EVALUATION OF THE DYNAMIC BEHAVIOR OF GEARBOX STAND MILL LF 02 PASS 1 BEFORE PROCESS VARIABLES*

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Abstract

Gearbox Stand Mill LF 02 Pass 1 presented failures of internal components, such as bearings and gears. In order to determine its root cause, a dynamic evaluation of the Gearbox before process variables was performed. Thus, the following engineering tools were applied: Vibration Analysis, Operating Deflection Shape – ODS and Numerical Simulation by Finite Element Method - FEM. Vibration and ODS highlighted excitation frequencies, operational modes, which in these frequencies caused relative movements. Results obtained by Vibration and ODS were correlated to simulation, allowing to calibrate FEM model. Also, during simulation, Component Mode Synthesis technique was used, allowing evaluation of the fatigue life internal components. Based on field and numerical results obtained in this study, recommendations were: to verify alignment conditions, machine geometry using laser systems, and also application of structural modifications on the support of gearbox to eliminate a resonance condition.

Keywords: Process, Operational mode, Natural Frequency, Bearings, Gears.

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1 INTRODUCTION

Solutions for constant vibration problems frequently have been performed by customers themselves from trial and error. Such actions can be more time consuming than necessary, increase cost and it does not reach the objective. In order to obtain success in vibration problems solving, a detailed study should be performed. In this article, numerical methods for structures and rotating machine modelling were correlated to experimental techniques, such as Vibration Analysis and Operating Deflection Shape – ODS, to obtain robust solutions for constant vibration problems. Thus, this robust approach provides the best solutions without creating new problems, and consequently, there will be a better planning to repair actions and also reduce downtime.

Vibration Analysis was applied to guide Operating Deflection Shape Analysis. This evaluation was performed by studying operational mode obtained by the ODS technique in the excitation frequencies source. If the source is close to natural frequencies, ODS mode would describe an approximate mode shape of the natural frequency [1]. Additionally in numerical simulation, Component Modal synthesis technique was used, which enabled the best bearing arrangement for the gearbox, and also was evaluated gears and belt transmission fatigue life.

2 MATERIAL AND METHODS

2.1. Vibration Analysis

Vibration monitoring is performed using acceleration, velocity or displacement parameters. Such vibrational data measured in the field are dealt with Fast Fourier Transform (FFT), by which it is possible to transform the vibratory movements in the time domain to frequency [2]. With the signals in the time domain or frequency, it is possible to verify the excited frequencies and its respective vibrational levels along structures or rotating machines.

2.2. Operating Deflection Shape- ODS

Operating Deflection Shape – ODS is used to identify operational modes of structures or rotational machinery. ODS can be performed in time or frequency domain from vibration measurements of some structure points. These modes can be obtained by data measured simultaneously or measured data considering a reference signal. Measurements are performed in housing, metallic base, structure and frame machinery, and it can be in displacement, velocity or acceleration. Such vibration measurements performed from ratio, which is defined by Frequency Response Functions - FRF, when excitation force is feasible to measure, is given by Equation 1:

$$T_{ij}(\omega) = \frac{S_{ij}(\omega)}{S_{jj}(\omega)}, \qquad (1)$$

where, S_{ij} is the cross power spectral density between the force and acceleration response and S_{jj} is the auto power spectral density. In case of ODS, when the excitation force cannot be measured, a response signal is assumed as reference [3]. Transmissibility measurements are often used among ODS techniques performed from response signals as reference. It is calculated in the same way as the Frequency Response Functions, (Equation 1), but whereas the FRF is the ratio of the response divided by force, transmissibility is the ratio of the response divided by the reference response [1].

2.3. Finite Element Method - FEM, Component Modal Synthesis Technique CMS and ISO 6336: 2006 Method B Standard

The Finite Element Method – FEM is a general method which can be used to simulate the dynamic behavior of structures and rotational machinery or part of them (components). In FEM the structure is subdivided into a large number of finite elements. For a good accuracy, a Finite Element model usually consists of thousands of degrees of freedom with associated equations [3]. The evaluation of dynamic behavior of systems allows studying root cause of failures due to high vibration levels. In this article FEM was used to calculate natural frequencies and its mode shapes of frame planetary gearbox, which were correlated to operational modes from ODS. Another numerical technique, Component Modal synthesis - CMS, was used to model the dynamic problem that involves rotational parts (bearings and gears), consists on dividing a complex structure among substructures, and solving problems in an optimized way [4].

