American Journal of Technology Advancement 2997-9382, Vol 03, No. 08, 2026

Article

Vibration Behavior Analysis of an Automotive Differential in the Time and Frequency Domains Under Various Crown Wheel and Pinion Fault Conditions

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Abstract: Differentials are vital and delicate pieces of equipment in the automotive power-transmission group, and unnoticed damage or malfunction in the crown wheel and pinion will lead to expensive downtime and even, in the worst possible scenario, dangerous failures. Vibration analysis is a widely accepted and non-destructive method of identifying such faults in autos. The paper describes an experimental study on the vibration behavior of an auto differential in five conditions healthy, with a damaged pinion tooth, with a fractured pinion tooth, damaged crown wheel tooth, and broken crown wheel tooth in two states of operation at the input speeds of the pinion of 2000 and 3000 rpm. A special stand was created and included the differential itself, an electric engine with belt-and-pulley drive, a speed regulator, vibration insulators, and a piezoelectric accelerometer connected with the data-acquisition system.The theoretical gear-meshing frequencies (GMF) of the pinion's crown wheel and of their harmony were gained through the use of gear geometry and revolution rate values compared to the experimental results achieved through measurements. The outcomes show that the vibration intensity follows a linear relationship with the increasing intensity of the fault (healthy < worn < broken) in both gears, that the vibration caused by a broken tooth happens in the form of sudden pulses and can be measured in figures, that additional harmonics of GMF are introduced and increase with the accumulation of defect severity in the crown wheel, and that the separation of the symptoms of defects from the background level increases due to the increase of rotational speed of 2000–3000 rpm. This observation makes it possible to better understand the behavior of a differential's vibration signal, and thus create the basis for a future automated fault classification.

Keywords: Automotive Differential; Bevel Gears; Condition Monitoring; Vibration Analysis; Fast Fourier Transform; Gear-Meshing Frequency

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1. Introduction

A gear is responsible for the transmission of rotational movement and torque from one shaft to another without slippage through the interaction of its teeth, making gears one of the most common options for mechanical power transfer. In the case of rear-wheel or front-wheel drive cars, the differential, including the crown and pinion, the gearbox, and prop shafts, provides the last step

in the speed reduction and torque conversion process while enabling different speeds of wheels when moving in corners. As the differential is the last stage in the drive line and has to withstand heavy loads, the damage of its gears often goes unnoticed and leads to serious issues at the subsequent stages as well as vehicle safety risks [1], [2].

Conventional inspection techniques consist of disassembly and visual inspection, or the use of operators' senses (hearing, touch), which has resulted in high costs and time consumed and reliance on the knowledge and experience of the inspectors [3], [4], [5]. In contrast, condition monitoring in modern maintenance focuses on continuous observation of the condition of machines expressed in measurable parameters, usually vibration, with the help of the deviation of the same from normal illustrating an emerging defect [6], [7], [8]. Concerning the task above-mentioned, vibration has numerous advantages: it is repeatable, safety-affecting factors have little impact on it, and it is measured non-intrusively [9], [10].

The findings from research conducted on automotive differential spiral-bevel gear sets backs the claim that flaws in crown-wheels and pinions lead to substantial vibration behavior changes, which vary in strength depending on the way the defect manifests, how severe it is, along with the speed of operation, load, and signal analysis approach carried out experiments, proving that old and artificial crown-wheel-and-pinion pairs have higher vibration and noise than new pairs, while the damage of a pinion tooth led to noticeable changes in both the time-domain and frequency-domain responses. In destructive testing of fatigue, it was shown that pitting developed after about 3×10^6 revolutions of the pinion although the tooth failure occurred after approximately 7.2×10^6 cycles of the pinions and noise and vibration increased sharply just before failure [11], [12].

A numerical study carried out on an SUV spiral-bevel pair having an artificially damaged pinion tooth has also indicated the presence of higher harmonic stress and vibration levels for the damaged pair, thereby further validating the hypothesis of using frequency domain as a viable means of identifying states [13]. In addition to the experimental and simulation results, it can be observed from the analysis of dynamic models of the differential gears that the presence of any local gear fault modifies the time-varying mesh stiffness, thereby producing vibration impulses and that the geometry profile defects or the shaft eccentricity also has an important role in raising the vibration levels [14].

Operating condition itself represents a significant element of fault physics and should not be regarded as mere inconvenience. Surányi et [15] proved experimentally that traditional acceleration-based monitoring of the differential failed to show the onset of a defect because of low speed high load conditions and proposed the Locking-Factor spectrum indicator, which managed to successfully monitor crack initiation and growth for a period of 14 to 81 hours of physical testing of the differential operational conditions, which were not expected to permit the amplitude-based approach applied in most of the lab studies, including this one, to work properly. Gharehbolagh et al. [16] dealt with acoustic rather than vibration signals and identified conventional, broken and worn states of the crown wheel and pinion of the differential using discrete wavelet features and neural networks, achieving 90.6% classification accuracy confirming that time-frequency features of the signal may provide useful information for diagnosing the component.

A great number of close studies have been conducted that applied vibration analysis and acoustics to gearboxes as a whole. Saravanan et al. [17] studied vibration spectra of a gearbox in good condition and in the states with failed teeth, fractured roots, and worn-out surfaces. Liu et al.[18] differentiated tooth failure and wear severity in a gearbox using tri-axial accelerometer signals and discrete wavelet coefficients. Li et al. [19] connected vibration and acoustic signals in their study and established 11 classes of gear failure. Zhixiong et al. [20] detected cracks in small gears of wind-turbine gearboxes using multi-sensor vibration signals and wavelet transformation. In turn, Odgaard and Stoustrup [21] created specific filters which helped to detect failure frequencies of wind-turbine gearbox. Jedliński and Jonak [22] created a study on shock absorber wear applying continuous wavelet transformation, and Ramadevi [23] used vibration signals to detect cavitation of

a centrifugal pump utilizing the method of discrete wavelet features. All the studies reviewed in these works share the same conclusion that the development of gear failure leads to the change of vibration signal within the frequency range which can be predicted from gearbox design and velocity. It is important to note the gear-meshing frequency is crucial for investigation.

