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Positive Effect for Using Novel Monitoring System on Reciprocating Compressor

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ABSTRACT

Enhancing asset reliability and optimizing preventive maintenance for reciprocating compressors, which are vital rotating equipment in various industries, particularly within oil, gas, and refinery plants, stands as a pivotal challenge for asset management today. A review of existing literature has shown that numerous development efforts have been made to establish monitoring techniques for reciprocating compressors. Nevertheless, previous research has not delved into the continuous monitoring of the mechanical behaviors and clearances of compressor crankshafts, crossheads, and cylinders. Here, we employ additional proximity probes within an online monitoring system to measure all compressor clearances, aiming to optimize preventive maintenance for reciprocating compressors. K-1101B reciprocating compressor from Egyptian refining company and solid works have been used based on actual compressor mechanical clearances obtained during at 4000, 8000 and 12000 running hour's compressor overhauls to trend the actual changes in clearance during different compressor overhauls. Stress analysis was conducted on the crankshaft journal bearing using SolidWorks contour analysis. The results demonstrated a direct proportional relationship between the Drive End (DE) and Non-Drive End (NDE) static pressure contour and the clearances, indicating that an increase in bearing clearance leads to higher stress on the bearing. Additionally, the findings indicated that implementing an online condition monitoring system is valuable for safeguarding the compressor against repetitive failures. This approach can enhance compressor availability and reduce downtime effectively.

1. Introduction

The reciprocating compressor, also known as a piston compressor, is a positive-displacement machine that employs pistons driven by a crankshaft to deliver gases at high pressures. Its significance in various oil, gas, and refinery processes cannot be overstated. Due to its complex and costly mechanical components, safeguarding this equipment from abrupt failures is paramount. This is typically achieved through preventive maintenance practices as recommended by the manufacturer. However, implementing this maintenance strategy can be prohibitively expensive, often costing three times more than maintenance for centrifugal compressors, and it results in production losses during compressor overhauls [1,2]. This situation has prompted the exploration of innovative systems capable of monitoring all compressor clearances during operation, thereby reducing downtime associated with scheduled overhauls and minimizing the overall cost of compressor maintenance. Condition monitoring, which continuously measures and analyzes relevant mechanical parameters during operation, offers valuable insights into performance through the Pressure-Volume (PV) system and piston rod assessment via the rod drop probe. Vibration analysis of the casing is also employed. However, these methods do not directly provide information about the mechanical condition and clearances of the compressor. Studies conducted by Jim Townsend and M. Affan Badar have examined the impact of cylinder pressure monitoring on compressor performance, revealing its positive influence on productivity compared to temperature monitoring [3]. Baker Hughes has introduced the 3500/22M Bentley Nevada system, capable of monitoring cylinder pressure, constructing PV curves for each cylinder, and monitoring temperatures for main bearings, crossheads, and suction valves [4]. Some compressors even utilize wireless connecting rod temperature measurements [5]. Nevertheless, previous research has not delved into the continuous monitoring of the mechanical behaviors and clearances of compressor crankshafts, crossheads, and cylinders [6]. This study aims to investigate the impact of real-time continuous monitoring of main bearing, crosshead, and cylinder clearances on compressor availability and reliability. By doing so, it aims to reduce the frequency of preventive maintenance (compressor overhauls), lower overall maintenance costs, and enhance compressor availability. This will be achieved by:

1. Monitoring changes in main bearing, crosshead, and cylinder clearances over operating time to

2. Calculating the failure rates for various compressor failure modes.

3. Analysing the effect of increased bearing clearance on compressor main bearing stress using contour analysis.

Through these approaches, a comprehensive understanding of compressor behavior and clearances will be gained, facilitating more efficient maintenance strategies and ultimately improving compressor reliability and availability.

2. Materials and Methods

This study focuses on the application of a comprehensive online monitoring system to optimize maintenance practices for reciprocating compressors. It involves monitoring and assessing changes in mechanical clearances (including main bearings, slipper, and piston bottom clearances) as the compressor operates over time.

Compressor Selection: K-1101B Naphtha hydro treating recycle gas compressor, manufactured by MITSU, located at the Naphtha hydro treating unit of the Egyptian Refining Company (ERC), serves as the basis for this study. To accurately simulate the actual compressor's behavior and perform stress analysis related to changes in bearing clearance over operating time, clearance data was collected during a complete overhaul of the compressor.

Simulation Steps: To ensure a close match between the simulation and the real compressor on-site, the following steps were undertaken:

1. Clearance Measurement (4000-Hour Overhaul): Initial measurements of all compressor clearances were taken during the first 4000-hour overhaul, providing a baseline for accurate simulation.

2. Clearance Measurement (8000-Hour Overhaul): Clearances were measured again during the 8000-hour and 12000-hour overhaul, and these measurements were compared with the data obtained during the 4000-hour overhaul to establish trends in compressor clearances to understand the deterioration rate for the mechanical components and evaluate the importance for developing the online monitoring system for compressor.

3. SolidWorks Simulation (K-1101A): Utilizing SolidWorks, a simulation was created to draw the pressure contour for crankshaft main bearings and study the stress analysis with change in bearing clearance and evaluate the value for installing the on line monitoring system on decrease compressor downtime, increase overall compressor availability, and enhance mechanical performance.

4. Online Monitoring System Data Integration: Input data from existing online monitoring system (consisting from two rod drops and two velocity transducers only) specifically the Bentley Nevada 3500 series, was used in conjunction with System 1 software to generate a comprehensive visualization of available compressor monitoring parameters from piston rod drop probes and crank shaft casing velocity transducers.

