INTEGRITY EVALUATION OF POWER PLANT COMPONENTS BY FRACTURE MECHANICS

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ABSTRACT

Power plant components can be subjected to unexpected failures with serious consequences, unless careful attention is paid to minute cracks and defects and their possible growth. The application of linear elastic fracture mechanics (LEFM) to structure integrity evaluation of power plant components is illustrated through several case studies. Projects related to material data generation and the development of structural analyses methods to make LEFM workable are described. Comments on the possible future development, in the next ten years are included.

KEYWORDS

Pressure vessel; transients; flaws; limit loads; fatigue crack growth; stress intensity; fracture toughness; turbine disc; personal computer.

INTRODUCTION

There has been significant development over the past ten years in the field of structural integrity (SI) evaluation. It is now generally accepted that all welded structures have small imperfections. The concept that all structures contain potential defects entails a substantial responsibility, in that the condition for defect extension must be limited on a quantitative basis.

Consider a structure in which a crack exists. Due to the application of repeated loads or due to a combination of load and environmental attack, the crack may grow with time. In the presence of the crack, the load bearing capacity of the structure may be reduced. The residual strength of the structure decreases progressively with increasing crack size. In order to ensure the safety of the structure, the rate of crack growth and the residual strength of the structure at a given time and at the end of the design life must be predicted. The aim of the fracture mechanics related work conducted by Ontario Hydro is to develop and evaluate methods for flaw-structure interaction analysis, conduct SI evaluation of power plant components and to determine fatigue and fracture properties of generating plant materials. The best way to illustrate the SI evaluation of power plant components is to use several case studies.

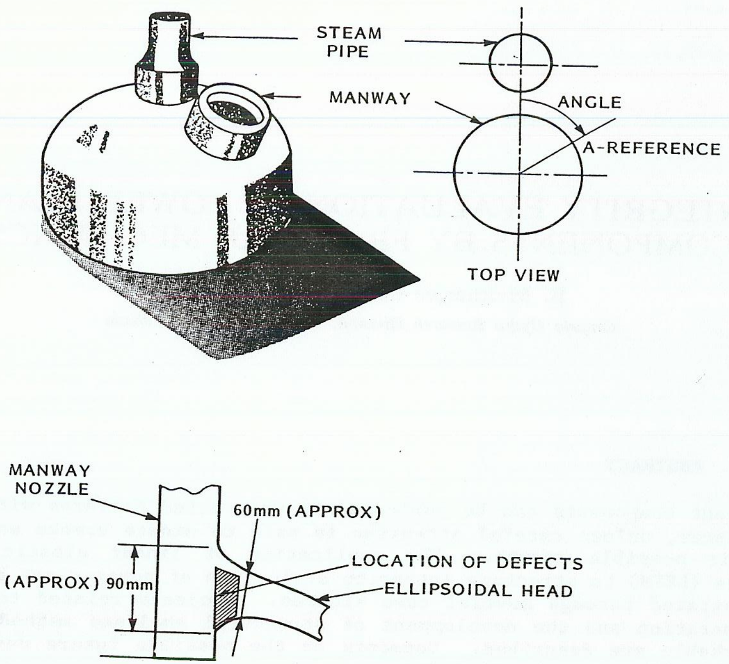

Fig. 1. Boiler upper head and section through manway (schematic).

MANWAY DEFECTS IN A STEAM GENERATOR

Figure 1 shows a sketch of a pressure vessel upper head. It also shows a section through the manway indicating the approximate location of defects. To characterize the flaws several simplifying assumptions were made. For example, although the defects were scattered throughout the weld zone, they were considered to be coplaner for the purpose of the analysis. The non-destructive inspection results reported several indications. These were measured using both 45° and 60° ultrasonic probes. No attempt was made to identify redundant indications; the size of overlapping indications was taken as those of the enclosing rectangle as indicated in 'the ASME Section XI Code. These assumptions increase the severity of the flaws for the purpose of analysis.

During operation, the vessel may be subjected to many different transients. Each transient is associated with a change of pressure and temperature and produces a specific distribution of membrane and bending stresses which act normal to the defects. The fatigue analysis, therefore, must consider the growth of all indications due to the different stress states with due attention to the interaction between the flaws. This analysis should also use appropriate fatigue growth data for surface and subsurface cracks including the effect of load ratio on the data. This case study illustrates our approach to this complex problem.

