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Condition assessment of generator rotor shaft of 108 MW hydro plant through phased array ultrasonic technique and estimation of remaining life through finite element analysis

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The Condition Assessment of Generator rotor shaft of typical 108MW plant has been carried out involving the application of advanced Phased Array Ultrasonic Technique (PAUT)to quantify the volumetric nature of the defects in the rotor. This technique involves mapping of the entire rotor volume in different segments and layers using multiple ultrasonic sensors. The imaging of defects in the rotor shaft was carried out in four different diameter sections 558 mm, 610mm, 737 mm and 914 mm. The scanning of entire shaft geometry regions was done following different patches from top to bottom portion of the shaft by PAUT and analysis of individual scanning results for evaluation of crack size, position inside the rotor geometry and orientation. The critical defects identified by this technique with their locations and orientation inside the rotor has been presented. The stress and fatigue life of the shaft under defect free conditions have been carried out by finite element analysis using the commercial code Fe-Safe™. The calculation of remaining life of the rotor shaft was carried out based on the principle of fracture mechanics using the code ZENCRACK™. The results of the crack growth rate data of critical cracks close to the surface were presented.

Keywords: *Hydro turbine generator, phased array ultrasonic inspection, stress analysis*

1.0 INTRODUCTION

The refurbishment and uprating of hydro plants in India are being perused in a big way in units which have completed more than 40 years service and a number of case histories in this regard has been reported [1-3]. While, the generator and turbine components undergo a major change in the design during uprating in terms of insulation, rotor poles, turbine runner etc., the shaft components are retained keeping in view the available margin on the factor of safety. The rotor shaft of hydro plants is generally very large in diameter and length. In view of the criticality of loading and high magnitude of torsional stresses during service, they are generally designed as a hollow section with a high factor of safety. As these rotors are manufactured through casting and forging route, they invariably considered to have process-induced defects like slag, blow holes etc. [4-5]. Owing to the non-availability of proven Non-Destructive evaluation techniques during late 70's, the rotor components made with controlled process parameters have been put into use. Thus, the structural integrity assessment of turbine-generator rotor shafts of hydro plants with particular reference to remaining life prediction is considered critical from the view point of achieving safe and reliable operation under the uprated conditions. Hence, the condition assessment of these rotors is being pursued on a regular basis

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by the utilities and forms an important exercise during annual maintenance program.

With the advent of advanced ultrasonic techniques in late 90's, the improved quality control measures have been put into practice. While the conventional ultrasonic technique has been routinely applied for the identification of various defects such as crack, discontinuity etc. along with its location inside the component, the evaluation of the orientation of the defect is not possible. The orientation of crack geometry with reference to the axis of rotation affects its growth during service. Any existing crack in the rotor which is in phase with the direction of shear stress grows at arapidrate under torsional loading conditions. The rate of crack growth is affected by the material properties as well as the crack geometry. Any crack reaching the surface is considered detrimental as it leads to catastrophic failure of the rotor.

FEA based damage Tolerance studies of Generator Shaft considering critical cracks identified by PAUT inspection in high-stress regions was conducted. The overall procedure followed in determination of the remaining life of the cracked shaft are summarized below.

- Identification of crack locations inside the rotor through PAUT.
- Detection of surface cracks through Magnetic and Dye penetrant methods
- Geometric modeling of the shaft -CAD modeling of the rotor shaft based on the design drawing
- Meshing of the shaft- Discretization into small elements for analysis
- Incorporation of critical crack geometries observed by the PAUT
- Stress analysis of rotor for prediction of fatigue life
- Prediction of crack growth using programs like Fe-safe and ZENCRACK™ modules for arriving at the extended life under uprated condition

2.0 DETAILS OF THE GENERATOR ROTOR STUDIED

The generator rotor shaft studied, consisted of four different diameter sections 558 mm, 610 mm, 737 mm and 914 mm along its full length as shown in Figure 1. The generator shaft of the hydro turbine supports the rotor part weighing 4, 37,269 kg and the turbine shaft and the pelton bucket runner. The weight of the turbine shaft and runner was 66,100 kg. The generator shaft rotates at a speed of 166.7 rpm. The hydrodynamic bearing supports the top region of the generator shaft. The self-weight of the shaft also acts as the load in the vertical direction to the shaft, which is compensated by the upward thrust by the hydraulic pump during normal service.

FIG. 1. VIEW OF GENERATOR SHAFT WITH ROTATING MASS

2.1 Phased Array ultrasonic inspection of rotor shaft

The application of Phased Array Ultrasonic Technique (PAUT) which has been developed primarily for complex geometry of turbine components has been adopted in the present study. A phased array system is normally based on a specialized ultrasonic transducer that contains many individual elements (typically from 16 to 256) that can be pulsed separately in a programmed pattern. An array transducer is simply one that contains a number of separate elements in a single housing, and phasing refers to how those elements are sequentially pulsed. These transducers are used with various types of wedges, both in contact and immersion testing. Their shape may be square, rectangular, or round, and test frequencies are most commonly in the range from 1 to 10 MHz. Phased array systems pulse and receive from multiple elements of an array as shown in Figure 2. These elements are pulsed in such away as to cause multiple beam components to combine with each other and form a single wave front traveling in the desired direction. Similarly, the receiver function combines the input from multiple elements into a single presentation.

