

A METHODOLOGY FOR APPLYING THE CHOY-WILLIAMS TRANSFORM TO QUALITY CONTROL IN AUTOMOTIVE GEARBOX PRODUCTION

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Abstract. *In this article a methodology based on the energy of the temporary segments associated to the tooth meshing frequencies and their lateral bands of the Choy_Williams Transform applied to vibration signals for quality control in automotive gearbox production is presented. A sensitivity analysis applied to simulated data in order to evaluate the potentialities of the proposed technique is made. Finally, the results associated to the measured vibration signals in a group of noisy gearboxes are compared with the results of a standard gearbox.*

Keywords: *Quality control, gears, signal analysis, automotive gearbox, vibrations.*

1. Introduction

Of the several mechanisms used for power transmission, the gearboxes are the simplest and most efficient. Their advantage resides in their capacity to broaden the operation range of mechanical systems thanks to the possibility of varying the rotation and the torque supplied by an engine. This characteristic, coupled with their robust, compact and reliable construction features, contributes to the widespread use of transmission systems with gears in numerous industrial sectors, allowing that the electric or combustion motors, present reduced dimensions and smaller complexity. This aspect is extremely important in any area of engineering, mainly in the automotive industry.

Due to the importance of the transmission boxes, a great amount of research has been undertaken with a view to extending their lifetime. Among the several research lines, the study of the factors related to the project, such as the effects of inadequate geometry, is one of the most important (Howard et alli, 2001; Amabili and Fregolent, 1998). Another field of research where the largest number of papers is found is the vibration signature analysis (Braun, 1986) that aims to predict the state of the gearboxes as prematurely as possible (El Badaoui et alli, 2001; Choy et alli, 1996).

However, in this paper a distinct focus from the two lines described in the previous paragraph is carried out. This approach consists in the use of non-traditional signal analysis techniques, not to detect flaws, but for the purpose of quality control in automotive gearboxes at assembly lines.

It is interesting to observe that the rejection of the gearbox tests is associated to some problem in the production or assembly line. Usually, after noisiness is verified, the definition of the causes that resulted in a problem is made by means of dimensional tests in all the gears, axes and housings that comprise the gearboxes. Considering that the complete characterization of a gear implies dozens of metrological procedures, conducting such verification is uneconomical and time-consuming.

The use of the classic tools of predictive maintenance by vibration monitoring applied to the quality control of automotive gearboxes has not proven effective due to the great number of gears involved, transmission ratios close to the unit and the fact that the gearbox is tested in a void (Guimarães, 2000), implying in great fluctuations in the vibration signal phase.

Another complicating factor in this system is that all pairs are simultaneously geared, although just one is actually conveying power, which means that a defect in any one pair will manifest itself, even if it is not conveying power. This problem is magnified when the defect is in the differential, as the effect induced by the latter is present in all gears, and thus requiring a specific methodology for determining the faults associated to this pair.

Noise also hinders the signal analysis making its interpretation more difficult. Carrying out the time domain average procedure to minimize the effects of these spurious inputs is quite an effective tool (Staszewski and Tomlinson, 1994). This procedure reduces noise and also minimizes the inconvenience of the resulting effects of the other geared pairs (Brie et al., 1997, McFadden, 1987a, Lin et al., 1997).

Theoretically, gear meshing vibration signals present a constant period; however, due to nonlinearities inherent to the process, the signal spectrum contains harmonics of the gear mesh frequency in addition to the fundamental frequency (Randall, 1982). Inconstant spacing between teeth results in contact out of the primitive diameter, causing frequency modulations that have as carriers the mesh frequency and their multiples, while irregularities in the teeth contact surfaces or axes eccentricity cause amplitude modulation, whose frequency is associated to the axis rotation (Randall, 1982; Brie et al., 1997, Ma and Li, 1996).

Part of the reason for existing techniques to have had limited applications resides in the need for specialist interpretation of their results. An approach that facilitates this task consists in obtaining a significant and reduced dimension parameters group. To this end, some mathematical or statistical techniques can be used, ranging from drawing out the representative values of the data -- such as peak values and root mean square - to the use of mathematical operators – such as wavelet transforms or resolution processes, denominated Matching Pursuits (Mallat and Zhang, 1993; Lépore Neto et alli, 2001; Jaggi et alli, 1998) or by the use of the Generalized Stockwell Transform (McFadden, 1999).

Bi-dimensional transforms, such as the Short Fourier Transform and Wavelets can be also used in the reduction of data. Staszewsk and Tomlinson (1997) applied the Short Fourier Transform to detect flaws in teeth of gearboxes wherein, to facilitate the signals analysis, an indicative parameter of the flaw condition was proposed for each narrow band spectrum of frequency.

