Challenging edge loading: a case for homogeneous polymer bearings for guidevanes

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This paper introduces a calculation method to determine the peak pressure on a bearing with a misaligned shaft. A case study is presented, where the guidevane bearings had experienced persistent excessive wear as a result of edge loading, possibly caused by insufficient shaft stiffness. A practical solution is provided, involving the use of non-metallic, homogeneous polymer bearings.

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 the case of guidevanes, there are two major causes of shaft misalignment leading to edge loading, namely imperfect installation and insufficient shaft stiffness. Edge loading caused by insufficient stiffness can occur within low to medium head, high specific speed and high power output turbines. In such cases, to achieve the required power output, the gate height may be designed to be quite tall with a guidevane area that is also rather large. Misalignment caused by installation can be corrected by careful and diligent procedures, but the shaft stiffness cannot be corrected during operation and can become an inherit issue to bearings. This paper deals with an edge loading problem which resulted in an excessive bearing wear possibly caused by insufficient shaft stiffness, and provides a practical solution involving the use of homogeneous polymer bearings.

1. Shaft bending and edge loading

Insufficient shaft stiffness causes an excessive slope when the shaft bends as a combined result of hydraulic pressure on the guidevane and pulling force by the operating mechanism. There are two methods to determine the shaft slope at the bearings the traditional analytical approach and the modern finite element method. Kovalev, [1965[1]] developed a part analytical and part numerical procedure to determine shaft slope caused by bending. This method has been widely used for guidevane calculations and even in today's age of prosperous computerization, it remains an effective method in practice because of its simplicity and reasonable accuracy. In addition to this practical method, the finite element method (FEM) has become a dominant technique because of its ability to handle complex geometry and virtually any type combination of physics. The guidevane shaft can be simulated using FEM and a relatively reliable result can be obtained. However, FEM can only be as accurate as can be 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 used to calculate the shaft slope and reaction forces. The 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 caused by shaft inclination. The resulting maximum pressure is then used as initial information to select a suitable material and determine a feasible design. It seems that very little attention is paid to this very special topic in the literature. Engineers at Thordon Bearings Inc have created a practical and useful method to predict the bearing pressure caused by edge loading. When the axis of the shaft is inclined to the axis of the bearing, the contact surface between the shaft and bearing is shifted to one end of the bearing and this creates a very high pressure zone at one end of the bearing. The contact surface is a 3D surface in shape (Fig.1), but the projected contact area becomes approximately an ellipse. The pressure from high at very end of bearing along shaft in the 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. An 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 the very end edge of the bearing and shaft slope. The bearing capacity can be expressed by following equation

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

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 on 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 the actual one.

$K_\eta = 0.0959 \cdot \eta^3 - 0.086 \cdot \eta^2 + 0.327 \cdot \eta - 0.0017$

This factor was obtained by integrating the bearing pressure over the actual contact surface, and is valid only if the length of the contact area is smaller than the bearing length.

μ = Poisson ratio of bearing material

E_c = Compressive E-modulus of bearing material (MPa)

D = Shaft diameter (mm)

W = Wall thickness of the bearing (mm)

S = Shaft slope at bearing

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

$\eta = \sin(\Phi_m/2)$ a parameter defined by eq. 2

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

It should be noted that contact the angle Φ_m is not an independent variable, but is a function of other variables. It follows that:

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

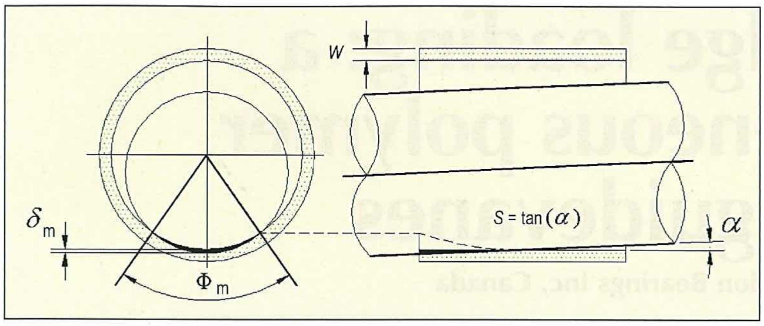

Fig. 1. Notations for the equations.