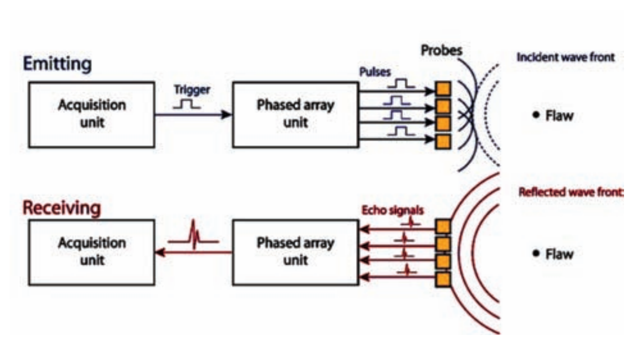

FIG. 2. BASIC PRINCIPLE OF PHASED ARRAY ULTRASONIC TECHNIQUE

Because phasing technology permits electronic beam shaping and steering, it is possible to generate a vast number of different ultrasonic beam profiles from a single probe assembly and this beam steering can be dynamically programmed to create electronic scans:

The capabilities of PAUT system used for the inspection includes the following:

- Software control of beam angle, focal distance, and beam spot size. These parameters can be dynamically scanned at each inspection point to optimize incident angle and signal-to-noise for each part geometry.
- Multiple-angle inspection can be performed with a single, small, multi-element probe and wedge, offering either single fixed angles or a scan through a range of angles.
- These capabilities provide greater flexibility for inspection of complex geometries and tests in which part geometry limits access.

Multiplexing across many elements allows motionless high-speed scans from a single transducer position. More than one scan may be performed from a single location with various inspection angles.

The typical PAUT output results showing various scans is shown in Figure 3.

3.0 COMPARISON BETWEEN CONVENTIONAL UT AND PAUT IMAGING

The benefits of phased array technology over conventional UT come from its ability to use multiple elements to steer, focus and scan beams with a single transducer assembly. Beam steering, commonly referred to sectorial scanning, has been applied for mapping of complete volume of the component at appropriate angles. This can greatly simplify the inspection of components with complex geometry. The small footprint of the transducer and the ability to sweep the beam without moving the probe also aids inspection of such components in situations where there is limited access for mechanical scanning. The ability to focus at multiple depths also improves the ability for sizing critical defects for volumetric inspections. Focusing can significantly improve the signal-to-noise ratio in challenging applications, and electronic scanning across many groups of elements allows for C-Scan images to be produced very rapidly in combination with A and B-scan images as shown in Figure 3.

4.0 METHODOLOGY OF INSPECTION

CPRI has identified four diametrical regions of the shaft i.e., 558 mm diameter, 610 mm, 737 mm and 914 mm diameter for PAUT mapping. In order to cover the entire volume of the shaft, the shaft regions were divided into a number of patches of 50 mm length