After using all inputs from the simulation system and pressure, contour analysis Maintenance Optimization and evaluating the using for complete online monitoring system can be done by studying culminated in the calculation of maintenance optimization strategies and the rescheduling of preventive maintenance activities. This was based on data and insights obtained from the online monitoring system, considering vendor recommendations.

By following these steps and utilizing advanced monitoring technology, this research aims to enhance the efficiency of maintenance practices, reduce compressor downtime, increase availability, and ultimately optimize the mechanical performance of reciprocating compressors.

2.1. Dynamic Model and Equations of Motion for Compressor Simulation:

Figure 1 illustrates a line diagram depicting the primary components of the model. Within the crank-slider mechanism, consisting of the crank arm, connecting rod, and piston, these components are treated as a sequence of rigid bodies. It's important to note that, with the exception of the crank arm-connecting rod joint, which houses a bearing, the other links are assumed to be free of any clearances or gaps [8]. Elastic compliances within the mechanism are addressed by introducing additional joints between the effective rigid bodies. The model takes into account-applied forces acting at the center of gravity of each link.

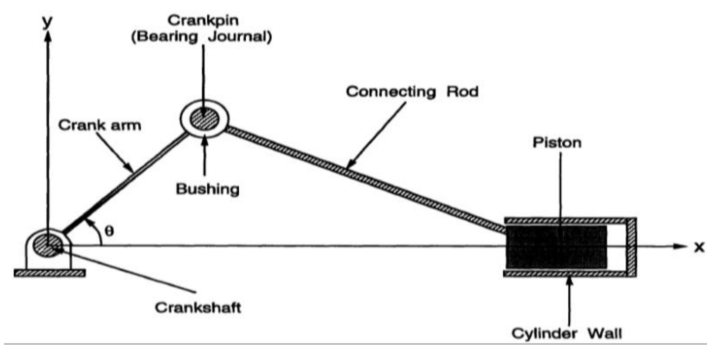


Figure 1: Main compressor components line diagram

Incorporating externally applied forces, including those generated by gravity, requires deriving equations of motion that relate the accelerations of each link to these applied forces and moments. These equations are subsequently integrated with the nonlinear constraint equations enforced by the connections between components, resulting in a system of differential equations. Solving these differential equations enables us to determine the motion and behaviour of the mechanism.

Equations of Motion for Piston (Slider):

Summation forces on the piston, in the x-direction are shown in Figure 2, where the resulting force is the product of the piston mass and the x-component of acceleration yields

$$M_{piston}\ddot{X}_{Piston} = -Fx_{conrod} - F_{xpiston} - F_{gas} - W_{piston} \quad (1)$$

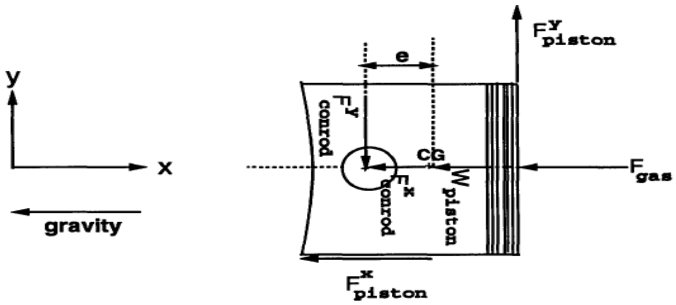


Figure 2. Forces on X and Y directions for piston

Constraint Equations for Piston Motion

The motion of the center of gravity of the piston is limited to movement in the x-direction exclusively, assuming no motion occurs in the y-direction. This assumption is made under the premise that there is no clearance at the piston-cylinder wall joint. Consequently, there is no motion in the y-direction, and rotational motion is absent. This leads to the formulation of constraint equations reflecting these limitations. $\ddot{y}_{Piston} = 0$ and $\ddot{\Theta}_{piston} = 0$. Considering both viscous and coulomb friction at the cylinder walls, the resultant force on the piston is given by

$$F^x_{piston} = -C_\upsilon \dot{X}_{piston} - C_c \left(\frac{\dot{X}_{piston}}{|\dot{X}_{piston}|}\right) \tag{2}$$

where C_υ is the coefficient of viscous friction between the piston and cylinder wall, and C_c is the coefficient of static friction between the piston and cylinder wall.

Equations of Motion for Connecting Rod

The connecting rod undergoes a combination of linear and angular motions, experiencing the effects of both dynamic influences. When summing forces acting on the connecting rod, both in the horizontal and vertical directions as depicted in Figure 3, the resulting force can be determined as the product of the connecting rod's mass and the corresponding horizontal and vertical components of acceleration [8].

$$M_{conrod}\ddot{X}_{conrod} = -F^x_{bearing} + F^x_{conrod} - W_{conrod} \tag{3}$$

$$M_{conrod}\,\ddot{y}_{conrod} = -F^y_{bearing} + F^y_{conrod} \tag{4}$$

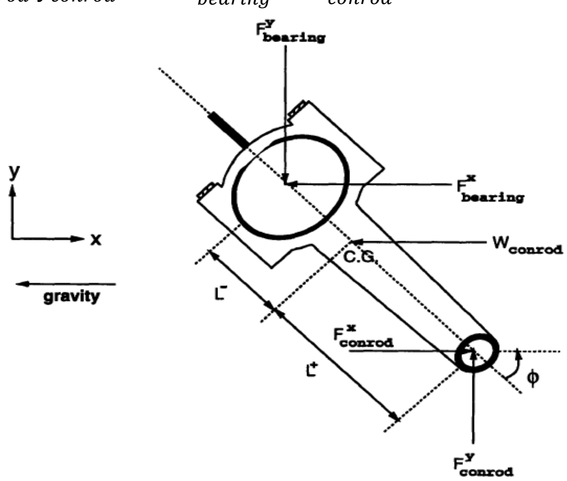


Figure 3: Horizontal and vertical forces on the connecting rod

Constraint Equations for the Journal (Crankpin)

The x and y components of the position of the center of gravity of the journal are given as $X_{journal} = Rcos\Theta$ and $Y_{journal} = Rsin\Theta$ where R is the radius of the crank arm.

