APPLYING THE PROBABILISTIC FITNESS-FOR-SERVICE CONCEPT FOR ASSESSING RELIABILITY OF WELD REPAIRED STEAM TURBINE ROTORS

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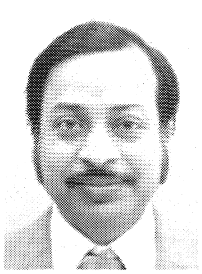

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ABSTRACT

The cost savings and fast turnaround offered by weld repair of turbomachinery rotors results in significant advantages to the end user when faced with the possibility of having to replace the rotor. However, it also carries a potential risk to reliability. Many rotors have been weld repaired and have been put back in service over the years. The reliability of the repairs have not been quantified, but have usually been proven only through the lack of failure. No standard way of decision making has developed because it is an ever evolving technology. The principal driver for repair is economics, and the technical justification for such repair is "fitness-for-service."

A method to quantify the risk for weld repairing a rotor *a priori* is presented. This couples material properties characterization, fracture mechanics concept, and probabilistic type calculation to determine reliability of the repair. This extends the fitness-for-purpose concept and captures the randomness and uncertainties in controlling variables; thus, it is called the "probabilistic fitness-for-service" concept.

INTRODUCTION

The technical basis of reliability evaluation, the results and implementation of a study undertaken to develop a method to provide for the quantitative assessment of rotor weldment reliability are presented. Also discussed are two specific test cases of reliability study and weld repair performed on steam turbine rotors. The study concluded that the use of probabilistic fracture mechanics as analytical technique, coupled with empirically measured nondestructive inspection detectability limits, would serve as a methodology for assessing the "fitness for purpose" as defined by Wells [1] of any rotor weld repair. This approach simultaneously provides a realistic, yet conservative, prediction for safe operation. Fracture mechanics was selected as the analytical technique because classical S/N type fatigue data are believed to be unrealistic and possibly anticonservative for weld repairs, which are inherently prone to generation of flaws. S/N type of fatigue data is a strong function of flaw size, frequency of its occurrence, and its distribution. It is common to observe large scatter in fatigue properties of weld specimens. Furthermore, the fatigue properties of a weldment are a function of joint configuration and loading conditions as described by Walker [2]. A development project was initiated in 1990 to develop gas tungsten arc weld processes specifically for ASTM A470 Class 4 and Class 8 rotor alloys. The results of the development effort have been described by Tipton [3]. The methodology used to evaluate the reliability of any weld repair requires detectability limit information for the nondestructive method utilized, and fracture mechanics type fatigue threshold data for weld filler metal, heat affected zone, and base metals for the rotor alloy of interest. The reliability analysis as described by Singh [4] is basically composed of two tiers, the first of which determines the probability of a crack equal to the detectability limit of the ultrasonic NDT method to grow under operating conditions. If crack growth is not predicted, no further calculations are performed. However, if crack growth is expected, then the probability of the number of cycles elapsed to reach critical crack size is calculated. A computer program has been developed to perform reliability type calculation utilizing linear elastic fracture mechanics. Description of the underlying theory will be discussed. Results of studies performed for two steam turbine rotors are included to demonstrate the applicability of the method. These show a need to quantify the reliability or risk so that a logical decision could be made and/or a possible design change could be instituted to achieve an acceptable reliability.

THEORETICAL BASIS OF RELIABILITY ANALYSIS

The following reasonings have been considered in the theoretical basis for evaluating the reliability of weld repaired rotor:

- Occurrence of minute crack-like flaws is inherent in welding processes.
- Weakness or strength of weldment is the result of existence or absence of flaws.
- Flaws occur in various sizes and orientations.
- Evaluation of such a structure can be done by utilizing fracture mechanics.
- There will be uncertainty in the precise characterization of size and orientation of flaws.
- Measured material properties exhibit variation from specimen to specimen. These properties can be described by a statistical distribution.

