TESTS ON A 1,75 m O.D. BOLTED FLANGE ASSEMBLY WITH SPIRAL-WOUND GASKETS

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ABSTRACT

Paper reports on a test programme involving leakage behaviour of a double gasketed 1.75 m (O.D.) bolted flange assembly subjected to hydrostatic pressure at room temperature. Fabrication of a heat exchanger for a CANDU-NPP was temporarily stopped so that the channel-cover - tubesheet assembly could be used for the tests. Special instrumentation was developed to measure bolt stress, inner and outer flange displacements, flange to tubesheet metal contact stress and intergasket leakage. The results of two of several flange-face configurations tested are reported. Main conclusions follow:

- Very high gasket seating stresses are necessary for adequate sealing.
- Flange movement alters contact pressure on inner and outer gaskets. Simple flange face modifications adequatly compensates for this and provides leak-tight joint.

1. INTRODUCTION

In recent years increasing attention has been paid to the better understanding of gasket behavior and the validity and basis of the m and y factors listed in the ASME Code [1]. A report by Raut and Leon [2] is indicative of the effort required to obtain useful experimental data on gasket leakage behavior. The Subcommittee on Bolted Flange Connections of the Pressure Vessel Research Committee of The Welding Research Council has formed a "Task group on Long Range Gasket Testing" (see Ref. [3]); widely used gaskets such us Spiral-Wound or Compressed Asbestos, etc. are prime candidates for immediate investigation.

In Canadian Nuclear Applications, Asbestos-filled spiral-wound stainless-steel gaskets are utilized almost exclusively in bolted flange assemblies. One special feature of the Canadian bolted flange design is its use of double gasketed joints: the two gaskets are separated by an annular chamber to which leakage detection instrumentation and leakage collection piping may be connected; both gaskets are thus within the bolt circle. For such designs proper knowledge of gasket behavior is even more critical than for single gasketed joints because of the increased lever arm between bolt and gasket circles and the wider overall gasket contact area.

In a previous paper [4], the author presented the results of mechanical compression tests on a wide range of spiral wound gaskets from 25 mm to 1300 mm in diameter. It was found that very high stresses are necessary to compress the gaskets to the value recommended by the manufacturer. It was also found that an important drop in gasket stress results from a small flange movement.

This paper reports on the results of a test program involving the sealing behavior of a large double gasketed bolted flange assembly subjected to hydrostatic pressure at room temperature. The test program has the following purposes:

1) Assess the sealing performance of several flange face geometries and seek improved design.
2) Compare gasket behavior during actual pressure testing with that of simple mechanical tests as reported in [4].
3) Determine approximate applicable gasket factors.

A total of seven test series have been performed in which some flange face configurations included one or more annular metal contact regions. Because of limited space, only the results of the two configurations of most practical interest are presented in this paper; they are representative of the various configurations and joint behavior considered.

2. TEST DETAILS

2.1 *Test unit*

Fabrication of a heat exchanger for a CANDU Nuclear Power Plant was temporarily stopped so that the channel-cover/tubesheet assembly could be used for the tests. Fig. 1 gives the overall dimensions of the assembly. As illustrated in Fig. 2 the test unit is instrumented for bolt stress, metal contact pressure, flange deflection, intergasket leakage and internal pressure. Temporary sealing of the 2380 tubesheet holes is achieved by means of a 3 mm rubber mat covering the entire perforated area and backed by a mild steel 10 mm thick plate held in place by twenty evenly distributed studs passing through corresponding tube sheet holes and fitted with washers and nuts on the opposite tubesheet face.

2.2 *Flange face configurations*

The two flange face configurations considered in this paper are detailed in Fig. 3 :

2.2.1 *Configuration A*

Provides a metal contact ring to prevent gaskets from compressing beyong 2.9 mm (from 4.6 mm thick) as recommended by the gasket manufacturer and assumes both gaskets compress an equal amount (minimal flange rotation).

2.2.2 *Configuration E*

This configuration was arrived at on the basis of test results for configuration A and subsequent intermediate ones. It compensates for flange rotation by providing a deeper groove for the outer gasket, also the metal contact ring is machined off.