Constraint Equations for the Bushing

The x and y components of the position of the center of gravity of the bushings are given as follows

$$X_{bushing} = X_{piston} - Lcos\,\phi \tag{5}$$

$$Y_{bushing} = Y_{piston} + Lsin\,\phi \tag{6}$$

where $L = L^- + L^+$ is the length of the connecting rod.

3. Results and Discussion:

The study involved measuring compressor clearances during complete overhauls conducted at 4000, 8000, and 12000-hour intervals to assess the actual lifespan of compressor spare parts. This data was utilized for simulating the compressor using SolidWorks, as a computer-aided design program. The objective was to monitor the deterioration rate for the compressor spare parts and evaluate the impact of installing the developed condition monitoring and protection system on increasing compressor availability. Additionally, System One (utilizing Bentley Nevada technology) was employed to track trends in the rod drop measurements and crank case velocity transducer readings, as illustrated in Figure 11 and Figure 12.

A- Piston Bottom Clearance

By collecting the measurements of compressor clearances at 4000, 8000, and 12000-hour complete overhauls, it was observed that the piston bottom clearance, measured using a filler, decreased from 1.85 mm at the 4000-hour overhaul to 1.3 mm at the 12000-hour overhaul, as illustrated in trend. 1. This reduction in clearance may be attributed to wear in the rider ring and a decrease in rider ring thickness. This wear and thinning could be linked to exposure of the rider ring to corrosive hydrocarbons present in the compressed gases within the compressor cylinder, as depicted in Figure 5.

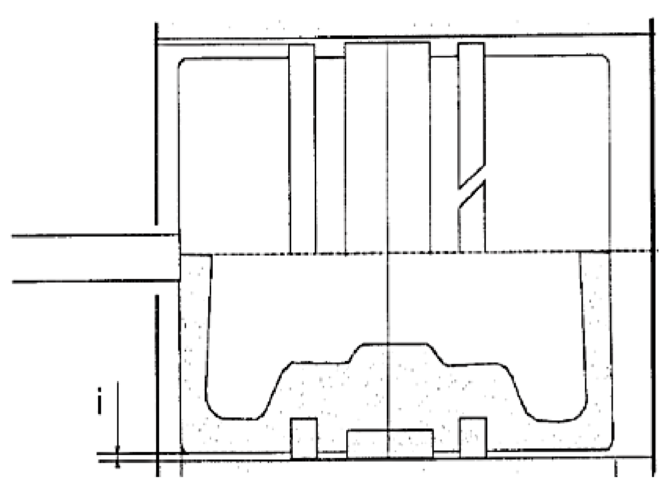
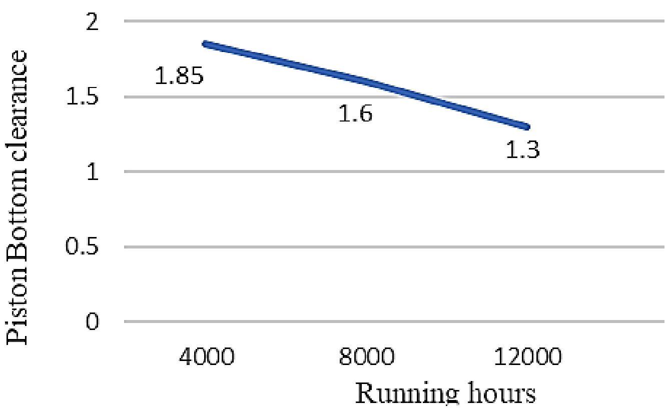


Figure 4: Piston bottom clearance



Trend 1: Piston bottom clearance at 4000, 8000 and 12000 running hours

B- Cross head and cross head shoe measurement (slipper clearance)

The slipper clearance, which refers to the gap between the crosshead and the crosshead shoe, was meticulously measured. It was observed that the slipper clearance for the compressor during the three overhauls was slightly outside the specified tolerance range of 0.20 mm to 0.30 mm, as indicated in Trend 2. This deviation can be attributed to the deteriorating condition of the shims used to adjust the clearance between the crosshead and the crosshead shoe, as depicted in Figure 6.

Furthermore, it's noteworthy that the slipper clearance varied between the two sides of the crosshead and the crosshead shoe. Such discrepancies can result in elevated stresses on the connecting rod bearings and piston rod packings during compressor operation. This situation necessitates attention and potential corrective actions to maintain the proper functioning and longevity of these critical components.

