Radial shaft seal failure investigation

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The purpose of this study is to use root cause failure analysis method to investigate and resolve reliability related problems of radial shaft seals in order to help designers and manufacturers to improve the product validation process. Previous studies have revealed that lifespan and performance of hydraulic motors are significantly affected by the reliability of the sealing solution. The key point for achieving lifetime of radial shaft seal improvement is to deeply understand seals applications duty cycle, which leads to specific failures modes. In this paper, the factors relating to life of radial shaft seals are investigated and discussed. The application of root cause failure analysis method used on field data, experimental tests and numerical simulations aimed at reproducing of a defect, accelerating test and estimating the lifespan of a seal have been presented. Obtained results show a good agreement between on field data and experimental tests, as well as the strong impact of operating conditions on the radial shaft seal lifespan.

1 Introduction

Poclain Hydraulics is an international group specializing in the design, production and marketing of hydrostatic transmissions. With more than 2,400 employees present worldwide through our numerous factories and commercial subsidiaries, we have been able to develop and have our expertise and our sense of innovation recognized by the largest manufacturers in many markets (transport, Agriculture…). To respond to constant expansion of market, Poclain is always looking to extend its knowledge and drives innovations even in field like sealing technologies, lubricant and lubrication and so on, though not its core business.

The elastomeric radial shaft seal has been extensively used throughout industry for many years due to their adavantages, which include low friction and zero leakage [1] (Figure 1). The sealing concept appears to be simple. The flexing of the elastomeric lip and the spring as they are installed over the shaft generates a force between the lip and the shaft that acts as a dam to keep fluid in and contaminants out of the sump.

However, as the practice had shown, the sealing mechanism is not fully explored despite the fact elastomeric lip seals have been used since the 1940's. This lack of understanding leads to trial and error in seal design, material compounding, and processing. It may results in seals that leak instantly for no apparent reason. A different seal design and material may have sporadic performance with early leakers combined with long-lived seals. Trial and error can also result in a seal design and material that will function smoothly until the ultimate defect is reached. Fatigue, wear and ageing are the main defect modes present on radial motor. When the material loses its flexibility the lip seal can no longer resist to the shear stress and unnable to follow the dynamics of the shaft. Thick lip seals, robust materials and garter springs are often used to add additional force, which compensates for the resistance to the shear stress, the radial load and flexibility loss that occurs when rubber materials are exposed to high pressure and high speed combined with high frequency of direction change for extended periods of time.

A great number of variables of lip design [2], material compounding, manufacturing processes and application conditions combined make a complete understanding of sealing mechanism a difficult task. Fortunately, many researchers have been studying the phenomena since the mid 1950's and important progress has recently been made. When the apex of the lip is pushed against the shaft, the apex flattens out, creating a small edge of the elastomer that is approximately parallel to the shaft. This parallel edge is often referred to as the sealing zone. An illustration of the sealing zone is shown in Fig. 1(b). As the shaft rotates within the seal, friction is created that causes the shaft surface to become smoother, yet also causes the rubber to wear away within the sealing zone. It is generally accepted that the sealing zone wears out in such a way that small asperities form on the surface of the rubber. These microasperities create a load-carrying capacity due to cavitation effects. This load-carrying capacity leads to the development of a film thickness within the sealing zone [3], which lowers the wear rate of the rubber, allowing the asperities to be effective for an extended length of time. It has also been demonstrated experimentally [4] that some of the lubricant in the sealing zone flows back into the sealed area instead of flowing out into the environment, following their natural route. This “reverse pumping” is what keeps the seal from leaking, and it has been found that the reverse pumping rate is related to the microasperity contact pattern [5].

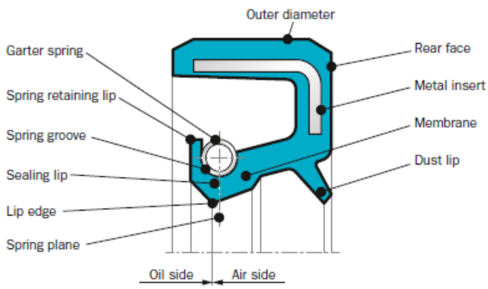


Figure 1 : Designation for rotary shaft lip seals (extract from ISO 6194)

As the working conditions of modern machinery develop in the direction of high temperature, high speed, and high pressure, rotary lip seals are facing more stringent performance requirements such as zero leakage, low friction, and long life[6][7][8][9][9]. Therefore, continuous research on the lubrication and sealing performance of radial shaft seals is of great significance and has attracted widespread attention from scholars at home and abroad. With the development of computing technology, numerical simulation has gradually become an important research method. Scholars have established numerical models or carried out experiments to study the formation of the film between the two surfaces of a lip seal and shaft [11][12][13][14][15] and study the influence of morphology of the asperities on the surface of the lip[16][17][18][19][20][21], the structural parameters of the lip seal [22], and the misalignment of the lip seal and shaft [23]on the sealing performance. Some researchers have also carried out experimental studies to explore the changing laws of the sealing performance under different operating conditions [24], and other factors. Studies have shown that there are two main aspects to evaluate the sealing performance of a lip seal: reverse pumping rate and friction torque. The reverse pumping rate is the key to the realization of zero leakage of the lip seal, and the friction torque reflects the degree of wear of the lip seal, which are two important factors for evaluating the performance of the lip seal.

