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GEAP-5622
AEC RESEARCH AND
DEVELOPMENT REPORT
APRIL 1968

A FAILURE DIAGRAM FOR AXIALLY FLAWED PIPES

M. B. REYNOLDS

U.S. ATOMIC ENERGY COMMISSION
CONTRACT AT(04-3)-189
PROJECT AGREEMENT 37

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United States Atomic Energy Commission
Contract No. AT(04-3)-189
Project Agreement 37

Printed in U. S. A. Available from the
Clearing House for Federal Scientific and Technical Information
National Bureau of Standards, U. S. Department of Commerce
Springfield, Virginia

Price: \$3.00 per copy

4197-NL-18
250-DAC-5/68

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A FAILURE DIAGRAM FOR AXIALLY FLAWED PIPES

M. B. Reynolds

ABSTRACT

The relationship between flaw configuration and the maximum pressure load which can be sustained by axially flawed pipes is discussed. Both axial through-wall flaws and part-through flaws are considered. A method of plotting burst strength versus flaw length data to produce a useful "failure diagram" is presented.

INTRODUCTION

The presence of a flaw in any stressed structure, such as a pipe, may be expected to reduce the maximum load which it can support. The design load for an unflawed structure is commonly well below the value at which general yielding would occur; and it is common for a structure made of a ductile material to undergo appreciable plastic deformation before the maximum load is reached. A large enough flaw in any structure can cause a failure at a below-yield load by virtue of simple loss of load-bearing cross section; how large a flaw is required depends on the shape of the flaw and the deformation properties of the material. If the material is brittle, a relatively low load may be enough to bring the local stress at the tip of a sharp flaw to the fracture level and cause the structure to fail as a result of unstable fracture propagation. If the flaw configuration is geometrically simple enough that the elastic stress field about its tip can be determined theoretically or experimentally, the gross stress at which failure will occur can be estimated in terms of a single material constant. This constant is the "plane strain fracture toughness," K_{Ic} , of linear elastic fracture mechanics. Within the range of applicability of linear elastic fracture mechanics, the gross failure stress is almost entirely dependent on material properties and the size and configuration of the flaw and is but slightly dependent on the size of the structure.

If a sharp flaw exists in a ductile material, an increasing load causes plastic deformation to occur first in the region of the highest local stress at the tip of the flaw. Deformation tends to reduce the acuity of the flaw tip and consequently the local stress concentration. Hence a larger flaw is required to produce a failure at a given stress level in a ductile material rather than in a brittle one.

For brittle materials, the mathematics of linear elastic fracture mechanics predict that the product of flaw diameter or length and the square of applied stress at fracture should be a constant. This prediction has been experimentally verified for simple flaw configurations, such as a semi-elliptical surface crack in a large plate under tension normal to the crack plane, an edge crack in a plate under tension, or a through-wall crack in a plate under tension. If the material is somewhat ductile (but not so ductile that plastic deformation extends very far from the flaw tip before failure occurs) the fracture failure stress can still be estimated by correcting the macroscopic flaw length by an amount comparable to the size of the plastic zone at the flaw tip. Since this correction is a function of the ratio of the fracture toughness of the material to its yield stress, another material constant has been introduced into the estimation of the failure stress and some of the simplicity of linear elastic fracture mechanics has been lost. The technique is useful,

however, for dealing with ductile materials in sections of thickness exceeding the size of the plastic zone. For ductile materials in thin sections, linear elastic fracture mechanics is of little or no value in predicting failure stresses under monotonic loading, and empirical techniques must be used.

PIPES WITH AXIAL THROUGH-WALL FLAWS

The plane strain fracture toughness, K_{Ic} , of a material is that value of the stress intensity factor in the expression for the elastic stress field about a crack tip at which unstable plane strain crack propagation sets in under monotonic loading. For a through crack in a wide plate under an applied stress normal to the crack, unstable crack propagation sets in when

$$K_{Ic}^2 = \pi \sigma^2 c \quad (1)$$

in which σ is the applied stress and c is the crack half-length. The similarity between such a crack and an axial through-wall flaw in a large diameter brittle cylindrical shell under internal fluid pressure is obvious; however, as the pipe diameter decreases, it is necessary to take into account bending stresses which are not present in the plate. In addition, since most pipe materials are not completely brittle, a plastic zone correction of some form is also necessary. Eiber and co-workers⁽¹⁾ have developed an expression based chiefly on the work of Folias⁽²⁾ which takes into account both bending stresses and plastic zone correction and gives an effective critical stress intensity factor or crack toughness for axially cracked pipes in terms of hoop stress and crack length. This expression is of the form

