Challenging Edge Loading – A Case Study for Homogeneous Polymer Bearings Operating in Wicket Gates

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ABSTRACT

Wicket gate bearings for hydro electric turbines can possibly be subjected to great levels of edge loading due to shaft misalignment. Shaft misalignment is used as a general term to describe a situation where several bearings on one shaft line are not concentric or the axes of shaft and bearings are not parallel. In terms of wicket gates, there are two major causes of shaft misalignment leading to edge loading, namely the imperfect installation and insufficient shaft stiffness. Edge loading due to insufficient stiffness can occur in low to medium head, high specific speed and high power output turbines. In such cases, in order to achieve the required power output, the gate height may be designed rather tall with a guide vane area that is also rather large. Misalignment due to installation can be corrected by careful and diligent procedures but the shaft stiffness cannot be corrected during operation and might become an inherit issue to bearings. This paper deals with an edge loading problem that resulted in an excessive bearing wear caused by possibly insufficient shaft stiffness and provides a practical solution by utilizing Thordon homogeneous polymer bearings.

1.0 Shaft Bending and Edge Loading

Insufficient shaft stiffness causes excessive slope when shaft bends as a combined result of hydraulic pressure on the guide vane and pulling force by the operating mechanism. There are two methods to determine the shaft slope at the bearings that are the traditional analytical approach and the modern finite element method. N. N. Kovalev, /1/ (1961) developed a partial analytical and partial numerical procedure to determine shaft slope caused by bending. This method has been widely used for wicket gate calculations and even in today's age of prosperous computerization, it still remains as an effective method in practice because of its simplicity and reasonable accuracy. In addition to this practical method, Finite Element Method (FEM) has become a dominant technique due to its ability to handle complex geometry and virtually any type combination of physics. Wicket gate shaft can be simulated using FEM and a relatively reliable result can be obtained. However, FEM can only be so accurate as defined by assumed physical and boundary conditions. It cannot completely replace a diligent analytical approach in certain circumstances. In this case study, the traditional analytical approach had been utilized to calculate the shaft slope and reaction forces and its result was compared with FEM analysis.

By knowing the bearing load and the shaft slope obtained by shaft bending analysis, it is possible to calculate pressure distribution over the contact surface due to shaft inclination. The resultant maximum pressure is then used as initial information to select a suitable material and determine a viable design. It seems there is very little attention paid to this very special topic in literatures. Engineers at Thordon Bearings Inc have created a practically very useful method to predict the bearing pressure caused by edge loading. When the axis of shaft inclined to axis of the bearing, the contact surface between shaft and bearing is shifted to one end of bearing that creates a very high pressure zone at one end of the bearing. The contact surface is a 3-D surface in shape (Figure -1), but the projected contact area becomes approximately an ellipse. The pressure from high at very end of bearing falls down along shaft in direction to bearing inside to zero. The integration of the pressure over the entire contact surface builds up the bearing capacity to support the shaft load. Analytical study revealed that the bearing capacity, the supporting force of bearing $F_{capacity}$, is a function of the compressive E-modulus of material, the wall thickness of bearing, the running clearance of bearing, the shaft diameter and the bearing surface deflection at very end edge of bearing and shaft slope. The bearing capacity can be expressed with following equation

$$F_{capacity} = K_{\mu} \cdot K_{\eta} \frac{E_c \cdot D \cdot \delta_m^2}{W \cdot S} \ . \qquad (1)$$

In this equation, the variables are:

$F_{capacity}$: Supporting force of bearing (N)

$K_{\mu} = (1-\mu)/(1+\mu)(1-2\mu)$. This factor is based upon the assumption that two strains other than loading direction are zero. This assumption could lead to a stiffer bearing and higher estimated edge load than actual.

$K_{\eta} = 0.0959 \cdot \eta^3 - 0.086 \cdot \eta^2 + 0.327 \cdot \eta - 0.0017$. This factor was obtained by integrating bearing pressure over actual contact surface and is valid only if the length of contact area is smaller than bearing length.