Time-frequency techniques and adaptive approaches have emerged in response to limitations in standard FFT analysis of stationary conditions due to variations in speed and load. For example, gear-mesh frequency amplitude changes with both speed and load, sideband power changes primarily with speed and various effects of the transmission route as well as time dependency blur the resulting spectra [24], [25]. Among the notable time-frequency techniques, one can mention, for instance, the generalized synchrosqueezing transform [26], frequency-domain averaging without a tachometer [27]and the variational mode decomposition method [28]to extract weak transients and dynamic gear-rotor-bearing models [29] with localized defects. These techniques are reported to allow for capturing features that can be missed by standard FFT once the speed changes.

Consequently, thus far it seems that while the relation of the crown-wheel/pinion problems and the changed vibration behavior is known and backed by both differentiated experiments and general gearbox literature, the data concerning the systematic and proper labelling of the vibration signature of the differential is still scarce as the associated differential studies are mainly of either analytical/ numerical nature, aimed at a special low-speed regime with a unique parameter or based on acoustics rather than vibrations.

This study aims to fill this gap by experimentally analyzing the vibration of an automotive differential in five conditions: undamaged, worn pinion, damaged pinion, worn crown wheel, and damaged crown wheel, through two constants of pinion rotation speed, 2000 rpm and 3000 rpm, in order to make the analysis based on FFT valid staying close to steady-state behaviors. The main objectives are to: (i) characterize the relation between the types of faults and their severity on the amplitude values of vibrations observed in time domain; (ii) determine how the given fault influences the frequency spectrum and the specific gear-meshing frequency and its harmonics; and (iii) investigate how the rotation speed influences the distinction of the fault types. The purpose of this analysis is to create a basis of interpretation for the automated fault diagnostics of automotive differentials and become a basis for future time-frequency studies.

2. Materials and Methods

2.1 Test Rig and Instrumentation

In order to obtain vibration data for an automobile differential, a test rig was specially designed and fabricated. The test rig as shown in fig.1 attached to the differential consists of electric driving motor, belt and pulley arrangement, vibration-proof mounts, speed-controlling for electric motor, tachometer, and vibration sensor that uses data acquisition system. See Fig.2.

The object of testing was the final drive (differential) unit of the Saipa car with model RA28 characteristics of which are presented in Table 1. The pinion gear is a 9-toothed one and the crown gear has 35 teeth.

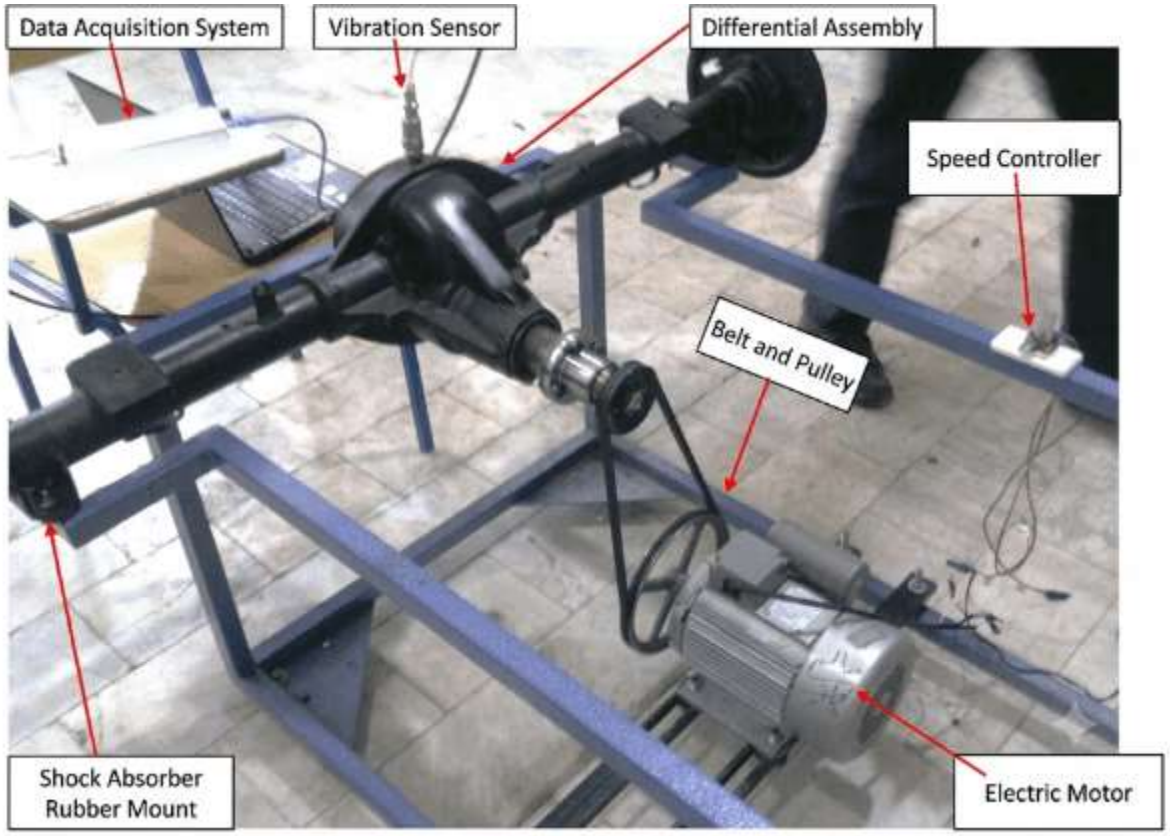


Figure 1. Test rig used for vibration data acquisition.