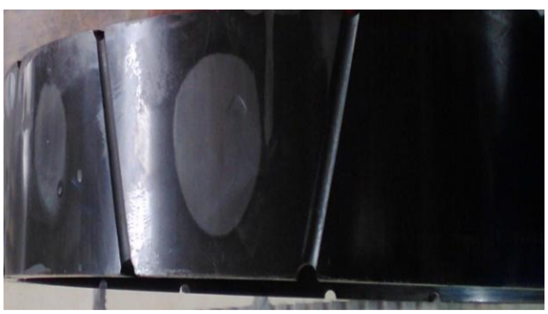


Figure 5: The wear in rider ring

Figure 6: The slipper clearance and adjusted shims

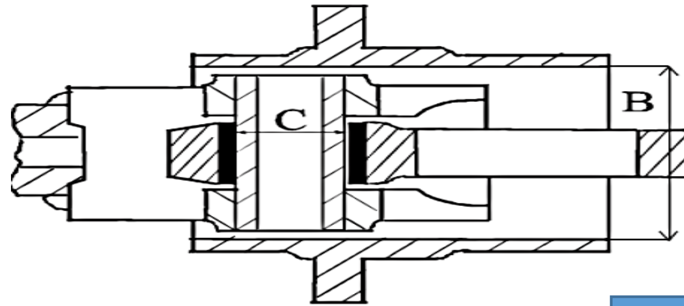
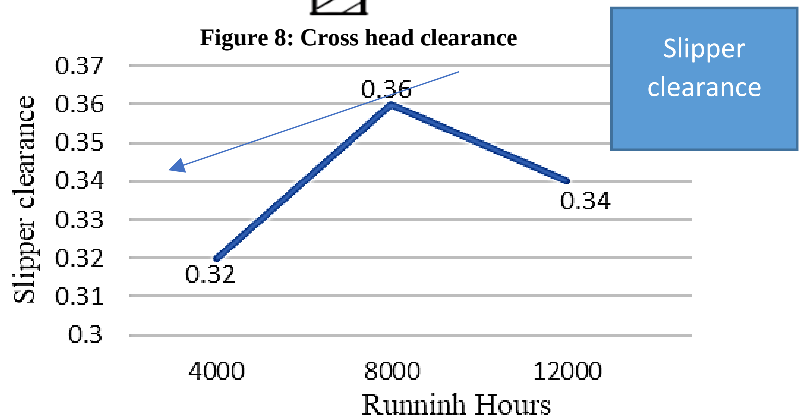


Figure 8: Cross head clearance



Trend 2: The slipper clearance at 4000, 8000 and 12000 running hours

C- Run out measurement

Vertical run-out measurements for the piston rod were conducted using a dial indicator fixture, as shown in Fig. 7. The vertical measurements revealed a discrepancy between the actual measurements and the reference measurements (where the reference measurement range is defined as 0.106 to -0.094 according to the Manual), as illustrated in Trend 3. This disparity can be attributed to wear in the rider ring and potential issues with the piston bottom and cross head clearance.

On the other hand, when assessing horizontal run-out, it was observed that the actual measurements were approximately consistent with the reference measurements (where the reference measurement range is defined as 0.100 to -0.100 as per the Manual). This suggests that horizontal run-out was within acceptable tolerances.

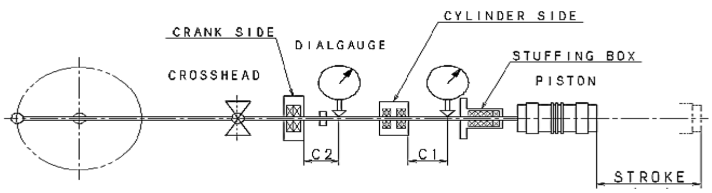
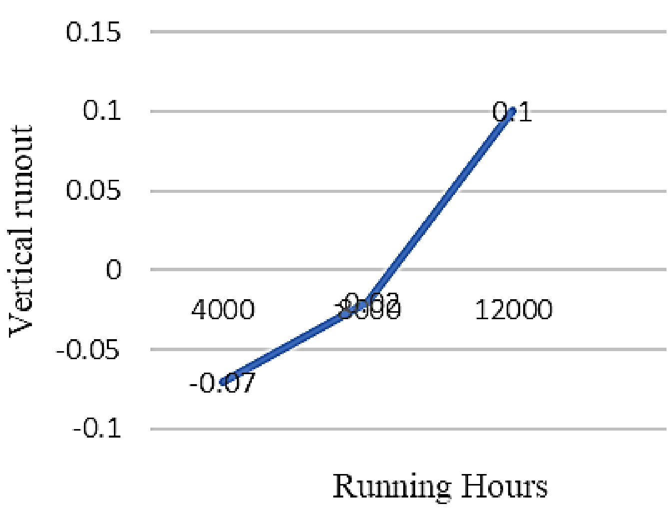


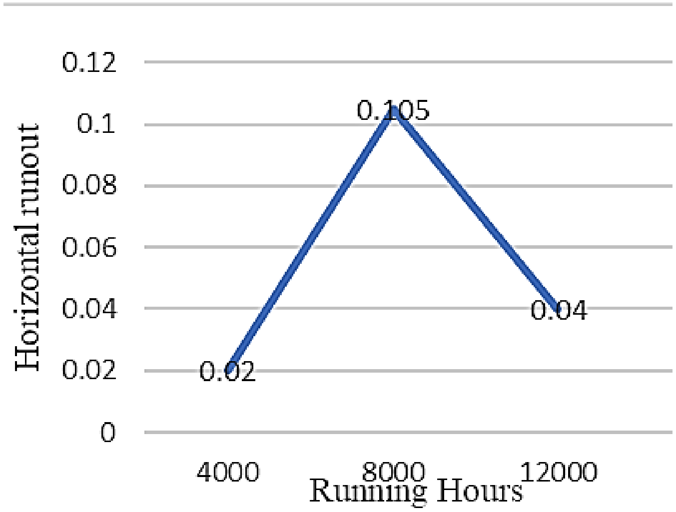
Figure 7: Dial indicators fixation during run-out measurement

D- Main Bearing Drive End (DE) and NON-drive end bearing Clearance:

The clearance between the crankshaft and the non-drive end journal bearing was measured to be 0.2 mm at 4000 hours of operation and increased to 0.24 mm at 12000 hours, as depicted in Trend 4. This trend suggests that over time, there has been a gradual increase in the clearance between these components.