The assessment was made on the basis of the ASME Boiler and Pressure Vessel Code, Section XI. This document provides tables of flaw indication for various pressure retaining components which are considered as allowable. Naturally, these flaw sizes are very small. Indications which are not "acceptable" are "conditionally acceptable" provided their growth to the end of service lifetime or next inspection is calculated and during this period, the maximum stress intensity factor K satisfies:

$$K < K_{Ia}/\sqrt{10} \quad \text{(normal condition)}$$

Where K_{Ia} is the arrest toughness given in the Section XI. The primary stress limits of the code should also be satisfied assuming a local area reduction of the pressure retaining membrane that is equal to the area of the detected indications.

Stresses

Stresses acting at the defect location were obtained from a three dimensional finite element analysis of the steam generator secondary side manway to head weld for the following cases:

(i) bolt loads
(ii) internal pressure
(iii) maximum thermal transient (at end of heatup)
(iv) worst case pipe reactions (due to the steam pipe)

The distribution of the membrane and bending stresses acting normal to the defects are shown in Table 1. These are for 'normal' conditions. For emergency conditions (eg, crash cool down) the thermal stresses are compressive and the pressure drops, consequently, the (Level A) heatup transient produces the most severe stresses. The temperature amplitude associated with other normal and upset transients are small (less than 15°C) compared to heat up (211°C). Stress amplitudes for these transients are dominated by the pressure amplitude alone.

TABLE 1 Total Membrane and Bending Stress Acting Normal to Defects

Location	Membrane		Bending (1)	
	MPa	Ksi	MPa	Ksi
0	188	27.2	230	33.3
30	165	24.0	194	28.1
60	141	20.5	184	26.7
90	125	18.2	179	25.9
120	101	14.6	162	23.5
150	76	11.0	125	18.1
180	57	8.3	116	16.8

(1) Inside surface is in compression relative to outer surface.

Fatigue Growth Analysis

Analysis of the growth of the defects due to cyclic loading was performed numerically and a computer program was developed for this purpose. The program treats all the indications simultaneously in order to account for their changing interactions (due to the characterization rules of the Section XI) during growth, and performs the following sequence:

(i) individual indications are characterized according to the code rules (combining defects if 'near enough', treating subsurface as surface defects if 'too close' to the surface, etc),

(ii) since membrane and bending stresses vary around the manway, a search for those producing the maximum ΔK for a given characterized defect is performed,

(iii) the size of each defect is then increased based on its maximum ΔK and the appropriate fatigue growth data (eg, for inside surface cracks steam environment data are used, for subsurface and outside surface cracks air environment data is used),

(iv) the size of individual indications are then increased (length and vertical extent) proportionately to their enveloping 'characterized' defect,

(v) the steps above are then repeated.

For this component one design life corresponds to 500 heat-up/cool-down cycles and 15250 cycles due to other transients such as power maneuvering, frequency control, turbine trip etc. The numerical analysis was performed considering only heat-up/cool-down transients. The effect of other transients was investigated analytically. The calculation indicates little change up to about 10,000 heat-up/cool-down cycles, since the largest and fastest growing defects remain subsurface. The effect of including all transients is estimated to be small. For subsurface defects or surface defects exposed to steam with $\Delta K < 19.5$ MPa$\sqrt{m}$, a reduction in the calculated number of design lives in the order of 10% is then expected. However, if defects were exposed to steam at $\Delta K > 19.5$ (during all transients) the reduction could be as large as 57%. Specific results are given in Table 2. The maximum stress intensities remain well below the allowables which are 69.5 MPa$\sqrt{m}$ for normal and upset and 155.3 MPa$\sqrt{m}$ for emergency and faulted conditions. The material properties as used in this analysis were from ASME, Section XI.