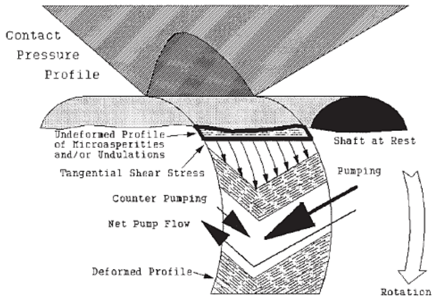


Figure 2 : Micro-visco seal by Kammüller [4]

The factors relating to life of rotary lip seals have been investigated, these are friction, compression, fatigue, wear, aging, thermal hardening and oxidation, etc. First of all, different modes could coexist between the rotary lip and the shaft in which they are defined as boundary and mixed, and hydrodynamic lubrications, and these three states may be continuously altered during working cycles. Boundary lubrication leads to wear and tear, hydrodynamic lubrication with too thick oil film is a cause of leakage, but the mixed lubrication represents an oil film layer interfaced between the lip and the axis which generated a stable fluid dynamics. This oil film layer, except to produce lubrication function, also provides the sealing effect. When the oil film layer is too thick, the fluid leaks; too thin oil film layer leads to weak lubrication function causes the lip wear. Secondly, excessive temperatures may result in hardening of the elastomer, which can lead to loss the sealing function, the permanent deformation or permanent damage of the seal would happen. During the working cycles rubber seals, under the action of mechanical forces, repeatedly generate fatigue failure under the third failure mode of the radial seal which manifest by tear and loss of stiffness. The knowledge of condition of seals throughout their working life is important since they are often used on high value engineering products. As the current material properties of a seal and the working environments vary, it is difficult to predict useful life of the rotary lip seal. Experimental data analysis combined with experiences are among the most commonly used methods to predict lifetime of the radial shaft seal [25]. The advantage of this method could include cost effectiveness; however, it leads to errors because of the subjective, variable results of manual vision inspection method. Data collection data for different kinds of seals is also time-consuming.

In this paper, the application of Ishikawa root cause analysis method based on field data and experimental tests to reproduce defect, accelerate test and estimate the lifespan of the radial shaft seal presented. Obtained results show a good agreement between on field data and experimental tests, as well as the strong impact of operating conditions on the radial shaft seal lifespan.

2 Final customer field investigation

In this paper, we will select one radial shaft seal design applied on only one machine type application in order to eliminate variable conditions, which can be introduced by other machine type condition. To select the following studied seal design, two perspectives have to be followed: on the one hand, field data has to be accessible and workable (such as customer warranty claims, quantity of machine manufactured, machine use conditions, expected lifetime and so on). On the other hand, the customer field analysis has to show some design weakness that can be easily identified and reproducible on test bench.

Obviously it's quite a complicated work to accomplish to put together both conditions mentioned above. However, thanks to the important range of radial shaft seal used within Poclain and very well organize warranty network, one seal reference that combine all required conditions has been highlighted.

Chosen seal design is a part of Poclain motor solution set on 4 wheels forklift machine application for worldwide manufacturer. This application has advantages to be very common machine type that offers us an important quantity of data and sealing conditions close to general use of seals in our global range of application.

Figure 3: 4 wheel forklift machine application

Recurrent failure mode on this part has been identified as main lip seal rupture in fatigue with crack propagation initiated from elastomer radius between both seal lips and propagated up to garter spring through the seal cross section and along the circumference.

In some case, crack propagation can occur in both direction from starting point up to spring location in one way and over the dust lip in the other direction.

Figure 4: Cross-section of seal after 600hrs on machine conditions with crack presence

It should be noted that even if main failure of the studied design is obviously a fatigue rupture, we also observed wear under main lip seal located on contact surface with rotating shaft. At this stage, it is important to specify that both failure modes start simultaneously at first machine use and fatigue rupture is predominant on wear failure mode.

The greatest challenge when studying product resistance to stress environment lie in insuring test representability in an acceptable test duration. Since the studied machine, application is design for 10.000 service hours it is obvious that test program must be somehow accelerated. The advantage of our study is direct data integration from machine fleet. Warranty service and worldwide repair center makes it possible for us to use real data to estimate seal reliability behavior according to known conditions versus true working hours.

The main task is to use real unreliability law of the seal from machine field analysis to be able to correlate it with Weibull law. This correlation will allow on the one a hand to determine acceleration factor of laboratory test reproduction and on the other hand, to continuously following any changes occurs between test bench defect mode and field defect mode.

Principle of this method is to define accordingly to usage distribution the proportion of a system that survives at a given number of cycle. In our case, the number of cycles will correspond to potential working hours of the machine from starting date up to 30 months of service life. Usage distribution parameter is defined as the ratio between machine hours meter and service life of the returned machine in repair center. Usage distribution is defined with each machines hours meter divided by total service life of the returned machine in repair center. In other words the average daily hours of machine use.

The aim of this study is not to properly analyze reliability of the system in final environment conditions but to use tools from reliability engineering to define an accelerated laboratory test that is fully representative of the on field failure mode observed. Figure 5 shows the estimation of stress distribution or usage distribution in motors within the machine fleet gathered by repair centers.

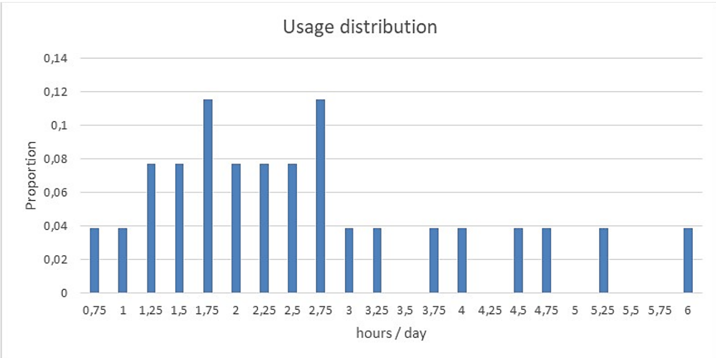


Figure 5: Daily usage distribution form machine field investigation (based on 667 samples)

According to this distribution, we create matrix of “potential service hours” with usage distribution as matrix column and month from 1 to 30 as matrix line.

