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THE SPACING OF BOLTS IN FLANGED JOINTS

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This paper presents methods to determine the spacing of bolts in large diameter or straight low pressure flanges. The results of work suggesting values for bolt spacing in bolted flanges without gaskets are discussed and compared with current flange standards. The effect of a gasket in a flanged joint is examined and the influence of the gasket's properties on bolt spacing is determined. Formulae are derived for calculating approximate values for maximum bolt spacing in flange and gasket joints. A brief outline is given of the standard methods for determining required bolt loads in flange and gasket joints.

NOMENCLATURE

s	bolt spacing
t	flange thickness
d	bolt head diameter
q	flange contact surface pressure
E	elastic modulus of flange
g	ratio of elastic modulus of gasket/elastic modulus of flange
k	modulus of foundation
f	compression load per unit length
y	vertical deflexion
b	flange width
w	ratio of gasket thickness/flange thickness
F	point load
I	second moment of area
x	distance along beam from point load
r	ratio of gasket width/flange width
m	gasket factor
P	pressure
di	pipe bore
do	outside diameter of flange

INTRODUCTION

During recent development work at B.S.R.A. in connexion with a draft marine standard for exhaust gas flanges for Diesel engines and boiler uptakes, it was necessary to examine literature references on flange design. These included the flange sealing criterion which gives an allowable bolt spacing which is significantly different from current practice for large-bore low-pressure pipe flanges. An investigation was therefore made to clarify the reasons for this difference, and as the results are of general interest, this study is being made available in the form of a report.

Flange Bolts

The bolts in a flanged joint must have sufficient strength to:

a) withstand the hydraulic end thrust on the joint caused by the internal pressure

b) provide a residual compressive stress or surface pressure on the flange faces when the joint is under pressure.

The spacing of the bolts is also limited because under the action of the bolt load the flange deflects, and leakage of the joint may occur between the bolts. This sealing requirement is considered in[(1)] where methods based on certain simplifying assumptions are given to calculate the maximum bolt spacing in a flanged joint without a gasket.

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BOLT SPACING IN FLANGES WITHOUT GASKETS

The surface pressure on the flange face varies between the bolts with the maximum value occurring under the bolts and the minimum value midway between bolts. The maximum allowable bolt spacing for flange sealing occurs when the pressure at the point midway between the bolts has zero value. Any further increase in bolt spacing will cause the pressure to go negative with separation of the flanges and leakage of the joint.

Fig. 1 gives the results of[(1)] for flanges without a gasket. Values of q_{min}/q_{max} are plotted on a graph of s/t and d/t where

q_{min}/q_{max} = ratio of minimum to maximum contact surface pressure
s = bolt spacing
t = flange thickness
d = bolt head diameter.

The maximum allowable bolt spacing is given by the curve $q_{min}/q_{max} = 0$

The assumptions made in this analysis are that the flanges are flat and that the bolt spacing and flange width are small in comparison to the bolt pitch circle diameter. The curvature of the flange is therefore neglected, and the results will apply to straight flanges, and approximately to flanges of large diameter. Finally the flange is approximated to a loaded beam with the strengthening effect of the pipe, and the bending of the flange in the radial plane being ignored.

The recommended maximum bolt spacing for flanges without gaskets given in Fig. 1 differ greatly from certain bolt spacings called for in flange standards. To illustrate this point the bolt spacings of typical flanges are plotted in Fig. 1. For low-pressure flanges the bolt spacings are seen to be much greater than the recommended values of[(1)].

The influence of the gasket is therefore of considerable importance and the analysis of the reference has been extended to determine the effect of gasket flexibility.

MAXIMUM BOLT SPACING FOR FLANGE SEALING WHEN A GASKET IS USED

Fig. 2 shows the dimensions of a flange, gasket and bolt assembly. The flange is assumed to be a long straight beam with point loads at each bolt. The gasket covers the full race of the flange and is assumed to undergo elastic deformation.

