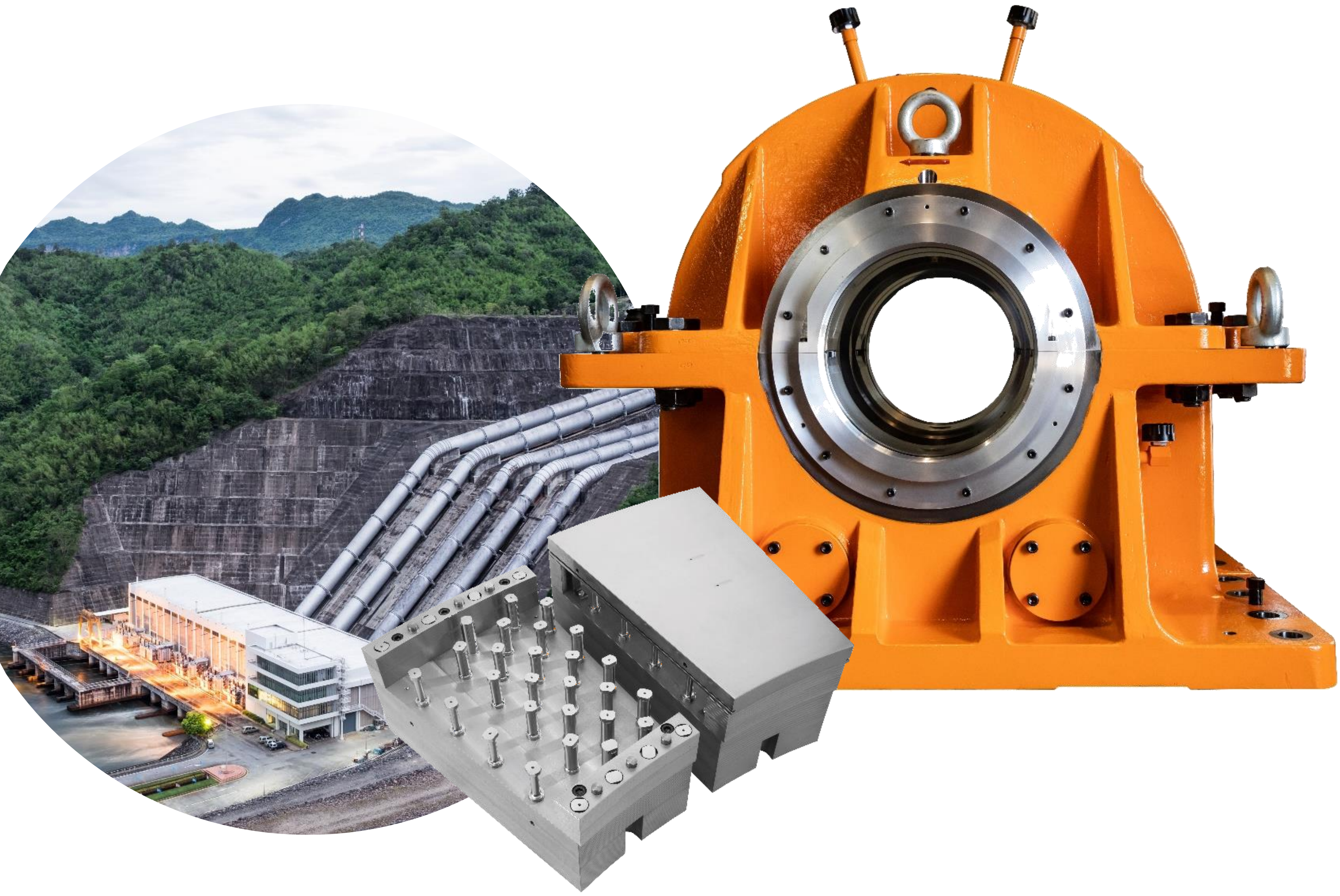
MIBA INDUSTRIAL BEARINGS

Hydropower Turbines
Engineered Success Cases

Miba Industrial Bearings Brasil

Turbine and generator manufacturers are often confronted with issues related to the project's specific characteristics, which influences the performance of the bearings used. In fact, some of these working conditions are a great challenge to the bearing manufacturer.

Turbine or generator design changes and consequently new requirements for bearings make a close partnership between machine and bearing designers necessary. In this paper we list some successful engineering collaborations resulting in bearing designs that fit the application best.



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1. Introduction

The main function of hydrodynamic sleeve bearings is to create a lifting force in an oil film developed by the shaft rotation inside the bearing. There are several key parameters affecting the oil film formation including, shaft speed, oil viscosity and flow rate, load amount and direction and the clearance between shaft's external diameter and bearing's internal diameter.

