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discussion

Elastic-Plastic Fracture Toughness — A Comparison of J-Integral and Crack Opening Displacement Characterizations[1]

B. A. FIELDS.[2] The author writes that the extent of crack flank linearity during deformation of HY-80 fracture toughness specimens has not been established. Recently a technique, initially due to Robinson [1],[3] has been developed [2] using a silicone rubber to take replicas of a crack at successive stages of deformation in a standard notched three-point bend test. The fluid rubber is placed in the notch, the specimen given a known flank linearity during deformation of HY-80 fracture toughness deformation in a standard notched three-point bend test. The amount of deformation and after 20 min under load the hardened rubber can be removed with tweezers and the process repeated. It was found for HY-100 that the same results were obtained using a machined slot of width 0.25 mm as for a fatigue-cracked specimen and hence the former was used for ease of observation.

Using the replica both the crack front and a section through the midsection of the crack were examined and recorded on tracings with the aid of a projection microscope at a magnification of ×50. The results for HY-80, Fig. 1, show clearly the formation of the stretch zone, the linearity of the crack flanks at all stages of deformation and the initiation of cracking (a diffuse process of microvoids coalescing with the crack tip). A direct measurement of COD could be made and compared with values predicted from the author's equation (7). For specimens with $W = 20$ mm, $B = 20$ mm and $a = 8$ mm a relation COD $= 0.37$ V_g - 0.37 was obtained. For $V_g > 0.5$ mm this result can be approximated by using equation (7) with $r = 0.33$. A value of COD$_i$ of between 0.25 and 0.29 mm was obtained for HY-80 in agreement with the results of the paper.

[1]By C. A. Griffis, published in the November 1975 issue of the Journal of Pressure Vessel Technology, Trans. ASME, Vol. 96, Series J, No. 4, p. 278.

[2]Cambridge University Engineering Department, Cambridge CB2, England.

[3]Numbers in brackets designate References at end of discussion.

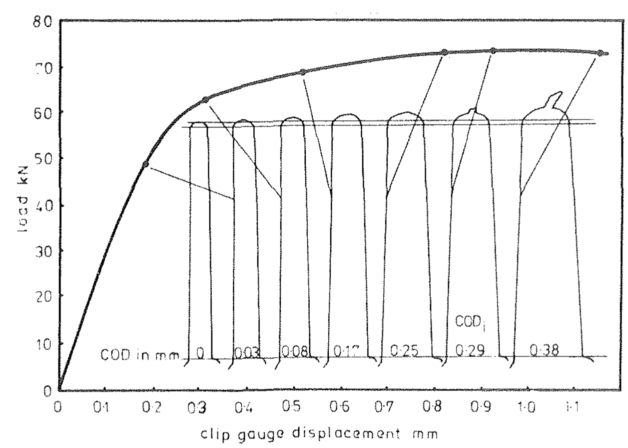

Fig. 1 Load versus deflection curve for HY-80 steel indicating COD values, stretch zone formation and crack flank linearity with increasing deformation

References

1 Robinson, J. N., and Tetelman, A. S., "The Critical Crack-Tip Opening Displacement and Microscopic Fracture Criteria for Metal," U. S. Army Research Office, Durham, DA HC04-69-C-0008, Technical Report No. 11, 1973.

2 Fields, B. A., and Miller, K. J., "A Study of COD and Crack Initiation by a Replication Technique," Cambridge University Engineering Department Technical Report, CUED/C-Mat/TR.17 Oct. 1974.

Author's Closure

The author is pleased to hear of Dr. Fields' experiments which tend to support the validity of equation (7), as well as the author's determination of the onset of crack extension for HY-80 steel using a heat-tinting procedure. Furthermore, the results reported by Dr. Fields suggest that a value of $\lambda = 1$ is reasonable, which establishes a quantitative link between the J-integral and COD approaches for sufficiently large specimens. It would be of considerable interest to apply the "replicating" technique over a range of specimen sizes to further investigate the apparent reduction in COD (and J-integral) with decreasing size as depicted in Fig. 8 of the author's paper. Such a study might establish whether this reduction reflects primarily: (1) a breakdown of COD as a fracture initiation criterion due to constraint relaxation, and/or (2) a dependence of the rotational constant, r, on thickness as suggested in Smith and Knott.[4]

[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.

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

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 strainless steel spiral wound gaskets. Using their test results and the

[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.

[6]Chief Engineer, Joseph Oat Corporation, Camden, N. J.

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,

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

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

$$W_{m2} = \pi G b y \tag{3}$$

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

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

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 nulcear 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.

Energy Approach for Creep-Fatigue Interactions in Metals at High Temperatures[9]

E. KREMPL.[10] *The author has attacked a very complicated* problem of elevated temperature design and has proposed the energy approach for the solution of the creep-fatigue interactions at elevated temperature.

The discussor is also of the opinion that the energy approach holds promise. The formidable problem of identifying the energy that is permanently stored in or released from the material has, however, not been solved. Only this part of the energy can be used to calculate damage as it is responsible for the structural rearrangements within the metals which ultimately lead to failure. The author proposes "to construct from data (*b*) a hypothetical curve for the material with a given strain history, and to compare the hypothetical curve with the real data . . . and assert that the difference must be due to structural change." The discussor does not consider this to be a satisfactory solution since the difference between a hypothetical and a real quantity is still hypothetical. It is further unclear how material responses under different histories can be compared in a meaningful way. It would be helpful if these points could be clarified.

