TILTING PAD JOURNAL BEARING REVERSE ENGINEERING AND FAILURE ROOT CAUSE ANALYSIS FOR PELTON HYDRO TURBINE

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Abstract: Pelton turbines though they operate at lower speeds experiencing a high-water jet load and varied direction of the load, their dynamic behavior is of greater importance especially considering the bearing life. A major overhaul was performed on a Pelton turbine which was first commissioned in the late 90's. During operation after overhaul, it was observed the temperature of the turbine guide journal bearing was shooting above 130C when both nozzles are in operation however with a single nozzle the turbine bearing temperature was within acceptable limits. Since the bearings were damaged it needs to be re-babitted. The present work is about reverse engineering and failure root cause analysis of the bearing for a Pelton turbine with 2 nozzles producing a 7 MW of power. Since there was limited information available for the bearings, a physical measurement was done to reverse engineer the bearings. Various parametric studies were performed in AxSTREAM Bearings™ and AxSTREAM RotorDynamics™ to study the influence of different factors on the turbine guide bearing performance to find out possible sources of non-reliable operation. Sensitivity studies were performed on bearing geometric parameters to determine the hydrodynamic characteristics under varying operating loads of the turbine. Rotordynamics analysis was performed to determine the vibrational characteristics for full and part load operation. The results of the sensitivity studies, the influence of misalignments, the direction of force applied, etc. are presented in this work. The results of the study helped to fix the issue with the turbine guide bearing.

Keywords: AxSTREAM; Journal Bearings; Pelton Turbine; Rotordynamics; Vibration;

Nomenclature:

K_{xx}, K_{xy}, K_{yx}, K_{yy}, K_{zz} – stiffness coefficients in (x – horizontal, y – vertical, z – axial);
C_{xx}, C_{xy}, C_{yx}, C_{yy}, C_{zz} –damping coefficients in (x – horizontal, y – vertical, z – axial);
TGB – turbine guide bearing;
NDE – Non-drive End;
H_{min} – minimal oil film thickness (MOFT);
N_{fr} – friction power losses;
Q_s – oil flow;
β – rotor attitude angle.
DE – drive end;
T_in – oil inlet temperature;
P_{max} – maximal oil pressure;
F_r – friction coefficient;
E – eccentricity ratio;

Introduction: Every rotating assembly in rotating machinery needs to be separated from the stationery part by the appropriate type of bearings. The bearing allows the surface of the rotating shaft to easily slide relative to the stationary part with a thin layer of lubricant, which forms an oil wedge between them. The bearing surfaces are subjected to wear especially in the case when the bearing design is not optimal with respect to minimal oil film thickness, heat exchange conditions, and friction forces. In large turbomachinery such as hydro-electric turbines and other electric power generators, oil film hydrodynamic journal bearings are used to separate the rotating and stationary parts. As the shaft begins to rotate, pressure begins to build in the lubricant causing it to lift the shaft and transfer the load from the journal to the bearing shell through the lubricant as shown in Figure 1. The changes in the hydrodynamic properties of the lubricant, speed of the shaft, or load result in increasing or decreasing the separation distance between the rotating and stationary surfaces. As the shaft speed increase, the lubricant pressure reaches a point where the surfaces no longer make contact with each other and the load is completely shifted to the lubricant.

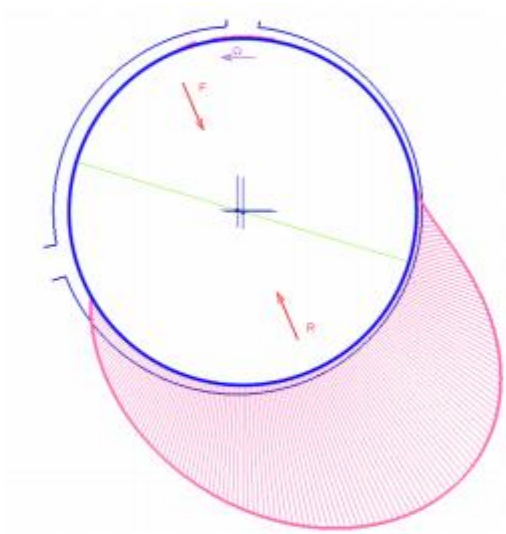


Figure 1: Hydrodynamic pressure profile in the Journal bearing

The change in clearance between the shaft and the bearing surface is affected by the hydrodynamic characteristics of the bearing system. Reduced operating clearances resulting in rubbing or even contact between the surfaces which would lead to faster wear, an increase of temperature, and damage to the bearing surfaces. The journal bearings can either be of plain cylindrical shape or with pads. The pads may be fixed type or tilted type depending on various factors.

