REMAINING LIFE ASSESSMENT OF A HP-ROTOR

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ABSTRACT

The remaining life of a HP-turbine rotor was evaluated considering both bulk creep damage leading to rupture and local damage leading to the development and growth of a creep crack. The turbogenerator and boiler were commissioned in 1966 and had operated for approximately 160,000 hours.

Finite element analysis was used to determine the initial stress distribution in the rotor and to identify the critical regions where the combination of high stress and temperature would cause the maximum creep damage. A subsequent finite element creep analysis provided the stress redistribution as well as the accumulated creep strain caused by the creep process.

KEYWORDS

HP-rotor, creep, remaining life, finite element, creep crack initiation and growth.

INTRODUCTION

The 200 MW steam turbogenerator has a bottle bored monobloc HP-rotor manufactured from 1Cr1Mo¼V creep resistant alloy steel. The turbogenerator has operated for approximately 160,000 hours and is known to have operated at its maximum steam design temperature of 565°C with output exceeding by 10% the maximum continuous rating (MCR) during the first few years after commissioning in 1966. In 1984 the average operating temperature was dropped to approximately 538°C. The aim of this investigation was to determine the remaining life of the HP-rotor considering both bulk creep damage leading to rupture and local damage leading to the development of a crack.

It is generally accepted that non-destructive inspections of identified critical areas should be carried out to substantiate theoretical evaluations and provide the final evidence during the latter stages of life on which to base the run, repair, replace decisions. The interval between inspections is critical as it must be shorter than the period that it could take a creep crack to grow to a critical size and cause the rotor to fail catastrophically. To determine such intervals the critical crack size was calculated and a preliminary estimate of the period of stable creep crack growth was made.

STRESS AND FRACTURE ANALYSIS

A finite element stress analysis of the rotor was performed to:

- locate areas where creep damage is concentrated and inspections are required,
- calculate the initial stresses and the stress redistribution caused by creep,
- calculate the accumulated creep strain,
- calculate the maximum representative stress profile for calculation of the critical crack size.

Elastic Stress Analysis

The model was generated using the graphics package PATRAN (Ref. 1) and analysed with the program ABAQUS (Ref. 2) on a Silicon Graphics Indigo workstation. The purpose of the analysis was to find the stresses at the critical locations around the rotor bore. Details of the blade attachments were therefore not included in this model. The same model was used for the calculation of stresses during a cold start. For the creep simulations the mesh at the location of maximum creep damage was substantially refined, as shown in Figure 1, increasing the size of the model to 2708 second order isoparametric axisymmetric elements.

The loading consisted of centrifugal forces acting directly on the rotor body and pressures, representing the centrifugal forces acting on the blades, applied to the disk rims. Additional stresses were caused by the stationary temperature distribution along the rotor axis.

In the area where maximum creep damage is predicted, ie around the dummy piston, metal temperatures are determined by the steam which is a mixture of the main steam leaking through the glands and the steam bled from the stage 13 area of the turbine. The temperature of this steam cannot be accurately calculated and it is also believed that this temperature varies over time due to the wear of the glands. In the analysis an upper value for this temperature was used and this was subsequently verified using a purposely installed thermocouple.

The highest stress at the rotor bore occurred at the nose of the rotor which creates a stress concentration in the dummy piston. The temperature in this region was found to be about 25°C lower than the temperature of steam entering the turbine. It is the combination of high stress and temperature that makes this location the most susceptible to creep damage. Figure 2 shows the stress distribution in this region. The maximum surface stress was found to be in excess of 225 MPa.

There is a second location of relatively high stress in the groove in the dummy piston outer surface. However, the creep damage at this location is smaller because temperatures are lower. The region between this groove and the bore nose experiences higher stress than the surrounding part of the rotor body. The maximum local stress at the nose was expected to reduce with time due to stress redistribution caused by creep relaxation.