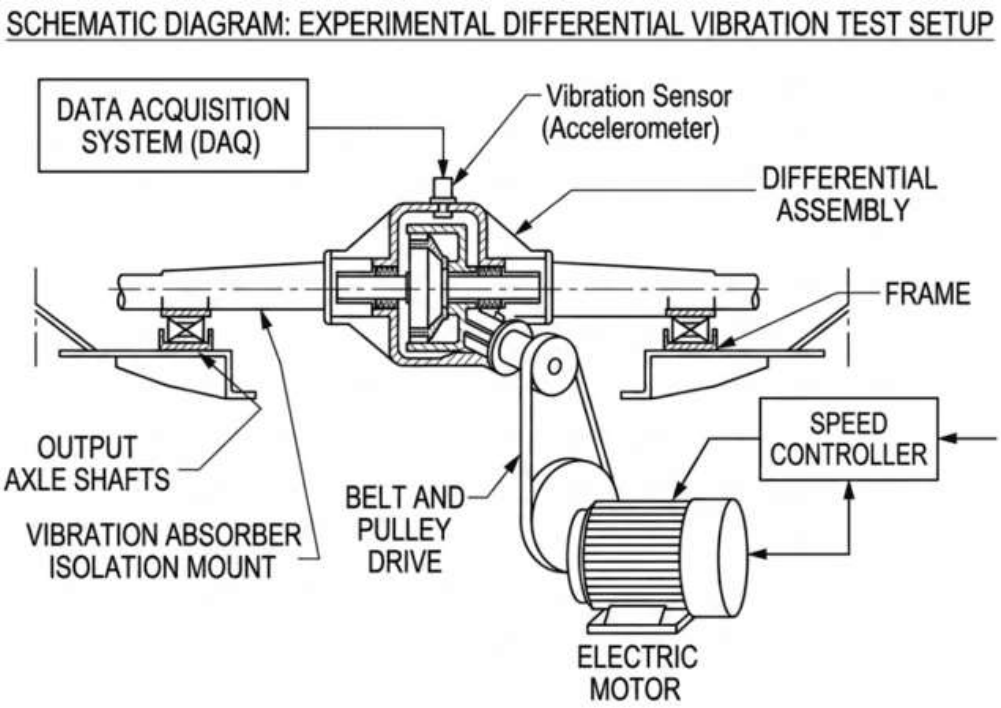


Figure 2. A schematic diagram of the experimental setup.

Table 1. Specifications of the test-vehicle final-drive assembly

Specification	Value
Model	RA28
Differential type	Rear differential
Number of pinion teeth	9
Number of crown-wheel teeth	35
Rated torque capacity	158 N·m at 3000 rpm

The differential was operated by an electric motor (Table 2) using a belt-and-pulley system (Fig. 3): since the motor speed (1440 rpm) is lower than the desired speed of pinion, the pulley diameter ratio of 3:1 is used to increase the speed of the pinion shaft. The speed control circuit (Fig. 4) was used to control the speed of the electric motor so that it can be adjusted from 0 to 1440 rpm by changing the frequency of the supply current. The pinion speed of 2000 rpm and 3000 rpm were set using an ARMA ETI-TACHO digital optical tachometer as shown in Fig.5.

Table 2. Characteristics of the drive electric motor.

Parameter	Value
Type	YC90L1-4
Power	0.746 kW (1 Hp)
Supply frequency	50 Hz
Voltage	220 V
Current	1.2 A
Rated speed	1440 rpm

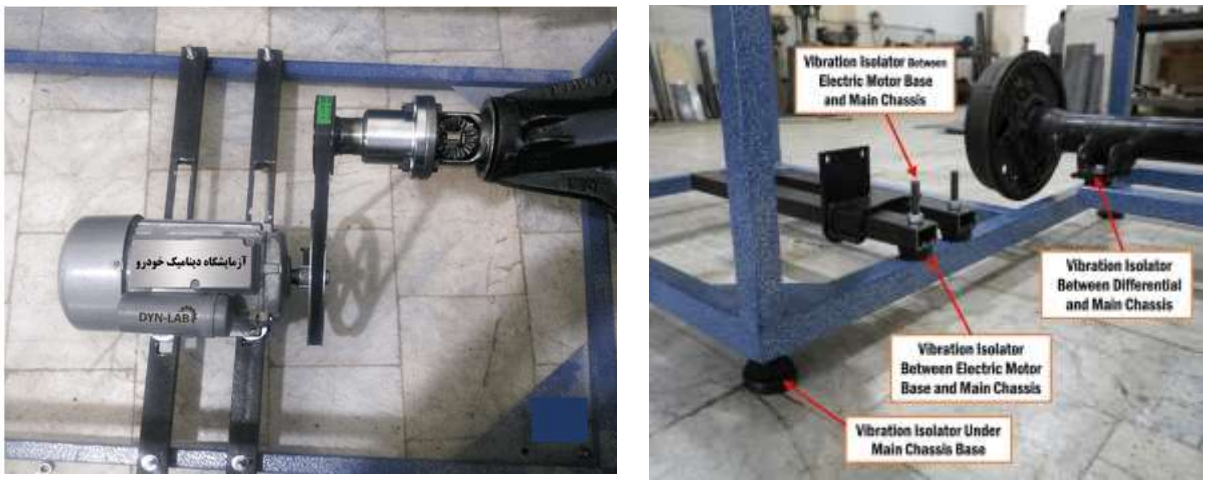


Figure 3. Transmission of electric-motor power to the differential via a belt and pulley.

