified, and, coated bolts / nuts may be used. Additional options may be cathodic protection, sacrificial anodes, mastics, tape wraps, shrink-sleeves, or encapsulation type devices and products. Refer to Appendix E

THE BOLTED CONNECTION:

If an HDPE Lap-Joint Flange Assembly leaks, there is a natural tendency to "blame the gasket" or "blame the flange". Leaks are much more complicated than that. There are two flange-adapter faces, two LJF's, perhaps a gasket, gasket centering, the bolt-size and grade, the nut grade, the flange-adapter face flatness and alignment, the interaction of the gasket and flange-adapter faces, proper bolt torque sequence, even and proper bolt loading, HDPE stress relaxation, flange-adapter face integrity (marred or dented), bolt self-loosening, low bolt-torque, and many other variables. The project engineer's written bolting specification should integrate all of these issues for clarity, proper installation, and long-term performance.

The **written bolting specification** should be supplied to the experienced pipeline contractor to assemble each different type of flange pair (ie: HDPE to HDPE; HDPE to Raised Face Weld-Neck Steel flange; HDPE to Ductile-Iron flat face valve, HDPE butterfly-valve-flange-adapter to metal butterfly valve: each with different contact areas and, thus, different torque requirements.)

Flange Face Inspection and Integrity:

The HDPE and Metal flange faces should be inspected to insure they are free from radial gouges across no more than 1/3rd of the face width. Some surface marring or denting is acceptable. The metal flange sealing faces should be free from rust, weld spatter, dirt, debris, etc. HDPE flange-adapter faces exhibiting surface marring or dents should limit such defects to less than 1/16" deep. (Sometimes, minor HDPE surface marring on flat-face flange adapters (not serrated faces) can be "flattened" by lightly striking the damaged area with a flat-faced 5-lb sledge-hammer to "work" the defect "flat".) The mating metal flange faces should be cleaned so as to remove preservatives, rust, corrosion, or old gasket material.

Alignment of the Flange Faces:

Align flange faces prior to bolting so that any gap is minimal. The mating flange faces should be aligned square and true.

As a general rule, the axial centerline off-set misalignment should not exceed 1/8" for smaller diameter pipes, up to 1/4" for large diameter pipes (24" to 54").

The angular misalignment of the flange-adapter face is usually limited to less than 0.005" per inch of diameter. For example: on nominal 12" diameter pipe, the flange-adapter faces can be touching on one side, with a tolerable gap of 0.060" on the other side; and for 48" pipe the flange-adapters can be in contact on one side with a tolerable gap of 0.240" on the other side. (1/16" per foot : 0.5%)

The tolerable axial gap between parallel flange-adapter faces should be zero under perfect circumstances. In imperfect installations, the axial gap should be less than 1/32" on small diameter flanges, and 1/8" on large diameter flanges.

The project inspector should record measurements of off-set, angularity, gap prior to bolt-up. Surface and above grade flanges should be supported properly to avoid beam-bending stresses in the pipe and flanged joint. Buried flanges connected to heavy appurtenances such as fire hydrants, valves, tanks, metal pipes, require a proper support foundation for the heavy component to prevent settlement with its resultant shear and bending strain on the flanged joint.

Measurement of Gaps:

During the first four rounds, take measurements of the gap between the Lap-Joint Flanges around the circumference in at least 3 to 4 places to validate that the flanges are being brought together evenly. The closure distance for each round should be about same for each position measured. The gap should be measured at four equally spaced locations for flanges with up to 8 bolts; at every other or every third bolt for flanges with more than 12 bolts. Record the gap position and gap closure distance after each rotational round. Retain this data with the Checklist on page 9. Analog or Digital calipers, linear scales, or other measuring devices are useful in measuring the gap distance.

Concentric Alignment: Flange Adapters & Butterfly Valves:

Align the LJF's to be reasonably concentric with the OD of the HDPE flange adapters. The weight of the LJF's will tend to cause them to "hang" eccentric with an un-even crescent contact area on the back face of the flange adapter. By snugging a few bolts first, the lap-joint flange can then be raised upwards and held concentrically in place by light bolt friction, so as to maximize, and make uniform, the contact area between the LJF and the flange adapter.