The increase in processing and data storage capacity in computers since the 90s made possible the implementation of time-frequency tools for signal processing, which offer advantages in relation to time or frequency domain techniques. In general, defects in gear systems generate transient components of vibration that can be identified through the extraction of the frequencies content excited by the defect in each instant of time (Wang and McFadden, 1993). Guimarães (2000) used bi-dimensional techniques in vibration signals from automotive gearboxes tests, performed with variable rotation, and therefore highly non-stationary.

Among the techniques for analysis of non-stationary signals, the Wavelet Transform is actually the most used due to its low computational cost and efficiency in compacting data (Gregoris and Yu, 1994). The method's philosophy consists in correlating the signal with a pre-determined function (wavelet mother), which presents similarities with the desired characteristics. The bandwidth variation for this function especially projected for the application of interest enables the visualization of all the short and long duration details in the signal. Therefore, all the vibration components provoked by defects in gearboxes can be characterized through this technique, provided a function that correlates with the characteristics of the defects is used (Wang and McFadden, 1996). Wavelet Transforms (Newland, 1993) have already been used as a pre-processor by Paya and Badi (1997) for obtaining relevant characteristics of gear and bearing vibration signals.

In another class of time-frequency tools, the density of energy per unit of frequency for each time instant is calculated (Cohen, 1995). The implementation of this concept results in the Wigner-Ville Distribution (Kadambe and Bartels, 1992) that has as an inconvenient the occurrence of energy values in areas where they not should really exist (cross terms of interference). For defect detection in gearboxes this constitutes a great problem because for signals with several basic frequency harmonics the components effectively caused by defects are masked (Guimarães, 2000). The Choi-Williams Distribution, in which filters are used to attenuate cross-term interference (Choi and Williams, 1989), results in a viable analysis tool with the advantage of possessing a greater resolution than that obtained by the Wavelet Transform (Jones and Parks, 1992). These time-frequency tools are very powerful, yet require a deep knowledge for interpretation of the maps (Guimarães, 2000).

In this work, simulated signals of gear meshing are used aiming to evaluate the effectiveness of the procedure based on the Choy-Williams Transform for defect detection caused by errors in the production or assembly of automotive gearboxes. Initially, a model for the gear meshing signal used in the numeric simulations will be presented.

Later, some considerations regarding the use of the root mean square applied to the time portions associated with the interest frequencies of the vibration signal -- seeking to aid in the interpretation of the data obtained by the Choy-Williams Transform applied to the simulated signals -- will be made. The evaluation of the efficiency of this methodology will be verified via sensitivity analysis, in which the effects of variation of the noise level in the signal and the magnitude of the defects will be analyzed.

Finally, the results obtained by the application of the proposed approach to the measured vibration signals in a gearbox without defect and in a faulty one will be compared and analyzed.

2. Model for the gear meshing signal

In the analysis of localized defects like tooth cracking or breaking, the transfer function variations about the gear meshing vibration can be assumed as relatively insignificant due to the high weighting factor of the periodic gear meshing signatures (Wang, 2001). Unlike what occurs with bearings, for instance (McFadden, 1986; Braun and Datner, 1979).

The tests in gearboxes are conducted in an automated way so as to ensure their repeatability. However, in the analysis of distributed and incipient defects featuring a low weighting factor in the gear meshing periodic signatures, such as those resulting from assembly or manufacturing errors, the path traveled by the signal -- even when the sensor coordinate remains constant -- has a striking influence in the amplitudes of the gear meshing frequencies and their harmonics. For the evaluated gearboxes, the displacement of the system's natural frequencies in relation to the expected values is of approximately 20%.

One way to measure the transfer function variations in the gear meshing frequency amplitudes in a theoretical model, would consist in conducting a model analysis in the gearbox and analytically determining an excitation force, which when applied in the obtained Frequency Response Function would result in the output signal of the system. However, the analytical formulation for this excitation force is quite complex, hindering the employment of this methodology.

The development of new diagnosis techniques to detect defects in gears presupposes the use of increasingly advanced models that take into account the effects of the different types of damage. This is necessary due to the impossibility of developing real prototypes which permit to vary, with precision and repeatability, the defects to be analyzed.

One methodology that has seen great development consists in obtaining dynamic models using the finite elements method in the exploration of several aspects, as in the analysis of the friction effect caused by the alterations in the geometry and in the material on the resulting vibration (Howards et alli, 2001). The finite elements model is also used for the determination of gear meshing rigidity in the bent and Hertz contact deformations, for the optimization of the evolving geometry (Brito, 1994; Brito and Lépore Neto, 1996). The great drawback in these approaches is the time needed to obtain the answers of the models which, associated with the great number of variables and simulated cases, would render its application prohibitive within the scope of this work.

