Fault Diagnosis of Lakvijaya Power Plant: A Case Study of an Anti-Rotational Pin Failure

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Abstract-*Phase 1 of the Lakvijaya power plant has suffered an inherent vibration problem leading to its temporary shutdowns in 2013, 2014 and 2015. The purpose of this study is to present a hypothesis to explain causes of these vibrations, test the hypothesis and make suitable recommendations to avoid such occurrences in the future.*

When the shell of one of the bearings adjacent to the intermediate pressure turbine was opened for repairs in January 2015, it was observed that the anti-rotational pin of the labyrinth seal of the bearing has been broken causing the observed unusual vibrations. There are evidences to believe that this has happened in two previous occasions as well.

The visual inspection of the fracture surface of the anti-rotation pin revealed that it has been failed from fatigue failure. Further, examination of the images of the facture surface obtained using scanning electron microscope indicated that the fracture has originated from the outer surface of the pin and propagated to final fracture by fatigue failure. Another region of the scanning by electron microscope image showed that the pin had been subjected to heavy deflection as well.

This paper presents a hypothesis to explain how the anti-rotational pin of the labyrinth seal was failed and test the validity of the hypothesis using vibration data. Computations carried out to test the hypothesis show that the stresses inside the anti-rotational pin due to the forces exerted on it by the labyrinth seal can exceed the its fatigue strength causing its failure.

Keywords*-Labyrinth seal, Finite Element Analysis, 1x vibration and 2x, 3x vibrations, Cascade plot*

1. INTRODUCTION

The Phase 1 of the Lakvijaya power plant was temporarily shut down in 2013, 2014 and 2015 following a vibration problem. It has been discovered during the repairs in all three occasions that at least one anti-rotational pins of the labyrinth seals of the bearing number 3 or 4(bearings of the either side intermediate pressure turbine) of the plant has been broken causing the observed high level of vibrations.

A schematic diagram of the Lakvijaya plant showing the bearings and labyrinth seals has been given in Figure 1. The turbine shaft of the Lakvijaya power plant rests on six oil pressurized bearings [1] with labyrinth seals [2] fixed at their both ends to prevent the leakage of oil. High pressure oil is continuously supplied to the bearings and labyrinth seals. The shaft rotate at rate of 3000 rpm. Vibrations of the plant is continuously monitored using a real time vibration monitoring system which acquires data from two proximity sensors[3]. Which are placed at 45^{o} to the vertical. An accelerometer is also fixed on the top of the bearing .The proximity plots indicate relative movements of the shaft center with respect to the bearing center measured in μm.

The unusual vibration caused the tripping of the plant consisted of a high level of proximity plots with amplitudes exceeding 250 μm [4]. In addition to the proximity plots the plants electronic system is able to produce bode and cascade plots.

Further when referring to the cascade plot of the broken bearing it was found that the super synchronous [5] vibration of 300 MHz has been reported. This is the natural frequency of the bearing. This was verified by a modal test carried out by the CEB engineers.

It was hypothesized that the natural frequency (super synchronous) vibration occurrence was due to hammering of the labyrinth seal of the bearing to the bearing shell.

The main objective of the study was to provide a possible explanation for the repeated failure of the "anti-rotational pin" by studying the forces acting on the pin and analyzing the fracture surface of one of the broken pins.