Butterfly valves require the rotating disk to be concentric to the HDPE flange-adapter. Typically the HDPE butterfly-flange is longer and ID tapered or beveled to accommodate the disk rotation. However, a ring spacer may be used to off-set a standard HDPE flange adapter sufficiently to enable disk rotation. After fitting the valve to the flange-adapter with light torque to frictionally hold it in place, the butterfly valve may be installed with the disk fully rotated to assist and help check valve alignment. Alternately, after lightly torquing and fitting the valve to the beveled HDPE flange adapter, operate the valve to insure full opening without interference. Re-align as required and fully tighten. This may require a crane to suspend the valve in concentric alignment while also centralizing the lap-joint flange with low torque. When both the valve and LJJ are concentrically aligned, proceed with full torquing to specification.

Proper Bolt Procedure and Bolt Sequence:

Table #1 gives the proper bolt sequences to use when torquing the bolts. Each bolt should be numbered to insure it is used in the proper sequence. Keeping track of the bolting sequence on large diameter flanges can be confusing. With large numbers of un-labeled bolts, errors and skipping will occur.

Torque Progression:

When tightening pipe flange bolts, the best even loading of the bolts, and the best even compression of the HDPE flange face, is achieved by progressing through several levels to the final torque value.

For pipe flanges less than 18" nominal pipe size, the rule of thumb is the 30-30 rule. The bolts are snugged up and the flange-adapter aligned flush with the mating flange. Begin by sequentially tightening the bolts to 30% of the final torque value. Return to the first bolt, add 30% more torque, and sequentially tighten to 60% of the final torque value. Lastly, return to the first bolt, and torque to the final torque value; followed by a clockwise rotational torque check on all bolts to insure they are evenly torqued at or above the specified torque value.

For 20" and larger nominal diameter flanges, the 25-25 rule applies in which the bolts are sequentially tightened in four (25%) stages, with a final clockwise torque check.

Residual Bolt Torque (RBT) & Mandatory 4-Hour Re-torquing

With time, the initial bolt torque will slowly decline to a residual level of about 35% of the initial bolt torque. This long term level of engineered torque is sufficient to seal the lap-joint flange assembly. The high initial bolt torque seats the HDPE flange-adapter face, and the residual bolt torque seals the flange face. This visco-elastic relaxation in torque is normal. The residual bolt torque (RBT) is the minimum torque necessary to provide the elastic HDPE face compression necessary to seal the pipe joint, with reserve included for surge pressure, bolt-tension scatter, and other variables. The high initial torque provides seating stress for no-leak path, with the residual bolt torque providing the long term sealing stress.

Re-Torque to Target Torque: The Polyethylene flange adapter and the gasket (if used) will undergo some compression set that decreases the bolt torque. About four hours or so after the first tightening to the target torque value, retighten each bolt's nut to the final target torque value. As before, retighten in the criss-cross pattern sequence and in small increments, followed by a final rotational round, to raise the torque back to its target value.

For pipes of diameters 12" and smaller, the re-torque after 4 hours is recommended.

For pipes of diameters 14" and larger, for environmentally sensitive, or for critical pipelines, a second re-torque is encouraged after an additional 4 to 24 hours.

In all cases, before pipeline and flange assembly burial, the criteria for residual bolt torque should be RBT not less than 35% of the initial target torque.

Checking RBT can be done by using a torque wrench, setting it at a low torque, and then trying to rotate the nut on the stationary bolt. Re-set to a higher torque and try again, and then again. When the nut slightly rotates while the bolt is stationary, the residual torque is then measured by the torque wrench.

Re-torquing after 4 hours to 24 hours compensates for partial seating of the plastic face and relaxation of the bolts, nut embedment, nut dilation, thread stretch, thread surface smoothing, torsional relaxation, bolt-creep, and initial gasket compression-set (if gaskets are used).

Safe Disassembly Procedures

When it is necessary to open an HDPE flanged assembly, special procedures must be adopted to insure there is no damage to the main components or personnel. The assembled flange is under tremendous compression. The resilient HDPE flange adapter face wants to recover to its pre-compression thickness. IF, one bolt is removed, its compressive load is transferred to the two adjacent bolts, increasing their tension by 1/3rd. If one more adjacent bolt is removed, the additional compressive load is transferred to the remaining two adjacent bolts, increasing their tensile load by 50%. Very quickly, one can see that un-screwing multiple bolts completely will over-load adjacent bolts causing them to be bent, or permanently stretched, or causing the metal lap-joint flange to be permanently distorted, or the HDPE flange-adapter face to be permanently distorted with a wavy face thickness with potential gaps upon re-assembly.