Another procedure consists in the mathematical-analytical modeling as an auxiliary tool for the diagnosis. Bartelmus (2001) investigated the effects of the design factors, manufacturing technology, operation and changes in the conditions of use to evaluate the state of the gearbox system. The influence of the form deviations and assembly errors on the gear meshing dynamics were modeled by Velex and Maatar (1996), assuming that the deformed state is represented by a nonlinear system of concentrated masses with six degrees of freedom for the pinion and the wheel. Arakere and Nataraj (1998) studied the phenomena that happen due to the fatigue generated by the excitation resulting from centrifugal and dynamic loading on the teeth, generating radial cracks by fatigue in spur gears operating in high rotations.

In some situations, an interesting approach consists in modeling the signal of gear meshing vibration, not taking into account the design factors. To the simulated signals are applied signals analysis techniques to verify the detection capacity for alterations in the degree of defects and noise level (McFadden, 1986; Arato Jr. and Silva, D. G., 2001).

This approach will be adopted in this work to obtain the gear meshing vibration signals to be used in the analysis of the proposed technique.

Considering a geared pair with constant speed and load, different teeth numbers and assuming the hypothesis that the two gears are ideal, not presenting form deviations and constant spacing among teeth and there being no eccentricity, we can represent the gear meshing vibration by Fourier's series with a fundamental frequency equal to the gear meshing frequency (McFadden, 1987a). In this case, the time domain signal can be modeled by Equation (1).

$$x(t) = \sum_{m=0}^{M} X_m \cos(2\pi m f_e t + \chi_m) \tag{1}$$

Where X_m is the amplitude, f_e the gear meshing frequency and χ_m is the phase.

When one gear of the pair does not have a uniform profile or different spacing between the teeth, resulting in a gear meshing vibration signal with amplitude or phase modulation, and considering that the other gear does not have defects, the modulations will be periodic with the rotation of the defective gear in the rotation frequency - f_r. The amplitude (a_m) and phase (b_m) modulation functions can be represented by finite Fourier series for the rotation frequency. These functions can vary from one gear meshing harmonic for other, and subscript m should be incorporated to equations (2.a) and (2.b) that describe the variations in the vibration of the system.

$$a_m(t) = \sum_{n=0}^{N} A_{mn} \cos(2\pi n f_r t + \alpha_{mn}) \tag{2.a}$$

and

$$b_m(t) = \sum_{n=0}^{N} B_{mn} \cos(2\pi n f_r t + \beta_{mn}) \tag{2.b}$$

Where A_{mn} and B_{mn} are the amplitudes and α_{mn} and β_{mn} are the phases for the modulation functions.

Combining equations (1) with (2.a) and (2.b) results in Equation (3) which describes the gear meshing modulated vibration, $y(t)$.

$$y(t) = \sum_{m=0}^{M} X_m (1 + a_m(t)) \cos(2\pi m f_e t + \chi_m + b_m(t)) \tag{3}$$

2.1. Expanded Gear Meshing Model

For better understanding and control of the defects that one wants to observe, the model presented in Equation (3) can be divided according to equations (4) (Randall, 1982)

a) Portion relative to the rotation harmonics of axis - $x_g(t)$:

$$x_g(t) = X_a sen(2\pi f_r t) \tag{4.a}$$

Where X_a is the amplitude and f_r the axis rotation frequency.

b) Portion due to gear meshing - $x_e(t)$:

$$x_e(t) = X_b sen(2\pi f_e t) \tag{4.b}$$

Where f_e is the gear meshing frequency, X_b is the amplitude for the considered gear meshing frequency.

c) Portion of the gear meshing harmonics - $x_h(t)$:

$$x_h(t) = \sum_{n=1}^{N} Xb_n sen(2\pi n f_e t + \phi_n) \tag{4.c}$$

Where Xb_n is the gear meshing amplitude and ϕ_n is the phase angle for the harmonic n.

