Initial gasket compression is key to safe, reliable flange joints

Jim Drago, PE, Senior Manager, Marketing Intelligence, Garlock Sealing Technologies, Palmyra, New York, USA

This article provides guidance on how to prevent gasket blow-outs in flanged piping connections. It shows that blow-out resistance is not primarily a function of gasket tensile strength, but rather a number of other factors, including clamping force, flange surface texture, temperature, gasket creep resistance, joint rigidity and internal pressure. The text explores the impact of clamping force and also includes supporting calculations.

All gaskets, whether compressed fibre sheet, rubber, metal, corrugated metal, Kammprofile, spiral-wound or polytetrafluoroethylene (PTFE), are susceptible to failure if they are not properly installed in the pipe and equipment flanges.

Gasket manufacturers' application engineers fixate on adequately tightening, or "torquing" flange bolts. Tightening the bolts develops a compressive load on the gasket contact area that both creates a seal and retains the gasket in the flanges (**Figure 1**).

Integrity

A number of factors contribute to the integrity of a flanged joint. Maintaining a tight seal depends on the clamping force on a gasket, flange surface-finish, service temperature, flange rigidity and internal pressure.

Figure 1. Typical bolted flange (courtesy of GPT).

The clamping force of the bolts transferred through the flanges produces compressive stress (force per unit area) on the surface of the gasket. Different types of gaskets require different minimum compressive stresses in order to effect and maintain a seal.

Surface-finish determines how effectively flanges grip a gasket. It is especially important to bear this in mind when dealing with soft, sheet-type gaskets.

A serrated flange surface, as specified by ASME B16.5 section 6.4.5.3 will "bite" into a gasket – holding it in place and preventing it from splitting and shearing as it would with a smooth flange.

Temperature and gasket creep are directly related. When the temperature increases, flange bolts relax as their yield strength decreases.

Gasket materials become thinner without an increase in compressive load. Bolt relaxation and gasket creep effectively reduce the clamping force on a gasket, increasing the probability of a leak.

Flange rigidity affects how bolt load will be transferred to a gasket. Internal system pressure develops forces within a joint that concurrently work to pull apart the flanges and push out the gasket. **Figure 2** shows these forces acting on a flanged joint.

To determine which is more important to gasket performance – the compressive force or strength of the material – consider a hypothetical scenario based on the assumptions summarised below.

- Soft gasketing materials (such as fibre sheet with rubber binders, PTFE, flexible graphite or homogenous rubber) are used for this illustration. These types of gaskets do not have the high tensile strength or the hoop strength of a metal gasket.
- The flanges are perfectly stiff with no bending or deflections when the bolts are tightened. This allows all the force produced by the bolts to be transferred to the gasket.
- The gasket does not creep. This is for illustrative purposes only as in the real world this is not true.
- There are no temperature effects on bolts, flanges or gasket.
- Assume the flanges are smooth – that is, no serrated finish as specified by ASME B16.5. This makes it easier to characterise the force due to friction of the gasket on the flange surface.
- When the gasket blows out it will break in two places, 180° apart. In reality soft gaskets can expand radially until they snap or segments of 15° to 30° sheer out (**Figure 3**).

Balanced forces

These assumptions are simplistic, but they enable one to look at the interaction of clamping force, gasket strength and internal pressure in a flanged joint. **Figure 4** (on page 12) shows a flange joint broken down into free-body diagrams.

The forces produced by the internal pressure and bolts are balanced when a joint is on the verge of opening and blowing out the gasket. The sum of the forces in the *X* direction is 0 .

$$F_G + F_C - F_{PR} = 0 \qquad (1)$$

where:

$F_G = \sigma_{GS}2((D-d)/2)t = \sigma_{GS}(t(D-d))$

$F_C = \mu(F_B - F_{PH})$, with

$F_B = \sigma_G\pi(D^2 - d^2)/4$ and $F_{PH} = P\pi\, d^2/4$

and

$F_{PR} = P\pi dt$

Substituting these terms into **Equation 1**:

$$[\sigma_{GS}(t(D-d))] + [\sigma_G(\mu\pi(D^2-d^2)/4) - P(\mu(\pi d^2/4)] - [P(\pi dt)] = 0 \qquad (2)$$

