CHAPTER 5

FAILURE ANALYSIS OF MAIN BEARING

5.1 BEARING LAYOUT

The two main bearings are used in the crankshaft assembly, one bearing is at the drive end side of the crankshaft and another bearing is on the non-drive end side. There was a failure observed in one of the industrial units, it was analysed to finalize the root cause, and corrective actions are discussed. The compressor layout along with bearing assembly is discussed in the chapter 3.

5.2 FAILURE OBSERVATION

The crankshaft main bearing failure happened in one of the Model-B machine used in the industrial application. The failure has occurred after two months of operation and it had crossed about 300 hours of operation.

5.3 ROOT CAUSE ANALYSIS (RCA) OF MAIN BEARING

Primary physical root causes of failure mainly fall into one of the four fundamental categories like design defects, material defects, defects related to the method of assembly and operational anomalies. The all-possible causes are listed and it was verified in the following section.
5.4 VERIFICATION OF POSSIBLE CAUSES

In order to find the root cause for the failure, all possible causes are listed in the fish bone diagram shown in Figure 5.1 and the same has been validated and verified as follows.

Figure 5.1 Fish bone diagram –Crankshaft main bearing failure
5.4.1 Load bearing capacity of the bearing

The radial load was calculated by considering the maximum gas pressure in HP cylinder and corresponding pressure from the low-pressure cylinder, pulley weight and belt tension. The dynamic load was calculated for 5000 hours of life and it was 55453 N by considering the service factor of 1.2. The selected main bearing (6309) has the dynamic load carrying capacity of 55300 N. The calculated value is slightly above the catalogue value. The static load safety factor is 3.9 and it is estimated from basic static load rating and equivalent static load, this is more than the bearing manufacturer recommendation (≥2). Therefore, the load bearing capacity is not the cause for the failure.

5.4.2 Fit between the shaft and bearing

Initial tolerance given in the crankshaft was k5(45 +0.002/+0.013 mm) and housing (Fly end cover) was K6(100 +0.004/-0.018) and normal clearance class (CN) bearing was selected based on the manufacturer’s catalogue recommendation. The radial internal clearance was worked out based on the fit selected, the temperature gradient between an inner and outer ring of the bearing and reduction in diameter due to smoothing (interference loss) and it varies from –0.035 mm to +0.026 mm.

The tolerance class of the bearing was modified to C3 based on the temperature gradient between inner and outer race of the bearing and to accommodate the manufacturing tolerances of the crankshaft and fly end cover bearing seating bore (bearing housing). The radial operating clearance was calculated by considering all the parameters and it varies from -0.023 mm to +0.039 mm.
5.4.3 Temperature withstanding ability of the grease inside the bearing

The bearing temperature was measured against the ambient condition. In case of continuous duty applications, the bearing temperature had reached to 70°C with an ambient temperature of 45°C. The critical temperature of the lithium soap base grease is 80°C and the safe margin is only 10°C as per bearing manufacturer’s data.

The standard grease for deep groove ball bearing at normal temperature is usually a filtered lithium soap based grease of consistency class 2 with a base oil viscosity of 100 mm²/s at 40°C. The modified grease is with polyurea thickener and synthetic base oil with a continuous operating temperature limit of 110°C and intermittent limiting temperature of 140°C.

5.4.4 More axial load

The deep groove ball bearing is not capable of taking a more axial load during operation. The end float is provided as per the design in the reciprocating compressor to avoid more preload on the bearing in axial direction. The end float was calculated based on crankcase, endcovers (bearing housings), crankshaft dimensions and bearing width and it varies within the specification. Therefore, it is ensured that there was no axial load acting on the bearing due to positive end float.

5.4.5 Duty cycle of operation

Duty cycle of operation is varied depending on the application at customer end. This compressor was designed to run 100% duty cycle, but from the failure data, it was observed that the main bearing failed in the compressor where the duty cycle was more than 80%. Duty cycle determines the bearing operating hours and the operating temperature of the bearing.
5.4.6 Assembly sequence of the bearing

The bearing was pressed into the crankshaft initially with proper tool and then the fly end cover (bearing housing) was assembled into the shaft to reduce the load on the outer ring of the bearing. If it was assembled with fly end cover first, the more load will act on the outer ring. This process has been verified in the assembly line.

5.4.7 Verification of other causes

Apart from the above-mentioned causes, the other related causes are validated and eliminated in a same way as explained in the chapter 3. Therefore, these causes are not contributing the main bearing failure.

5.5 VISUAL OBSERVATION

The abnormal noise was observed in the failed unit during running. The compressor was disassembled and observed that there was no free rotation of the fly end side bearing. Except the main bearing, all other parts were found in good condition. The failed bearing was opened and it was observed that the residue of the grease found in black colour. The inner and outer ring of the failed bearing raceways have shown with axial preload cum splinter indentation.

Figure 5.2 Fresh bearing
The balls were found in the dark brown condition. The fresh bearing raceway is shown in Figure 5.3 for visual comparison.

5.6 HARDNESS OF THE BEARING

The hardness of the inner and outer ring measured and it is varied from 62.5 to 64 HRC. It was well within the specified limit of 62-65 HRC.

5.7 VALIDATION DETAILS

The changes are implemented in two field trial units. The bearing condition found normal and free rotation observed after 11 and 18 months of operation. The one of the field trial units crankshaft main (Fly end side) bearing picture shown in Figure 5.4. Since, number of failures observed was less compared to other types of bearing failures; the validation was also
carried out on the machines where the big end bearing failures happened in the industrial application.

![Main bearing](image)

**Figure 5.4 Crankshaft main bearing after 11 months of validation**

### 5.8 DISCUSSION

It can be observed from the analysis and considering the running hours (300 hours), the cause for the failure is not due to the fit and tolerance class. There is a possibility of misalignment during assembly and due to the lesser temperature margin between the operating temperature of the bearing and limiting temperature of the grease, which accelerates the splinter indentation. The same kind of failure had not happened in the other machines. However, in order to increase the reliability of the bearing, the tolerance class was modified to C3 to increase the internal clearance of the bearing and the lubricant was modified to high temperature grease as like in the connecting rod big end bearings.
5.9 CONCLUSION

From the root cause analysis, it was concluded that the bearing failure was due to misalignment during assembly. However, in order to increase the reliability of the bearing as like in the connecting rod big end bearing, the main bearing was selected with C3 clearance filled with high temperature grease. The modified C3 model bearing was implemented after in-house testing, field validation and no failure has been reported from the application.