The correct disassembly protocol is to reverse the assembly process. Using the star pattern, rotate the nut, to un-screw it, by about 10 to 30 degrees (less than one-half of a flat on a six sided nut.) Repeat this two or three or several times more, until the assembly torque is gradually and evenly diminished, and the HDPE flange face is gradually and evenly loosened. Once the HDPE flange face is un-bolted, the HDPE flange-adapter should drop loose or pull free from the mating flange by its own self weight. DO NOT USE WEDGE TOOLS to separate the HDPE Flange Adapter from the mating flange, as such tools will damage the sealing surface area. DO NOT HAMMER the pipe wall to "shake" the pipe loose. Lifting straps may be used on the HDPE pipe a ways back from the flange to lift the pipe, changing its effective lay length, and causing the flange face to pull-back from the mating flange, and then lift up. Once loosened, the HDPE flange adapter's face seal surface should be protected from gouging or marring in a manner acceptable to the maintenance/project superintendent.

The un-bolted pipeline pipe invert should be cradled to bear the weight of the pipe, flange, and LJF. The LJF should not rest on the ground bearing the weight of the pipeline on the "thin" ID edge of the lap-joint flange.

The Nuts and Bolts should be removed from the ditch, cleaned and oiled, and examined to see if they may be re-used, as corrosion may have damaged the bolts. They may need to be replaced, as required, upon flange re-assembly. Rusty threads will dramatically reduce the deliverable bolt-load (sealing force) at equal torque compared to new, lightly oiled threads.

Warning: When working on pipelines that transport pressurized fluids, the contained energy may be dangerous to workers. Typically, a pipeline is depressurized before it is worked on so as to avoid injury in the event of a leak. Generally speaking, never tighten nor loosen a flange

joint while the pipeline is pressurized. Always de-pressurize the pipe section before tightening or loosening flange bolts.

Employers should develop, implement and enforce a written safety program which includes task-specific training and lockout / tag-out procedures; and employers should ensure that when more than one employee is exposed to hazardous hydrostatic energy, a procedure is in place for group lockout / tag-out. It is the responsibility of the management, engineering, and operations groups to insure such written procedures exist and are followed.

Hydro-Testing & Leak Closure Guideline

Normally, after initial torque and the optional 4 - 24-hour re-torque, a hydrotest is applied, usually to 1.5 times operating pressure or 1.5 times pipe working pressure rating. Experience has shown that if the above procedures have been followed, virtually none of the flange joints will leak. Refer to ASTM F2164 for Hydro-Static Testing Procedures.

If drip or spray leaks are discovered during hydrotest, the principle corrective action is to measure the existing bolt torque with a torque wrench, increase it by 10% to 15%, and apply that larger torque to the bolt(s) in the center of the leak, and to each side of the leak. Tighten, slightly-more, each bolt adjacent to those bolts. Repeat, slightly increasing the torque on the bolts neighboring the leak, until the leakage stops and the pipeline remains sealed. Do not loosen the bolts on a pressurized pipe system! However, if 150% of the specified torque value is reached and the flange assembly still leaks, stop the hydrotest, de-pressurize, and safely disassemble the flange joint. Something else is probably wrong

NOTE: Safety in the ditch or, around pressurized pipelines is of primary concern. Strategies for fixing leaking pipelines must always include the safety manager, and possibly the corporate OSHA representative to insure the maximum safety and the minimum chance of an injury or accident. Procedures should be sufficiently thought through and rehearsed, and re-checked by project management before performing the work-plan, so as to avoid accidents, injury, or even death.

APPENDIX B

Wrench Size Chart for: HEAVY HEX BOLTS and HEAVY HEX NUTS:

Nominal Bolt and Nut Diameter	Heavy Hex Wrench Size
1/2 inch	7/8" (0.875)
5/8 inch	1 1/16" (1.063")
3/4 inch	1 1/4" (1.250")
7/8 inch	1 7/16" (1.437")
1 inch	1 5/8" (1.625")
1 1/8 inch	1 13/16" (1.813")
1 1/4 inch	2" (2.000")
1 3/8 inch	2 3/16" (2.188")
1 1/2" inch	2 3/8" (2.375")
1 5/8 inch	2 9/16" (2.563")
1 3/4 inch	2 3/4" (2.750")
1 7/8 inch	2 15/16" (2.938")
2 inch	3 1/8" (3.125")

APPENDIX – C

GASKETS (Ref. 11)

This Technical-Note does not provide guidance on gasket selection. This Appendix is added to inform users about most of the parameters involved in gasket selection. Selecting gasket material for a particular application is not an easy task. Consult the gasket supplier for detailed recommendations, including “TAMP” and gasket parameters mentioned below. If gaskets are to be used, PPI members strongly recommend the flange assembly design engineer make an informed and documented gasket selection. Reference 11 is an excellent resource.