One of the Pelton turbines with a rated capacity of 7MW, installed at a site in Malaysia after a major overhaul was experiencing higher bearing temperature even though the vibration was within limits. The said turbine has two nozzles and was operating without any issues until this overhaul. Since there was wear on the bearings, the bearings were re-babitted during the overhaul. As no original drawings of the bearings were available, the design was recreated based on the shaft dimensions by the maintenance team and reinstalled. During operation, it was observed that when the load was increased higher than 50% the temperature was shooting up beyond 130C whereas at a half load the temperature was within limits.

The present work is about reverse engineering and failure root cause analysis of the oil film hydrodynamic tilting pad bearing for the Pelton turbine. Since there was limited information available for the bearings, a physical measurement was done to reverse engineer the bearings. Various parametric studies were performed in AxSTREAM Bearings™ and AxSTREAM RotorDynamics™ software to study the influence of different factors on the turbine guide bearing performance to find out possible sources of non-reliable operation. Sensitivity studies were performed on bearing geometric parameters to determine the hydrodynamic characteristics under varying operating loads of the turbine. Rotordynamics analysis was performed to determine the vibrational characteristics for full and part load operation. The results of the sensitivity studies, the influence of misalignments, the direction of force applied, etc. are presented in this work. The results of the study helped to fix the issue with the turbine guide bearing.

Methodology: Prior to the bearings / rotordynamics simulations, potential reasons for the TGB non-reliable operation were analyzed. Table I, below shows the potential sources which may result in the bearing issue and actions which shall be conducted to check the probability of the critical influence of the corresponding source.

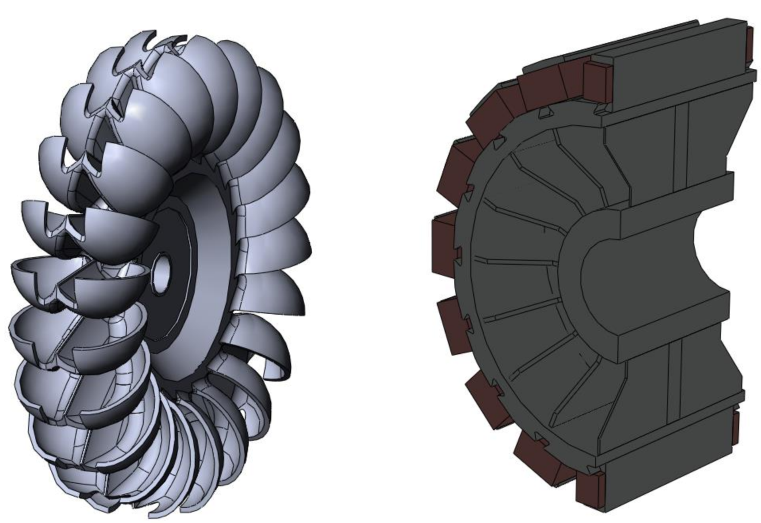
Table I: Potential Reasons for the TGB Bearing Non-Reliable Operation.

N	Potential reasons for TGB non-reliable operation	Actions
1	TGB incorrect geometry.	Check bearing dimensions, clearances, preloads, offsets of the existing machine. TGB sensitivity study to check the influence of bearing geometrical parameters on hydrodynamic characteristics.
2	TGB load change or direction of the load acting on the bearing.	TGB sensitivity study to check the influence of load increase and direction of applied force.
3	TGB (NDE & DE) generator bearings misalignment compensation pins (wrong dimensions, dirt)	Check NDE & DE bearings misalignment compensation pins. TGB sensitivity study to check the influence of load increase and direction of applied force.
4	Electrical pitting and shaft erosion due to electrical pitting.	Analyze TGB hydrodynamic parameters at full load and half load (lower nozzle operation). Check/confirm that the issue with electrical pitting was resolved.
5	TGB lubrication: not a proper operation of the oil supply system (not enough oil flow).	Check the oil level in the oil bath; oil cooling (water flow rate/pressure).

As the first step of the work, the TGB hydrodynamics sensitivity studies were performed to understand which factors and geometrical parameters deviations may result in bearing improper operation. Bearing clearance, inlet oil temperature, preloads, and offset of the bearing pads, bearing load and direction were some of the selected parameters that were varied to analyze the influence of these parameters on the mechanical and hydrodynamic characteristics of the TGB bearings. The effect of these parameters was used to determine the data required for performing the rotordynamics analysis for different operational conditions of the hydro turbine.