3 RESULTS AND DISCUSSION

3.1. Vibration and ODS Analysis

Instrumentation and software used in the measurement of vibration analysis, and in ODS measurement, as well as in the numerical simulation are described in the Table 1.

Table 1. Instrumentation and software used in the measurement of Vibration and ODS Analysis, as well as, in the Numerical Simulation

Collector/Analyzer (SKF):	SKF IMX-P;
Sensors (SKF):	Accelerometers CMSS2200
Management/analysis of measurement data	@ptitude Observer
3D CAD Software:	Creo Parametric 2.0;
ODS, FEM, CMS and Gears Software:	Dynamic Analysis, ANSYS® R1 2019 SimPro Expert 4.2 and KissSoft Release 03/2018D

In Tables 2 and 3 technical data of motor and gearbox is described. Such information is relative to operation conditions used during vibration measurement and ODS.

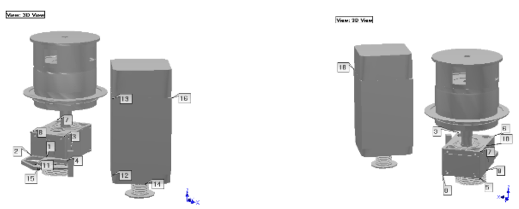
Table 2. Motor data

Equipment:	Motor Siemens 1PH7 226-2HD110EB7-Z
Power:	105 kW
Rotation:	1150 RPM (19 Hz)
Pulley:	Ø 192 mm

Table 3. Gearbox data

Equipment:	Gearbox Rossi Motoriduttori
Input speed:	613 RPM (10.2 Hz)
Output speed:	124.7 RPM (2.08 Hz)
GMF:	132.8 Hz
Bearing (input shaft):	32312 J (2X)
Bearing (output shaft): shaft):	32026 X (2X)
Lubricant:	Klübersynth EG 4-220
Pulley (input shaft):	Ø 362 mm
Tooth number (Z1):	13 (Pinion right angle helix)
Tooth number (Z2):	64 (left angle helix)
Belt	Trapezoidal SPB (8X)

Vibration and ODS measurements were performed using the following set up: 100 mV/EU; 1600 lines, end frequency 5,000 Hz, Hanning window and operational condition was high load. Figure 1a and 1b present all points where vibration and ODS measurements were performed on gearbox system.

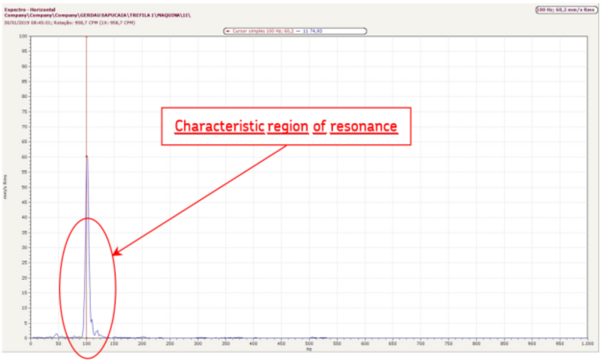


(a) (b)

Figure 1. (a) and (b): Measurement points used in the vibration and ODS analysis.

In the measured direction 11Z (vertical) and 9X (horizontal), the spectra placed on the belt support and gearbox frame, respectively, presented the following frequencies: 100 Hz and 500 Hz. Such frequencies that are highlighted in the spectra vibration represent 1st and 2nd harmonic of second stage gear meshing frequency, respectively, due to variable rotation motor.

In 100 Hz region is possible to verify exciting characteristic of natural frequency in Figure 2a, and in 2nd harmonic of the GMF (500 Hz) there is a gear meshing frequency misalignment according to [1] (Figure 2b). According to [1] and also [5], the base of energy spikes is very broad and may be made of many small energy spikes, which is on indication of natural frequency excitation.



