TCO optimization during design phase – assessment of bearing concepts by calculation and simulation

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Abstract. Energy efficiency has been a key issue for screw machines for many years. This article describes a new analytical procedure for determining rolling bearing friction and the use of the procedure for designing compressor bearing supports. Further analytical options for a calculative assessment of the overall efficiency of bearing support alternatives are presented in the design phase based on parameter analyses performed by the BearinX® calculation software, which includes the new friction calculation by means of a mechanical, tribological model.

1. State of the Art, Determining Bearing Friction Empirically
Energy efficiency has been a key issue for screw machines for many years. Operating and energy costs make up a considerable percentage of the total cost of ownership (TCO) and therefore are increasingly coming to the fore for operators.
In addition to intelligent control and drive technology, many compressor manufacturers are working intensely to increase the efficiency of the screw group itself. An optimized rotor and gearbox bearing support offers good savings potential.
Primarily rolling bearings in a great variety of bearing designs and arrangements are being used to provide bearing support in screw machines.

Previously, the selection of bearings was mainly determined according to the following aspects:
- Reliability and long operating life
- Available installation space, bearing rigidity
- Ease of assembly
- Bearing cost and availability
- Experience from existing solutions

Established and internationally standardized calculation methods for bearing operating life (ISO 281 [1]) are on hand for optimizations designed to increase productivity.
To date, it has not been possible to sufficiently assess friction and power loss due to relatively imprecise calculation formulas.
Bearing manufacturers offer empirical calculation models that enable calculative estimations based on the load, speed, and operating viscosity of the lubricant and bearing design. The formulas are based on research by Palmgren in 1957 (Figure 1) leading to the ISO 15312 [2]. They were derived from measuring results in tests with laboratory test stands. However, they provide approximations of the measuring results and are only sufficiently precise in narrow ranges. The friction calculation of individual bearing arrangements for specific operating conditions is therefore only possible on a very rough scale.

2. Determining Bearing Support Friction Analytically
In contrast to the empirical determining process, an analytical calculation begins with a mechanical, tribiological model of a rolling bearing. The mechanical model serves to precisely map force application, load distribution in the bearing, shaft deflection and skew position, operating clearance, the internal bearing structure, and more. The linked tribological model describes the behavior of the different tribological phenomena. Figure 2 shows the most important friction components of a rolling bearing.
Unlike with the empirical method, factoring in these friction components results in the addition of fundamental parameters such as those shown in Figure 3.

![Figure 3: Rolling bearing friction, influencing parameters](image)

The BearinX® calculation software uses this kind of mechanical, tribological model to calculate the friction of rolling bearing supports. While BearinX® is a quasi-static model that offers fast calculation times and high accuracy, a dynamic simulation with the CABA3D calculation software provides enhanced analysis possibilities with additional consideration of cage friction, rolling element contact, and rolling element skewing. A comparison of the calculated friction coefficients and those measured in the test confirms a high degree of conformity and thus the quality of the procedure developed (Figure 4).

![Figure 4: Comparison of the measured and calculated friction torque force](image)

Figure 4 clearly shows that the empirical method according to Palmgren only provides realistic friction coefficients in a narrow range (approx. 1 to 5kN here). However, the friction calculated using BearinX® is so precise that it lies within the measuring value spread established in the test (0.5 to 15 kN).

3. Extended Application Analyses
The calculation procedure integrated into the BearinX® bearing design program offers a powerful tool for the individual modeling and analysis of bearing supports.

BearinX® makes it possible to calculate individual bearings and also to model shaft systems and gearboxes while allowing for elastic shaft systems. For instance, in the process, it is possible to investigate details of the individual rolling contact, lubricant film formation, bearing rigidity, shifting, rigidity in the operating points, and even the dynamic analysis of natural vibration behavior (Figure 5).