μ: Poison ratio of material

E_c: Compressive E-modulus of bearing material (N/mm^2)

D : Shaft diameter (mm)

W: Wall thickness of bearing (mm)

S : Shaft slope at bearing

δ_m: Maximum deflection of bearing surface at edge of bearing end (mm)

$\eta = \sin(\Phi_m/2)$

Φ_m: Contact angle between shaft and bearing at the very end edge

It is to note that contact angle Φ_m is not an independent variable rather a function of other variables. There exists a relation

$$\eta = \sin\frac{\Phi_m}{2} = \sqrt{1 - \left[\frac{(1+\psi)^2 + \left(\frac{2\delta_m}{D} + \psi\right)^2 - 1}{2(1+\psi)\left(\frac{2\delta_m}{D} + \psi\right)}\right]^2} \ . \qquad (2)$$

ψ: The ratio of bearing running clearance to shaft diameter

Equation (1) and (2) shows that the only unknown variable is the maximum surface deflection at the edge of bearing end δ_m . Other parameters are either obtained during shaft bending analysis such as shaft slope or the geometric information of bearing and material property. In effect, equation (1) is a relationship between bearing load capacity $F_{capacity}$ and maximum surface deflection at the edge of bearing end δ_m . The true use of equation (1) is to calculate the maximum surface deflection δ_m by a known bearing load

$$\delta_m = \sqrt{\frac{W \cdot S \cdot F_{capacity}}{E_C \cdot D \cdot K_\mu \cdot K_\eta}} \quad . \qquad (3)$$

It needs to emphasize that variable δ_m and η are interdependent. Iteration technique is required to solve equation (3) for the maximum surface deflection δ_m . After obtaining the maximum surface deflection at bearing end, the peak pressure is calculated using following equation

$$p_m = K_\mu \cdot \frac{E_C}{W} \cdot \delta_m \quad . \qquad (4)$$

The projected contact area is approximately

$$A_{pro} = \frac{\pi \cdot D}{4} \cdot \sin\frac{\Phi_m}{2} \cdot \frac{\delta_m}{S} \quad . \qquad (5)$$

Note that Φ_m is the contact angle between shaft and bearing at the bearing end (see Figure -1). Similar to the projected bearing area under perfectly aligned condition, the projected contact area for misaligned cases is the real contact surface projected to the horizontal plane through bearing central axis. The pressure over projected contact area is then obtained using equation (6)

$$p_{pro} = \frac{F}{A_{pro}} . \qquad (6)$$

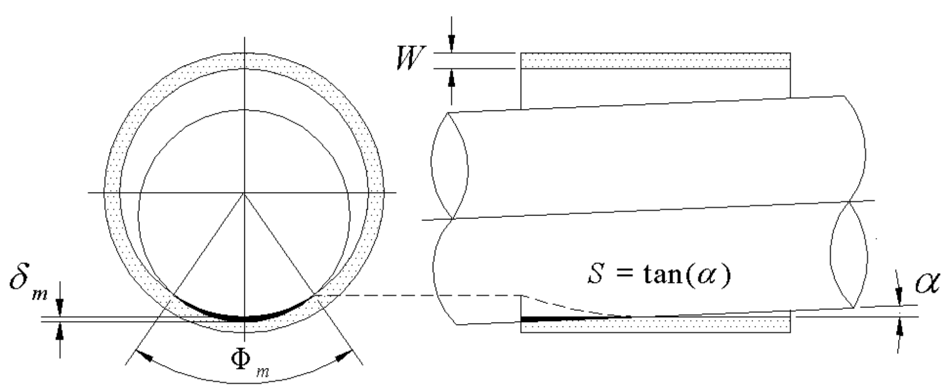

Figure -1: Notions on equations

The equations above are derived from elastic theory. Examining the predicted edge peak pressure with equation (4), it can be noted that in real bearing situations, the behavior of deformation at very edge may be quite different from the rest of inside of the bearing. The material at the bearing end is less restrained and the relation $K_\mu = (1-\mu)/(1+\mu)(1-2\mu)$ may be invalid and it rather approaches to $K_\mu = 1/(1-\mu^2)$. Consequently it is resulted in a decreased bearing peak pressure at bearing end. Peak pressure predicted by equation (4) shall therefore be higher than real peak pressure. Figure -1 provides the meaning for some of the most important dimensions.