To comprise the set of defective gear meshing signals misalignment and eccentricity defects were considered. In gearboxes with misalignment, the frequency rotation harmonics and mainly the gear meshing frequency of the gears fixed in a shaft are observed. Another outstanding characteristic is that the second and third harmonic feature larger amplitudes than that of fundamental frequency. The critical limit is reached when the second or third harmonic amplitudes of the rotation frequency or of the gear meshing frequency surpass by 50% the fundamental frequency amplitude (Arato Jr. and Silva, 2001). Equation (4.d) represents this defect.

$$x_d(t) = \sum_{n=0}^{N} Xg_n sen(2\pi n f_r t + \phi_n) + \sum_{k=o}^{K} Xb2_k sen(2\pi k f_e t + \phi_k) \tag{4.d}$$

Where: Xg_n is the amplitude for the rotation frequency harmonic; N is the number of rotation frequency harmonics; ϕ is the phase angle; $Xb2_k$ is the amplitude of gear meshing frequency harmonic and K is the number of gear meshing frequency harmonics.

The eccentricity, caused by inadequate assembly, manufacturing errors or bent shaft, is characterized by presenting an elevation in the gear meshing frequency amplitudes and the appearance of lateral bands. This defect reaches its maximum limit when the gear meshing frequency amplitude is higher than the triple of the normal operation amplitude or when the amplitude of any lateral band surmounts by 70% the fundamental gear meshing frequency amplitude. Equation (4.e) represents the model for this defect.

$$x_f(t) = \sum_{n=0}^{N} Xb_n \left[1 + \sum_{p=0}^{P} Xa_p sen(2\pi f_r t) \right] . sen(2\pi n f_e t + \phi_n) \tag{4.e}$$

Where: P is the number of lateral bands; N is the number of gear meshing harmonics and Xa_p is the amplitude of the lateral band.

Therefore the meshing signal can, in a general way, be modeled by the sum of all these contributions.

$$x(t) = x_g(t) + x_e(t) + x_h(t) + x_d(t) + x_f(t) \tag{5}$$

The signals were computationally generated with the same transmission relationships presented in a real gearbox, mainly those close to the unit. The effects of the eigenvector and eigenvalue variations in the frequency response functions of the gearboxes were taken into account by the introduction of an amplitude factor, *fa*, applied to the meshing frequencies amplitudes, ranging from 2% to the limit of 20%. It was considered, as in the real case, that all of the gears are rotating simultaneously with the gear which is conveying power.

In all simulated cases, faulty vibration signals with misalignment and eccentricity were considered, which implies in amplitude and phase modulation, respectively.

3. Sensitivity analysis

In the simulation of signals for gear meshing with misalignment oscillations in the meshing frequencies amplitudes of up to 20% were considered, hence the second and third harmonic amplitudes of the rotation frequency or of meshing frequency feature increments from 20% to 80% in relation to the fundamental frequency amplitude. For the same reason, the fundamental frequency amplitude for the eccentricity defect is multiplied by a factor that varies from 2.4 to

3.6 and the lateral band amplitudes will be 56% to 84% larger than the amplitudes of the respective meshing frequencies.

For the third speed, signals without defect were generated with amplitude variation of up to 20% for the meshing frequency and their respective harmonics. Considering a unit amplitude, the amplitudes of the good signals group varied from 0.8 to 1.2. Assuming that real perfect meshing systems do not exist, the lateral bands amplitudes of the good signals were adopted as 10% of the respective gear meshing frequency or harmonic.

The defective signals were generated with amplitude factors equal to 1.4; 1.8; 2.2 and 2.6 and for each factor were as lateral band factors the values: 0.2; 0.4; 0.6 and 0.8.

All generated signals, both those without defect and the defective ones, were contaminated with four levels of addictive random noise. The first level corresponds to a noise factor - *fr*, of 0.2, that represents approximately 0.5% of the root mean square signal level. The second one corresponds to a noise factor of 0.4, which represents approximately 2.0% of the energy level existing in the signal. The two remaining levels correspond to noise factors 0.8 and 1.2, which represent around 5.0% and 15.0%, respectively, of the energy level present in the signal without noise. For the group without defect signals, in addition to these considered noise factors, a noise factor of 0.1 was used, representing 0.1% of the energy level existing in the signal without noise.

After the application of the Choy-Williams transform to the signals, vectors were built with the matrix lines associated to the concerned frequencies. These vectors represent the temporary signal behavior for the fundamental meshing frequency and its first harmonic, as well as their respective lateral band frequencies. Finally, the root mean square which is associated to the energy of the signal was calculated for each one of the vectors.

To the energy of the good and faulty signals, the averages and the standard deviations were estimated for each one of the considered frequencies. The energy values were expressed with respect to the standard deviations calculated for the good signals. The classification of the signals is made according to the comparison between the estimated values for the faulty signals, having as acceptable limit one-fold the standard deviation obtained for the good signals. This analysis was made with respect to the variation in: amplitude, lateral band and noise factors. The averages and standard deviations of the good signals are shown in Table 1.