$$K^2 = \frac{\pi c \sigma_h^2}{\cos \theta} \left(1 + \frac{5\pi}{32} \lambda^2 \right) \left(\frac{4-\kappa}{2} \right), \quad (2)$$

in which

K = Stress intensity factor, ksi $\sqrt{\text{in.}}$,

c = Crack half-length, inches,

$$\lambda^2 = \frac{c^2}{Rt} \sqrt{12(1-\nu^2)},$$

ν = Poisson ratio,

R = Radius of vessel, inches,

t = Wall thickness, inches,

$$\theta = \frac{\pi}{2} \frac{\sigma_h}{\sigma_c},$$

σ_h = Nominal hoop stress, ksi,

σ_c = Failure hoop stress for unflawed vessel, ksi,

κ = $(3-4\nu)$ plane strain; $(3-\nu)/(1+\nu)$ plane stress.

For very large diameter brittle pipes, Equations (1) and (2) become equivalent. The authors of Equation (2) make no claim for its accuracy at large values of c^2/Rt . The present author has found that application of Equation (2) to data from hydrostatic tests on ASTM A106B pipes with axial through-wall flaws in the range of $0.13 < c^2/Rt < 18.5$ leads to values of K which increase with crack length. ⁽³⁾ This variation in K with crack length is greatly reduced by use of an expression of the form

$$K^2 = \frac{\pi \sigma_h^2 c}{\cos \theta} \left(1 + 0.4 \frac{c^2}{Rt} \right). \quad (3)$$

CRACKS WITH LONG AXIAL PART-THROUGH CRACKS

A pipe containing a long (several pipe diameters) axial part-through flaw of constant depth may be considered analogous to a series-connected pair of cylindrical tensile elements. In this analogy, the ligament remaining at the base of the flaw would be represented by the smaller, plastically deforming member of the pair and the unflawed portion of the pipe would be represented by the larger, elastically loaded element. It may be easily shown that the load maximum for a two-element structure of this kind coincides with the onset of plastic instability in the smaller of the two elements. This coincidence suggests that a pipe containing a long part-through flaw should fail when plastic instability occurs in the ligament at the base of the flaw.

To apply this hypothesis to a pipe containing a long part-through flaw, it is necessary to estimate (a) the three principal stresses effective in the ligament at the base of the flaw and (b) the plastic instability stress for the material under triaxial stress conditions. By denoting axial, circumferential, and radial stresses in a pipe under internal pressure by σ_1 , σ_2 , and σ_3 , respectively, the average principal stresses in the ligament can be estimated in terms of the nominal applied hoop stress σ_h :

$$\sigma_1 = \sigma_h/2, \quad (3a)$$

$$\sigma_2 = \frac{T}{x} \sigma_h, \quad (3b)$$

and

$$\sigma_3 \cong \frac{\nu}{2} (\sigma_1 + \sigma_2). \quad (3c)$$

in which T represents the unflawed wall thickness, x represents the thickness of the ligament, and ν is the elastic Poisson ratio for the material. The ratio of the nominal hoop stress based on unflawed pipe dimensions to the effective stress* in the ligament can be calculated as a function of x/T by using the principal stress values given in Equation (3).

The estimate of the plastic instability stress in the ligament is rather arbitrary. Low carbon steels, such as ASTM A106B, commonly exhibit a rather sharp tensile yield point, and it is not unreasonable to assume that the effective instability stress for such a material is approximately equal to its uniaxial tensile yield stress.

* Effective stress $\bar{\sigma}$ is defined by $\bar{\sigma} = \frac{1}{\sqrt{2}} \sqrt{(\sigma_1 - \sigma_2)^2 + (\sigma_1 - \sigma_3)^2 + (\sigma_2 - \sigma_3)^2}$

The validity of the plastic instability hypothesis has been tested with experimentally determined burst hoop stress values for ASTM A106B pipes containing long part-through flaws. ⁽⁴⁾ Ratios of burst hoop stress to uniaxial tensile yield stress were compared with hoop stress to effective stress ratios calculated by using the principal stress values given in Equation (3). Considering the rather arbitrary choice of a value for σ_3 and the effective instability stress, reasonable agreement was obtained over a considerable range of x/T .