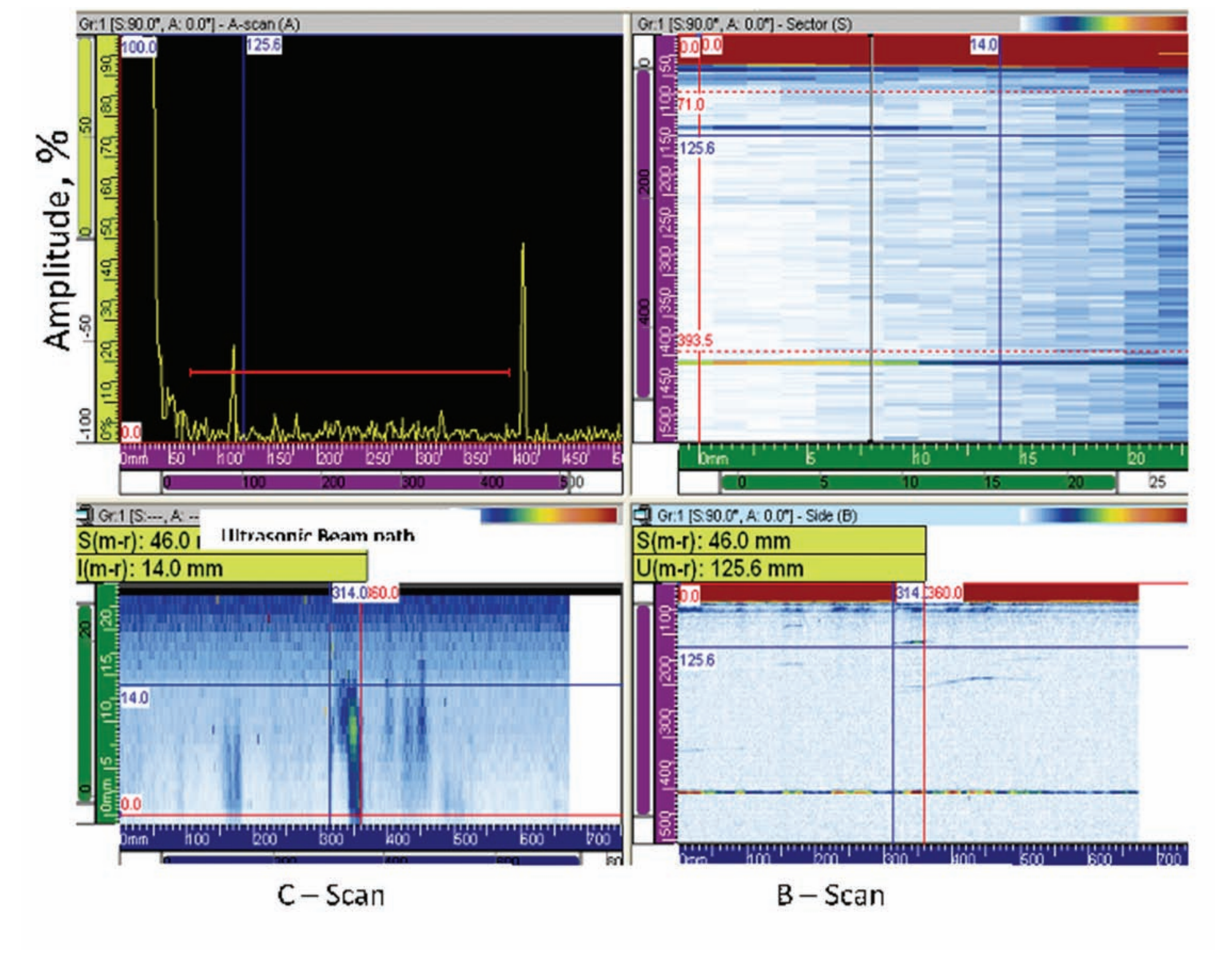

FIG. 3. TYPICAL PAUT OUTPUT RESULTS

matching with that of the phased array probe length. Further, the complete circumference of the shaft was divided into four different quadrants in the clockwise direction. The no. of patches made in each of the diametrical region and the section thickness of the shaft is given in Table 1.

FIG. 4. PATCHING MADE ON 914 MM DIAMETER REGION OF ROTOR SHAFT FOR PAUT

The PAUT probe was scanned along each of the patches and data acquisition was carried out on four segments in a single patch. All the marked patches were scanned and UT spectrum response data has been acquired. The patching made on typical shaft region during the ultrasonic testing is shown in Figure 4 and the identification of surface cracks through the dye penetrant inspection is shown in Figure 5.

5.0 STRESS ANALYSIS OF THE SHAFT BY FINITE ELEMENT METHOD

The generator shaft supports the rotor part of the generator which houses the electric poles, the turbine shaft, and the runner. The top region of the generator shaft is supported by ahydrodynamic bearing. The self-weight of the shaft also acts as the load in the vertical gravity direction.

FIG. 5. DYE PENETRANT INSPECTION OF THE ROTOR SHAFT

TABLE 1				
MEASUREMENT OF SHAFT DIMENSIONS				
OD OF SHAFT REGION (MM)	BORE SIZE (MM)	SECTION THICK-NESS (MM)	PERIME-TER (MM)	NO. OF CRACK INDICA-TIONS
565	88.9	238	1774	11
609.6	88.9	260	1914	19
914.4	88.9	413	2871	1
736.6	88.9	323	2312	1
1134	171.45	485	3561	-
965.2	171.45	397	3031	4

As a part of life estimation, the following analysis was carried out on the meshed model of the shaft.

1. Estimation of stress levels of the shaft under defect free conditions for two different conditions of rated design capacity and uprated capacity conditions – to identify critical stress regions
2. Estimation of fatigue life of rotor shaft using Fe-SAFE™ program
3. Estimation of service life of cracked shaft under uprated load conditions through fatigue crack growth analysis using ZENCRACK™ program
4. Prediction of the remaining life of the rotor shaft at uprated load conditions so as to have a minimum service life of one year.

5.1 Material properties and Boundary conditions used

The generator shaft is made up of UNS S41000 and the crack growth rate linked parameters such as Paris constant and Paris exponent was selected matching with the strength properties of the rotor shaft material as given in Table 2.

TABLE 2	
MATERIAL PROPERTIES OF ROTOR SHAFT	
Young's modulus	200 GPa
Poisson's ratio	0.3
Yield strength	275MPa
Tensile strength	540 - 640MPa
Fracture toughness	105 MPa $\sqrt{m}$
Paris constant, C	10^{-9}m/cycle
Paris exponent, m	2.8

The 3D CAD model of the hydro turbine rotor assembly was created in CATIA V5™, as shown in Figure 6. Generator shaft was meshed with tetrahedral elements in ABAQUS™, with a total of 79,171 elements as shown in Figure 7. The boundary conditions considered for the stress analysis is given in Figure 8. During the operating condition, the turbine torque is considered consumed by the generator rotor. The shaft is allowed to rotate in the Y-direction and the translations displacements were constrained for the shaft.