Daily usage (hours / day)	hours/1 month	…	hours/20 month
0,75	22,9	…	457,5
1	30,5	…	610,0
1,25	38,1	…	762,5
1,5	45,8	…	915,0
1,75	53,4	…	1067,5
2	61,0	…	1220,0
2,25	68,6	…	1372,5
2,5	76,3	…	1525,0
2,75	83,9	…	1677,5
3	91,5	…	1830,0
3,25	99,1	…	1982,5
3,5	106,8	…	2135,0
3,75	114,4	…	2287,5
4	122,0	…	2440,0
4,25	129,6	…	2592,5
4,5	137,3	…	2745,0
4,75	144,9	…	2897,5
5	152,5	…	3050,0
5,25	160,1	…	3202,5
5,5	167,8	…	3355,0
5,75	175,4	…	3507,5
6	183,0	…	3660,0

Figure 6: Potential service hour matrix for 20 months according to usage distribution

By crosschecking this matrix with on-field failure and machine unit production we create a second matrix that wrap the quantity of machine that will perform enough service hours to have probable failure. Second step is dedicated to calculation of the unreliability probability, according to methodology described by Baro et al [28]. That can be explained as the sum of the product of occurrence probability and system resistance. Thus, using the second matrix we calculated the proportion of system that survive according to on-field failure occurrences. Then divided it numbers by the number of failure occurrence at the time t and the result obtained is the probability of failure occurrence according to service hours. This probability of failure occurrence equation (1) and then (2) allows us to fit real occurrence data with parametric Weibull law [28].

$$R(t) = e^{-\left(\frac{t}{\eta}\right)^{\beta}} \tag{1}$$

R(t) is Weibull Reliability Function represents the proportion of surviving specimens from the time 0 to a given time “t”, β shape parameter (Scatter) and η scale parameter

$$F(t) = 1 - e^{-\left(\frac{t}{\eta}\right)^{\beta}} \tag{2}$$

F(t) is Weibull Cumulative Failure function or Unreliability. To estimate Weibull parameters, several methodologies has been developed over the last 50 years, a non-exhaustive list can be found in [28]. In our study we linearize the unreliability function excel regression and then use median ranks method to determine Weibull parameter.

$$Ln\left[Ln\left(\frac{1}{1-F(t)}\right)\right] = \beta.Ln(t) - \beta.Ln(\eta) \tag{3}$$

Parameters found with correlation factor of R^2=0.987 with β =2.01 and η =7220. Figure 7 shows the product unreliability from field data compare to parametric law approximated.

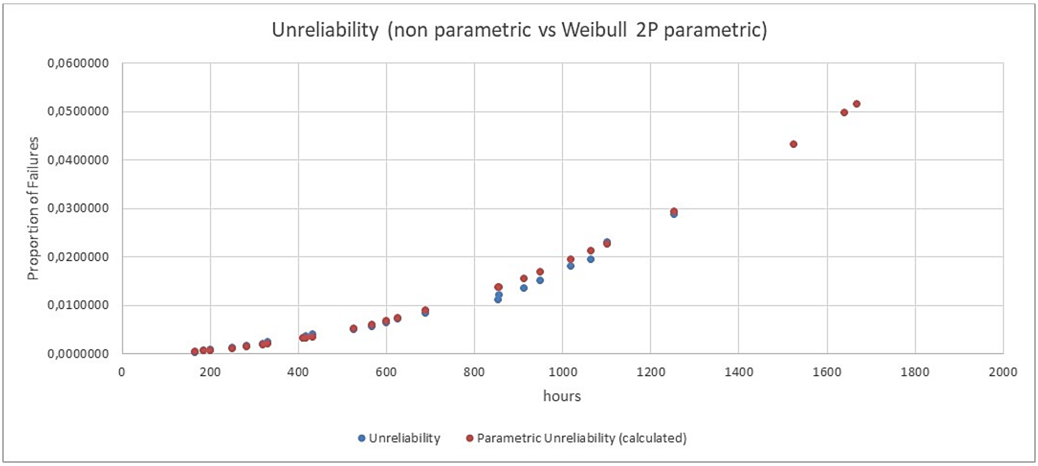


Figure 7: Failure proportion distribution in function of machine working time (adjusted Weibull parameters vs field data)

From data analysis point of view we highlight that failure mode is related to fatigue wear out behavior [28][29], which is confirmed by on-field analysis.

In this chapter, we analyzed data from field to described real system behavior through parametric equation that will be used as reference to evaluate representability of laboratory test.

3 Test setup and method

3.1 Test bench

The bench test was carried out on a self-made lip seal-testing machine; its main structure is shown in Figure 8. The main composition and working principle of the test bench: the test bench is mainly composed of a main engine, an electric control cabinet, a computer, and a lubricant source for the oil chamber. The motor can achieve steeples speed change from 20 to 2000 rpm through the frequency change of the frequency converter, the maximum output torque of the motor is 200 N.m; two ball bearing to support the shaft to avoid any radial runout or misalignment. The test cell is made of stainless steel, and equipped with an oil inlet valve, an oil return valve, and two temperature sensors embedded in the inlet and outlet of the oil chamber to monitor the oil temperature, a pressure sensor embedded in the inlet of the oil chamber used to measure the pressure in the oil chamber. Tow radial shaft seals are installed face to face in the oil chamber, the friction torque generated by both seals is supposed the same. A torque sensor, with a resolution of 0.01N.m and a measurement range of 50N.m, related between the electric motor shaft and the test cell shaft is used to measure the friction torque between both radial shaft seals and the rotary shaft and the friction torque generated by ball bearing. At the begging of each test the friction, torque generated by ball bearing is characterized and subtracted from the total friction torque generated by both seals.

