

INTELLIGENT GEARING SYSTEM – SOLUTION TO UNDERSTAND AND CORRECT OPERATIONAL CONDITIONS THAT REDUCE THE FATIGUE LIFE OF A GIRTH GEAR

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Abstract

Horizontal rotating mills are being used all over the world associated with different mineral and power generation industries. David Brown South Africa, with over 140 years of experience in gear manufacture and design, has been supplying these industries with innovative designs at exceptional quality.

The gearing systems associated with these industries are challenged daily with operating conditions which are not always fully understood. Failures are therefore experienced, over time in the industry, which can therefore not always be fully explained.

David Brown South Africa has developed a gear monitoring system, the Intelligent Gearing System, in association with Transmission Dynamics (UK), in order to monitor and analyze these operating conditions in real time. The system has been designed on the principles of Six Sigma, incorporating Design for Six Sigma.

The purpose with this system design is to monitor the operating conditions on the pinion, record the information and analyze the data in relation to the rate of fatigue damage to the girth gear and pinion. This is done in real time and unattended. The output from the Intelligent Gearing System can be fed back into the operating system of the mill user, in order to display an alarm for operating conditions exceeding the design criteria as well as information on the rate of fatigue damage due to these operating conditions.

When the mill operating conditions are better understood, and the effect on the fatigue life of the mill gearing components, mill users can forecast critical spare holding levels with an increased accuracy. The Intelligent Gearing System also allows early actions to remove reasons for system overloads.

1. Introduction

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The gearing systems associated with these industries are challenged daily with operating conditions which are not always fully understood. Failures are therefore experienced, in the industry over time, which can therefore not always be fully explained.

In this article the author demonstrates, through experimental work done on a ball mill in service, that the current methods used in industry in general, are not adequate to prevent fatigue failures associated with girth gear alignment running conditions. SixSigma design philosophies, Design For Six Sigma (DFSS), is used to validate the IntelGear^(T) system through Design of Experiments (DOE), and other DFSS tools, which are developed specifically to overcome the problem of operational gear mesh misalignment. Refer to Picture 1.



Picture 1 Typical misalignment damage on a girth gear

2. Background

Different mills are used in the mineral processing and power generation industry. The work done in this article concentrates on horizontal rotating mills i.e. ball mills, but the same would apply for other types of horizontal rotating equipment. The work discussed in this article was done on a ball mill used in the power generation industry in South Africa.

The prime purpose of the mill is to pulverize the coal that is used for fuel for the boilers of the coal fired power generation plant. Refer to Figure 1 for a schematic description of the mill process flow.

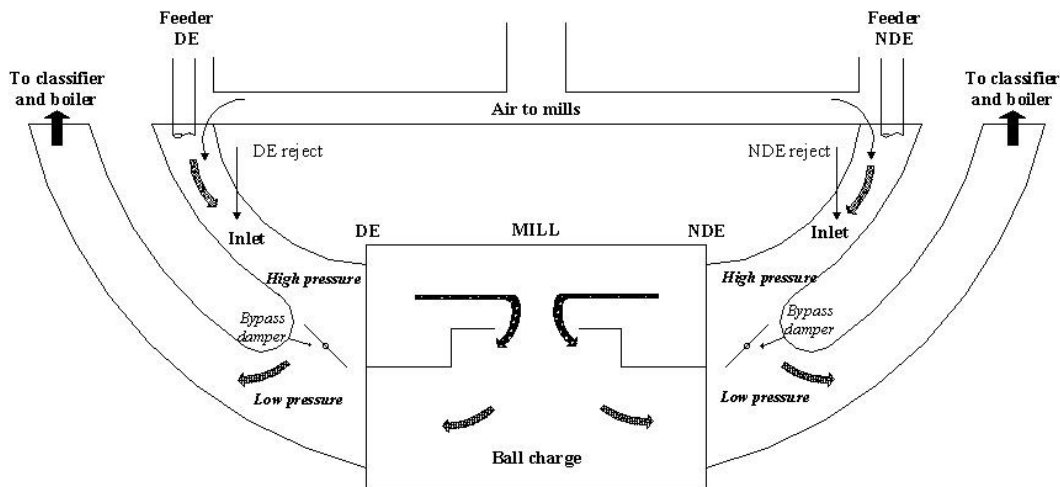


Figure 1: Schematic diagram of the milling process in the ball mill ^[1]

The coal is fed to the mill from both sides of the mill mixed with air. The milling process of the coal happens through the ball charge inside the mill where the coal is crushed through the rolling action of the balls on each other as well as the side of the mill. This continuous milling takes place until the coal is pulverized enough to be taken up by the air passing through the mill and then fed to the classifier and then to the boiler. The purpose with the classifier is to separate the coal that is not suitable for the combustion process yet and is therefore fed back to the mill via the reject feed pipes.

