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Maintenance and Reliability Improvement of Roller Bearings Operating at High Temperature: Thermal Stress Analysis Approach

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ABSTRACT- In this paper the authors present the outcome of an experimental maintenance and reliability investigation conducted to prevent operation failure of pinion bearings caused by excessive temperature with a case study of a 1360kW ball mill driveline system in mineral processing application. Aiming to improve the overall plant performance, the lubrication regime of pinion bearings as key component within the ball mill driveline system was studied to determine possible causes of temperature increase at 80 to over 100°C with a 25 - 30°C gap over the maximum operating temperature. An analytical review of maintenance data is conducted to mitigate future operational risks by implementing a new lubrication regime and shutdown frequency for overall statutory inspection. Accelerated life test based inverse power law has been applied to identify the root causes of excessive bearings temperature to establish immediate short-term solution paths towards a guaranteed less required maintenance asset improvement practice, less operational cost, and plant availability enhancement. Furthermore, long-term solutions have been proposed based on modelling by thermal network approach of pinion bearings for thermomechanical stress prevention. A single parameter based accelerated test on pinion bearings was applied to establish operational fault tolerant factors to be considered for asset capability improvement, which remains an open question for future studies. These actions have shown significant improvement with temperature decrease between 15 and 25°C below the maximum required temperature.

KEYWORDS- Pinion Bearing, Lubrication Regime, Maintenance Data, Thermal Modelling, Reliability Improvement.

I. INTRODUCTION

The pinion bearings studied in this case study are essential mechanical components of ball mill drive lines. The pinion as an assembly consists of a shaft, gear, and bearings. The gear is locked onto the shaft and the shaft on its both ends is connected to dynamic bearings to allow a relative movement between them. The pinion gear is coupled with the girth gear, which is mounted around the circumference of the ball mill shell, and this transfers motion from the drive to the shell. In mining industry, grinding process is continuous and requires necessary loads [3] to grind coarse product into fine particles. During operation, the pinion's shaft is subjected to different loading modes, these are their own weights, static loads, and dynamic loads due to an alternative function, machining, or assembly defects. The bearings, on which different loading modes have a great influence, are the main elements that support the radial loads and the rotating guidance during the operation and become the cause of many incidents [1]. Pinion bearings consist of an outer and inner bearing rings and rolling elements. Lubrication of these bearings is not oil wedged nor pressurised. However, the internal dynamic pressure within the rolling elements can variate depending on grease injection and release rates. In case of no grease release provision, the pressure increases so the temperature. This can also be characterized by the grease thermophysical behaviour in the bearing assembly considered as a closed space [4]. In a steady-state lubrication and operation conditions, the friction coefficient between rolling elements and races is minimum and this is theoretically and experimentally well understood [6]. However, it is rare to maintain steady-states conditions in real applications. Hence thermomechanical stability of the rolling elements is subject to transient phenomena. This phenomenon is a result of load changes when the rolling element enters the loading zone, which can be caused by vibrations due to dynamic unbalances of the pinion shaft, surface irregularities, lack or over greasing, and surrounding constraints. Furthermore, the transient phenomenon can be caused by the interface interaction of viscous fluid and solid at very minimal dynamic clearance within bearing rollers and races. This interaction at solid-solid interfaces under dynamic conditions due to poor lubrication practice and/or regime, generates high temperature in the bearings. Where, the integrity of the bearings raises questions due to the decrease in thermoelasticity and surface conditions [4] [5]. The use of greasing lubrication is to minimize frictional forces at rolling elements and races contact points [1]. This process prevents wear and tear on the materials, and significantly reduces the coefficient of friction. The ball mill pinion bearings lubrication is supplied by an automated greasing system provided with an injection pump. The temperature on both drive end and non-drive end bearings increases after a period of run time with recorded temperature above 80°C as shown in Figure 1, which is considered as high against a maximum of 75°C as per the manufacturing specification. Evidence suggests that excessive greasing would be the root cause of the overheating problem. Excessive amount of grease causes rollers to work harder to push the grease out of the races. No provision in the current bearing housing to relieve or remove excess grease indicates a failure in design for reliability (DfR).

II. LITERATURE REVIEW

Glovnea [6] has conducted research on the transient phenomena in hydrodynamic lubrication with a focus on both theoretical and experimental scales. This research further illustrates rolling element bearings lubrication behaviour in fixed geometry approaches Amaranth[11], gives an explanatory idea of bearing failure, where he studies an experimental test rig development to investigate the wear in rolling element bearings. The variations in micro-geometry of the bearing components have been observed using scanning electron microscope. Major cracks have been reported on the inner race surface and solid particles found on the inner race surface. The lubricating grease capacity to provide hydrodynamic lubrication between the inner race and the rolling elements failed[12]. Thus, conducted an experimental work to analyse the behaviour of two different greases considering the wear of bearing as context. The analyzed grease microstructure revealed that the lubricant film thickness decreases as the grease fibroid structure deteriorates [13], observed the effect of the variations in the micro-geometry in a rolling contact on wear processes through the development of shakedown and deformation mechanisms in rolling contacts. They concluded that the total deformation on roller bearings is a combination of macro scale deformation and plastic deformation of asperities [12]. However, Tasan, et al [13], performed experimental research and analysed the wear response of roller bearing along with its vibration response and demonstrated how wear rate increases with the increase in bearing temperature and reduction in lubricant film thickness in roller bearings[13].