In both cases the surface finish at the gasket contact is 3200 μm RMS.

2.3 *Gasket type*

Both inner and outer gaskets are manufactured by the Flexitallic Gasket Co. Ltd. and consist of:

- a stainless steel winding, 0.18 mm thick
- blue asbestos filler
- initial gasket thickness 4.6 mm
- gasket width: nominal = 19 mm , actual = 17.4 mm (excluding bead)
- total number of windings 32
- inner gasket I.D. = 1232 mm
- outer gasket I.D. = 1297 mm

2.4 *Instrumentation details* (*Fig. 2*)

1) Electric resistance strain gage deflectometers are positioned at 3 locations, 120° apart, at the inner and outer flange diameters. The deflectometers consist of a small contilever leaf spring fitted with an adjustment screw at its end; the base of the spring is attached to the channel cover and the tip of the screw is in contact with the tubesheet or a bracket attached to it. Two strain gages connected in a half Wheatstone bridge configuration are mounted on the springs. Several layers of protection coatings are applied to the strain gage installation in order for sufficiently reliable readings to be obtained from the submerged deflectometers in spite of the adverse pressurized water environment to which these gages are subjected.

2) A caliper gage is used to measure the deflection of the outer periphery of the flange. Also measured is the gap in the metal contact region with a set of feeler gages. Both these additional deflection measurements permit a continuous check on the deflectometer readings.

3) Three "epoxy resin contact pressure plugs" are embedded in the shell flange in the

region of the metal contact ring. Two strain gages are encapsulated in each of the plugs [5] , mounting procedure is detailed in Fig. 4 (see 4.1 (d) for calibration).

4) Three studs are instrumented with strain gages. For this purpose, the threads are machined to the root diameter over a short length of stud in a region coinciding with the gap between the flange and the tubesheet as indicated in Fig. 2. The instrumented studs are calibrated in compression on a laboratory press.

5) The intergasket leak detector made of transparent PVC tubing (3 mm ID) is sensitive to a leak of a single drop (approx. 10 mm^3). The graduated U-tube is filled with green-colored water for ease of viewing.

6) Two calibrated pressure gages are used to monitor the hydrostatic pressure.

2.5 *Testing procedure*

The tests are carried out in the following sequence:

1) Stud tightening usually in a number of steps with increasing stud tension up to the required minimum level (see 2.6 for bolting-up procedure).
2) Filling of unit with water.
3) Pressure testing, usually in 6 steps up to p=18.6 MPa or when a leak appears (a few drops per minute). Pressure cycling is performed when no leak is detected (i.e. usual waiting time: 1 hour).
4) Further stud tightening (at p = 0).
5) Repeating (3), (4) until maximum allowed stud tension stress is attained (350 MPa).
6) Draining of unit.
7) Stud untightening, in reverse sequence of (1).
8) Removal of cover and refitting of new gasket for subsequent test series.

2.6 *Bolting-up procedure*

Initial attempts at controlling torque resulted in important differences in stud tension even though torquing-up was performed in three or more passes.

Significantly more consistent results are obtained by the turn-of-the-nut method using a hydraulic torquing machine. The following procedure is followed:

1) With the sequence of Fig. 5, the nuts are tightened by hand (see note)*.
2) Starting with instrumented stud No.8, nut tightening is carried out until a stress value above the required level is reached.
3) Studs 1 to 7 are tightened in sequence with the same extent of nut rotation as stud No.8.
4) Steps (2) and (3) are repeated if the stress level in stud No.8 drops below a predetermined level.

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* Note: Control of this operation is difficult because the extent of hand tightening varies from one operator to the other. Only small fluctuations are observed however from one stud to the other in any particular bolting-up cycle by the same operator.

5) All remaining studs are tightened in sequence with the same total nut rotation as stud No.8.

Stress levels in instrumented studs 8, 14 and 28 are usually found to be within ± 10% of each other indicating a well controlled bolting-up procedure.