The bearing simulations are performed in AxSTREAM Bearing™ software, which allows determining hydrodynamic and mechanical characteristics for hydrodynamic journal bearings. The methodology for the bearing characteristics simulation is based on the mass-conserving mathematical model, proposed by Elrod & Adams [1], which is well-established as the accurate tool for simulation in hydrodynamic lubrication including cavitation. The Elrod & Adams model is expanded with heat conduction equations. Additional modifications to account turbulent flow regime and compressible fluid are incorporated in the solver. The numerical solution of the Elrod & Adams equations is generated using the Finite Difference Method with successive over-relaxation (SOR) algorithm. The detailed description of the methodology for a bearing simulation is given in [2] and [3]. Additional rotordynamics analysis for the Hydro Turbine to check the level of vibration for the machine was performed in AxSTREAM RotorDynamics™ software, which is based on finite element methodology and allows to perform the full scope of rotor dynamics analyses.

Simulation and Results: The finite element model of the hydro turbine was created to determine the load acting on the bearings. The 3D CAD model of the turbine and generator (Figure 2) was created based on the provided drawings to determine the mass-inertia characteristics of the attached to the shaft equipment and develop FE model for rotordynamic analysis.



a) *b)*

Figure 2: Components CAD Models: (a) Runner; (b) Generator Core.

Rotordynamics model for the Hydro Turbine is presented in Figure 3. The shafts were modeled as a series of Timoshenko beam elements with 4 degrees of freedom per node (horizontal and vertical displacements and rotations). The Z-axis is aligned with the rotor rotation longitudinal axis. The mass and stiffness of each element were calculated based on the material properties assigned. The shaft line on the upper half represents the mass diameters for each section and the bottom shaft line represents the stiffness diameters. All rotor components assembled to the shaft (runner, generator core) were modeled as lumped masses with moments of inertia in order to consider the gyroscopic effect in the rotordynamics analysis. Polar and diametral moments of inertia for runner and generator core were calculated based on CAD models.

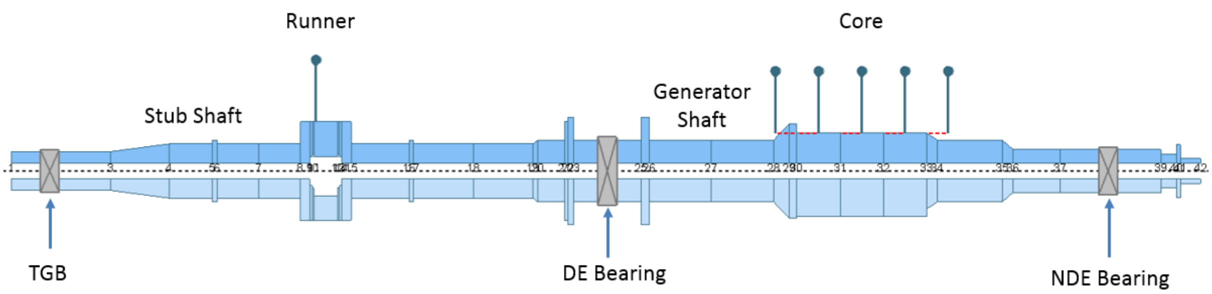


Figure 3: Hydro Turbine FE Rotordynamics Model.

Bearing Load Simulation: Hydro turbine rotor static deflection analysis was performed to obtain the loads acting on each bearing. The rotor weight and forces exerted by the water jet on the disk were considered as loads. Water jet forces were calculated for a full load – 7.5MW (2 nozzles) and half load– 3.5MW (lower nozzle operation) according to the scheme presented in Figure 4 and equation (1) and (2).