Where Ψ is the ratio of bearing running clearance to shaft diameter

Eqs. (1) and (2) show 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, eq. (1) is a relationship between bearing load capacity $F_{capacity}$ and maximum surface deflection at the edge of bearing end δ_m. The real use of eq. (1) is to calculate the maximum surface deflection δ_m by a known bearing load:

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

It should be emphasized that variable δ_m and η are interdependent. An iteration technique is required to solve eq. (3) for the maximum surface deflection δ_m. After the maximum surface deflection at the bearing end has been obtained, the peak pressure is calculated using following equation

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

The projected contact area is approximately:

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

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

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

The equations above are derived from elastic theory.

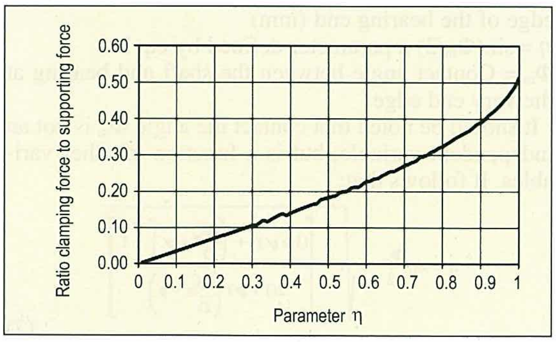

Fig. 2. Ratio of the clamping to supporting force defined by eq. (8).

Examining the predicted edge peak pressure with eq. (4), it can be noted that in real bearing situations, the deformation behaviour at the very edge may be quite different from that inside the bearing. The material at the bearing end is less restrained and the relationship

$$K_\mu = (1-\mu)/(1+\mu)(1-2\mu)$$

may be invalid, and it may approach to $K_\mu = 1/(1-\mu^2)$. Consequently, there will be a decreased bearing peak pressure at the bearing end. The peak pressure predicted by eq. (4) will therefore be higher than the real peak pressure. Fig. 1 explains some of the most important dimensions.

2. Clamping force and friction force

Peter Bakker [2009[2]] discusses the effect of clamping force on non-metallic bearings. If the contact angle between the shaft and bearing increases, so will the actual friction force, as a result of the so-called clamping effect. This phenomenon is not new, and has been used for some time by traction elevators and other mechanical devices where increased friction is preferable. However, this clamping force is not desirable for bearings where a minimum friction is advantageous. There are two main factors determining the amount of clamping force: the running clearance and the bearing stiffness. If the running clearance is small, this 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 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 inappropriately high stiffness of the bearing will cause high edge loading and premature failure. The clamping force for the inclined shaft can be calculated using eq. (7)

$$F_{clamp} = K_\mu \cdot K'_\eta \frac{E_c \cdot D \cdot \delta_m^2}{W \cdot S} \quad \text{... (7)}$$

The equation above appears very similar to eq. (1). The only difference is that a different modification factor is used, which 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 the clamping force to the bearing supporting force is calculated by:

$$\Lambda = \frac{F_{clamp}}{F_{capacity}} = \frac{K'_\eta}{K_\eta} \quad \text{... (8)}$$

Fig. 2 shows the ratio of the clamping force to the supporting force. It should be recalled that:

$\eta = \sin(\Phi_m/2)$, $\eta = 1$ means $\Phi_m = 180$ degrees.

From Fig. 2 it can be seen that if the contact angle approaches 180°, the amount of clamping force can be as high as 50 per cent of the bearing supporting force, which means the friction coefficient increases to 1.5 times the material coefficient of friction. However, for well aligned guide 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 per cent. It must be pointed out that the eqs. (1) to (8) apply only to cases where:

$$\delta_m/S < L$$

here L is the bearing length. If this condition is not fulfilled, then a more complicated procedure is required for this type of analysis.

3. Case study of homogeneous polymer bearings operating in guidevanes

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 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 the material chosen. However, there are indeed cases where edge loading may be the dominant factor to consider, as a result both of imperfect installation and insufficient shaft stiffness. The following case describes one of the challenges which was solved by the use of a bearing material produced by Thordon Bearings.

