1998

Theoretical and Experimental Evaluation of the Friction Torque in Compressors with Straddle Bearings

R. Puff
Embraco - Empresa Brasileira de Compressores S/A

A. L. Manke
Embraco - Empresa Brasileira de Compressores S/A

H. R. Seibel
Embraco - Empresa Brasileira de Compressores S/A

Follow this and additional works at: https://docs.lib.purdue.edu/icec/1233

https://docs.lib.purdue.edu/icec/1233
THEORETICAL AND EXPERIMENTAL EVALUATION OF THE FRICTION TORQUE IN COMPRESSORS WITH STRADDLE BEARINGS

Rinaldo Puff
Adilson L. Manke
Hugo R. Seibel
Embraco – Empresa Brasileira de Compressores S/A
Rua Rui Barbosa 1020, Cx.P. 91, 89219-901, Joinville – SC – Brazil

ABSTRACT

The assembling of the mechanical pump of compressors with straddle bearings requires the alignment of the secondary with relation to the main bearing. When the misalignment between these two components reaches a limit value, high increase of the mechanical losses can be observed. The evaluation of the bearing alignment, after the pump is assembled, is difficult to be done. One manner to verify the correct alignment of the bearings is by measuring the resistance torque, with the use of a torque transducer. The measured torque is directly related to the alignment of the bearings, and this information can be used as a control item for the pump assembly acceptance.

A theoretical and numerical evaluation was done using a simplified formulation for misaligned bearings, together with the evaluation of assembled pump friction torque. The simplified model was compared to the results obtained with a finite bearing simulation model. The numerical results obtained with the simplified formulation, compared to experimental data, were used to define the torque limit for the acceptance of the pump assembly.

NOMENCLATURE

- $h$ = Oil film thickness
- $c$ = Bearing radial clearance
- $L$ = Bearing length
- $D$ = Bearing diameter
- $R$ = Bearing radius
- $Y$ = $y / R$
- $D_m$ = Misalignment degree = $\alpha / \alpha_{max}$
- $\varphi$ = Angular position on bearing
- $\alpha_{max}$ = $\tan^{-1}(2c / L)$
- $\mu_o$ = Oil viscosity
- $\omega$ = Shaft driven velocity
- $T_{rf}$ = Piston faces friction torque
- $R_r$ = Piston external radius
- $R_{ecc}$ = Eccentric radius
- $\omega_r$ = Piston angular velocity
- $c_{rf}$ = Piston faces clearance
- $\alpha_{mc}$ = Minimum clearance
- $T_{mc}$ = Minimum clearance friction torque
- $F_{mc}$ = Minimum clearance force
- $\beta_{mc}$ = Minimum clearance angle
- $R_{cyl}$ = Cylinder radius
- $H_{cyl}$ = Cylinder height
- $I_r$ = Piston Inertia Moment
- $T_{re}$ = Piston x Eccentric friction torque
- $T_{mb}$ = Main bearing friction torque
- $T_{sb}$ = Secondary bearing friction torque
- $T_{ar}$ = Piston translation friction torque
INTRODUCTION

In the assembly line of compressors with straddle bearings, one important factor for the correct run of the mechanical pump is the good alignment of the secondary bearing with relation to the main one. Between the two bearings an intermediate part is assembled, such as a piston-vane-cylinder in rotary, or a connecting rod in reciprocating compressors. The objective is to achieve an assembly with the minimum shaft x bearing frictional torque.

One way to achieve a good alignment between the two components is to assemble the secondary bearing with the shaft in rotation, in the presence of setup oil. In the process, a fixture supports the main bearing and keeps its axis in a vertical position. The shaft is assembled with the main bearing. The intermediate part is assembled, and the secondary bearing is assembled over the intermediate part. Setup oil is applied to the components.

The shaft is driven in rotation. Hydrodynamic forces bring the shaft to an approximately coaxial position with relation to the main bearing. At the same time, the hydrodynamic forces that are present in the secondary bearing, bring this component to a position that is approximately concentric with the shaft rotation axis. In this situation the main bearing, the intermediate part and the secondary bearing are fixed by a clamping device and screwed together. As a result of this operation, both bearings are approximately concentric. The final verification is done using a torque transducer, in order to approve or reject the assembly.

The main problem with this method is to correlate the measured torque with the bearing alignment. Once assembled, it is quite impossible to measure the relative position between the bearings. The objective of this article is to perform a numerical analysis of the friction torque behavior in the presence of bearing misalignment, define a numerical torque limit, and compare it with measured results.

BEARING DYNAMIC TORQUE CONSIDERING SHAFT INCLINATION

Figure 01 shows the situation of the shaft inclined with relation to the bearing. This occurs when we have a bearing misalignment. According to Shigley (1986), the product of the friction force and the bearing radius results in the frictional torque. The friction force is inversely proportional to the bearing radial clearance. In our case, due to the shaft inclination, we do not have a constant clearance. It is function of the geometrical parameters (y, φ and E₀).

In Pinkus et al (1979), we can find the correlation for \( h(y, \phi, E₀) \).

\[
h = c \left[ 1 + E₀ \cdot \cos(\phi) + \frac{D_m(1-E₀)}{(L/D)} \left( \frac{L}{D} - Y \right) \cos(\phi) \right]
\]

Assuming that we have only shaft inclination, with \( E₀ = 0 \), and an infinitesimal area element \( dA = d\phi.dY \), the infinitesimal torque \( dT \), is given by:

\[
dT = \frac{\mu \omega R^2}{h(\phi, Y)} \cdot d\phi.dY
\]
Integrating numerically this equation using \((0 \leq \varphi \leq 2\pi)\), and \((0 \leq Y \leq L/R)\), as limits, we can obtain the total friction torque \(T\), for each bearing.