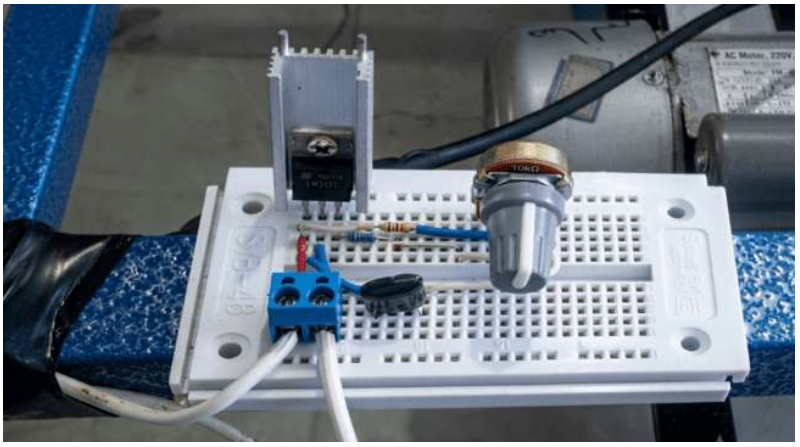


Figure 4. Speed-control circuit board used to set the electric-motor speed



Figure 5. Speed gage.

Vibration isolating mounts or shock absorbing mounts were installed underneath the mainframe of the rig, in the place of the motor, between the motor and frame of the rig and also next to differential in between differential unit and chassis. These vibration isolating mounts were used to limit the effect of any external vibration caused due to rig frame imperfections, working of the motor, or belt and pulley drive on the vibration signal of interest. The data was collected using a VMI-102 piezoelectric accelerometer which was placed vertically on the differential housing (Fig 6), which then was amplified by APC-40 spectrum analyzer data acquisition system and then sent to Pulse software which was responsible for data export of the collected time series data to be processed in MATLAB.



Figure 6. Data-acquisition system, vibration sensor, and sensor mounting position on the differential housing.

2.2 Simulated Fault States

Four fault conditions have been defined in addition to the reference condition. These include pinion tooth wear, pinion tooth failure, crown wheel tooth wear, and crown wheel tooth failure. The wear simulation was achieved by removing precisely 1 mm of material from the surface of each tooth of the relevant gear and smoothing it out with the help of a grinding machine, thus mimicking the tooth tip rounding feature of natural wear. In the case of failure simulation, a part of a tooth in the meshing area is removed, thus making it not engage any longer with the mating gears. The crown wheel and pinion in pristine, worn, and ruptured tooth conditions are shown in Fig. 7.



Figure 7. Crown wheel (left) and pinion (right), each shown in the healthy, worn-tooth, and broken-tooth condition

2.3 Gear Kinematics and Characteristic Frequencies

The gear ratio for two gears in mesh is given through the following equation.

$$i = \frac{Z_2}{Z_1} \quad (1)$$

Where Z_1 and Z_2 are the number of teeth on the driving gears and driven gears. In the current scenario involving the crown gear/pinion set (Z_1 = 9, Z_2 = 35) the differential generates a constant speed reduction between the crown gear and the input shaft connected to the pinion.

The gear-meshing frequency which is defined as the rate at which the teeth mesh is given by:

$$GMF = T \times S \quad (2)$$

Wherein T is the number of teeth of the gear and S is the revolutions per minute expressed in hertz. As far as the driving and driven gears of a gearpair are concerned, they also operate at same GMF.

The following calculations were done for the two cases of pinion input speeds studied.

At 2000 rpm: f_{shaft} = 33.33 Hz, GMF (pinion) = 300 Hz

At 3000 rpm: f_{shaft} = 50 Hz, GMF (pinion) = 450 Hz

Therefore, it must be stated that the higher harmonics of the GMF are influenced by it, together with the main harmonic, as far as used marker for data interpretation is concerned.

2.4 Signal Acquisition and Processing

Vibration signals in time domain were recorded for five states (healthy, worn pinion, broken pinion, worn crown wheel, broken crown wheel), and two speeds (2000 rpm and 3000 rpm) using an accelerometer and data acquisition chain described in Section 2.1. Time-domain signal was converted into frequency domain using Fast Fourier Transform (FFT) technique, which has been implemented in MATLAB. The obtained spectra was then analyzed state-wise and speed-wise and compared with theoretical values of shaft and meshing frequency obtained in Section 2.3 so that changes in the state with respect to the fault could be understood.

3. Results and Discussion

3.1 Time-Domain Vibration Signals

The figures 8 and 9 depict the time-domain vibration signals obtained for five different conditions at the speeds of 2000 rpm and 3000 rpm. It is seen in both cases that as the severity of the fault increases as compared to the healthy condition, the vibration amplitude also increases, and to be more precise, for a specific gear, the amplitude connected with tooth fracture is bigger than that connected with tooth wear. Likewise, the amplitude change linked to pinion faults is always bigger than that caused by analogous faults of the crown wheel, which can be explained by the fact that from one to nine pinion revolutions happen in the time period of one revolution of the crown wheel (due to the 9:35 tooth ratio), which illustrates that a single defect of the pinion tooth re-engages with the crown wheel much more frequently in comparison to a similar defect of the crown.