Bearings must thereby be capable of sustaining all static and dynamic forces occurring during different working condition. Besides the oil film formation, the bearing designer has further points to consider, such as, bearing housing stiffness, oil film stiffness and damping, accessibility to the bearing, assembly interface matching to the equipment, oil flow rate, oil sealing, proper working temperature, monitoring probe positioning, mechanical resistance of parts, etc.

The bearings are a performance-critical part of every rotating equipment and therefore a close look on a practical, yet efficient design has to be taken.

2. Synchronous generators - Fixed oil disc

Customer: Ideal Electric (USA)

Site: Jordanelle, Utah, USA

Application: Two horizontal synchronous generators working with horizontal Francis turbine

Rating: 6500 kW at 360 rpm, 12470 V, 3ph, 60 Hz

Customer's main request: No external oil lubrication system

Challenge: MIBA's standard pedestal design, with loose oil ring, did not provide the necessary oil flow rate requested by the thrust tilting pads.

Parameters		
Inputs	**Nominal**	**Run away**
Shaft speed (rpm)	360	620
Radial load (kN)	161.3	100.7
Thrust load (kN)	144.5	144.5
Environment temperature (°C)	40	40
Water inlet temperature (°C)	18	18
Outputs (4)		
Radial specific pressure (N/mm²)	1.79	1.13
Radial min. oil film thickness (mm)	0.048	0.067
Radial max. temperature (°C)	65	75
Thrust specific pressure (N/mm²)	2.23	2.23
Oil sump temperature (°C)	39	52
Thrust min. oil film thickness (mm)	0.021	0.022
Thrust max. temperature (°C)	69	81
Requested oil flow rate (l/min)	15	25.6

	Oil flow rate required by the hydrodynamic calculation	Oil ring capacity according to Lemmon / Booser calculation theory (1)	Oil ring capacity according to Keysell calculation theory (2)
Nominal	15 l/min	3.3 l/min	0.83 l/min
Runaway	25.6 l/min	4.1 l/min	1.36 l/min

Other theories like Innes; Dowson and Tayler (3) also indicate that even if we would have used two loose oil rings, they would not have supplied enough oil to the bearing. Therefore, Miba Industrial Bearings Brasil decided to use a fixed oil disc instead. However, Miba had little experience using oil discs during these conditions. There were some points that needed to be checked and confirmed whether they were suitable or not:

1. Would the fixed disk take on enough oil at both shaft speeds?
2. How should the fixed oil disk and its scraper be designed to provide the necessary oil flow rate?
3. The easiest place to position the oil disk would be on the shaft collars, but then it would collect a warmer mix of oil from the bearing sump in comparison to if it were located at the middle of the bearing. So, should we design a center collar bearing shell? If we do so, the shell would not have a self-alignment unless increasing the housing size to permit a much larger spherical seat. If we kept two separate shaft collars at the shell edges, we would need to open a slot at the shell centerline, which would make the shell body less stiff. The more we enlarge the disk width and its external diameter to increase its capacity to transport oil, the more we lose shell stiffness, which is quite critical for bearings with thrust load capacity.
4. Should the thrust faces get direct (fresh oil) lubrication or can they receive oil from the radial part (warm mix oil)? Would this warm oil mix prejudice the oil film thickness of the thrust pad in a significant way?

We decided to place the oil disk in the center of the bearing leaving the two separate thrust collars at their standard DIN bearing position, keeping the shell's self-alignment capacity during assembly. We also decided to not feed the thrust faces directly with fresh oil, but instead feed them with the radial oil.

The decision was backed by various tests and calculations including the following:

a) We calculated the amount of oil that the fixed oil disk would transport at both shaft speeds (nominal and runaway) for different widths and external diameters (5).
b) We had different options for the fixed oil disk dimensions, varying its width and its external diameter. We calculated the effect on the bearing shell stiffness, for each dimension of oil disk we had planned to open at the shell radial part. After that, we finally selected the fixed oil disk dimensions.

c) We created a small test rig to check if the actual oil amount delivered by the oil disc and its oil scraper design matched the calculated value at the given shaft speeds of the application.
d) We selected the clearance between the oil disk and the scraper according to the proper oil amount and possible shaft orbit.
e) We heated up the oil in the test rig sump to see how heat would affect the flow rate.