2.0 Clamping Force and Friction Force

Peter Bakker /2/ discussed the effect of clamping force on non-metallic bearings. If the contact angle between shaft and bearing increases so does the actual friction force due to so-called clamping effect. This phenomenon is not new and it had been used since long time by traction elevators or other mechanical device where an increased friction is preferred. However, this clamping force is unwanted for bearings where a minimum friction is desirable. There are two main factors determining the amount of clamping force that are the running clearance and bearing stiffness. If the running clearance is small, it tends to result in a large contact angle and an increased friction force while a large clearance tends to reduce the friction, but may cause other issues such as high edge loading. Therefore, proper clearance is a trade off and can only be determined according to practical need. Increasing bearing stiffness reduces contact angle and therefore reduces friction force. However, an improper high stiffness of bearing will suffer from high edge loading and premature failure. The camping force for inclined shaft can be calculated with equation (7)

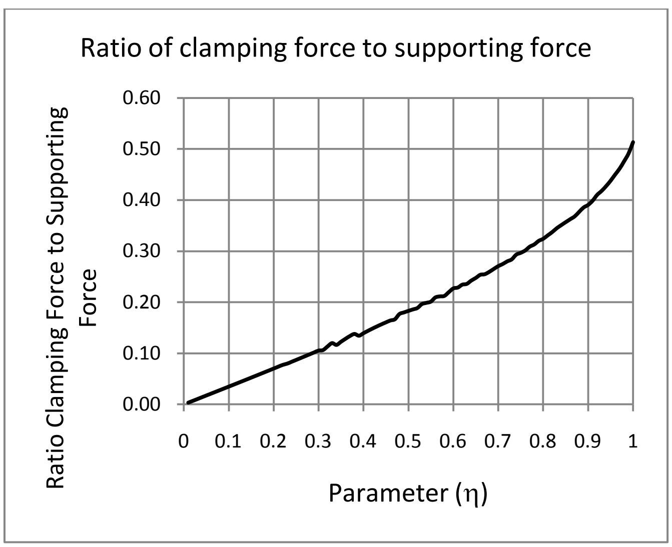

Figure – 2: Ratio of Camping to Supporting force defined by equation (8)

$$F_{clamp} = K_{\mu} \cdot K_{\eta}^{'} \frac{E_c \cdot D \cdot \delta_m^2}{W \cdot S} \quad . \tag{7}$$

Above equation appears very similar to equation (1). The only difference is another modification factor is used that is $K_{\eta}^{'} = 0.758 \cdot \eta^5 - 1.475 \cdot \eta^4 + 1.054 \cdot \eta^3 - 0.192 \cdot \eta^2 + 0.026 \cdot \eta$ instead of K_{η}. Thus the ratio of clamping force to bearing supporting force is calculated by following equation

$$\Lambda = \frac{F_{clamp}}{F_{capacity}} = \frac{K_{\eta}^{'}}{K_{\eta}} \, . \tag{8}$$

Figure – 2 illustrates the ratio of clamping to supporting force. If one recalls $\eta = \sin(\Phi_m / 2)$, η = 1 means Φ_m = 180 degree. From above diagram is to see if contact angle approaching to 180 degrees, the amount of clamping force can be as high as 50% of bearing supporting force which means the friction coefficient goes up to 1.5 times of material coefficient of friction. However, for well aligned wicket bearings, the contact angle is typically not more than 40 degrees, so parameter η may fall about 0.35 and the coefficient of friction may possibly increase 10 to 15%. It must point out that the equation (1) to (8) is applicable only to cases where $\delta_m / S < L$, here "L" is bearing length. If this condition is not fulfilled then a more complicated procedure is required for this type of analysis.

3.0 A Case Study for Homogeneous Polymer Bearings Operating in Wicket Gates

The previous sections have provided the theoretical background for calculating the maximum bearing pressure and clamping force for an inclined shaft. This section provides a case study where a high edge loading is the predominant factor. For most applications, only a general review of nominal bearing pressure may suffice to justify the design chosen since the nominal bearing pressure is typically designed at a much lower level than the strength of material chosen. However, there are indeed cases where edge loading may be the dominant factor to consider due to both of imperfect installation and insufficient shaft stiffness. Following case that Thordon Bearings Inc had solved was one of the challenges.

A hydraulic turbine owner in South America has experienced persistent wicket gate bearing problems at their hydroelectric station. The root cause was a result of excessive edge loading that had been evidenced by increased friction force and excessive wear and wear pattern of wicket gate bearings. The power plant houses two 230 MW medium head (99 metres), high power vertical Francis turbines generating 230 MW of power at 125 rpm.