Table 1 - Mean (μ) and standard deviation (σ) of the energy for the third speed.

Indicator	1st Mesh Frequency		2nd Mesh Frequency		Lateral Band (1st Mesh Frequency)		Lateral Band (2nd Mesh Frequency)	
	μ (mm^2)	σ (mm^2)	μ (mm^2)	σ (mm^2)	μ (mm^2)	σ (mm^2)	μ (mm^2)	σ (mm^2)
Energy (mm^4)	340.65	24.38	171.26	10.46	75.05	6.46	29.43	2.49

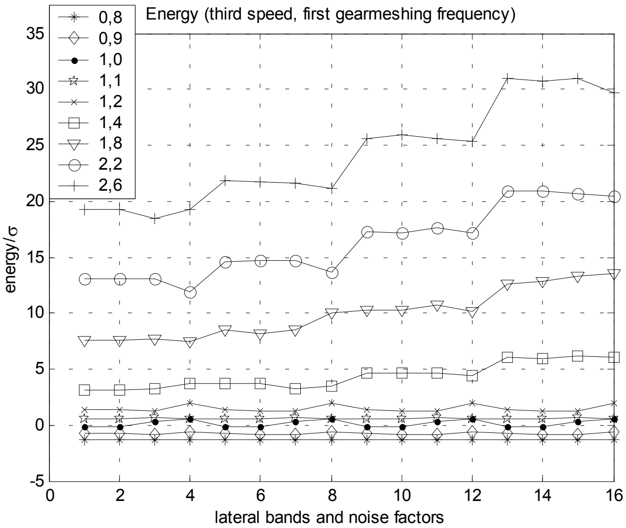
In this sensitivity analysis, the behavior of the energy in the simulated signals was evaluated simultaneously with respect to the lateral bands and noise factors for the first and second meshing frequency and their respective lateral bands.

3.1. Analysis for the simulated signals of the third speed.

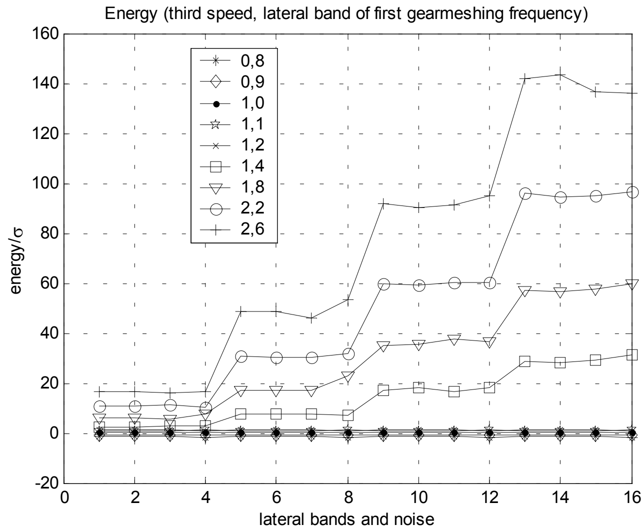
In Figures (1.a), (1.b), (1.c) and (1.d), energy values for the temporary portions of the signals associated to the first and second meshing frequencies and their respective lateral bands are presented in function of the respective standard deviation. The first four positions of the horizontal axis represent the points obtained for the lateral band factor equaling 0.2. The points of fifth to the eighth position correspond to the lateral band factor 0.4. The energy for the lateral band factor equal to 0.6 is associated to the ninth at twelfth abscissas. Finally, the last four positions show the energy values for the lateral band factor of 0.8. In these Figures, for all lateral band factors, the four positions are associated to the noise factors 0.2 to 1.2.

In Figures (1.a) and (1.b) it is observed for the first and second meshing frequency that the energy levels increase with the amplitude factor. Considering each amplitude factor, one can notice an energy increment due to the increase in the lateral band factor. Another important characteristic is the insensitivity to noise of the energy values, as can be seen in a same lateral band. For the signals without defect featuring an amplitude factor equal to 1.2, all the energy values are bigger than 1σ.