Let the elastic modulus of the flange be E
Let the elastic modulus of the gasket be gE

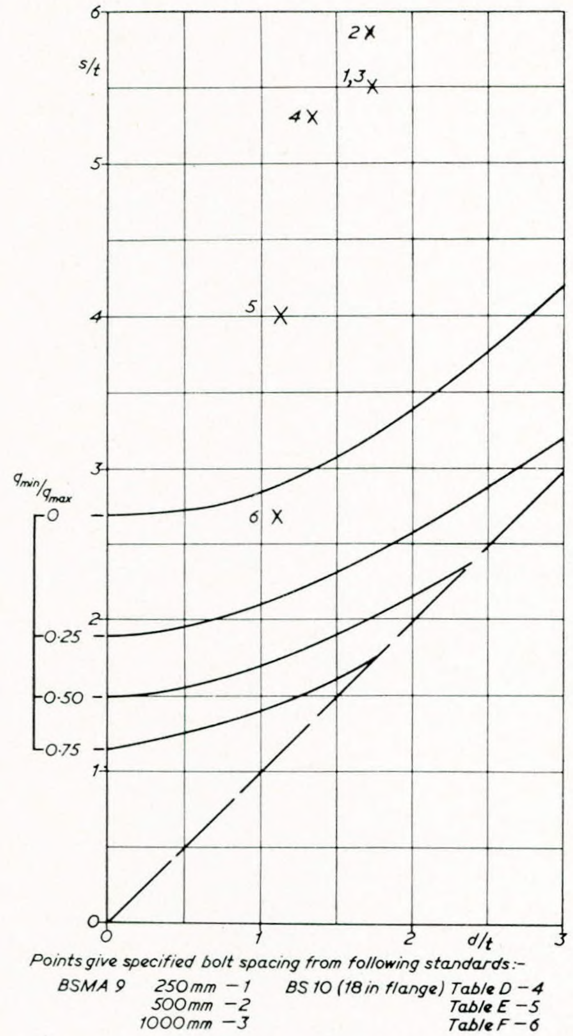

FIG. 1—*Curves for bolt spacing in flanges without gaskets*

Under the action of the bolt, the compression force F causes the flanges to deflect. The deflexion of both flanges is symmetrical about the mid-plane of the gasket, with the materials in the flange and gasket deforming elastically. Considering one flange, the deflexion of its neutral axis can be considered to be analogous to the deflexion of a loaded beam on an elastic foundation. The foundation for one flange is comprised of the parts of the gasket and flange itself, that lie between the flange's neutral axis, and the centre line of the gasket.

Determination of the Modulus of the Foundation

The modulus of the foundation k is defined by the equation $f = ky$, where f is the compression load per unit length that causes a vertical deflexion y.

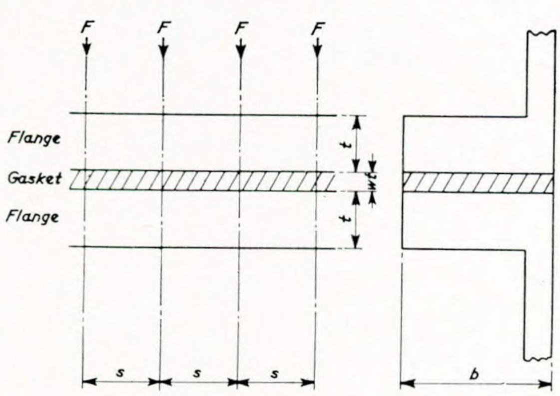

FIG. 2—*Flange and gasket assembly*

Now k = force on unit length to produce unit deflexion of foundation. Using the standard elastic theory, for uniformly distributed leading

$$\text{(Vertical Deflexion)} = \frac{\text{(Force)}}{\text{(Area)}} \times \frac{\text{(Vertical Thickness)}}{\text{(Elastic Modulus)}}$$

(of foundation)

i.e. for unit deflexion of foundation

$$1 = \left(\frac{k}{bx1} \times \frac{t/2}{E}\right) + \left(\frac{k}{bx1} \times \frac{wt/2}{gE}\right)$$

$$\text{Therefore } k = \frac{2bEg}{t(g+w)}$$

Determination of Flange Deflexion

The theory of beams on elastic foundations gives the deflexion produced by a single force F

$$y = \frac{F}{8\beta^3 EI} \times e^{-\beta x} (\cos \beta x + \sin \beta x) \quad (1)$$

where β is defined as $\left(\frac{k}{4EI}\right)^{\frac{1}{4}}$ (2)