3. RELATED COMPUTATIONS

3.1 *Gasket stress*

The system of forces acting on the flange is given in Fig. 6. From force equilibrium considerations one obtains:

$$N_B - P - G - M = 0 \qquad (1)$$

where N_B = Total stud force (N)
P = Total pressure force (N)
G = Total gasket force (N)
M = Total metal contact force (N)

Knowing that

Total stud root area = $32 \times 4.067 \times 10^{-3} = 0.130 \text{ m}^2$
Total gasket area = $w\,\pi \left[1.251 + 1.314\right] = 0.153 \text{ m}^2$
Total pressure area = $\frac{\pi}{4}(1.232)^2 = 1.192 \text{ m}^2$

Equation (1) gives for gasket stress:

$$\sigma_G = 0.849\,\sigma_B - 7.78\,p - 6.53\,M \qquad (2)$$

where σ_B = Stud (bolt) tensile stress (Pa)
p = Pressure (Pa)
σ_G = Average compressive gasket stress (Pa)
w = Gasket width (m)
(nominal width of 0.019 m is used throughout in this paper)

Values of σ_G, computed from (2) are plotted together with experimental data in the graphical results. Equation (2) is also the basis for obtaining an approximate calibration for metal contact force as a function of contact-pressure-plug readings (see 4.1 (d)).

3.2 *Flange rotation and gasket deflection*

Assuming that the flange rotates essentially as a rigid body, and knowing the distance between the inner and outer deflectometers (295 mm, Fig. 3) the angle of rotation of the flange with respect to the tubesheet is:

$$\theta = 0.194\,(\delta_o - \delta_I) \quad \text{(deg)} \qquad (3)$$

where δ_I and δ_o are the displacements indicated by the inner and outer deflectometers (mm).

With the gaskets located at 28.6 and 60.4 mm respectively from the internal deflectometer, flange displacement at the gaskets is thus computed:

$$\delta_{G1} = \delta_I + 0.1\,(\delta_o - \delta_I)$$
$$\delta_{G2} = \delta_I + 0.205\,(\delta_o - \delta_I) \qquad (4)$$

4. RESULTS

4.1 *Configuration A*

Figs 7 and 8 show the results for stud tightening and pressure testing pertaining to configuration A. The following parameters are plotted:

a) Inner and outer flange deflections δ_I, δ_o as measured by the strain-gage deflectometers and confirmed by caliper and feeler gage measurements.

b) Flange rotation θ, from equation (3).

c) Inner and outer gasket deflection, from equation (4), in Fig. 7 only.

d) Average gasket stress σ_G from equation (2).

e) Metal contact force M as indicated by the embedded plug readings (see 2.4(3)) and calibrated by considering equation (2) and the deflection measurements (Fig. 7):

i) Initial metal contact occurs at

$\sigma_B = 190$ MPa

$p = 0$

Hence, from (2), with M = 0,

$\sigma_G = 161$ MPa

ii) δ_I remains essentially constant once metal contact is attained. It is therefore assumed that σ_G also remains constant at 161 MPa. Hence from (4)

$$M = 0.13\ \sigma_B = 24.7 \times 10^6 \quad \text{(N)} \qquad (5)$$

iii) A calibration curve (not shown) relating equation (5) with actual contact force readings was confirmed throughout the various series of test in which metal contact was present.

In the results presented, the curves for stud tightening alternate with those for pressure testing, as indicated by the step number at the top of Figs 7 and 8 in the following manner:

- Steps 1, 2 and 3 are three consecutive levels of stud tightening.
- Steps 3 to 12, 13 to 22, 23 to 34 and 35 to 42 are four cycles of pressure tests with increasing initial stud stress. Only the first and third pressure cycles are shown in Figs 8 (a and b) and represent typical behavior. Leakage (of the order of a few drops per minute) begins to occur, in all cases, at a pressure of about 15 MPa.

Fig. 9 shows gasket stress as a function of gasket deflection, from equation (3), for the first pressure cycle as computed from Figs 7 and 8 (a) . Also plotted is a curve obtained during mechanical compression/decompression testing of a similar gasket [4] . If initial zero shift is neglected the similarity between the two curves is evident thus confirming the validity of the computed parameters plotted in Figs 7 and 8.