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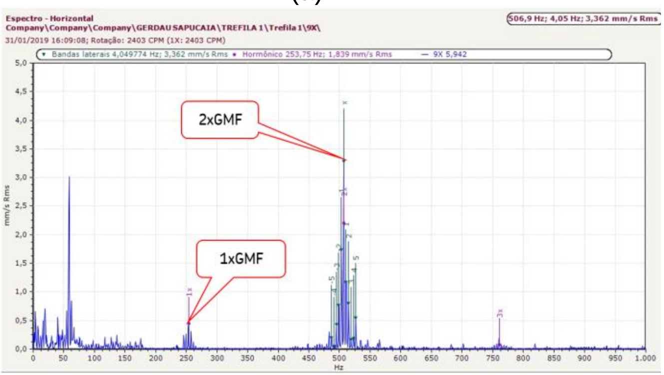

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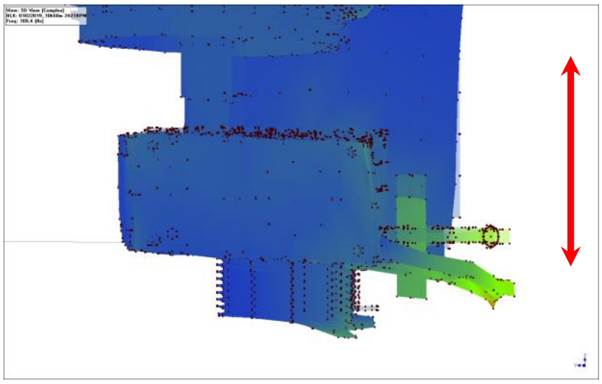
Figure 2. (a): vibration spectrum measured in the 11Z present characteristic of resonance and (b): vibration Spectrum measured in the 9X with misalignment event.

From ODS Analysis it was verified that the belt support presented operational mode that generated relative movement, which caused frame gearbox dynamic misalignment. Spectrum from Figure 2a measured in 11Z direction shows GMF (100 Hz) with characteristic of natural excitation frequency, and the operational mode obtained by ODS in this region frequency (109 Hz) shows the belt support displaces on the vertical direction (Figure 3a). This movement can cause amplitude of GMF sidebands increase, as verified in spectrum (Figure 2b). Spectrum of Figure 3(b) shows that dynamic misalignment in 500 Hz also occurred, thus operational mode in this frequency presents horizontal direction displace (Figure 3b).



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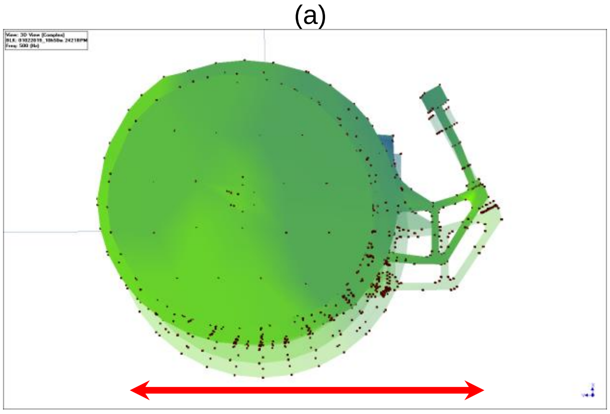


(b)

Figure 3. (a) and (b): Operational modes obtained by ODS in 109 Hz and 500 Hz frequencies highlighting the vertical and horizontal displacement, respectively.

3.2. Numerical Simulation of Gearbox System by Finite Element Analysis – FEM

Modal Analysis was used to calculate natural frequencies and its mode shapes respectively of gearbox system, and then, assure numerically the possible existence of natural mode shape. The 3D CAD (Computer Aided Design) model was built based on the technical drawings (Figure 4) and used in the FEM software.

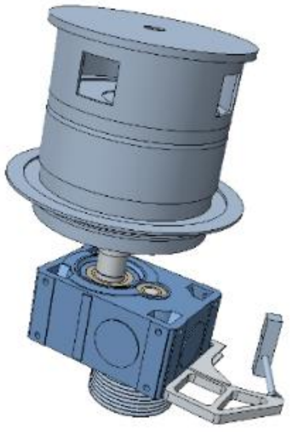
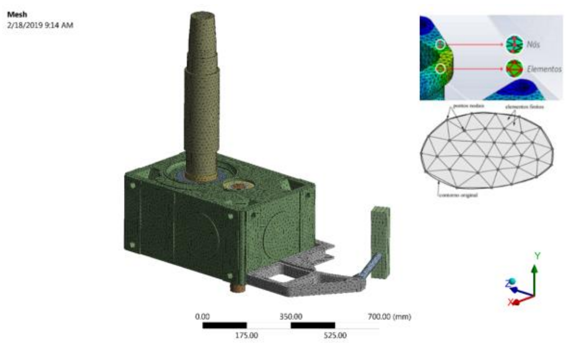


Figure 4. Gearbox System (gearbox, belt support, pulleys and drum).