Not all gaskets are created equal. For example, some sheet-stock black rubber gaskets used in larger flanges are limited to an operating pressure of 80-psi or less. Some red rubbers are limited to an operating pressure of 100-psi or less. Some internally reinforced sheet rubber gaskets are limited to an operating pressure of 150-psi or less. Some micro-cellular, non-rubber gaskets are limited to 300-psi or less. Note: Be sure to include working pressure plus surge pressure in evaluating gaskets, along with a design or safety factor applied, and with all the design parameters previously discussed in Equation #1, plus those listed here below.

When gaskets are being evaluated, the flange assembly designer should evaluate gaskets that are able capable of sealing at the clamping pressure imposed on it (seating stress) , and also resist blow-out at this load level, without suffering excessive compression set. The gasket thickness should be no thicker than that which is necessary for the gasket to conform to the unevenness of the mating flange, which is defined by its flange flatness and flange warpage during use. It must have adequate conformability into the micro-surface of the mating flange to create frictional forces, and there-by resist radial motion due to internal pressure (blow-out).

Epoxy coated valves and HDPE have low friction coefficients; the gasket might need to be internally reinforced, or have been tested with HDPE/epoxy surfaces verify that it does not radially creep nor slip, due to the low surface friction.

Some flat gaskets for an HDPE flange to HDPE, or to Ductile-Iron, or to Steel flanges, are cut from internally reinforced elastomeric sheet rubber. The reinforced gasket and the flange and the bolts are interactive, with the gasket selection (seating stress) dominating the design. The gasket selection must be appropriate for the over-all design, as specified by the joint designer, or project engineer.

(Note: When gaskets are considered, calculations should be performed using the seating stress, blow-out resistance, crush resistance, and other performance values obtained from the gasket manufacturer. The seating stress for many rubber gaskets is limited to about 600-psi to 1200-psi. Hence the bolt torque is low; but thermal contraction forces and operating pressures may require a bolt torque in excess of the rubber seating stress to keep the flanges together, thus exceeding the crush strength or compression set of some rubbers. Consult with the rubber gasket manufacturer to know that the rubber gasket will sustain the total bolt load calculated in equation #1.)

Gaskets must have:

- ** Zero leakage over the gasket face
- ** Resistance to flow-stream fluids
- ** Compensate flange surface alignment
- ** Minimum loss of clamp load bolt torque
- ** Exhibit sufficient Elastic Recovery
- ** Possess Resiliency against bolt-load
- ** Sealability
- ** Macro-Conformability to accommodate flange distortion and waviness
- ** Micro-Conformability to cold flow into the irregularities of the mating surface.
- ** Zero leakage (no weeping) through the gasket
- ** Have anti-stick properties
- ** Be uniformly flat
- ** Handle thermal strain
- ** Creep Resistance to sustain sealing

Rubber gasket compounds are typically specified (cell-classification) by ASTM D2000.
Non-Rubber, non-metallics are typically specified (cell-classification) by ASTM F104.

The Typical Gasket Specification Sheet includes data on the following **TAMP** data:
(T= Temperature; A= Application; M= Material; P= Pressure)

Color & Density	Composition	Reinforcement
P x T value	Max Pressure	Temperature Range
Min Seating Stress & Max Compressive Stress	Fluid Resistance	Hardness – Shore “A” or Shore “D”

ASTM F35, Compressibility
ASTM F36, Recovery
ASTM F 37, Sealability of Gasket Materials
ASTM F38, Creep Relaxation
ASTM F152, Tensile Strength
ASTM F 145, Evaluating Flat-Face Joint Gasket Compression
ASTM F434, Method for Blow-out Testing of Preformed Gaskets
ASTM F585, Flange Gasket Leak-Rate versus “y” stresses & “m” factors
ASTM D395, Compression Set (Method B: constant deflection)
ASTM D2240, Hardness (Shore D)