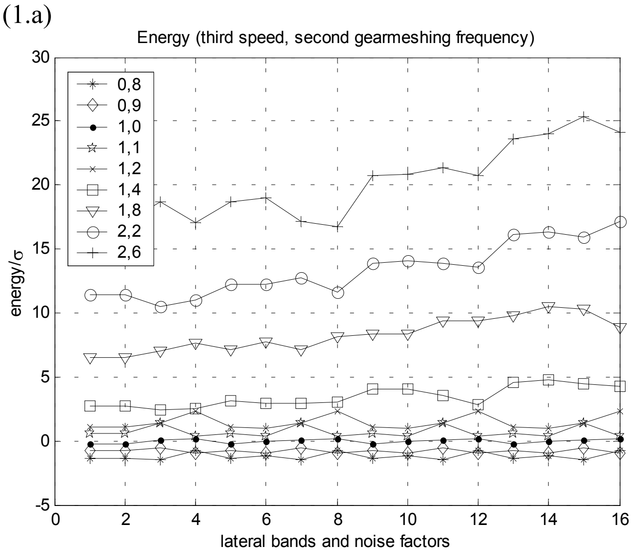
The graphs corresponding to the lateral band frequencies presented in Figures (1.c) and (1.d), show the same behavior observed for the meshing frequencies. However, the sensitivity to the variations in amplitude and lateral band factors for the lateral bands frequencies is superior.



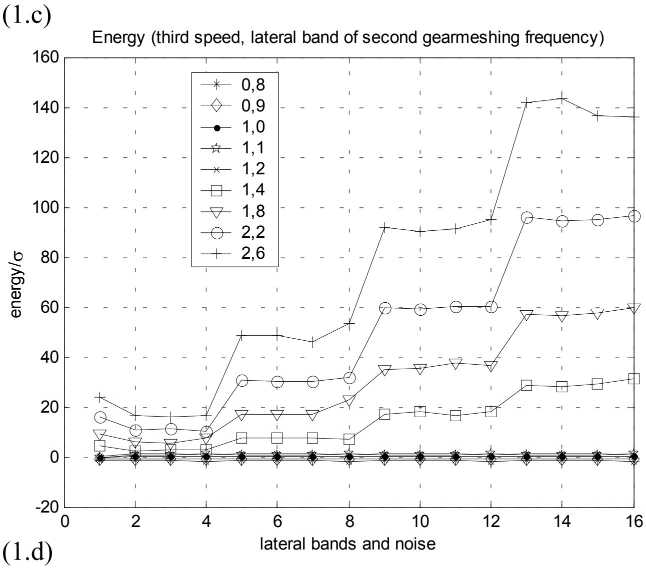
(1.a)



(1.c)



(1.b)



(1.d)

Figure 1. Energy with respect to the meshing frequencies and lateral bands of the third speed.

In general, the estimator based on the use of the time energy portions associated to the meshing frequencies and lateral bands of the Choi-Williams transform presented high consistence and reliability for defects classification when applied to the simulated gearbox vibration signals.

4. Analysis of the measured data for the gearbox 14-58.

In this stage the results obtained in the classification of the vibration signals for the third and fifth speeds and differential for the gearbox model 14-58 are analyzed. The teeth number for each pair is shown in Table (3).

Table 3. Number of teeth for the gears - Gearbox model 14-58.

Speed	Third	Fifth	Differential
Primary axis	25	37	15
Secondary axis	38	31	61

4.1. Description of the Applied Strategy.

Vibration signals of a faulty gearbox group and of a non-faulty gearbox, which was considered as standard, were measured. The faulty state definition for the gearboxes were obtained using the analysis system developed by the Mechanical Engineering Department of the Federal University of Uberlândia, in operation in the FIAT gearboxes test room, located in Betim, MG.

The defective gearboxes were labeled according to their main faulty cause. When a same defect in more than one gearbox occurs, sequential numbering is added to the label. Seven samples comprise the defective gearbox group: faulty, faulty I, faulty II, differential beat, differential beat I, differential beat II and differential beat III.

For each speed the acceleration vibration signals were recorded. The sampling rate is 16384 Hz, resulting in a Nyquist frequency of 8192 Hz. For each speed, the test comprises three stages: acceleration up to approximately 3600 RPM; maintenance of this rotation for approximately four seconds, depending of the pair tested; followed by deceleration until stoppage.

In the analyses of each pair the test segment performed with constant rotation was used. The signals were pre-processed by time domain average, with the period relative to the rotation frequency of the primary axis as well as the period associated to the rotation frequency of the secondary axis being used for each geared pair, so that the harmonics associated to the differential could be highlighted. This reduces the noise influence in the signal, improving the signal to noise ratio. As the defects targeted for this evaluation are not impulsive ones, such as fatigue cracks in teeth, but assembly or manufacturing defects expressed basically as phase or amplitude modulation, which are distributed and occurring throughout the signal, the average procedure does not harm the analysis.

To evaluate the state of the defective gearboxes, the energy contained in the vibration signal to the first and second meshing frequency and their respective lateral bands was estimated. This was done after the application of the Choy-Williams transform to the vibration signal followed by the construction of a vector with the time portion associated to these frequencies.