A hydraulic turbine owner in South America experienced persistent guidevane bearing problems at its hydroelectric station. The root cause was a result of excessive edge loading which had been evidenced by increased friction force and excessive wear, and the wear pattern of guidevane bearings. The powerplant houses two 230 MW medium head (99 m), vertical Francis turbines operating 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 guidevane bearings with a non-metallic, homogeneous polymer. It uses a practical method to calculate edge loading on the bearings, as described earlier, and explains in details why non-metallic polymer materials such as ThorPlas® are more suitable, and able to handle edge loading better than stiffer metallic bearing components. An excellent performance result was obtained from an eight-month trial conducted in early 2008 by the owner, using ThorPlas®, which eventually had led them to replace all existing bearings in April of 2009 for one turbine, followed by a retrofit of the second turbine in April 2010.

3.1 Determination of the shaft slope at the bearings

The shaft slope at the bearings was calculated using the analytical method described by N.N. Kovalev [1965[1]] and the result was compared with the finite element analysis [Endesa 2007[3]]. The maximum load used for the calculation was 10 bar hydraulic pressure acting on the guidevanes in combination with a 160 kN operating link load. For investigation purposes these loads are considered to operate in the same plane and in opposite directions. Fig. 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 centres. There are three bearings, known as the lower, intermediate and upper bearing.
- The moment of inertia of guidevane was calculated in the weakest bending resistant direction.
- The 3 bearings (Lower, Intermediate and Upper) are ideally aligned
- Bearing surface material is homogeneous elastic

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

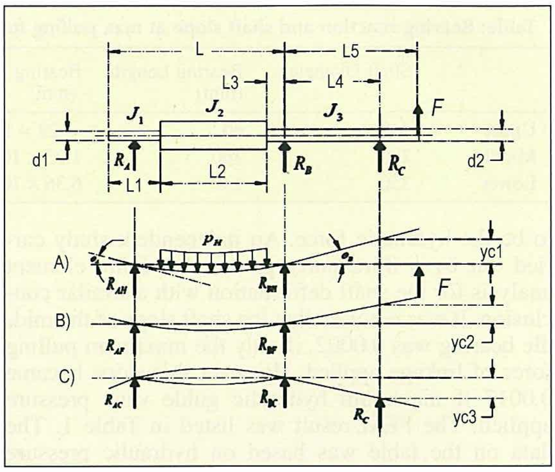

Fig. 3. The loading diagram.

- Loading Case (a): Loaded by hydraulic pressure on guidevane only;
- Loading Case (b): Loaded by gate ring pulling force only; and,
- Loading Case (c): Loaded by reaction force of upper bearing R_c.

Now each individual loading case is statically determined. After calculating each case and adding all the results together, one can obtain the reactions and shaft slope of the entire system [N.N. Kovalev, 1965[1]].

In loading case (c), there are three unknowns, including bearing reaction R_c. The two-dimensional system, however, can only solve two unknowns, and thus one more equation is required to solve all the variables. Shaft deflection at the upper bearing is chosen as such a variable. This means all reactions for the lower, middle and upper bearings can be expressed as a function of deflection at the upper bearing. It is designated y_c. According to Fig. 3:

$$y_C = y_{C1} + y_{C2} + y_{C3} \qquad \text{... (9)}$$

The final value of y_c will be equal to the running clearance of the upper bearing, which is a known design parameter. Now there is sufficient information to solve the problem. Without going into detail about the solving procedure, the next section of this paper is a discussion of the results.

Fig. 4 shows the calculated shaft slope at the lower and middle bearing. The pulling force of the linkage arm applied at the top end was assumed to be the maximum. It is worth noting that neither the pulling force of the linkage nor the clearance of the upper bearing has a significant effect on the shaft slope of the lower and middle bearings. The dominant factor was found

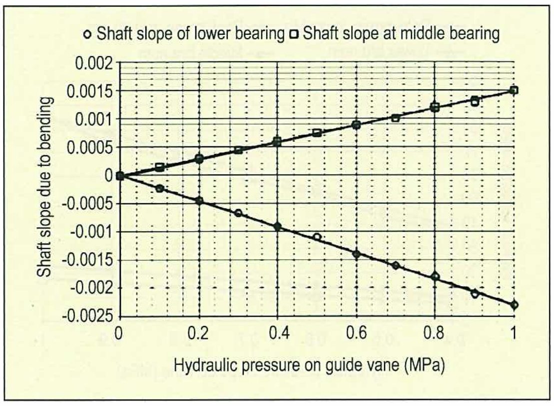

Fig. 4. Shaft slope for the middle and lower bearings.