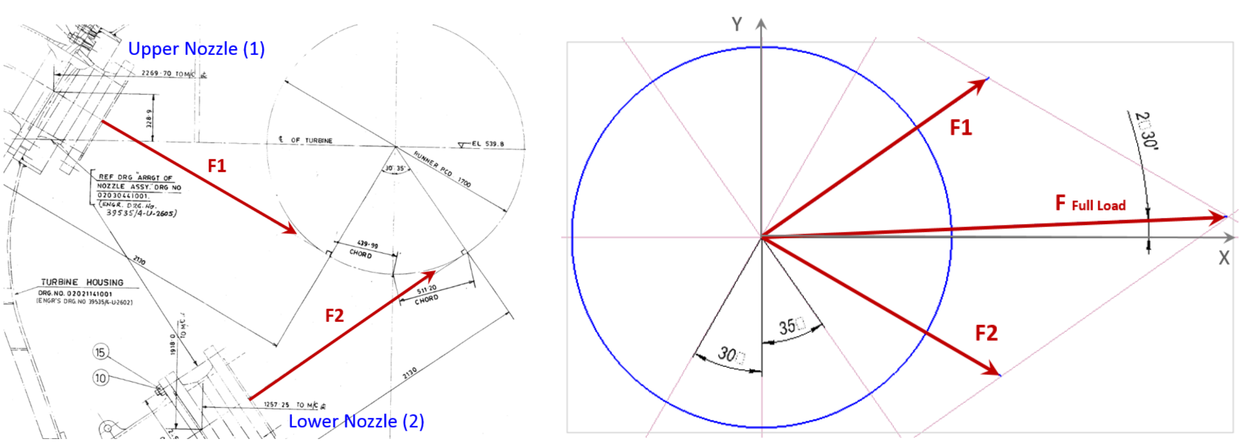


Figure 4: Water Jet Force Calculation

For the case of full load condition:

$$M_t = \frac{N_{FP}}{\Omega}; \; F \quad = \frac{N_{FP}}{R}; \; F_1 = F_2 = \frac{F}{2}; \tag{1}$$

$$\vec{F}_{FL} = \vec{F}_1 + \vec{F}_2, \tag{2}$$

where M_t – hydro turbine torque moment; N_{FP} – hydro turbine power; R – radius of applied force (Runner radius); F – total force; F_1 – a force from the 1st nozzle; the F_2 – a force from the 2nd nozzle; $\overrightarrow{F_{FL}}$ – actual force, which acts on the runner at full load conditions, calculated as a vector sum.

The calculated components of jet forces acting on the runner are presented in Table II.

Table II: Calculated Water Jet Force Components.

Parameter	Force	Units	Value
Full Load 7.5 MW	F_{FL}	N	189501
x-component	FX_{FL}	N	189321
y-component	FY_{FL}	N	8266
Half Load 3.5 MW (Lower Nozzle)	$F_{HL3.5}$	N	104855
x-component	$FX_{HL3.5}$	N	85892
y-component	$FY_{HL3.5}$	N	60142

Static deflection analysis was performed for the hydro turbine rotor for three conditions: no load, half load (3.5MW), and full load (7.5MW). Calculated bearing reaction forces for TGB, DE, and NDE generator bearings are presented in Table III.

Table III: Bearings reaction forces

	No Load		3.5MW Load		7.5MW Load	
	Rx	Ry	Rx	Ry	Rx	Ry
	N	N	N	N	N	N
TGB	0	18684	-34239	-5290	-75468	15389
DE	0	193029	-63791	148362	-140607	186890
NDE	0	95110	12138	103609	26753	96278

Hydro Turbine rotor deflection forms for the considered cases are presented in Figure 5.

