discussion

Application of Spiral Wound Gaskets for Leak-Tight Joints[5]

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K. P. SINGH.[6] Authors Stevens-Guille and Crago have rendered an important service to the pressure vessel industry by publishing the deflection characteristics of asbestos-filled stainless steel spiral wound gaskets. Using their test results and the

[4]Smith, R. F., and Knott, J. F., "Crack Opening Dispoacement and Fibrous Fracture in Mild Steel," Conference on Practical Application of Fracture Mechanics to Pressure Vessel Technology, The Institution of Mechanical Engineers, London, 1971, pp. 65–76.

[5]By P. D. Stevens-Guille and W. A. Crago, published in the February 1975 issue of JOURNAL OF PRESSURE VESSEL TECHNOLOGY, TRANS. ASME, Vol. 97, Series J, No. 1, p. 29.

ASME Code[7] formulas, Stevens-Guille, et al., deduce that the minimum seating stress of spiral wound gaskets is in the order of 20,000 psi. This conclusion may be erroneous. This subject, being of profound importance in pressure vessel closure design, warrants a close scrutiny.

The minimum seating stress used in the Code is associated with an assumed effective width b, which for the most configurations is defined by

$$b = 0.5\ (0.5N)^{1/2} \tag{1}$$

where N is nominal gasket width. The value of b is strongly dependent on the rotational flexibility of the flange. The custom flanges, with an eye to economy, are far more flexible than their standard product line counterparts made by flange manufacturers.

Furthermore, certain flange types such as lap joints exhibit greater flexibility than the welding neck construction. The Code ignores such refinements, presumably for the sake of simplicity. However it is evident that a seating load W_{m2} defined by Code[7] (p. 243) as,

$$W_{m2} = \pi bG \cdot y \tag{2}$$

where b is given by equation (1), will in general produce a peak pressure different from y. For flexible flanges, the effective width is less than b, and consequently the peak pressure is higher than y. In other words, for flexible flanges, the actual peak seating stress may be much higher than the nominal value of y used in computations. On the other hand, Stevens-Guille and Crago employ flat rigid platens to compress the gaskets in their tests. This results in an effective width greater than that given by Code (equation (1)), and hence a high seating load W_{m2} is recorded in the experiments.

If one assumes that the gaskets are uniformly compressed in the reported experiment, then b should be taken equal to the actual width of the gasket, N. Based on $b = N$, the following table gives the modified values of S_g for the seating loads reported in Table 1 of the paper.

Table 1 Value of S_g based on $b = N$

Gasket data O.D. (in.)	Gasket data I.D. (in.)	Width N (in.)	Compression load $W_{m2} \times 10^{-3}$ #	Stress on modified compression area, psi, S_g
20.75	19.25	0.75	436	9252
18	16.5	0.75	370	9103
17.5625	16.0625	0.75	339	8558
14.4375	13.4375	0.5	126	5755
12.8125	11.5625	0.625	185	7730
5.5	4.75	0.375	71	11759
5.125	4.25	0.4375	113	17539
4.75	3.75	0.5	88	13182
4.125	3.375	0.375	50	11318
2.375	1.5625	0.4063	43	17113
2.2188	1.6875	0.2657	14	8586

It is seen that the value of S_g is generally in the neighborhood of 9000 psi; which incidentally is the value of y suggested by some leading gasket manufacturers.

We at Joseph Oat Corporation have used the values of y in the range of 8000–10,000 psi with great success.

Finally, our experience has shown that the filler material used in the spiral wound gaskets has an important effect on its resilience, and perhaps its stiffness. In one instance, we were unable to seal a joint using a spiral wound gasket with teflon filler. Using an identical gasket with asbestos filler sealed the joint without any trouble.

[7]ASME Boiler and Pressure Vessel Code, Section VIII, Div. 1, New York, 1974, pp. 233–260.

Author's Closure[8]

Mr. Singh's observation that, due to flange flexibility, nonuniformity of gasket stress occurs is correct. Both radial and circumferential variations exist. Circumferentially, peak stresses occur under the bolts and minimums between the bolts. Flange rotation under load results in a lower stress being applied to the inside edge of the gasket than is present at the outside edge. This consequence is particularly serious in double-gasketted designs where the primary (inner) gasket is far inboard from the bolt circle. Differential groove depths may be necessary in this case to compensate for flange rotation. Our experiments have utilized extra thick flanges to minimize both radial and circumferential flexibility effects.

Rigorously defined, the preload or yield factor, y, is the stress required to cause a gasket to seal against zero pressure differential. By definition of y, then, the equation

$$W_{m2} = \pi Gby \tag{3}$$

can only have application at zero pressure. Since the Chalk River program has been exclusively concerned with sealing at CANDU primary heat transport system conditions, no experimental determination of y has been made. I feel that Singh has inferred far more from the gasket compression data presented in the paper than is legitimate. This data was presented mainly to illustrate that if a spiral wound gasket is compressed to the manufacturer's recommendation for optimum performance based on their experience, then a preload substantially in excess of that indicated by the Code factor y is required. No implication that satisfactory leakage performance would necessarily result from so doing was intended.

In addition it can be stated that had the test gaskets been compressed with radial constraint, the compression load at the manufacturer's recommended thickness would have been substantially greater, particularly for the larger gaskets. For example, the 14.4375-in. OD and the 12.8125-in. OD gaskets listed in Table 1 require approximately 15,000 psi stress at 0.130 in. when constrained in grooves having 0.031-in. radial clearance inside and outside, as recommended by the manufacturer.

The residual or maintenance factor, m, provides the additional preload necessary to stress the gasket such that it will continue to seal after differential pressure is applied. The Code equation

$$W_{m1} = \frac{\pi G^2 P}{4} + 2\pi bGmP \tag{4}$$

implies that the gasket stress required to effect a seal using a given gasket, is linearly dependent upon system pressure only. I am skeptical that this is true. It seems more probable that m is a function of fluid thermodynamic and transport properties as well as being dependent upon the choice of gasket. The use of equation (3) by the Code to determine the required gasket load for certain low-pressure applications may be an attempt to empirically correct for the nonindependence of m from fluid properties, including pressure.

Most of my tests have utilized stainless steel-Chrysotile asbestos gaskets. However experimental evidence confirms Singh's observation that the choice of filler material may substantially affect the performance of a gasket in any given service.

In closing, I wish to emphasize that much remains unknown about the performance of spiral wound gaskets. However it seems axiomatic that the only measure of gasket performance is the leakage which occurs over its service life. It is recognized that acceptable performance is different for different applications depending on the toxicity and value of the fluid and the cost of making repairs. To satisfy the stringent requirements of the

[8]The viewpoints expressed herein belong to Mr. W. A. Crago alone. Mr. Stevens-Guille was not available for comment.

CANDU nuclear power system, the experimental work reported in this paper and the experimental data accumulated since this paper was prepared indicates that prudent design of spiral wound gasketted joints requires that the gasket be initially seated to a mean stress over the total contact area of about 18,000 psi. The definitions and values used in the Code equations are relatively unimportant provided this condition is satisfied and subsequent leakage performance is satisfactory.