Adding more coal to the process changes the load of the mill but the ball charge stays constant. The amount of ball charge in the mill is a function of the condition of the mill liners.

The mill drive system consists of the main mill motor, the main reduction gear box, mill pinion and girth gear. The mill is also equipped with a barring drive which is either connected to the main drive reduction unit or alternatively mounted at the back of the main motor. Refer to Figure 2 for a schematic of the mill drive system.

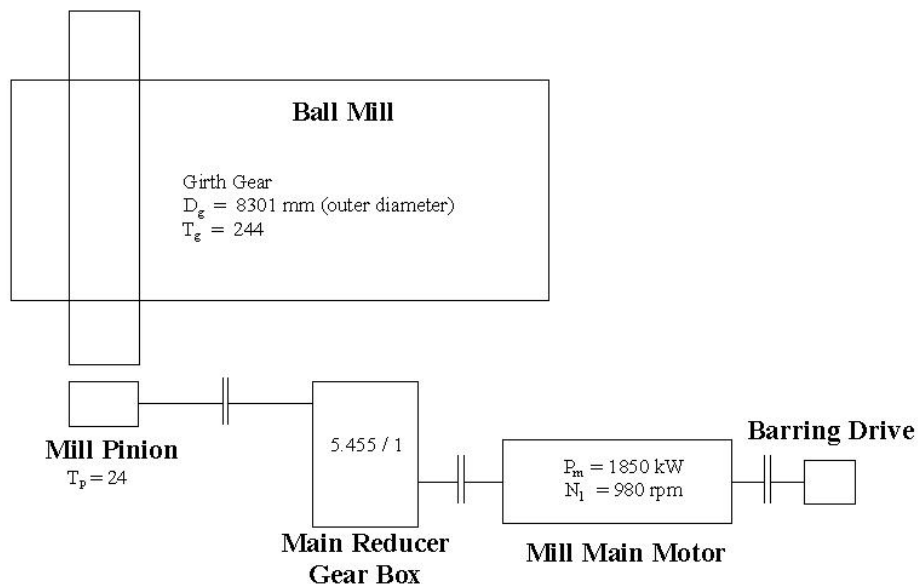


Figure 2: Schematic diagram of the mill drive system

The girth gear is considered as the most critical component of the mill drive system due to the replacement cost of such a gear as well as the lead-time for a replacement gear. The maintenance of these drive components is therefore critical for the economical operation of a mill over an extended period. Common maintenance techniques used in the industry include (a) vibration analysis on the mill pinion bearings, main reduction gear box and main motor, (b) oil analysis of the main reduction gear box, (c) spray pattern analysis on the girth gear, (d) stroboscope inspection of the girth gear teeth during operation and (e) temperature measurement across the face width of the girth gear. Annual shutdowns are also planned where NDT techniques such as MPI are done on the girth gear teeth to look for cracks. The gear teeth are then also inspected for abnormal wear and distinct wear patterns.

Over the years of girth gear inspections on site, David Brown has found that the most predominant failure mode is surface distress of the girth gear teeth (pitting), where the pitting is bias to the one side of the girth gear. Refer to Picture 1 of typical gear teeth with Pitting. This is always an indication of misalignment between the gear teeth of the mill pinion and girth gear. In most of the instances, the above maintenance techniques have been adhered to and the failure mode can therefore not be explained. Common practice is then to turn the girth gear around on the mill and run it on the back flank.

In order to maintain alignment of the girth gear and mill pinion, standard practice in the industry is to measure the temperature difference across the face width of the girth gear. The David Brown standard is a maximum temperature difference (ΔT) of 6 degrees Celsius across the face width of the girth gear. In the absence of any better method to tell the misalignment, have most mill users in the industry employed this standard.

The work that has been done and therefore reported by this article is to prove an alternative method to detect misalignment that is far more accurate and sensitive to misalignment than a DeltaT across the face width of the girth gear. The purpose with this study is therefore to prove the hypothesis that a temperature difference, DeltaT, is not sensitive and therefore not accurate enough to align the girth gear and pinion to its optimum alignment for continuous operating conditions under load.