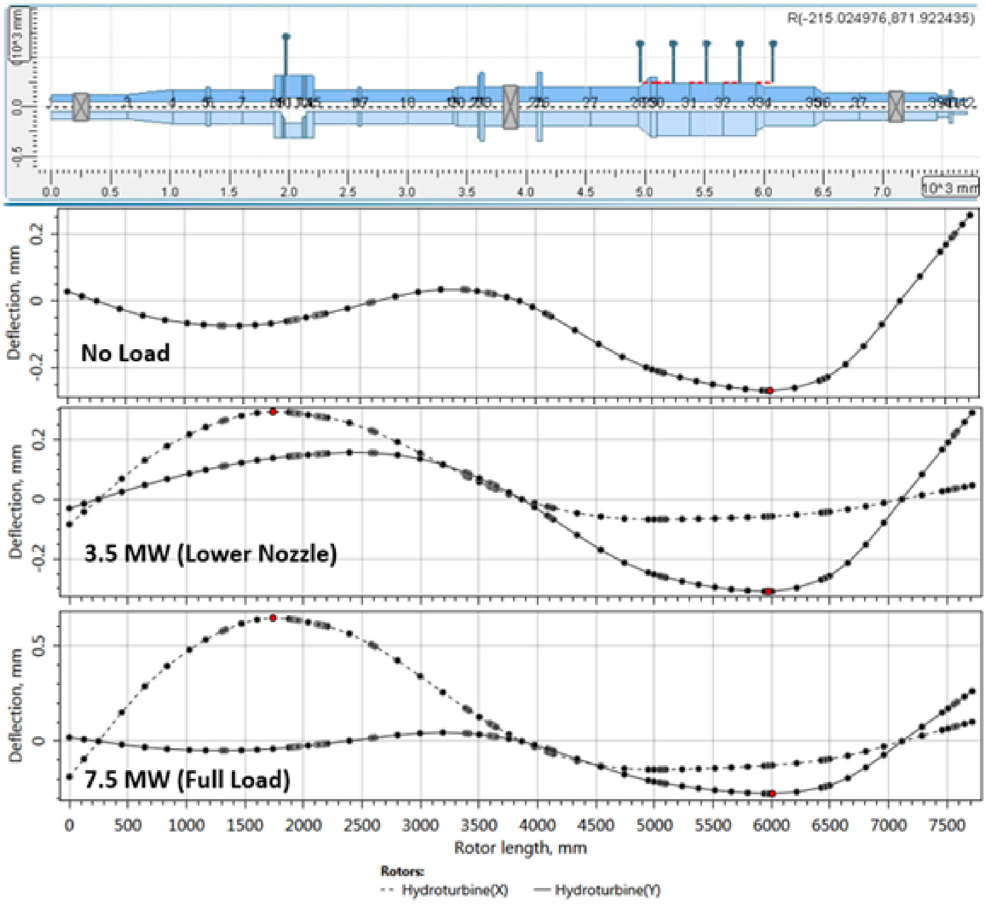


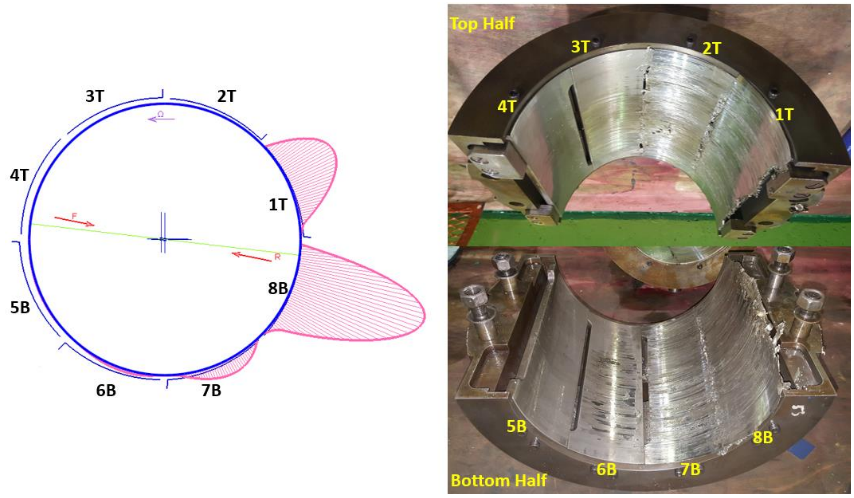
Figure 5: Hydro Turbine Deflection Form under static gravity for different loads

Bearings Sensitivity Study: Turbine guide tilting pad bearing (TGB) parameters, summarized from drawings and scanned data, are presented in Table III. TGB model has been created in AxSTREAM Bearing™ software.

Table III: TGB Initial Data

Bearing Parameters	Symbol	Units	Value
Bearing Length	L	mm	270
Shaft Diameter	D	mm	250
Clearance (Diametrical) Min	C_{bd} (Min)	mm	0.38
Clearance (Diametrical) Max	C_{bd} (Max)	mm	0.57
Pads Number		-	8
Pad angle	d_Alpha	deg	43.47
Offset angle	Alpha_off	deg	25.15
Offset (Offset=Alpha_off/d_Alpha)	Offset	-	0.58
Pads preload	Preload	-	0.568
Oil Grade	ISO VG 68 / Shell T 68		
Shaft Speed, Normal	Omega	rpm	375

Pressure distribution profiles for the TGB simulation results are in good agreement with TGB failure incident consequences picture – see Figure 6. The highest load according to simulation results falls on pad '8B' and pad '1T' (Figure 8a). The most damages on the pads in the TGB failure incident pictures are the same – '8B' and '1T'. This result confirms that the load direction is not changed and can't be the reason for the TGB failure.



a) *b)*

Figure 6: Comparison of calculated results of pressure distribution profiles (a) with TGB Failure Incident Picture (b).

Simulation results for the TGB at full load (7.5MW) for minimum and maximum clearances for varying preload (from 0 to 0.89) for three different oil temperatures of 45, 50, and 57 °C are presented in Figure 7.