![Figure 5: Modular calculation with BearinX®](image)

So-called parameter analyses can be compiled for calculation models. To do this, a user-defined number of model input parameters is given with an initial value and a final value. The program then calculates independently specified output parameters and graphs their change as a function of the input parameters.

For example, a typical analysis would be the operating clearance of a bearing as a function of component temperature with the specified shaft and housing materials/seats and bearing play class. In the design phase, these parameter analyses are used to investigate the parameters’ effects on the command variables being analyzed.

It becomes clear in the process which parameters have the greatest effect. They can then be systematically altered so that they change the overall system behavior in the desired direction. This frequently makes it possible to avoid or at least reduce complicated, expensive, and time-consuming tests.

3.1. Bearing Friction Analysis:
At the outset of friction analysis, it is interesting to see what kinds of friction the various types of bearings produce.

For instance, the bearing friction analysis for various types of bearings as a function of the induced force as shown in Figure 6 is helpful in this respect. Instead of individual bearings, bearing combinations were selected for this, of the kind frequently used on the pressure side in screw compressors.

The profiles confirm the known friction behavior of the various bearing types. Here, tapered roller bearings generate the greatest friction forces, whereas angular contact ball bearings and four point contact bearings exhibit considerably lower levels of friction.
3.2. **Bearing Rigidity:**
The radial and axial bearing rigidity also represent important parameters.
The following figure shows the axial shift of the various alternatives as a function of the axial bearing load.
The greatest axial rigidity levels for the tapered roller bearing support are given here. The ball bearing solutions exhibit lower levels of rigidity.

![Figure 6: Bearing friction depending on the load](image1)

**Figure 6: Bearing friction depending on the load**

The radial rigidity depends on the type and size of the bearing as well as on the radial bearing play. Figure 8 indicates the radial rigidity of an NU308-JP cylindrical roller bearing.

The radial bearing play has been reduced here between CN, C3, and C2 in stages. The graph shows that the bearing spring deflection can be reduced by restricting the operating clearance. It is necessary to conduct a temperature analysis, particularly in the case of restricted operating clearance, to prevent inadmissibly high bearing preload levels due to thermal expansion.

![Figure 7: Parameter analysis of the axial rigidity](image2)

**Figure 7: Parameter analysis of the axial rigidity**
3.3. Bearing Kinematics at High Speeds

For high-speed screw machines such as oil-free compressors, it is essential to factor in the bearing kinematics in addition to questions of clearance and operating life. The speed coefficients (nxdm) typically range between 600,000 mm/min. and 1.5 million mm/min. Centrifugal forces result in unfavorable spin/roll conditions that produce large sliding movements. In favorable lubrication conditions, these sliding movements lead to increased frictional heat or, in mixed friction conditions, to increased fretting. The frictional heat produced must be conveyed away from the contact through respectively high quantities of oil. These oil quantities, in turn, lead to increased churning losses, which once again reduce efficiency. Suitable measures must be adopted to guarantee a minimum bearing load for all operating conditions to avoid negative circumstances and ensure a sufficient operating life.

Parameter analyses of the specific model make it possible to very accurately predict the conditions at which the bearing kinematics will begin to be adversely affected.

Figure 9 below shows that the current bearing configuration exhibits considerable spin/roll effects as of approx. 10,000 rpm, corresponding to a speed coefficient (nxdm) of approx. 435,000 mm/min. These effects may be reduced to admissible levels by increasing the axial preload accordingly or by using ceramic balls, for example.

The use of range springs in the calculation model serves to identify shifting at any position in the model. For instance, this makes it possible to analyze in detail the effect of centrifugal force on the axial skew of the face surfaces with angular contact ball bearings, as shown in Figure 10. Not taking this into due consideration can lead to rotor/housing contact at high speeds.
4. Example of a bearing optimization of a twin screw compressor HP side

Target is to optimize the bearing arrangement of a twin screw compressor on the High Pressure (HP) side regarding friction and life rating.