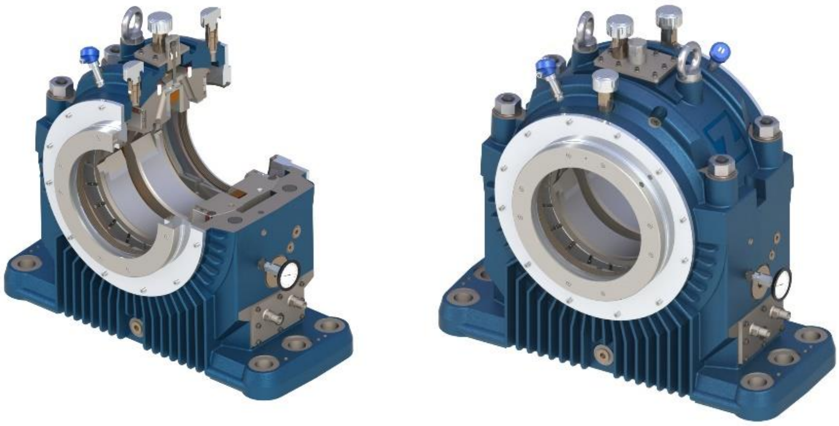

Figure 1: Miba engineered solution: ZR WFA 35
Bearing with fixed oil disk

Final Specification: MIBA customized pedestal bearing, with water cooler in the bearing oil sump, fixed oil disk & scraper, cylindrical profile and thrust tilting trapezoidal shape pads on both sides, shaft diameter 355 mm: ZR WFA 35 – 355

Right after this project, we received another order from the same customer for bearings with the same requirements, but larger shaft diameter (400 mm). This time, we used the same design concept in a bigger bearing housing.

Three horizontal synchronous generators working with Bulb turbines, rating 2635 kW at 157 rpm, 2400 V, 3ph, 60 Hz

Final Specification: MIBA customized pedestal bearing, with water cooler in the bearing oil sump, fixed oil disk & scraper, cylindrical profile and thrust tilting trapezoidal shape pads on both sides, shaft diameter: 400 mm – ZG WFA 45- 400.

3. Francis Turbine – 2 Bearing Configuration

In the past, turbine generator configurations used to have four bearings: Two bearings to support the turbine shaft and two additional ones to support the shaft of the generator.

Later this configuration changed to three bearings. One bearing to support the turbine shaft and the others to support the generator shaft.

This last-mentioned configuration was used e.g. for the projects PCH (PCH means Small Hydropower Plant in Portuguese) Colino II, Brazil, (HISA turbine) and PCH Sete Quedas, Brazil, (VOITH), which had the following data:

	PCH Colino II, Brazil	**PCH Sete Quedas, Brazil**
Nominal shaft speed (rpm)	900	900
Height (nominal fall) (m)	196.91	152.3
Flow rate (m^3/s)	4.65	5.592
Power (kW)	8200	7650

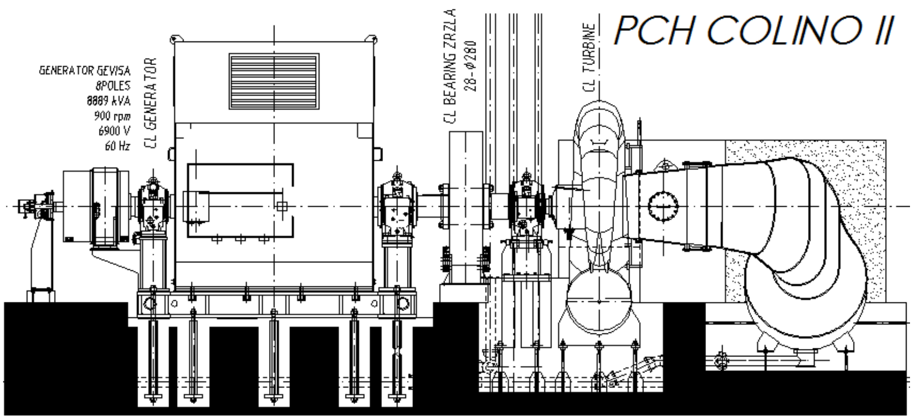

Figure 2: PCH COLINOII - 3 bearings configuration; Francis turbine

3.1 Francis: 2 Bearing Configuration

Nowadays the machine configurations changed again and often only two bearings are used. One guide bearing and one combined (guide and thrust) bearing support the generator shaft.

The turbine shaft is overhung and has no bearing. In this condition, the generator receives thrust loads and has to deal with the shaft deflection and following the turbine clearances. The next examples show the named configuration:

	São Tadeu HISA	Santa Laura HISA	Santa Rosa HISA	Queixada VOITH	Malagone VOITH
Nom. speed (rpm)	900	327	600	327.3	400
Height (fall) (m)	190	37.37	124	38.36	53.57
Flow rate (m^3/s)	5.36	23.04	9.15	22.564	20.36
Power (kW)	9278	7733	10310	7770	9800

From the bearing point of view, the configuration of PCH São Tadeu shown in figure 3 (position of the flywheel) allows a better distribution of the radial load on the combined bearing. In some other cases, the radial load might be quite light.