The electric control cabinet is mainly used to control the opening and closing of the frequency converter, the frequency conversion speed regulating motor, and the oil supply pump of the oil chamber pressure, whose signal line is connected with the computer to implement the automatic control of the experimental process, data display and processing, and over limit protection function in the experiment process. The test software matched with the hardware system of the testing bench to carry out the functions of experimental parameter setting, parameter testing, display of dynamic data and graphics, and export of experimental results.

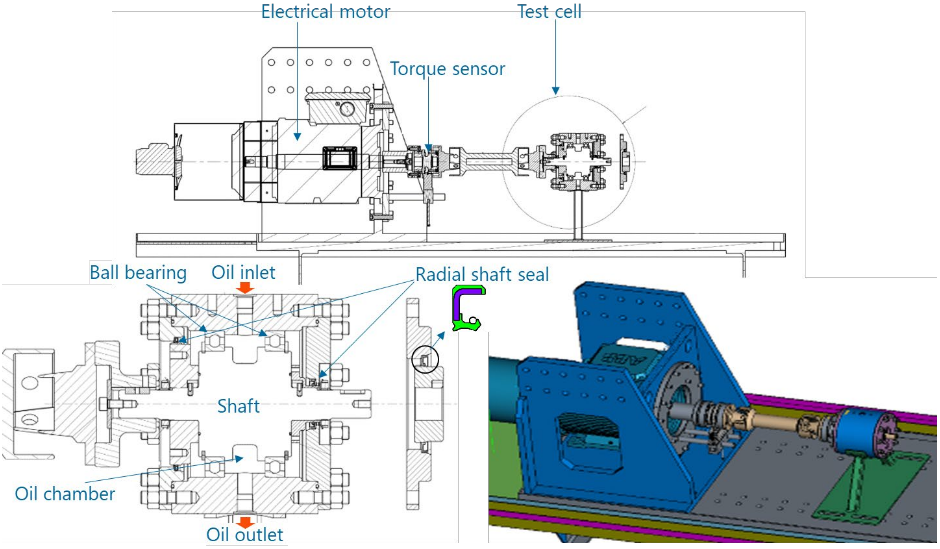


Figure 8: Test bench for radial shaft seals

3.2 Sample preparation

The outer surface of the rotary shaft sample was designed to be ground with a roughness Ra meet the standard [27]. Figure 9 shows the two-dimensional profile curve of the shaft surface, the surface roughness of the rotary shaft was measured by the MAHR surface profilometer. The two-dimensional roughness values of the shaft surface are Ra = 0.4321 μm and Rz = 2.7933 μm obtained by multiple measurements and taking the average value, which meet the standard requirements [27] .

The hardness of the shaft surface reaches HRC>45 after cylindrical grinding without strokes. The base material of the test rotary shaft is made out of steel, and its diameter is Ø65 mm.

The material of the radial shaft seal sample Figure 10 is made out of FKM (Fluoroelastomer materials) with hardness of 80 shore A, 1.5 mm thick metallic insert from

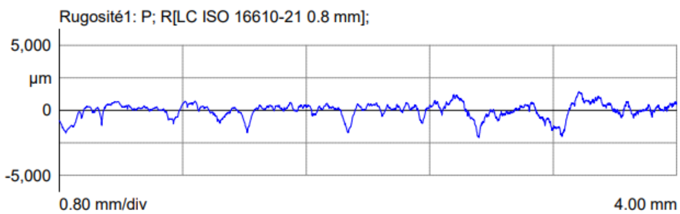


Figure 9: shaft roughness

Figure 10: radial shaft seal profile

standard low carbon steel C10 (1.0301), garter spring on main lip seal and additional dust lip seal, the specification dimensions are 63.1 mm × 85 mm × 8.5 mm (inner diameter × outer diameter × width), and the range of applicable temperature is −20~200°C. The lubricant is ISO VG 46 (46HV) hydraulic oil, whose kinematic viscosity is 46 Cst at 40℃ and 8.1 Cst at 100℃. The density is about 0.87 g/cm3. The test cell was completely fulfilled with oil. The leaking lubricant oil was collected by a beaker and weighed using a precision electronic scale with a resolution of 10e-4g.

4 Measurement

Friction torque is one of the key parameters for evaluating the sealing performance of radial shaft seal. The measurement of friction torque is mainly completed by the torque sensor of the test bench and the test software. There are also two temperature sensors on the test cell in this study: one is installed on the inlet of oil test cell and the other is installed in the outlet. This temperature measurement allow to maintain the oil test cell temperature constant. The test results show that the temperature of the oil between inlet and outlet of the test cell is quite similar, with the dynamic of the system, it is not possible to measure the temperature generated by seals friction [24]. The measurement results of the torque, oil pressure, temperature and shaft speed will be dynamically displayed on the interface of the supporting software during the experiment, see figure bellow.