Initially the radial loads are carried by a radial Cylinder Roller Bearing (CRB) and the axial loads are carried by a Taper Roller Bearing (TRB). Basic compressor data and the BearinX model of the bearing set up is shown in the following slide (Figure 11).

TRB offer high axial load capacity. Due to the internal design of TRBs an axial load to the tapered rollers is generated by axial or radial loads to the bearing. This axial force is transferred from roller to rib of the related bearing ring producing sliding friction. Bearings without this kind of sliding friction (e.g. Angular Contac Ball Bearings (ACBB)) create smaller friction.
Figure 11: Basic compressor parameters and BearinX model

Figure 12 shows the cumulated calculated friction for this specific arrangement depending on the shaft speed.

The TRB arrangement shows significant higher friction than ACBB arrangements, which even grows with higher speeds. But aside reduced friction the alternative bearing arrangement needs to reach comparable and sufficient life times as the initial bearing solution. Here the modified reference rating life $L_{hmr}$ offers the best calculable life time to judge load, contamination and lubrication.

The table in figure 13 gives an overview regarding life times, friction and contact pressure for each bearing alternative at nominal conditions of the compressor.

With pressures below 1000 N/mm² the initial solution shows low bearing loading. This results in very high calculated bearing life ratings with values $> 650,000$ h. With life time target of 40,000 h at failure rates $< 1\%$ the calculated life ratings should show values above 160,000h.

For the CRBs values like this are reached by using bearings with the same shaft and housing diameter, but reduced width.

For the first alternative the TRBs are replaced by ACBBs size 7319-B in tandem arrangement. The life time and pressure values reveal that even a single bearing arrangement is possible. A single 7319-B
still reaches life ratings > 570,000 h. So a friction reduction of 1200W which is about 48% of the initial value is possible.

In comparison to the TRB 31319 the ACBB 7319-B reaches smaller axial stiffness. So the shaft displacement will be bigger at nominal loads which results in a bigger axial clearance between rotor and housing. The calculated displacements are shown in figure 14. The 7319-B tandem arrangement nearly reaches the TRB displacement, but the single 7319-B displacements are increased by ~20 µm. Here the compressor manufacturer needs to judge the efficiency reduction based on this.

Due to centrifugal forces to the balls single ACBB or ACBB in tandem without counter ACBB produce a shaft displacement depending on axial load and speed. This shaft displacement reduces the axial clearance between rotor and housing.

For high axial load conditions the displacement is small. For low load conditions the displacements may reach values where the rotor gets in contact with the housing.
Figure 15 shows the behavior at idle mode, where just 1500 N axial load is generated. According to the BearinX parameter analysis the displacement at 3000 rpm is about 0.25 mm for a single 7319-B. The tandem arrangement even produces bigger displacements in the range of 0.35 mm. So it is necessary to carry out a test at idle mode to check whether the oil lubrication is sufficient to separate the axial contact surfaces rotor/housing while idle operation. Alternatives to prevent this rotor displacement are to use an additional ACBB in X arrangement or to use a 4 point contact bearing (QJ series).

![Figure 15: axial displacement at idle condition depending on shaft speed.](image)

As final result the analysis reveals possible friction reduction of 1200W which offers savings of energy costs about 3800,- €.

These savings are based on energy costs of 0.1 €/kWh and 40.000 h operating time by taking 20% idle mode into account.

5. Summary / Conclusion:
In order to assess the overall efficiency of a compressor, aspects besides friction, such as rigidity, temperature behavior, and bearing kinematics, need to be considered. Calculation software with a mechanical, tribological model enable calculative, comprehensive preliminary investigations during the design phase, in this way presenting decisive optimization options already at this stage and leading compressor manufacturers to the sought-after solution more quickly.

References
[1] ISO 281 Rolling bearings — Dynamic load ratings and rating life
[2] ISO 15312 Rolling bearings — Thermal speed rating – calculation and coefficients