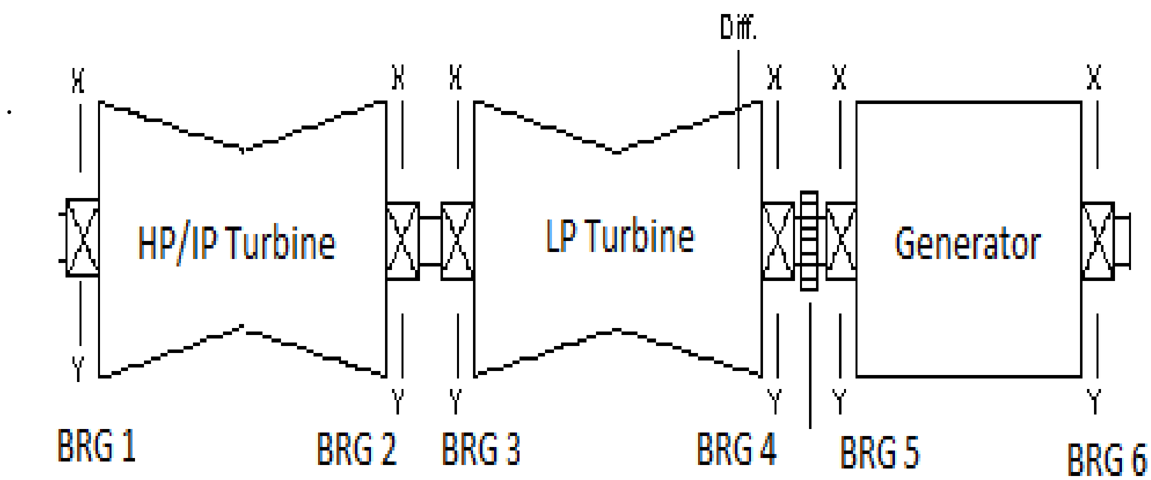

Figure 1: Arrangement of the Turbine Shaft of Lakvijaya Phase-1

2 PROPOSED EXPLANATION FOR THE FAILURE OF THE ANTI-ROTATIONAL PIN

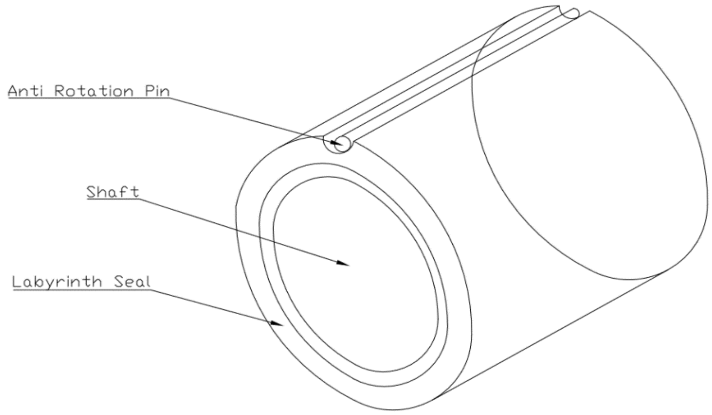

Figure 2: Orientation of the Labyrinth Seal and the Bearing

As illustrated in Figure 1 the shaft of the turbine is rested on eight bearings filled with pressurized oil and two labyrinth seals placed either side of each bearing to prevent leaking of oil to the environment. When the shaft is rotating, usually at a speed of 3000 rpm [4], the oil layers adjacent to it also rotates with the same speed and this motion is transmitted to adjacent layers finally applying a tangential thrust on the labyrinth seals. Movement of the seal due to this thrust is prevented by an anti-rotational pin which is placed in a grove and rests against the wall of the grove (Figure 2). When the seal attempts move the wall of the grove applies a thrust on the pin causing it to bend and hence the stresses on the pin get concentrated at two edges which are in contact with the wall. Periodic variation of the position of the shaft causes these stresses to fluctuate over a large range with the possibility of causing fatigue failure[6] due to the repeated variation of stresses.

3 METHODOLOGY

In order to understand the failure of the anti-rotational pin it is necessary to understand the nature of forces acting on the pin and how they arise.