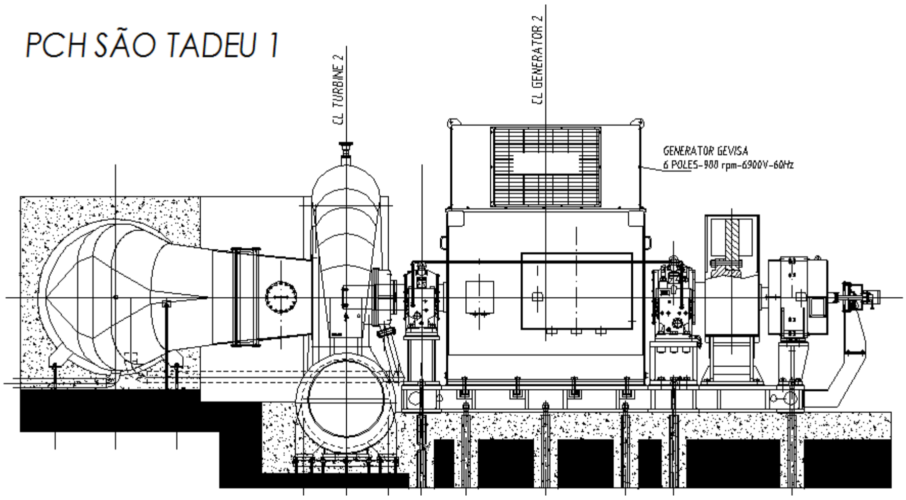

Figure 3: PCH SAO TADEU- 2 Bearings configuration Francis turbine

From the turbine manufacturer's point of view, the closer the generator's combined bearing is to the turbine; the stiffer is the shaft. The limitation for this approach is the civil foundation configuration.

Francis turbines increased in power, but did not grow in size, in fact they are even more compact. Therefore, the requirements on bearings increased too.

The generator bearings start receiving loads and deflections imposed by the turbine. The bearing designer has to pay special attention to the deformations of the bearing housing, shaft collar and thrust pads, and the resulting influence of these deformations on the oil film thickness, on the thrust load distribution and on bearing's temperature. However, these parameters are interrelated. We have to keep in mind that the mechanical deformations squeeze the oil film which consequently generates an increase in temperature. A higher temperature in turn generates a smaller oil film thickness.

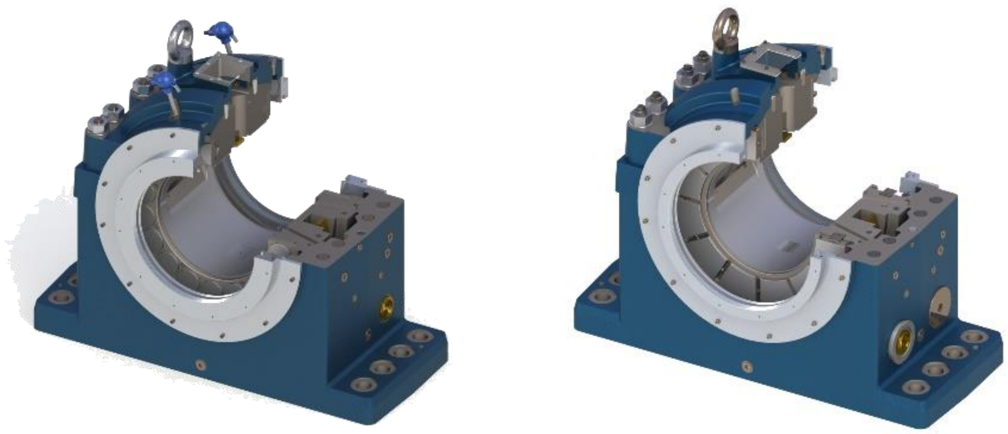

Figure 4: Left: standard Miba pedestal bearing with round thrust tilting pads. Right: special Miba thrust trapezoidal tilting pads

To solve this problem MIBA had to change the standard DIN thrust pad design (round pads) to a trapezoidal shape in order to increase the pad's load capacity and to allow it to work off-set instead of centered, which increases the capacity of the oil film formation and its thickness (6). The different pad designs are shown in figure 4.