In this FEM model was used the following material: cast iron, structural steel ASTM A-36 and SAE 8620 [6]. 3D finite element (MEF) model of the gearbox was discretized with a mesh of 136879 solid elements (249156 nodes) (Figure 5a). In this model, the element SOLID187 was used. It is a finite tetrahedral element that allows quadratic interpolation of the displacements and is suitable for use in non-uniform meshes. This finite element consists of 10 nodes and each node has 3 degrees of freedom, corresponding to the displacements in the directions x, y and z, according to Figure 5b.



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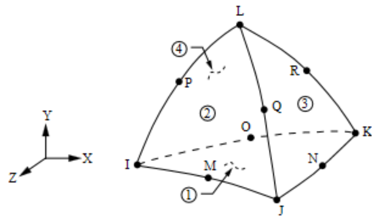

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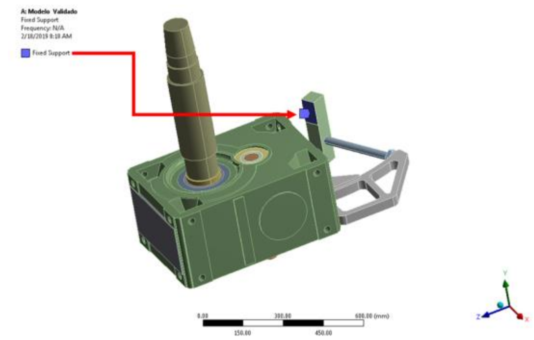
Figure 5. (a): element SOLID187 and (b): Mesh



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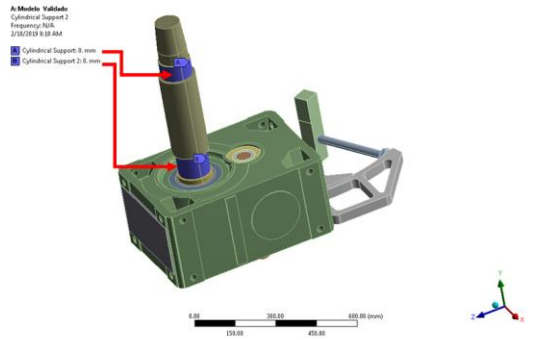

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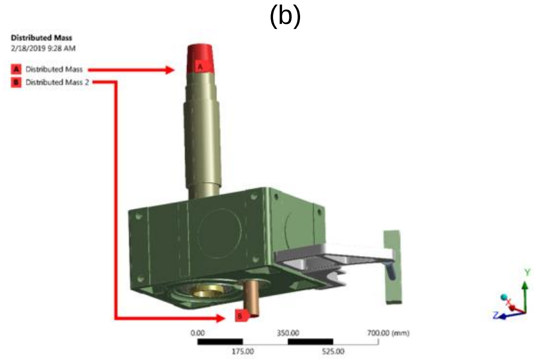


(c)

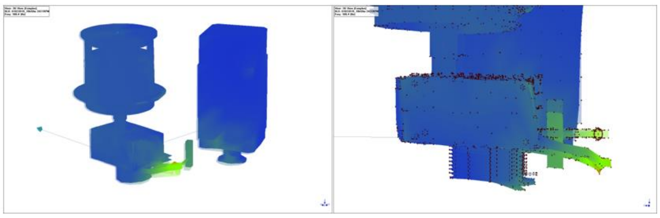
Figure 6. (a-c): Boundary Conditions applied to frame gearbox system.

Results obtained by Modal Analysis were first 15 natural frequencies and its respective mode shapes, represented in Table 4.

Table 4. Natural Frequencies of the first 15 and its respective mode shapes gearbox system

Mode Shapes	Natural Frequency
1	3.06
2	95.15
3	103.01
4	126.91
5	137.24
6	202.61
7	272.82
8	272.96
9	283.94
10	377.78
11	413.72
12	419.38
13	495.17
14	522.35
15	566.49

Operational mode in 109.04 Hz obtained by ODS, Figure 6(a) was correlated to mode shape in 103.01 Hz. Then, it was verified that operational and shape mode causes bending of pulley support in vertical direction, as showed in the Figures 6(a) and 6(b), respectively. Also, operational mode in 511 Hz was correlated to mode shape in 522 Hz, and was observed horizontal displacements, as showed in Figure 7(a) and 7(b), respectively. Therefore, it was observed that mode shape by FEM and operational mode by ODS in each respective frequencies presented the same dynamic behavior. Then, Finite Element model is calibrated, and so will be presented the structural modification proposal to displace the natural frequencies in 103.01 Hz, avoiding pulley support resonance, and consequently reducing vibration levels in other frequencies.