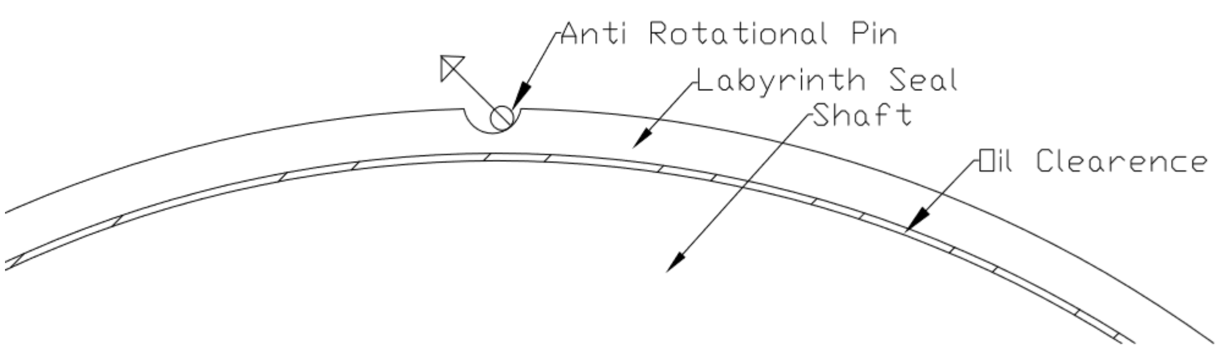

Figure 3: Types of Forces Applying on the Anti-Rotational Pin

There are mainly two forces acting on the labyrinth seal and the anti-rotational pin. The first is a shear load generated by the viscous action of the rotating turbine oil on labyrinth seal shaft due to the rotation of the turbine (Figure 3) while the second is the normal load in the form of pressure generated by the to and fro motion of the shaft. Since this motion is very small, the normal force generated by it is also small and can be neglected.

As the shaft moves the fluid adjacent to it also moves applying a viscous force on the labyrinth seal forcing it to rotate. However, it will be stopped by the anti-rotational pin and in the process, the anti-rotational pin will be pressed against one of the sides of the pin groove experiencing a force of large magnitude. When the shaft moves to and fro, the distance between the shaft and the seal also changes changing the velocity gradient of the oil causing the viscous force to fluctuate. This in turn causes fluctuations of the force acting on the anti-rotational pin.

As explained earlier, tangential viscous drag on the labyrinth seal originates due to the rotation of the shaft in the viscous oil. Velocity of the outer surface of the shaft can be calculated. Assuming the oil layer adjacent to the shaft rotates with the same velocity and that adjacent to the labyrinth seal is stationary, velocity gradient and the viscous force on the labyrinth seal can be calculated. It should be noted that for a given position of the centre of the shaft the gap between the surface of the shaft and inner surface of the labyrinth seal is not a constant and hence the viscous drag also varies along the seal. The following method was adopted to calculate the total viscous drag on the seal.

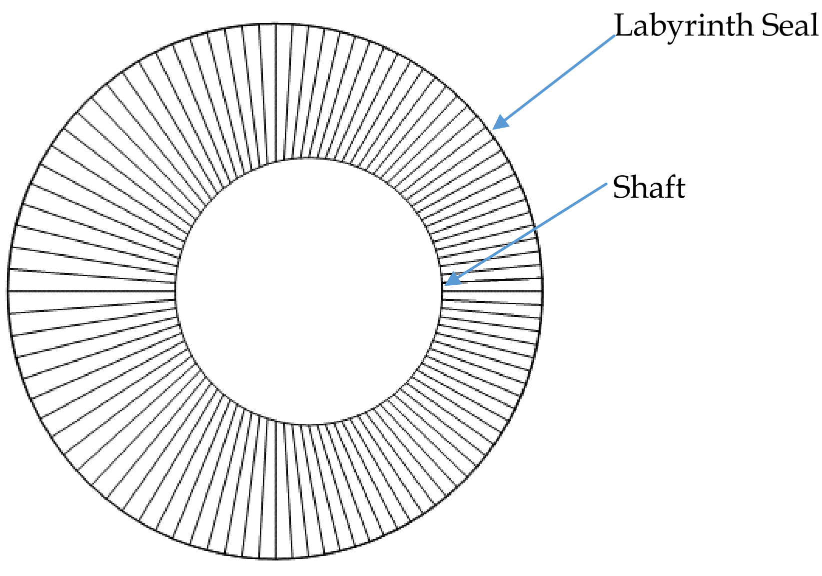

Figure 4: Division of the space between the labyrinth seal and the shaft for the calculation of the viscous drag on the labyrinth seal