The study provides an analysis identifying the possible effects of gate deflection on the bearings and the performance after replacing the existing wicket gate bearings with a non-metallic, homogeneous polymer. It uses the practical method to calculate edge loading on bearings introduced before and explains in details why non metallic polymer materials such as ThorPlas® are more suitable and able to successfully handle edge loading better than stiffer metallic bearing components. An excellent performance result was obtained from an 8-month trial conducted by the owner using ThorPlas® in early 2008 which eventually had led them to replace all existing bearings in April of 2009 for one turbine followed by the retrofit of second turbine in April 2010.

3.1 Determination of Shaft Slope at Bearings

The shaft slope at bearings was calculated with using the analytical method per N.N. Kovalev /1/ and the result was compared with the finite element analysis /3/. The maximum load used for the calculation was 10 bar hydraulic pressure acting on guide vanes in combination with 160 KN operating link load. For investigation purposes these loads are considered to operate in the same plane and in opposite directions. Figure - 3 shows the assumed loading arrangement for this investigation. The assumptions applied to the calculation were

- The shaft is considered as an elastic beam, point–supported at bearing centers. There are 3 bearings named as Lower, Intermediate and Upper bearing.
- The moment of inertia of guide vane was calculated in the weakest bending resistant direction.
- The 3 bearings (Lower, Intermediate and Upper) are ideally aligned
- Bearing surface material is homogeneous elastic

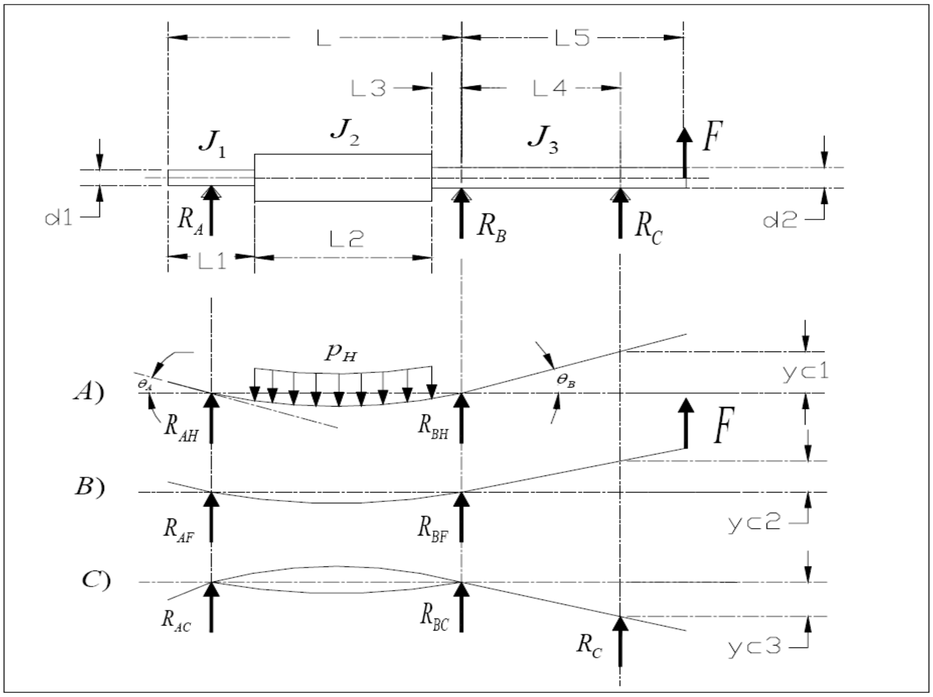

Figure - 3: Loading diagram

As shown in Figure-3, the system is static over-determined. However, since the system can be fairly assumed as linear, the superposition principle can be applied to it. This means one can separate loading case into 3 individual conditions (a), (b) and (c).

Loading Case (a):	Loaded by hydraulic pressure on guide vane only
Loading Case (b):	Loaded by gate ring pulling force only
Loading Case (c):	Loaded by reaction force of upper bearing R_C

Now each individual loading case is static determined. After calculating each case and add all result together, one is able to receive the reactions and shaft slope of entire system (see N.N. Kovalev /1/).
In loading case (c), there are 3 unknowns including bearing reaction R_C. The 2-dimension system however can only solve two unknowns and thus one more equation is required to solve all variables. Shaft deflection at upper bearing is chosen as such variable. This means all reactions for lower, middle and upper bearings to be expressed as a function of deflection at upper bearing. It is designated with “y_C”. According to Figure -3, there is:

$$y_C = y_{C1} + y_{C2} + y_{C3} \tag{9}$$

The final value of "y_C" shall be equal to the running clearance of upper bearing which is a known design parameter. Now information is sufficient to solve the problem. Without going into very details of solving procedure, following section of this paper focuses on discussion of result.