4.2 *Configuration E (Fig. 8)*

Figs 10, 11 and 12 give, for configuration E, results of the same type as those presented for configuration A in Figs 7, 8 and 9 respectively.

Since no metal contact occurs in this case, M is always set at zero in equation (2) when computing average gasket stress. It should be noted that there may be an important difference between the stress applied on the inner and outer gaskets because of the difference in gasket compression. This is illustrated in Fig. 12 where the complete set of gasket stress-deflection loops is plotted for this series.

An important leak (several drops/second) developed at the end of the first cycle (steps 3 to 9) because of loss of contact stress at the gaskets. No further leak developed in subsequent pressure cycles (steps 10 to 16 and 17 to 23).

5. DISCUSSION

Comparison of the results of configuration A with those of configuration E will serve to explain why the latter performs so much better than the former:

1) Metal contact: The purpose of the metal contact ring is to limit the compressive stress on the gasket. Because this requirement is satisfied, additional stud tightening will not improve sealing. Removal of the metal contact corrects the situation. It is worth noting that, in spite of the very high stress levels applied on the gaskets during stud tightening, no damage is observed on any of the gaskets tested.

2) Flange rotation causes unequal deflection of the two gaskets (Fig. 7) resulting in the inner gasket being less stressed (Fig. 9). Sinking of the outer gasket in a deeper groove, as in configuration E, overcompensates for flange rotation (Fig. 10) and creates a higher stress on the inner gasket as pointed out in Fig. 12.

3) Gasket decompression: The curves of Figs 9 and 12 confirm results previously reported by the author [4] and show that a small flange movement (of the order of 0.1 mm) will cause an important drop (150 MPa) in gasket stress upon decompression. Configuration E compensates for this lack of sufficient gasket spring-back by allowing an increase in initial gasket stress. In other designs it may be advantageous to increase the overall stiffness of the bolted flange connection.

4) Gasket factors: Although the present results are limited in number it is of interest to compute the gasket factors y and m as defined in the ASME Code [1].

   i) Factor y is defined as the minimum gasket seating stress, and is based on an effective gasket area approximately 1/2 x actual area.
   Hence, $y \cong 2$ x initial gasket stress as computed here.

   The present results indicate initial stresses as high as 270 MPa (Fig. 11 (b)), however in Fig. 11 (a), a value of approx. 180 MPa is sufficient to ensure a leak-tight joint (i.e. y = 360 MPa = 52000 psi). Such a high value was predicted in [4] and is much above y factors listed in the ASME Code for spiral wound gaskets.

   ii) Factor m ensures that the residual stress on the gasket is at least equal to m times the pressure for proper sealing; it is applicable twice the effective gasket area, (i.e. nearly full gasket area).

   From Fig. 12 (step 16), an inner gasket stress of approx. 25 MPa is sufficient to seal at a pressure of 18.6 MPa, giving a value of as low as m = 1.3. Results of configuration A however indicate that, before leakage occurs, at a pressure of approx. 14 MPa, the gasket stress is 60 MPa on the average, so that m = 4.3 which approaches the m-factor suggested in the ASME code.

It may be observed, in a general sense, that the steep gasket decompression curve is the reason that why such high initial stresses must be applied on the gasket. In summary therefore, not only is there an interaction between the m and y factors, but the overall stiffness of the bolted flange assembly (including the stiffness of the gasket upon decompression) must also be taken into account.

6. CONCLUSIONS

Room temperature pressure testing of a double gasketed full size flange assembly with two flange face configurations yield the following main conclusions:

1) The 19 mm-wide spiral-wound gaskets require compression stresses of the order of 180 MPa (based on nominal gasket width) for initial gasket seating and in order to compensate for an important drop in gasket stress resulting from a small flange movement.

2) Sealing is maintained for a value of m of about 4 .

3) Removal of the metal contact ring and compensation for flange rotation yields an improved flange face design.