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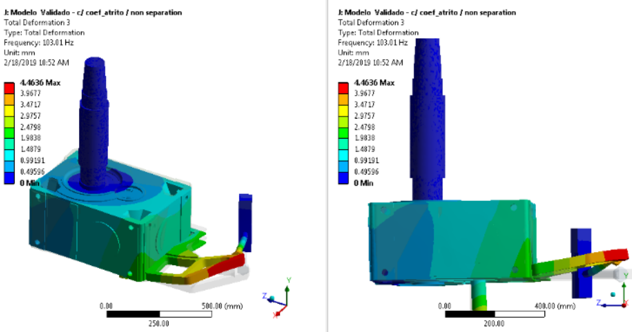

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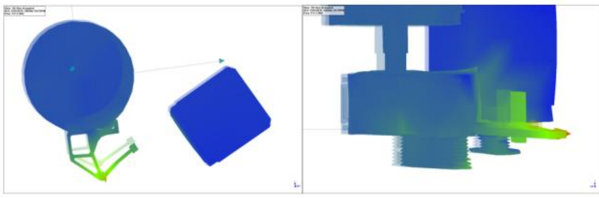
Figure 6. (a): Operational mode obtained by ODS in 109.4 Hz and (b): mode shape obtained numerically by FEM in 103.01 Hz.



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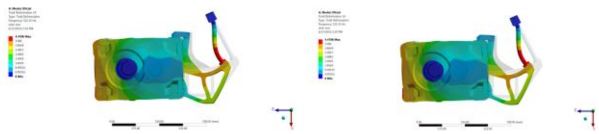


(b)

Figure 7. (a): Operational mode obtained by ODS in 511 Hz and (b): mode shape obtained numerically by FEM in 522 Hz.

Proposal for the solution to avoid resonance in 103.01 Hz region is presented in Figure 8.

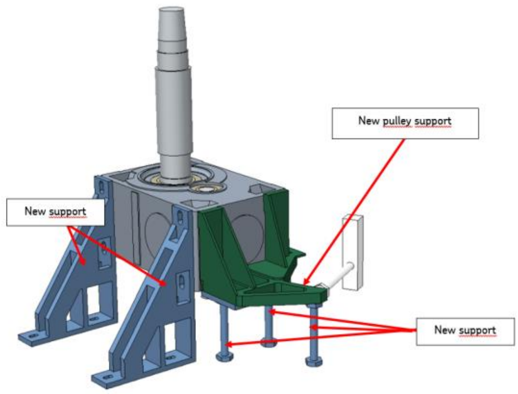


Figure 8. New supports applied to gearbox system.

Results of Modal Analysis obtained for the support proposal, presented in Table 5.

Table 5. Natural frequencies of the first 15 proposal modification

Mode Shapes	Natural Frequency
1	2.79
2	146.74
3	156.10
4	272.82
5	272.96
6	281.80
7	343.24
8	413.72
9	419.64
10	465.35
11	529.96
12	613.28
13	620.00
14	691.43
15	713.86

Based on the Modal Analysis results could be verified that the structural modification proposal obtained increase stiffness gearbox system, where the first mode shape in 103.01 Hz (current condition) was eliminated. This structural modification eliminated resonance risks.

3.2. Numerical Simulation of Internal Component of Gearbox

Internal components and belt analysis were evaluated from information in Table 1 and 6. Figure 9 and 10 shows CMS current and structural modifications model, and also boundary conditions applied to gearbox system.

Table 6. Load cycle of the gearbox system

Material	Linear speed [m/s]	Input Shaft Rotation Gearbox [rpm]	Power [kW]	% Time
CA60 3.40 mm	17.9	1103.4	83.2	26.35
CA60 4.20 mm	17.9	1321.6	78.4	15.81
CA60 5.00 mm	15.3	1259.0	58.3	28.01
CA60 5.60 mm	10.0	809.0	68.6	1.45
CA60 6.00 mm	10.0	920.0	77.5	28.18

3.2.1 Numerical Simulation – Bearings Fatigue Life

Fatigue life of bearing was evaluated to operational conditions and environment, where it operates. The fatigue life was obtained by three methods: Basic Rating Life, Modified Rating Life (ISO 281) and SKF Advanced Fatigue Calculation, which are considering the following condition:

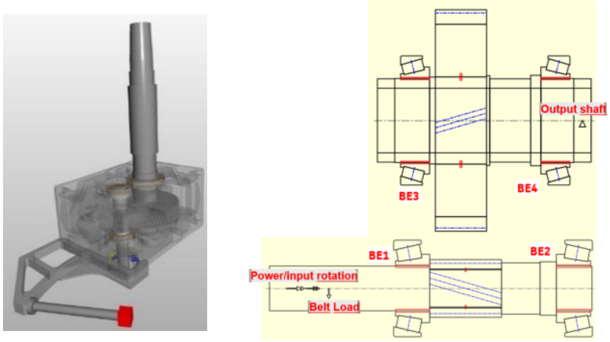
- Current condition, as shown Figure 9a and 9b;
- Proposal modification structural condition and new bearing arrangement, as shown Figure 10a and 10b.

Basic Rating Life: consider load and basic load ratings applied to bearings. Lubricant effects, contamination and internal bearing characteristics are not evaluated in this method.

Modified Rating Life (ISO 281): consider equivalent bearing load. Some factors, such as: lubricant, contamination and internal bearing characteristics are in the fatigue life calculation. In this method is not possible to evaluate internal misalignment.

SKF Advanced Fatigue Calculation: is obtained to subsurface stresses ring. In this case, were considered contamination, internal bearing characteristics and misalignment. Therefore, an accuracy estimative of fatigue life.

Obs.: water contamination is not evaluated in all methods.



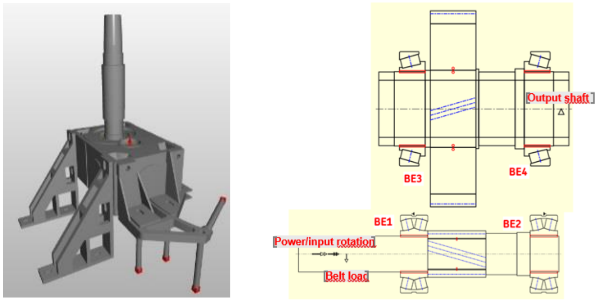
(a) (b)

Figure 9. (a): CMS current condition model and (b): boundary conditions applied to gearbox system.

Tables 7 shows bearings designation used to CMS model and fatigue life calculations, on current gearbox.

Table 7. Current gearbox – Bearings (TRB)

Bearing	Type	Designation
BE1	TRB	32312
BE2	TRB	32312
BE3	TRB	32026 X
BE4	TRB	32026 X



(a) (b)

Figure 10. (a): CMS structural modifications condition model and (b): Proposal modification structural condition and new bearing arrangement (SRB).

As there was not information about oil cleanliness of gearbox, according to ISO 4406, were considered in the fatigue life calculations, three levels of contamination: ISO-/15/12; ISO-/17/14 and ISO-/19/16.
Results of the Table 8 highlight that BE1 bearing (32312) has low fatigue life, due to high loads presents in this bearing. Also, it was observed in Table 8 that oil cleanliness has significant impact on fatigue life of bearings. BE1 (32312) was evaluated to high level of oil contamination by particles, and it presented 5700 hours, and low level it presented 29295 hours. Fatigue life calculated by SKF Rating Life of BE1 (32312) was compared SKF Advanced Fatigue Calculation, and it was observed that second method presented a reduction of fatigue life, because it considers misalignment between rings, differently of first method. 5.77' to 6.12' are misalignment values obtained to condition in Table 6, which are upper to limit of BE1, that is 4'.

Table 8. Fatigue life of bearing – Current condition

Variants	Bearing	Basic rating life [h]	SKF rating life [h]	SKF advanced fatigue calculation [h]
Load cycle –	SH1->BE1	15420	174231	29285
(According to	SH1->BE2	41578	898605	> 1e06
Table 6)	SH2->BE3	336909	> 1e06	823486
(ISO -/15/12)	SH2->BE4	> 1e06	> 1e06	> 1e06
Load cycle –	SH1->BE1	15420	30547	10132
(According to	SH1->BE2	41578	122324	268258
Table 6)	SH2->BE3	336909	399265	301115
(ISO -/17/14)	SH2->BE4	> 1e06	> 1e06	> 1e06
Load cycle –	SH1->BE1	15420	12502	5722
(According to	SH1->BE2	41578	43995	92636
Table 6)	SH2->BE3	336909	209560	180252
(ISO -/19/16)	SH2->BE4	> 1e06	> 1e06	> 1e06
Maximum load	SH1->BE1	7318	5196	4114
(According to	SH1->BE2	9435	7170	15434
Table 1*)	SH2->BE3	135220	71113	87825
(ISO -/17/14)	SH2->BE4	> 1e06	> 1e06	> 1e06

Tables 8 shows bearings designation used to CMS model and fatigue life calculations, on new bearing arrangement.