The shaft thrust collars had to be enlarged and their external diameter increased to support the higher thrust loads and to keep the specific pressure on the thrust pads below safe limits. This new slenderness of the thrust shaft collar had to be analyzed in order to check its effect on the bearing thrust tilting pads. We also had to analyze the housing deformation effects.

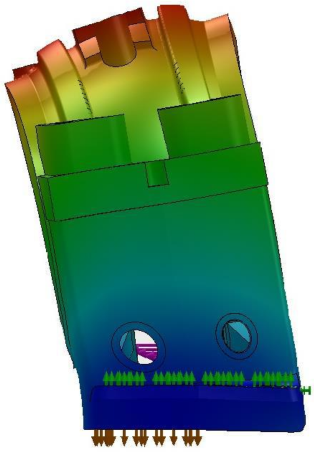

Figure 5: Housing Deformation Simulation

4. Small Size Bearing Bulb or Open Pit Turbines

This type of turbine is normally used for water heads of 20m with high flow rates. Due to the bigger rotor size, high radial loads are applied to the bearings, but the thrust loads are not so significant.

The challenge here was to design a bearing capable to support high radial loads while being as small as possible. Not only to fit inside the machine bulb, but also to place it as close as possible to the turbine. The nearer it is to the turbine, the stiffer is the shaft and the better are the machine's rotor dynamics.

Limitations are the high loads and the need to use spherical seating for a better self-alignment. There was also a concern regarding the accessibility of the bearing's monitoring sensors.

MIBA made an engineered design, eliminating the bearing housing oil sump and adding huge oil outlets. We also modified the spherical seat diameter design to make the bearing's outer dimensions as small as possible. An example of this case is seen in figure 6 PCH Riacho Preto, Brazil.

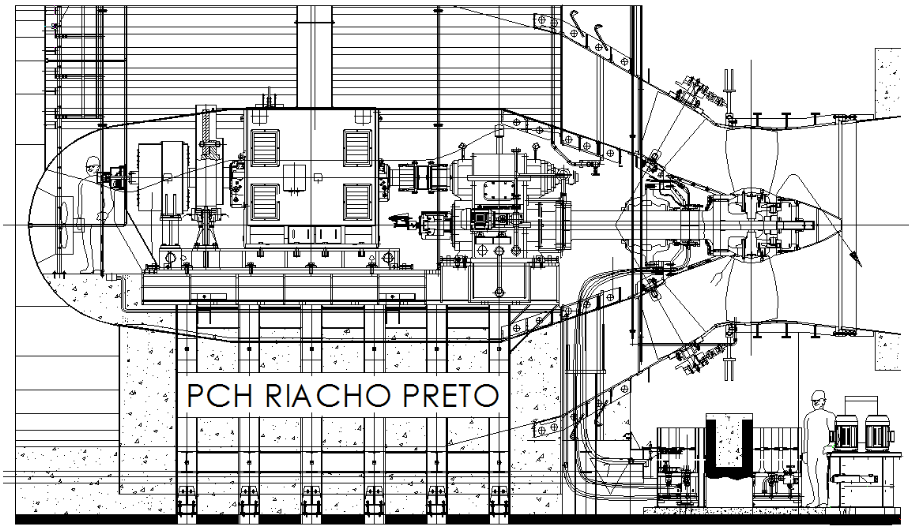

Figure 6: PCH Riacho Preto

The speed multiplier allowed a reduction in generator size, decreased its number of poles and consequently reduced its cost.

	PCH Riacho Preto, Brazil
Nominal shaft speed (rpm)	190
Height (nominal fall) (m)	10
Flow rate (m³/s)	55.3
Power (kW)	5057

In this configuration, we used two standard MIBA DIN bearings ZF type (end-flanged housing) on the generator and a special flanged guide bearing on the turbine which is shown in figure 7.