III. LUBRICATION REGIME ANALYSIS

Maintenance data analysed throughout this study have shown that the root cause of bearings overheating is the build-up of lubrication grease in the Plummer blocks due to non-provision to drain old grease at the time of replacement. The existing greasing regime, via the lubrication pump, of 55g/day in average as shown in Table 1 has been reviewed with a recommendation of 25 g/day, which represents above 50% less grease injection into the pinion bearing system compared to the old regime.

Table 1: Grease regime via the pump

Grease Discharge in grams per shot : 1 shot per valve per cycle per bearing						
	NDE			DE		
	Valve A	Valve B	Total	Valve A	Valve B	Total
1	2.153	2.166	4.319	2.241	2.616	4.857
2	2.281	2.308	4.589	2.248	2.626	4.874
Average grams per cycle	2.217	2.237	4.454	2.2445	2.621	4.8655
Average grams per day	53.448			58.386		
Average Kilograms per month	1.60344			1.75158		

In addition to the greasing regime review, a preventative maintenance strategy review has been conducted with a recommendation of 6 monthly bearings inspection and grease change during plant major shutdown and a spare for full pinion assembly. The below figure shows the state of temperature trends for both drive-end and non-drive end bearings before grease regime review.

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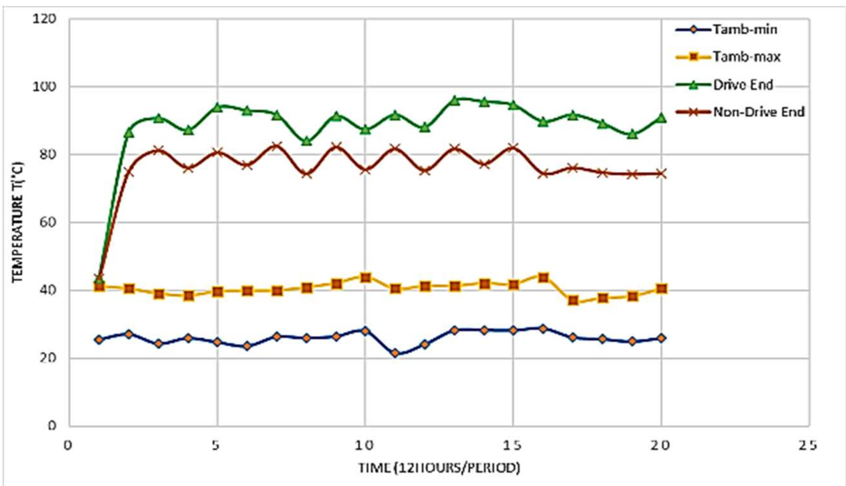

Figure 1: DE and NDE Bearings temperature before grease regime review

Figure 1 shows the temperature situation on both DE and NDE bearings during summer with ambient temperature ranging from 20-44°C. Recorded maximum temperature reached 95.9°C for DE and 82.5°C for NDE bearing. History revealed that 105°C was reached on the DE leading to an instant stoppage of the ball mill driveline.

IV. RELIABILITY AND MAINTENANCE DATA

To evaluate the reliability and maintainability of an industrial system, it is necessary to know the failure modes at which the system is submitted [23]. This knowledge makes it possible to create a database that will then be exploited for a modeling of its reliability. This section focuses on the evaluation and exploitation of reliability and maintainability models of the pinion bearing system. Mastering the reliability of a system is an important economic challenge for any company [24]. The measure of this greatness is an indispensable first step towards its mastery. The estimation and evaluation of reliability can be based on real data [20]. Two data sources are possible:

- Reliability test results
- System operating returns

Reliability covers multiple aspects: system failure analysis, predictive reliability, reliability databases, reliability tests, operational reliability, predictive methods of reliability and safety, reliability, and quality assurance [25]. In this case study, it is therefore important to know the failure mechanisms to determine the optimal architecture of the pinion bearing system and to evaluate its reliability. This work consists of the evaluation and exploitation of the reliability and maintainability of the pinion bearing system on a ball mill used in mineral processing plant by Pareto analysis [18]. The proposed Pareto principle for reliability and maintainability data analysis is to facilitate the determination of the pinion bearing's reliability problem with the most economical use of available resources and the elimination of failure causes by creating a Pareto plot of the failure data. Therefore, the following common reliability definitions and models are approached throughout the analysis:

Table 2: Common reliability models

Function	Definition
Reliability: $R(t) = \Pr\,(T > t)$	Reliability is defined as the probability that a system will perform its function during a given period and under given operating conditions.
Failure time distribution: $F(t) = \Pr(T \le t) = 1 - R(t)$	Probability that a failure will occur before a given time, indicated by failure law.
Probability density: $f(t) = dF(t)/dt$	
Failure rate: $\lambda(t)dt = f(t)dt/R(t)$	Probability that a system will fail between t and t+dt knowing that it was operating at a given time t.
Weibull's law: $F(t) = 1 - \exp\left[-\exp\left(\frac{lnt - \mu}{\sigma}\right)\right]$	General expression of Weibull's law.