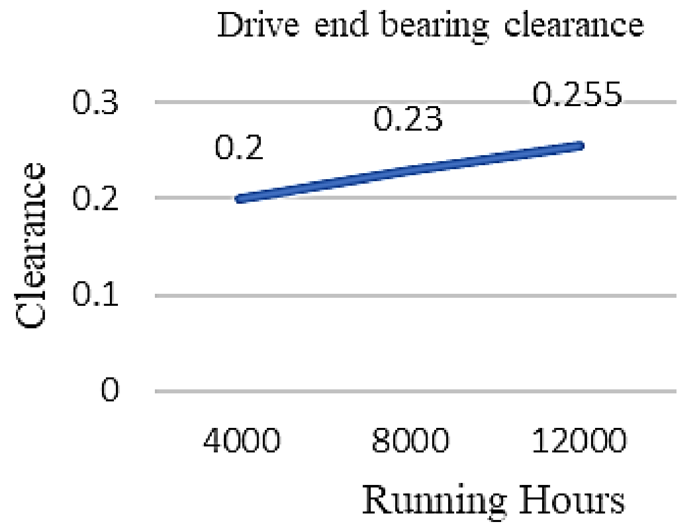


(a)

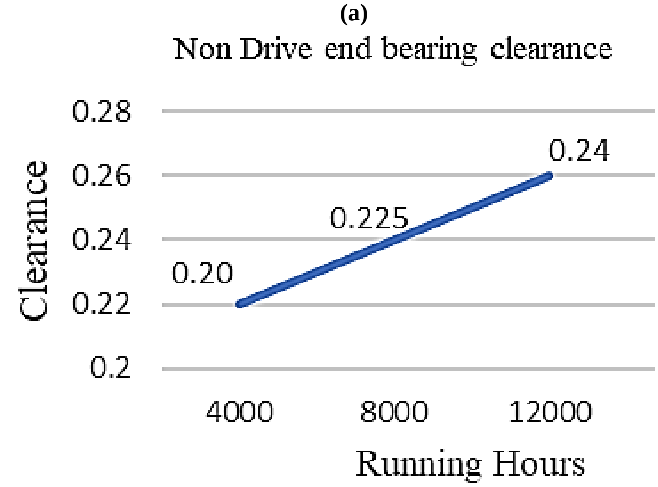


(b)

Trend 3: The Vertical and horizontal run out measurement at 4000, 8000 and 12000 running hours



(a)



(b)

Trend 4: The Drive End main bearing clearance at 4000, 8000 and 12000 running hours

Indeed, the findings discussed above highlight the critical importance of implementing a comprehensive monitoring and protection system for the compressor. The increase in clearances between various mechanical components within the compressor underscores the need for continuous monitoring. Such a system not only enables real-time tracking of these clearances but also plays a pivotal role in enhancing compressor availability while minimizing downtime.

By providing early warnings and insights into the changing mechanical conditions of the compressor, a monitoring and protection system allows for proactive maintenance and adjustments. Such discrepancies can result in elevated stresses on the connecting rod bearings and piston rod packings during compressor operation, in turn, can significantly extend the lifespan of compressor components, reduce the risk of critical failures, and optimize the overall efficiency of the compressor system.

3.1. The repeated failure during the 3 years of compressor operation

The pie chart presented in Figure 10 vividly illustrates that during the operation of the compressor and between compressor overhauls intervals, failures have occurred for the most critical compressor spare parts, albeit at varying percentages. These failures collectively contribute to increased maintenance costs, extended equipment downtime, and decreased compressor reliability and availability.

Given this scenario, the utilization of a compressor condition monitoring system emerges as a paramount solution. Such a system serves the crucial purpose of safeguarding the equipment from experiencing catastrophic failures. By continuously monitoring the condition of various components and providing timely alerts and insights, it allows for proactive maintenance and targeted interventions. This proactive approach not only reduces maintenance costs and equipment downtime but also substantially enhances the overall reliability and availability of the compressor system, as mentioned before.

Figure 9: The bearing clearance location

The consequences of a complete failure of the rider ring can be quite severe. It can lead to the failure of both the piston and the cylinder liner, resulting in exceptionally high maintenance costs. Furthermore, piston ring failure has a detrimental impact on the operational performance of the compressor. It disrupts the uniform distribution of dynamic pressure inside the compressor cylinder, affecting the balance of the compressor's mechanical motion [10].

As for the valves, their failure can have significant repercussions on the smooth operation of processes. In a reciprocating compressor, valves play a critical role in regulating and controlling the flow of pressurized gas. When valves operate optimally and as intended, compressed air is distributed efficiently and evenly across various processes. However, when a valve becomes faulty, it can introduce delays and disruptions in the system, adversely affecting processes.

