REFERENCE

CRACK PROPAGATION IN MAIN COOLANT PUMPS

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ABSTRACT

The Byron Jackson® main coolant pumps in pressurized and boiling water reactors have experienced cracking in the shaft and cover. Based on detailed metallurgic studies and analyses it is now clear that crack initiation is due to thermal cyclic fatigue. The cracks propagate due to temperature cycling and mechanical loading phenomena.

In the vast majority of cases, the cracks propagate only to small depths in spite of many thousands of hours of operation as the propagation is mostly due to the thermal cycling occurring in the annulus region. However, it appears that circumferential cracks can be driven to unacceptably large depths when mechanical loads or other cyclic loads of large magnitude occur during operation. Analytical calculations give good bounding predictions for axial cracks. Countermeasures to overcome the thermal cracking problem are briefly described.

1.0 INTRODUCTION

Byron Jackson boiling water recirculation pumps, (Figure 1.1) have provided many years of reliable, trouble-free service. In the early 1980s, however, significant linear indications (shallow cracks) began to appear in the thermal barrier area of these units. These were observed both on the shaft and in the cover bore in a region extending one inch axially into the thermal barrier to approximately one inch below the barrier. Subsequent in-service inspections have shown that most BWR recirculation pumps are affected. Crack depths range in magnitude from mils to fractions of an inch on both the cover and the shafting. Fractographic examination of the cracked shaft surfaces and analytical studies have convincingly shown that these cracks propagate as a result of thermal cycling fatigue.

The mechanical seal cavity area in these pumps is separated from the hot system water by the provision of the thermal barrier, through which cold water from the closed cooling water (CCW) system is circulated. Further a small quantity of cold water (about 3 gpm.) is injected into the seal cavity to keep the area free of contaminants and to provide a substantially constant thermal environment to the seal. Part of this injection water (0.75 gpm) flows upward through pressure reduction devices in the seal cartridge providing the pressure breakdown for the seal stages. The remaining water (about 2 gpm or more) flows down the annulus between the shaft and the thermal barrier and mixes with the system water. Since mixing is an unsteady phenomenon, it can be expected that the shaft and cover metal surfaces will experience alternating temperatures. If the magnitude of temperature fluctuations is sufficiently large, and if the alternations are repeated enough number of times, cracks will be initiated on the surfaces as a result of thermal cyclic fatigue.

This phenomenon has also been observed on the reactor coolant pumps of pressurized water reactors, where the physical configuration is substantially similar. Cracks have also been observed in pumps without injection. In these pumps, the controlled bleed-off flow for the seal is drawn from the system water. This hot water flows up the annulus and it is cooled by the CCW flowing within the thermal barrier. The water then mixes with the ambient cooler water at the bottom of the seal cavity and hence thermal cracking is observed in that area.

The dynamics of thermal mixing is not very well understood. The frequency and magnitude of thermal fluctuations can be expected to be random if turbulent mixing alone is the cause. However, other phenomena may also be expected to contribute to this problem. Pressure fluctuations are known to exist in the mixing region. These are generated by the nature of the impeller flow field (which characteristically contains once per revolution fluctuations as well as low frequency components generated by rotating stall and separation). Also fluctuations in the pressure generated by the injection pump could be felt in the mixing area. The position of the shaft within the thermal barrier may be eccentric due to radial loads and this will create a non-uniform pressure distribution around the periphery resulting in a circumferentially non-uniform velocity and temperature field. The reactor piping system may also contribute to pressure fluctuations. It is evident that alternating pressures will drive the hot reactor water in and out of the thermal barrier causing the metal surfaces to be exposed to cyclic temperatures.

The orientation of the cracks is axial in the area where there are ACME threads. On the smooth portion of the shaft, the cracks have a circumferential orientation. Also, it has been observed that axial cracks after some degree of propagation into the solid metal can turn circumferential. Whereas the axial cracks driven by thermal cycling tend to reach only finite depths, the circumferentially oriented cracks can be driven by mechanical loads (occurring in the radial direction) and can reach catastrophic depths.

To determine the long-term implications of fatigue cracking, Byron Jackson has developed predictive models of the cracking phenomenon for their recirculation pump geometries. These analyses have been based on certain assumptions regarding frequencies of temperature changes, metallurgical susceptibility to cracking magnitude and direction of mechanical loads and other considerations.

This paper will first describe the observations from the field for BWR pumps. These involve crack depth measurements as well as detailed metallographic examination of the crack surfaces. The paper then summarizes the methods of analyses and compares the results of the calculations against field data. Finally, we will touch briefly upon the countermeasures undertaken to eliminate thermal cracking.

2.0 FIELD OBSERVATIONS

2.1 Rotating Element

Thermal cracking has been found on both BWR and PWR pump shafts. The fatigue cracks range in depth from a few mils to fractions of an inch. The severity of the cracks is governed by the interacting effects of the rate and temperature of seal injection flow (or absence of injection), the presence of pressure oscillations, and the particular pump geometry. For this reason, there is considerable variation from one pump to another.