Pareto analysis is a formal technique used when there are numerous possible courses of action raised [26]. In fact, the problem-solving tool calculates the benefit that each action provides, then selects the most effective activities that lead to the maximum achievable action. Pareto analysis is a creative method of analyzing problem causes since it stimulates and sets up cognition [27]. However, it may have limitations due to its exclusion of other significant problems that could potentially be considered, which may seem to be insignificant at first but with the potential to grow over time. The Pareto technique is effective when it is used in conjunction with other analytical methods, such as FMEA and fault tree analysis. This technique helps to identify the primary factors that must be addressed to resolve numerous problems encountered in a system. Once the primary causes have been identified, tools such as the Fish-bone Analysis can be used to determine the root causes of the problems. The below table shows pinion bearing's maintenance data in a sub-system set up within the ball mill's driveline

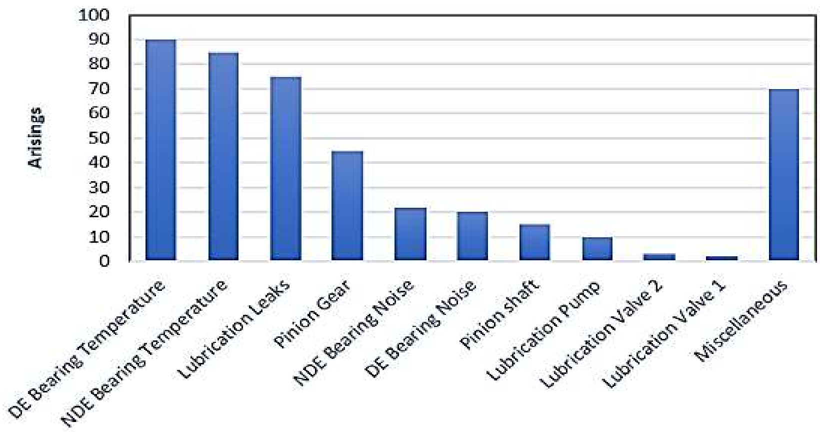

Figure 2: Pareto plot for failure data

Table 3: Maintenance data

Work Description	Budget Hours	Actual Hours	Budget Cost (AU$)	Actual Cost (AU$)
Change Ball Mill rotable pinion assembly	40	50	900	1500
Inspect and service Ball Mill pinion assembly	16	20	1200	1200
Fabricate & install new pinion gear cover	8	12	600	850
Investigate why the pinion is not engaging	8	8	600	1000
Open the 2 bearing caps for inspection	30	42	2250	2700
Strip, inspect and repair Ball Mill Pinion	24	30	25000	51548

The Pareto plot of figure 2 shows that bearing temperatures, lubrication system leak and the pinion gear are very critical failures in terms of their occurrence within the pinion bearing system and are to be looked at carefully. However, bearing noise, shaft and lubrication pump can be considered for condition-based maintenance, such as vibration analysis and thermography. The below figures 2 and 3 show maintenance time and cost history for a period of 6 months.