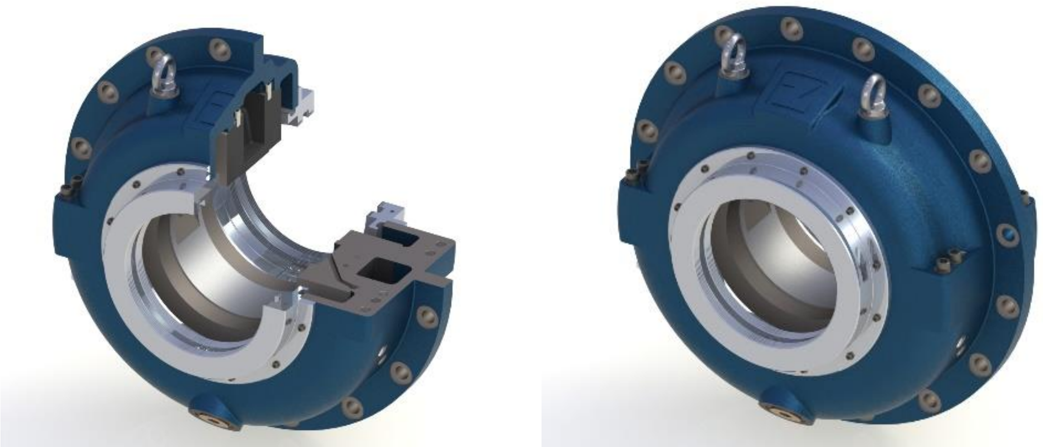

Figure 7: Smaller Size Engineered Flange Bearing

5. Kaplan S turbines Tail Water Level

This type of turbine is used for heads up to 60 m with high flow rates in areas with uneven raining periods. In order to reach a better performance and efficiency curves, the rotor blades can change their position to reach the best angle for different levels of water flow.

The challenge here was to design a bearing which is as small as possible, to be placed very close to the turbine, with spherical seat and with accessibility to the monitoring sensors. Yet, capable of supporting high thrust loads.

In relation to the previous case, MIBA's additional concerns on this case were the access to the thrust pads and to their sensors. Such type of turbines can be seen in PCH Salto Jauru and in PCH Boa Sorte, both using HISA turbines. Also, in PCH Indaia Grande and PCH Anhanguera, both VOITH turbines.

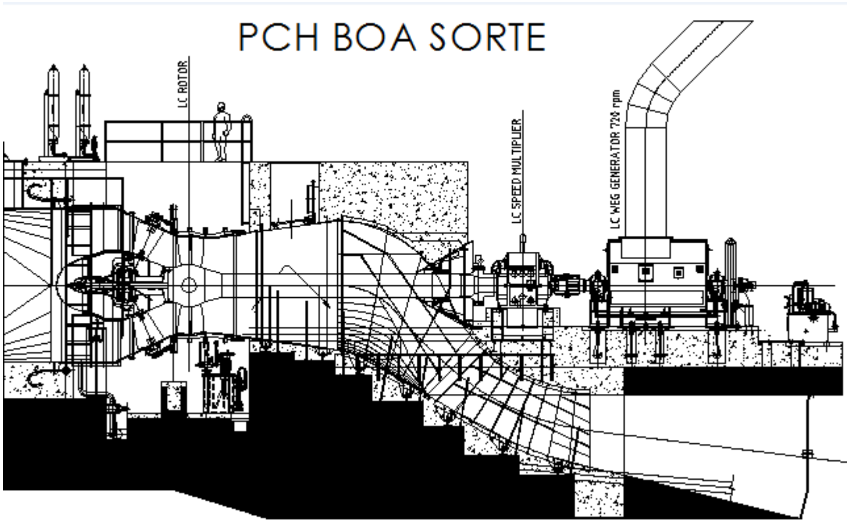

Figure 8a: PCH BOA SORTE Kaplan S Turbine

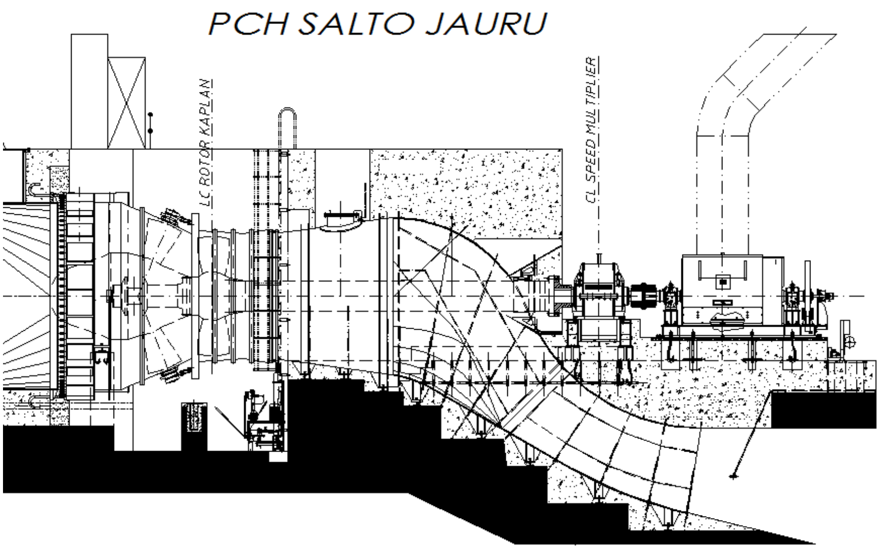

Figure 8b: PCH SALTO JAURU Kaplan S Turbine

	Salto Jauru HISA	Boa Sorte HISA	Indaia Grande VOITH	Anhanguera VOITH
Nominal speed (rpm)	200	270	400	400
Height (nominal fall) (m)	18.2	22.9	26.7	16.7
Flow rate (m^3/s)	60	41.5	28.76	49.86
Power (kW)	9918	8578	6899	7500

As we can see in figure 8a and 8b, there is no other possible place to install the bearing in this type of turbine. Since the water flows all around the turbine, it is not possible to use a pedestal bearing.