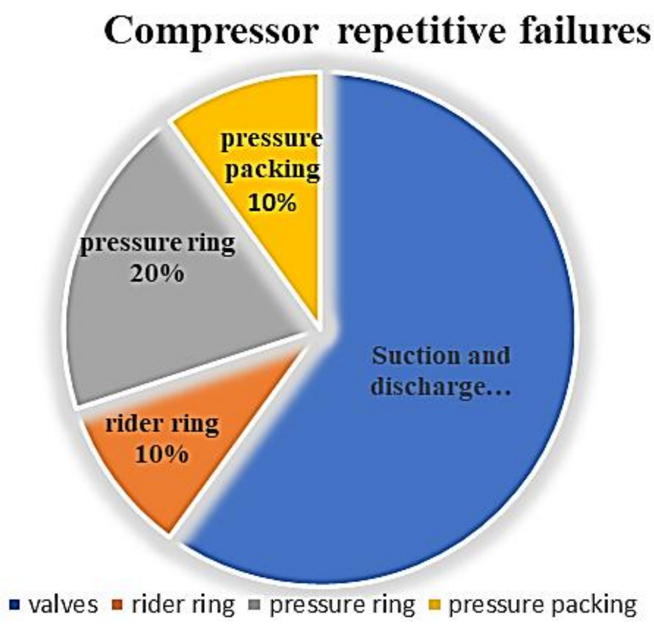


Figure 10: Pie chart for the compressor parts repetitive failure percentage

Valve defects can arise from two main categories of causes: environmental and mechanical. Common environmental causes include: (a) Corrosive Contaminants (b) Foreign Material (c) Liquid Slugs (d) Improper Lubrication. On the other hand, common mechanical causes comprise: (a) Off-Design Operation (b) High Impact Stresses (c) Spring Failure. For the compressor K-1101B, valve failures were attributed to issues such as liquid slugs and spring failures. Addressing these root causes and implementing proper maintenance and monitoring systems is crucial to prevent such failures and maintain the efficient operation of the compressor.

The failure of the pressure ring was attributed to improper material selection, which was not suitable for the operational conditions. Consequently, the pressure ring broke shortly after being put into operation. To address this issue, Hoerbiger Company made a change in piston ring material, switching from (HY54) to (HY101) during testing to ensure it could withstand the rigors of the operational conditions.

Regarding the pressure packing, its failure was linked to the piston rod's vertical run-out not being properly adjusted, as indicated in trend (3), as well as issues with the crosshead clearance, as shown in trend (2). However, it is worth noting that the incorrect piston rod vertical run-out was detected for a second time after recalibrating the rod drop system.

These issues underscore the importance of precise maintenance and monitoring of various parameters to prevent failures and ensure the reliable operation of the compressor. Addressing material suitability and clearances are critical steps in maintaining the compressor's functionality and longevity.

3.2. The Complete Monitoring System for the Compressor

The evaluation of the overall vibration readings from the rod drop probes installed in the compressor distance piece, which are intended to monitor the piston bottom clearance and rider wear was compared to the actual clearance measurements obtained during compressor overhauls, aiming to verify the accuracy of the online monitoring system provided by the compressor manufacturer. This system comprises:

1. Two-rod drop probes for monitoring the piston bottom clearance and assessing the rider ring's condition.
2. Two velocity transducers, one installed on the drive end and the other on the non-drive end of the crankshaft, to monitor crankshaft-casing vibrations.
3. Two temperature sensors for monitoring piston rod temperatures.

In particular, the ZI-5002B rod drop reading for cylinder number one, as shown in Fig. 11, indicated a value of 79 mils, equivalent to 2 mm. Meanwhile, for ZI-5003B, which corresponds to cylinder number two, the reading was 75 mils, equivalent to 1.9 mm. Upon comparing these rods drop readings with the actual piston bottom clearances, a discrepancy of approximately 0.4 mm was observed.

This variance can be attributed to the fixed position of the rod drop probes, which are situated at a distance from the actual piston bottom clearance and the condition of the piston rider ring. The findings emphasize the importance of considering the probe location and its limitations when interpreting the monitoring system's data.

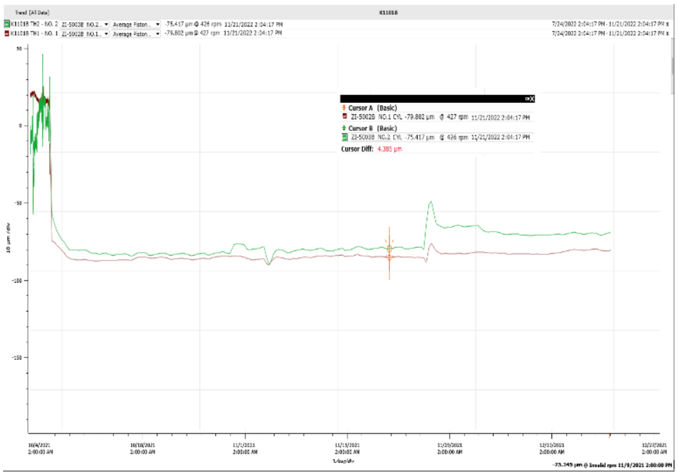


Figure 11: Rod drop reading for two cylinders

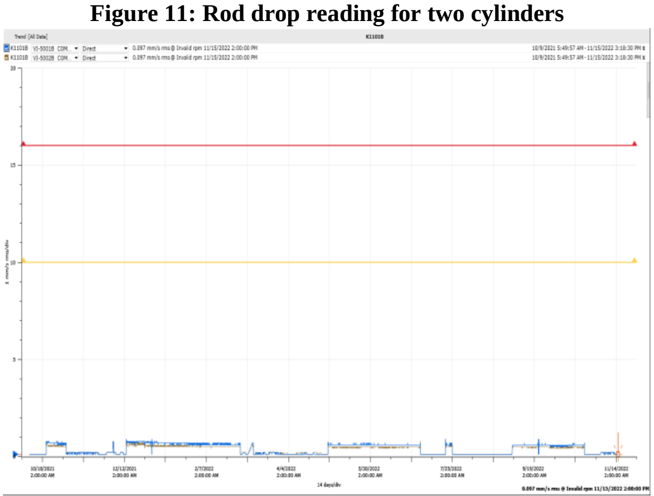


Figure 12: DE and NDE crank shaft vibration trend

Using velocity transducers to measure crank casing vibration provides valuable data, but it lacks direct information about the dynamic behavior of the shaft and the forces acting on the crankshaft main bearings, particularly in cases where there is a high casing-to-rotor weight ratio. Therefore, velocity transducers may not provide a precise indication of crankshaft vibration or the condition of the crankshaft main bearings. To address this, the use of direct proximity probes for monitoring the crankshaft journal bearings can offer a more accurate assessment of bearing clearances and the physical motion of the shaft in relation to the bearings, which will have a positive effect for monitoring the compressor mechanical performance. This approach can reduce the frequency of compressor overhauls and protect the compressor components from sudden failures.