For the fifth speed, the inexistence of values for the second meshing frequency and its lateral bands is due to these frequencies being higher than the maximum frequency obtained for the Choy-Williams transform of these signals. The maximum frequency for this time-frequency transform is defined as the half of the Nyquist frequency, in this case being 4096 Hz (Flandrin and Martin, 1985).

4.2. Faulty Index (FI)

To simultaneously measure the information contained in the meshing frequencies and in the lateral bands energy a faulty index (*FI*) was proposed. This index is a weighed average of the energies calculated for the lateral bands and meshing frequencies. For the gear meshing frequencies a larger weight was attributed, as its energy content represents a significant portion of the global gear meshing signal energy.

$$FI = (4E_{gear\ meshing\ frequency} + 6E_{lateral\ band})/10 \tag{6}$$

4.3. Evaluation of the differential.

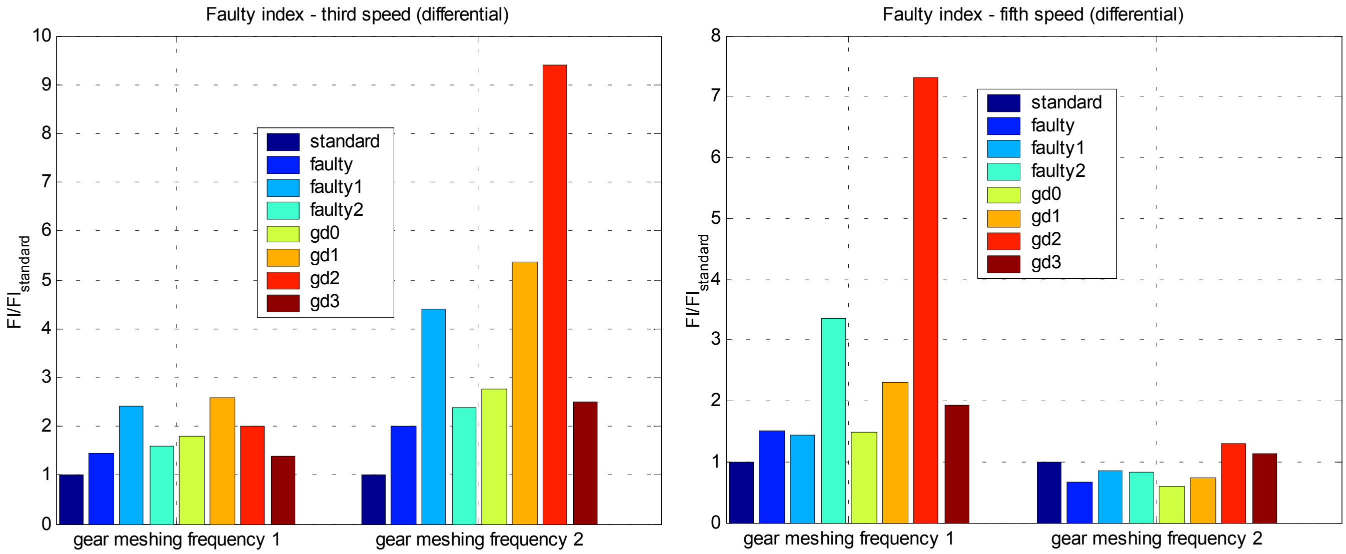
Figure (2) shows the faulty index graphs for the differential with respect to each speed. The faulty indexes, relative to the third and fifth speeds are respectively shown in Figures (2.a) and (2.b). The non-dimensional values were obtained having as reference the *FI* for each meshing frequency of the standard gearbox ($FI_{standard}$)

When the second speed is geared, the faulty, faulty I and II gearboxes and differential beat I, II and III present higher *FI* than that calculated for the standard gearbox in all of the gear meshing frequencies, indicating a high probability of manufacturing or assembly defect in the differential.

For the gearbox labeled "gd0" - differential beat, an *FI* higher than the standard gearbox is observed just for the second gear meshing frequency, implying a low defect probability.

As can be seen in Figure (2.a), all the faulty indexes for the third pair are higher than those obtained for the standard gearbox, regardless of the gear meshing frequency considered. In this case, all of the gearboxes have a great probability of presenting manufacturing or assembly defect.

Figure (2.b) shows faulty indexes related to the differential while the fifth speed was engaged. Only for the gearboxes denominated "differential beat II and III", did the faulty indexes calculated for the second gear meshing frequency present higher values than those obtained for the standard gearbox for this same frequency. For the first gear meshing frequency, all of the gearboxes feature higher *FI* than that obtained in the standard gearbox for the same frequency. This points to a high probability of defects associated to the differential.