The following fracture mechanics equations are used to describe the behavior of a structure containing a flaw.

$$K = Y\sigma\sqrt{a} \tag{1}$$

$$\Delta K = Y\Delta\sigma\sqrt{a} \tag{2}$$

$$\frac{da}{dn} = C(\Delta K)^n \tag{3}$$

where
K = stress intensity factor, ksi $\sqrt{in}$
ΔK = stress intensity factor range, ksi $\sqrt{in}$
Y = factor dependent on geometry
σ = stress, ksi
$\Delta\sigma$ = stress range, ksi
a = crack length, in
$\frac{da}{dn}$ = crack growth rate, in/cycle
C,n = material constants

Equation (3) can be integrated to yield the number of cycles which can be achieved.

$$N_f = \sum_{a_i}^{a_j} da + \int_{a_j}^{a_f} \frac{da}{CY^n\Delta\sigma^n a^{n/2}} \tag{4}$$

where
a_i = initial crack length
a_f = final crack length
da = incremental crack growth

Fracture toughness (K_{IC}) and threshold (ΔK_{th}) are two limit values of fracture properties of a material. K_{IC} is compared to the calculated value of K (Equation (1)) to determine how rapidly the crack length will grow in a half cycle application of stress (σ). This crack length is defined as critical crack length (a_c). If the magnitude of ΔK as determined from Equation (2) is less than ΔK_{th}, the existing crack of length, a, will not grow. However, if the magnitude of ΔK exceeds the value of ΔK_{th}, growth of the crack is predicted. Equation (4) is then used to calculate the number of cycles before rapid growth or when the crack length equal a_c.

Existence of crack-like flaws in weldment and applicability of fracture mechanics for analysis can easily be understood, but the uncertainty in flaw size and variation in material properties pose some difficulties. It is well understood that response of a structure under influence of uncertain input parameters will be random. Thus, a deterministic type of analysis does not provide a realistic prediction. A probabilistic type of calculation provides meaningful information which could be used to decide on a particular repair. The question is not that there will be a variability in the input parameters, but what is the influence of such variation on the performance of the repaired rotor.

Assuming that ΔS is the imposed stress range, and a is the existing crack length then from Equation (2)

$$\Delta K = Y\Delta S\sqrt{a} \tag{5}$$

The existing crack under the applied ΔS will not grow if

$$\Delta K \leq \Delta K_{th} \tag{6}$$

The exact magnitude of ΔK is not known because stress range and crack length both have a statistical distribution. Also, the magnitude of ΔK_{th} is defined by a statistical distribution. At this point, it should be asked what the chances are or what the probability is that calculated ΔK is less than or equal to ΔK_{th}? Mathematically, it is expressed as follows:

$$P_s = P(\Delta K \leq \Delta K_{th}) \tag{7}$$

$$P = P(G(\Delta s, a, \Delta K_{th}) \leq 0) \tag{8}$$

where G is a "limit state function" and depends on stress range, crack length, and ΔK_{th}. Magnitude of p_s is the probability of the crack not growing.

If it is found that there is a probability for the existing crack to grow, an equation can be written to predict probability of attaining a given number of stress cycles before rapid growth of the crack.

$$P_s = P(N \geq N_f) \tag{9}$$

where N_f is defined by equation (4)

Closed form solutions of Equation (8) and Equation (9) are difficult to obtain. A numerical scheme to evaluate the probabilities has to be employed.

First and second order reliability methods (FORM/SORM) are used in risk assessment of mechanical structures and can be used in assessing reliability of rotors. A detailed description and explanation of reliability theory is beyond the scope of this presentation. The purpose herein is to describe the application rather than the theory. Theory of reliability, its background, and description of FORM/SORM have been given in the literature [5].

JOURNAL BEARING REPAIR

Background

An inquiry was received to build up the coupling end journal area of a rotor by welding. The unit consists of seven stages and was commissioned in the 1950s to develop 7,460 hp at a rated speed of 7,100 rpm; the machine was rebuilt in 1973 to produce 8,500 hp at a rated speed of 7,100 rpm. The rotor was fabricated in 1954 from ASTM A293 Class 6. The rotor had severe damage to the coupling end journal area (scoring and rehardened material): the cause of the damage was reported to be loss of oil to the bearing. Hardness of the rehardened area was measured to be HRC 43-56 with a portable hardness tester.