TABLE 2 Flaw Size and Stress Intensity Factor

Total Verticle Extent (mm)		Total Length (mm)		Max K (MPa$\sqrt{m}$)	
Initial	At 17 Design Lives	Initial	At 17 Design Lives	Initial	At 17 Design Lives
22.0	22.7	47.0	67.7	26.8	29.4

Primary Stress Limits

The ASME Section XI code rules require that the primary stress (Tresca equivalent stress) due to mechanical loads (pressure and pipe reactions) remain below the allowable limit (1.5 Sm = Yield Stress, Sy) when the wall thickness is reduced because of the presence of defects, as characterized by the Code. In the present analysis, the maximum defect size was governed by the primary stress limits.

It is suggested that the calculated minimum wall thickness and therefore maximum allowable defect size may be overly pessimistic because the finite length of defects is not taken into account. If yielding were to occur at some location, a redistribution of loads around the manway would be expected. An appropriate method for analyzing this effect is through plastic limit analysis as is allowed for in Section III NB-3221, NB-3228-2.

THERMAL PLANT CASE STUDY

The structural integrity of components in thermal plants is of major interest to utilities. Large components, such as headers, turbines and generators might fail catastrophically and cause major safety problems as well as economic loss due to downtime and component replacement costs. Components may undergo increasing fatigue cracking, with unit age and the incidence of cyclic operation. The most obvious result is corrosion fatigue cracking in waterwalls, economizers and low temperature headers.

On January 5, 1981, a stub tube failed on the economizer inlet feeder of a coal-fired power station. Metallographic examination showed severe cracking on the inside of the tube, and on the header outlet hole in an axial direction (shown schematically in Figure 2). The cracks were present around most of the inside of the stub tube, and were the typical broad, oxide-filled cracks associated with corrosion fatigue. Removal of the header handhole cover at one end showed substantial internal cracking. Examination of other economizer inlet headers in identical units and others of different design showed similar cracking, though generally not so severe.

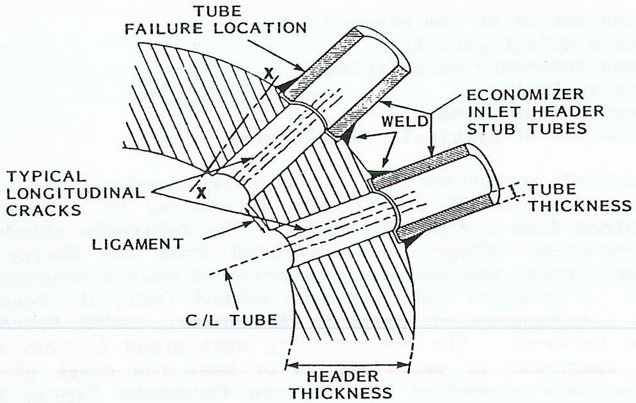

Fig. 2. Typical cross-section through economizer inlet header at tube attachments (Courtesy of D. Sidey).

The two headers in the worst unit were replaced immediately; three others are scheduled for later replacement. Destructive examination of one of these two headers showed that the worst cracks were about 1/3 through the 62 mm wall (see Figure 3). The presence of cracks in a pressure vessel obviously has implications with respect to the structural integrity of the vessel so a program was developed to (1) perform a fracture mechanics analysis to determine if the cracked header would 'leak before burst' and (2) develop a nondestructive evaluation (NDE) technique to measure crack depths in headers in service.

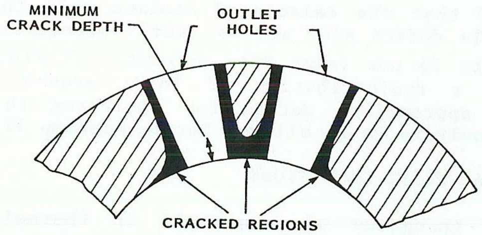

Fig. 3. Schematic showing minimum crack depth in header.

Fracture Mechanics Analysis of an Economizer Inlet Heater

The procedures adopted for a leak-before-break analysis involve first, a stress analysis and second, a fracture analysis. Considering the primary pressure stresses and the stresses arising from a thermal shock applied to the header internal diameter, the stresses at the critical location on the header were determined (Hoff and Byrne, 1982).