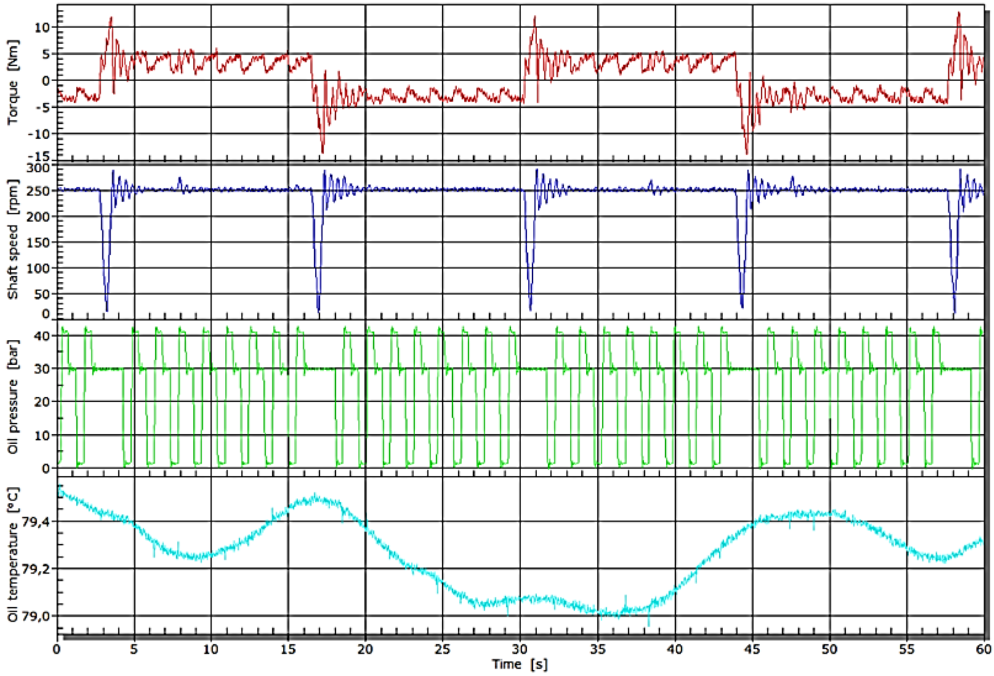


Figure 11: The interface of the supporting software with dynamic measurement of friction torque and shaft speed, oil pressure and oil temperature.

In order to understand the results presented below it is important to specify that, the shaft speed is considered moving Clockwise (CW) when the friction torque is positive and Counterclockwise (CCW) when friction torque is negative.

During the test, three parameters are imposed: the oil pressure in the test cell, the shaft speed, and the oil temperature. Figure 11 shows an example of a typical cycle, shaft speed, oil pressure (pressure levels 30, 40 and 0 bar), oil temperature and dynamic friction torque vary over a typical CW and CCW cycle. As we can observe, parameters are controlled with high accuracy.

5 Test conditions

5.1 Machine working conditions

As described on the paragraph final customer field investigation, the motor works under average speed of 80 rpm and maximum of 250 rpm and pressure levels of 30, 40 and 0 bar. Moreover, during the inversion of direction (CW and CCW) the pressure remains constant at 30 bar. The circuit oil temperature of the machine is quite stable around 80°C.

The shaft seal used on this machine was validated by two types of Poclain standard test procedures, Pressure X Speed procedure and Pulse test procedure. After these tests the seal is supposed validated for 10.000hrs of working machine.

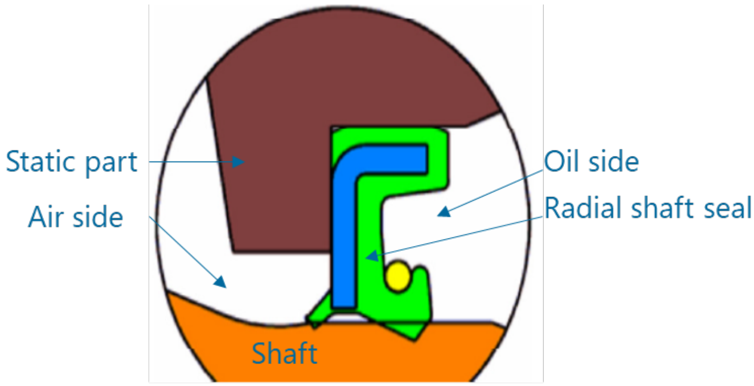


Figure 12: Radial shaft seal environment

5.2 Pressure X Speed test

Pressure X Speed procedure is used to validate the shaft seal resistance to wear. It is carried out in one sample with three steps, as presented in the table below. The test duration is 640hrs without stop and no shaft moving inversion.

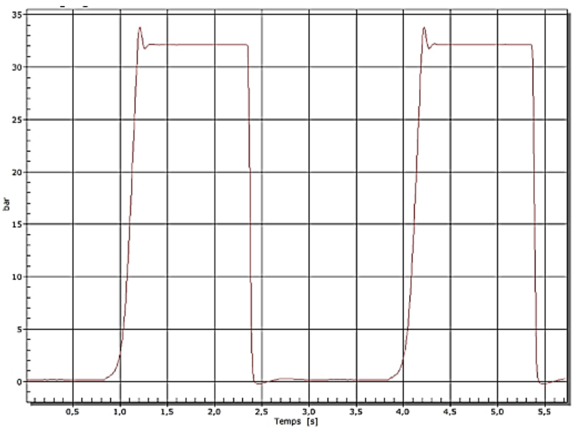
Phase	Oil pressure [bar]	Speed [rpm]	Duration [hrs]
1	12	250	160

2	20	250	160
3	30	250	320

Tableau 1: Speed X Pressure conditions

5.3 Pulse test procedure

This test is used to valid the shaft seal resistance to fatigue. The shaft oil seal specimen is subjected to a constant pressure spike with frequency of 1Hz that is increased with 8 bar after each step of 50000 cycles until the rupture.