Description	Nominal pipe size	Gasket tensile strength σ_{GS} (psi)	Contribution of gasket strength (psi) to P	Contribution of compressive stress on the gasket (psi) to P	P (psi) at the point of leak
Flexible graphite, σ_G = 4800	2	600	51	3075	3126
Metal reinforced flexible graphite, σ_G = 4800	2	4500	382	3075	3457
Flexible graphite, σ_G = 7500	2	600	51	4804	4855
Metal reinforced flexible graphite, σ_G = 7500	2	4500	382	4804	5187
Flexible graphite, σ_G = 4800	6	600	15	2260	2275
Metal reinforced flexible graphite, σ_G = 4800	6	4500	112	2260	2371
Flexible graphite, σ_G = 7500	6	600	15	3531	3546
Metal reinforced flexible graphite, σ_G = 7500	6	4500	112	3531	3643
Flexible graphite, σ_G = 4800	12	600	6	1541	1547
Metal reinforced flexible graphite, σ_G = 4800	12	4500	41	1541	1583
Flexible graphite, σ_G = 7500	12	600	6	2408	2414
Metal reinforced flexible graphite, $_{\sigma G}$ = 7500	12	4500	41	2408	2450
Flexible graphite, σ_G = 4800	24	600	2	1257	1260
Metal reinforced flexible graphite, σ_G = 4800	24	4500	18	1257	1275
Flexible graphite, σ_G = 7500	24	600	2	1964	1967
Metal reinforced flexible graphite, σ_G = 7500	24	4500	18	1964	1982
Flexible graphite, σ_G = 4800	60	600	1	968	968
Metal reinforced flexible graphite, σ_G = 4800	60	4500	6	968	973
Flexible graphite, σ_G = 7500	60	600	1	1512	1513
Metal reinforced flexible graphite, σ_G = 7500	60	4500	6	1512	1518

Table 1. The pressure at the point of leak for a variety of cases for different flange pipe sizes and gasket strengths (also see Figure 5).

Solving for P:

$$P = \sigma_{GS}[4t(D - d)/\mu\pi d^2 + 4\pi dt)] + \sigma_G[\mu(D^2 - d^2)/\mu d^2 + 4dt)] \quad (3)$$

where F_B is the force on the gasket from tightening the bolts (lb); $F_C = \mu(F_B - F_{PH})$ is the net radial force retaining the gasket in the joint (lb); F_G is the force caused by gasket tensile strength (lb); F_{PH} is the force that works to spread apart the flanges because of the internal pressure, known as the hydrostatic end-force (lb); $F_{PR} = P\pi dt$ is the radial force caused by the internal pressure working to push the gasket out of the joint; D is the outside diameter of the gasket contact area (inches); d is the inside diameter of the gasket contact area (inches); P is the internal pressure (psig); σ_G is the compressive stress developed by the bolts on the gasket contact area (psi); σ_{GS} is the tensile strength of the gasket (psi); t is the gasket thickness (inches); and $\mu = 0.2$ is the friction factor between the gasket and flange surface (the value is the factor used for graphite on a smooth steel surface).

Equation 3 gives the internal pressure required to open a flanged joint and cause a leak or gasket blow-out at a given flange size, compressive stress and gasket tensile strength.

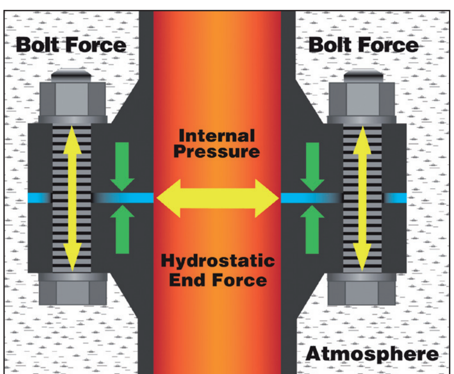


Figure 2. Forces acting in a bolted flange joint.