The temperature difference, DeltaT, has a fundamental error due to the fact that the one end of the mill pinion is a free end and therefore exposed to cooling due to convection, while the drive end is connected to the main gear box and therefore exposed to heat due to conduction and radiation from the main drive. For a typical mill application, due to the weight of the pinion and girth gear, the soaking time of the mill system takes a substantial time and could therefore lead to “false” temperature gradients across the face width of the gear.

For a typical installation of 1800 kW, the losses in the main reducer gear box due to gear mesh and bearings could amount to 24 kW where the loss at the mill mesh could be 14 kW. The mechanical losses inside the mill, due to the milling process, are another source of heat generation. With the unbalanced heat input into the mill pinion and the rest of the thermodynamic forces due to cooling of the free end into the air, cooling by the spray system, etc. the temperature gradient across the face width of the girth gear and pinion may be significantly influenced by other factors, which are not related to gear misalignment.

It is therefore probable that the gear is set to run with a misalignment to compensate for the temperature gradient due to the thermodynamics of the total system. It is also a lengthy process due to the heat inertia of the girth gear and pinion in order to detect a DeltaT after an adjustment has been made.

3. Design Principles

There are several standards used in industry today to design the girth gears and pinions for mills of which the most common standards are AGMA 321.05, AGMA 6004 F88 and ISO 6336. The gears are designed for surface durability (pitting) and tooth bending strength. For the purpose of this study, only design for surface durability will be considered because the failure mode, misalignment, under investigation in this study is associated with surface durability.

The load distribution factor for AGMA 321.05 is read from a graph for gears with a face width of up to 1016 mm. AGMA 6004 F88 is a revision of AGMA 321.05 where the load distribution factor is limited to a maximum of 2.

ISO 6336-1 method A, recommends that the misalignment be measured, but method B to D does show how to calculate the face load distribution factor. The face load distribution factor is a function of gear tooth manufacturing accuracy, alignment of the axes of rotation, elastic deflection of gear unit elements, bearing clearances, Hertzian contact deformations

at the tooth surface, thermal deformations due to operating temperature, total tangential tooth load which includes the dynamic and application factor to name a few. The advantage of method A is that the actual face load distribution factor is measured in operation and therefore already accounting for all the above-mentioned influences.

According to ISO 6336-2 the contact stress is calculated as follows:

$$\sigma_H = \sigma_{HO} \sqrt{K_A K_V K_{H\beta} K_{H\alpha}} \leq \sigma_{HP} \quad (1)$$

Where:

σ_{HO} = nominal contact stress at the pitch point

K_A = Application factor

K_V = Dynamic factor

$K_{H\beta}$ = Face load distribution factor for contact stress

$K_{H\alpha}$ = Transverse load distribution factor for contact stress

σ_{HP} = Permissible contact stress

From equation 1 it is therefore clear that the contact stress of a gear pair and hence its life, is influenced by the face load distribution factor which is strongly influenced by gear misalignment.

This study will now further focus on the face load distribution factor ($K_{H\beta}$)

4. Experimental work

Design of Experiments, DOE, has been used to proof the IntelGear^(T) pilot system and hence proof the hypothesis that a temperature difference across the face width of the girth gear is not sensitive enough to set the alignment for optimum aligned prolonged running conditions. The experimental work has been done on a ball mill at a coal fired power station in South Africa.

4.1 Experimental setup

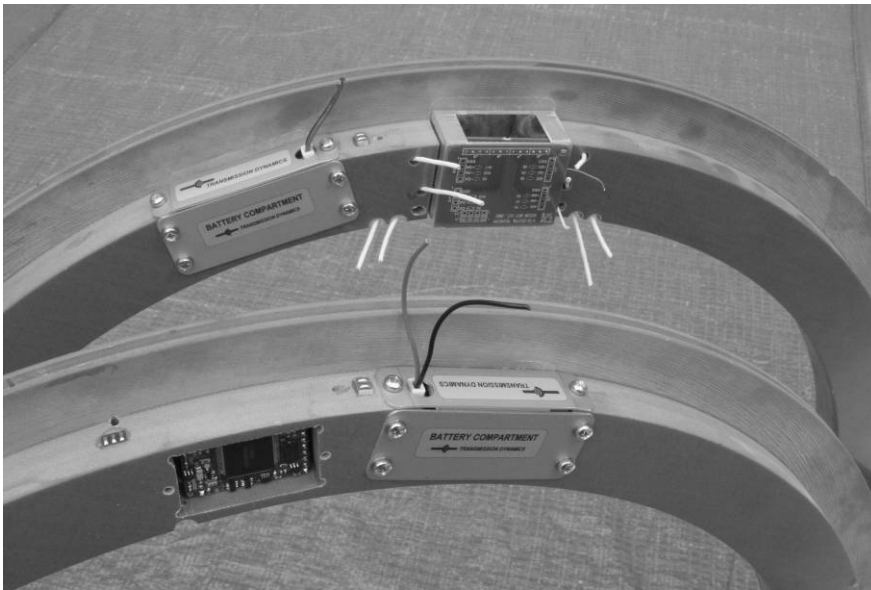
The experimental set up took 3 working days to install the IntelGear system on the mill pinion of the ball mill (Refer to Picture 2). The experimental system consists of two major components (a) the torque measurement module and (b) the gear alignment module which are both mounted inside a tufnol disk that is mounted next to the pinion gear teeth (Refer to Picture 3). The torque measurement module is used to measure the application factor, K_A , while the gear alignment module is used to measure the load distribution factor, $K_{H\beta}$, for the mill pinion.