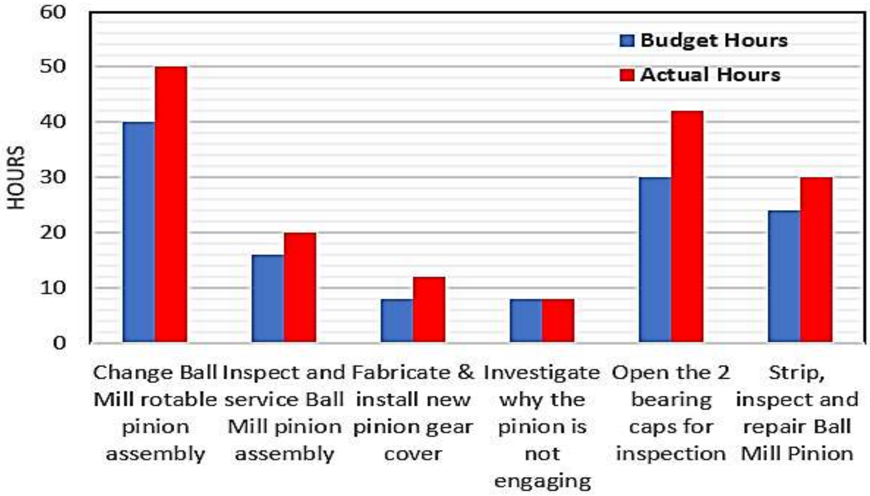

Figure 3: Maintenance budget hours vs actual hours

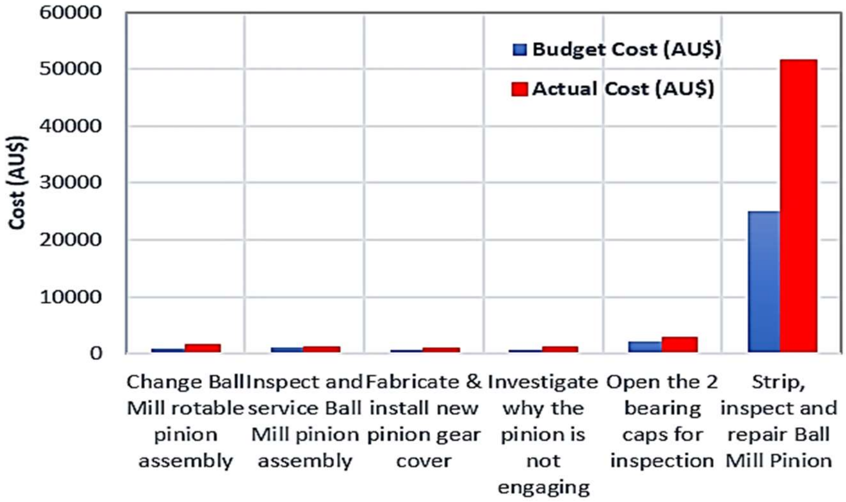

Figure 4: Maintenance budget cost vs actual cost

V. ACCELERATED LIFE TEST

Accelerated life test is the process used in reliability engineering to determine asset performance considering its failure time [28]. Throughout this process, the asset is subjected to excessive stresses to uncover defects in a short period of time [29]. The obtained results from accelerated life test are utilized to establish the asset's normal usage conditions or asset life model [18]. This process is mainly used for long life assets and under extreme temperature and pressure. In this case study, the accelerated life test is conducted on pinion bearings due to two factors: their long life-pan and criticality within the ball mill driveline. Accelerated life test evaluates the relation between asset life before failure and at least one acceleration variable[30]. It also addresses the below element for asset life modeling:

- Time for reliable components failure threshold
- Effect of an acceleration factor on asset life
- Factor's parameters that maximize asset life

The accelerated life test in this case was conducted based on a single parameter: bearing temperature 〖(T〗 _be) for both drive-end and non-drive end pinion bearings. Experimentally, subjecting the test bearings at high temperatures expands the inner and outer races diameters and reduces the clearance between the casing and the outer race [4], [5]. This phenomenon can potentially crack formation and eventually crack failure as shown in the below figure

.

Figure 5: Pinion bearing crack failure outer race

The results in the reduction of number of revolutions a bearing could possibly hold is expressed by the equation [5]: L_10=(C/P)^3 with L10 the bearing life in million revolutions with a 90% confidence, C is the dynamic capacity of the bearing, and P is the sum of radial and axial loads. The method used for the accelerated life testing is the inverse power law [16], with a single variable which is the temperature. The bearings' life ranges between 2 and 2,5 years as per the original equipment manufacturer's service life guarantee and operational constraints. The expression of bearing life under testing condition is given by:

Life(S)=A_F/S_i (1)

with Life(S), bearing life as function of stress, A_F maximum tolerated temperature ($A_F = 100°C$) and S_i given stress. The case study ball mill pinion bearings are cylindrical rolling elements type lubricated by an injection pump. The bearings are grease pre-filled on assembly by the manufacturer. Both drive end and non-drive end pinion bearings overheats after a period of run time with recorded temperature above 80°C which is considered over the acceptable temperature limit. The average ambient temperature during summer at site is 40°C. The consideration of ambient temperature during summer jeopardises the lubrication strategy. Hence an extra 250g is manually greased to cool the bearings down in high temperature alarm events. It is inevitable that bearing failure can be caused by material being subjected to high temperatures. The main high temperature failure mode on cylindrical roller bearings is softening, weakening, and melting of used materials, poor lubrication, or lubricant contamination. Cognisant of materials different thermal coefficients of expansion (TCE), mechanical stresses have the potential to set up within the bearings thus leading to fatigue failures, in the form of cracks on both outer and inner races. The reliability process rate of cylindrical roller bearings can be expressed by Arrhenius' Law given in equation $R = Kexp[-E_A/kT]$ [18]. Where: R the reliability process rate, K thermal conductivity of bearing material, E_A activation energy for the reliability process depending on the material, k Boltzmann's constant and T Absolute temperature. R increases by factor of 2 for every 10-20°C rise in temperature. This expression is used to calculate the failure rate of the bearings. The below figure is the design model for accelerated life test conducted on the pinion bearings.