Because the thrust load is much higher than the counter thrust load, smaller thrust pads on one side of the bearing can be used which might result in a significant cost reduction.

Notice when using this kind of configuration, the shaft is pulled out instead of compressed against.

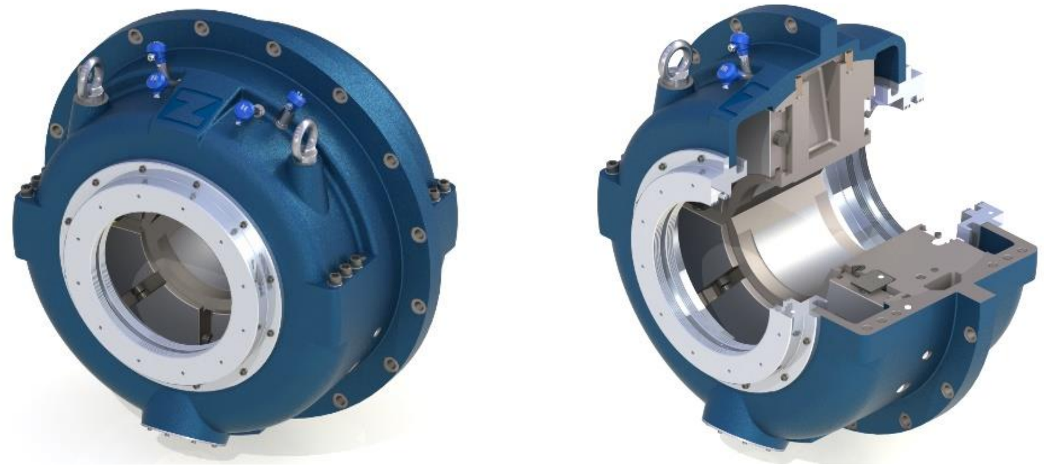

Figure 9: Special flanged bearing with trapezoidal *tilt pads*

Both examples above use two standard MIBA DIN bearings ZR (pedestal) on the generator and a special flanged combined bearing on the turbine.

6. Kaplan S Head Water Level

When comparing the Head Water Level to the Tail Water Level, we found that with the first one the civil work costs are cheaper. The Tail Water Level type needs a smaller waterfall height and the Head Water Level type allows an easier access to the machine.

The more the fall height increases, the bigger is the thrust load and consequently the bigger the bearing has to be. At a specific point it is not possible anymore to fit the bearing inside a Tail Water Level type; in this case, the solution can be a Head Water Level type.

The main concern here for the bearing designer is the very high thrust load that the combined bearing has to support while maintaining the necessary stiffness of pedestal bearing housing and of the shaft thrust collar.

At that time, MIBA's engineering team had never faced such thrust load requirement (1500 kN). In order to avoid housing deflections, it was important to keep the shaft centerline as close as possible to the floor and due to the thrust load value, we had to use trapezoidal shape thrust pads.

The big thrust tilting pads demanded a huge amount of oil. In case there was a failure in the external oil pump system the oil taken by oil rings or fixed oil disks would not have been enough, so we decided to remove the bearing housing oil sump, which reduced the shaft centerline height.

We placed the shaft thrust collar at the middle of the bearing to reduce the bearing total length and the shaft weight. It was easier to make a stiffer thrust collar having a unique centered part instead of two (one at each shell side). The importance of having a smaller shaft centerline height to decrease housing deflection overruled the benefit of using a spherical seat to help alignment during the assembly. Due to this thrust collar configuration, we had to elaborate a process and a device for assembling and disassembling of the thrust pads at the site. The housing design managed to become quite stiff.

In order to compensate the loss of the self-alignment of the spherical seat, the assembler has to raise the generator guide bearing or lower the combined bearing, looking for an even distribution of the radial load on the combined bearing, since the shaft thrust collar splits its radial part.