Thermal fatigue cracks occur both on the ACME thread region within the thermal barrier and on the smooth portion of the shaft immediately below the threads. Crack length varies from one thread width (0.062 inch) up to six threads (1.000 inch). (Figure 2.1) shows a typical crack pattern. The longer, continuous cracks are believed to be deeper. In the groove region the orientation of the cracks is axial. On the smooth portion, cracks occur in a random, 45 degree, or "turtle back" pattern. These cracks can coalesce and form single circumferential cracks, which generally tend to grow deeper.

In the ACME thread region, the fatigue cracks appear to be axial. However, a destruction examination of one BWR pump shaft (which had operated for about 110,000 hours) showed that at a depth between 0.100 and 0.200 inch, the cracks tended to bend into a circumferential direction. On this shaft portion, circumferential crack lengths up to three-quarters of an inch were measured. Metallographic examination indicated that the crack sides had moved relative to each other and the crack tips showed a large plastically deformed material zone.

Thermal cracks have also extended down to the fillet weld that joins the upper side plate of the hydrostatic bearing journal of the shaft. This may jeopardize the integrity of the journal support plate attachment weld.

2.2 Cover/Heat Exchanger

Thermal fatigue cracks occur in the lower portion of the thermal barrier where the two fluids mix. The observed fatigue cracks have been axial in direction, ranging from one thread width (0.062 inch) in length to as long as six threads (1.000 inch). (Figure 2.2) shows a typical thermal crack pattern. On one cover, for which measurements are available, the thermal cracks were found to have propagated into the cover bore to an elevation coinciding with the tapered bottom of the water jacket drilled holes. Thus, crack propagation into the ligament region of the thermal barrier in these BWR covers could breach the pressure boundary. However, there has been no documented evidence in this or other covers of such a pressure boundary breach.

3.0 ANALYSIS

3.1 Crack Initiation

Cyclic thermal stresses are induced in the shaft material as a result of the mixing of the cold injection water with the hot system water. The modelling of this phenomena is complex. However, based on certain assumptions regarding pressure pulsations, an unsteady flow and heat transfer analysis can be performed (1). If a further assumption is made, the analysis can be greatly simplified. This assumption is that the bounding value of temperature fluctuations is that given by steady-state results for the extreme cases of purge flow on and off (i.e., zero flow through the annulus). This quasi-steady state assumption is justified in view of the thermal boundary layer thickness which can be shown to be less than 0.025" for the once per revolution driving frequency.

Steady-state thermal and stress calculations were made for annulus flows of 5 gpm, and 0 gpm (no flow). (Figure 3.1) shows the temperature distribution along the shaft length for these flows, and (Figure 3.2) shows the hoop stress variation along the shaft length at the bottom of the thermal barrier.

Element 64 is near the critical shaft area where cracks have been observed. At this location the hoop stress changes from a tensile value of about 20,000 psi to a compressive value of -12,000 when the flow rate changes from 5 gpm to 0 gpm. Thus, if the flow were to change from 5 gpm to 0 gpm cyclically, the shaft element will see a stress variation from +20,000 to -12,000 psi at the same frequency, thus giving rise to a peak-to-peak alternating stress magnitude of 32,000 psi. If the driving frequency were of the order of shaft speed, it is easy to see that high cycle fatigue can be set up and cause the shaft to have crack initiation.

3.2 Crack Propagation

After initiation, cracks can continue to grow if the alternating stresses through the shaft are adequate to drive the crack. The driving stresses are calculated using a set of transient boundary conditions on a sufficiently refined finite element model.

Transient Stress Analysis

The steady-state analysis results presented in (Figure 3.1) show that the fluid temperature in the annulus at the critical location changed from 450°F when there was no flow to 120°F where there was a 5 gpm downward flow. During cycling conditions induced by mixing, the actual temperature difference ΔT will be less than the one computed for steady-state conditions. Therefore, if we assume that ΔT will remain as 330°F, (450°-120°F), independent of frequency, then the results will provide a conservative estimate for crack propagation.

The metal temperature difference, of course, will be a function of the frequency. Unfortunately, the frequency at which the pressure pulsations occur is not known at this stage. Therefore, parametric calculations were carried out for several frequencies using a detailed finite element model of one land with individual element sizes of about 0.005". Boundary conditions for the land model were extracted from the analysis results using the overall model of the cover and shaft.

For the calculation of crack propagation, the variation of hoop stresses with time was established by inputting the water temperature as a sinusoidal function of time at a specified frequency (25 Hz, 12.5 Hz and 2.5 Hz) with an amplitude of 330°F. At each element, stresses were calculated as a function of time and the maximum and minimum stresses were established. These stresses vary with depth and a typical variation is shown in (Figure 3.3). It can be seen that at this frequency of 2.5 Hz, beyond about 1/4 in. the stress is independent of time i.e., the cyclic fluctuations of water temperature are not felt beyond 1/4 in. depth. The depth of penetration decreases rapidly with increasing frequency.