Finite Element Creep Analysis

The creep simulation was performed to find the amount of accumulated creep strain and the redistribution of the initial elastic stress. The constitutive equation describing the creep behaviour used in the simulation was in the form of a power law:

$$\dot{\bar{\varepsilon}} = A\bar{\sigma}^n t^m$$

where $\dot{\bar{\varepsilon}}$ is the equivalent creep strainrate,
$\bar{\sigma}$ is the equivalent stress, MPa,
t is the time, hours,
A, n, m are temperature dependent material constants.

This law is valid for the primary and secondary stages of creep. It cannot represent the increase in the creep strain rate that typifies tertiary creep.

The creep simulation was carried out using the measured temperature history. The simulation was performed for the past life of the rotor and continued until the accumulation of 250,000 hours, assuming the current nominal main steam temperature of 538°C.

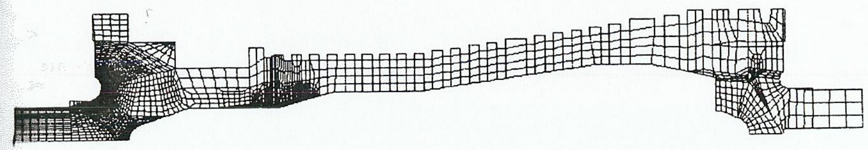

Figure 1: Finite element mesh for the 200 MW HP-rotor.

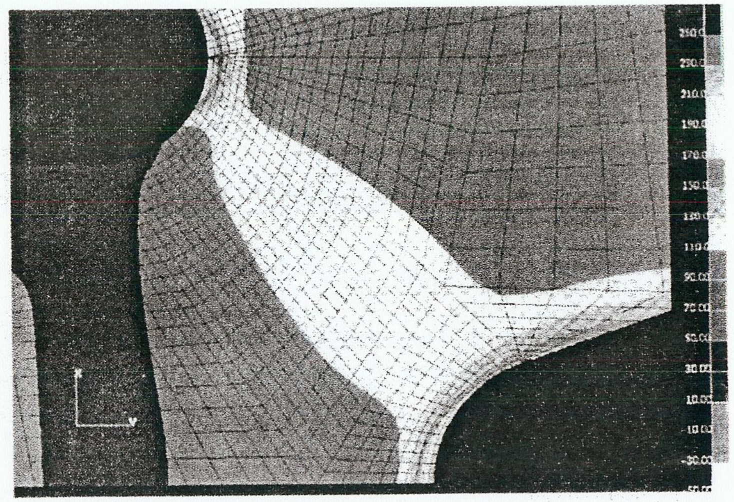

Figure 2: Finite element analysis showing the distribution of steady state stresses at the rotor bore nose and at the stress relief groove.

The redistribution of the local stress at the nose was smaller than expected. As shown in Figure 3, the surface stress relaxes to only approximately 75% of the initial value. The consequence is that the high creep damage at the nose is very localised and the most probable mode of incipient failure is the initiation of cracking at this location.

Critical Crack Size

The integrity assessment of the rotor shaft is based on the assumption that a crack may initiate in the vicinity of the highest stress concentration as a result of creep deformation processes. This crack would then grow under slow stable creep crack growth conditions to a critical size at which point the crack would extend rapidly resulting in failure of the rotor shaft.

The critical crack size at the nose is determined by the most adverse combination of low temperature and high tensile stress, which occurs during a cold start. The steam entering the turbine during a start causes compressive stress at the rotor surface and a balancing tensile stress at the bore. This stress is superimposed on the tensile stress from the centrifugal forces.

The CEGB program "Fracture Zero" (Ref. 3) was used to determine the critical crack sizes in the HP-rotor shaft. Semi-elliptical and extended surface defects were assumed in the analysis. The results of the analysis are given in Figure 4. For a semi-elliptical surface defect the critical crack depth is 8-10 mm for a length of 22 to 15 mm.

HP-ROTOR REMAINING LIFE CALCULATIONS

Three possible modes of rotor failure by a creep mechanism were assessed:

- bulk creep damage leading to rupture,
- accumulation of excessive localised creep strain,
- development and propagation of a creep crack.

