# THE FATIGUE DESIGN OF GAS STORAGE SYSTEMS USING FRACTURE MECHANICS

**P. Hopkins and D. G. Jones**

*British Gas Corporation, Engineering Research Station, Newcastle upon Tyne, NE99 1LH, England*

## ABSTRACT

This paper describes the analytical and experimental studies conducted to produce a suitable fatigue design curve for large scale underground high pressure gas storage vessels. These vessels can be operated at stress ranges up to 2/3 yield stress magnitude and are particularly difficult to inspect during service.

Full scale fatigue tests on vessels containing artificial defects and fatigue fracture mechanics analyses allowed the fatigue lives of the vessels to be quantified. These studies and subsequent ductile and fatigue crack growth fractography of the failed test vessels were used to demonstrate the ability of a periodic high level pressure test to define a safe fatigue life. Therefore, major periodic inspections can be replaced by direct revalidation with a hydrotest.

## KEYWORDS

Fatigue design; gas storage vessels; fracture mechanics; pressure test; periodic inspections.

## INTRODUCTION

Industrial and domestic demand for gas has never been constant and there has always been a need for storage systems to store gas during the night and release it during peak periods in the daytime (Sykes, Brown, 1975).

Gas storage can be achieved using a variety of plant and methods. An attractive form of storage is known as a below-ground pipe array. These pipe arrays are generally 1066 mm diameter linepipe laid underground in parallel rows with a common manifold system, Fig. 1. By definition, these facilities will be subjected to fluctuating stresses and therefore must be designed to resist fatigue.

Pressure vessel and pipeline standards normally require a vessel/pipe subjected to cyclic operation to be designed to prevent fatigue failure. Although these standards have this common aim, the approaches adopted differ significantly. The approaches can be summarised as:

i) Pressure Vessel Approach

UK and American pressure vessel codes, BS.5500 (Anon., 1982) and ASME VIII (Anon., 1976), incorporate fatigue design curves based on data from nominally defect free, plain or welded specimens; the philosophy being that this type of specimen simulates a vessel that has passed rigorous inspection requirements and contains no 'major' defects.

ii) Pipeline Approach

In the United Kingdom, the Institution of Gas Engineers are soon to issue guidelines on the fatigue design of transmission pipelines. These guidelines, Edition 2 of IGE/TD/1 (Anon., 1977), will contain fatigue design recommendations based on experimental studies (Fearnehough, Jones, 1978) on linepipe containing defects just able to survive a hydrotest (proof test). The philosophy is that a successful hydrotest demonstrates a definable fatigue life for a pipeline taking account of defects remaining after the test. This approach has the major advantage that at the end of a defined fatigue life, a pipeline can be re-hydrotested to demonstrate its fitness for further service without recourse to major inspections.

As pipe arrays are buried, periodic inspections are both costly and introduce the risk of damage during excavation. Therefore, the 'Pipeline Approach' to fatigue design is particularly attractive in that inspection can be replaced by direct revalidation with a hydrotest. The IGE/TD/1 recommendations for pipelines are based on extensive experimental studies on linepipe, but data at stress levels relevant to pipe arrays ($>124$ MN/m$^2$) were relatively sparse. Therefore, before recommending fatigue design criteria for pipe arrays, a further test programme was undertaken at the Engineering Research Station of British Gas. This paper details the full scale fatigue tests, Section 2, and analytical work, Section 3, which formed the basis of the resulting design curves.

## FULL SCALE TESTS ON PIPE ARRAYS

Eight fatigue tests were carried out on 3500 mm lengths of 1066 mm diameter x 14.3 mm wall thickness Grade 5LX60 pipe, each containing a flat-bottomed defect, 533 mm to 1067 mm long x 0.15 mm wide, machined into the root of the seam weld as shown in Fig. 2. Each pipe was then made into a pressure vessel by welding on domed ends, Fig. 2.

For the first three vessels tested, the defects had depths such that they just survived a hydrostatic pressure test to a stress of 436 MN/m$^2$ (105% SMYS). Following the hydrotest, and 24 hour hold period, each of the three vessels was then fatigued to failure at a constant stress range (276, 241 and 186 MN/m$^2$ respectively). For the remaining five tests, the defect in each vessel was machined approximately 50% shallower than the initial tests, to allow for appreciable fatigue crack growth. Each vessel was then subjected to a hydrotest at regular prescribed intervals during fatigue to simulate revalidation procedures.

Table 1 gives details of all the above tests and Fig. 3 plots all the above and previous data (Fearnehough, Jones 1978) in standard S-N format. In the tests which involved repeated hydrotests the regions and depth of fatigue crack growth could be determined from fracture surface markings, Fig. 4.