A distinctive feature of the vibration generated by a fractured tooth is that it is impulsive and damped instead of continuous: the impulse appears only in the meshing zone of the broken tooth, thus disappearing when a sound tooth occupies its place. This impulsive, self-damping signature provides a new qualitative characterizing feature for distinguishing fracture from wear even before carrying out frequency-domain analysis.

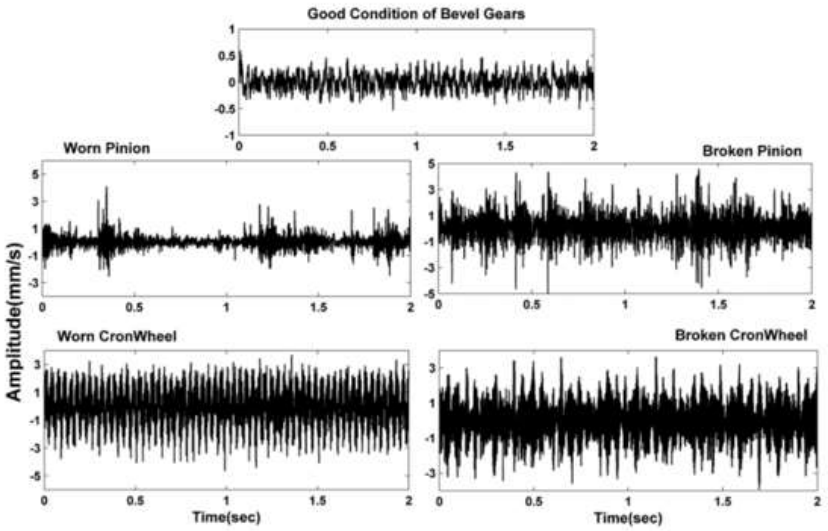


Figure 8. Time-domain vibration signals of the differential at 2000 rpm for the healthy, worn-pinion, broken-pinion, worn-crown-wheel, and broken-crown-wheel states.

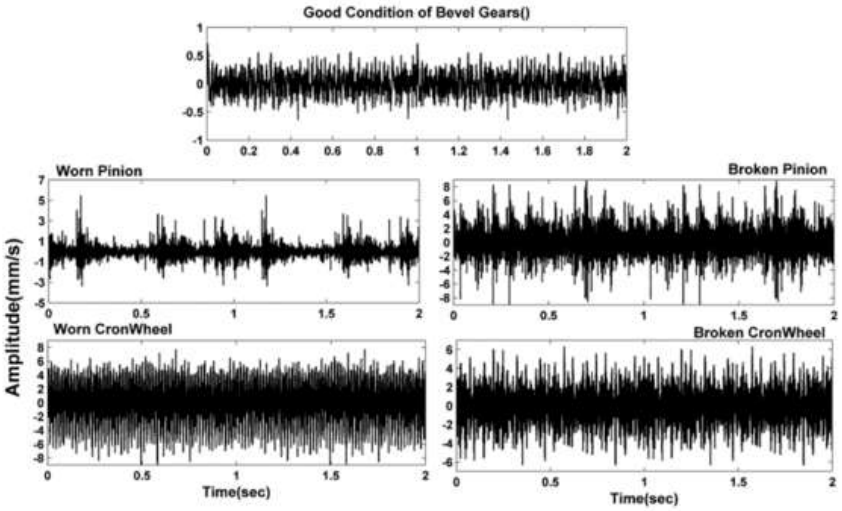
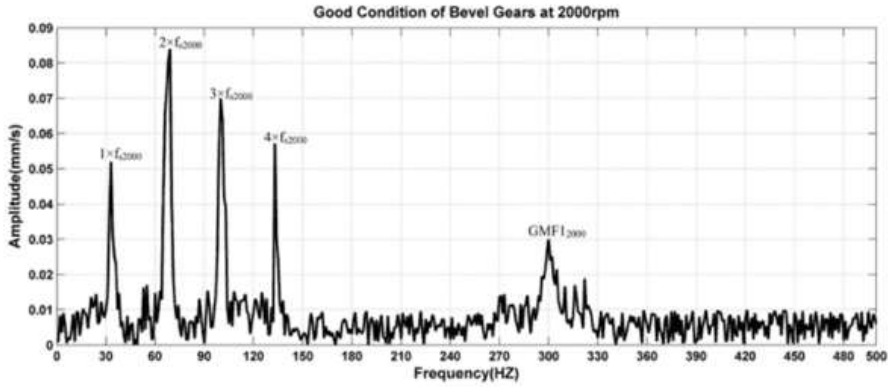


Figure 9. Time-domain vibration signals of the differential at 3000 rpm for the healthy, worn-pinion, broken-pinion, worn-crown-wheel, and broken-crown-wheel states.

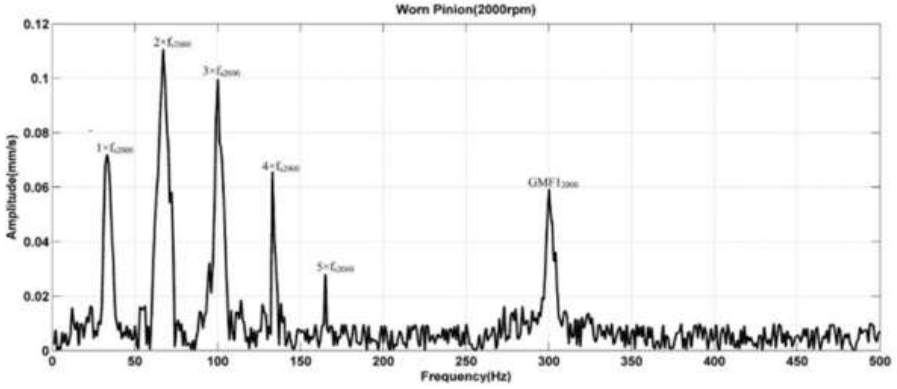
Comparing Figure 8 and Figure 9 illustrates how the change of pinion speed from 2000 rpm to 3000 rpm leads to an increase of the total vibration amplitude in the time domain, although the ordering of the five states would remain the same. Nevertheless, since the time domain presentation consolidates all frequency content into a single amplitude vs. time trace, it can provide only limited information about the meshing behavior.