In the Figure 4 the labyrinth seal is repressed by the outer circle while the inner circle represents surface of the shaft. When the centers of the both coincide the gap between the two surfaces is a constant. However, usually when the shaft is rotating, its centre moves to and fro making this gap a variable. To calculate the total tangential viscous drag on the seal for a given position of the centre of the shaft, the gap between the two surfaces were divided into 100 segments and it was assumed over each segment the gap is a constant and hence viscous force is also a constant. Since the gap between two surfaces is very small, this assumption can be justified easily. The average gap of each segment is calculated considering the geometry of the figure using a computer routine written employing the "Mathematica" software package. Then the viscous drag on the seal along each segment was separately calculated and added together to give total viscous drag.

This tangential force makes the seal including the anti-rotational pin to move around and it will be stopped when the pin rest against the wall of the cylindrical grove (Figure 3). Now there are two forces acting on the seal; tangential viscous force and the thrust on the pin of the seal by the wall of the grove. The thrust was calculated resolving forces taking into the fact that the pin is cylindrical and the thrust on the pin is acting along the radial direction of the cylinder.

Now if we consider the pin alone it is subjected to a uniform thrust in the radial direction throughout its length. After determining the radial thrust on the pin a finite element analysis of the pin was carried out. As explained earlier initially the pin is subjected to a continuous thrust by the wall of the grove causing it to bend. The bending of the pin was modeled using the finite element method by the Lisa open source software. When the pin is bended only the edges of the pin are in contact with the wall of the grove and it was assumed contact region in each end is of 1 mm length. In fact this length could be much smaller than this value. Bending of the pin and the stress distribution under this new pressure distribution was also calculated.

4. RESULTS AND DISCUSSION

As explained earlier, tangential viscous drag on the labyrinth seal originates due to the rotation of the shaft in the viscous oil. Considering the values of the radius of the shaft and its angular speed obtained from turbine manual, velocity of the outer surface of the shaft was calculated and found to be157.07 ms^{-1}. Assuming the oil layer adjacent to the shaft rotates with the same speed and that adjacent to the labyrinth seal is stationary, velocity gradient and the viscous force on the labyrinth seal were calculated for different gaps between the shaft surface and the labyrinth seal inner surface.

The above mentioned gaps were selected based on the readings of the two proximity plots which gives the relative displacements of the centre of the shaft along two perpendicular directions (X and Y). Proximity plots give the relative displacements the shaft centre. Typical amplitudes of relative displacement of the centre of the shaft recorded by the X and Y sensors are 110 μm and 120 μm respectively. A relative displacement of 110μm on the X sensor can be produced due to the real displacement of the shaft along the X direction over several ranges of displacements. For the purpose of this calculation ranges 0-110 μm, 110-220 μm, 220-330μm were considered for the displacement along X-direction. Similarly the ranges 0-120 μm, 120-240 μm, 240-360 μm were considered along the Y direction. Table 1 and 2give values of the thrust applied on the edges of the pin for the three ranges along X and Y directions respectively.

Table 1: Variation of the Stress in X Direction

Displacement of the Shaft Center in X Direction	Thrust on the Edges of the Pin(MPa)
110 μm	1110
220 μm	1283
330 μm	1890

Table 2: Variation of the Stress in Y Direction

Displacement of the Shaft Center in X Direction	Thrust on the Edges of the Pin (MPa)
120 μm	1130
240 μm	1250
360 μm	1780

It is expected to use the "Von Misses" stress criterion[7] to examine the possibility of failure of the pin under the applied pressures at its ends. Therefore the "Von Misses" stress distribution of the pin was calculated under the above pressures applied on the pin at its ends using the finite element method. It was observed that the maximum "Von Misses" stress levels occur at the ends of the pin.Table-3 shows the magnitude of the maximum "Von Misses" stress occurs when the shaft center displaces 110μm, 220μm and 330 μm from the center of the labyrinth seal along the X-direction.