Typical rubber compounds are:

Nitrile (Buna-N) – ----- NBR
Styrene Butadiene (Buna-S) – SBR
Polychloroprene (Neoprene) ---CR
Ethylene Propylene ----- EPDM, EPM, EPR
Isobutylene (Butyl) -----BR
Fluorocarbon (Viton) ----- FKM

The gasket is a flat spring in series with the bolts and deformable LJF (springs). The spring constant of the gasket can be combined with the other spring constants to plot a joint diagram for joint calculations. However, the softness of the gasket dominates the elastic behavior of the assembled joint. If gaskets are used, the minimum seating-stress, maximum crush stress,

extrusion resistance, and blow-out resistance are often the predominant criteria for determining MOBL and its resultant torque.

Rubber displaces; it does not undergo volumetric compression. During compressive loading, significant internal shear stresses develop. Sufficiently large shear forces can result in fracture (cracking) of the rubber matrix, such that the degree of compression must be limited.

The non-reinforced rubber gasket stiffness exhibits an initial elastic response at low compressive stress, followed by a visco-elastic response at intermediate loads, and finally its viscous response at high loads or long times. This means the gasket has large hysteresis and will eventually take a permanent set, thus allowing creep, stress relaxation and some degree of torque loss.

In some applications, or when using “softer” rubber, the entire gasket or certain locations on gaskets, may be subjected to compressive stresses of sufficient intensity to cause “extrusion” (compressive yielding). When not internally reinforced to restrain radial extrusion, the gasket’s Shape-Factor greatly affects the relaxation characteristics, especially for highly compressible materials. Some of the stress relaxation is derived from radial expansion or bulging. Thus, the greater the area for lateral expansion, the greater the relaxation. The Shape Factor is defined as the ratio of the area of the load bearing face to the area free to bulge.

$SF = \text{Area}(\text{load}) / \text{Area}(\text{bulge})$ For ring type flange gaskets, it is the area of the contact face divided by the bulge area, which is the gasket thickness (h) times the sum of the ID and OD perimeter. This mathematically computes to be:

$$SF = (OD - ID) / 4 h \quad \text{Equation \#7}$$

As the area free to bulge increases, the shape factor decreases, stress relaxation increases, the retained stress decreases, and the bolt-torque decreases.

The shape factor decreases with increasing gasket thickness, thus thinner gaskets are desirable. This must be balanced against macro-conformability. However, as some compensation, the clamp-area can be made as large as possible, based upon the sealing & seating stress requirements.

The dynamic of radial extrusion (compression yielding) is as follows: As a non-reinforced, higher compressibility gasket begins to radially extrude, the gasket has to become thinner, because the volume of the gasket is conserved. As the OD enlarges, the shape factor decreases, and the stress relaxation accelerates. Then, as the sheet-gasket’s ID radially moves outward and into the gap between the hard flanges, the flow-stream pressure begins pushing and wedging and shoving the gasket outwardly, further accelerating radial extrusion with further thinning and increasing stress relaxation of the compressive load. With decreasing bolt-load, the radial extrusion becomes even easier, and will occur over time.

Use of internally reinforced or micro-cellular gaskets is one preventative measure to inhibit radial extrusion and to maintain the seal.

Fabric molded into the rubber sheet artificially increases the macro-stiffness of the gasket, and provides sufficient resilience and resistance to radial extrusion, creep, and stress relaxation. The volume of rubber contained within relatively in-extensible or much stronger fibers or yarn, or wire, act like tiny cubes of confined rubber, such that the mechanical stiffness, strength, and overall properties are enhanced. However, some cloth fabrics subjected to high pressure may permit weeping radially from ID to OD. Some do not. Ask. Hence other gaskets should also be considered and evaluated, perhaps like the micro-cellular gaskets.

Typical fabric reinforcing materials may be Kevlar, cotton, steel wire, polyester, etc. The open area between woven threads, the number of layers of fabric, and the strength of the fabric will affect the gasket sealing and extrusion resistance. Again, slow “weeping” over time may be a concern.

HDPE flange adapters, subjected to sufficient torque, do not require gaskets, unless the sealing surface is damaged. When gaskets are used, re-torquing after 4 to 24-hours is strongly recommended, so as to compensate for gasket creep and compression set..