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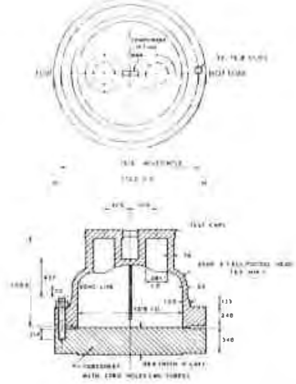

Fig. 1

Overall dimensions of double gasketed bolted tubesheet/channel-cover assembly used throughout test series. All dimensions are in millimeters.

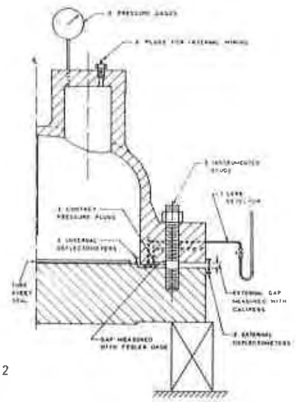

Fig. 2

Test assembly showing instrumentation scheme. When three stations are indicated they are spaced 120° apart around the flange periphery.

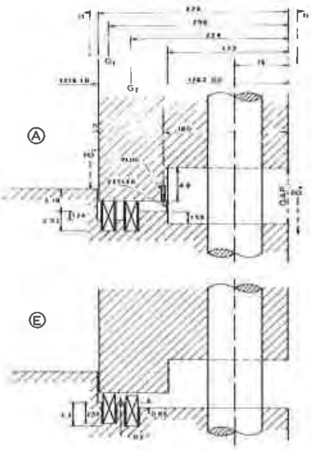

Fig. 3

The two flange face configurations (dimensions in mm. Note that vertical scale is 10 times larger than horizontal scale).

A. Two 2.9 mm gasket grooves and metal contact between bolt circle and outer gasket.

E. Inside gasket groove 2.9 mm deep, outside groove 3.3 mm deep; no metal contact.

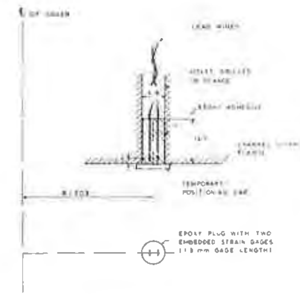

Fig. 4

Details of contact pressure plugs.

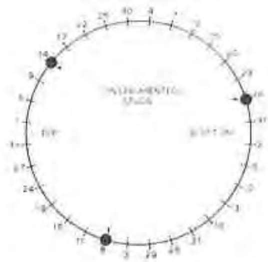

Fig. 5

Stud tightening sequence.

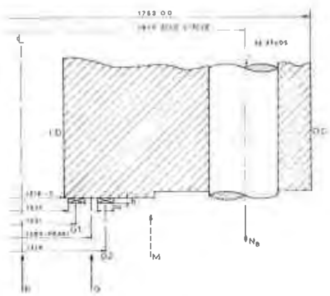

Fig. 6

System of forces acting on flange and sign convention.

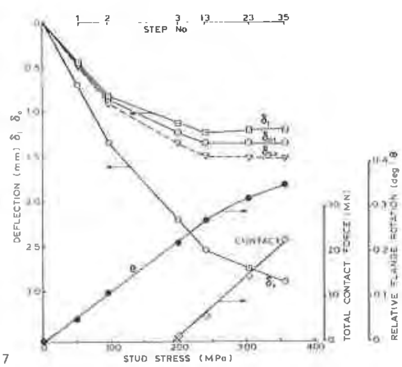

Fig. 7

Inner and outer flange deflections, gasket deflections, flange rotation and metal contact force as a function of stud stress during tightening. Configuration A .