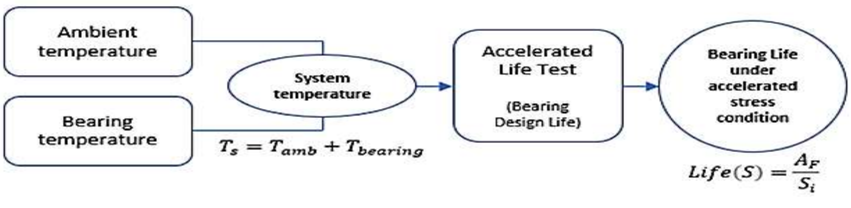

Figure 6: Accelerated life test design model

The below figures 7 and 8 show the results of Accelerated Life Testing conducted on both drive-end and no-drive end pinion bearings.

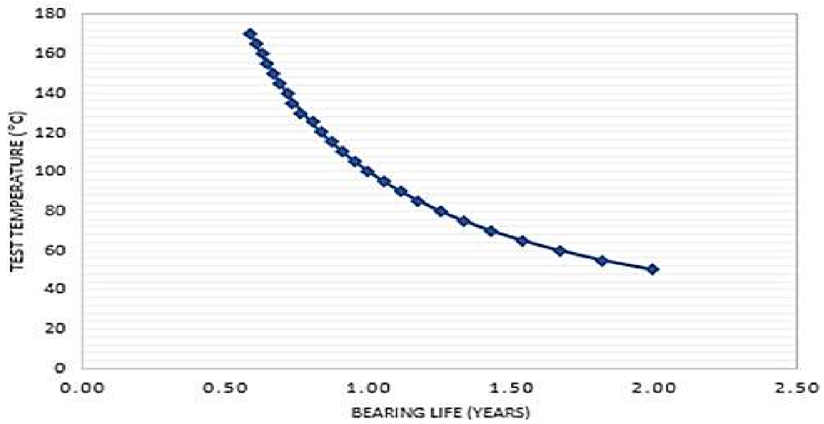

Figure 7: NDE Bearing (ALT) with an average operating temperature of 50°C min.

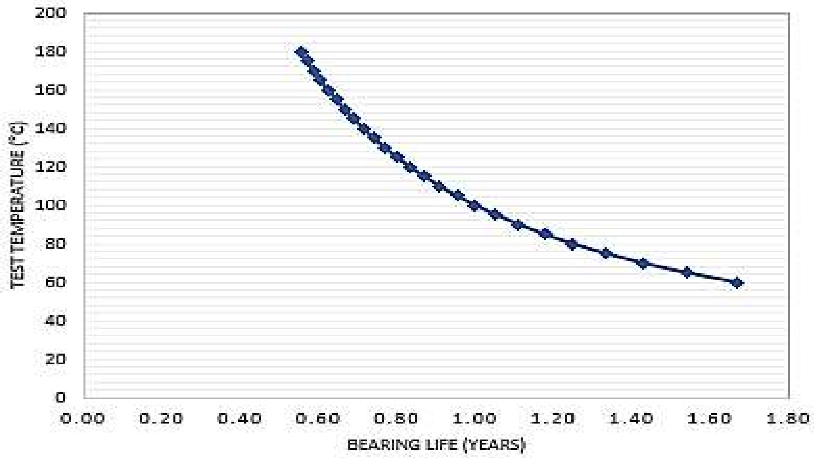

Figure 8: DE Bearing (ALT) with an average operating temperature of 60°C min.

The above figures show that consistent operating temperature of 80°C at minimum reduces the life and reliability of the pinion bearings with at least 40% lifespan reduction, which represents a major risk of sudden failure. The drive-end bearing has a minimum of 50°C test temperature, 10°C lower than the non-drive end bearing due to its proximity with the gearbox compartiment which is a high temperature environment. Corrective actions have been implemented to change temperature set-points in programable logic control as per below table.

Table 4: Bearing tempaerature set-points before and after ALT

	Before ALT		*After ALT*	
Bearing/Warning points	DE Bearing	NDE Bearing	DE Bearing	NDE Bearing
High temperature alarm	**80°C**	**100°C**	**75°C**	**90°C**
High-high temperature alarm	**80°C**	**100°C**	**75°C**	**90°C**

The bearing temperature set-points change shown in the above table is an important action post- improment in terms of ball mill driveline system reliability, and it is an indicator for operation stretch for fault tolerance viewpoints. History has shown that to replace the pinion in a planned shutdown event takes at minimum 12 hours. This is considering resource and spare parts availability. Thus, a breakdown on pinion bearings would take much more time to replace in case spare parts and equipment specialists are not available on site. This can lead to a possible production loss of over US$1000,000.00 on a single failure event. The recorded numbers of the ball mill driveline failures caused by high pinion bearing teamperature over a periode of 7 years are given in the table 4. The bathtub of figure 9 is representative of the ball mill driveline failure behaviour over the same period of runtime and could be a continuous pattern in the coming years of operation if no improvement works were done.