I = second moment of area of beam

Substituting for k in equation 2

$$\beta = \frac{1.565}{t}\left(\frac{g}{w+g}\right)^{\frac{1}{4}} \quad (3)$$

The deflexion of the beam, equation (1), is of the form of Fig. 3 and by inspection has zero values at

$$\beta x = \pm\left(\frac{3\pi}{4}, \frac{7\pi}{4}, \frac{11\pi}{4}, \text{etc}\right)$$

For a series of bolts spaced at pitch s, the surface contact pressure or deflexion at any point is obtained by summing the individual distributions for each bolt. The maximum bolt pitch occurs when the deflexion is zero midway between the bolts.

Considering only the bolts adjacent to the point under consideration, Fig. 4 the limiting bolt pitch occurs for:

$$\beta s = \frac{3\pi}{2}$$

$$\text{i.e. } s = \frac{3}{2}\frac{\pi}{\beta}$$

Substituting for β from equation (3)

$$\text{Bolt Pitch} \leqslant 3{\cdot}02 \, x \, t \, x \left(\frac{w+g}{g}\right)^{\frac{1}{4}} \quad (4)$$

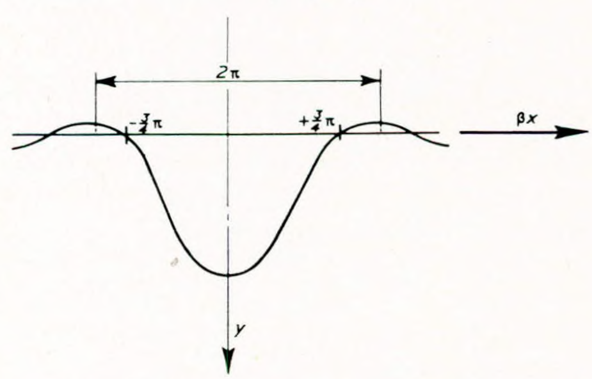

FIG. 3—*Form of deflexion of upper flange under single bolt load*

Equation 4 gives the limiting bolt pitch for full face gaskets. It can be modified for the case of soft gaskets that do not cover the whole face of the flange. When the gasket is soft so that w is appreciably larger than g

$$\text{Bolt Pitch} \leqslant 3{\cdot}02\, x\, t\, x \left(\frac{w}{r \times g}\right)^{\frac{1}{4}} \qquad (5)$$

where r = (gasket width)/(flange width)

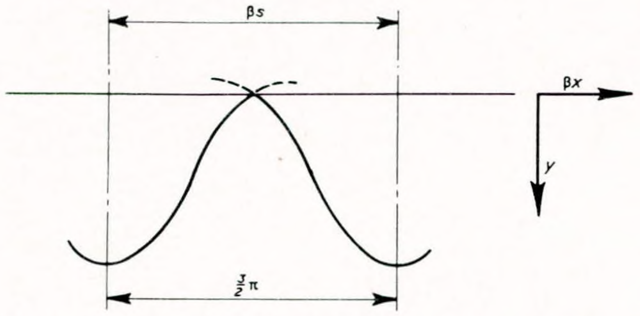

FIG. 4—*Form of deflexion of upper flange under number of bolts at limiting pitch*

DISCUSSION OF RESULTS

Equations (4) and (5) give the maximum bolt spacing for flange sealing in a bolted joint with a gasket. The assumption of elastic gasket deformation will not apply accurately to non-metallic gaskets but if the gasket can be considered to approximately deflect in proportion to the intensity of the applied pressure, the assumption is not too unrealistic.

Using, for examples, flanges from Fig. 1, Table 1 gives the maximum allowable bolt spacing calculated by equations (4) and (5). Gaskets have in all cases been assumed to be compressed asbestos fibre 1·5 mm thick. Full face gaskets are called for in exhaust gas flanges to BS MA9[2] while for BS. 10 flanges[3] gaskets inside the bolt circle are taken from BS.3063[4].

The modulus of elasticity for compressed asbestos fibre is in the order of 1/200 that of steel.

