A Spinning Cylinder Tensile Test Facility for Pressure Vessel Steels

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ABSTRACT

A novel facility for investigating the fracture characteristics of thick section steel under mechanically and thermally generated stresses is described. The basic concept is that the stress distribution in a thick-walled cylindrical pressure vessel may be closely simulated, for purposes of crack propagation from an internal axial flaw, by rotating a section of the cylinder at high speed about its principle axis. The report deals with the necessary design features of such a facility and the background stress and fracture considerations upon which the specification of the tests and the analysis of results will rest. It also outlines operating experience with a small scale prototype of the proposed apparatus.

1. OUTLINE SCHEME FOR A SPINNING CYLINDER RIG

This paper describes a novel facility for investigating the fracture mechanics of thick section steel under mechanically and thermally generated stresses. The facility can be used to induce and measure crack growth on engineering scale precracked specimens of the highly ductile steels used in modern pressure vessels. It will test specimens which are sufficiently big to reproduce the stress conditions which would occur in a large flawed pressure vessel, viz plane strain at the flaw and elastically constrained crack tip plasticity. Also since it is large, it is comparatively simple to apply uniform thermal shocks to the specimen surface so that their influence on crack growth can be studied. Design and construction of the facility has started.

The specimens are in the form of hollow cylinders of the required wall thickness in which a hoop stress is generated by rotation at high speed about the longitudinal axis. Typically, for a high quality pressure vessel steel, the cylinders will have an outer diameter of 1.4m, a wall thickness of 0.2m and a length of 1.3m, requiring a rotational speed of about 3500 rpm to induce yield stress. These dimensions are consistent with dynamic stability and lead to a specimen weight of 8 tonnes.Thermal stresses can be produced by water-spray cooling the inside surface of the pre-heated cylinder.

The method of spinning a cylinder has advantages over tests on pressure vessels made of full thickness steel, or tensile tests using large hydraulic actuators applied to thick plates or cylinders. The major advantages are that it is a comparatively inexpensive method of applying large stresses to large specimens and the specimens themselves are simple, involving no welding. Another advantage is that the spray cooling method allows large and uniform thermally induced stresses without major interaction with the primary stress. Also, as in a pressure vessel,there is considerable stored energy available to drive the crack. The major disadvantages of the rotating cylinder method are those of providing temperature, strain and crack growth instrumentation which will withstand the large centrifugal force, and the need for massive safety containment to allow for the possibility of specimen detachment or disintegration.

Many different design concepts were considered for supporting and spinning the cylindrical specimen. It is essential that the method should accomodate a considerable and varying imbalance of the spinning cylinder since, even if it was dynamically balanced initially, imbalance will occur due to distortion and crack growth during test. The need to heat and thermally shock the inside of the cylinder by spraying with cold water requires open access to the cylinder. This leads to the chosen method of suspending the cylinder from a gimbal mounted bearing in the manner of a free pendulum,using a support disc and semi-flexible shaft as means of interconnection. The bottom of the hollow cylinder is free from obstruction. Driving torque is transmitted through the shaft, which is able to bend elastically to accomodate dynamic imbalance of the cylinder. The conceptual design which emerged is shown in Figure 1. A cylinder freely suspended in this way rotates about an axis which passes through its centre of mass and the pivot point. The fact that the attachment point of the shaft to the support disc does not lie precisely on this axis, because of imbalance, is not normally of consequence. The shaft bends through a small angle and remains at that angle as the specimen rotates, so no cyclic stressing or fatigue damage is induced. Cyclic flexure of the shaft occurs only for a non-synchroneous disturbance where the rotational axis does not pass through the pivot point - a situation that can occur under aerodynamic forces at high speed. The torque to drive the cylinder is governed predominantly by the requirements of the test programme to vary the hoop stress in the cylinder at a specified rate, and also by windage losses at the highest speeds. It has been specified as 1000 Nm maximum. The rotating cylinder is sited below ground level, with massive shielding to provide containment against missiles and rupture.

2. TESTS WITH A SMALL-SCALE PROTOTYPE RIG

The proposed design raised several issues, which required investigation, such as the dynamic stability of the cylinder, the power required to overcome air resistance, the heat transfer coefficient and uniformity of cooling obtained when injecting a cold water spray onto a hot rotating cylinder, and the ability to use essential instrumentation for temperature, strain and crack growth. It was therefore decided to build a small-scale rig to investigate these .

Because of the high energy in the rotating cylinder, the model rig (Fig 2) was built in an existing underground bunker. The power unit is a 1.1L Ford petrol engine with hydraulic torque convertor and automatic gear box. The horizontal drive shaft coupled to the motor transmits through a right-angled 2:1 step-up gear box to a hollow vertical shaft which is flexibly gear coupled to the hollow support shaft of the cylinder. The hollow shafts enable instrument cables to pass from the cylinder to a slip-ring unit above the right-angled gear box. The cylinder is supported, via the shaft, on angular contact ball bearings contained within a spherical bearing for self alignment. This bearing arrangement is supported by a welded channel structure to the wall of the bunker. Two sizes of test cylinder have been used, with external diameter 0.55m and wall thickness 0.05m, lengths of 0.25m and 0.55m, and masses of 153 kg and 337 kg respectively. In essentials the design is similar to the proposed larger rig. It has been operated intermittantly over two years and runs have regularly been made to speeds of 7000 rpm, which produced stresses approaching the yield point of the mild steel used for the specimens.