Steps	Pressure [bar]
Initial step	32
Step 1	40
Step 2	48
Step 3	56
Step 4	64
Step 5	72
Step 6	80

Figure 13: Pulse test conditions

6 Experimental results

In this section, the experimental results obtained for the various tests conditions are demonstrated. The friction torque behavior of the radial shaft seals is presented and analyzed as a function of oil pressure (continues and spikes pressure), Shaft kinematic (shaft speed, change direction, and start/stop ramp up). The friction torque presented in all curves are given for both seals. For reliability reason, the seals have undergone a running-in phase of 50 hrs, time need to get a stable friction torque also all tests are repeated 03 times, each time we test two samples.

6.1 Test cycle

Based on the machine working conditions a typical duty cycle was defined. The shaft oil seal undergone three levels of continues pressure (30, 40 and 0 bar) as show in Figure 14. The machine is supposed to work at normal speed of 75 rpm and maximum speed of 250 rpm. During forward and backward motion, the pressure remain constant 30 bar. For 4 wheel forklift machine application, we do not have on field data about the frequency of forward and backward motion, also the speed ramp-up of the machine.

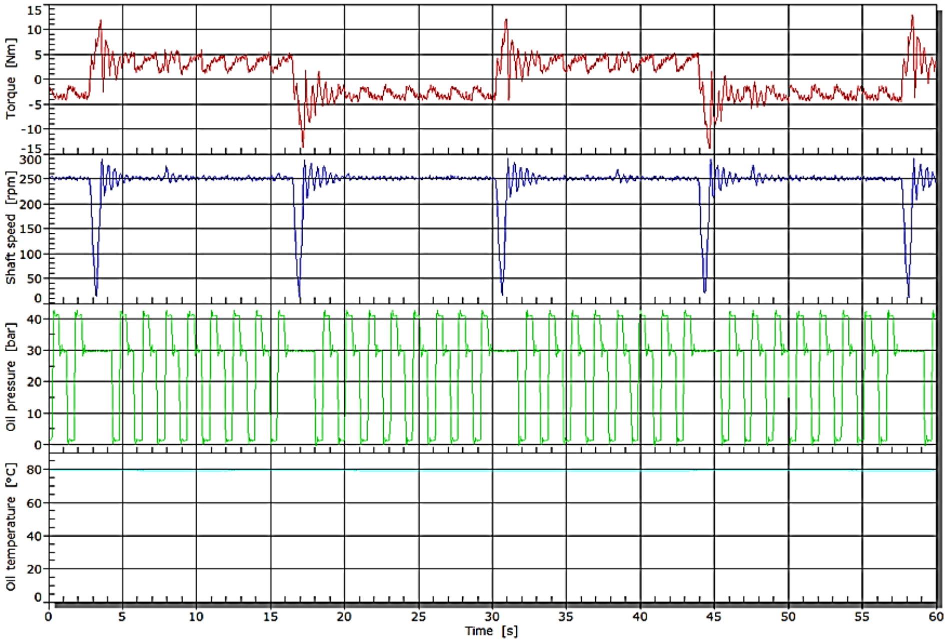


Figure 14: Typical cycle to accelerate failure

In the continuation of the study, we will present the effect of shaft kinematic, oil pressure and time on the friction torque behavior. The aim of this characteristic study is to lead us to define a representative accelerated cycle with unchanged defect mode.

6.2 Effect of speed ramp-up

The results presented in this section mainly relate to the effect of speed ramp-up on the friction torque. Figure 15 shows the friction torque in function of speed ramp-up at 30bar oil pressure and 250 rpm shaft speed. It can be noted that the friction torque spike at forward and backward motion is strongly influenced by the speed ramp-up. The friction torque is tripled at ramp-up of 0.4s compared to that of 5s ramp-up. The torque peak is mainly generated by seal friction; the contribution of the mass inertia of the rotating parts is negligible.

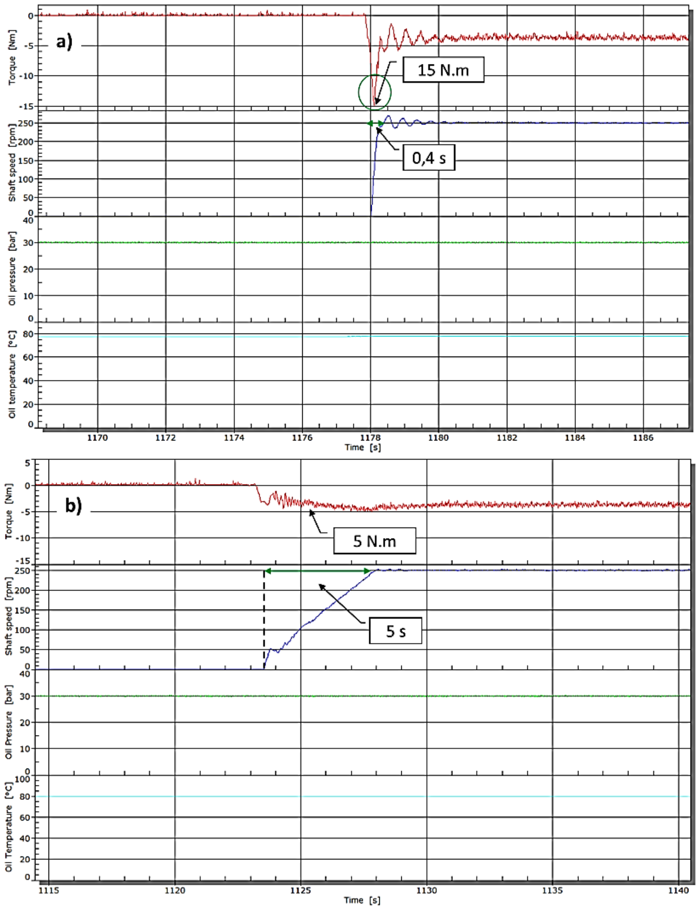


Figure 15: Effect of speed ramp-up on torque friction, a) High ramp-up, b) Low ramp-up