Using this stress data, critical crack lengths for brittle failure (following linear elastic fracture mechanics) and for ductile burst (using limit analysis) were evaluated. The method of analysis is similar to those given by Chell (1979) and Zahoor and Kanninen (1981). The following aspects were addressed in this analysis:

1. Stress acting on the assumed defect,
2. Assumed defect geometry,
3. Stress intensity calculation,
4. Limit analysis,
5. Material Properties,
6. Estimation of critical crack lengths.

The mechanical properties and the fracture toughness values of the actual header material (SA106B steel) were determined by testing specimens which were machined from a replaced header. The reference nil-ductiblity transition temperature (RT_{NDT}) was estimated from the Charpy impact absorbed energy and lateral expansion curves measured over a temperature range of 0°C to 100°C. A proposed ASTM standard method (E813-81, Standard Test Method for J_{IC}. A Measure of Fracture Toughness. ASTM Standards. Part 10, 1982) was followed. The property J_{IC} determined by this method characterizes the toughness of materials at or near the onset of crack extension. J_{IC} values were converted to fracture toughness $K_{IC(J)}$ by the following relationship where E is Young's modulus:

$$K^2_{IC(J)} = J_{IC} \cdot E \qquad (1)$$

TABLE 3 Fracture Toughness Data for SA106 B* Steel

Sp. ORIENTATION	TEST TEMP °C	σ_{FLOW} N/m²	MAX J_{IC} MEASURING CAPACITY OF SPECIMEN kN/m	J_Q kN/m	J_{IC} kN/m	K_Q MPa$\sqrt{m}$	$K_{IC(J)}$ MPa$\sqrt{m}$
L-R	23	327	274	153	153	177	177
	100	304	255	>255	>255	>229	>229
	200	293	246	366	>246	275	>225
	300	293	246	354	>246	270	>225
C-R	23	327	141	149	>141	175	>170
	100	304	131	213	>164	209	>164
	200	293	126	192	>161	199	>161
	300	293	126	130	>126	163	>161

* Chemical Composition 0.3 C, 0.52 Mn, 0.025 S, 0.22 Si, 0.014 P; Yield 214 MPa, UTS 414 MPa

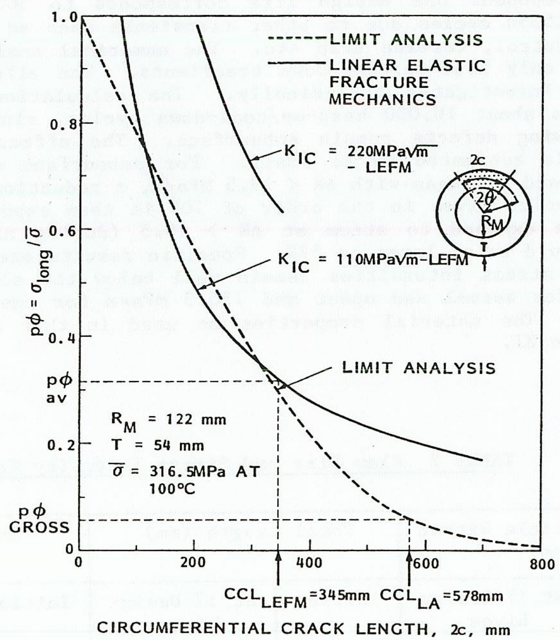

Fig. 4. Failure analysis diagram for circumferential through wall defect in inlet header at 100°C.

Table 3 is a summary of all test results. The results from the linear elastic fracture mechanics and limit analysis are shown in Figure 4. It shows the ratio of assumed longitudinal stress to flow stress ($p\phi$) at which either the fracture toughness or limit load is achieved for a given defect length. Two curves are plotted corresponding to two values of fracture toughness, and also a curve corresponding to the limit analysis. The ratio $p\phi$ is based on thermal plus pressure stress for the two curves corresponding to allowable fracture toughness, and on pressure stress alone for the limit-load curve. The critical crack length at the mid-thickness for a through wall circumferential flaw may be obtained from Figure 4, based upon the geometry of the inlet headers.

TABLE 4 Critical Crack Length (CCL) Estimates

Basis	CCL (mm)	CCL / L+2D*
Limit Load	578	6.9
LEFM, K_c = 161 MPa$\sqrt{m}$, @ 300°C	505	6.0
LEFM, K_c = 170 MPa$\sqrt{m}$, @ 23°C	>570	6.8
LEFM, K_c = 110 MPa$\sqrt{m}$	345	4.1

* L+2D = mean circumferential ligament length + 2x tube inner diameter.