The discussor wants to emphasize the need for data of material response listed under (*a*)-(*f*) on p. 217. Since there is a nonlinear, cycle dependent relation between stress and strain it would appear that stress-controlled data in the categories (*a*)-(*f*) are also important. In general, creep-fatigue interaction tests usually do not report deformation data although they are taken during testing. Hysteresis loops and cyclic deformation data are absolutely necessary for the development of energy approaches. Since many of these data are not reported the author had to rely on hypothetical numbers which made some of the arguments not as convincing as they would have been if real data would have been available.

In reference [14] of the paper a status report on our constitutive equation development was given. The objections (footnote 10) of the author have been met in a paper which has appeared in *Acta Mechanica*, *22*, 1975, p. 53.

S. S. MANSON[11] and G. R. HALFORD.[12] We wish to address ourselves to the comments made by Dr. Fong regarding the Strainrange Partitioning method which we have been developing in recent years. In defending our method it should be made clear that we are not criticizing Dr. Fong's new and interesting method nor are we criticizing Dr. Fong himself for briefly referring to a "limitation" of our approach in setting the stage for the presentation of his new treatment. All contributions to the treatment of high-temperature creep-fatigue interaction are highly welcome, as is Dr. Fong's because of its novelty. But we cannot resist the temptation of seizing on one of the points made by Dr. Fong in his references to Strainrange Partitioning because it focuses on common misunderstandings of the method. In clearing up these points we can, perhaps, also clarify the generality of our method and how it can, indeed, be the possible answer to some of the questions raised by Dr. Fong relative to the desirable features of a comprehensive creep-fatigue treatment method.

Reference is made by Dr. Fong, for example, to the work of Lord and Coffin on Rene 80 at 1600 F, as shown in Fig. 2 of the paper. Strainrange Partitioning is cited as being inadequate to explain the results of these tests on the ground that the curve for "equal hold" crosses the curve for "no-hold." The point is made that for finite values of N_{cc}, it is highly improbable that the two endurances could be shown to be equal on the basis of the Strainrange Partitioning framework. This, however, is not the case. First, it should be pointed out that the conditions designated "no hold" *do not necessarily exclude the presence of* $\Delta\epsilon_{cc}$. *It simply* means that the material is continuously cycled. But, as the cycle time is increased, the uniform strain rate is decreased for such "no-hold" tests. Thus, there is essentially a short-hold period at each strain level everywhere along its path toward the maximum value, rather than a single long-hold at only the maximum strain. The fact that the life decreases as cycle period is increased along the "no-hold" curve indicates that $\Delta\epsilon_{cc}$ is being introduced. There is, therefore, no reason to assume that there does not exist a cycle period such that the $\Delta\epsilon_{cc}$, gradually introduced, cannot equal the $\Delta\epsilon_{cc}$ introduced during the single longer-time hold (for equal net cycle periods), resulting in equal lives.

Furthermore, there is no reason to suppose that the N_{cp} life is necessarily much lower than the N_{pp} life. If, for example, they are equal, then the life values could be equal at all cycle-periods without violating the framework of Strainrange Partitioning.

It should also be pointed out that Rene 80 is a "maverick" material in which compressive-hold is more damaging than tensile-hold, as seen in Fig. 2, compared to many materials of interest which suffer more from tensile-hold. Strainrange Partitioning has been shown, Reference [10], to be able to explain the behavior of such materials, using 2-¼ Cr-1Mo as the example which shows the same type of behavior. Rene 80 may also be subjected to property changes with longer exposure time (which may explain the increased life with long tensile hold). Again Strainrange Partitioning has been shown to be able to handle materials displaying such aging property changes with time-temperature exposure. Manson[13] treats by Strainrange Partitioning the alloy A-286, in which creep exposure caused progressive changes in ductility; good results were obtained. Thus a wide range of material characteristics, including those displayed by Rene 80 can be accommodated by the Strainrange Partitioning approach.

To illustrate the application to Rene 80 in a more quantitative way, the report by Lord and Coffin was re-examined by the discussors. It was found that all the basic information was actually contained in this report to permit direct determination of the Strainrange Partitioning lives for the 0.0032 strainrange at 1600 F presented in Fig. 2. Thus,

$$N_{pp} = 600; N_{cp} = 450; N_{cc} = 190; N_{pc} = 80$$

These values conciliate most of the results in Fig. 2, and specifically explain:

(*a*) that no-hold can produce about the same fatigue lives as

[9]By J. T. Fong, published in the August 1975 issue of the JOURNAL OF PRESSURE VESSEL TECHNOLOGY, TRANS. ASME, Vol. 96, Series J, No. 3, p. 214.

[10]Professor of Mechanics, Rensselaer Polytechnic Institute, Troy, New York.

[11]Case Western Reserve University, Cleveland, Ohio.

[12]NASA Lewis Research Center, Cleveland, Ohio.

[13]Manson, S. S., "The Challenge to Unify Treatment of High Temperature Fatigue—A Partisan Proposal Based on Strainrange Partitioning," ASTM STP 520, 1973, pp. 744–775.