Growth Analysis

The rate of crack propagation depends upon the stress intensity factor range ΔK. The method of ΔK calculation for arbitrary stress loading is described in Appendix 1. It is assumed that the actual crack conditions resemble an edge crack in a semi-infinite plane subjected to concentrated forces. The calculated ΔK values for frequency = 2.5 Hz are shown in (Figure 3.4).

The crack growth rate is a function of the ΔK values. A Paris Law form is used for this purpose.

$$\frac{da}{dN} - C\ (\Delta K)^m$$

where C and m are constants, being properties of the material and

a: crack depth

and N the number of cycles.

A piecewise integration technique is used to calculate the number of cycles required for the crack to grow to various depths.

The constants, C and m are material properties and they vary somewhat with material condition. For the calculations shown in the report, the following values were used for the 2.5 Hz case

$m = 3.86$

$C = 5 * 10^{-24}$ for ΔK expressed in psi $\sqrt{in}$

It is known that below a certain threshold value for ΔK, crack growth is arrested. This ΔK (threshold) is estimated to be anywhere between 3000 and 6000 psi $\sqrt{in}$. The present calculations do not assume a threshold limit so that conservative predictions can be made.

(Figure 3.5) shows the predicted crack depths as a function of operating hours. As may be expected, the 2.5 Hz case shows larger crack depths than the higher frequency cases.

3.3 Effects of Mechanical Loads

Centrifugal pump impellers experience radial forces during operation. These forces consist of steady-state and sinusoidal components. If the shaft contains a circumferential crack, as the shaft turns, the steady radial force appears as an alternating moment to the crack. Consequently such cracks can be driven by steady-state radial loads.

Most double volute pumps operating at their best efficiency point (B.E.P.) experience very low radial loads. (3). A radial force coefficient F_r is commonly defined as follows

$$F_r = \frac{2.31 \cdot F}{H \cdot D_2 \cdot B_2}$$

where F : Radial force in lbs.
H : Pump head in ft.
D_2 : Impeller diameter in inches
and B_2 : Impeller exit width in inches

For most recirculating pumps, with specific speeds of 2500, the value of F_r is about 0.03 at B.E.P. However, for some pumps operated at off design conditions as in BWR6 plants, the value can increase up to 0.06 or more. Further, unsteady dynamic loads almost equal in magnitude to the steady loads are also possible (4). Also, the axial thrust does not pass exactly through the centerline because of pressure non-uniformities and this generates an additional bending moment. If all of these factors work in phase, the net radial force tending to drive the circumferential crack can indeed be well in excess of the 0.03 value quoted above.

Whether a radial load will drive a circumferential crack depends upon its magnitude and the crack depth. The crack tip stress intensity factor increases with load and crack depth. When this value exceeds the threshold value (~ 3000 psi $\sqrt{in}$), crack growth will occur. The critical crack depth at which such growth is possible is shown plotted as a function of F_r in (Figure 3.6). It can be seen that in order to drive a crack of 0.3" depth (a value that is likely from thermal growth alone), F_r has to be in excess of about 0.12. The foregoing considerations suggest that this eventuality of crack growth through mechanical loads cannot be ruled out. (Figure 3.7) shows crack growth scenarios for different assumed F_r values superimposed on 2.5 Hz thermal growth. If the radial load is large (F_r = 0.2), a very short life of 40,000 hours is indicated. But for normal radial loads (F_r < 0.12) applicable to most recirculating pumps, the potential for mechanical loads driving a thermally induced crack is very small.

3.4 Cover Crack Analysis

The analysis for prediction of crack growth in the cover was carried out using an advanced time dependent method in which the unsteady flow and heat transfer in the annulus were solved simultaneously. Full details may be found in (1).

4.0 COMPARISON WITH FIELD OBSERVATIONS

Crack depth data were obtained from some of the operating pumps. The measurements were made using the A.C. potential drop method or by progressive machining. These data are plotted as a function of operating hours in (Figure 4.1). Axial and circumferential cracks are shown separately. The prediction for thermal cracks is shown for 2.5 Hz and 25 Hz. It can be seen that nearly all axial cracks are bounded by these two curves. Some of the circumferential cracks clearly fall well outside the thermally expected range, demonstrating that mechanical loads have driven them. It is also interesting to observe that a number of circumferential cracks are bounded by the thermal curves indicating that for these cases, the mechanical loads are small enough not to be able to drive these cracks.

Of the four circumferential crack points clearly driven by mechanical loads, three are from the same plant and a great deal of study has gone into understanding the root cause of the problem. A report on this activity is beyond the scope of this paper.