3.2 Frequency Spectrum at 2000 rpm

In Figure 10, the frequency spectrum for healthy differential at a 2000 rpm shows distinct features, namely, the presence of harmonics 1 to 4, and the pinion gear-meshing frequency (GMF, 300 Hz). In the presence of pinion-tooth wear (Fig.10 a), there is an increase in the amplitudes of harmonics 1 to 4, and the fifth harmonic appears. Most significantly, GMF amplitude increases considerably.



a. healthy state



b. pinion tooth wear

Figure 10. Frequency spectrum of the differential at 2000 rpm: healthy state (a) and pinion tooth wear (b).

The spectrum of a broken pinion tooth (Fig.11) resembles that for worn pinion since the same fundamental harmonics of first to fifth and the pinion GMF peak are observed in both cases. However, the third and fourth harmonics turns out to be much greater in amplitude for the case of broken tooth than for the case of a worn tooth thus providing a frequency domain indicator that can provide a distinction between the two types of severity in theory.

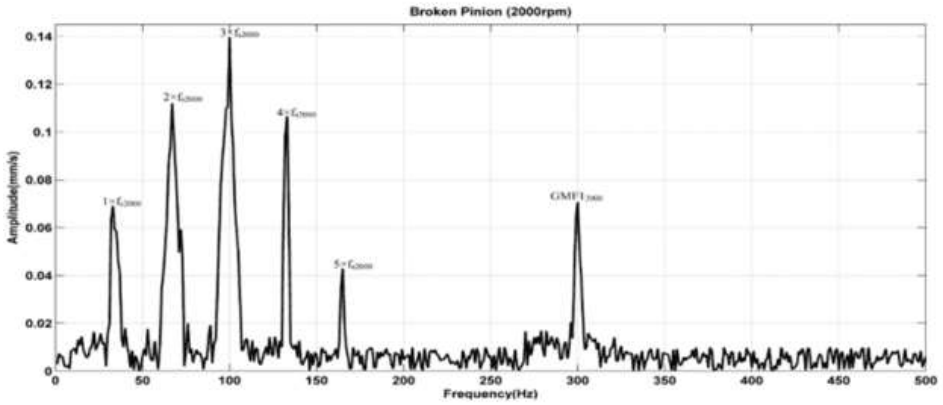
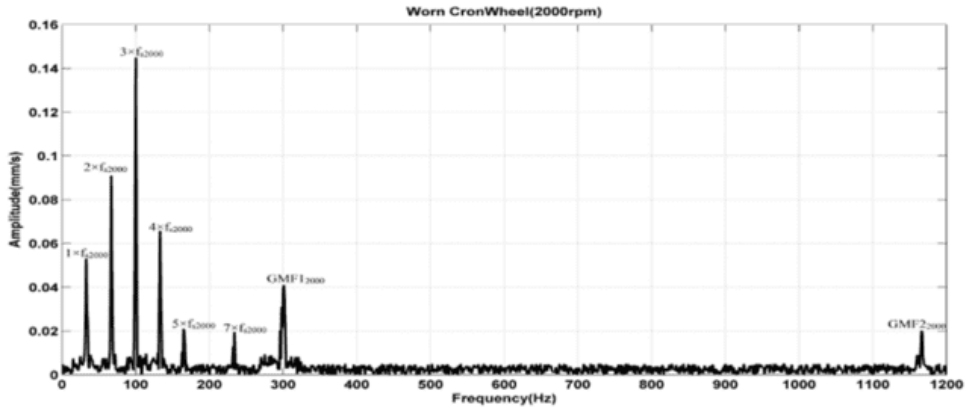
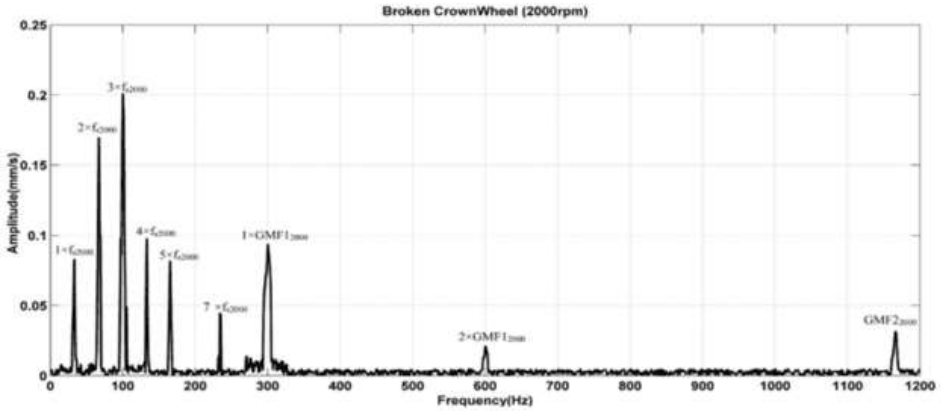


Figure 11. Frequency spectrum of the differential at 2000 rpm with a broken pinion tooth