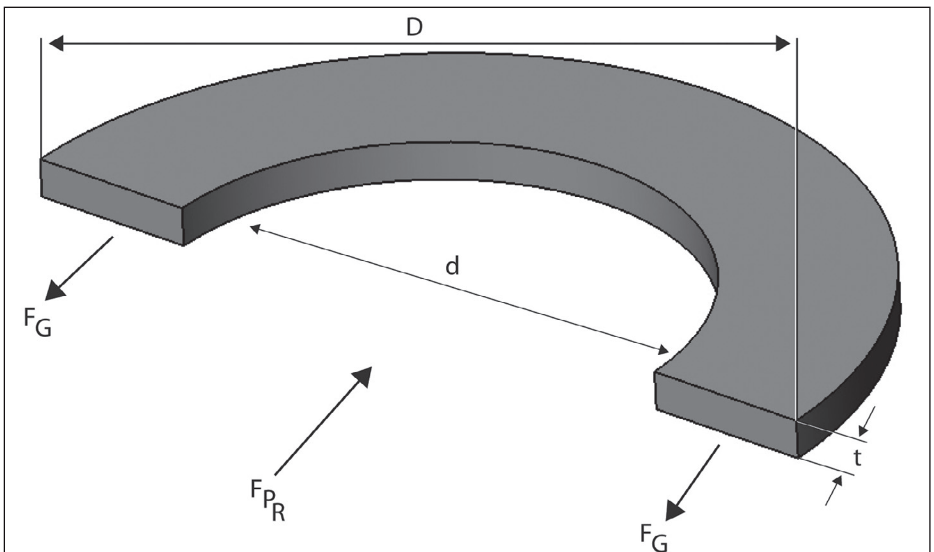


Figure 3. Hypothetical break point of the blow-out illustration (F_B is the force on the gasket from tightening the bolts; $F_C = \mu(F_B - F_{PH})$ is the net radial force retaining the gasket in the joint; F_G is the force caused by gasket tensile strength; F_{PH} is the force that works to spread apart the flanges because of the internal pressure, known as the hydrostatic end-force; and $F_{PR} = P\pi dt$ is the radial force caused by the internal pressure working to push the gasket out of the joint).

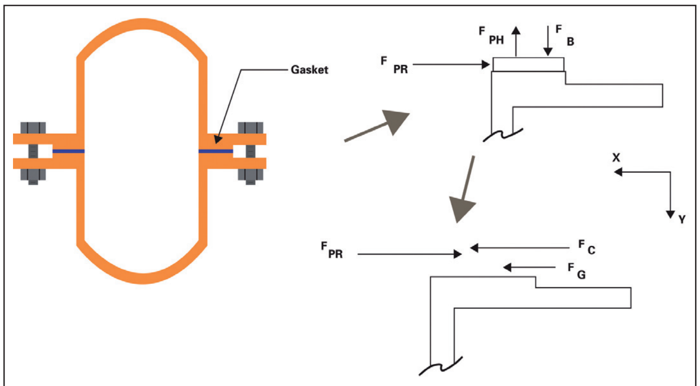


Figure 4. Development of free-body diagrams of a flange joint.

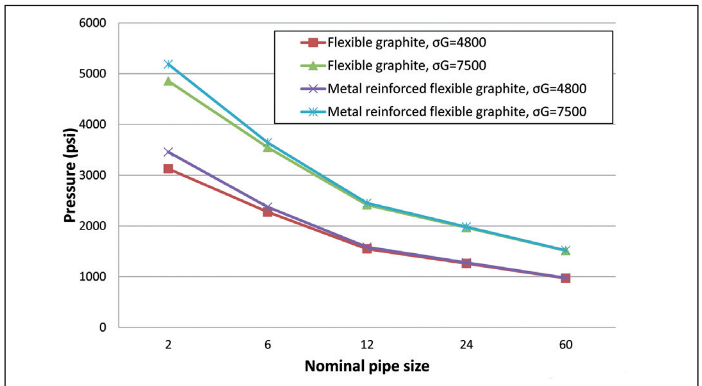


Figure 5. Pressure at leak/blow-out versus pipe size for varying compressive stress (also see Table 1).

Table 1 (on page 11) and **Figure 5** detail a variety of cases for different flange pipe sizes and gasket strengths.