For fracture tests on the full size cylinders it will be necessary to produce full length axial cracks on the inside wall. These will cause an imbalance of about 0.1% of the cylinder mass, which will be easily accomodated by shaft flexure. To confirm this, tests have been made on the small-scale rig with deliberately introduced out of balance masses equivalent to 0.3% cylinder mass, both on the short and long cylinders, with no obvious effect on the rotational stability of the specimens. Considerable effort has been directed to the development of damper designs to eliminate the aerodynamically induced vibrations which occur at high speed. Although the energy of these vibrations is only a few joules, the induced cyclic bending of the support shaft can lead to rapid fatigue failure. Eventually it was found that a damping system acting directly on the support bearing housing completely suppressed the instability.

There have been few difficulties in using thermocouples to record cylinder temperature. The voltage output is fed through a slip ring unit, and there has been no problem with slip-ring noise. When applying water cooling to the inner face of the cylinder, the water is centrifuged into any available crevice. To avoid spurious readings of through wall metal temperature it is necessary to seal the thermocouple pocket with a high temperature sealant. Resistive strain gauges have also been used successfully to measure the strains induced in the vicinity of the machined groove in the cylinder. Measurements of crack growth in a rotating cylinder have not yet been attempted. It is intended to use the method of measuring the potential drop across the crack in the presence of a high frequency current, a technique which in principle is well suited to a rotating cylinder.

In many engineering applications thermally generated stresses are larger than stresses generated by gas pressure or mechanical load. The thermal stress of most practical interest is induced by a rapid cooling of one face of a pressure vessel. To produce the largest stresses likely to be generated in practice, a heat transfer coefficient of about 7000 watts/m^2 °K is necessary. Tests were made on the model rig to measure the attainable heat transfer.Two 3kW electric heaters mounted inside the cylinder heated it to 300°C from ambient in about 2 hours. A water spray of up to 24 gallons a minute, delivered from two sparge pipes, induced rapid cooling on the inside face measured by thermocouples mounted in drilled holes through the cylinder wall. Tests were made at various rotational speeds of the cylinder. The results were analysed by comparing the measured temperatures with those calculated from the one dimensional heat conduction equation. As would be expected, the heat transfer coefficient varies during the test; it was close to the required value of 7000 watts/m^2 °K in the boiling regime, and somewhat less in the non-boiling regime. The latter effect was caused by the limited water supply available on the model rig, and so a greater flow per unit area of cylinder will be provided on the full size rig.

3. STRESS AND FRACTURE ANALYSIS OF THE CYLINDER

If the rotating cylinder is to be used to study fracture processes which might occur in a pressure vessel it is desirable that the essential characteristics of the stress distribution are reproduced. The stresses in a cylindrical pressure vessel under internal pressure loading are given by the Lame equations and these stresses, ie hoop stress σ_θ, longitudinal stress σ_z, and radial stress σ_r, are plotted in Figure 3 for vessels of different inner radius to wall thickness ratios (a/t). The corresponding stresses in a rotating cylinder have been calculated, assuming a condition of engineering plane strain, and are also plotted in Fig 3. Note that b is the outer radius of the cylinder, ρ is its density and ν is Poisson's ratio.

The main feature of the comparison is the similarity in the form of the hoop stresses. For both the pressure vessel and the rotating cylinder it is the dominant stress and has its peak at the inner surface. For a large pressure vessel and for the rotating cylinder the radial stresses are small. The major difference between the stress distribution in the two cases is the absence of a longitudinal stress in the rotating cylinder. This is not important to the growth of the longitudinal cracks which it is proposed to study using the rig.

3.2 Thermal stress

The magnitude of the elastic stresses generated at any time during a thermal downshock may be calculated from the temperature distribution, using equations for an infinite length thick-walled cylinder as given by Timoshenko and Goodier.(1) The thermal stress distribution is dependent on cylinder wall thickness but is not strongly sensitive to cylinder radius, so small diameter test cylinders of correct wall thickness adequately reproduce the thermal stresses in a large cylinder. The rate of heat transfer from the surface has a large effect on the thermal stress, and the spinning cylinder method can emulate the highest values found in practice.

3.3 Cylinder end effects

The stress evaluations in Figure 3 assume an infinitely long cylinder. In practice end effects arise because both thermal and mechanically, induced axial stress set up in the cylinder must fall to zero at the free ends. The end effect due to the distribution of rotational stress is small but the thermal shocks produce considerable stresses longitudinally. The most significant effect of this longitudinal stress is a bending moment which causes the cylinder to become barrel-shaped during a thermal downshock. The end effects are significant for about one quarter of the cylinder length, but the remaining central half of the cylinder length experiences an approximately constant stress giving an ample length of specimen for the fracture testing.