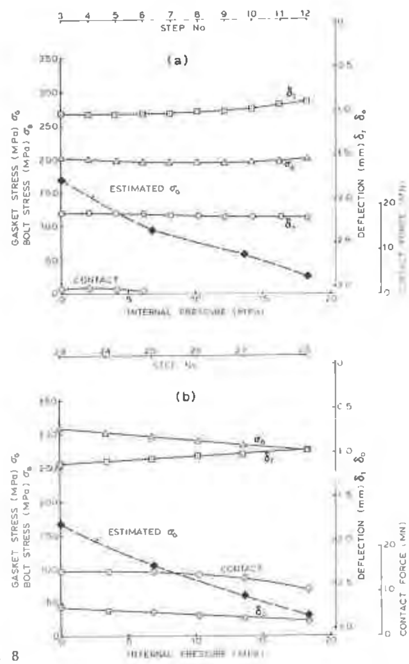

Fig. 8

Flange deflections, stud stress, metal contact force and gasket stress as a function of pressure. Configuration A.

a) 200 MPa initial stud stress.

b) 310 MPa initial stud stress.

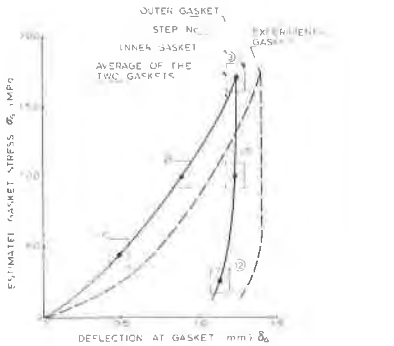

Fig. 9

Gasket stress as a function of gasket deflection comparison of test gaskets wit lab-tested gasket (Ref. [4]). Single pressure cycle steps 1 to 12. Configuration A.

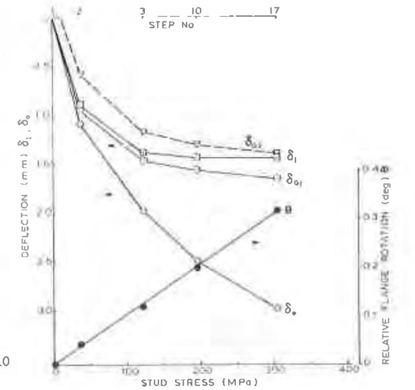

Fig. 10

Inner and outer flange deflection, gasket deflection, and flange rotation as a function of stud stress during tightening. Configuration E.

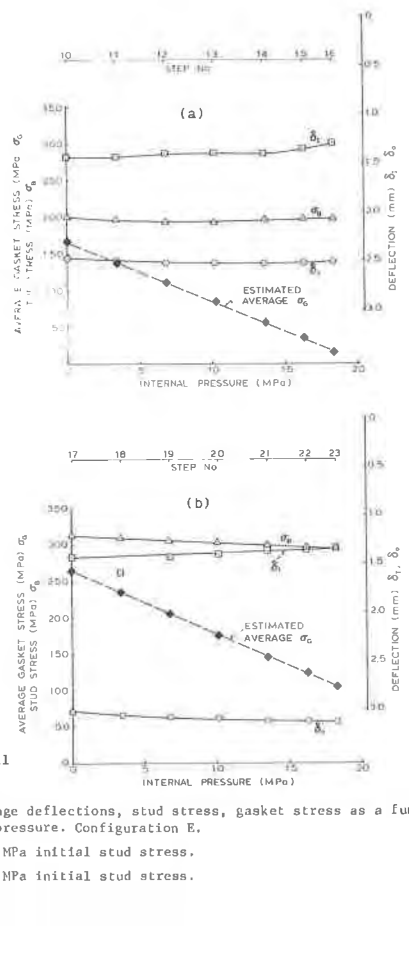

Fig. 11

Flange deflections, stud stress, gasket stress as a function of pressure. Configuration E.

a) 203 MPa initial stud stress.

b) 317 MPa initial stud stress.

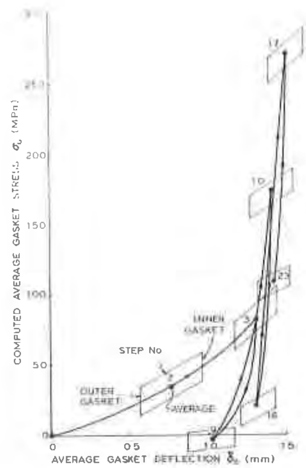

Fig. 12

Gasket stress as a function of gasket deflection for three cycles of pressure test with increasing stud tightening. Configuration E.