DESIGN OF RECIPROCATING
COMPRESSOR CRANK MECHANISM PLAIN BEARINGS

BY A NEW DEDICATED SOFTWARE — PART II

The study was conducted on a typical case for the uprating of a medium-speed, medium-sized reciprocating compressor

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Editor’s Note: This is the second part of a technical article by the authors. The first part, “A Practical Mathematical Model For The Design Of Reciprocating Compressor Plain Bearings,” was published in the October 2011 issue of COMPRESSERTech®. For additional information or answers to questions, contact the authors at their e-mail addresses: renzo-ciuffi@virgilio.it, andrea.fusi@cstfirenze.com or alessandro.rossi@cstfirenze.com.

The BearingPerf software developed by Compression Service Technology (CST), based on the Mobility Method, analyzes both squeeze and rotation effects and enables evaluation of the most significant bearing performance parameters. These include minimum oil film thickness, maximum pressure, maximum temperature and lubricant feed hole locations of dynamically loaded crank mechanism bearings. In the following, some applications are described to show how the software works and how the significant performance parameters are influenced by input design data variations, clearance, load profile, speed, etc.

The study was conducted on a typical case for the uprating of a medium-speed, medium-sized reciprocating compressor. All the bearings were checked and redesigned. The study considered two operating conditions: the first giving the maximum load on the crank mechanism, and the second giving the minimum load reversal on the crosshead pin, Load Conditions 1 and 2 of Figure 1, respectively.

Load Condition 1 was applied to all the three bearings — main, big end, crosshead pin bushing — while the second was applied only to the crosshead pin bushing, whose performance is affected when the load reversal is inadequate. As stated in the API 618, 5th edition, paragraph 6.6.4, “The duration and magnitude of this reversal shall be consistent with the oil distribution design of the crosshead bushing in order to maintain proper lubrication.”

Other input data for this study were: the 600 rpm rotation speed of the compressor; the oil was an ISOVG220; the oil inlet pressure, 50.7 psi (3.5 bar); and the oil inlet temperature, 122°F (50°C).

Crosshead Pin Bushing

First to be analyzed was the original crosshead pin bushing design with Load Condition 1. Figure 2 and Figure 3 show the results for the oil film thickness vs. the crank angle in Cartesian and Polar coordinates, respectively.

Figure 1. Crosshead pin load vs. crank angle: Load Condition 1 (black solid line) and Load Condition 2 (red dashed line).

Figure 2. Crosshead pin, original design, Load Condition 1: normalized film thickness (red dashed line) and normalized pressure (blue solid line) vs. crank angle for the last two cycles calculated.

Figure 3. Crosshead pin, original design, Load Condition 1: polar diagram of film thickness [m] and eccentricity ratio vs. crank angle.
Figure 2, in particular, reports both the normalized oil film thickness and pressure for the last two cycles calculated by the software. One observes that the curve relevant to the first 360° is almost identical to that of the following 360°, which means that the convergence was reached. An appreciable delay between minimum oil film thickness and maximum load can be observed, due to the inertial effect.

The Polar diagrams of Figure 3 respectively show the oil film thickness and the normalized eccentricity ratio vs. the crank angle. These diagrams are useful to immediately identify the most critical crank angle intervals, where the oil film thickness is lower than the minimum acceptable value and the eccentricity ratio is close to 1. In these conditions, in fact, the oil temperature increases rapidly and, consequently, the oil viscosity decreases with further reduction of the bearing load capability. The minimum film thickness value calculated is approximately 2 µm: too small to guarantee the correct operation of the compressor. The authors’ experience with the subject machines suggests having an oil film thickness, calculated with this method, possibly greater than 5 µm.

The above diagrams confirmed that the original crosshead pin bushing was not able to work correctly in the new operating conditions. Therefore, the bearing was redesigned introducing several modifications including the following: bearing internal diameter increased to 3.54 in. (90 mm), minimum radial clearance decreased to 0.0012 in. (0.0315 mm).

The software was re-run for the new configuration, obtaining acceptable values for the minimum oil film thickness (6 to 7 µm) and the eccentricity ratio (lower than 0.8), as it can be seen in Figure 4 and Figure 5. Figure 6 and Figure 7 show a comparison of the above parameters between original and improved designs.

For the improved design, the case of minimum load reversal (Load Condition 2) was also studied by means of the new software. Figure 8 compares the minimum oil film thickness for both Load Conditions 1 and 2. As shown, the improved design can guarantee, also in this case, an acceptable performance in terms of minimum oil film thickness 0.2 in. (>5 µm) even if Load Condition 2 results heavier than the 1 since the bushing works with low film thickness for a much longer period. Also, the oil pressure peak calculated with Load Condition 2 is significantly higher than that calculated with Load Condition 1 8702 vs. 5802 psi (600 vs. 400 bar).