Picture 2: Typical installation on a 1800 kW Ball Mill (courtesy of Kendal Power Station, South Africa)



Picture 3: Tufnol disk mounted on the mill pinion next to the gear teeth



Picture 4: Torque measurement module and gear alignment module inside Tufnol disk before installation

The temperature gradient, ΔT , is measured with a hand held Laser temperature measurement device that is commonly used in the industry today. After each alignment adjustment is made, the system was left to run for 12 hours in order for the temperature to stabilize, before temperature measurements were taken across the gear face width. In order to validate the experimental setup a Measurement System Analysis, MSA, was done on the temperature measurement system as well as on the IntelGear system. The purpose with the MSA is to determine if the variance measured is due to the variance in the measurement system, repeatability and reproducibility or if the variance measured is a true reflection of the response of the system being measured.

The alignment adjustments were made by installing pre-determined stainless steel shims under the mill pinion bearing housings. The DOE simulated the effect of misalignment from one end of the mill pinion through the mesh to the other end of the mill pinion. The alignment convention used was that, anti-clock wise relative movement of the pinion, (shim the drive end bearing housing), denotes a negative specific alignment and if the movement is positive, (shim the non drive end bearing housing), denotes a positive specific alignment.

The order in which these alignment adjustments were made was done according to the run order from the DOE. The standard order was randomized to prevent any special causes in the recorded data during the experiment that could lead to misinterpretation of the experimental data. One center point was added in order to investigate if there is curvature present in the output data. Refer to Table 1.

The DOE is designed as a 2k factorial with two factors (a) load and (b) alignment. The purpose with the DOE is to test the sensitivity of the IntelGear towards misalignment and

then also to find a prediction equation that describes the load distribution factor $K_{H\beta}$ as a function of the load and misalignment as described in equation 2.

$$y = f(x_1, x_2) \tag{2}$$

Where:

$$y = K_{H\beta}$$

$$x_1 = \text{specific alignment } [\mu\text{m}]$$

$$x_2 = \text{Load } [\%]$$

StdOrder	RunOrder	CenterPt	Blocks	Alignment	Load	$K_{H\beta}$
2	1	1	1	102.2	40	1.45
1	2	1	1	-102.2	40	1.63
5	3	0	1	0.0	60	1.17
3	4	1	1	-102.2	80	1.63
4	5	1	1	102.2	80	1.45

Table 1: DOE run order and specific alignment data per gear face width

The MSA was done by three people by taking four measurements with two replicates each. A R&R gauge study (crossed) was performed on the data to test if the variance of the data is in the data itself or in the repeatability and reproducibility of the measurement system.

4.2 Experimental results and discussion

The results from the DOE are tabled in Table 1 with a centre point at 0.0 μm specific alignment and 60% load. The results from the DOE demonstrated that the load change in the mill is not a significant factor to be considered at a confidence interval of 95% (Refer to Figure 3).