TABLE 1

Exhaust Gas Piping Flanges to BS MA9	Max. Allowable Spacing (Full Face Gasket) Equation (4)	Spacing Specified in Standard
250 mm	102 mm	88 mm
500 mm	121 mm	117 mm
1000 mm	121 mm	109 mm
18″ flanges to BS.10	(Max. allowable spacing (Inside bolt gasket) Equation (5)	Spacing Specified in Standard
Table D	196 mm	156 mm
Table E	179 mm	117 mm

CONCLUSIONS

The general form of equations (4) and (5) show that the maximum allowable bolt spacing for flange sealing is affected considerably by the properties of a gasket. The allowable bolt spacing increases if the gasket is thicker, softer or of less width. In the Appendix it is shown however that using thicker gaskets requires greater compressive stresses and hence the bolt load will increase. There are also practical minimum gasket widths.

The analysis does show that for a large diameter bolted flange with a gasket, that there is a maximum value of bolt spacing which is independent of the internal pressure or the compressive load imparted by the bolts. This maximum spacing is governed by the dimensions of the flange and gasket, and their approximate elastic properties.

In view of the assumptions made to simplify the configuration of the bolted joint, equations (4) and (5) will give only approximate values of maximum allowable bolt spacing for flange sealing. In the case where the stress/strain properties of the gasket are completely unknown this analysis should give the qualitative effects of the various factors of the problem.

Appendix

Determination of Required Bolt Loads in a Flanged Joint with a Gasket

The required total bolt load must be sufficient to withstand the hydraulic end force on the joint, and to exert a certain residual compressive stress on the gasket.

The value of this residual stress is usually given as a ratio of the internal pressure. This ratio is called the gasket factor m, i.e.

$$m = \frac{\text{(total bolt load)} - \text{(hydraulic end load)}}{\text{(Gasket Area)}} \times \frac{1}{\text{(Internal Pressure)}} \qquad (6)$$

Values of m are somewhat arbitrary, but a value of about 2·5 is suitable for many types of asbestos gaskets[5]. A higher value of m is needed for sealing gases, and for various types of metal gaskets. A value of m less than 2·5 should be adequate for asbestos gaskets whose width exceeds 25 mm.

As there is still some discussion over the suitable value of m it is suggested that hydraulic end load is calculated using the bore area of the gasket, i.e. Hydraulic End Load = (Bore of area of gasket) × (Internal Pressure), and that the actual gasket area is used in expression (6).

Considering as an example a circular gasket with inside and outside diameters di and do and an internal pressure P,

$$m = \frac{\text{(total bolt load)} - (P \times \pi/4\, di^2)}{\pi/4\,(do^2 - di^2)} \times 1/P$$

Therefore total bolt load = $P \times \pi/4\,(mdo^2 \times (m - 1)\, di^2)$

For low pressure flanges the gasket factor criterion does not provide adequate compressive stress to seat the gasket into the surface irregularities of the flange. A minimum value of compressive stress is therefore needed. This in the order of 15 — 30 MN/m² for most asbestos gaskets, rising to 150 MN/m² for some metal gaskets.

Thickness of Gaskets

The use of a thin gasket is advantageous for high pressure joints, as there will be a smaller spreading force on the inside edge of the gasket. For a given value of m, this force will cause a thick gasket to have a lower leakage pressure as against a thin gasket[6].

For compressed asbestos gaskets the properties of the material are such that the thinner the gasket, the higher is the compressive stress that it can sustain. A common rule given by asbestos gasket manufacturers is that the suitable gasket thickness is four times the largest flange surface irregularity, which for normal machined faces means a 1·5 mm gasket.

Relationship Between Bolt Torque and Tension

This is a complicated problem and depends very much on the coefficient of friction in the screw thread and on the bearing face under the nut. Unless friction conditions are known accurately a very useful relationship is given in BS.3580—"Guide to Design Considerations on the Strength of Screw Threads."

i.e. Applied Torque = $\dfrac{\text{(Bolt Tension} \times \text{(Thread Diameter)}}{5}$

This relationship is based on a coefficient of friction of approximately 0·15.

These notes, on bolts loads in flanged joints should give sufficient information to calculate the required torque and cross sectional area for flange bolts, where adequate guidance from standards is not available. Where high pressure flanges with high bolt loads are involved more advanced texts should be consulted, e.g. [5] and [6].

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