Finally, to better withstand the minimum load reversal situation, the bushing internal grooving was modified introducing the concept reported by D.F. Wilcock and E.R. Booser for two-cycle engines and shown in Figure 9.
**Conrod Big End Bearing**

Figures 10, 11 and 12 show oil film thickness, eccentricity ratio and oil pressure vs. the crank angle for the original bearing in Load Condition 1. The minimum film thickness calculated is approximately 10 µm and the eccentricity ratio is slightly greater than 0.8. The bearing was not modified in its main dimensions because these values were judged to be sufficient to guarantee the correct operation also in transient conditions.

![Figure 10. Conrod big end, Load Condition 1: film thickness [µm] vs. crank angle.](image)

![Figure 11. Conrod big end, Load Condition 1: polar diagram of film thickness [m] and eccentricity ratio vs. crank angle.](image)

![Figure 12. Conrod big end, Load Condition 1: oil pressure [bar] vs. crank angle.](image)

**Main Bearings**

Figures 13 and 14, which report the software output diagrams for film thickness and eccentricity for the main bearings, show that there was an acceptable situation for these bearings.

![Figure 13. Main bearing, Load Condition 1: film thickness [µm] vs. crank angle.](image)

![Figure 14. Main bearing, Load Condition 1: polar diagram of film thickness [m] and eccentricity ratio vs. crank angle.](image)

**Sensitivity Analyses**

The CST–BearingPerf software is very useful to perform sensitivity analyses to determine the effect of the variation of a design parameter on the performance of the bearings.

Figures 15 through 20 show the results of two typical sensitivity analyses, done for the minimum oil film thickness of the three types of bearing (crosshead pin, conrod big end, main) of a small compressor.

![Figure 15. Crosshead pin bushing: minimum film thickness vs. crank angle for different speeds (600 rpm solid blue line, 1000 rpm dashed green line, 1500 rpm dashed red line, 1800 rpm dot-dashed black line).](image)

![Figure 16. Conrod big end bearing: minimum film thickness vs. crank angle for different speeds (600 rpm solid blue line, 1000 rpm dashed green line, 1500 rpm dashed red line, 1800 rpm dot-dashed black line).](image)
In Figures 15 through 17, the clearances are kept constant at nominal value and four different values of rotational speed are considered (600, 1000, 1500 and 1800 rpm). In Figure 18 through 20, the compressor rotational speed is kept constant at the nominal value (1500 rpm) and four different values of the clearance are considered for each bearing type.

Figure 15 shows that the minimum oil film thickness of the crosshead pin bushing has acceptable values for speeds of 1500 and 1800 rpm while it becomes critical for a speed of 1000 rpm and even more of 600 rpm. Note that the new software calculates the bearing lubrication parameters considering the load variation as a function of speed. As a matter of fact, the software takes account of the variation of the contribution to the total load of the inertia forces as well as of the valve losses when the speed varies.

Figure 16 shows the effect of speed variation on the conrod big end bearing. One can see that the minimum oil film thickness has marginal values at 600 rpm, it starts having acceptable values at 1000 rpm, it further increases at 1500 rpm but it decreases at 1800 rpm. Figure 17 shows the behavior of the main bearing with a clear reduction of the minimum oil film thickness when the speed decreases. In fact, 600 and 1000 rpm show very low thickness values.

As far as the oil film thickness variation as a function of the clearance (see Figures 18 through 20), as expected, for all types of bearings, the minimum film thickness increases when the clearance decreases.

The above sensitivity analyses are a very useful and a practical tool both during the design stage of new machinery and in case of applications with variable operating conditions. One example can be the underground gas storage where reciprocating compressors are required to run with variable speed to adjust the throughput to the variable reservoir requirements.

Conclusions

The authors developed a user friendly program (CST-Bearing Perf), in a Matlab environment without any add-in modules, to apply the “mobility iterative method” for the design of plain bearings for the crank mechanisms of reciprocating compressors and other reciprocating machinery. Dynamic input loads, variable in amplitude and phase angle, can be considered.

The values of minimum film thickness, maximum oil pressure and temperature, and minimum oil pressure points can be easily calculated so that the effects of modifications in the design and in the operating conditions can be promptly evaluated, thus enabling useful sensitivity analyses. The software can be used for the design of new units or for the revamping of existing units and also to evaluate the bearings performance according to the applicable regulations.

Calculations with input data taken from operating compressors gave results consistent with the experience, and equally good results were obtained by comparing the output of the new software with examples taken from the literature.

More advanced tools can be used whenever the approximation required is of a superior order of magnitude, for instance when the structural deformation of the bearing has to be taken into account. In those cases fluid structure interaction (FSI) software, based on structural finite element analysis (FEA) and computational fluid dynamics (CFD), should be used.

References