Metallurgical Evaluation

A complete baseline inspection was performed on the entire rotor, consisting of visual, dimensional, magnetic particle, ultrasonic, and hardness testing. The inspection revealed that in addition to the high hardness in the exhaust end journal area, the rotor was also grooved to a depth of approximately 0.050 in, and the shaft end was bowed, resulting in a 0.003 in plus runout. A quantitative chemical analysis was performed on drillings from the subject rotor, and the results are presented in Table 1. The composition of the rotor conformed to current ASTM A470 Class 4 requirements, and no changes to the inplace process parameters were deemed necessary. The degree of distortion which may occur as a result of welding could not be determined *a priori*, and it was proposed to the customer that if the degree of distortion as a result of welding was acceptable, the rotor would be straightened by "spot heating." A schematic of the proposed weld repair buildup is shown in Figure 1. The actual weldment was drastically different from that shown in Figure 1, because of distortion resulting from the relieving of residual stresses in the rehardened zone of the journal area. It was recommended that the rotor be welded per the established process specifications with no modifications. It was also required that the rehardened zone be turned down further or stress relieved prior to welding so that the area have a maximum hardness of HRC 35. The subject area would be preheated to 350°F, and the interpass temperature would be limited to 550°F. Subsequent to welding, the weldment would be stress relieved at 1025°F, plus or minus 25°F for two hours per inch of shaft diameter. The appropriate fatigue data, shown in Figures 2 and 3, was used for the analytical evaluation.

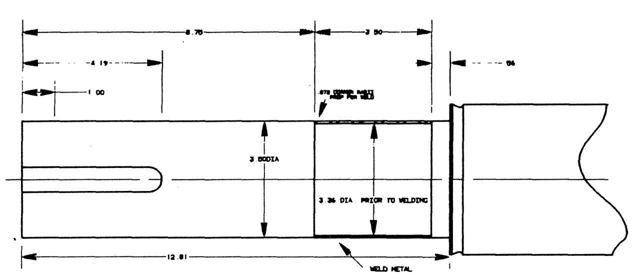

Figure 1. Schematic Representation of Original Rotor 1 Repair Weldment.

Table 1. Chemical Composition of Rotor 1.

Sample	C	P	S	Ni	Cr	Mo	V	Si	Mn
Rotor	0.23	0.01	0.01	3	0.34	0.49	0.05	0.18	0.47
ASTMA293 Class 6 Spec	0.40 max	0.04 max	0.04 max	2.50 max	0.5 max	0.45–0.65	0.03 min	0.15–0.30	0.9 max
ASTMA470 Class 4 Spec	0.28 max	0.012 max	0.015 max	2.50 min	0.75 max	0.25 min	0.03 min	0.15–0.30	0.20–0.60

Reliability Analyses

Reliability calculations were performed for the proposed weld repair geometry, as shown in Figure 1. Linear elastic fracture mechanics (LEFM) formula were used. The location of possible discontinuities arising due to welding was assumed to be on the surface and oriented in the radial direction, thus providing the worst case. Two types of loads were considered, both for crack initiation and crack growth. The first type of loading was the maximum torque load imposed on the rotor during startup and shutdown cycle. The second type was alternating bending loads due to misalignment of the coupling between turbine and compressor rotors. Other variable loads typical to the application were covered by the use of a "service factor." The magnitude of alternating bending load depends on the coupling type and allowable misalignment, and was calculated based on the method provided by Singh and Ramsey [6].

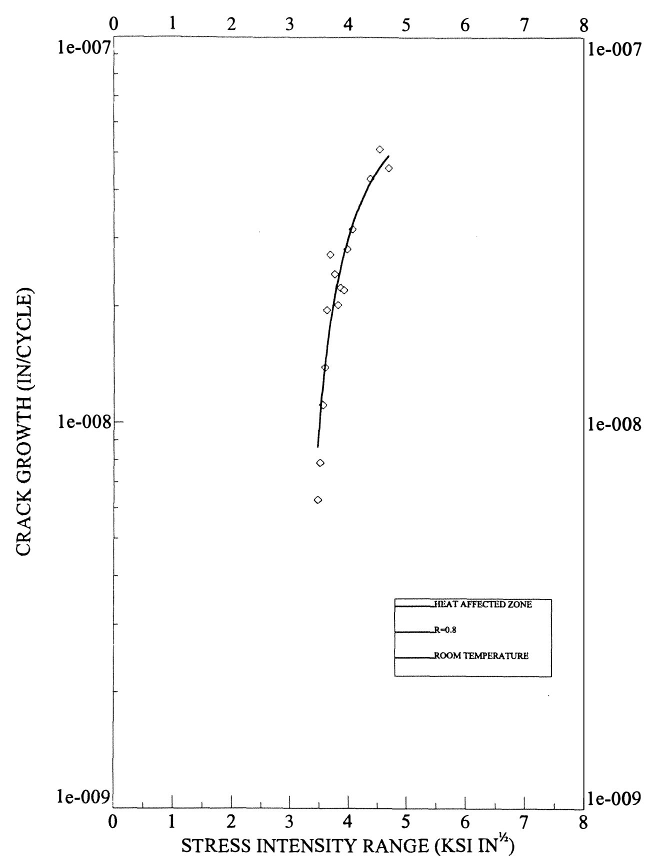

Figure 2. Near Threshold Crack Growth Curve for ASTM A470 Class 4 Heat Affected Zone Specimens.