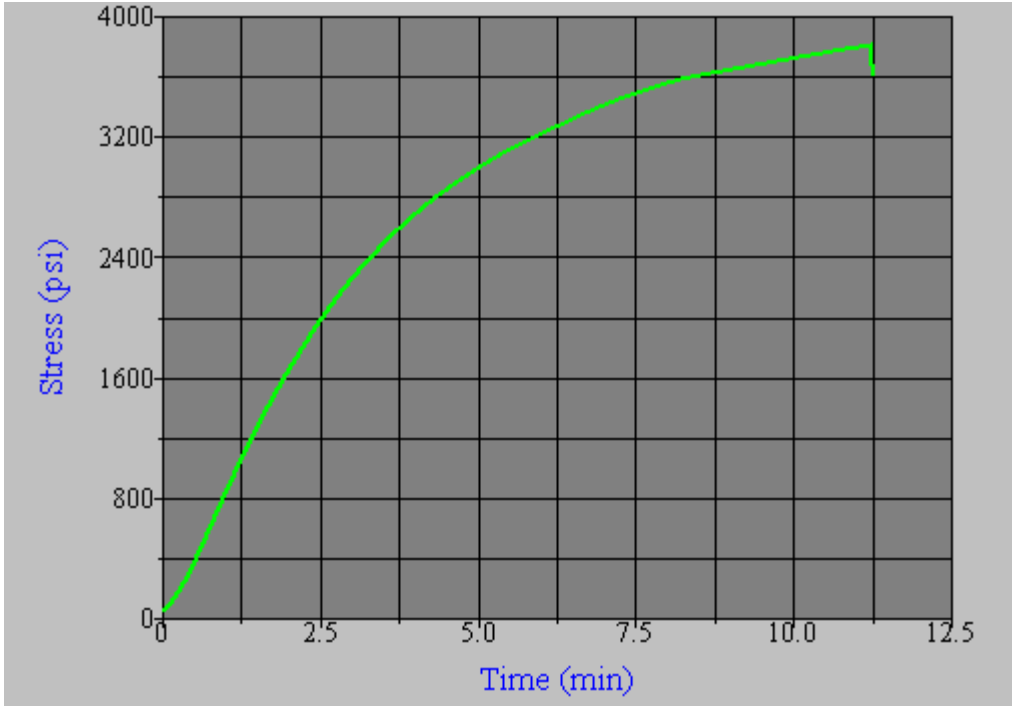
IF any gasket is used for sealing ductile HDPE flange adapters, the internally, fabric-reinforced, monolithic gasket typically is of a 60 to 75 Shore-A hardness, with low compression set, higher resilience, good mechanical tear strength, high sealability, low creep relaxation, moderate compressibility, a seating stress near to polyethylene’s compressive yield stress, with good deformation recovery, and good micro-conformability.

Micro-cellular Teflon faced gaskets (exemplified by Gylon 3545) may be an excellent choice.

Warning: When gaskets are thought to be required, or when specified as required, the flanged assembly design should not be finalized without independent evaluation for suitability of all components. Failure to specify the proper sealing products could result in personal or property damage. Consult with all component suppliers for their guidance and recommendations.

APPENDIX D

Typical HDPE Compressive Stress-Strain Curve



Note: Multiplying the compression rate (0.0068"/min) by time (above) gives the absolute strain. Because the sample is 1" thick, the absolute strain divided by the specimen thickness times 100 gives the percentage strain; stress divided by strain gives the apparent modulus at that point:

Time (minutes)	:	zero	1.25	2.5	3.75	5.0	7.5	10.0
Approx. Comp. Stress:		zero	1125psi	2000psi	2600psi	2975psi	3500psi	3700psi
Compressive Strain:		zero	0.014"	0.028"	0.042"	0.056"	0.084	0.112"
Compressive % Strain:		zero	1.4%	2.8%	4.2%	5.6%	8.4%	11.2%

APPENDIX E

CORROSION CONTROL REFERENCES

AWWA C116 / A21.16 TITLE:

Protective Fusion-Bonded Epoxy Coatings for the Interior and Exterior Surfaces of Ductile-Iron and Gray-Iron Fittings for Water Supply Service

(Fusion-Bonded Epoxy (FBE) Coating is a one-part, heat curable, thermosetting epoxy coating powder designed for corrosion protection of pipe and pipeline components in buried (only) service. FBE has a limited above ground UV exposure service life of nominally less than one year. NOTE: Only Fusion Bonded Polyester should be used above ground due to polyester's solar UV resistance.)