FE analysis takes into account the geometric, material and loading histories in the preprocessing stage. Stresses and strains are taken from the FE analysis to the fatigue calculations. The fatigue analysis was carried out by Fe-SAFE™ software tool adopting Brown Miller Algorithm. The calculation of fatigue life due to each load block and compute the damage for each load block. Finally, the total damage for the defined load spectra and calculation of a number of cycles for failure is carried out.

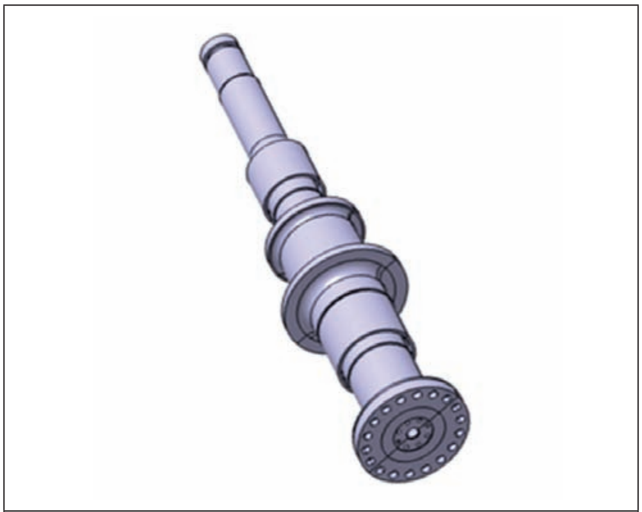

FIG. 6. 3D GEOMETRY OF SHAFT

FIG. 7. MESHED MODEL OF SHAFT

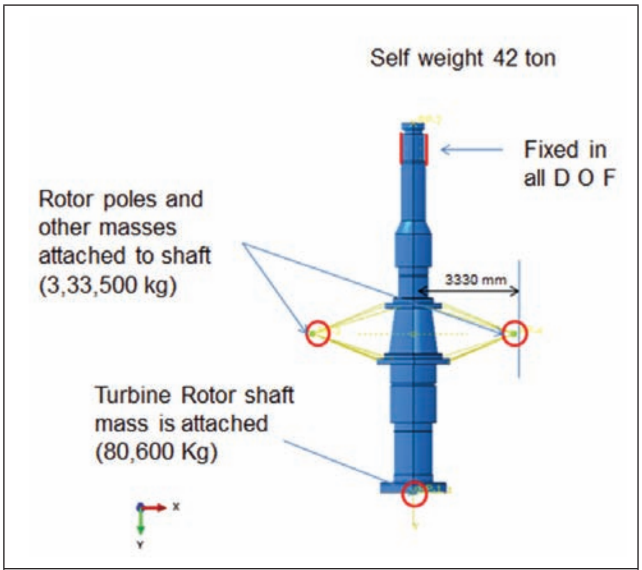

FIG. 8. BOUNDARY CONDITIONS USED FOR STRESS ANALYSIS

6.0 RESULTS AND DISCUSSION

6.1 PAUT crack location results

The B-Scan mapping clearly identified multiple cracked regions inside the shaft in three regions namely close to the outer surface, mid wall as well as close to ID side bore sections. Multiple indications with inter connectivity is as seen in the scanned areas.

Considering the type of indications observed and the maximum depth suggests that the indications are pertaining to service oriented which has originated from the segregation available in the shaft and cyclic loading resulting in fatigue crack initiation and growth. The results indicated that the available integral section thickness is less than 16% of the original thickness. The observed crack locations along with their dimensions are given in Table 4.

6.2 Results of Stress analysis

Finite element (FE) analysis of the generator shaft was carried out at a maximum rotational speed of 166.7 rpm. Global Y direction is considered as the axis of rotation.

The average stress in a rotating component in all three directions of x, y & z is expressed in terms of von-mises stress. The maximum von-Misses stress observed from the static analysis with a rotational body force is observed to be 68.2 MPa against the allowable maximum tensile stress of 91.7 MPa (Figure 9) and the maximum stress location is observed in the region just above the turbine rotor bolting flange as shown by the green color contour region. The top end of the generator shaft above the rotating mass mounting flange region has been observed to experience comparatively very low-stress levels. The tensile stress acting along the length of the rotor shaft in vertical direction