Table-4 shows the magnitude of maximum "Von Misses" stress occurs when the shaft center displaces 120μm, 240μm and 360 μm from the center of the labyrinth seal along the Y-direction

Table 3: Variation of the Internal Stress with Thrust on the Pin (X direction)

Shaft Center Displacement in X direction	External Pressure Applied on the Ends of the Pin (MPa)	Maximum "Von Misses" Stress (MPa)
110 μm	1110	1332
330 μm	1890	2270
220 μm	1283	1543

Table 4: Variation of the Internal Stress with Thrust on the Pin (Y direction)

Shaft Center Displacement in Y direction	**External Pressure on the Edge of the Pin (MPa)**	**Maximum "Von Misses" Stress (MPa)**
120 µm	1130	1333
240 µm	1750	2080
360 µm	1250	1503

As can be seen from the Table 3 when the position of the shaft center varies from 220µm and 330 µmin X direction the maximum Von Misses" stress in the pin varies between 2270 MPa and 1543 MPa. This variation exceeds the endurance limit [8] of 400 MPa of the material of the anti-rotation pin. This situation also occurs when the centre of the shaft moves from 240 µm to 360 µmin Y direction as the corresponding variation of the stresses is 577 MPa exceeding the endurance limit(Table 4). Based on these results we can suggest that it is possible for the anti-rotational pin to undergo fatigue failure as a result of variation of pressure applied at its ends originated from the viscous drag on the labyrinth seal.

The above suggestion can be further strengthen using the scanning electron microscope images of the fracture surface of the pin. Scanning electron microscope images of the fracture surface of the pin are shown in Figures 5 to 8. These images show characteristic features that can be seen in an image of a surface formed as a result of a fatigue failure. They show the presence of beach marks indicating propagation of the fatigue failure and granular surface indicating occurrence of final fracture of the pin. It can be inferred from the Figure 5 that the crack initiation has taken place at three sites labeled as 1, 2 and 3. A 48x magnification of the location-1 is shown in Figure 6. The direction of propagation cracks have been illustrated by the white arrows.

Figure 6 shows continuation of the fatigue failure initiated at location –1.The white arrows in Figure 6 shows the propagation of the crack in white arrows. Figure 7 also shows the location-2 where the crack initiated. Figure 8 shows the beach marks initiated near the location-3 leading to the final fracture resulting a granular surface. Therefore it can be concluded based on the electron microscope images that the pin has been failed due to the "fatigue failure".

Considering the calculations presented in relation to the "Von Misses" stresses subjected the pin and electron microscope images it is concluded that the anti-rotational pin has undergone fatigue failure as a result of variation of pressure applied at its ends originated from the viscous drag on the labyrinth seal as hypothesized in this study.