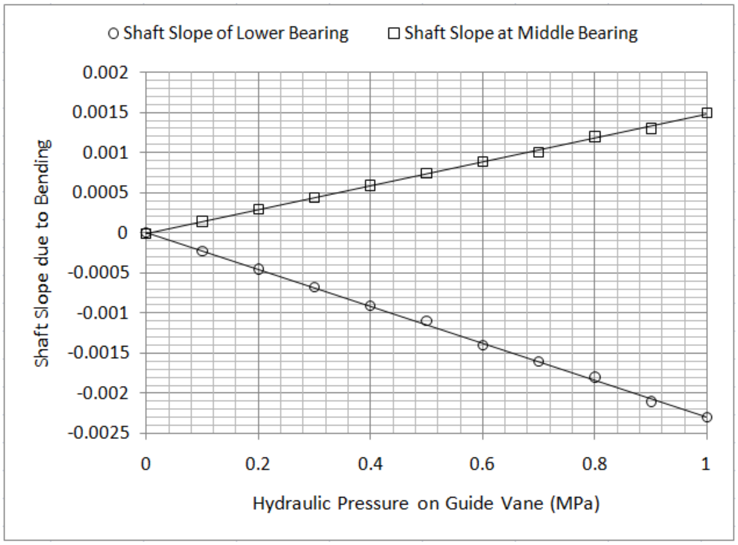

Figure - 4: Shaft Slope for Middle and Lower bearings

Table -1: Bearing reaction and shaft slope at max pulling force and hydraulic pressure

	Shaft Diameter (mm)	*Bearing Length (mm)*	*Bearing Reaction (N)*	*Shaft Slope*	*Bearing Reaction/3/ (FEM) (N)*	*Shaft Slope (FEM)/3/*
Upper	248	60	-3.29×10^5	N/A	-2.54×10^5	N/A
Middle	280	160	1.03×10^6	0.0015	0.83×10^6	0.0017
Lower	250	140	6.36×10^5	-0.0023	6.16×10^5	-0.0021

Figure - 4 shows the calculated shaft slope at lower and middle bearing. The pulling force of linkage arm applied at top end was assumed at maximum. It is worth noting that neither the pulling force of linkage nor the clearance of top bearing has a large effect on shaft slope of lower and middle bearings. The dominant factor was found however to be the hydraulic force. An independent study performed by a third party institution provided a finite element analysis for the shaft deformation with similar conclusion. It was reported that the shaft slope of middle bearing was 0.0002 if only the maximum pulling force of linkage applied. However this slope became 0.0017 if maximum hydraulic guide vane pressure applied. The FEM result was listed in Table -1. The data on the table was based on hydraulic pressure applied to guide vane = 145 psi (1.0MPa) and the pulling load of linkage applied to top end of shaft = 36000 lbs (160530 N). The pulling force of this turbine had been increasing in operation in the history. This was suspected due to the damaged bearing with increased friction.

3.2 Maximum edge pressure

The reason behind of edge loading calculation is to determine the uneven distribution of bearing pressure over the actual contact surface caused by shaft inclination. The algorithm is partially analytical and partially numerical using equation (1) to (6). To solve the bearing surface deflection at end edge, numerical iteration procedure was applied. Peak and nominal bearing pressure is shown on Figure -5.

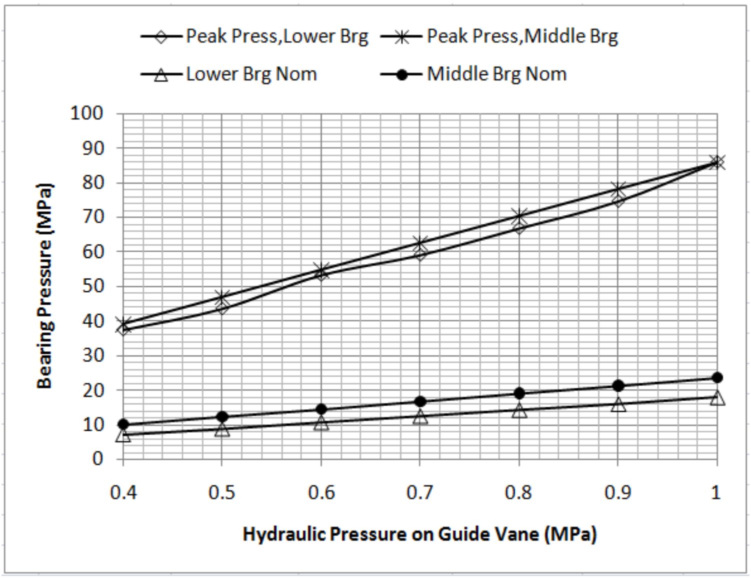

Figure – 5: Peak and Nominal Bearing Pressure (MPa)