## ANALYTICAL STUDIES

i) Basis of Studies

The maximum size of credible defect able to survive the hydrotest can be estimated using standard formulae (Shannon, 1974). Similarly, the maximum defect size able to survive the peak operational stress can be estimated. The difference between these two defect sizes is available for fatigue growth and the rate of defect growth can be calculated using standard fracture mechanics formulae such as the Paris Law (Paris, Erdogan, 1963). Consequently the fatigue life of a pipeline can be defined for a given hydrotest and operational stress level. In the following section this type of analysis is performed on the full scale tests.

ii) Analysis

The maximum sized defect which can survive a pipeline proof test ($a_i$) and the critical defect at operating pressure ($a_f$) are determined from the equation for failure through-the-wall of a part-wall, infinitely long defect (Shannon, 1974):

$$\delta_H = 1.15\ \delta_Y \quad (1 - \frac{a}{t}) \qquad (1)$$

where $\delta_H$ = hoop stress at failure, $\delta_Y$ = yield stress of pipe

$a$ = defect depth, $t$ = wall thickness

From $a_i$ and $a_f$, the number of cycles, N, to failure can then be calculated from the Paris crack growth equation (Paris, Erdogan, 1963).

$$\frac{da}{dN} = C\ (\Delta K)^m \qquad (2)$$

where $\frac{da}{dN}$ = crack growth per cycle, $\Delta K$ = cyclic stress intensity factor

C and m = material constants.

The stress intensity factor (K) for a defect in a pressurised pipe at the deepest penetration point on the defect periphery is given by (Erdogan, Ratwani, 1974):

$$K = \delta_H \sqrt{\pi a} \quad \frac{C_t\ C_c}{Q} \qquad (3)$$

where $\delta_H$ = hoop stress, $a$ = defect of depth

$C_t$ = thickness correction factor

$C_c$ = shell curvature correction factor,

$Q$ = defect shape and plasticity correction factor

The fractography confirmed that the defects grew in reasonable accordance with the Paris crack growth equation. Figure 5 plots the results of the analysis showing the stress intensity range ( $\Delta K$) versus crack growth rate per cycle (da/dN)[1]. From the plot of (da/dN) versus ( $\Delta K$), regression methods provided a mean fit to the data with material constants C and m in the Paris equation, $2.42 \times 10^{-15}$ and 4.9 respectively (m/cycle, $MNm^{-3/2}$). These results laid the foundation for the proposed design curves and Fig. 6 shows the relationship between predictions made using the above growth law and empirical material constants and the full scale vessel tests.

## DERIVATION AND DISCUSSION OF FATIGUE DESIGN CURVES FOR PIPE ARRAYS

The full scale tests and analytical curve, summarised in Fig. 7, provide a sound base for recommending design curves for pipe arrays (with the exception of a single data point[2].

Having established an empirical/analytical curve it is standard practice to apply suitable safety factors before considering a design curve. A safety factor of 10 is compatible with factors generally applied to the prediction of the fatigue life of engineering structures from laboratory fatigue data.

A design curve for storage arrays incorporating a safety factor of 10 is shown in Fig. 8. The design curve defines the fatigue life which is guaranteed by successful repeat hydrotesting to approximately 1.5 times the maximum operational stress, thus eliminating the need for a major inspection programme.

An alternative design curve incorporating a reduced safety factor of 3 is also shown in Fig. 8 in recognition of the following factors:-

i) The fatigue data have been generated using full scale tests on the actual component i.e. the stresses and service conditions have been accurately simulated.

ii) Safety factors are used to accommodate unknowns in fabrication, operation, etc. Pipe arrays are fabricated, operated and maintained to the highest engineering standards and are buried on secure sites where damage is unlikely.

iii) Pipe arrays are simple structures (lengths of pipes) in which stress concentrations (e.g. nozzles) and fabrication defects (e.g. ovality) are either absent or easily prevented. Similarly temperature fluctuations are neglible.

Adoption of this curve allows the pipe array to be designed for a higher fatigue duty. However, use of this curve would only be permitted after a detailed analysis of the entire pipe array system (including materials) to confirm that conditions (i)-(iii) are satisfied.

Finally it is of interest to compare the recommended design curves with the pressure vessel standard BS.5500 (Anon, 1982). Figure 8 shows the fatigue design curve from BS.5500 to fall within the two curves. This illustrates the conservative nature of the BS5500 curve as the pipe array curves are based on vessels with major seam weld defects whereas the BS5500 curve is based on fatigue tests on butt welds. However it is reassuring to note that the BS5500 curve can accommodate the full scale test data reported in this paper.

[1] Bars attached to points on the graph shown in Fig. 5 represent the range in $\Delta K$ for the corresponding value of da/dN.