The metallographic finding that axial crack tips show blunting and oxide deposits is supported by the theoretical calculation that beyond about 50,000 hours, crack growth is very slow, if not arrested. For circumferential cracks, once they start growing by mechanical loads, the growth rate will be rapid and blunting of the crack tip can not be expected.

The fact that the analysis predicts the axial crack depths reasonably well suggests that no major phenomena is left unaccounted for. In other words, environmental effects, material imperfections etc. are unlikely to have played an important part in this phenomenon.

(Figure 4.2) shows the comparison between calculation and measurement for the cover. The more advanced calculations differentiates the effect of purge flow quantity. The field points do not belong,, specifically, to a given flow rate as the purge flow was changed during operation. Field observations to date support the reduction of purge flow. A recommendation was made to the operating utilities to reduce the purge flow from the 3-5 gpm range to a value consistent with the flow control system capability and accuracy.

5.0 COUNTERMEASURES

Based on the research work done to date and field observations, a two-phase strategy has been adopted as countermeasure to the cracking problem. The first phase applies to installations which have reached a number of operating hours at which it was considered prudent to conduct an in-service inspection. For these installations, new components of the same basic design but with some significant improvements were offered. These improvements include welded impeller, welded journal, optional provision of U.T. hole through the shaft to facilitate future inspections, higher strength/lower carbon materials, optimized drilled hole construction for the cover, optional inspection ports, double gaskets etc. These upgrades, by themselves, do not eliminate the mechanism for thermal cracking, but provide a greater margin for safe operation in combination with reduced purge flow operation.

The second phase in this program is the development of a permanent countermeasure which will substantially eliminate the thermal cracking mechanism. The basis for this concept, as described in (5), is the elimination of the temperature differential between the mixing streams by elevating the temperature of the purge flow after emerging from the seal cavity and just prior to mixing with the system water. The concept has been fully tested in mock-up arrangements and other configurations in Japan. It is scheduled to be installed at a new plant, in the near future.

6.0 CONCLUSIONS

- The thermal cracks observed on the shaft and cover of Byron Jackson reactor recirculating pumps are initiated by thermal mixing between the cold injection water and hot system water.
- Cracks propagated purely by thermal cycling phenomena reach depths not much in excess of about 1/4 in. Such cracks by themselves do not cause significant impact on safety or operability of the pump.
- Thermal cracks can reach circumferential orientation and in this configuration can be driven by mechanical loads. The potential for the crack to reach unacceptable depth under these circumstances can not be minimized.
- An analysis based on a quasi steady approximation of the temperature field gives a bounding

prediction for axial cracks.

- Unbounded crack growth is predicted only for mechanical loads well in excess of normal design values such as may be expected in pumps operated at off-design conditions.
- An interim countermeasure consisting of the same basic design but with some significant upgrades has been offered for installations for which the 10 year in-service inspections had to be performed.
- A permanent countermeasure which will substantially eliminate the thermal cracking problem has been developed and has just completed final verification tests. It will be offered for installation in the very near future.

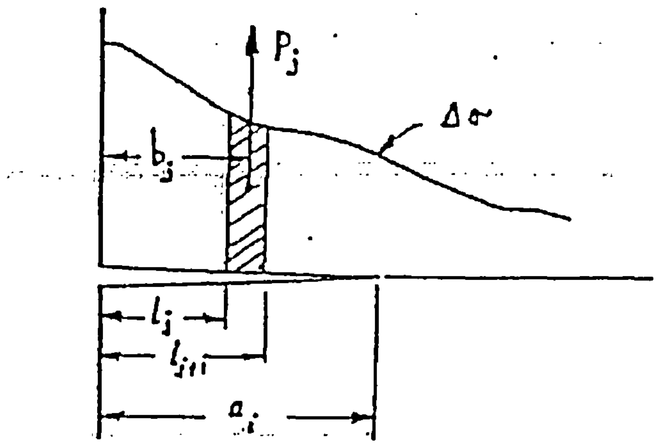
7.0 REFERENCES

1. S. Gopalakrishnan, P. Payvar and C. Reimers, "An Analytical Investigation of Thermal Cracking in Reactor Recirculating Pumps". Paper to be presented at the VDMA Pumpentagung at Karlsruhe, October 1992.
2. G.C. Sih ed, "Handbook of Stress Intensity Factors for Researchers and Engineers", Lehigh University, PA, 1973.
3. E. Makay, "Centrifugal Pump Hydraulic Instability", EPRI Report No. CS-1445 (1980).
4. J. Verhoeven, "Rotordynamic Considerations in the Design of High Speed, Multistage Centrifugal Pumps", 5th International Pump Users Symposium, Texas A&M University, 1988.
5. C. Boster, S. Gopalakrishnan, C. Reimers and G. Vaghasia, "Pump with Seal Purge Heater" Patent applied for.