3.4 Fracture analysis

The test programme, for which the spinning cylinder is being built, is aimed at a more detailed study of the fracture of ductile steels. Fracture of both a full-length longitudinal inner surface crack and a semi-elliptical axial crack,has been analysed using small scale yielding corrections to linear elastic fracture mechanics. The analysis assumed plane strain conditions at the crack, and showed that in all tests the ligament ahead of the crack would not be fully yielded. Stress intensity factors (K_1) were calculated using the method given by Buchalet and Bamford[2] and also by finite elements. An equation was developed to permit interpolation to various cylinder radius to thickness ratios. The calculation of temperature distribution, stress distribution, including end effect correction, and evaluation of stress intensity factors has been written as a computer programme, called SIF,which has been validated by comparison with calculations made by Cheverton and Bolt[3] on a reference PWR pressure vessel. SIF was used to indicate the influence on crack extension of changes in specimen radius, thickness, rotational speed and heat transfer coefficient during thermal shock, for assumed values for the fracture toughness of the steel specimen. The chosen values of cylinder dimensions were based on this assessment.

Fig 4 shows a typical fracture analysis for a proposed test to measure ductile crack growth of a longitudinal crack in the inner cylinder wall under rotational hoop stress. This is an 'R6' failure assessment diagram for an A508 Class 3 steel cylinder with a crack depth to wall thickness ratio (a/t) of 0.45. In the R6 diagram the locus of failure assessment is a function of Kr(the ratio of the stress intensity factor to the plane strain fracture toughness) and Sr (the ratio of the stress to the plastic collapse stress). The example assumes a flow stress of 500 MPa and a crack initiation fracture toughness of 190 MPa $\sqrt{m}$. Crack growth initiation was calculated to occur at a rotational speed of 2325 rpm. The speed required for various amounts of ductile crack growth depends on the assumptions about the increase of fracture toughness. For the example in Fig 4, a lower bound curve for the increase in toughness with crack length, based on small-scale metallurgical tests, was taken. It is seen that stable ductile crack extension of several millimetres with increasing speed is predicted before unstable tearing and plastic collapse of the cylinder eventually occur. Such predictive methods will be used to analyse the test data.

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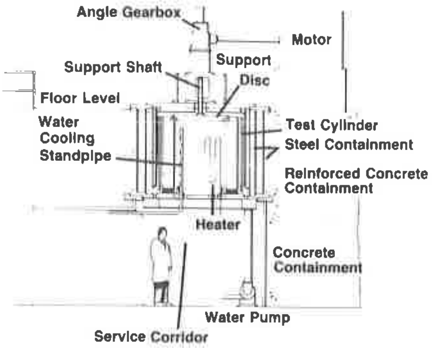

FIG 1 CONCEPTUAL ARRANGEMENT OF A PROPOSED FULL-SCALE TEST RIG

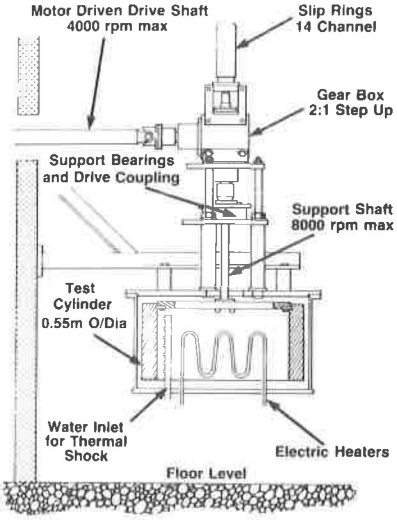

FIG 2 ARRANGEMENT OF MODEL RIG

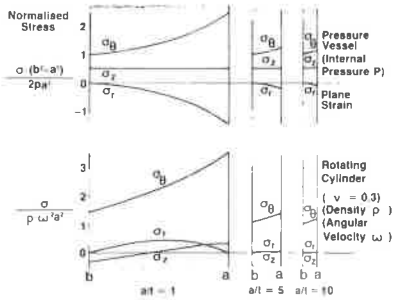

FIG 3 COMPARISON OF STRESSES (PRESSURE VESSEL V ROTATING CYLINDER) AT DIFFERENT RATIOS OF RADIUS (a) TO WALL THICKNESS (t)

(1) a/t initial = 0.45
σ_f = 500 MPa
K_I = 190 MPa $\sqrt{m}$, lower bound K_R

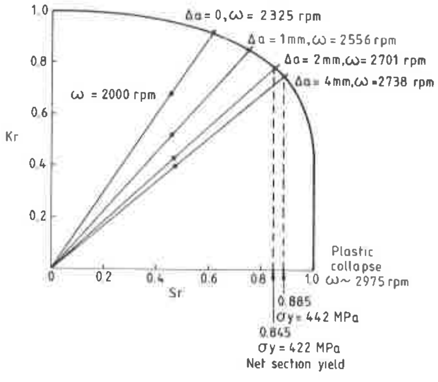

Fig 4 R-6 crack growth analysis : a/t = 0.45, σ_f = 500 MPa, K_I = 190 MPa $\sqrt{m}$, lower bound K_R