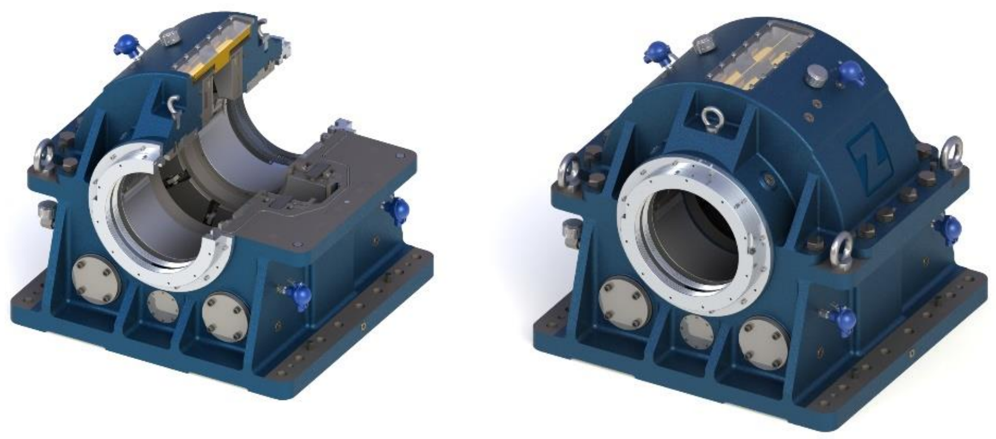

Figure 10: MIB Cataguases' ZT Bearing for Kaplan S Turbines

The huge amount of oil flow requested made us pay special attention to the bearing seal design and to the oil path, jet intensity, oil distribution inside the bearing housing, radial shells and thrust pads.

Later, we made three new designs using the same solution or configuration, for 1000 kN, 2000 kN and 2500 kN. The projects turned out so well that now this product is a MIBA Brazil standard product, known as ZT bearing.

Examples of this type of turbines we find in PCH Santo Antonio do Caiapó and in PCH Caju, both in Brazil.

	PCH S. Antonio do Caiapó Brazil	PCH Caju Brazil
Nominal shaft speed (rpm)	257	360
Height (nominal fall) (m)	29.67	28.87
Flow rate (m³/s)	57.3	20.05
Power (kW)	15465	5150
Thrust load on the combined bearing (ton)	195.9	81.12

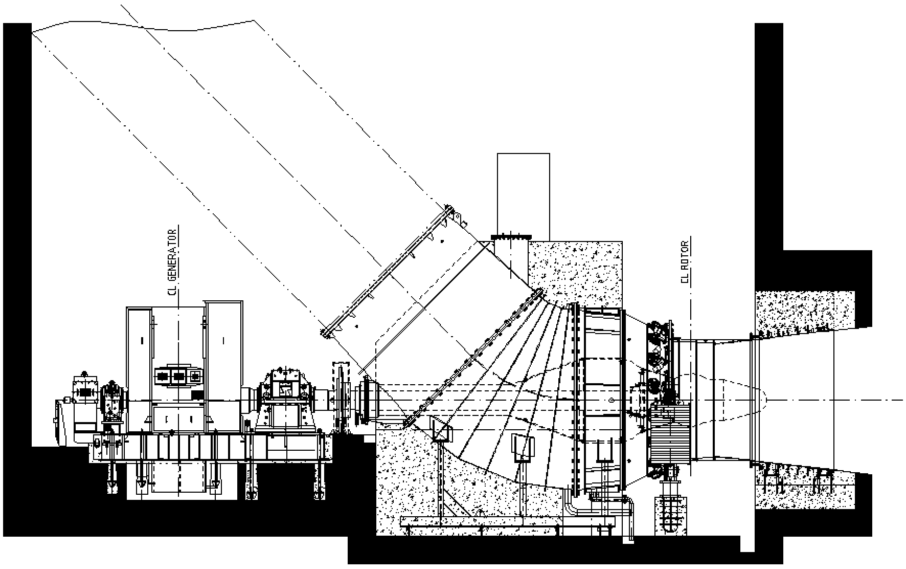

Figure 11: ZT Bearing in use; Kaplan S Turbine

7. Conclusion

The hydro market is developing constantly. The changing market requires new turbine or generator solutions. The bearings as performance critical part of rotating equipment need to be analyzed and revised to support these new requirements and allow the best solution for the specific application. Therefore, a close partnership between turbine or generator manufacturer and bearing designer is expedient.

8. Acknowledgement

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About the Author

Fernando Correia holds a master’s degree in mechanical engineering and is the Application Engineering Manager at Miba Industrial Bearings Brazil, a company based in Cataguases that designs and manufactures bearings for hydro generation, industrial and electrical machine applications since more than 40 years.

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