The proposed complete monitoring system for the compressor will consist of the following components:

1. Two proximity probes for crankshaft main bearings for measuring the clearance between the journal bearing and the crankshaft, as depicted in Figure 14.
2. Two proximity probes fix on the crosshead shoe both sides for measuring the slipper clearance between the crosshead and the crosshead shoe, on both sides of the crosshead.
3. Two proximity probes for measuring the piston bottom clearance at both top dead center and bottom dead center for each compressor cylinder.

By incorporating these proximity probes to the monitoring system in addition to proximity probe (rod drop) to measure the aligned for connecting rod with the center for crosshead and piston. The monitoring system can offer more comprehensive and accurate data regarding the condition and behavior of critical compressor components, leading to enhanced maintenance strategies and reduced risk of unexpected failures.

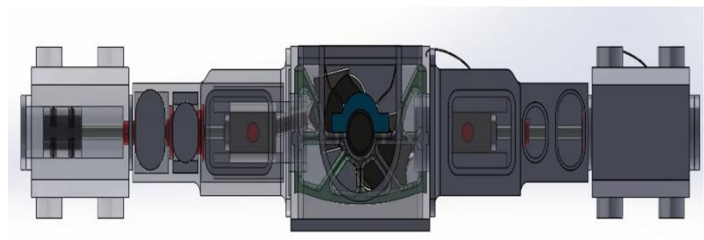


Figure 13: The fixed probes on the compressor

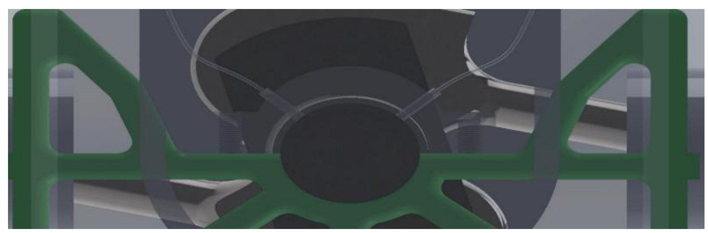


Figure 14: Main bearing proximity probes

3.3. Stresses contour analysis for crankshaft main journal bearing

A journal bearing is a type of bearing that consists of a bearing surface without any rolling elements. In this type of bearing, a journal, which is a cylindrical component of a rotating machine, slides over the bearing surface. This design allows for relative motion between the journal and the bearing surface [11].

In the context of the study, pressure contour analysis was conducted for the compressor's main bearing to illustrate the stress analysis applied to the bearing surfaces at various clearances, as shown in the figures below. Here are some key observations from the analysis:

1. Drive End Bearing: In the case of the drive end bearing, the contour static pressure exhibited a direct proportional relationship with the bearing clearance. As the clearance increased, the static pressure also increased. For instance,

static pressure increased from 9.05 x 10^5 Pascals at diametral clearance 0.21 mm to 1.01 x 10^6 Pascals at diametral clearance 0.23 mm, as illustrated in Fig. 15, Fig. 16 .This increase in static pressure was attributed to the growth in the minimum fluid film thickness [12], leading to an increase in the coefficient of friction. However, once the bearing clearance reached 0.26 mm, the static pressure became constant as illustrated in Fig. 17 due to the fixed minimum fluid film thickness and coefficient of friction.

2. Non-Drive End Bearing: In the case of the non-drive end bearing, there was a slight increase in static pressure when the clearance changed from 0.20 mm to 0.24 mm (stress) as shown in Fig. 18, Fig. 19 and Fig. 20. However, this increase was lower than that observed on the drive end bearing. This difference in pressure behavior is because the drive end experiences higher load concentration, causing the rate of change in static pressure to be higher on the drive end side compared to the non-drive end side.
3. Surface Texture Area: An increase in the surface texture area for the journal bearing led to a decrease in the oil pressure film. As a result, the oil film pressure and thickness emerged as crucial factors influencing and increasing the static pressure on the journal bearing [7].

These findings highlight the importance of understanding the relationship between bearing clearance and static pressure in journal bearings, as it can have significant implications for the performance and reliability of rotating machinery, such as compressors.

The static pressure contour analysis indeed demonstrates that an increase in the diametral clearance has a detrimental effect on bearing performance. This finding underscores the critical importance of maintaining proper bearing clearances to ensure optimal performance and reliability in rotating machinery, like compressors. The static pressure contour analysis indeed demonstrates that an increase in the diametral clearance has a detrimental effect on bearing performance. This finding underscores the critical importance of maintaining proper bearing clearances to ensure optimal performance and reliability in rotating machinery, like compressors.

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Furthermore, factors such as oil viscosity, which can be influenced by the selection of oil type and grade, as well as engine operating temperature, play a direct role in bearing performance. Proper oil selection and maintenance are essential to ensure the lubrication system functions effectively and that the bearing surfaces are adequately protected.