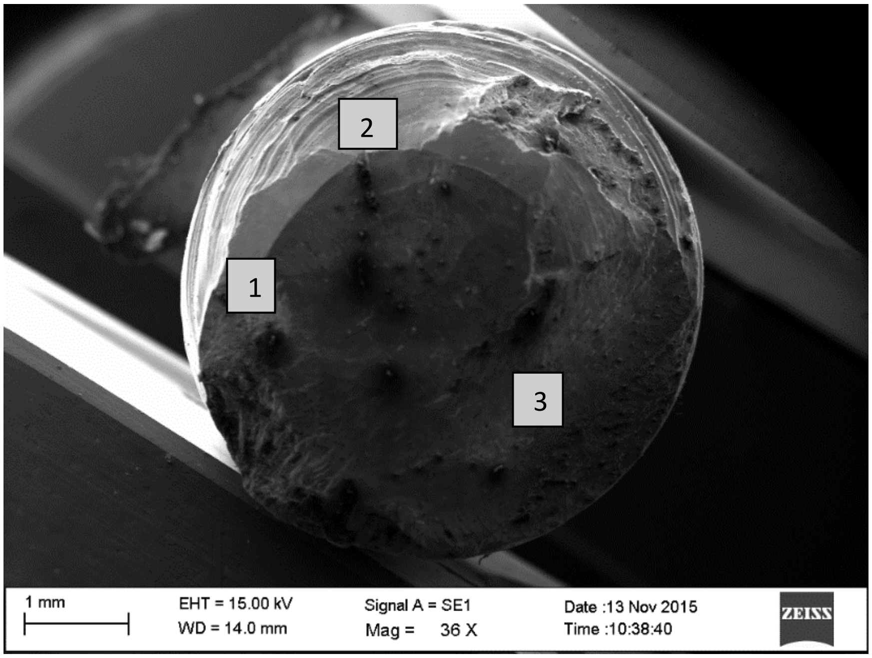

Figure 5: Scanning Electron Microscope Image of the surface of the broken Anti-Rotation Pin" showing identified Locations (1, 2 and 3) for further study

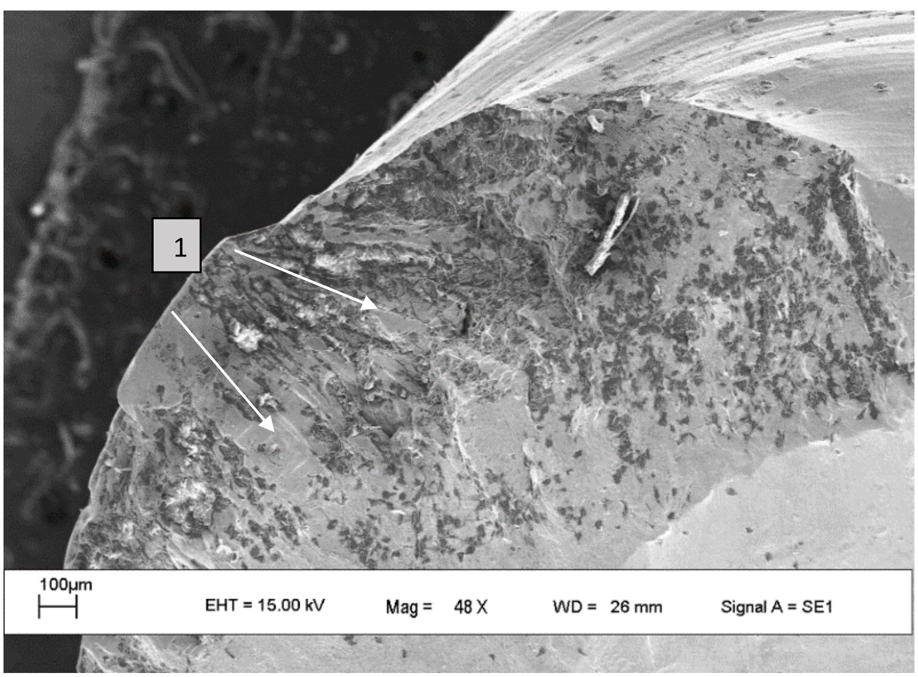

Figure 6: Scanning Electron Microscope Image of the surface of the Pin at Location-1