This critical crack length needs to be compared with the physical dimensions of the inlet header in the region of the tube attachments. This has been carried out in Table 4 where the value of critical crack length is compared with the dimension L + 2D where L is the remaining ligament between the two inlet header stub tubes and D, the internal diameter of the tubes. Here L + 2D is taken to be the assumed worst case flaw, ie, complete failure of the ligament between the two stub tubes. From Table 4 it is apparent that a continuous through wall crack of at least 4.1 times longer than the assumed worst case flaw, ie, complete failure of the ligament between tube pair in the circumferential direction, would be required to initiate a brittle fracture around the circumference of the inlet header. Any failure of the headers would, therefore, occur by leak and not break.

Fatigue Life Prediction of an Economizer Inlet Header

Experimental and analytical work described in the previous section indicated that the header would leak, rather than burst should a failure occur. This section attempts to predict the number of operating cycles that an existing or postulated flaw might endure before propagating through the vessel wall. Special consideration was given to the effect of warmup and cooldown rates on flaw growth since the stress field is dominated by thermal effects and these are, to some extent, controllable.

The following procedure was adopted to predict the cyclic life of the headers, considering only fatigue growth of pre-existing flaws:

(i) simplify the defect geometry as a representative single, fully longitudinal or circumferential flaw,

(ii) simplify the detailed stress distributions and separate the pressure and thermal contributions,

(iii) combine pressure and thermal stresses according to postulated operating transient,

(iv) final tolerable flaw sizes were arbitrarily selected to be about 60% of the wall thickness. A check was subsequently made to verify that the fracture toughness was not exceeded for the postulated loading,

(v) predict the number of cycles required to extend defects to their final size for a matrix of combinations of initial sizes and through wall temperature differences.

In this analysis, the results were bracketed between the maximum and minimum stress levels. Some conservatism was built into the assumed flaw models and their final size. The basic approach employed was to provide as realistic a prediction as possible within our understanding of the current technology, as opposed to demonstrating the satisfaction of preestablished safety margins.

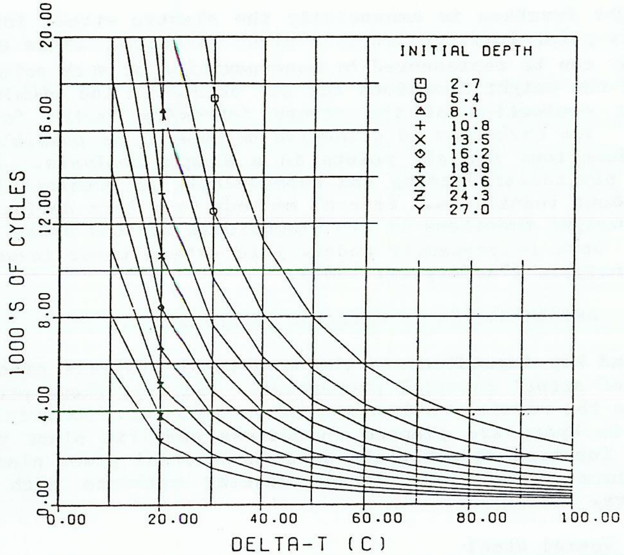

Fig. 5. Fatigue life predictions from maximum stress circumferential flaw model.

Results are presented graphically as a series of curves giving the predicted fatigue cycles versus through wall temperature difference obtained from thermocouple measurements, each curve corresponding to a given initial flaw depth (Figure 5). The trends in the fatigue predictions shown in Figure 5 are as expected; a decrease in life with inital flaw depth and increasing through wall temperature difference, ΔT. Evidently the thermal membrane contribution to the stress intensity factor dominates the bending effect, and its reduction with ΔT is responsible for the increase in predicted cycles in the circumferential ligament.

To summarize, this project on the thermal fatigue cracking of economizer inlet headers established the structural integrity of these components using fracture mechanics providing information usable by operations personnel.