6.3 Effect of Pressure and Speed over time on friction torque

In order to understand the operating mode of the shaft seal. Test were carried out at 0 and 30 bar oil pressure at shaft speed of [30, 75 and 250] rpm.

At the beginning of the test

Figure 16 presents the friction torque in function of pressure and speed (Stribeck curve). It can be noted that the friction torque is strongly influenced by pressure compared to speed. As can be seen also, at low speed, less than 70 rpm, for both pressure the seal is operating in boundary regime, this mode of operating is more significant at 30 bar. At speed higher than 70rpm, for both pressures the seal is working on hydrodynamic regime. Around 70 rpm, for both pressure, the seal is working in the optimal conditions (mixed regime).

At high speed and 0 bar, the seal is strongly working in hydrodynamic regime. The high speed and no oil pressure allows to generate a hydrodynamic pressure on the contact lip which leads to a thin film and high viscous shear stress [6][12][13][14].

At high pressure and low speed, the oil seal is strongly in the boundary regime, which can leads to high wear. Even at high speed the friction torque remains less than at low speed, this can be explained by the fact that the viscous shear generate by speed is low.

At 400hrs

As we can see, the friction torque increases with time. After 400hrs and 0 bar, the friction torque increases by 10 % at low speed, 28 % at 75 rpm and 22% at 250 rpm. Although this increase is significant, it is less significant than at 30 pressure, the friction torque was increased by around 10% at 30 rpm, 32% at 75rmp and 25% at 250 rpm.

From this result, we can conclude that at high pressure and after 400 hrs there is not effect of speed on the friction torque, the friction torque is completely dominate by oil pressure. Moreover, at low pressure the sealing mode is completely in hydrodynamic regime. After 400hrs, we noticed that the friction force remains constant over time.

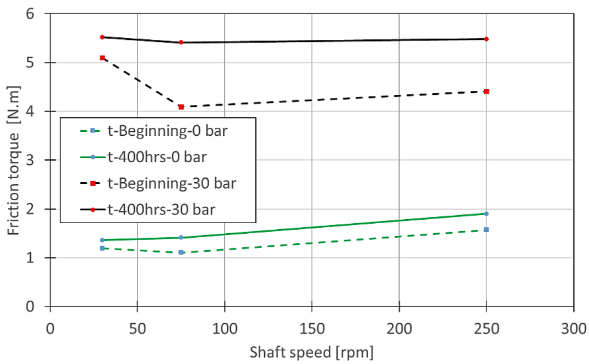


Figure 16: Friction torque evolution over time and operating conditions

6.4 Effect of forward and backward

To reproduce radial shaft seal defect and accelerate the test, two duty cycles was proposed based on the machine measurement data and the result of the previous conclusions. The typical cycle presented on the Figure 14 is used at high speed ramp-up (Figure 15) and shaft speed of 250 rpm and oil pressure levels of (30, 40 and 0) bar (Figure 16). The only difference between both cycles is the frequency of the backward and forward motion. For the first cycle, the motion is repeated each 3 minutes, with the second cycle, the motion is repeated each 15 seconds. Figure 17 shows the evolution of friction torque spikes at forward/backward motion in function of time for both cycles. As we can see, the friction torque spike increases with time. For the first cycle, seals failed between 297hrs (16.9N.m) 432hrs (21.5 N.m) and 761hrs (22.4N.m). However, we can note that with second cycle all seals failed in less than 85 hrs under conditions of less friction torque spike but with high friction torque rate, 4 to 5 times higher than the first cycle.

Moreover, there are no significant differences of friction torque between cycle 1 and cycle 2 at constant conditions over time.

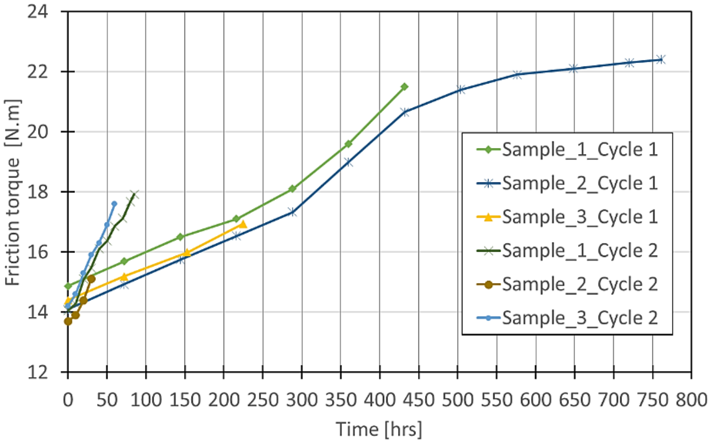


Figure 17: Friction spike evolution over time

During this tests seals failed with a sudden leakage (Figure 18.a) with consequence of losing the pressure on the test cell. The failure times were 432 h, 761 h and 297 h for the cycle 1 and 85h, 30h and 60h for the second cycle. All seals and shafts were measured and examined after test. The examinations of the seal indicate a crack initiated from the elastomer radius between seal lips and propagated up to garter spring through the seal cross section also the crack is propagated along the circumference (Figure 18).