Analysis of Bulk Creep Damage

The analysis of bulk creep damage leading to rupture required material properties, obtained from accelerated creep rupture tests, and the temperature history of the unit.

KWU design data and experimental data from ERA and English Electric for 1Cr1Mo¼V rotor steels were found in the literature and are reproduced in Figure 5. Test results from British rotor steels are for material from a similar era to the HP-rotor under assessment. While there is some scatter in these results, the data appears to fall about the KWU mean design line, thus confirming the general validity of the KWU data.

An accelerated creep-rupture program undertaken at HRL provided a good indication of the actual creep-rupture properties of the HP rotor steel. Material has been sampled from the vicinity of the stress relief groove at the dummy piston. The results of these tests are shown in Figure 5. Results for the present HP-rotor exhibit considerable scatter but generally fall around the KWU mean line. Consequently the KWU design data were used to assess life in this case where a creep-rupture failure criterion was assumed.

A temperature history based on the secondary superheater outlet header temperatures was adjusted downwards to reflect the temperature loss as the steam travels from the secondary superheater outlet to the turbine inlet.

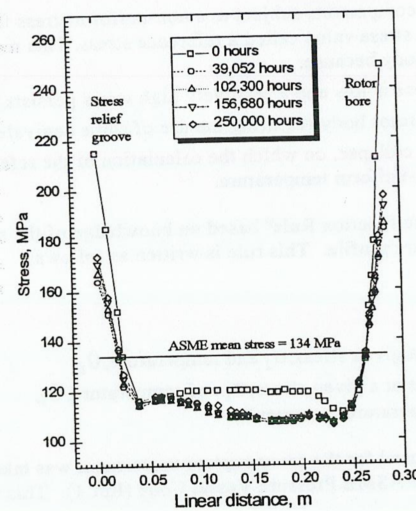

Figure 3: Distribution of stress between the rotor bore nose and the stress relief groove on the dummy piston. Note that the relaxation of the stress occurs in the first 40,000 hours of operation.

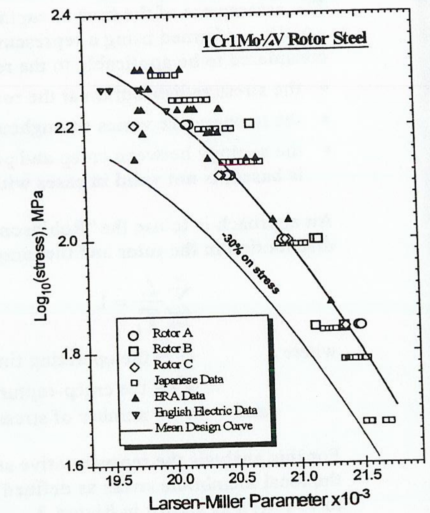

Figure 5: Experimental creep-rupture results for three rotors, together with data from the literature.

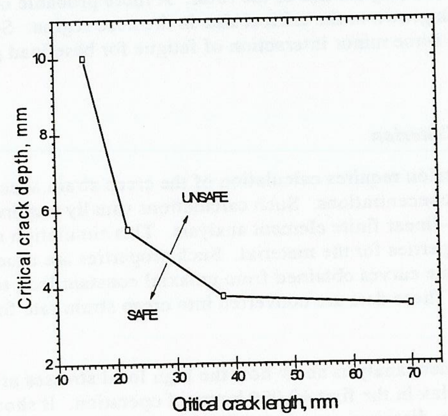

Figure 4: Critical crack size

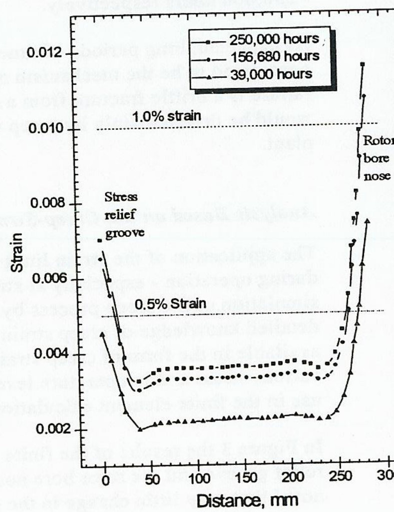

Figure 6: Strain distribution along the line of maximum stress.