[2] This has been shown to be the result of inadvertent equipment failure not suspected at the time of the tests in 1974. The result is not corroborated by any other of the full scale test results, nor by 20 similar tests on defective pipe ring samples, (Fearnehough, Jones, 1978).



## CONCLUSIONS

Full scale fatigue tests and fracture mechanics analyses have allowed the fatigue life of pipe arrays to be quantified. The design curves obtained from these data can be used to define a safe fatigue life guaranteed by a hydrotest. Consequently major periodic inspections can be replaced by hydrotests as a means of revalidation.

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TABLE 1 Full Scale Tests on Pipe Array Vessels, 1067 mm Diameter x 14.3 mm wt, X60

| Test Number | Defect Length 2c, mm | Mean Defect Depth a, mm | Cyclic Stress Range $(MN/m^2)$ | Cycles to Failure After Hydrotest | Time of Failure | Yield Stress of Pipe $(MN/m^2)$ |
|---|---|---|---|---|---|---|
| A | 787 | 2.3 | 186 | 21,470 | Fatigue | 434 |
| B | 787 | 2.5 | 243 | 7,858 | Fatigue | 434 |
| C | 787 | 2.8 | 276 | 5,125 | Fatigue | 434 |
| 1 | 686 | 1.35 | 276 | 3,073 | Fatigue++ | 434 |
| 2 | 686 | 1.32 | 241 | 7,000 | Sixth Hydrotest | 455 |
| 3 | 737 | 1.37 | (a) 186* (b) 276 | 5,000 | Second Hydrotest | 455 |
| 4 | 762 | 1.60 | 276 | 3,000 | Fourth Hydrotest | 455 |
| 5 | 737 | 1.40 | 241 | 6,000 | Third Hydrotest | 455 |

*Test 3 survived 9 hydrotests and 8 periods of 21,500 cycles at 27,000lbf/in$^2$. Vessel failed on hydrotest after 5000 cycles at cyclic stress range of 40,000 lbf/in$^2$.

++Failed during fatigue subsequent to fourth hydrotest.