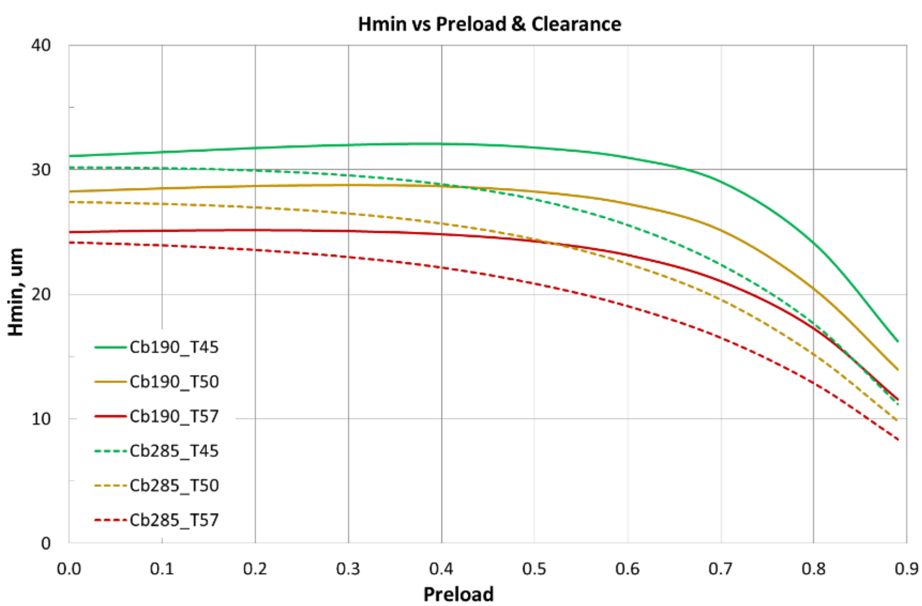


Figure 7: Minimum oil film Thickness vs TGB Preload & Clearance (Full Load Operation).

The recommendation for the value of minimum oil film thickness (MOFT) is to have MOFT > 20μm. The analysis results show that the highest values of minimum oil film thickness (MOFT) for the tilting pad TGB at full load operation correspond to preload 0 – 0.6 and decrease with higher preload. The MOFT takes unacceptable values with temperature increase (to 57°C) and preloads higher than 0.7 for the case with maximal clearance.

Figure 8 shows a variation of minimum oil film thickness versus tilting pad bearing offsett. From the simulation, it is observed that optimal values of TGB pads offset are 0.58 – 0.6 provide maximum value of MOFT and correspond to actual TGB parameters (offset=0.58).

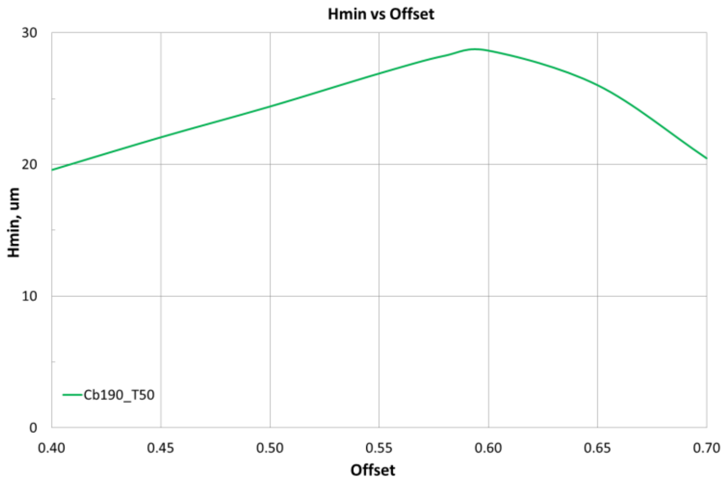


Figure 8: Minimum oil film Thickness vs TGB Offsett (Full Load Operation).

Table IV, shows the variation of MOFT for different clearances along with the stiffness and damping coefficient. The results show that for the bearing radial clearance 325 μm the MOFT=22.9 μm and still have an acceptable level (>20 μm). The calculated bearing stiffness and damping characteristics vs. rotor speed will be used in further steps of rotor dynamics analysis for the Hydro Turbine.

Table IV: TGB Simulation Results for Different Radial Clearances.

Clearance	**Hmin**	**Pmax**	**Nfr**	**Fr**	**ΔT**	**β**	**E**	**Kxx**	**Kxy**	**Kyx**	**Kyy**	**Cxx**	**Cxy**	**Cyx**	**Cyy**
um	um	MPa	W	-	C	deg	-	N/um				N*s/um			
190	28.3	5.2	1492	0.0039	11.9	353.3	0.83	3314	-614	-635	599	57	-8	-8	10
285	24.4	6.4	1329	0.0035	9.4	354.4	0.91	3405	-803	-812	513	55	-9	-9	7
325	22.9	6.8	1297	0.0034	8.5	355.1	0.93	3424	-835	-847	497	51	-9	-9	7

Hydro Turbine Lateral Rotor Dynamics: The purpose of the study is to present the rotor dynamic lateral analysis for the hydro turbine and prove that the rotor-bearing system has no issues with dynamics which may affect bearing life. Rotor model was developed according to API 684 practices [4] using data from drawings and CAD models. The rotor FE model which was used in the analysis is presented in Figure 3.
Critical Speed Map analysis has been performed to analyze potential resonance conditions for the Hydro Turbine rotor system and is presented in Figure 9 for the case of full load conditions and it shows that resonance conditions are avoided for the considered rotor-bearing system.