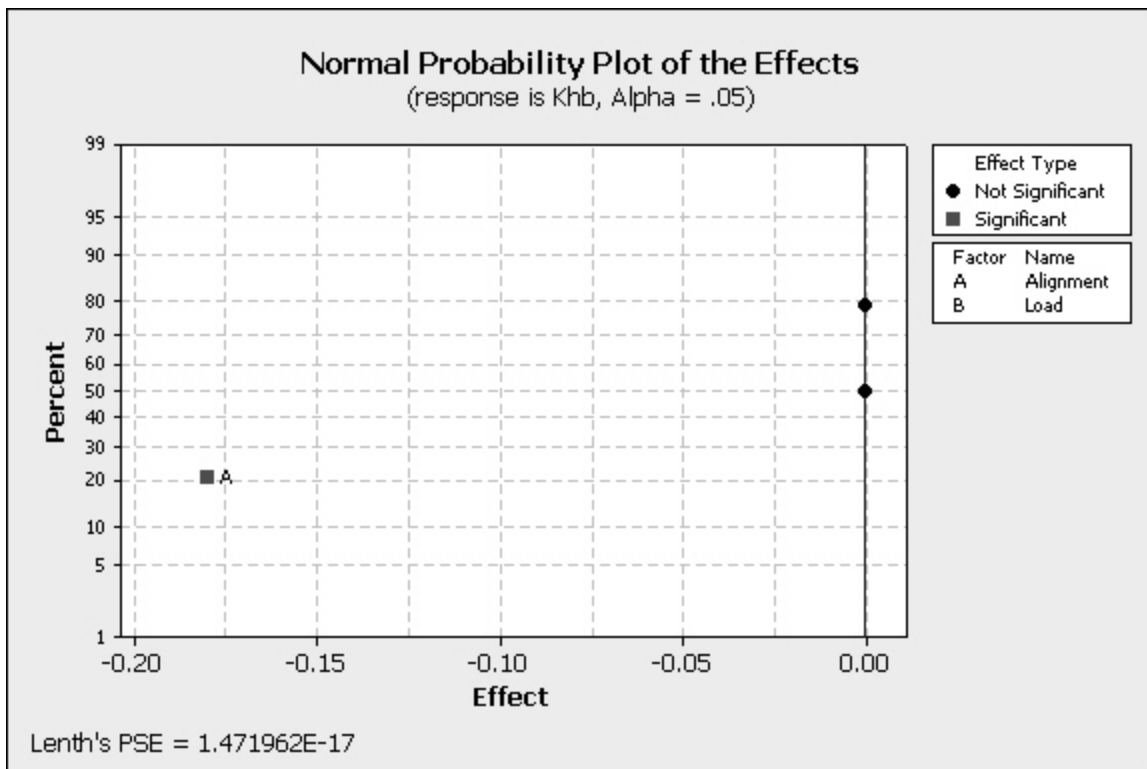


Figure 3 The load is not a significant factor with a confidence of 95%

The results further demonstrate that curvature exists in the data. Refer to Figure 4 for a main effects plot for the alignment. The centre point is offset to the data demonstrated by the line between the corner points. The centre point suggests that the value of K_{HB} decrease to zero and then increase again as the alignment decrease to zero, (perfect alignment), and increase again on the opposite end of the pinion.