PIPES WITH SHORT PART-THROUGH FLAWS

The pressure load maximum for a pipe containing a short part-through flaw is not so simply related to plastic instability in the ligament at the base of the flaw as it appears to be for long flaws. The series-connected tensile element analogy obviously does not apply to the short flaw case, and more sophisticated computer techniques for dealing with the problem remain to be developed. Experimental tests on specimens containing artificially prepared flaws appear to be the most economical source of information on the behavior of short axial part-through flaws in pipes. Because of the two-dimensional nature of such flaws as contrasted to the axial through-wall or long part-through flaw, considerably more data points are needed to treat empirically the short, through-wall flaw than are needed, for example, to adjust the constants in a relation, such as Equation (3).

THE FAILURE DIAGRAM

A continuing series of hydrostatic tests to determine pressure load maxima for typical ASTM A106B pipes containing axial through-wall flaws and machined external axial flaws of different lengths and depths has been carried out. The part-through flaws in the present series of test were cut with a 2.75-inch-diameter, 45° Vee milling cutter with a 0.010-inch tooth tip radius. Earlier in the program, short part-through flaws were cut in some specimens with slitting saws or abrasive wheels ranging from 0.015 to 0.020 inch in thickness. Full-length axial flaws have been cut with milling cutters ranging from 0.020 to 0.188 inch in thickness. * Specimens with through-wall cracks were prevented from leaking during the hydrostatic test by a laminated patch described earlier. ⁽³⁾ In addition, unflawed pipes were tested to obtain load maxima in the absence of defects. Hydrostatic pressure was supplied by an air-drive, piston-type pump with a 300:1 piston area ratio. Actuation of the pressure recorder chart drive, by electrical pulses from a contact on the pump, made it possible to obtain directly a pressure versus deformation (ΔV) plot for the specimen under test. Test temperature was 60°F ± 2.

Results of tests performed to date, chiefly on 6-inch Schedule-80 pipe are given in Figure 1 in which load maxima, expressed as a fraction of the burst hoop stress for an unflawed pipe, are plotted against the reciprocal of the flaw length. In a "failure diagram" of this type, it is possible to present on one diagram load maximum data representative of different flaw types. In each case x represents the thickness of the ligament at the base of a part-through flaw and T the wall thickness for an unflawed pipe. The lower curve represents load maxima for pipes with axial through-wall flaws. The dotted line at the top of the diagram represents the burst strength of an unflawed pipe. Load maxima for pipes with full-length part-through flaws of constant depth (i. e., $\frac{1}{L} = \infty$) are plotted along the ordinate at the left of the diagram at heights dependent on the

* Burst hoop stress was found to be dependent on flaw depth and rather insensitive to flaw width.

FIGURE 1. FAILURE DIAGRAM FOR ASTM A106B PIPE WITH VARIOUS FLAWS

value of x/T . All points representing load maxima for pipes having short part-through flaws should lie between the curve and the upper dotted line at a location dependent on the value of x/T for the particular flaw. Flaws with low values of x/T give data points near the curve; high values (i. e., shallow flaws) give points near the upper dotted line.

In certain cases, significant extension of the crack beyond its original length occurred; in others, leakage with no crack extension was observed. Where a clear-cut identification of crack behavior on this basis was possible, the plotted data points in Figure 1 are so identified.

A line indicating the ASME code allowable stress for ASTM A106B carbon steel pipe is shown for reference in Figure 1. It will be noted that rather severe flaws are required to reduce the load maximum under monotonic loading to the level of the code allowable stress.

A rather large number of tests is required to define completely the behavior of pipes containing short part-through flaws, and the diagram for A106B pipe as presented at this time cannot be regarded as complete. It appears that with sufficient data it should be possible to prepare for any common alloy a "failure diagram" on which zones representing definite types of crack behavior, such as leakage or catastrophic crack extension, can be clearly delineated. This "failure diagram" method of data presentation is described at this time because the author feels it should prove useful in predicting pipe behavior. The work on ASTM A106B is continuing with this in mind.

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