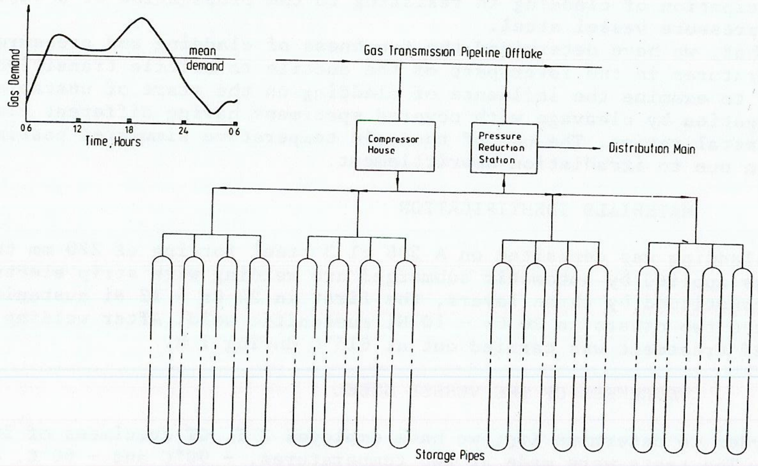


*FIG 1:* TYPICAL SCHEMATIC ARRANGEMENT OF A PIPE ARRAY STORAGE FACILITY



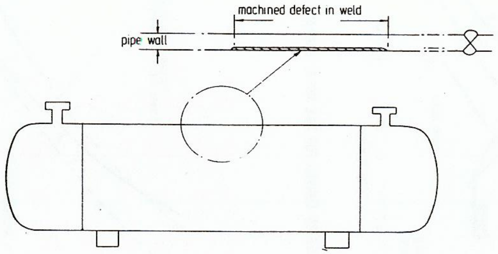


*FIG 2 :* SCHEMATIC OF TEST VESSEL WITH SEAM WELD DEFECT

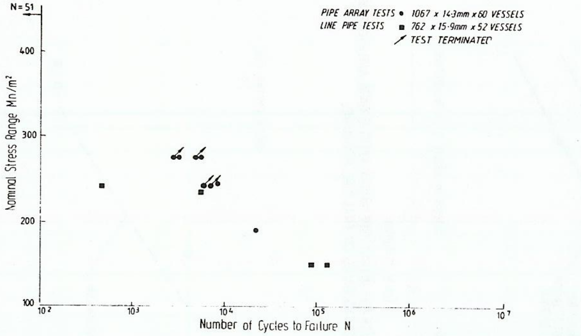


*FIG 3:* SUMMARY OF BRITISH GAS FULL SCALE FATIGUE TEST ON LINEPIPE & PIPE ARRAYS

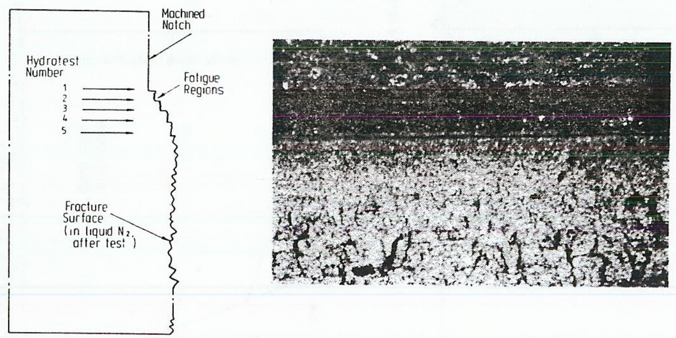


*FIG 4 :* FRACTURE SURFACE OF MACHINED DEFECT SHOWING CRACK GROWTH

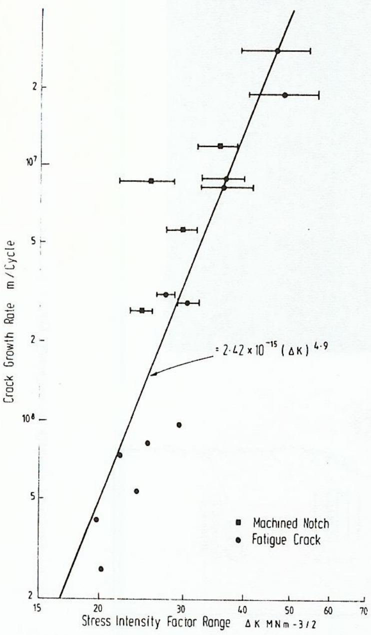


FIG 5: FATIGUE CRACK GROWTH DATA FROM FULL SCALE TESTS

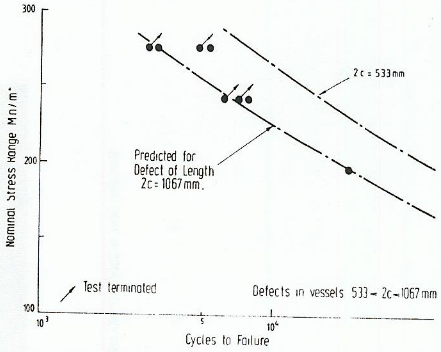


FIG 6 (a) COMPARISON BETWEEN RECENT PIPE ARRAY FATIGUE DATA AND PREDICTIONS BASED ON FRACTURE MECHANICS

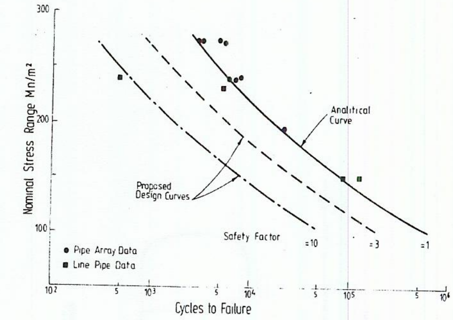


FIG 7: PROPOSED FATIGUE DESIGN CURVES FOR PIPE ARRAY STORAGE FACILITIES

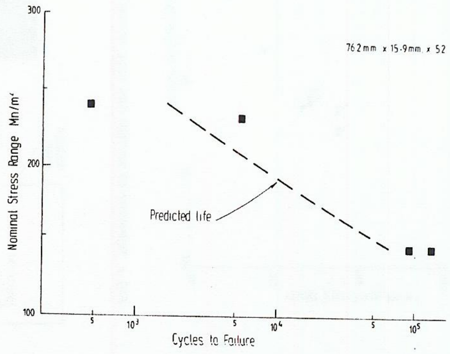


FIG 6 (b) COMPARISON BETWEEN PREVIOUS LINEPIPE VESSEL TESTS AND FRACTURE MECHANICS PREDICTIONS

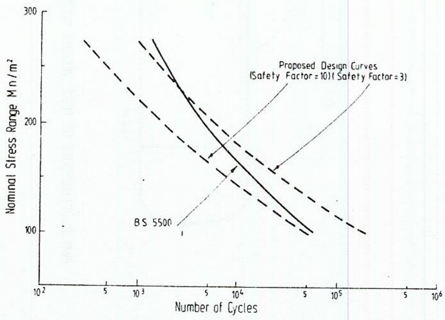


FIG 8: COMPARISON BETWEEN PROPOSED PIPE ARRAY DESIGN CURVES AND B S 5500