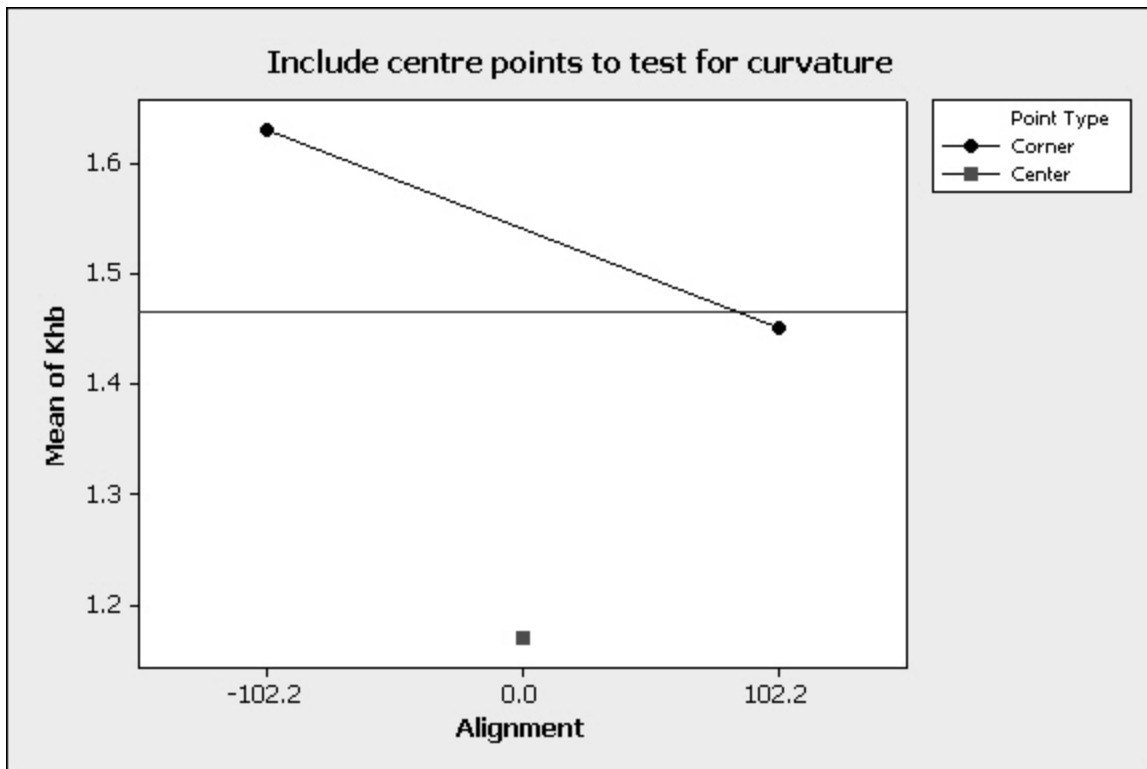


Figure 4 Centre point included to test for curvature

An extra set of data points were taken in between during the DOE for alignment conditions as displayed in Table 2. This data has been used to find the prediction equation by using a fitted line plot with a confidence interval of 95% and prediction interval of 95%. Analysis of the residuals reveals a normal distribution with no distinct patterns (Refer to Figure 5).

Alignment	K_{HB}
-102.201	1.63
-61.3208	1.52
-40.8805	1.17
-20.4403	1.29
20.44025	1.25
40.8805	1.3
61.32075	1.35
102.2013	1.45

Table 2: Additional data to find prediction equation

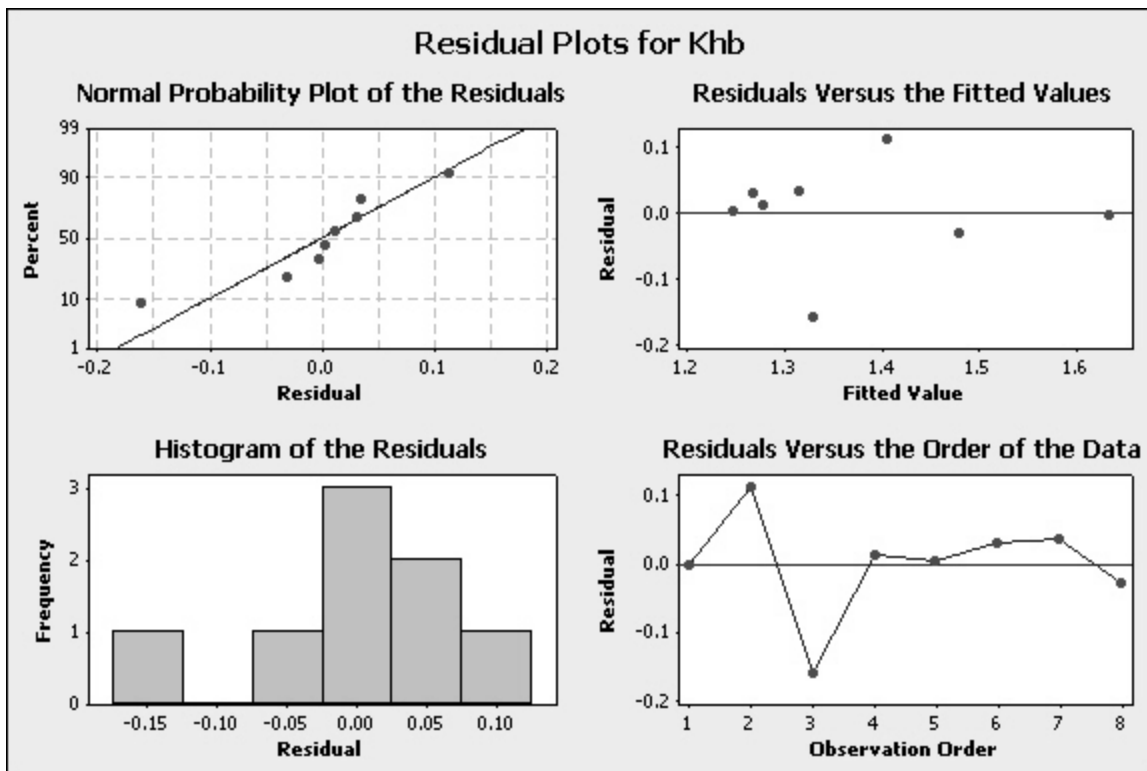


Figure 5: Residual plots for the fitted line $K_{H\beta}$

Figure 6 displays the prediction equation that is quadratic of nature. Figure 6 also displays the Confidence Interval as well as the Prediction Interval, both at 95%. This equation can therefore be used to predict $K_{H\beta}$ for a known specific alignment value.