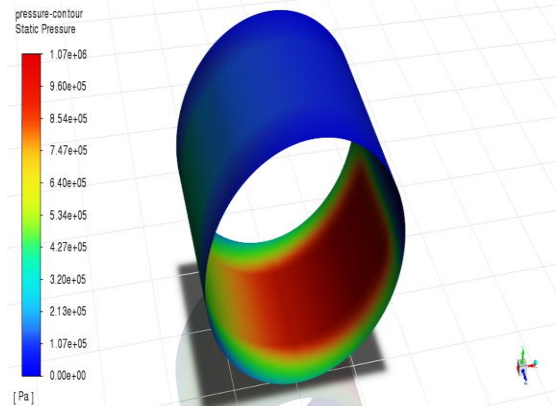

Figure 17: DE counter at 0.255 mm clearance

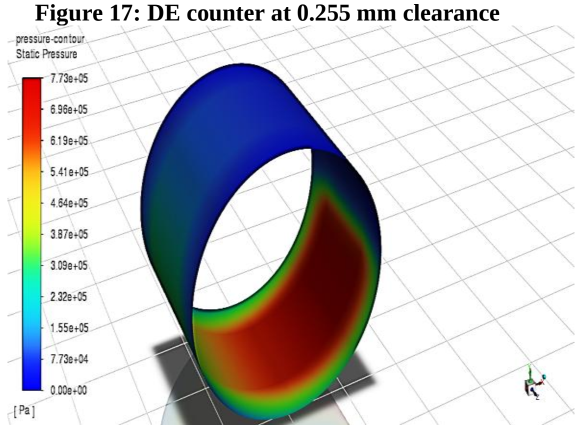

Figure 18: NDE counter at 0.20 mm clearance

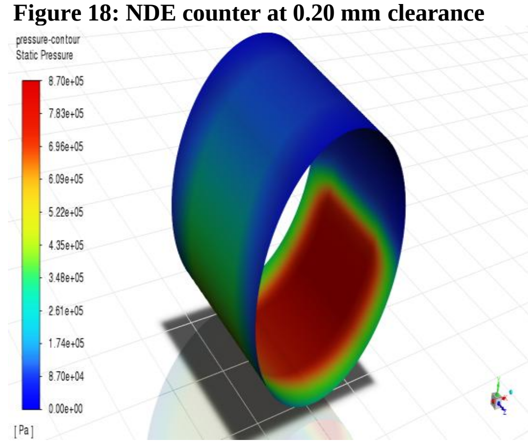

Figure 19: NDE counter at 0.225 mm clearance

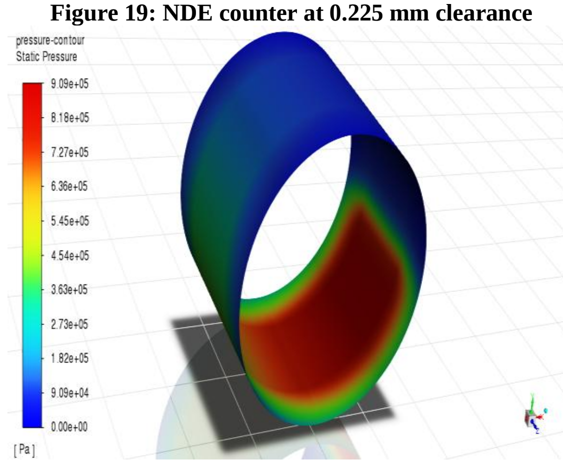

Figure 20: NDE counter at 0.24 mm clearance

Additionally, the study acknowledges the challenge posed by the non-linear characteristics of vibration signals in reciprocating compressors. Early faults can be masked by strong background noise in the vibration signals, making fault diagnosis more challenging [13]. In this context, the implementation of a protection system, specifically an online monitoring system, for monitoring bearing clearances and compressor mechanical clearances can have a significant positive impact. Such a system can continuously track and assess bearing and mechanical components performance, providing early warnings and insights that help prevent unexpected failures and maintain the reliability of the equipment.

4. Conclusion

The results of this study provide valuable insights into the performance and behavior of a reciprocating compressor's critical components, as well as the benefits of implementing a comprehensive online monitoring system. Key findings and conclusions from the study include:

1. The wear rate in the rider ring (piston bottom clearance) increases with extended compressor operating time.
2. The slipper clearance, responsible for aligning the piston rod in the center for the pressure backing and partition packing, fluctuates and consistently falls out of tolerance (greater than 30 thou or 0.3 mm). This is primarily due to defective lower shoe shims, which can reduce the lifespan of the pressure and intermediate packing.
3. The rate of increase in the clearance for the drive end (DE) bearing is slightly higher than that for the non-drive end (NDE) bearing. This is attributed to the higher loads experienced by the DE bearing during operation, including flywheel load and motor coupling (torsional load).
4. The analysis of static pressure contours for both DE and NDE bearings reveals that static pressure increases with an increase in bearing clearance. This increase in pressure is associated with a thicker minimum fluid film and an increased coefficient of friction.
5. A pie chart illustrates the rate of failure and repetitive failure for the compressor rider rings, pressure rings, and pressure packing's. These findings support the value and positive effect of using an online condition monitoring system to protect the compressor from repetitive failures.
6. The static pressure contour analysis for DE and NDE bearing clearances demonstrates a direct proportional relationship between bearing clearance and bearing stress. As bearing clearance increases, so does bearing stress, due to a thicker minimum fluid film and higher coefficient of friction. Notably, static pressure remains constant for the DE bearing when the diametral clearance reaches 0.26 mm, indicating a fixed minimum fluid film thickness. However, the increase in static pressure for the NDE bearing when the clearance increases is lower than that for the DE bearing due to load concentration at the drive end side. Consequently, the rate of change in static pressure is higher for the drive end bearing.

In conclusion, this study highlights the critical importance of monitoring and maintaining proper clearances and the positive effect of implementing an online monitoring system to monitor all compressor clearances and to enhance the reliability and performance of reciprocating compressors.

Conflict of Interest

The authors declare no conflict of interest.

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