Conclusions

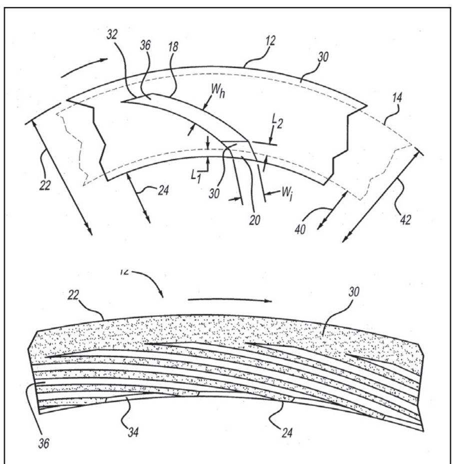
The calculations show that regardless of the tensile strength of a gasket the internal pressures that will cause it to blow out or leak are virtually the same at a given size and compressive stress. Note that the calculated amount of internal pressure causing the flange assembly to leak decreases as the flange diameter increases. This is due to the increased effect of the hydrostatic end force. In the case of soft-sheet gaskets, compressive stress and not the tensile strength of the material is the major determinant of flanged joint reliability.Therefore, gasket installation demands the utmost diligence. The use of torque wrenches and following gasket manufacturers' guidelines will go a long way to assuring reliable joints for improved worker safety and plant productivity.

Contact:
Jim Drago, PE, Senior Manager, Marketing Intelligence, Garlock Sealing Technologies,
1666 Division Street, Palmyra,
NY 14522, USA. Tel: +1 315 597 4811,
Fax: +1 315 597 3216,
Web: www.garlock.com

PATENTS

Hydrodynamic face seal

Applicant: **Eaton Corp, USA**
This disclosure relates to a hydrodynamic face seal. It comprises a rotating first ring and a stationary second ring. The first ring has an inner face. The stationary second ring also has an inner face, which is adjacent to the inner face of the first ring. The inner face of the first ring includes a groove that contains a fluid-inlet portion and a means of generating a hydrodynamic force. The fluid-inlet portion of the groove may have a depth greater than the force-generating part of the groove. A minimum depth of the fluid-inlet portion may be configured to create a higher pressure than that surrounding the rotating first ring, while not generating a hydrodynamic or hydrostatic force in the fluid inlet portion. As illustrated in the accompanying figure, the portion (36) responsible for generating the hydrodynamic force is relatively shallow in depth compared with the fluid-inlet portion (34) – both relative to the inner face (30) of the first ring (12). Portion (36) is configured to develop a hydrodynamic force to produce a lifting action during operation. When the first ring (12) is rotated at a particular speed, fluid enters the shallow the portion (36) that generates the hydrodynamic force, and is accelerated by the inertia of the first ring (12) towards the dam (32). The accelerated fluid may increase the pressure between the rings, and may produce a hydrodynamic air film. In one embodiment, the depth of the force-generating portion (36) may have a substantially consistent or constant depth (which may be chosen/configured for an intended or anticipated rotational speed associated with the first ring). For example, in various embodiments the depth of this force-generating portion may range from about 150 µinches to 900 µinches.
Patent number: WO/2013/006560
Inventors: J.F. Short, E.N. Ruggeri and G.M. Berard
Publication date: 10 January 2013



Cross-sectional views of the hydrodynamic face seal detailed by embodiments of patent WO/2013/006560.

"Eccentricity tolerant" valve-stem seal assembly

Applicant: **Dana Automotive Systems Group Llc, USA**
An "eccentricity tolerant" valve-stem seal assembly (40) has been developed. It includes an elastomeric seal that has an upper metal retainer (56), extending downwards and within an upper seal portion (44), which has an inwardly directed sealing lip (50) for contacting the valve stem. There are also flexible middle (46) and lower (48) seal portions. The latter is connected to a lower metal seal retainer (60) that is in contact with the lower edge of the middle seal portion (46). The flexible middle seal portion (46) enables the upper seal portion (44) to flex from an eccentric valve-stem without causing the lip to lose its sealing function. A separate cover may be used in cooperation with this structure. This forms has a tight tolerance fit with the lower seal retainer (48), over the top of the eccentricity tolerant valve-stem seal, in order to limit axial displacement of the upper portion of the seal because of high manifold pressure.
Patent number: WO/2013/009479
Inventors: T.A. Hegemier, R.W. Lehmann and A.S. Williamson
Publication date: 17 January 2013