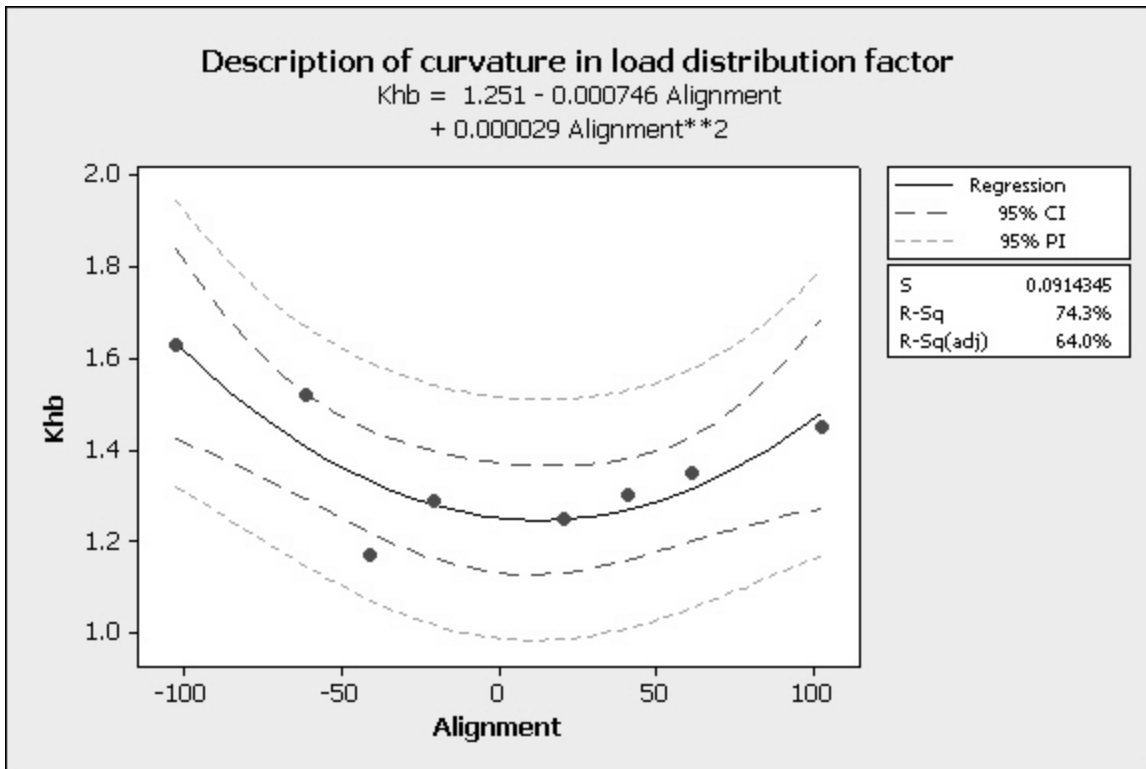


Figure 6: Prediction equation for $K_{H\beta}$

The prediction equation for $K_{H\beta}$ is therefore calculated as follows:

$$y = 1.251 - 7.46e^{-4}x + 2.9e^{-5}x^2 \quad (3)$$

Where:

$$y = K_{H\beta}$$

$$x = \text{alignment } [\mu\text{m}]$$

Based on recommendations from ISO 6336-1(method C)^[2] mesh stiffness is

$$C_\gamma = 20 \text{ N/mm}/\mu\text{m} \quad (4)$$

From the experimental torque measurements, a specific loading for the girth gear at a face width of 650 mm is calculated at

$$\frac{F_t}{b} = 550 \text{ N/mm} \quad (5)$$

The mean mesh deflection is therefore:

$$f = 27.5 \mu m \quad (6)$$

When gear misalignment errors are of the same order as the mean mesh deflection, the load distribution along the face width of the gear teeth, will become non uniform across the gear face width, which could significantly increase the localized stress at the mesh, which in turn could lead to fatigue of the gear teeth surface. In the industry today, with the existing employed methods, is it therefore extremely difficult to align the girth gear and pinion to 30 μm .

By substituting x in equation 3 with a value of +30 μm , a $K_{H\beta}$ value of 1.25 is predicted or -30 μm a $K_{H\beta}$ value of 1.29 is predicted. With this method explained by this study, is it therefore possible to align girth gear and pinion systems to 30 μm which, for this type application will give a $K_{H\beta}$ value of less than 1.3

The current method widely used in the industry, was also investigated. Temperature readings were recorded, refer to Table 3, but the measurement system used failed the MSA and no correlation could therefore be found. The system tolerance is set at 6 degrees Celsius.