Table 5: Ball mill driveline maintenance data

Year (48.3week period)	Failure rate	Failure	MTBF
2016	0.10	5	9.66
2017	0.06	3	16.1
2018	0.04	2	24.15
2019	0.04	2	24.15
2020	0.04	2	24.15
2021	0.04	2	24.15
2022	0.10	5	9.66

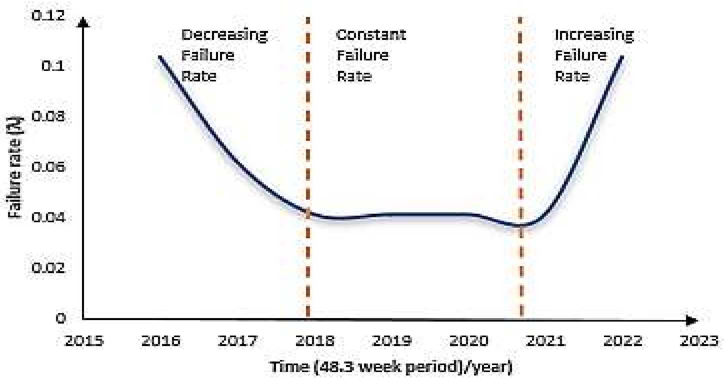

Figure 9: Bathtub curve for ball mill driveline failure rate.

The performance of the pinion bearings is vital to the whole ball mill operation. Lubrication practices within these elements have a direct effect on the ball mill drive line reliability and efficiency. An effective bearings lubrication with no chemical and/or metal particle contamination is capital to prevent failure and enhance plant availability. In the other hand the increase in temperature above 80□C as mentioned previously, has a negative impact on the pinion performance, hence the risk of unplanned shutdown in a failure event. Failure evidence shown in figure 9 is characterised by bearing material expansion and cracking due to thermomechanical stress. The production revenue for 100% plant availability over a period of 24 hours through the mill is estimated at $US1,092,400.00 for 200dmt of production capacity.

VI. THERMAL STRESS ANALYSIS APPROACH: HEAT GENERATION MODEL

The generation of bearing heat is related to the friction phenomena of the solid-solid interface; rolling elements and races in relation to the speed of the rotating parts which causes the sliding friction between the roller and the track, the sliding friction between the end face of the roller and the ribs, and the resistance to the flow of the roller.

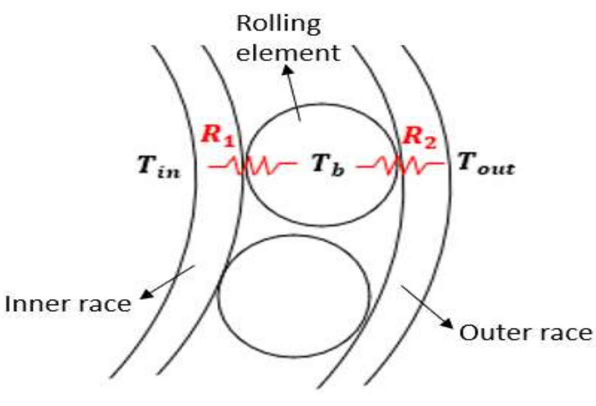

Figure 10: Thermal resistance model

The energy balance equations for inner and outer raceways are written considering the thermal resistance network shown in figure[31].

$$\frac{T_{ss}-T_{in}}{R_{S_{in}}}+\frac{T_b-T_{in}}{R_1}+\frac{T_l-T_{in}}{R_{l-r}}=m_{in}C_{in}\frac{\partial T_{in}}{\partial t} \quad (2)$$

$$\frac{T_b-T_{out}}{R_{S_{out}}}+\frac{T_{irs}-T_{out}}{R_{S_{out}}}+\frac{T_l-T_{out}}{R_{l-r}}=m_{out}C_{out}\frac{\partial T_{out}}{\partial t} \quad (3)$$

Where T_ss and T_irs are the temperatures of shaft and inner race respectively, with R_(S_in) and R_(S_out) thermal contact resistance between the inner race and the shaft, and thermal resistance contact between the outer race and the bearing housing [31] where he shows a temperature behaviour on a bearing housing as cooling condition with almost constant air convection constant

$$\frac{R_{l-r}(T_{ss}-T_{in})}{R_{S_{in}}}+\frac{R_{l-r}(T_b-T_{in})}{R_1}+(T_l-T_{in})=\frac{\partial \bar{T}_{in}}{\partial \bar{t}} \quad (4)$$

$$\frac{R_{l-r}(T_b-T_{out})}{R_{S_{out}}}+\frac{R_{l-r}(T_{irs}-T_{out})}{R_{S_{out}}}+(T_l-T_{out})=\frac{\partial \bar{T}_{out}}{\partial \bar{t}} \quad (5)$$

In predictive maintenance approach, the initial viscosity of grease and bearing initial temperature should be considered to determine the actual viscosity of the grease. The actual viscosity of the grease can be determined using the expression: μ_v=μ_vi e^(-β) (T_oil-T_i) with μ_vi and μ_v initial viscosity and actual viscosity of the grease respectively, and β and T_i viscosity-temperature coefficient and initial temperature respectively.