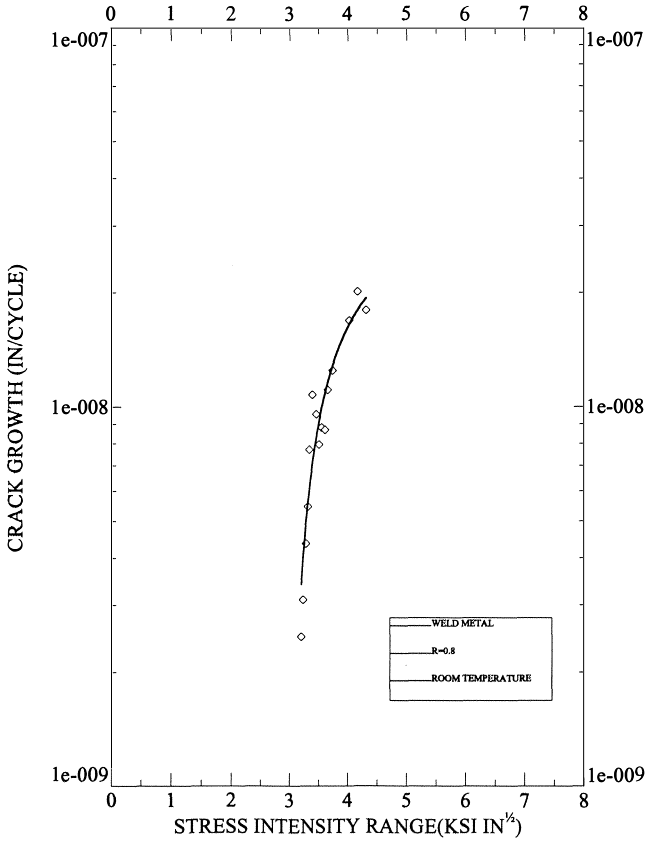

Figure 3. Near Threshold Crack Growth Curve for Proprietary Weld Metal.

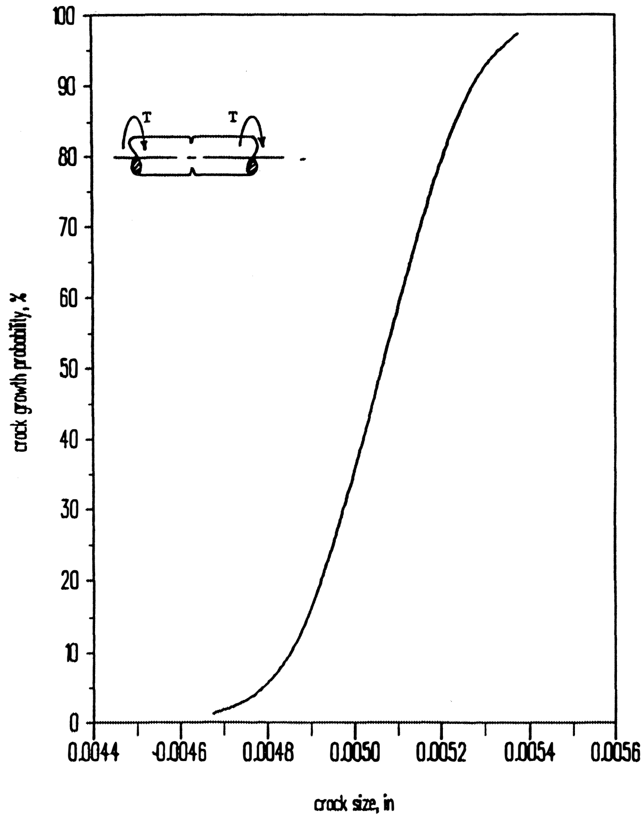

Figure 4. Probability of Crack Growth vs *Crack Depth for Startup Cycle for Rotor 1.*

A normal design check was also performed to take advantage of improved design criteria which have evolved in many years of experience and utilize new technology. Stresses were calculated based on the existing gear type coupling. The diameter of the rotor coupling was 3.5 in and transmitted a torque of 131,900 in-lbs. The alternating bending moment at the maximum misalignment was estimated to be approximately 30,700 in-lbs. The factor of safety based on von Mises criteria [5] was calculated to be about 1.52, which was less than the desired value of 2.0. Therefore, it was recommended that the coupling should be replaced by a diaphragm type to reduce the alternating bending stresses.