APPENDIX I

CALCULATION OF STRESS INTENSITY FACTORS AND CRACK GROWTH RATES

For a half-plane containing an edge crack subjected to a pair of equal and opposite concentrated force, the stress intensity factor is given in Reference (2). These results can be extended to an arbitrary stress distribution as shown below



$$P_j = \frac{\Delta\sigma_j + \Delta\sigma_{j+1}}{2} * (l_{j+1} - l_j)$$

$$b_j = (l_j + l_{j+1})/2.$$

$$\Delta K_i = \sum_{j=1}^{i-1} \frac{2}{\sqrt{\pi}} P_j \sqrt{a_i} \frac{1 + F(b_j/a_i)}{\sqrt{a_i^2 - b_j^2}}$$

Let $b/a = r$

$$F(r) = (1-r^2)\,[.2945 - .3912\,r^2 + .7685\,r^4 - .9942\,r^6 + .5094\,r^8]$$

In the above equations for deriving P and ΔK, it is assumed that the stress distribution is tension at all times.

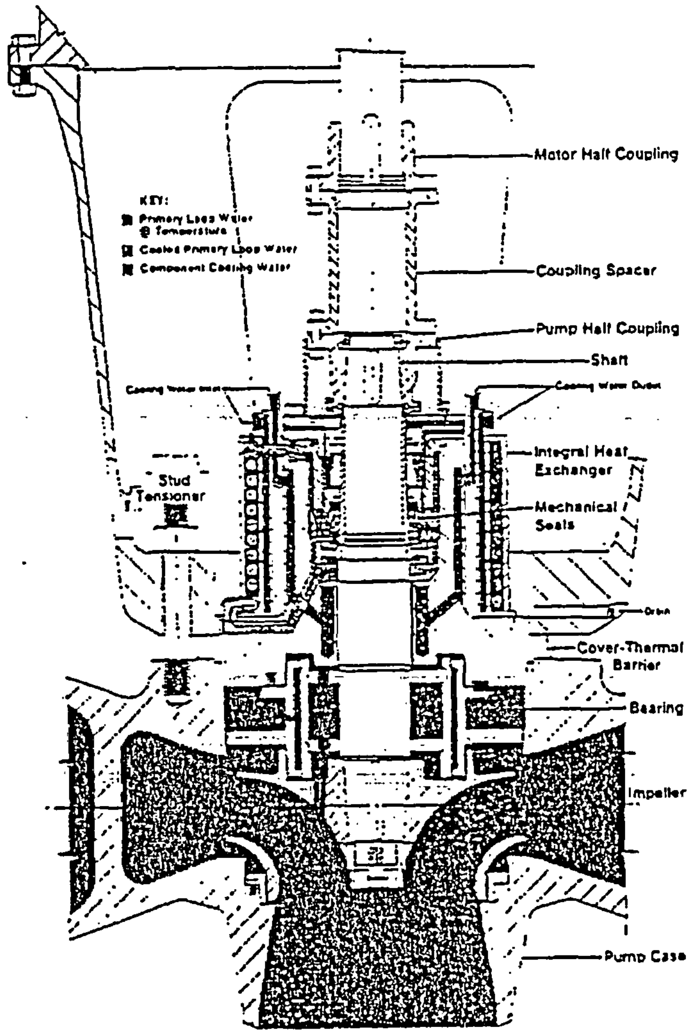


Fig. 1.1 BJ PRIMARY NUCLEAR PUMP

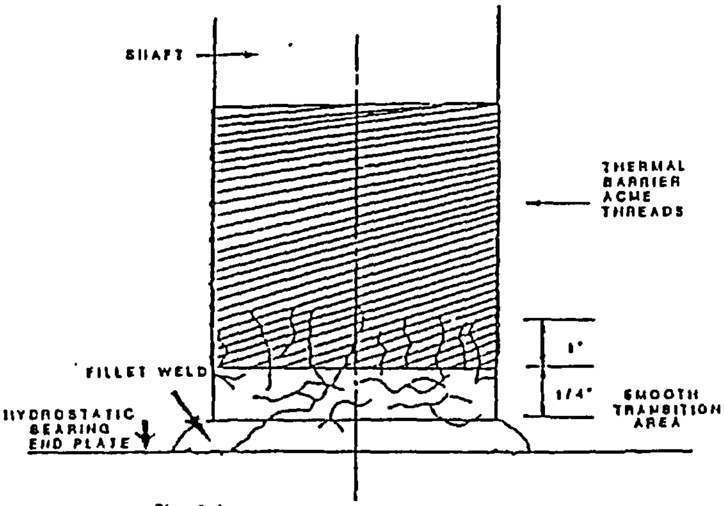


Fig. 2.1
TYPICAL CRACK PATTERNS ON SHAFT

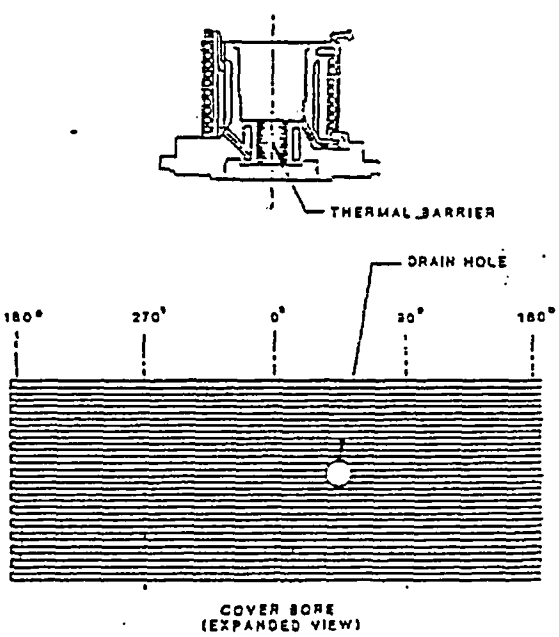


FIG. 2.2
COVER THERMAL FATIGUE CRACKS
(TYPICAL FOR PUMPS WITH INJECTION)

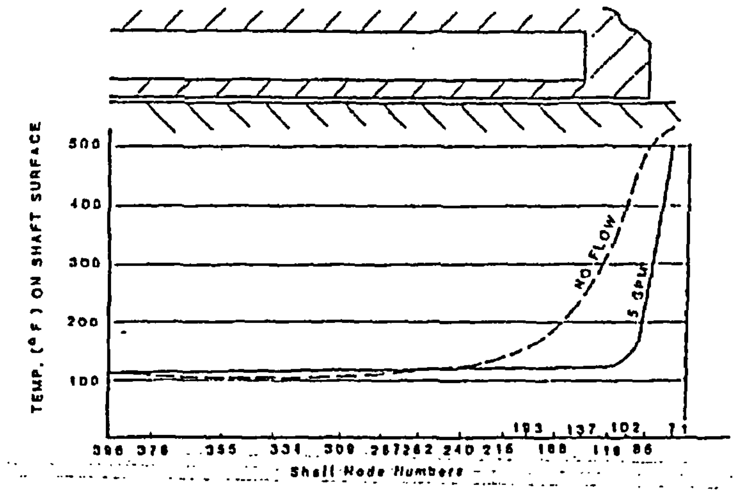


FIG. 3.1
TEMPERATURE DISTRIBUTION ALONG SHAFT SURFACE

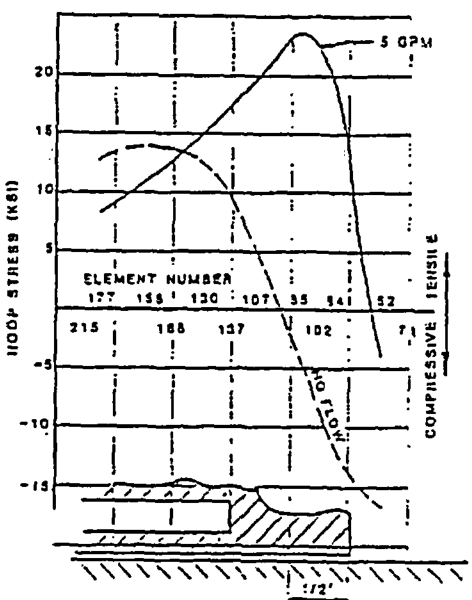


FIG. 3.2
HOOP STRESS ON SHAFT SURFACE

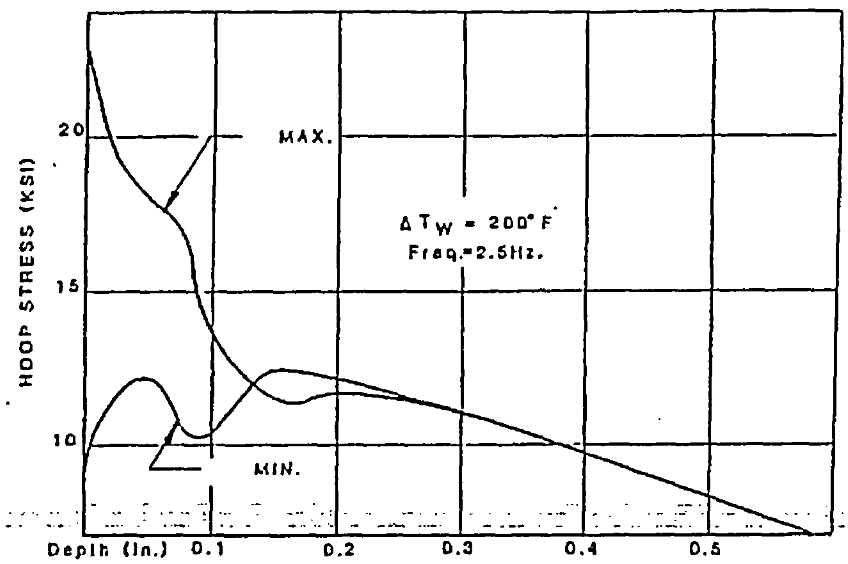


FIG. 3.3
VARIATION OF HOOP STRESS WITH DEPTH (2.5Hz)