METHODS FOR STRESS INTENSITY FACTOR DETERMINATION

The calculation of a crack driving force for particular flaws and stress distributions is a key ingredient of the flaw evaluation process. Many solutions for particular geometries and loadings are available in the open literature. However, situations often arise in which the stress intensity factor K (crack driving force for a linear elastic case) or the J integral values (crack driving for elastic-plastic case) must either be estimated or calculated by a mathematical analysis. Estimation of stress intensity factors from two dimensional models or from the peak stress on a section instead of the actual distribution, may lead to overly pessimistic results. Calculation of stress intensity factors is usually done by employing one of a variety of numerical methods (Rice, 1972; Vanderglas, 1980). The weight function technique provides an efficient means for translating stresses in an uncracked component into stress intensity factor and research is still being directed by Ontario Hydro toward refining this techique.

The weight function is essentially the elastic stress intensity factor due to a unit point load applied at a given location. Since any loading applied to a body can be represented by superposition of such point loads, the knowledge of the weight functions for all points on and within the region allow a direct evaluation of the stress intensity factor for the distributed loading. The technique is effective because it is possible to calculate the weight functions for all points in a single analysis. Stress intensities for any particular loading can subsequently be obtained from this information without reanalysis. Several methods have been devised for the calculation of weight functions in two dimensional regions using the finite element method. Work is presently underway to extend these ideas for three dimensional analysis (Vanderglas; 1983).

DETERMINATION OF MATERIAL PROPERTIES

The second key ingredient in the structural analysis process is the availability of actual material properties. Ideally, these properties should be known for the service environment of a component. Material related projects at Ontario Hydro are planned to address specific plant related issues and they are focussed on materials used in critical power plant components such as pressure vessels and large rotating machines such as turbines and generators.

Pressure Vessel Steel

Large vessels such as steam generators, pressurizers, headers and condensers in Canadian nuclear stations are fabricated from SA516 GR 70 pressure vessel steel. Experience has shown that these steels provide large tolerance against general corrosion and fracture. Ontario Hydro has made extensive use of the material data for low carbon steel (σ_{yield} < 345 MPa) given in the ASME Boiler and Pressure Vessel Code, Section XI, in the structural integrity consideration of these vessels. These data are in the form of fracture toughness as a function of a normalized temperature and fatigue crack growth rate as a function of the range of stress intensity factor and load-ratio for air and water reactor environment. In order to justify the use of the Code data, experiments were conducted to determine fatigue crack growth rates of SA516 GR 70 pressure vessel steel, associated welds and

SA106B piping steel and the results were compared with the Section XI air reference curve (Mukherjee, 1980; Mukherjee and Vanderglas, 1980).

In these experiments, standard compact tension specimens were subjected to cyclic loads. The experimental matrix included three different plates; two orientations where the crack plane was perpendicular and parallel respectively to the rolling direction; two temperatures, 20°C and 270°C; frequencies between 8 and 0.02 Hz and parent, weld and heat-affected zone material. Test results in Figure 6 show that within this test matrix the Section XI air reference curve remains a good representation of fatigue crack growth properties of the steel in an air environment.

Fatigue cracks that grow in an operating component are usually elliptical embedded cracks or semi-elliptical surface cracks. Surface cracks in SA516 Gr 70 material were grown in fatigue under pure bending loads. Fatigue crack growth rate and shape change were measured. For surface cracks, fatigue crack growth rates at a given stress intensity amplitude appeared to be somewhat lower than that expected from results using compact-tension specimens for the same material under similar conditions (Vanderglas and Mukherjee, 1981).