The nominal design check was repeated for diaphragm type coupling. The new value of alternating bending moment for the maximum allowable misalignment was estimated to be about 2,220 in-lbs. The factor of safety for this configuration was calculated to be above 3.0.

It was decided to replace the gear type coupling with a diaphragm type coupling. The list of parameters and its numerical values which were used in reliability assessment are provided in Table 2. A service factor of 1.1 was assumed, and the resulting torsional stress was about 2,850.0 psi. First, the probabilities of crack growth for different starter crack sizes were calculated for startup cycle. Crack depth (ranging from 0.0046 in to 0.0056 in) has a variable degree of probability for growth, shown in the results in Table 2. For example, a crack less than 0.0046 in has almost zero probability for growth, and a crack more than 0.0056 in in depth is sure to grow. The next question is the safe number of startup and shutdown cycles. The probability of the safe number of cycles when the starter crack is assumed to be equal to NDE limit of 0.0625 in is shown in Figure 5. Similar calculations were performed for the effects of imposed alternating bending moment due to coupling misalignment. Based on the results, it was determined that even a crack larger than the weld thickness had almost zero probability for growth.

Actual Weld Repair

The damaged bearing journal was turned down an additional 0.090 in and the hardness rechecked. The hardness was still too

Table 2. Description of Input Statistics of Rotor 1.

Name of Variable	Mean	Standard Deviation	Distribution Type
Shaft Radius at Base of Crack, in	Varies	Varies	Normal
Outer Shaft Radius, in	1.75	0.313E-4	Normal
Stress Range, psi			
Torsion	15686	196	Normal
Bending	526	6.28	Normal
ΔK_{th}, psi in$^{1/2}$			
ΔK_{IIIth}	1985	24.8	Lognormal
ΔK_{Ith}	3440	435	Lognormal
K_C, psi in$^{1/2}$			
K_{IIIC}	18000	225	Lognormal
K_{IC}	30000	375	Lognormal
C	0.182E-16		Constant
n	2.62		Constant

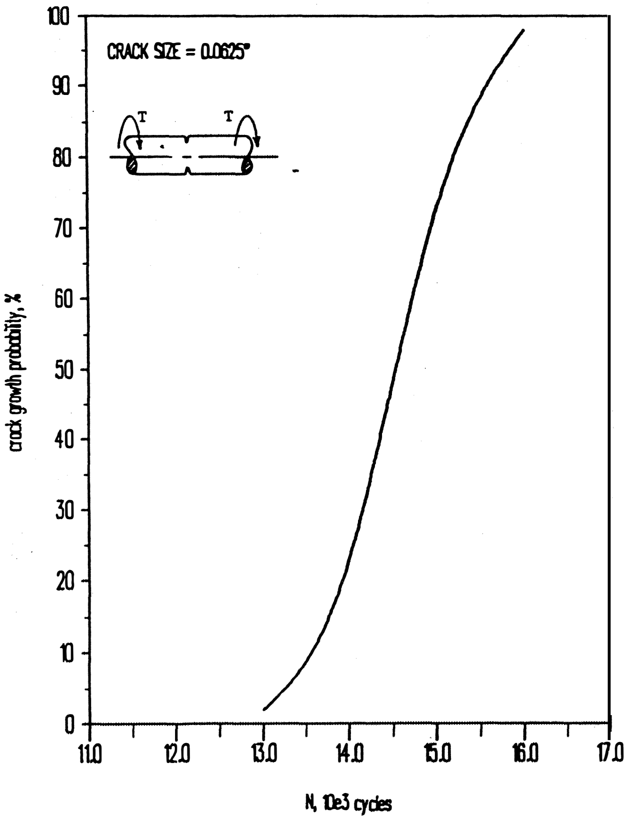

Figure 5. Probability of Unstable Crack Growth vs *Number of Startups for Rotor 1.*