The assessment of the time to rupture in components subject to a non-uniform stress field is usually performed using a representative stress value called a reference stress. This method is not considered to be applicable to the rotor body because:

- the stress redistribution at the rotor nose is low and a relatively high stress persists there,
- the temperature varies throughout the rotor body requiring the use of some equivalent value,
- the analogy between creep and plastic collapse, on which the calculation of the reference stress is based, is not valid in cases with non-uniform temperature.

An approach is to use the "Robinson's Life Fraction Rule" based on knowledge of the stress distribution in the rotor and the temperature profile. This rule is written as follows;

$$\sum_{i=1}^{n} \frac{t_i}{T_i} = 1$$

where t_i = the operating time at a given stress, σ_i and temperature, θ_i,
T_i = the creep-rupture time at a given stress, σ_i and temperature, θ_i,
and n = number of stress/temperature increments.

For this analysis the representative stress used for the creep rupture assessment was taken to be the local membrane stress as defined in the ASME Pressure Vessel Code (Ref 4). This was equal to 134 MPa as shown in Figure 3.

The creep-rupture time was calculated using the KWU mean design data as shown in Figure 5. The creep life fraction consumed to date for the HP-rotor assuming a representative stress of 134 MPa, was 0.29. The remaining life of the HP-rotor, assuming the upper bound future metal temperature of 530°C, and a lower bound metal temperature of 520°C, was 210,000 hours and >250,000 hours respectively.

As expected, long periods of time to rupture were calculated. However, bulk creep is not considered to be the mechanism most limiting the life of the rotor. A more probable mode of failure is a brittle fracture from a crack grown to the critical size in the nose region. Such a crack would be driven mainly by creep with some minor interaction of fatigue for base load generating plant.

Analysis Based on the Creep-Strain Criterion

The application of the strain limit criterion requires calculation of the creep strain accumulated during operation - especially at stress concentrations. Such calculations usually require the simulation of the creep process by non-linear finite element analysis. This simulation requires a detailed knowledge of creep strain properties for the material. Such properties are usually available in the form of creep strain-time curves obtained from uniaxial constant load tests at various stress and temperature levels. These data are converted into creep strain rate formulas for use in the finite element calculations.

In Figure 3 the results of the finite element analysis show how the high local stresses at the stress relief groove and the rotor bore nose relax in the first 40,000 hours of operation. It should be noted that very little change in the stress distribution occurs after this period of time as the rotor reaches a "steady condition". Reference to Figure 6 shows the strain distribution between the rotor bore nose and the stress relief groove. Like the stresses in Figure 3, the strains are higher at the highly stressed surface areas. The calculated accumulated equivalent creep strain at the nose reached the value of 1.08%, after the current operating time of 157,000 hours. The CEGB recommendation is that local peak strain should not exceed 1% (Ref 5).

The creep simulation was performed using the actual temperature history represented by temperature records sampled at approximately 2 week intervals throughout the rotor's operational life. The simulation was repeated using the nominal temperature distribution. The difference between the calculated value of creep strain accumulated to date, based on the actual temperature history and the nominal temperature distribution, was 0.03%.

It was assumed that the required life of the HP-rotor would be up to 250,000hours, and so this time period was simulated in the analysis. The strain at the rotor bore nose after 250,000 hours of simulated operation was shown to increase to 1.16%, whilst the strain at the stress relief groove was approximately 0.7%. The results show that the future increase of the local creep strain between 157,000 and 250,000 hours is very slow, due to the reduced operating temperature and the small degree of stress relaxation and redistribution.