(2.a) (2.b)

Figure 2 – Faulty Indexes for the differential with respect to the speed.

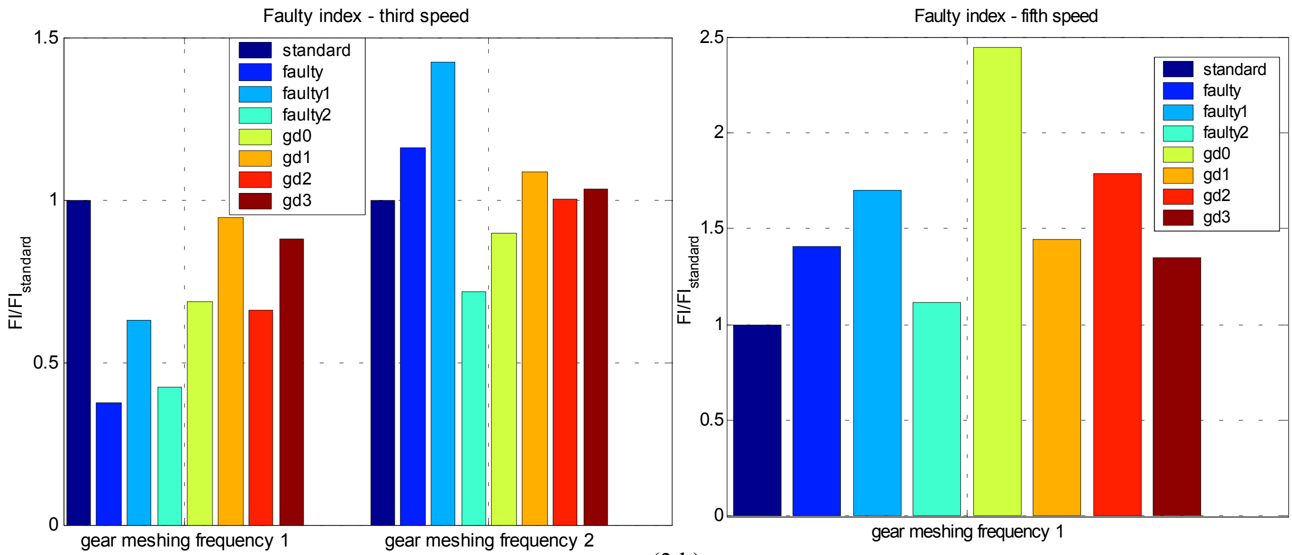
It is observed that all the faulty indexes calculated for the differential vibration signals of the analyzed gearboxes are above the estimated value for the good gearbox, at least while the two speeds are engaged.

4.4. Evaluation of the speeds

In Figure (3) the graphs of the faulty indexes for each one of the speeds are shown. The faulty indexes for the third and fifth speeds are presented in Figures (3.a) and (3.b), respectively. Again the non-dimensional values were obtained having as reference the FI of each standard gearbox gear meshing frequency.

As can be seen in Figure (3.a), all the faulty indexes calculated for the first meshing frequency of the third speed are lower than those obtained for the standard gearbox. The faulty indexes for the second gear meshing frequency are lower only for gearboxes "faulty II" and "differential beat".

In Figure (3.b) the faulty indexes associated to the fifth speed's meshing frequencies are shown. For the first gear meshing frequency, all the faulty indexes presented values higher than those obtained for the standard gearbox at this same frequency, which implies in a high probability of defects associated to this pair.



(3.a) (3.b)

Figure 3 – Faulty Indexes for the differential for the third and fifth speed.

5. Conclusions

In the differential evaluations, all gearboxes presented *FI* above the one calculated for the standard gearbox when the third speed is engaged with no occurrence being observed for the first meshing frequency. For the fifth speed, only the faulty indexes associated to the first gear meshing frequency were higher than those estimated for the standard gearbox. Only gearboxes "gd2" and "gd3" presented faulty indexes higher than those obtained for the standard gearbox at the second gear meshing frequency.

In the overall evaluation of the speeds, the third one featured a smaller number of occurrences, and none for the first gear meshing frequency. The fifth speed was the one featuring more occurrences.

Of the evaluated transmission boxes, the one with lower occurrences was that labeled as "faulty II", with incidences for the first gear meshing frequency of the fifth speed.

The results achieved with the use of this methodology were very coherent with the previously known state of each gearbox. The adoption of the faulty index, which weighs with higher incidence the energy related to the gear meshing frequency, was quite effective and the overall definition of the state of the gearbox was made easier

6. Acknowledgement

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