high to attempt welding. Another cut was made of 0.035 in, and the hardness was again checked. Areas of extreme hardness were still being found. A preweld stress-relieve process was chosen to reduce the unacceptable hardness levels. Two cycles of stress relieving were performed before acceptable hardness was achieved. The first cycle increased the shaft end runout to approximately 0.007 in, and the second returned the runout to 0.0027 in. It was decided to weld the rotor from the journal area out to the end of the shaft to allow for truing up after welding instead of "spot heating." The rotor was now prepared for welding both the undersized journal area and to correct the out-of-tolerance runout. The shaft end was undersized to completely remove the coupling keyway, and liberal chamfers were placed at each area of shaft diameter reduction, as shown in Figure 6. Welding was performed with continuous monitoring of the voltage, weld amperage, weld travel speed, filler wire speed, longitudinal travel, and the interpass temperature; all parameters were controlled with the exception of the interpass temperature. Upon completion of the repair weld, the rotor was removed from and set in the post weld heat treatment stand. The rotor received localized stress relieving per original specification requirements. When stress relieving was complete, the welded area was turned to an overprint dimension for post weld NDT. This inspection was identical to the baseline inspection taken before welding. Once the inspection was complete, the rotor was finished to the original manufacturing print dimensions, including recutting the keyway.

INTEGRAL COUPLING HUB REPAIR

Background

An inquiry was received from a major U.S.A. chemical company to build up an integral coupling hub on a rotor. The subject

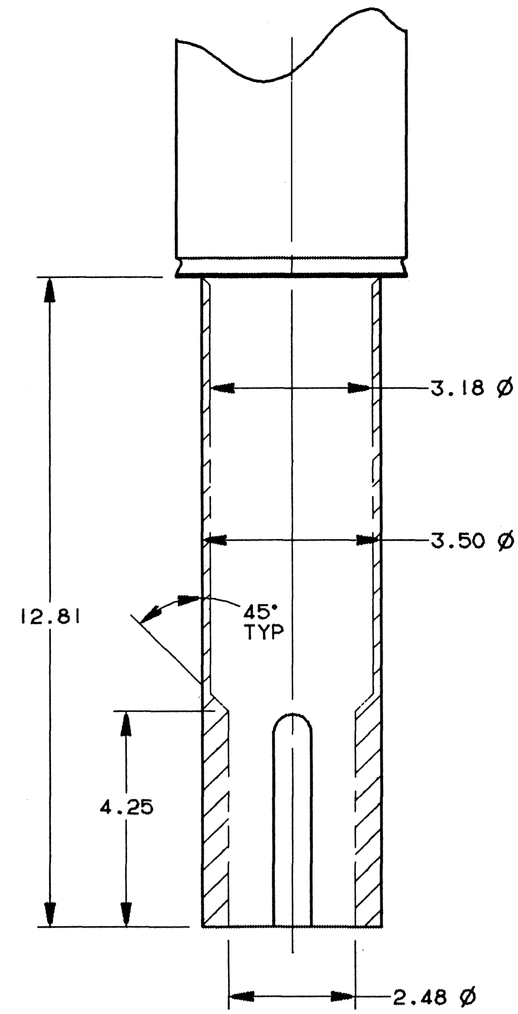

Figure 6. Schematic of Actual Rotor 1 Repair Weldment.

turbine was put into operation in June 1967. The specific turbine operating conditions and other pertinent data are given in Table 3. The subject rotor had experienced cracking at the back of the keyway after approximately four years of service and was subsequently undercut in the area and shortened. The current configuration of the shaft end are presented in Figure 7.

Metallurgical Evaluation

Quantitative chemical analysis was performed on drillings from the subject rotor, and the results are presented in Table 4. The composition was found to conform to the current requirements of

Table 3. Turbine Operating Characteristics of Rotor 2.

Item	Original	Current
Steam Conditions	450 psig, 650°F	490 psig, 700°F
Power	17,785 HP	22,150 HP
Speed	4370 RPM	4590 RPM
Shaft Size at Coupling End	5.5 Inches Diameter	Rotor Undercut and Shortened ('76)
Coupling Type	Gear	Dry
Driven Machine	Compressor	Same

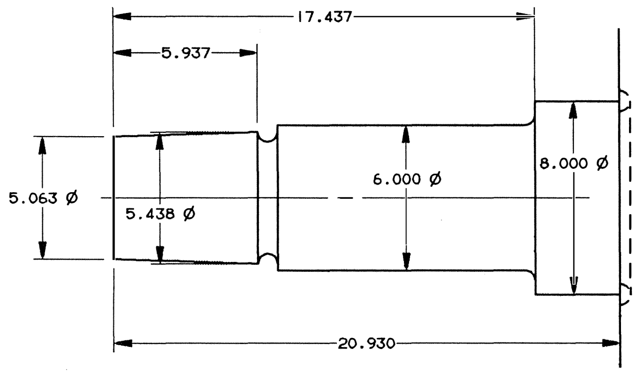

Figure 7. Schematic of Current Evaluated Rotor 2.