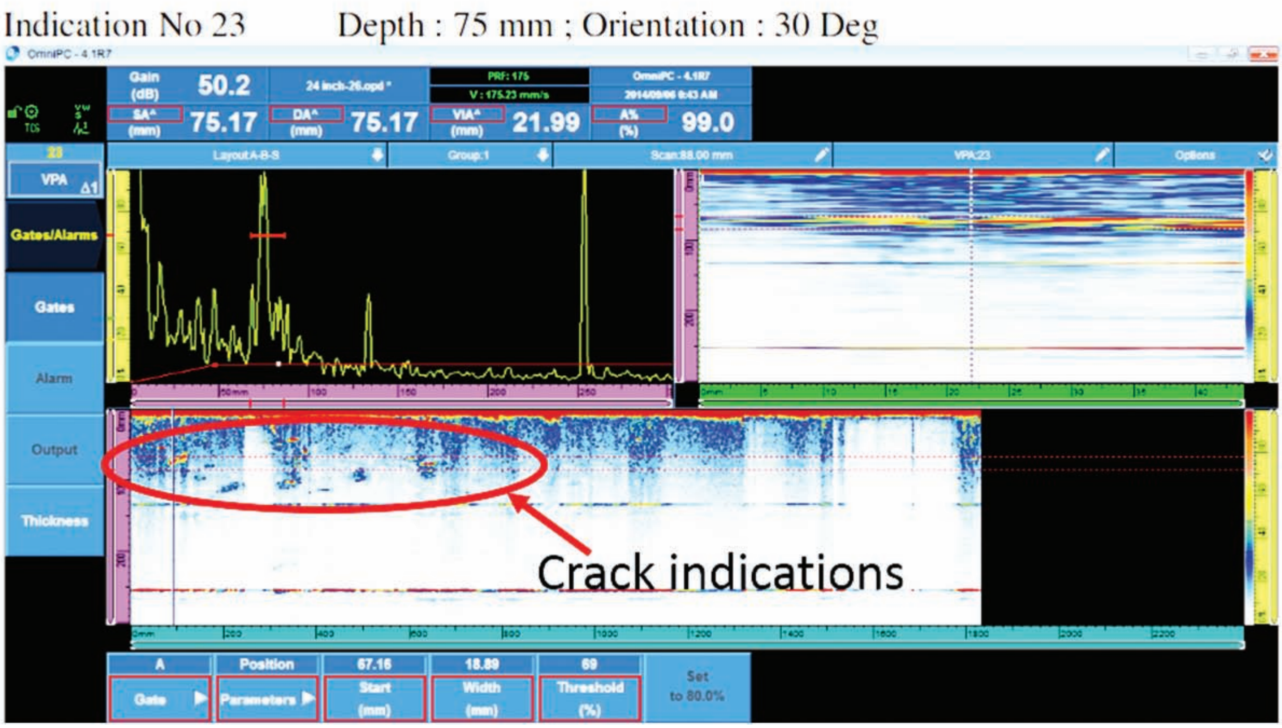

FIG. 9. THE A, B AND C-SCAN OUTPUT RESULTS SHOWING MULTIPLE CRACK INDICATIONS CLOSE TO THE OD SURFACE

Based on the analysis, the summarized critical crack results observed are given in Table 3.

TABLE 3

SUMMARY OF THE CRACK LOCATIONS

Shaft Diameter (mm)	Section thickness (mm) / Available thickness	No of indications	Depth (mm)	Circumferential Length (mm)	Axial length (mm)	Depth - Min/ Maximum
558	238	14	77	20	25	Min
	55		183	10	10	Max
610	258	46	40	15	10	Min
	34		224	110	30	Max
914	412	94	73	25	10	Min
	38		374	200	10	Max
737	319	21	51	10	10	Min
	49		270	110	10	Max

TABLE 4

MEASUREMENT OF CRACK LOCATION ON THE SHAFT

Shaft Diameter region (Ø, mm)	Indication No.	Location of discontinuity: Depth from OD (mm)	Location of discontinuity: From down streamline (mm)	Location of discontinuity: Clockwise / anti clock wise	Location of discontinuity: Frombolt-ing Flange (mm)	Circum-ferential length (mm)	Axial length (mm)	Equivalent flaw size (mm)
565	1	87.8	915	CW	170	37		
	2	162	1055	CW	185	7	10	2.2
	3	130	934	CW	255	38	30	2.3
	4	167	823	CW	330	9	20	2.4
	5	182	885	CW	440	15	45	4.8
	6	132	919	CW	515	13	25	2.5
	7	132	920	CW	560	13	25	3.5
	8	160	1255	CW	570	52	220	4.1
	9	217	1080	CW	452	8	30	2.9
	10	246	1090	CW	620	12	100	3.8
	11	180	790	CW	610	35	35	6.7
610	12	195	310	CW	1310	7	210	4.2
	13	214	25	CW	1550	14	40	4.1
	14	224	1820	CW	1620	63	30	5.7
	15	246	1595	CW	950	20	200	3.5
	16	205	120	CW	2120	13	120	3.5
	17	225	5	CW	2000	5	30	4.0
	18	225	30	CW	2130	11	100	2.5
	19	76	1855	CW	2210	22	35	2.5
	20	113	1745	CW	2220	40	25	2.8
	21	94	1715	CW	2110	65	25	3.5
	22	117	1585	CW	2060	10	85	3.3
	23	75	1595	CW	2220	22	25	2.0
	24	240	1125	CW	1590	13	55	2.0
	25	265	1025	CW	1600	5	40	3.2
	26	250	1065	CW	1880	8	160	3.8
	27	190	1215	CW	2090	30	90	3.0
	28	241	1325	CW	1030	6	75	3.5
	29	240	1285	CW	935	38	100	6.2
	30	160	815	CW	935	28	50	2.8
914	31	235	2700	CW	2600	24	200	4.1
737	32	192	2210	CW	3770	10	15	2.8
965	33	338	2650	CW	400	13	30	8.2
	34	370	2650	CW	320	25	40	6.9
	35	337	2320	CW	440	6	20	3.6
	36	346	1340	CW	370	15	30	5.0

i.e. (σ_{22}) due to self-weight of the shaft as well as the rotating mass of the generator is predominantly counter balanced by the upward thrust level of 370 tons. Under these conditions, maximum tensile stress in the vertical direction of the rotor σ_{22} is observed to be close to 26.53 MPa. (Figure 10). The calculated maximum torsional shear stresses at uprated conditions are observed to be 43 MPa in regions close to the bolting flange of the Generator-turbine shaft against the design value of 54.9 MPa.