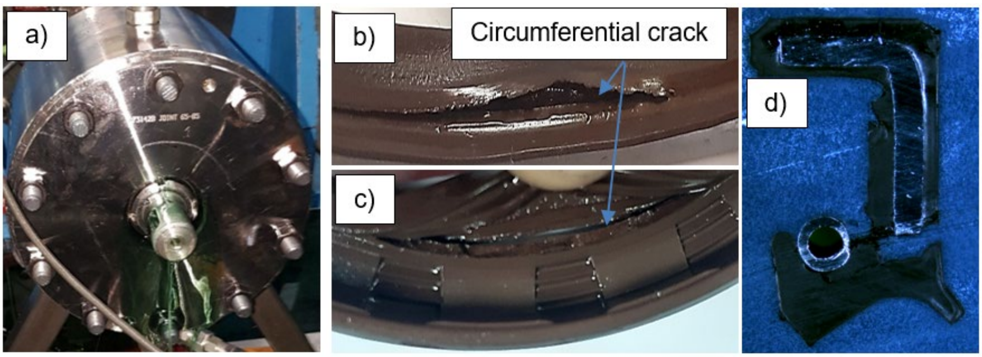


Figure 17: Radial shaft seal failure, a) Oil flow leakage, b) Crack on airside, c) Crack on oil side, d) Crack through the profile

7 Accelerated factor calculation

Estimation of test cycle acceleration factor on bench must allow us to evaluate the test cycle severity regarding current machine use conditions. The comparison of unreliability function for each conditions brings us all needful data to compare effect of conditions on seal lifetime.

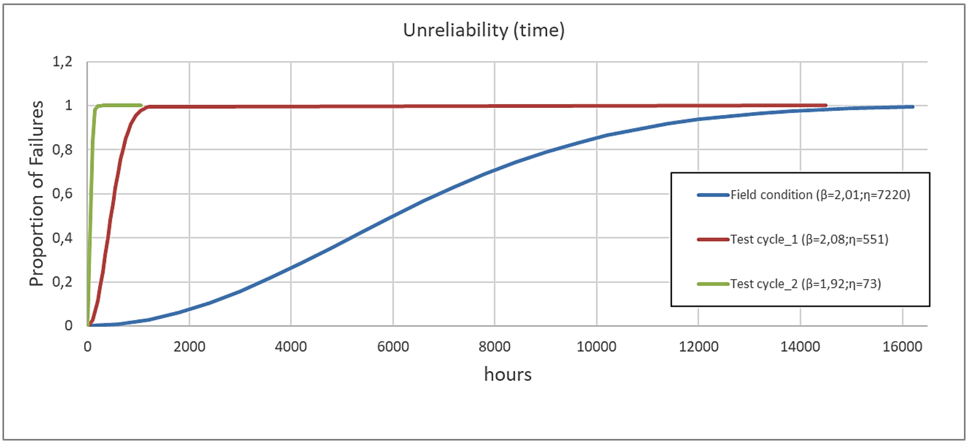


Figure 19: Failures proportion comparison between test cycles and field

In order to set our results obtained on machine lifetime scale, we define acceleration factor as ratio of Mean Time To Failure (MTTF) [28].2

$$AF = \frac{MTTF_{Field}}{MTTF_{Bench\_Cycle}} \quad (4)$$

According to test result (cycle 1 & 2) and machine field statistical analysis, we found an acceleration factor of 13 for cycle_1 and 100 for cycle_2.

The test cycle_2 gives us opportunity to observe 100 time faster the seal behavior along whole system lifetime as well as its state at the end of machine service life about 10 000hrs.

8 Conclusions

Radial shaft seal failure was investigated by analyzing field data and characterization under test bench. In this context, an experimental test bench has been designed and built. It allows to reproduce the machine operating conditions, which a shaft oil seal supposed to meet during whole life service.

Field data analysis showed that the radial shaft seal failure follows Weibull distribution with β of 2 and η of 7220. This distribution parameters represent a fatigue failure mode which confirm the failure mode observed on filed and presented in literature.

In order to reproduce the defect and accelerate the test the number of tests were performed under different operating conditions in terms of oil pressure, shaft speed, shaft speed ramp-up and frequency of forward/backward motion. Results showed that:

- At constant pressure and speed, friction torque increases over time,
- Shaft speed ramp-up has a significant impact on friction torque spike, the higher speed ramp-up, the higher is the friction torque spike. Also, we noted that the friction torque spike during backward/forward motion increases with time,
- Rate of friction torque spikes increase with the frequency of forward/backward motion,
- Fatigue defect was accelerated by increasing the frequency of the backward/forward motion without changing the stress level.
- The radial shaft seal failure was accelerated by 8 when the frequency of backward/forward motion increased by 12 with unchanged failure mode (fatigue).
- Both cycle lifetime function fit the Weibull distribution with β around 2 and η of 551 and 72 respectively.
- Failure was accelerated by 100 and 13 times with cycle 2 and Cycle 1 respectively with the fatigue defect mode.

The results presented in this paper can be used to validate and to develop theoretical models aimed at studying the effect of operating conditions on fatigue failure mode. Further developments and tests will be done to improve the design of the radial shaft seals to reach the lifetime target of 10.000hrs. With these results and ongoing study, it will be possible to create a model for the lifetime of radial shaft seal based on tests and simulation with known testing and field parameters.

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