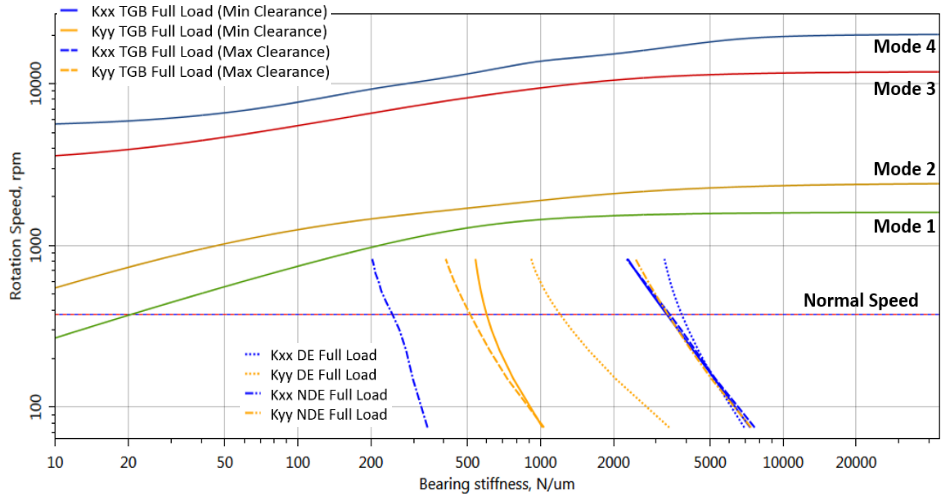


Figure 9: Critical Speed Map for the Hydro Turbine Rotor (Full Load).

Vibration modes which are the result of undamped analysis are presented in Figure 10.

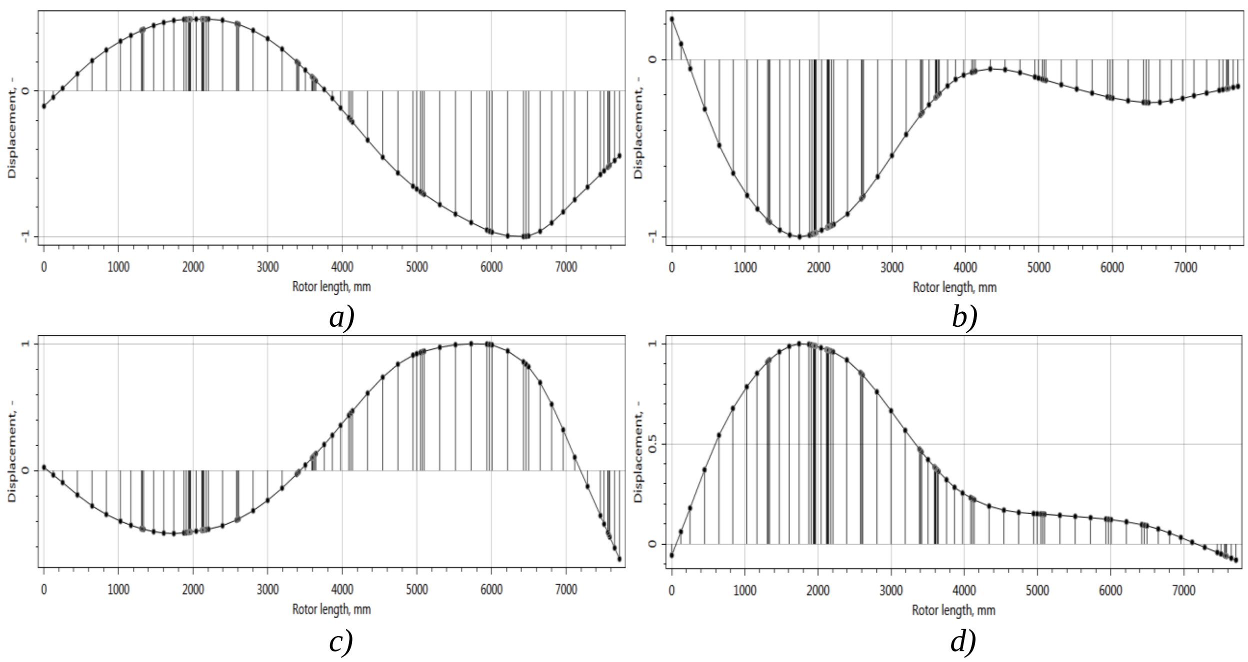


Figure 10: Mode Shapes: (a) 1st Mode, X; (b) 2nd Mode, X; (c) 1st Mode, Y; (d) 2nd Mode, Y.