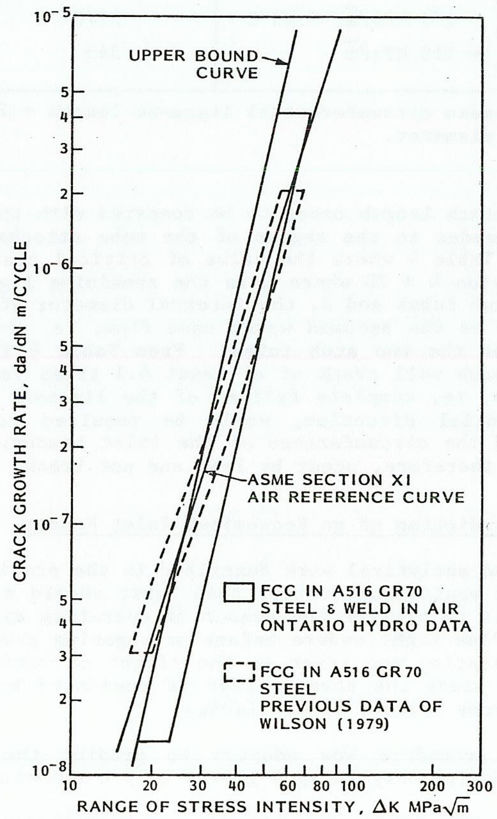

Fig. 6. Fatigue crack propagation rate in air environment.

Low Pressure Turbine Discs

In order to facilitate ultrasonic inspection and refurbishing of several low pressure turbine discs, 12 and 15 mm thick rings were removed from the rim and the bore of these discs. Structural integrity consideration of rotor discs is based on plane strain fracture toughness values (K_{Ic}) which are required to estimate critical crack lengths and useful service lives of rotors in a stress corrosion cracking situation. These discs were manufactured before the use of linear elastic fracture mechanics as an effective fracture control tool was introduced in the turbine manufacturing industry. Therefore, K_{Ic} values were not known for these discs and an experimental program was initiated to develop a method to measure K_{Ic} using miniature specimens. Charpy size specimens were machined from the disc off-cuts, precracked in fatigue and were used to measure a crack initiation toughness J_{Ic} using a resistance curve approach. The J-integral values were determined from the area under the load displacement curve and physical crack extensions were calculated from periodic unloading during a test. The J-integral values versus crack extensions are shown in Figure 7. Table 5 is a summary of test results showing J_{Ic}, fracture toughness at crack initiation. J_{Ic} is converted to K_{Ic} using the Equation (1) (Albrecht and co-workers, 1982).

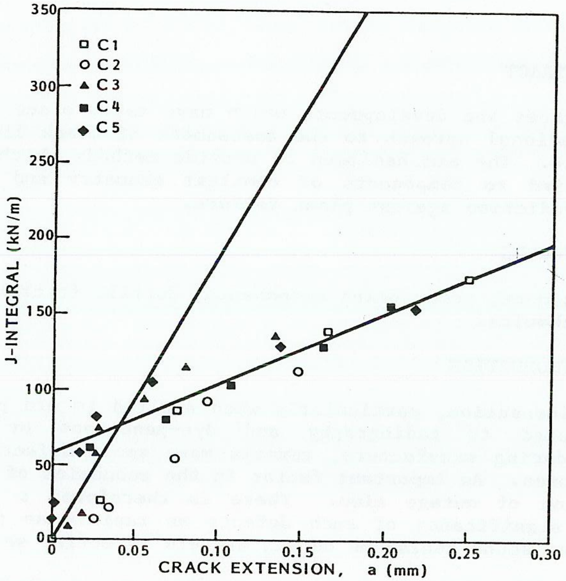

Fig. 7. J-integral versus crack extension.

Fracture toughness values from small specimen tests taken primarily in the post yield regime may be used to evaluate structures operating in a linear elastic regime. However, extreme care must be taken in interpreting these results, since the fracture toughness values may be either conservative or unconservative depending on the fracture mode.

In steels which exhibit a fracture mode transition, the fracture behaviour must be considered in three different ranges, brittle fracture at low temperature, a transition range and a ductile fracture range at higher temperature. In the transition range fracture toughness increases rapidly with increasing temperature and toughness values determined from small specimens may be much higher than those determined from large specimens. In the ductile fracture range, fracture toughness measured at the initiation of the ductile fracture process in small specimens gives a conservative value (Landes, 1979).

STRUCTURAL INTEGRITY EVALUATION - NEXT TEN YEARS

It is anticipated that the need to perform structural integrity evaluations of nuclear pressure-retaining components will continue to grow as further nuclear plants are commissioned in Ontario. In order to meet rigid time constraints for evaluating flaws in nuclear components, all background work (eg. determination of operating stresses, compilation of material data and selection of methods and computer programs for calculating stress intensity factors) must be addressed before inspection of a specific component.