Table: Bearing reaction and shaft slope at max pulling force and hydraulic pressure						
	Shaft Diameter	Bearing Length (mm)	Bearing Reaction (mm)	Shaft Slope (N)	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

to be the hydraulic force. An independent study carried out by a third party provided a finite element analysis for the shaft deformation with a similar conclusion. It was reported that the shaft slope of the 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 guidevane 1.0 MPa) and the pulling load of linkage applied to top end of shaft = 160.5 N). The pulling force of this turbine had been increasing in operation in the history. This was suspected because of the damaged bearing with increased friction.

3.2 Maximum edge pressure

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

The ThorPlas® bearing was chosen for this application. It was an optimal choice from balancing many factors. Since the shaft slopes at the bearings are relatively large, it is absolutely necessary to choose a material that is able to mitigate edge loading. The material must also have the capability to minimize the clamping force. The wall thickness is restricted to 5 mm by the existing space. The material has to maintain an appropriate interference force to keep within the housing. Low friction and low wear rate material are basic requirements and the material must be grease free as well. According to the reaction forces obtained above, the nominal bearing pressure at the lower and middle bearing is quite low. The average pressure over the actual contact area (a half ellipse) is not excessively high either. However, the peak pressure at the very edge of the bearing is much higher than the nominal pressure. Based on Fig. 5, the peak pressure is about four times higher than the 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 the nominal in the same conditions. To reduce the peak pressure at the very edge, all ThorPlas® bearings for this application were machined a chamfer at both ends. In effect, this has changed the bearing edge towards the inside of the bearing. The estimation of maximum bearing pressure was from shaft bending perspective. However, the deformation of supporting structure of the guidevane shaft, the misalignment of shaft from installation, fluctuation of hydraulic pressure on the guidevane, all these factors that might make a precise determination of the true maximum bearing pressure extremely difficult.

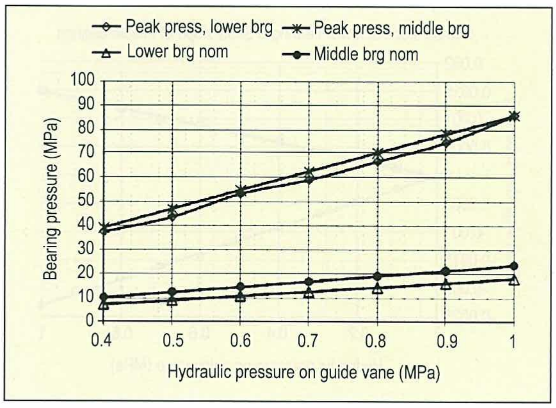

Fig. 5. Peak and nominal bearing pressure (MPa).

3.3 Installation and operational results

The trial installation of the bearings for two gates in one of the two turbines was 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 or wear on the bearing surface. The customer was extremely satisfied with this exceptional result. By the end of 2008, the customer had ordered 29 bearing kits including seals to protect abrasive substances from entering the lower and middle bearings. By 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 servomotor. The servo system pressure was kept constantly at the expected level, and with no damage and leakage observed. In December 2009, the customer reported that all bearings were operating well. The photograph below was the lower bearing after eight months operation.

4. Summary

In this case study, by an analytical calculation of the reaction forces and the shaft slope at the bearings, it was identified that an insufficient stiffness of guidevane shaft was the dominant factor contributing to the high edge loading and excessive wear on guidevane bearings. The real problem, however, is that the edge

A lower bearing after eight months of operation.

loading caused by shaft bending cannot be corrected just by careful installation. The problem only shows up when the turbine begins operation. The only way to mitigate the problem is to choose a suitable material with an appropriate bearing design, or to stiffen the shaft and guidevane as suggested by the third party analysis [2007[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 the bearing surface to deflect slightly 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. ThorPlas® bearings are made of new homogeneous polymer produced by Thordon Bearings, with a very low coefficient of friction. It is easy to machine and install. The typical designed nominal pressure for ThorPlas® bearings is 31 MPa. It can be concluded that factors such as identification of a problem, finding the root cause, selecting the right material, using a suitable design and careful manufacturing and installation all are the keys to a success project. ◊

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References

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

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