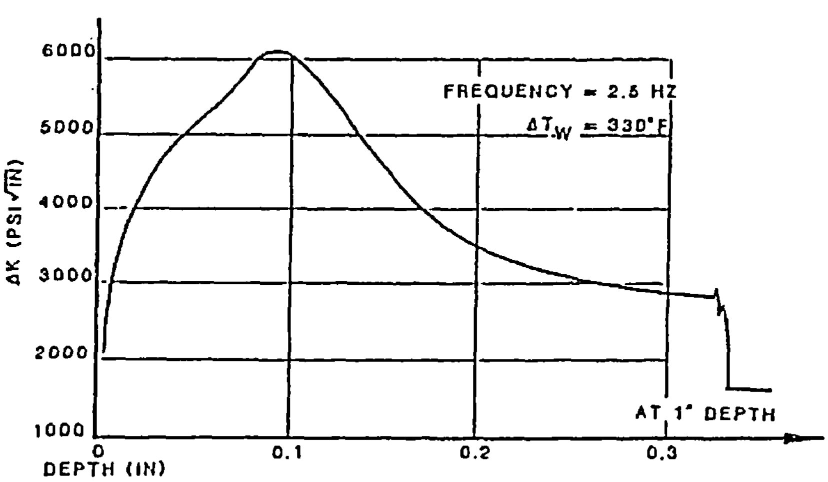


Fig. 3.4
VARIATION OF STRESS INTENSITY FACTOR RANGE

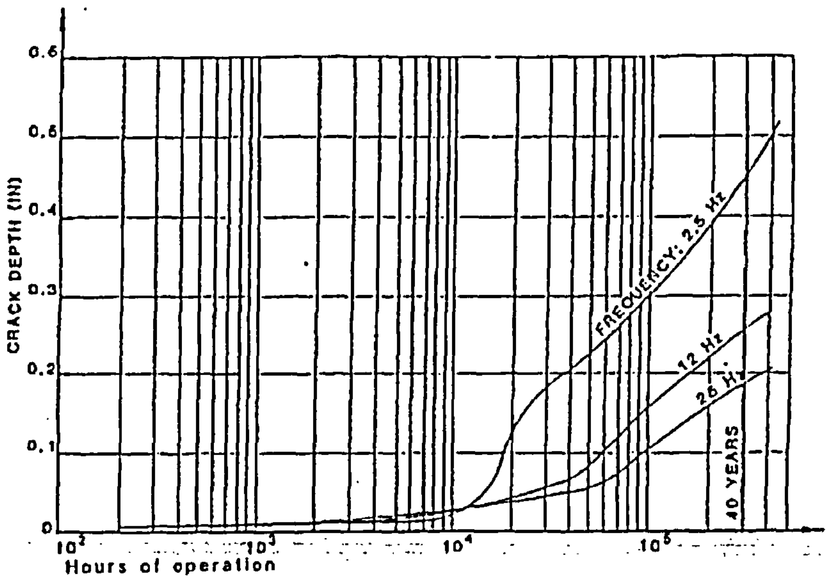


Fig. 3.5 CRACK GROWTH vs. TIME

CRITICAL CRACK DEPTH

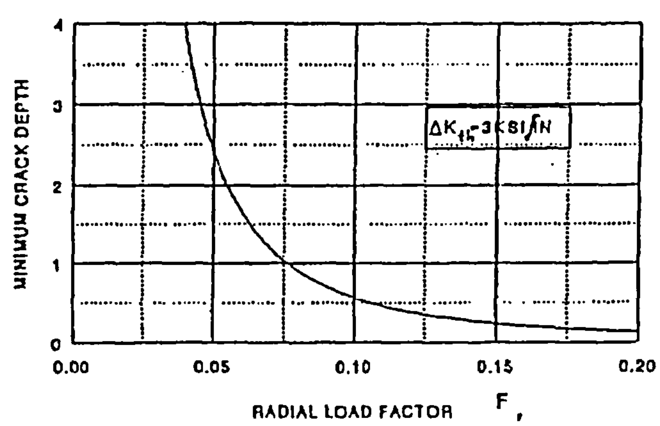


Fig. 3.6

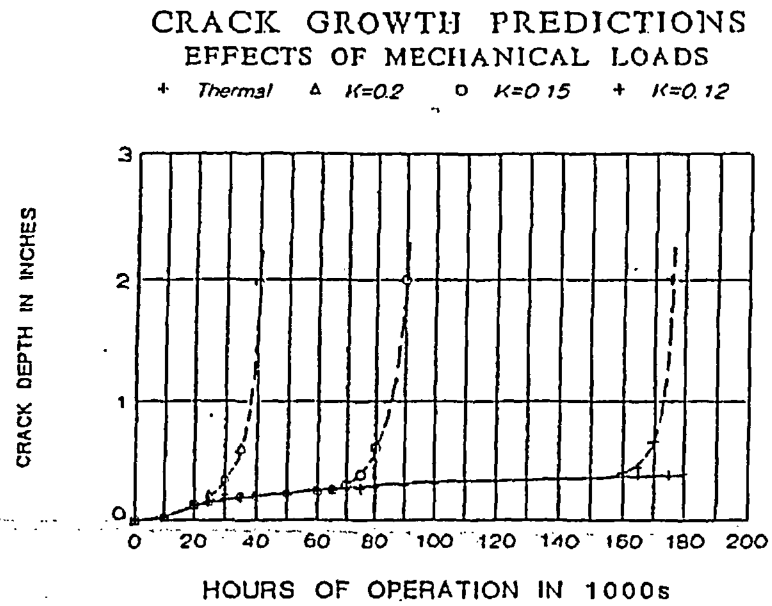

CRACK GROWTH PREDICTIONS
EFFECTS OF MECHANICAL LOADS
+ Thermal ∆ K=0.2 o K=0.15 + K=0.12
CRACK DEPTH IN INCHES
0
1
2
3