Table V,**Table** summarizes the critical speeds analysis results and the first critical speed is much higher than the nominal operating speed.

Table V: Undamped Rotor System Critical Speeds.

Mode	Minimal clearance	Maximal clearance
1st Mode, X	1249.0	1248.9
2nd Mode, X	2136.7	2136.4
1st Mode, Y	1485.7	1483.5
2nd Mode, Y	1847.3	1827.4

The unbalance response analysis predicts the rotor displacements for a given unbalance condition. API 684 SP6.8.2.7 requires to perform a separate damped unbalance response analysis for each critical speed within speed range 0-125% trip speed. For the present case, all critical speeds are higher 125% of trip speed, and damped unbalance response is not required. However, the damped unbalance response has been performed to check the level of rotor vibrations at full load operation. The unbalances were calculated following standard paragraphs recommendations of API 684. The first and second mode shape with the lowest frequencies was excited to check the rotor response. The unbalance amount according to API corresponds to Grade 0.7 ISO [4] Standard residual unbalance level. For the ISO specification, turbomachinery rotors are assigned the Grade of 2.5. Figure 11 shows the applied unbalances to the hydro turbine rotor system.

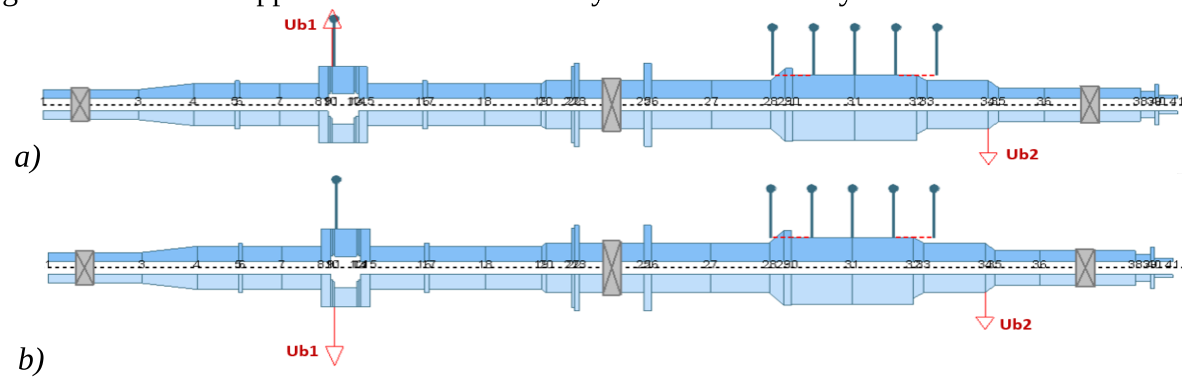


Figure 11: Applied Unbalances to Excite: (a) 1st; (b) 2nd mode.

The vibration amplitudes at the rated speed are presented in Table VI**Table** .

Table VI: Vibration Amplitudes at Full Load Operation at Rated Speed.

Unbalance placed to excite the mode	TGB Clearance	Probe location	UX	UY	Major Amplitude
			um	um	Um
1st	Min	TGB	1.3	5.6	5.7
		DE	0.7	1.0	1.1
		NDE	37.7	10.7	37.7
	Max	TGB	2.1	8.7	8.9
		DE	0.7	1.0	1.1
		NDE	37.7	10.7	37.7
2nd	Min	TGB	0.6	3.0	3.1
		DE	4.3	6.4	7.1
		NDE	33.1	9.4	33.1
	Max	TGB	1.1	4.7	4.8
		DE	4.3	6.4	7.1
		NDE	33.1	9.4	33.1

The unbalance response analysis results show that the vibration amplitudes are within the acceptable limits.

Conclusion: The sensitivity studies presented in the report were performed with the aim to analyze potential factors that may cause the issue with hydro turbine journal bearing. The turbine guide tilting pad bearing has been studied with regards to the hydrodynamic and mechanical characteristics at different initial data, such as variation of bearing clearance, inlet oil temperature, pads preload and offset, different bearing load, and direction of the applied hydro force. The rotordynamics analysis for the hydro turbine has been proved that the rotor-bearing system has no issues with dynamics which may affect bearing life. Based on the study it was concluded that TGB improper geometry parameters have resulted in the bearing failure. It was recommended to double-check the bearing preload and offset values and change to correct values. After the implementation of the recommended changes, the hydro turbine reached full power without TGB temperature increase.

References:

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