Analysis of Creep Crack Initiation and Propagation

Under conditions of uniform loading in the creep regime, general rupture is caused by the development and spread of damage, in the form of cavities and microcracks, throughout the creeping material. Components that operate at elevated temperatures can also fail by the propagation of a single macroscopic crack initiating from a pre-existing flaw, weld defect or site of stress concentration. Such crack growth at elevated temperatures is termed creep crack growth. A knowledge of the creep crack growth rate can enable the non-destructive inspection interval to be established so that crack growth between inspections would not produced defects of a critical size.

The calculation of the period to creep crack initiation can be based on the Local Stress History Method (Ref. 6). For the HP-rotor the stress at the nose remains at a relatively high level, and this method indicated that creep life at this location had been exhausted. For practical purposes it should therefore be assumed that the theoretical life to crack initiation has been exhausted, and hence that the remaining life of the rotor depends on the time for a crack to grow to the critical crack length. Crack length is then measured using non-destructive techniques at regular inspections, and it is essential that the inspection interval should be based on the calculated period for a crack to grow to the critical size by creep (and high strain fatigue).

While an exhaustive study of the literature has not been undertaken, some creep crack growth data were found that related to 1Cr1Mo¼V rotor steels at 540°C. The data can be expressed in the form:

$$\frac{da}{dt} = A.C^*$$

where da/dt is the creep crack growth rate, mm/h
A is a material constant, and
C^* is the creep crack parameter.

The C^* parameter can be estimated from the following approximate formula:

$$C^*_{Ref} = \sigma_{Ref}\,\varepsilon_{Ref}\left(\frac{K}{\sigma_{Ref}}\right)^2$$

Where C^*_{Ref} is the C^* parameter related to the reference stress,
σ_{Ref} is the reference stress, MPa,
ε_{Ref} is the creep strain rate corresponding to σ_{Ref}, and
K is the stress intensity factory.

As explained above, the reference stress defined in the R5 Procedure (Ref.7) is not applicable to the rotor. Instead, an averaged stress value should be used in the above formula. The local membrane stress of 134 MPa was used for this purpose.

Based on the above equations and the range in the values of the constants used, the growth of such cracks over a year would be between 1.3 and 6.6 mm/year. On this basis, for a critical crack size of 8-10 mm, the re-inspection period would range from 1 to 7 years, depending on the creep crack growth rate data that is used.

It should be emphasised that this estimate of the crack growth rate is simplified as a detailed analysis of this damage mechanism was not included in the original scope of this investigation. The accuracy can be increased by calculating the value of C^* from the finite element model rather than from the approximate formula . The fatigue crack growth component, which has been assumed to be small for this base load unit, could also be calculated.

CONCLUSIONS

The results of this assessment of the remaining life of the HP-rotor indicate that there is no danger of bulk creep damage leading to general rupture in the hot part of the rotor within the desired total operating time of 250,000 hours. However, a localised creep crack could initiate and grow from the nose of the bore as the accumulated creep strain is slightly in excess of the conservatively acceptable value of 1%. Therefore continuing operation of the rotor should be based on results from non-destructive inspections of the critical areas carried out to detect cracks. The intervals between inspections should be based on an accurate assessment of the crack growth rate to ensure that the crack does not reach the critical crack size between inspections.

This work confirmed certain limitations of the current life assessment techniques. The finite element creep simulations demonstrated that the reference stress is not always applicable. The reference stress concept assumes a redistribution of stress caused by creep. At the bore nose the relatively high local stress, due to three dimensional stress state effects, was only partially redistributed rendering the reference stress invalid.

There is also a need for a reliable method for calculation of creep crack initiation. The local stress method seems too simplistic and lacks a proper mechanistic basis. The method proposed in the R5 Procedure is not fully validated and suffers from a shortage of pertinent material data.

The approximate formula for the estimation of C^* has also only limited application, as it requires a knowledge of the reference stress. In cases where the reference stress is not valid, it is not certain what representative stress should be applied. The crack growth analysis should therefore be based on the value of C^* derived directly from the finite element model.

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