6.3 Fatigue analysis of uncracked shaft

The Brown Milleralgorithm which is based on strain based approach, primarily used for the ductile material is used for the determination of fatigue life [6]. The fatigue analysis of the generator shaft was carried out under defect free situations at the uprated loading condition.

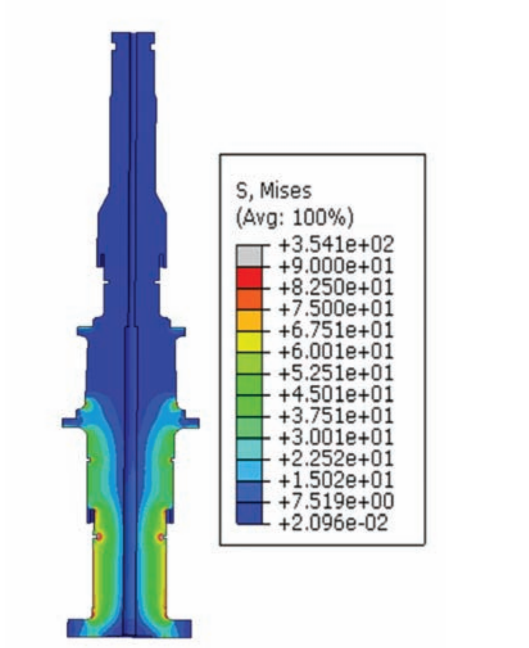

FIG. 10. VON MISSES STRESS ON SHAFT

The fatigue life of the shaft with no cracks was observed to be 1.0 x 10^{12} cycles and the fatigue analysis results are shown in Figure 12. From the plant operation data, it is found that the generator shaft has running hours of approx. 233180 hrs and which corresponds to 2.33x10^{9} cycles.

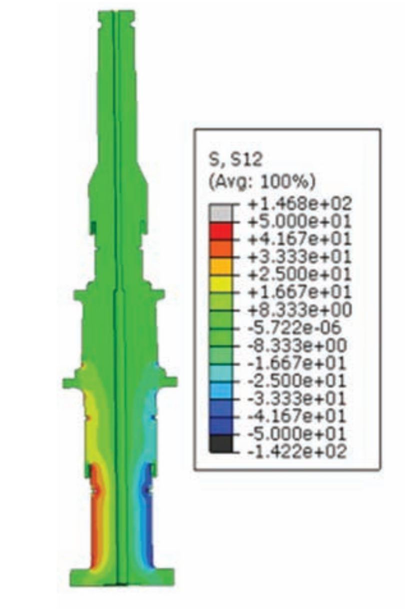

FIG. 11. TORSIONAL SHEAR STRESS PROFILE

Thus under defect free condition, the fatigue life would have been much more than 60 years. However, the actual life is limited by the number of cracks, size and their orientation as observed by the PAUT results.

The presence of cracks and other defects decreases the fatigue life. Further crack growth analysis is carried out to predict the life of the shaft in the presence of defects

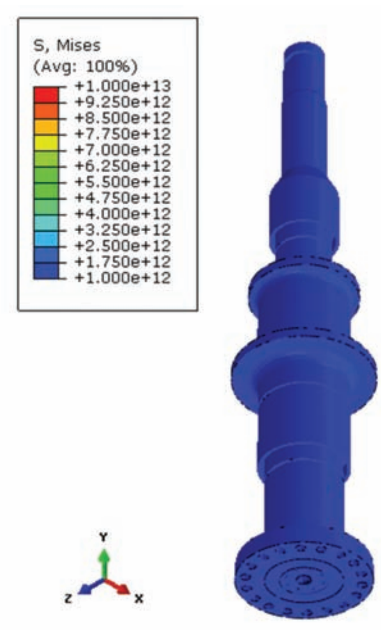

FIG. 12. FATIGUE LIFE CONTOUR OF THE GENERATOR SHAFT.

6.4 CRACK GROWTH RATE STUDIES

Out of 36 defects observed, seven defects which are of large in size and located close to the OD surface were considered for the crack growth analysis. The growth rate of these cracks under the uprated load conditions was monitored through finite element simulation, taking into account of the results of the dynamic stress analysis profiles. It is generally accepted that fatigue crack growth rate correlates with these verity of crack tip loading characterized by the elastic stress intensity factor, 'K' [7-8]. The fatigue crack growth rate has been predicted from the known stress intensity factor through the well established empirical relationships

$$\frac{da}{dN} = C(\Delta K)^n \quad \ldots(1)$$

where 'ΔK' is the change in 'K' during each fatigue cycle in MPa$\sqrt{m}$; da/dNis the increase in crack length in m/cycle; and 'C' and 'n' are experimentally derived constants. The constants used for the analysis is based on the yield strength properties of the rotor steel specification and the values are given in Table 3.