ASTM A123 / A123M-02 TITLE:

Standard Specification for Zinc (Hot-Dip Galvanized) Coatings on Iron and Steel Products

Glossary:

Adapter: A fitting used to join two pieces of pipe, or two fittings, which have differing joining systems. (i.e.: flange adapter)

Alloy Steel: a type of steel that contains another material that is added intentionally to alter the properties of the ferritic metal.

Assembly Required Tightness (ART): The total bolt-load necessary to insure a seal against flow-stream pressure when considering hydrostatic and hydrodynamic pressure and the variability of bolting efficacy under applied torque with elastic interaction between bolts considered.

Bolt Bearing Surface: The circular underside of the bolt-head that makes contact with the Lap-Joint Flange upper surface around the bolt-hole.

Bolt Stretch: The amount of tension in a bolt after the wrench has been removed. Bolt stretch determines the strength of the bolted joint.

Bolted Joint: Two pieces of metal joined together by the use of threaded fasteners.

Carbon Steel: A type of steel made of iron and carbon and no other elements.

Clamping Force: The compressive force that a fastener exerts on a joint.

Compressibility: the measure of the HDPE flange face's ability to deflect and conform to the mating flange face. This compressibility compensates for flange irregularities such as minor nicks, non-parallelism, metal corrosion, and variation in surface roughness or grooving depth.

Compressive Force: The force that occurs when opposing loads act on a material, crushing, or attempting to crush it.

Creep: the change in strain of a gasket under constant stress, (compression drift).

Design Factor: For flanges, it is the ratio of the maximum anticipated bolt load to the minimum anticipated bolt load; sometimes referred to as "scatter". It is used to insure the minimum load is applied to each bolt to insure a seal.

Flange Adapter: A device for mechanically connecting and sealing two pipe sections at full pressure rating. It is designed with a neck of pipe which is heat fused to the pipe main, and with a hub of larger diameter than the pipeline diameter. The hub face is the seating and sealing face for the joint. The hub OD fits just inside the bolts. Each plastic flange adapter must use a metal Lap-Joint Flange.

Grip Length: The length of the unthreaded portion of the bolt shank.

Head Style: The shape of the fastener head (i.e.: hex, socket, etc)

Hex Bolt: A type of bolt that has a head with six sides (flats, wrench-pads)

Hex Socket: A type of driving recess with a hexagonal indentation designed to accommodate a hex wrench.

Hydrostatic Test: A pressure test of a completed fabrication to confirm acceptable quality. Typically, the vessel, pipe, or system is filled with water, and held at the selected pressure while checking for leaks.

Identification Marking: The marking on a fastener or bolt or nut that often indicates the manufacturer, the material grade, and fastener capability.

Joining: The act of connecting two separate components of a pipeline system together.

Joint: A term used to describe an individual length of pipe; the actual joining mechanism connecting two pieces of pipe.

Lap Joint Flange Assembly: This is a two piece device consisting of: **1.** a polyethylene flange adapter (stub-end), with, **2.** a loose, metal, Lap-Joint Flange. The metal LJF cross-section geometry may be a rectangular solid, or a contoured cross-section. The rectangular cross-section typically is machined from metal plate; the contoured cross-section is typically cast using molten ductile-iron or stainless-steel. The LJF is typically in flat-face contact with the polyethylene flange adapter hub, and, by definition, has a radius on the contact side of the LJF ID which mates with the fillet radius of the matching polyethylene flange-adapter (stub-end). The LJF slips over the pipe; is not welded to the pipe; is loose until bolted; and is free to rotate into bolt-hole alignment with another flange. The bolt-load is transferred to the flange adapter sealing face by the pressure of the LJF against the back-face of the HDPE hub.

Minimum Required Tightness (MRT): The total load exerted by bolt extension in equilibrium to the force generated across the full bore pipe ID by the hydrostatic plus hydrodynamic flow-stream pressure. It excludes thermal or other external mechanical forces, such as pipe bending.

Minimum Operating Bolt Load (MOBL): the minimum total bolt load required to seal against the force of internal pressures plus external mechanical and thermal loads.