ASTM A470 Class 4; the original specification would have allowed higher levels of carbon, phosphorous, and sulfur. The welding would be performed per established process specifications with preheat, interpass temperature, and post weld stress-relief treatments as previously mentioned. A schematic of the proposed weld repair is shown in Figure 8 and consists of seven steps as follows:

- The rotor would be turned down to approximately 4.60 in from the coupling end to just rearward of the undercut area; beveled surfaces will be provided to step up to the next diameter.
- The 6.0 in diameter section would be cleaned up with no significant material removal.
- The 4.60 in diameter will be built up in excess of 6.0 in and machined to 6.0 in.
- The entire length, now at 6.0 in, will be built up in excess of 6.5 in in diameter.
- The two integral slingers would be built up locally in excess of their finished dimensions.
- The first 4.75 in of shaft length, which would be at a diameter of slightly more than 6.5 in, will be built up in excess of 7.875 in in diameter.
- The integral hub would be built up last to a diameter in excess of 14.5625 in.

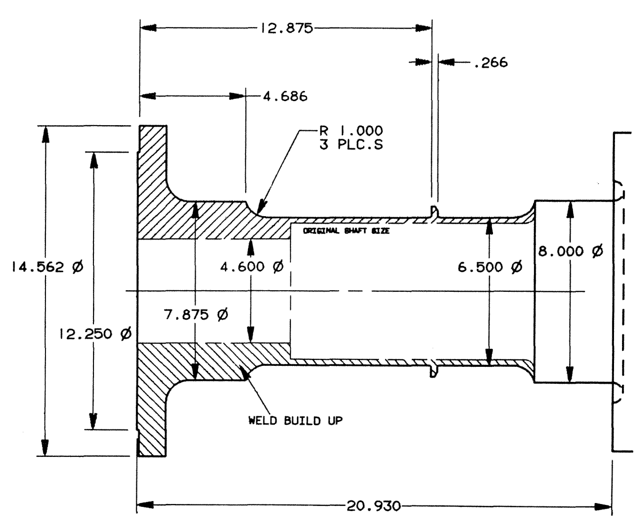

Figure 8. Schematic of Proposed Rotor 2 Weld Repair.

Table 4. Chemical Composition of Rotor 2.

Sample	C	P	S	Ni	Cr	Mo	V	Si	Mn
Rotor	0.28	0.01	0.01	2.94	0.12	0.5	0.06	0.18	0.58
ASTMA293 Class 4 Spec	0.35 max	0.035 max	0.035 max	2.50 min	0.75 max	0.25 min	0.03–0.12	0.15–0.30	0.7 max
ASTMA470 Class 4 Spec	0.28 max	0.012 max	0.015 max	2.50 min	0.75 max	0.25 min	0.03 min	0.15–0.30	0.20–0.60

The weldment would be nondestructively inspected at various stages. The appropriate fatigue properties for analyzing the reliability of the proposed weld repair were the same as shown in Figures 2 and 3.

Reliability Analysis

As mentioned earlier, both the original and the spare rotor developed cracks after operating for a number of years. The rotor was shortened, and the original gear type coupling had been replaced by a dry type coupling. Since the manufacture of this turbine, a new design criteria had evolved. It was decided that before a reliability type calculation is performed, the design should be checked using current design tools and methods. A shaft size of 5.50 in in diameter at the coupling end was calculated to be acceptable when a dry type coupling is used with a hydraulic fit hub. However, the existing coupling had slipped in operation. It was determined that it was due to sudden change in torque load. It was decided to treat the effect of this load at the coupling shaft juncture as an impact load. After a simplified calculation it was decided to use 300 percent of the original nominal torque load as the basis of weldment design and reliability calculation. Hence, this value of load was used in further study as the nominal load. It seems prudent in such a situation to provide the shaft end with an integral hub construction and to size the coupling end to withstand occasional overload. Reliability assessment was done following the method, as described for the previous case.

After nominal design checks, it was decided that the shaft should be made larger to increase the reliability of repair. A shaft size of 6.5 in was recommended to be within the physical limitations of bearing housing. Results of reliability analysis on 6.5 in rotor are presented in Figures 9 and 10. The magnitude of reliability for startup and shutdown cycle for varying sizes of crack assumed to exist on the rotor surface are shown in Figure 9. The crack size of about 0.0039 in has about zero probability for growth. However, a crack in excess of 0.0049 in will grow. The next set of results, as shown in Figure 10, for a crack equal to the NDE limit, show that at least 3.0×10^5 cycles are attainable. The maximum cycle which one should expect is about 4.4×10^5 cycles.