The following steps were followed to generate the crack in the Generator shaft.

Step 1: Replace elements with alarge crack block and define tie constraint between the surfaces.

Step 2: Define the crack size in ZENCRACK™ and generate the crack front mesh. Two corner crack blocks are introduced to generate a semicircular crack front

Step 3: Define fatigue loading cycle and run the simulation.

The crack blocks were inserted in appropriate diametral regions of the shaft. The crack growth analysis of the inserted cracks was monitored considering the load spectra obtained from the static analysis. Depending upon the local stress intensities experienced in each of the crack tips, the growth of crack takes places in preferential direction. The incremental growth of crack length (da) is estimated corresponding to the number of fatigue cycle (dN). The rate crack growth with cycles (da/dN) is analyzed for all the critical cracks considered.

Figure 13 shows the sum of incremental crack length da v/s Number of cycles under fatigue loading. It can be observed that the crack needs 25,000,000 cycles to reach a crack of length 0.28 mm. Considering the rpm of the generator as 166.7 rpm, then the crack will grow at ($25x10^6$/166.7) = 149970 min = 2499 hrs. So the crack will grow by 0.9826 mm per year. The same procedure is followed for all the seven cracks. The rate of growth of each of these cracks has been observed on a continuous basis up to $25x10^6$ cycles equivalent of 2500 hrs of operation. Extrapolation of the crack growth data was done. The analysis was continued for a maximum of 25 million cycles and the increase in crack length observed in each of the cracks is given in Table 5.

TABLE 5

SUMMARY OF CRACK GROWTH RATE DATA

	Crack 1	Crack 2	Crack 3	Crack 4	Crack 5	Crack 6	Crack 7
Initial crack size – Axial length (mm)	15	30	220	25	35	25	25
Depth of crack from OD (mm)	87.8	130	160	94	76	113	75
Increase in crack length in $25x10^6$ cycles (mm)	0.049	0.286	0.127	0.177	0.042	0.034	0.280