Table 8. New bearing arrangement (SRB).

Bearing	Type	Designation
BE1	SRB	22312 E/C3
BE2	SRB	22312 E/C3
BE3	TRB	32026 X
BE4	TRB	32026 X

Comparing the results obtained to condition with the structural modifications and new bearing arrangement applied to input shaft, it was observed relevant increase of the fatigue life. Considering the level of contamination ISO -/15/12, the fatigue life of the BE1 (22312E/C3) was about 90,000 hours (Table 9). [7] shows that typical fatigue life value for gearbox application of rolling mill are in the range of 30,000 to 50,000 hours (Figure 11), and therefore 90,000 hours is higher, and meets the modified condition (Figure 10), with a level of contamination ISO - / 15/12.

Table 9. Fatigue life of bearing – Proposal modification structural condition and new bearing arrangement (SRB)

Variants	Bearing	Basic rating life [h]	SKF rating life [h]	SKF advanced fatigue calculation [h]
Load cycle –	SH1->BE1	24518	359042	91117
(According to	SH1->BE2	197936	> 1e06	> 1e06
Table 6)	SH2->BE3	> 1e06	> 1e06	> 1e06
(ISO -/15/12)	SH2->BE4	> 1e06	> 1e06	> 1e06
Load cycle –	SH1->BE1	24518	56644	28827
(According to	SH1->BE2	197936	> 1e06	423240
Table 6)	SH2->BE3	> 1e06	> 1e06	> 1e06
(ISO -/17/14)	SH2->BE4	> 1e06	> 1e06	> 1e06
Load cycle –	SH1->BE1	24518	22219	15643
(According to	SH1->BE2	197936	333864	173441
Table 6)	SH2->BE3	> 1e06	> 1e06	> 1e06
(ISO -/19/16)	SH2->BE4	> 1e06	> 1e06	> 1e06
Maximum load	SH1->BE1	11399	8909	9832
(According to	SH1->BE2	21238	19879	24969
Table 1*)	SH2->BE3	477716	339938	559271
(ISO -/17/14)	SH2->BE4	> 1e06	> 1e06	> 1e06

Table 4

Guideline values for the requisite basic rating life L_{10h} for gearboxes for various applications

Gearbox application	L_{10h} (operating hours)
Machines and equipment infrequently used: Household appliances Agricultural machinery Medical equipment	300 to 3 000
Machines used for brief periods or intermittently: Cranes Lifts and elevators Construction machinery	3 000 to 10 000
Machines for daily (8 hour) use: Machine tools Woodworking machines Fans Conveyor drives Centrifuges	10 000 to 30 000
Machines for 24-hour use: Rolling mills Compressors Pumps Barges	30 000 to 50 000
Machines for 24-hour operation where high reliability is required: Cement mills Rotary furnaces Power generating plant Large-size open cast mining equipment Wind and water turbines Ocean-going ships	50 000 to 100 000

Figure 11. Fatigue life of gearbox application [8]

3.2.2 Numerical Simulation – Gears According to ISO 6336: 2006 Method B

Contact and bending Stress on gears were calculated according to International Standard ISO 6336: 2006 *Method* B, which is implemented in KISSsoft software. Following condition was considered: there is uniform contact between pinion and gear, and therefore, other conditions, such as: contamination by particles, water or improper mounting are not considered, consequently premature failure can occur. Also, misalignment is a condition that is not deal with by *Method* B. Figure 12 shows gears modeled in KISSsoft software.

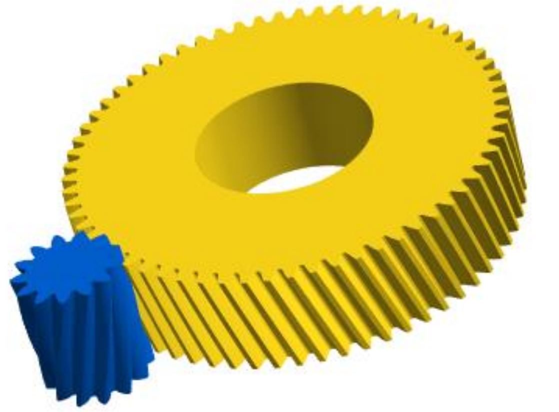


Figure 12. Gears modeled in KISSsoft software.