VII. RESULTS AND DISCUSSION

The below figures show that there has been significant change on the drive-end pinion bearing with a drop of the average temperature from 88.14°C to 59.48°C and a drop from 95.9°C to 65.2°C of maximum temperature. However, no major change has been recorded for the non-drive end pinion bearing. Maintenance data review has shown that the replenishment of grease from the non-drive end bearing was not conducted as defined in the preventative maintenance instructions. This has been classified as non-compliant maintenance practices and addressed to be rectified in the future.

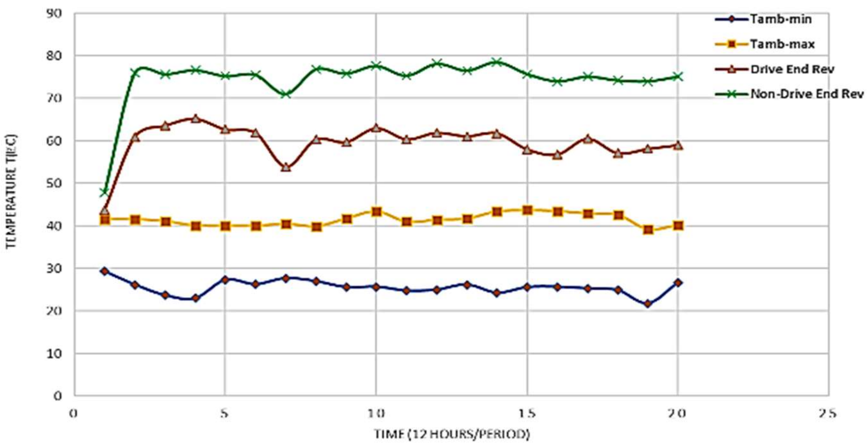

Figure 11: DE and NDE Bearings temperature after grease regime review

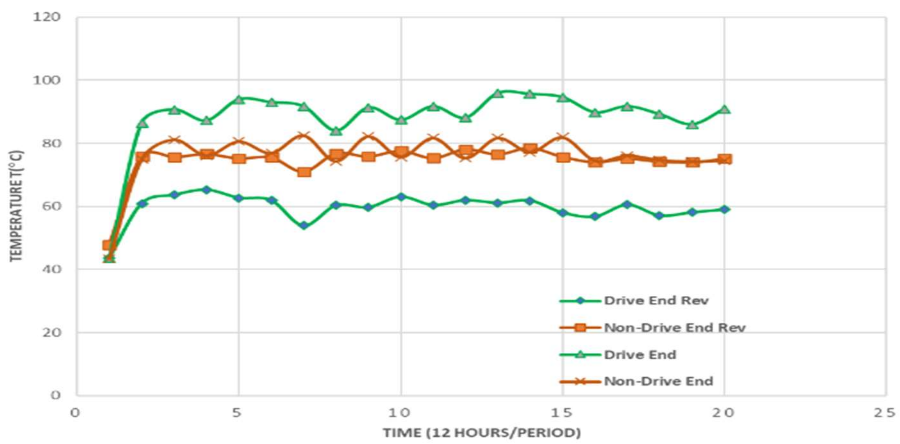

Figure 12: DE and NDE Bearings temperature before review vs after review

Figure 12 shows clear contrast between bearings temperature trends before review and after. The approach of maintenance data analysis significantly assisted in implementing immediate shot-term solutions to prevent bearings temperature increase and sudden failure that could potentially lead to a major production loss and maintenance cost. Although, there has not been considerable change on the NDE pinion bearing compared to the DE, a 75°C maximum operating temperature is acceptable during summer. The overall numerical comparison between the two regimes as illustrated in table 1 and section 3 with a variation of 30g/day in average is presented in the below figure 13.