Due to the fact that the loads applied to the bearings are extremely small during the assembling of the secondary bearing, it is expected that the results obtained with the above equation will be satisfactory, once the shaft runs inclined, but concentric with the bearing. To confirm the results of this equation, they were compared with data obtained with a bearing simulation program developed by Manke et al (1992). The graphic on figure 02 shows this comparison which was done using power and not torque results.

It can be observed that for low inclination, the difference was lower than 2.5 %. For higher inclination, it can be observed a difference not higher than 7.0 %.

For the objective of this simplified formulation, to obtain pump acceptance limits, such differences are acceptable.

APPLICATION TO ROLLING PISTON COMPRESSOR PUMP ASSEMBLING

Rolling piston compressors typically use straddle bearings, and we will use such concept to evaluate the results obtained with assembling process results.

Bearings friction torque is obtained with the use of the simplified formulation for inclined bearings. It is necessary to obtain the piston angular velocity, which is a consequence of the torque balance on this component. Figure 03 shows this balance when the pump is closed. The torque between piston and eccentric \(T_{re}\) is computed with the simplified formulation. Once we have the bearings misaligned, and as a consequence, the shaft inclined inside the bearing, the same will occur with the piston x eccentric. The torque between piston upper and lower faces and bearings \(T_{rf}\), according to Krueger (1988) are given by:

\[
T_{rf} = \frac{2\pi \mu \omega_r R_{s}^4}{c_{rf}} (R_{s}^4 - R_{mc}^4) \omega_r
\]

The friction torque at the minimum clearance is given by:

\[
T_{mc} = F_{mc} \cdot R_r
\]

Where:

\[
F_{mc} = \frac{\mu \beta_{mc} R_{cyl}^2 H_{cyl} \left[R_{cyl}^3 \omega + R_r (\omega_r - \omega)\right]}{c_{mc}}
\]

And finally, the piston angular velocity is given by:

\[
\omega_r = \omega_r + d\omega_r
\]

Where,

\[
d\omega_r = \frac{(T_{re} - T_{rf} - T_{mc})}{I_r} \, dt
\]
Figure 04 shows the behavior of the piston angular velocity during one turn of the shaft. In the computation of this velocity, it is considered the offset assemblage, very common in rotary compressors.

The total torque of the pump is given by:

\[ T_{pump} = T_e + T_{mb} + T_{rb} + T_{ar} \]

where

\[ T_{ar} = \frac{2\pi \mu \left( R_f^2 - R_{ecc}^2 \right) ecc^2 \omega}{c_r} \]

RESULTS AND DISCUSSION

A fortran 77 program was developed in order to compute the frictional torque. Considering specified setup oil viscosity and shaft driven velocity for the process, we obtained the results shown in the chart of figure 05. The graph presents the pump friction torque as a function of the percentage of bearings misalignment. The chart shows us that misalignments until 50% of the maximum range are admissible on the process.

To compare simulation results with process data, pumps with the same clearance range considered in the numerical results, were assembled on the process, and friction torque was measured and compared to the numerical limit. The limit of 50% of the maximum misalignment range, which gives a torque of 23.0 N.cm, was considered as the approbation limit.

Figure 06 presents a graph with the comparison of two sets of data with the torque limit specified using the program. The first set of data shows torque measurement of the assembled pump after alignment operation was done. We can see that all results are below the torque limit. That yields us to the conclusion that bearings were well aligned.

The second set of data were measured applying a radial force over the secondary bearing, in order to misalign the two bearings, and then measure the friction torque. It can be seen that torque results are all above our imposed limit, showing that bearings are clearly misaligned.

Finally, the statistics graph shown in Figure 07, shows the results for a set of 15 pumps, assembled on the process, considering specified torque limit. It can be observed that pump torque measured on process, showed acceptable results, yielding to the acceptance of the assemblies.

CONCLUSIONS

- The simplified formulation showed to be a good tool for the specification of the torque limits for the pump assembling process.
- For the process, a misalignment degree of 50 % of the maximum possible, showed to be a good limit. For higher misalignments, greater inclinations of the shaft occur inside the bearing, and the friction torque becomes more sensible. This can be explained by the fact of having very small clearances in the extremity of the bearing, tending to the metallic contact. As torque is inversely proportional to the clearance, these very small clearances result in higher friction in these regions.
- In this analysis, there was not considered form errors present in the parts, as well as the roughness and oil properties variation. This can explain measured torque values higher than
the maximum numerical obtained. If these machining irregularities were considered in a more powerful formulation, higher accuracy would be obtained. Otherwise, this was not the objective of the work.

REFERENCES


FIGURES

Figure 01 – Schematic view of the inclined shaft inside the bearing.

Figure 02 – Comparison between simplified method and finite bearings simulation program.

Figure 03 – Schematic view of Piston x eccentric x cylinder interaction.
Figure 04 – Piston angular Velocity x Shaft angle

Figure 05 – Pump friction torque x Misalignment degree

Figure 06 – Measured results x specified torque limit

Figure 07 – Process statistical results