Gears analysis were evaluated from information on Table 1, 6, as well as, some parameters showed in the **Erro! Fonte de referência não encontrada.**.

Table 8. Parameters used to estimate gear performance

Parameters	Pinion	Gear	Obs.
Profile Shift Coefficient x	0.4506	-0.5019	Estimated from center distance = 200 mm. Objetive optimize the slide
Quality (ISO 1328:1995)	6	6	
Surface Hardness	60 HRC	60 HRC	Minimum required in design
Service Factor	1.0	1.0	Considering load cycle

Table 9. Results of gear performance calculations

	Estimate Life	Root Safety	Flank Safety
	H [hours]	SF	SH
Minimum Value	-	> 1.1	> 1.0

required					
Load cycle	> 1.000.000	4.965	1.166	2.749	1.000
Maximum Load	7349	3.981	1.166	2.273	1.050

Safety factor and estimate life of gears results, presented in the Table 9, have showed that there is not fatigue failures on gears considering the load cycle of Table 6. In maximum operational load and permanent condition, fatigue life is reduced to 7,000 hours, therefore, it is sensible to overload. To obtain maximum life of components, it should keep alignment between gears, lubrification and avoid overload.

3.2.3 Belt Transmission

Belt transmission was evaluated from SKF *Power Transmission Calculations* software, which it used information of that Table 10 and 11.

Table 10. Parameters of belt transmission calculations

Parameters	Pinion	Gear
Speed [rpm]	1150	613.3 +/- 5%
Max. pulley diameter [mm]	200	400
Center distance [mm]	883 +/- 2%	
Power [kW]	96.8	

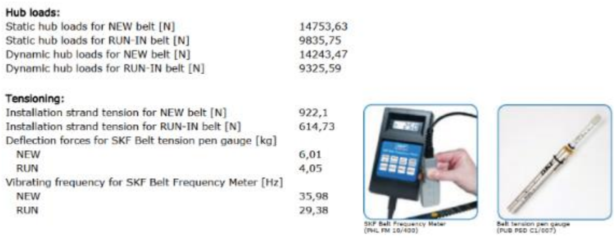
Table 11. Information belts

Belts	PHG SPB2650XP	PHG XPB2650
Nº Belts	8	8
Pulley motor [mm]	190	190
Pulley moved [mm]	355	355
Speed moved [rpm]	623.17	617.44
Center distance [mm]		
Belt speed [m/s]	11.9	11.9
Real Service factor	0.92	1.03

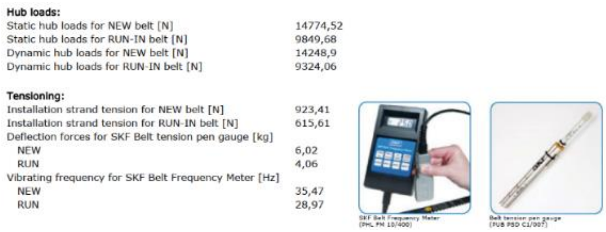
Obs.: SPB2650XP can operate in adverse environment, while PHG XPB2650 has high load capacity and it can be mounted with small pulleys.

Comparing information presented in Table 11, it was verified that quantity of belts is according to application. The difference would be the service factor of each belt. SPB belt presented service factor lower than 1.00, and consequently, it will have a reduction in its life.

Other important point to avoid failures in bearing is the tension applied to belts and alignment between pulleys, according to Figure 14.



(a)



(b)

Figure 14. (a): Tension applied to SPB2650XP and (b): PHG XPB2650

4 CONCLUSION

The gearbox presented internal components failures (bearings and gears). A study was performed in this machinery through the following techniques: Vibration and ODS Analysis, Numerical Simulation (Structural, bearings, gear and belt transmission). Such techniques highlighted excitation frequencies with natural characteristic, operational modes, which in these frequencies caused relative movements. Results obtained by Vibration and ODS were correlated to numerical simulation, which allowed calibrate FEM model. Afterwards, a proposal solution was given: structure (support) and bearings arrangement, avoiding system resonance.

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