Currently, some utilities are keeping records of all significant pressure and temperature transients experienced by critical nuclear components, so that actual loading history rather than the anticipated design conditions can be used for fatigue evaluation. Bookkeeping of operating transients may well become a requirement for all nuclear stations.

TABLE 5 Summary of Toughness Values for Turbine Rotor Discs*

Disc Number	J_{Ic} (kN/m)	K_{Ic} (MPa/m)	dJ/da (GN/m^2)	UTS (MPa)	0.2% YS (MPa)
A	138	170	0.54	772	620
B	174	189	0.66	903	784
C	78	125	0.47	1014	950
D	154	178	0.67	785	652
E	>154	>178	(Not Applicable)	804	664

* Specification: C 0.40 Max, S 0.018 Max, P 0.015 Max, Si 0.15-0.35, Mn 0.70 Max, Ni 2.0-4.0, Cr 2.0 Max, Mo 0.02-0.70, V 0.005 Min; Yield 586 MPa Min, UTS 758 MPa Min

The use of elastic-plastic fracture mechanics methods for structural integrity evaluation will probably be incorporated in design and in-service inspection codes and guidelines.

Among the new uses found for personal computers has been that of automating laboratory testing. Low cost off-the-shelf systems are being used to control and monitor tests, acquire data and to analyze test results. Personal computers have been used in the fracture mechanics laboratory of Ontario Hydro to monitor long-term stress corrosion cracking tests, to measure and plot load amplitudes during strain controlled low cycle fatigue tests, to conduct single specimen J_{Ic} tests using an unloading compliance method and to measure crack length using a potential drop method. The mass use of inexpensive computers to aid material testing will reduce testing costs and will result in the availability of a large pool ofquality and objective material data.

CONCLUDING REMARKS

This paper was intended to illustrate the use of fracture mechanics methods in the assessment of the structural integrity of power plant components. Several examples of case studies and material related testing were presented. Some other aspects of structural integrity evaluation including quality assurance of a pressure boundary, in-service inspection procedures, estimation of flaw sizes that may be missed during an inspection, probabilistic approach to integrity evaluation and the availability of a suitable in-situ repair procedure are outside the scope of this article.

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REFERENCES

Albrecht, P. and co-workers (1982). JTEVA. Vol. 10., No. 6, Nov 1982, p 245.

Chell, G.G. (1979). In G.G. Chell (Ed). Developments in Fracture Mechanics-1. Applied Science, London, Chap 3, pp 67-105.

Hoff, R., and T.P. Byrne (1982). Ontario Hydro Research Division Report, No 82-491-K.

Landes, J.D. (1979). Proceedings of the 5th Structural Mechanics In Reactor Technology Conference, Vol 6, Paper No G3/4, North-Holland, Amsterdam.

Mukherjee, B., (1980). Proceedings of the 5th Symposium on Engineering Application of Mechanics, Ottawa, N.R.C., pp 35-40.

Mukherjee, B., and M.L. Vanderglas (1980). J. of Pressure Vessel Tech., Vol 102, pp 294-302.

Rice, J.R. (1972), Some Remarks on Elastic Crack Tip Stress Fields. Int. J. Solids and Structures, Vol 8, pp 751-758.

Vanderglas, M.L. (1980), In D.R.J. Owen and A.R. Luxmore (Eds)., Numerical Methods in Fracture Mechanics, Pineridge Press, Swansea, pp 51-65.

Vanderglas, M.L. and B. Mukherjee (1981). Proceedings of the 6th Structural Mechanics in Reactor Technology Conference, Vol L, Paper No L8/4, North-Holland, Amsterdam.

Vanderglas, M.L. (1983). Proceedings of the 7th Structural Mechanics In Reactor Technology Conference, Vol G/F, pp 241-247, Paper No G/F 3/6, North-Holland, Amsterdam.

Wilson, A.D. (1979). J. of Eng. Materials and Tech., Vol 101, pp 265-274.

Zahoor, A., and M.F. Kanninen (1981). J. Eng. Mat and Tech., Vol 103, pp 194-200.