ThorPlas® bearing was chosen for this application. It was an optimal choice on balancing many factors. Since the shaft slopes at bearings are relatively large, it is absolutely necessary to choose a material that is able to mitigate edge loading. The material also must have a capability of minimizing the clamping force. The wall thickness is restricted to 5 mm due to existing space. The material has to maintain proper interference force to keep within housing. Low friction and low wear rate material is a requirement and it must be grease free as well. According to reaction forces obtained above, the nominal bearing pressure at lower and middle bearing is quite low. The average pressure over the actual contact area (half ellipse) is not excessively high either. However, the peak pressure at very edge of bearing is much higher than nominal pressure. Based on Figure - 5, the peak pressure is roughly 4 times higher than nominal pressure. This is however much lower than the edge pressure for a rigid metallic bearing which would be about 350 MPa and 15 times higher than nominal under the same conditions. To reduce peak pressure at very edge, all ThorPlas® bearings for this application were machined a chamfer at both ends. In effect, this has changed the bearing edge toward to inside of bearing. The estimation of maximum bearing pressure was from shaft bending perspective. However, the deformation of supporting structure of wicket gate shaft, the misalignment of shaft from installation, fluctuation of hydraulic pressure on the guide vane, all these factors might render a precise estimation of the true maximum bearing pressure extremely difficult.

3.3 Installation and Operational Result

The trial installation of bearings for two gates in one of the two turbines was firstly completed in March 2008. In November 2008, the bearings were inspected to verify the performance. The inspection confirmed that there was no evidence of damage and wear on bearing surface. Customer had been delighted by this exceptional result. By end of 2008, the customer ordered 29 wicket gate bearing kits including seals to protect abrasives from entering into lower and middle bearings. By time of April 2009, one of the turbines from this customer had been completely equipped with ThorPlas® bearings. After completion of the installation, the system was monitored carefully, especially the oil pressure of the servo motor. The servo system pressure was kept flatly at expected level and with no damage and leakage observed. In December 2009, customer reported that all bearings were operating well. Figure - 6 below was the lower bearing after 8 months operation.

Figure - 6: One Lower Bearing after 8 months operation

4.0 Summary

In this case study through an analytical calculation of the reaction forces and shaft slope at bearings, it was identified that an insufficient stiffness of wicket gate shaft was the dominant factor contributing to the high edge loading and excessive wear on wicket gate bearings. The real problem however is that the edge loading caused by shaft bending cannot be corrected by careful installation. The problem only shows up when the turbine going into operation. The only way to mitigate is to choose a suitable material with suitable bearing design or stiffen the shaft and guide vane as suggested by the third party analysis /3/. The application concerning this study is a rare case having such high-level shaft slope and edge loading. ThorPlas® Bearings were chosen according to estimated edge loading pressure and existing space available. The actual performance of the bearing confirmed that this was the right choice for the challenging application. The homogeneous elastic nature allows bearing surface to slightly deflect and spreads the load over a relatively large contact surface so that the resulting peak pressure is controlled within the acceptable range of material strength. Thordon ThorPlas® bearings are made of new ThorPlas® homogeneous polymer with very low coefficient of friction. It is easy to machine and install. Typical designed nominal pressure for ThorPlas® bearings is 31 MPa. It concludes that the factors, such as identification of problem, finding the root course, selecting right material, using suitable design and careful manufacturing and installation all are the keys to success.

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Reference:

/1/ N. N. Kovalev, Hydroturbines, Design and Construction, 1961, translated from Russian by Israel Program for Scientific Translation, Jerusalem, 1965
/2/ Peter Bakker, New Methods Optimise the Performance of Composite Bearings in Hydropower Applications, Hydro Power 2009, Trelleborg Sealing Solutions
/3/ Independent Study for Wicket Gate Bushing Wear Analysis; prepared for Endesa Chile, 2007
/4/ ThorPas® Bearings Engineering Manual, Version TP2006.1

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