Alignment	Temp Δ	Bias
	2	NDE
20.4	2	NDE
-20.4	2	DE

Table 3 Temperature recordings after alignment adjustments

From the analysis of the MSA the total % contribution (92.85%) to variance is in the Gage R&R, where most of the variance is in the repeatability (58.82%) but also in reproducibility and operator at 35% each. For an acceptable measurement system, the total % contribution to variance should preferably not be more than 1% but up to 10% can be tolerated pending the application. The number of distinct categories is only 1 where it should be 5 or more for an acceptable measurement system.

The variance Part to Part is very little at 7.15%, which therefore indicates that the DeltaT measured in Table 3 is of no significance and can therefore not be used to check for correlation. The analysis further indicated that the measurements taken by two operators were outside the control limits for the measurement system. Refer to Figure 7.

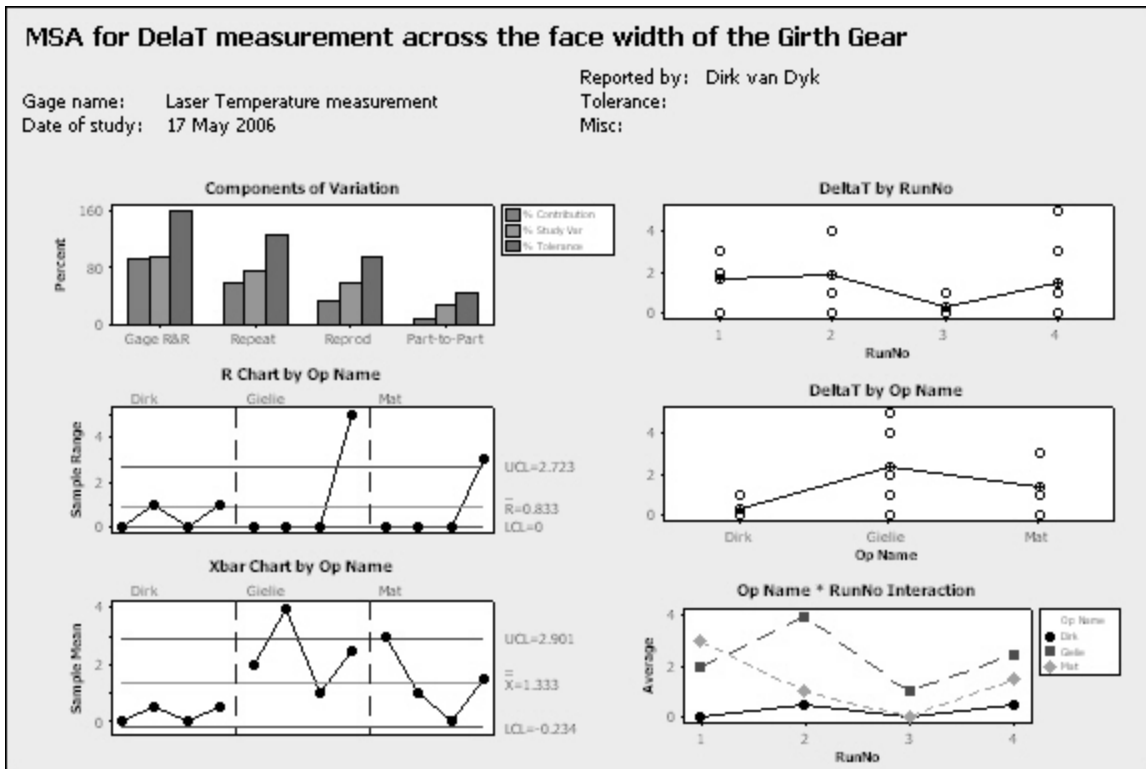


Figure 7: MSA results for R&R gauge study (crossed)

5. Conclusion

This study successfully demonstrated that the face load distribution factor, $K_{H\beta}$, can successfully be measured on typical mill applications by the IntelGear system. By measuring the $K_{H\beta}$, the alignment between the mill pinion and girth gear can therefore be adjusted much more accurately than the current systems used in the industry today. Being able to adjust the alignment accurately, the failure mode pitting, due to misalignment can therefore be eliminated. Misalignment being one of the most common gear failure modes would therefore be eliminated.

REFERENCES

- [1] Process flow as supplied by Mathew Miller, Plant Engineer, Kendal Powerstation
- [2] Calculation of load capacity of spur and helical gears – Part 1. Basic principles, introduction and general influence factors