Minimum Seating Force (MSF): The total bolt load required to effectively compress the HDPE flange face (or gasket) so as to embed the HDPE flange face into all contours and irregularities of the mating flange, so as to provide an elastic mechanical compliance with no possible leak-path, and to provide sufficient sealing pressure when the pipeline flow-stream is hydro-tested and operating.

Proof Load: The applied tensile load that a fastener must support without evidence of axial deformation. The proof load is just at/under the bolt's tensile yield load.

Roughness: the irregularities in the flange face surface texture from production processes.

Slip On: Metal flange ring that is slipped over a shell or pipe, and back-welded to it.

Smooth Bore: The bore of the flange coincides with the ID bore of the shell or pipeline.

Stepped Bore: The bore of the flange is different from the ID bore of the shell or pipeline.

Stress: The measure of force distributed over an area, calculated in pounds per square inch.

Stress Relaxation: the change in stress “s”, on a gasket under constant strain; it is usually graphed as percent relaxation (ratio of retained stress versus initial stress) versus initial stress, for various gasket thicknesses at temp.

Surge Pressure: A transient pressure increase due to rapid changes in momentum of flowing fluids. Water-Hammer is one type of surge pressure.

Thermal Expansion (Contraction): The increase (decrease) in dimensions of a material (pipe) resulting from an increase (decrease) in temperature.

Thrust Force : The force or load resultant from momentum changes in direction of a moving column of fluid; The axial force developed at end closures like caps or valves, resulting from the hydrostatic pressure across the pipe bore area.

Torque: The multiple of a force applied to a lever arm, so as to force rotation of an object. It is usually expressed in foot-pounds.

Van Stone Flange Assembly: An alternate name for a two piece joining device consisting of a stub-end hub or adapter, with a loose, rotating, metal lap-joint flange. The hub OD is nominally equal to a “raised face” diameter. The LJF is contoured to match the hub adapter geometry. The LJF is slipped onto the flange-adapter prior to welding it to the pipe main. The LJF is loose until bolted, and is free to rotate for proper alignment to mating pipeline components. (The assembly is alternately called a lap-joint flange assembly.)

Water Hammer: Pressure surges in a pipeline system caused by sudden fluid velocity changes imposed by a pump, valve, or other component.

Waviness: that component of surface texture upon which surface roughness is superimposed; widely spaced repetitive irregularities; the combination of flange roughness and flange waviness is called "profile".

Working Pressure Rating (WPR): The maximum anticipated, continuous long term hydrostatic pressure (excluding dynamic surge pressure) a manufacturer recommends for a given pipeline component.

Yield Strength: The load at which a fastener experiences a specified amount of permanent deformation.

References

1. "An Introduction to the Design and Behavior of Bolted Joints" by John H. Bickford; 3rd Edition, published in 1995 by: Marcel Dekker, Inc. New York, NY
2. "Bolted Flange Assembly: Preliminary Elastic Interaction Data and Improved Bolt-Up Procedures" Bulletin # 408, published in 1996 by: Welding Research Council, New York, NY.
3. "Modern Flange Design: Bulletin #502" ; the 7th Edition was published in 1978 by: The Taylor Forge Corporation. PO Box 999, Southfield, MI 48037
4. "Formulae for Stress and Strain" by R.J. Roark, published by McGraw-Hill Book Company; Article 72, page #307 (Flanged and Bolted Pipe).
5. "ASME Unfired Pressure Vessel Code" Divisions I and II (1974).
6. "Discussions of Formulae for Stresses in Bolted Flanged Connections" Transactions of ASME , 60:627 (April, 1938)
7. "Flange Design", Bureau of Reclamation Memorandum (July 15, 1947).
8. "Theory of Plates and Shells", by S. Timoshenko, 2nd Edition published in 1959, by: McGraw-Hill Book Company, New York, NY.
9. F704-81(2001) Standard Practice for Selecting Bolting Lengths for Piping System Flanged Joints
10. American Society for Testing Materials (ASTM): Technical Committee F 16.96: Bolting Technology; ASTM International, West Conshohocken, PA 19428-2959
11. "Gaskets: Design, Selection, and Testing" by Daniel Czernik, McGraw Hill Publications, 19th Street, New York, NY 10011 ISBN-0-07-015113-x
12. ASTM F-46 Committee for Structural thru-Hardened Steel Washers
13. ASME Document #PCC-1: Guidelines for Pressure Boundary Bolted Flange Joint Assembly