SUMMARY

Results of the two case studies presented here should be viewed not simply as a weld repair of the rotors but as a redesign. In the first example, the type of coupling was changed to increase the reliability. In the second example, the shaft size was increased to provide more reliability utilizing the experience gained in operation, and utilizing the new technology which has evolved since the manufacture of the rotor.

The "probabilistic fitness-for-service" method adds the influence of uncertainties to the deterministic type "fitness-for-service" concept.

The availability of quantitative information regarding the anticipated reliability of a candidate rotor provides a logical input to the decision making process.

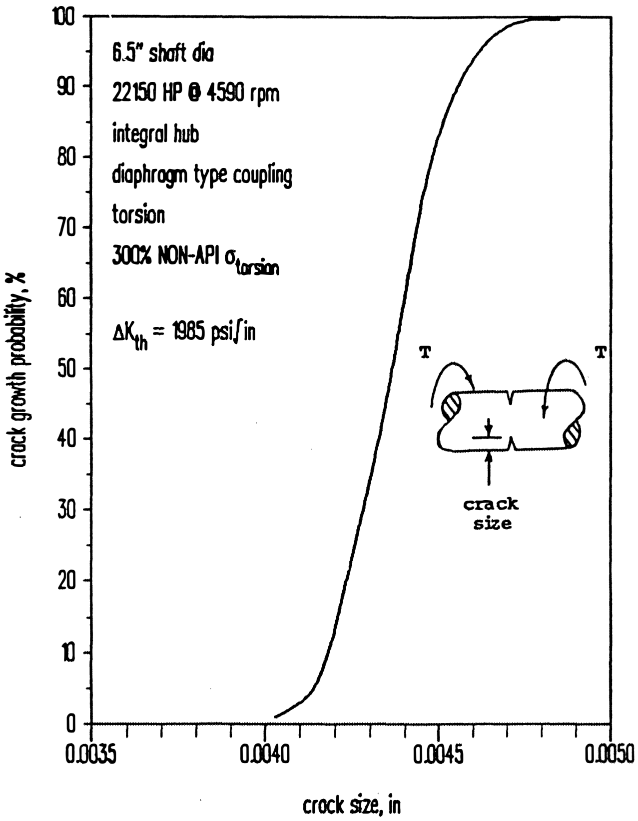

Figure 9. Probability of Crack Growth vs *Crack Depth for Startup Cycle for Rotor 2.*

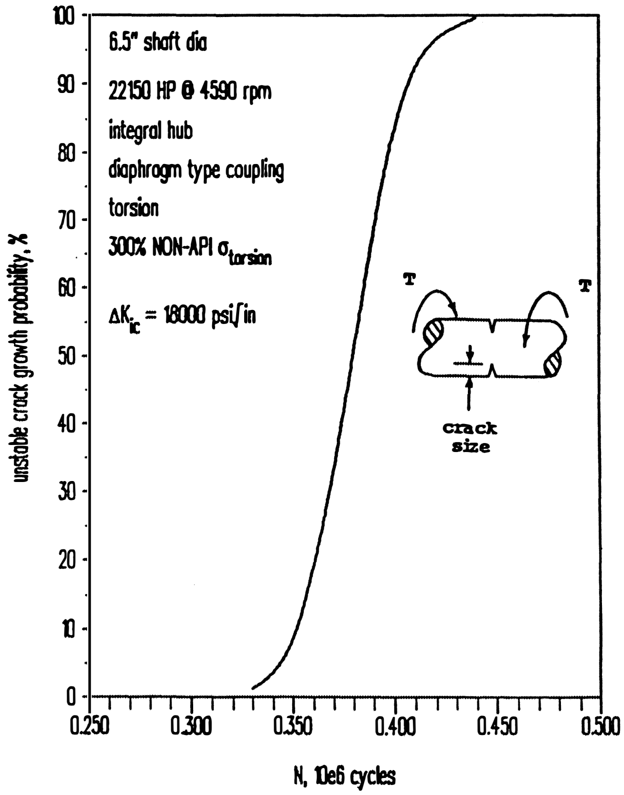

Figure 10. Probability of Unstable Crack Growth vs *Number of Startups for Rotor 2.*

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