For crown-wheel tooth wear and fracture as shown in Fig. 12, there is a distinct appearance of the crown-wheel GMF. A comparison of the spectra for wear and fracture indicates that in the fracture condition, the pinion GMF's second harmonic and some extra harmonics of the crown-wheel GMF appear much more distinct. These ratios at the crown wheel mimic the growth of the harmonics observed for the pinion in Fig. 11.



a. crown-wheel tooth wear

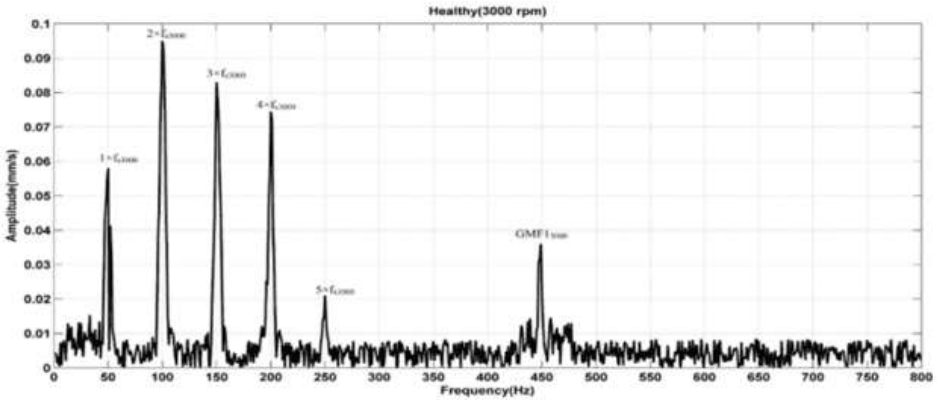


b. crown-wheel tooth fracture

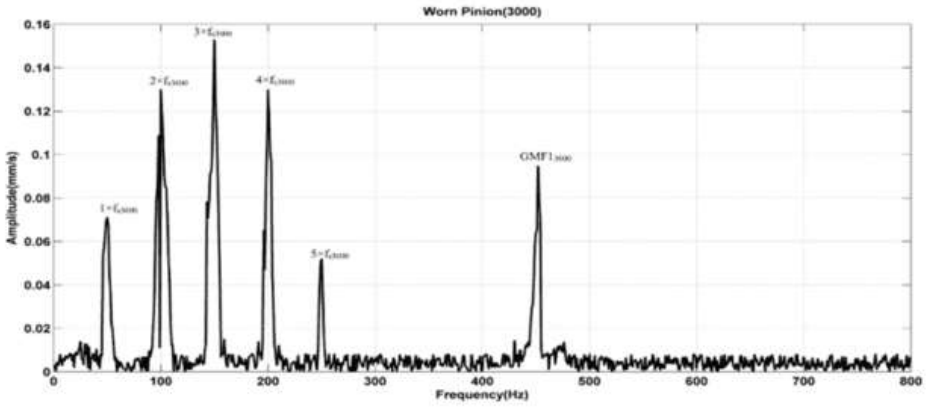
Figure 12. Frequency spectrum of the differential at 2000 rpm with crown-wheel tooth wear (a) and crown-wheel tooth fracture (b).

3.3 Frequency Spectrum at 3000 rpm

At 3000 revolutions per minute, the spectrum in a normal state (Fig. 13, a) demonstrates the fifth harmonic frequency along with the pinion gear mesh frequency (450 Hz) corresponding to a higher shaft speed. Also at 2000 revolutions per minute, abrasion of the pinion has increased the amplitude at certain harmonic frequencies of the vibration without introducing new frequencies but has increased significantly the value of the amplitude precisely at the pinion gear mesh frequency.



a. healthy state



b. pinion tooth wear

Figure 13. Frequency spectrum of the differential at 3000 rpm: healthy state (a) and pinion tooth wear (b).

In the case of a broken tooth of the pinion at a rotation speed of 3000 rpm (see Fig. 14), the amplification of the first and second harmonics of the pinion GMF is high enough to exceed their amplitudes in the wear stage already seen at a speed of 2000 rpm (see Section 3.2), but the changes are much more pronounced at this speed.