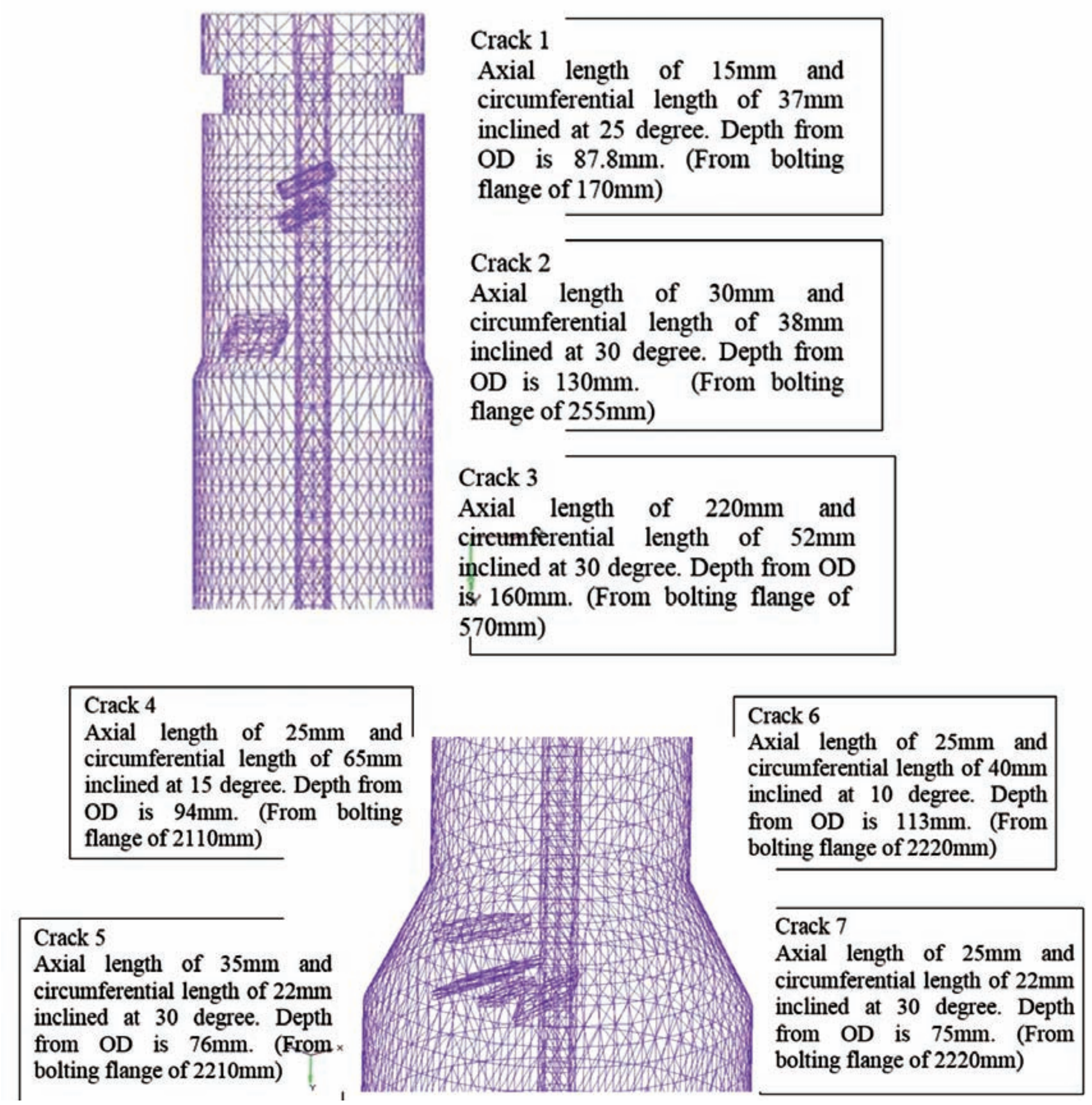

FIG. 12. INSERTION OF CRACK BLOCK ELEMENTS IN DIFFERENT DIAMETRAL REGION OF SHAFT

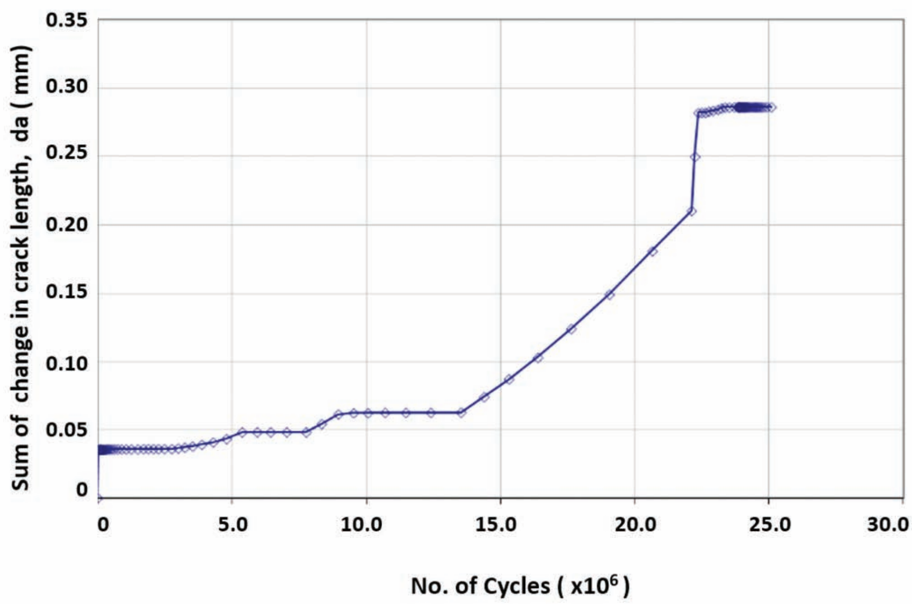

FIG. 13. SUM OF INCREMENTAL CRACK LENGTH V/S NUMBER OF CYCLES

Though the rate of growth of crack No. 7 included in 736 mm diameter region has been observed to be maximum, the depth of the crack from the surface is high (~70 mm). Thus for the crack to reach the surface, the no. of years of operation required is considered high (>10 years) .

7.0 CONCLUSIONS

The systematic study on the cracked rotor shaft through a combination of advanced PAUT inspection and crack growth rate analysis under the uprated loading conditions indicated the following.

- The Magnetic particle inspection at the roots of taper regions of the shaft as well as the Dye penetrant test all along the surfaces did not reveal any indications of surface and near surface defects.
- The PAUT technique has identified all the crack like defects present in the rotor shaft. Most of the cracks were observed to be close to the ID side of shaft and their orientation is observed to be slightly angular with respect to the shaft axis (less than 20°)
- The minimum depth to which the defect (crack tip) observed is close to 70 mm from the OD of the shaft. Thus any crack needs to grow 70 mm to reach the surface, which is considered the end of life.
- The stress analysis of the shaft, the area averaged maximum shear stress was observed to be approx. 39.3 MPa in 966 mm diameter regions close to Generator-Turbine bolting flange. The allowable design vale of shear stress provided by OEM is 54.9 MPa.
- The maximum von Mises stress under uprating conditions is observed to be 68.2 MPa against the design value of 91.5 MPa.
- The fatigue life of the generator shaft without any defect for the rated design capacity is 10^{12} cycles which is equivalent to life in excess of 75 years.
- The critical crack growth rate in cases of selected large size cracks studied was estimated to be 0.983 mm per year. The actual no. of service hours of the unit is approx. 233180 hours equalling to 2.33 x10^9 cycles. Keeping in view the critical crack location as 70 mm from OD, the crack is expected to reach the surface in not less than 10 years at the estimated growth rates.

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