0 20 40 60 80 100 120 140 160 180 200
HOURS OF OPERATION IN 1000s

Fig. 3.7

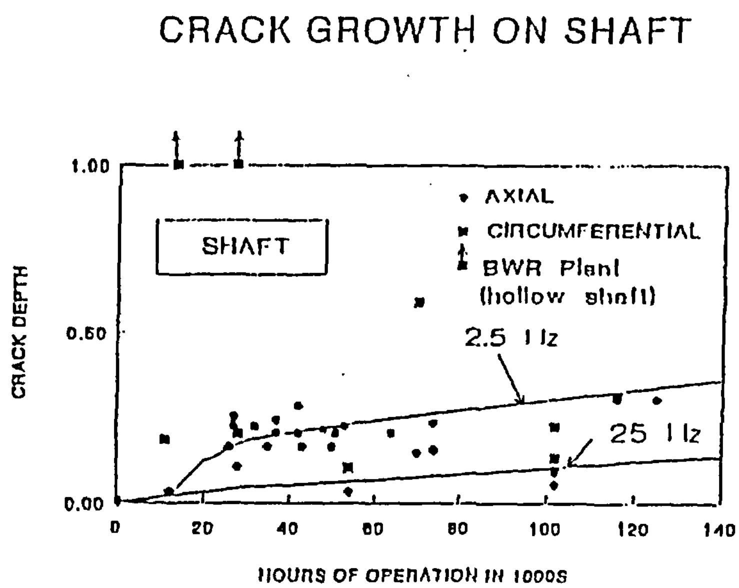

CRACK GROWTH ON SHAFT
SHAFT
AXIAL
CIRCUMFERENTIAL
BWR Plant
(hollow shaft)
2.5 Hz
25 Hz
CRACK DEPTH
0.00
0.50
1.00
0 20 40 60 80 100 120 140
HOURS OF OPERATION IN 1000S

Fig. 4.1

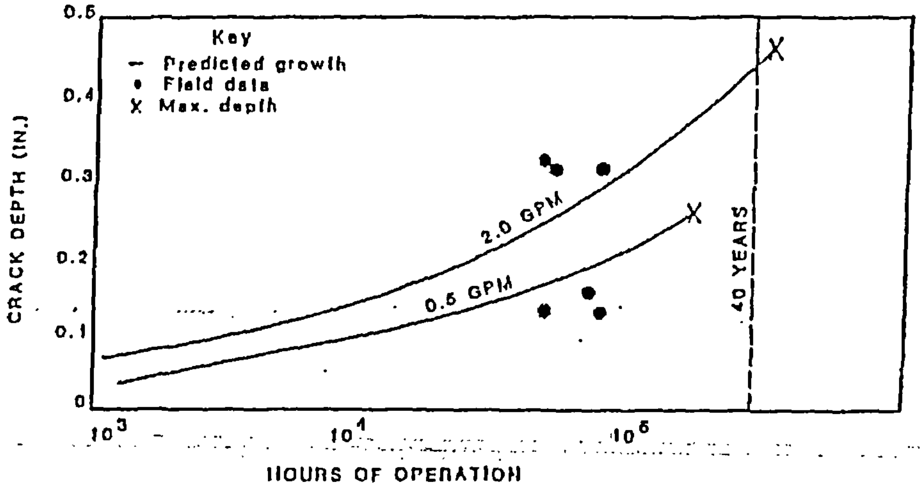


FIG. 4.2
COMPARISON OF COVER THERMAL FATIGUE PREDICTIONS WITH FIELD DATA