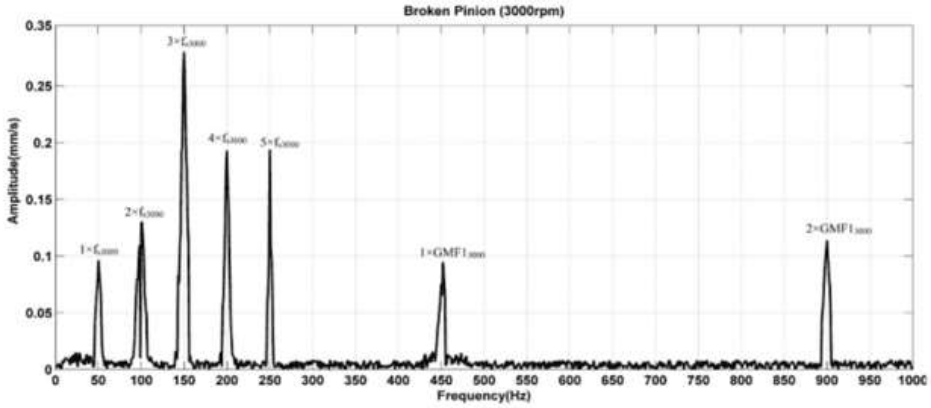
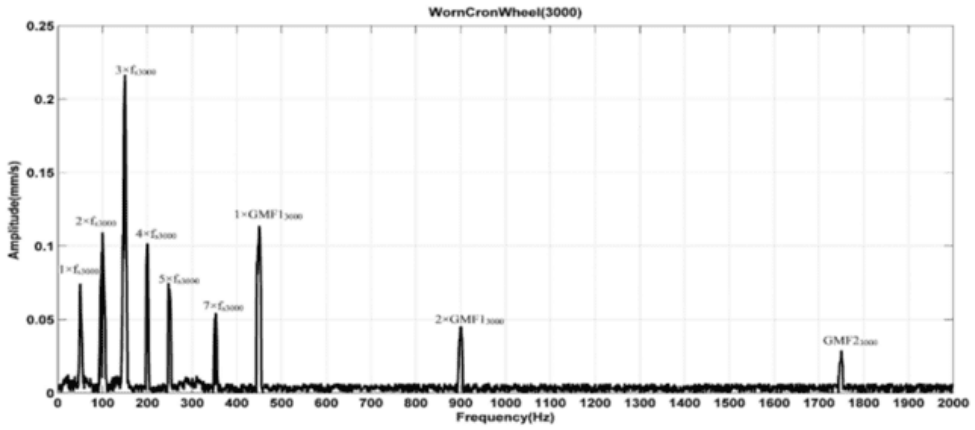
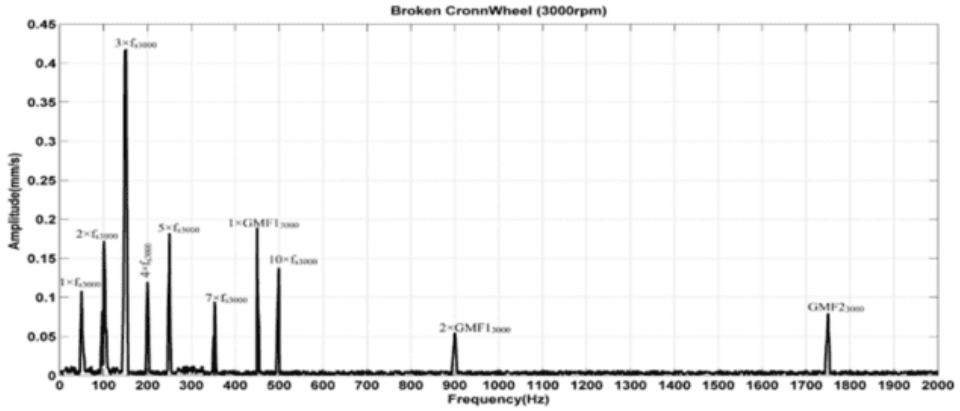


Figure 14. Frequency spectrum of the differential at 3000 rpm with a broken pinion tooth.

One of the most significant speed-related effects observed relates to crown-wheel fault conditions (see Fig. 15). As the damage severity increases, from wearing to fracturing of the crown-wheel teeth, additional harmonics of the GMF of the crown-wheel appear in the spectrum until at the rotating speed of 3000 rpm one can see the harmonics of up to the 10th order in case of broken tooth. This way, the progressive appearance of higher order harmonics with fault intensity characterized by its increasing prominence for higher speed can be considered as one of the most remarkable frequency characteristics of crown-wheel damage found in this work.



a. crown-wheel tooth wear



b. crown-wheel tooth fracture

Figure 15. Frequency spectrum of the differential at 3000 rpm with crown-wheel tooth wear (a) and crown-wheel tooth fracture (b).

3.4 Effect of Rotational Speed on Fault Separability

The increase of pinion speed from 2000 to 3000 revolutions per minute led to increased amplitude of vibration and amount of GMF harmonics associated with the given fault for all the fault conditions. When the pinion speed was increased to higher values, it made it easier to distinguish the faulty cases from the baseline healthy case. This is in accordance with the physical understanding that higher rotational speed leads to increased kinetic energy and therefore more impact energy involved in every meshing process of the damaged tooth. It also corresponds with the previous finding that the amplitude of gear-mesh frequency is affected by speed, and load. From the practical side, it means that if it is possible, taking diagnostic vibration measurements of the shaft during high-speed operation gives better results than at idle or low speed. However, Suranyí et al. note that at the low speed and large load, it can be the opposite when such indicators cease functioning, and additional specialized indicators are needed.

4. Conclusion

The paper contains experimental observations on the behavior of the automotive differential in the time and frequency domains in five different states: a healthy state, a worn pinion state, a broken pinion state, a worn crown wheel state and a broken crown wheel state for pinion operation at the speed of 2000 and 3000 rpm. The conclusions made in the paper are: Vibration amplitude increases in a monotonic way in the time domain with the increase of the defect in comparison with

that of the healthy state for both the pinion and crown wheel. With the same gear, tooth breakage leads to a larger vibration amplitude than tooth wear, and for pinion defects, the amplitude is higher than for a similar defect at the crown wheel due to a higher speed of rotation of the pinion as compared to that of the crown wheel. The vibration caused by tooth breakage is sporadic and has damping features due to intermittent contact of the broken tooth in the contact area. In the frequency domain, the presence of a fault is indicated by a rise in amplitude at the harmonics of the shaft's rotational frequency and, most prominently, at the gear-meshing frequency of the faulty gear. As the severity of fault increases from wear to fracture, additional harmonics of the gear-meshing frequency appear in the spectrum. Rising the operating speed from 2000 to 3000 rpm increases vibration amplitude and harmonic content, making the different states of failure easier to differentiate from the healthy level of vibration. The obtained results constitute the physical basis for the characterization of the occurrence of crown wheel and pinion faults in the vibration pattern of the automotive differential, where the gear-meshing frequency is one of the most important indicators in the diagnosis of this failure.

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