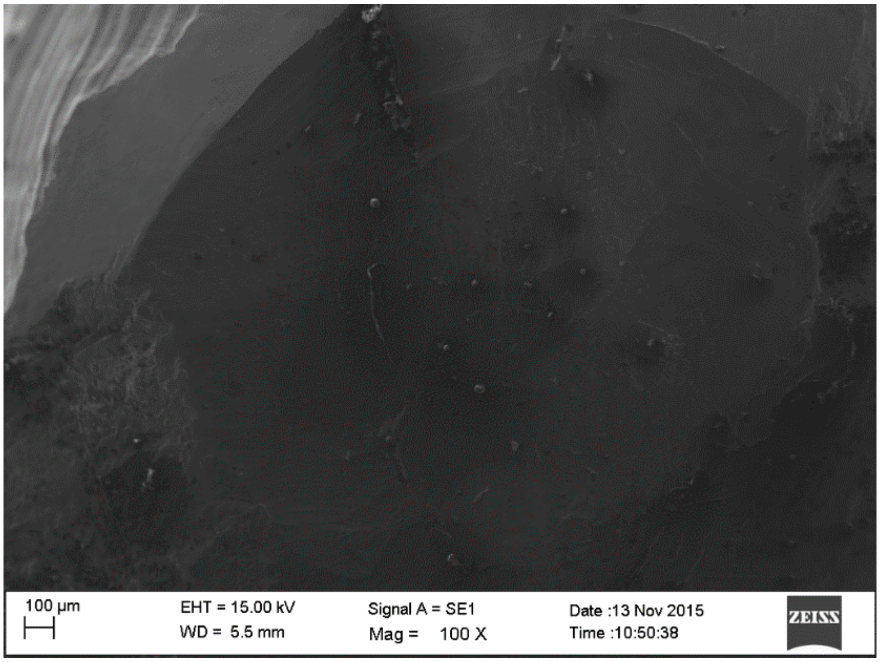

Figure 7: Scanning Electron Microscope Image of the surface of the Pin at Location-1(cont.)

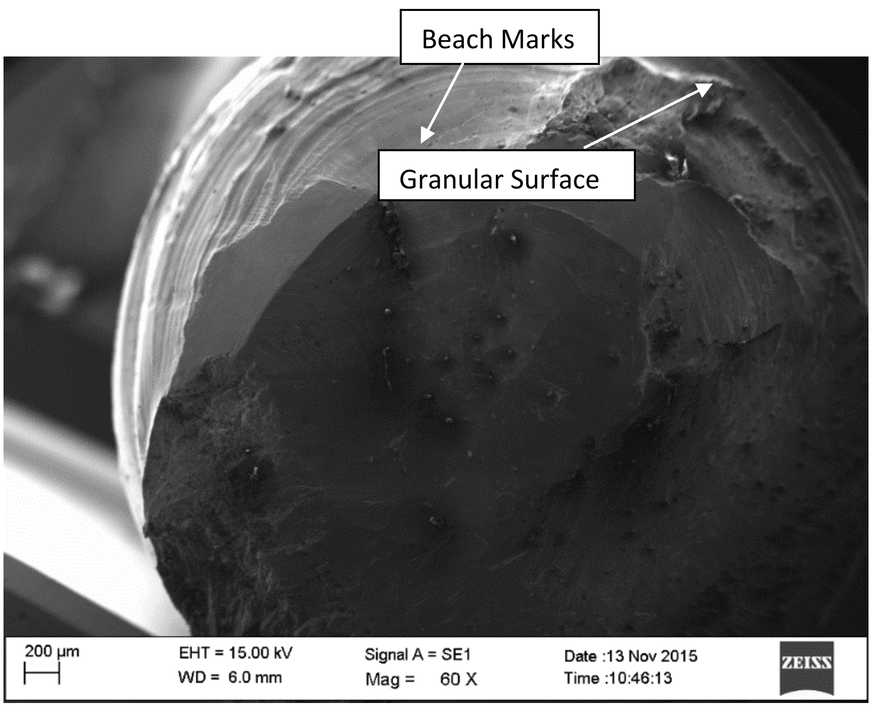

Figure 8: Scanning Electron Microscopic Image of the surface of the Pin at Locations-2 and 3

CONCLUSION

A possible reason for breaking down of the anti-rotational can be explained as follows. When the labyrinth seal tries to rotate, it exerts a pressure on the anti-rotational pin which attempts to stop this rotation. This pressure causes the pin to deform. Inspection of proximity plots revealed that the relative position of the shaft center undergoes periodic displacements causing the gap between labyrinth seal and the shaft periphery to vary. This in turn changes the viscous force acting on labyrinth seal and hence the pressure acting on the anti-rotational pin.

Therefore it can be concluded that the "anti-rotational" pin has subjected to the "Fatigue Failure" and the pin's diameter is not sufficient to the stress variation as the shaft position varies.

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