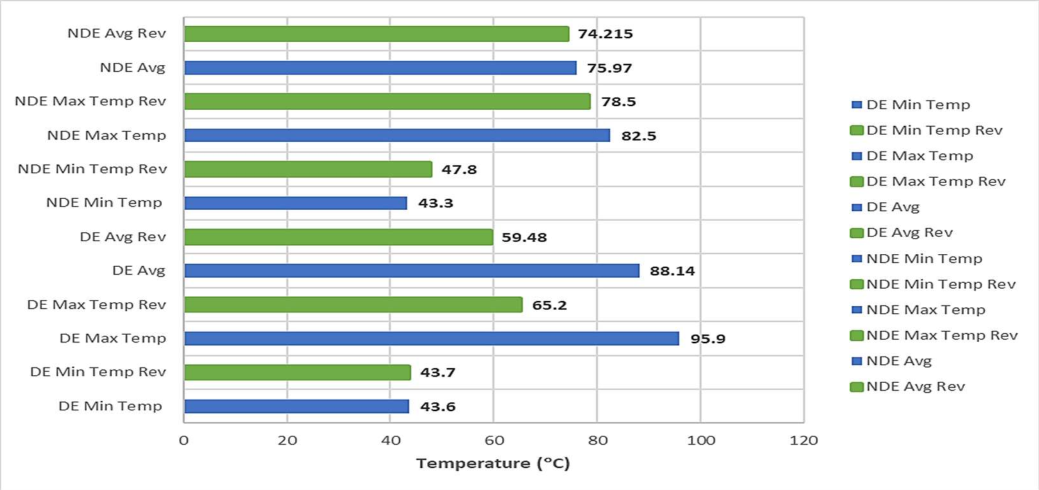

Figure 13: Overall temperature before lubrication regime review vs after review

Maintenance costs and hours as shown in Figures 3 and 4 have been significantly minimised for the first 5 months observation within the new maintenance strategy KPIs. The below figures show maintenance cost and time saving since the implementation of the new strategy adopted as part of immediate solution to prevent bearings overheating issues. The high actual cost and time are established based on 6 monthly preventive maintenance on both DE and NDE bearings which includes: strip, inspect and repair ball mill pinion bearings. This PM task is considered as statutory inspection within the new strategy with additional work scopes to cover maintenance planning and scheduling requirement.

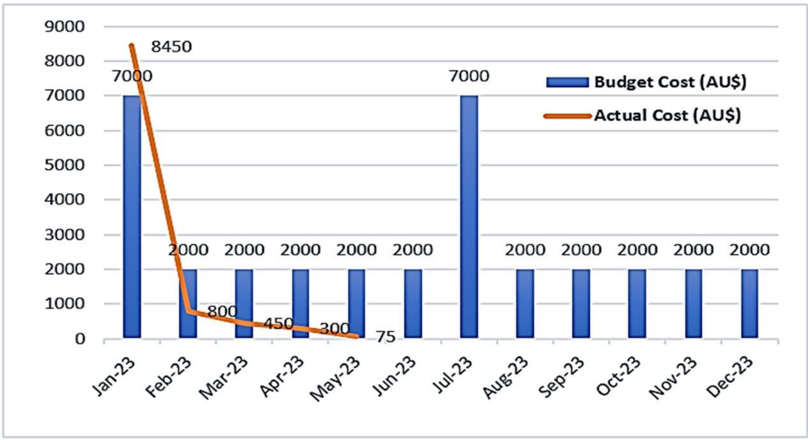

Figure 14: Reviewed maintenance budget after PM strategy change

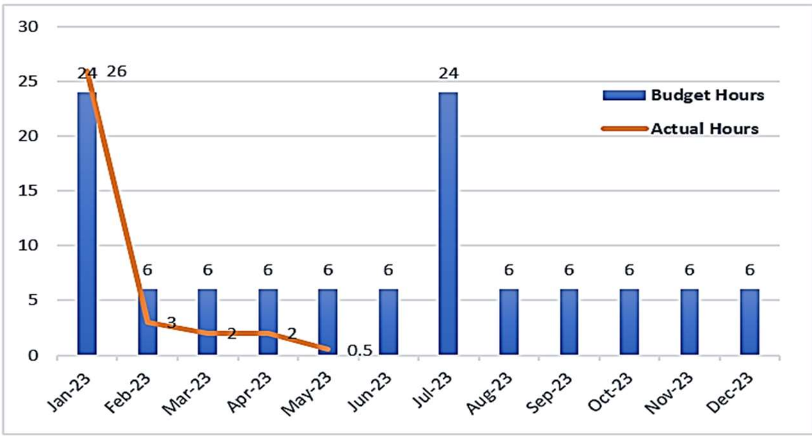

Figure 15: Reviewed maintenance hours after PM strategy change

This reviewed maintenance budget and run time have been rolled out for 2023 and is still in observation. The KPI was not met in January due to non-maintenance practice compliance on the non-drive end bearing as mentioned previously. However, a positive progress has been observed in February following with significant cost saving and time reduction in March and April. These positive impacts on maintenance cost are partially the result of the implementation of a consistent preventive maintenance and reliability techniques such as vibration, thermal and lubrication analysis adopted within the new strategy. The above changes have no risk on maintenance or production. On the contrary, they have positive impact on the overall ball mill drive line performance. As long-term solution, a complete change of lubrication system from grease to oil has been proposed and still under studies.

VIII. CONCLUSION

The execution of this project has shown significant improvement on the bearing's temperature with the application of maintenance and reliability applications from lubrication regime investigation as part of the new maintenance strategy, to maintenance practices and reliability. The application of accelerated life testing technique on the bearings revealed to be a successful indicator for an urgent consideration to propose long-term solution for a guaranteed reliability and maintainability of the ball mill drive line as an integrating system with fault-tolerance approach. Summer temperature as shown in figure 10 revealed not to be a contributing factor to temperature raise. Below are the outcomes of this project:

- DE temperature decreases from 88.14°C to 59.48°C.
- NDE temperature decrease from 95.9°C to 65.2°C.
- Lubrication regime and PM strategy review through reliability and maintenance data analysis has considerably contributed to obtain rapid solutions in the prevention of overheating.
- Grease replenishment on NDE bearing needs further improvement on PM method statement or personnel training.
- For a sustainable solutions RCM perspective, a possible study to transition from grease to oil lubrication needs to be conducted with consideration to economic approaches.

CONFLICTS OF INTEREST

There is no conflict of interest in the realization of this work.

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