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Advanced Fault Detection in Spur Gear Transmission Systems Using Vibration Analysis and Definite Integral-Based Damage Assessment

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Keywords： Fault detection, Frequency response function (FRF), Fast-Fourier transformation (FFT), Tooth-cut gear, Crack detection, Vibration.

ABSTRACT

This study focuses on the vibrational characteristics and fault detection in spur gear transmission systems, introducing a novel mathematical approach for damage identification. The method, based on definite integrals, calculates the area under the curve (AUC) of vibration signals for accurate detection gear damage. By comparing AUCs for normal, 2mm tooth-cut, fully tooth-cut gears, and cracked gears, the technique consistently identifies structural damage. Finite element analysis and acoustic isolation were employed to mitigate external disturbances and study system behavior. Fast-Fourier Transformation (FFT) and Frequency Response Function (FRF) techniques were also used to capture resonance frequencies. The findings confirm the method's reliability in detecting gear faults, offering a robust solution for optimizing rotating machinery performance.

INTRODUCTION

Rotary machines are essential components of various mechanical systems, including industrial gearboxes, aircraft engines, and power plants. Their dynamic behavior, particularly the vibration characteristics of rotating shafts, plays a critical role in ensuring the functionality, stability, and reliability of these systems. Many rotating components in dynamic systems produce transverse vibrations due to fatigue, leading to potential damage and catastrophic failure of machinery. Common defects such as rotor hub misalignment, shaft cracks, coupling faults, and gear issues like wear, cracking, pitting and scuffing contribute, to complex transient vibration signals during operation (Zhen et al., 2017; Chaari et al., 2006; Vasilev et al., 2019). Gears, frequently mounted on rotating shafts, introduce additional complexity to the system's dynamic response. Gear-related phenomena such as tooth meshing, gearbox resonance, and shaft rotation produce impulsive, high-frequency, and random vibrational signals, complicating the detection of faults (Randall & Antoni, 2011; Karpinski et al., 2021). Traditional signal processing methods-power spectrum, crest factor, kurtosis, and time-domain averaging-have been proven effective for fault detection (Zhao et al., 2017; Zimroz & Bartelmus, 2010; Xu et al., 2020). However, these techniques assume linearity and stationarity in signal generation, limiting their accuracy in detecting transient vibrational behavior (Peng & Chu, 2004; Antoni, 2006; Inman et al., 2018).

Utilizing vibration signals obtained from sensors offers a powerful method for diagnosing faults in machinery, estimating remaining machine lifespan, and identifying early signs of structural failure (Nandi et al., 2005; Koç et al., 2019).

Advanced techniques such as neural networks, fuzzy logic, wavelet transforms, and support vector machines are increasingly being used to process these signals and predict failure modes (Yan et al., 2014; Widodo & Yang, 2007; Guo et al., 2018). In particular, waveform analysis through time-domain signal analysis has been found effective in analyzing non-steady conditions and brief transient impulses (Jardine et al., 2006; Samanta et al., 2003; Abry & Veitch, 2006). A renewed focus on structural damage

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identification and dynamic performance optimization has been advancing the field (Gao, 2007). Natural frequency shifts have proven effective in detecting beam damage through iterative schemes grounded in perturbation methods and generalized inverse theory (Ullah, Chan, & Chang, 2025). Robust Frequency Method (RFM) and Improved Frequency Method (IFM) were proposed to analyze variations in frequency and mode shapes in truss structures with uncertain parameters (Xu, Zhu, & Emory, 2007). Experimental and numerical studies have further validated the use of natural frequency changes for structural damage detection (Chan, Ullah, & Chang, 2023). Finite element analysis, combined with experimental validation, was employed to optimize the structural accuracy of a high-precision machine tool, yielding a strong correlation and an error margin of less than 1.56%.

This study is divided into two key sections. The first investigates the effects of transmission shaft geometry on the system's modal properties, including natural frequencies and mode shapes, through finite element analysis. Two distinct allowable transmission shaft lengths are examined to understand how geometric factors influence the system's vibrational response (Gao et al., 2018; Furuya et al., 2018). The second section explores the vibration behavior of the gear transmission system in the absence of ambient noise by constructing an acoustic isolation enclosure around the experimental setup (Bendat & Piersol, 2010; Hwang et al., 2017). Using accelerometers and an impact hammer to generate excitations, FRFs of the transmission shaft were measured and compared to numerical finite element results for validation.

A key contribution of this work is the development of a novel mathematical method based on definite integrals to calculate the area under the curve of vibration signals. This technique allows for precise detection of damage within the gearbox. By comparing the areas under the curve for a normal gear, a 2mm tooth-cut gear, and a fully tooth-cut gear, the method effectively identifies structural damage. Furthermore, the approach was extended to analyze gears with 2mm and 3mm cracks between teeth, producing consistent results that confirm the method's reliability in detecting faults. This innovative technique significantly enhances fault detection capabilities and has practical implications for improving the design, operation, and maintenance of rotating machinery.

EXPERIMENTAL SETUP

The physical device and experimental apparatus are depicted in Fig 1. The motor serves as the power supply and uses a set of pulleys to drive the gear drive shaft. The driven wheel shaft's housing is where the accelerometers are mounted. As shown in Fig. 1, the accelerometers are oriented in orthogonal directions, two in the radial direction, namely X and Y axes, and one in axial direction, namely the Z axis, of the shaft to pick up the measured vibration signals from the housing. The measured signals are processed through the data acquisition (DAQ) system, which is then connected to the computer. Data from the X, Y, and Z axes are collected using piezoelectric-type accelerometers.

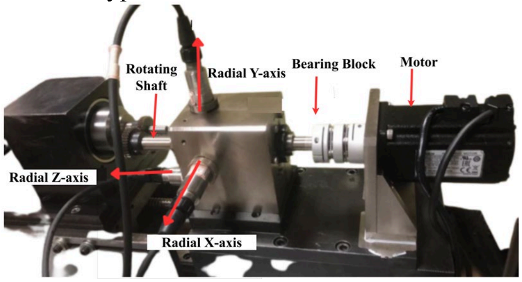


Fig. 1. Experimental apparatus showing placement of accelerometers for vibration measurements.

Design of shaft for investigation of geometric effects

Two spur gears are positioned within the gearbox at the end of the rotating transmission shaft, as shown in Fig. 2, to assess vibrational characteristics. The length of the transmission shaft plays a key role in the system's modal characteristics, directly impacting natural frequencies and mode shapes. In this study, transmission shafts of 120 mm and 60 mm, as depicted in Fig. 2(a) and Fig. 2(b), are used to examine the geometric influence on modal properties.

Both numerical finite element method (FEM) analyses and experimental modal analyses (EMA) are employed. FEM helps to predict modal properties by discretizing the shaft into finite elements (Zhang et al., 2015; Xie & Gao, 2012), while EMA validates these results by capturing real-world vibrations using accelerometers (Jiang et al., 2016; Shih et al., 2010). Previous studies have demonstrated that shaft length significantly affects vibrational behavior (Xue & Shao, 2008; Park & Kim, 2007), especially in gear systems (Li & Kahraman, 2005; Liew et al., 2017). The findings align with existing research, confirming that shaft geometry is critical in determining system performance (Sun et al., 2013; Akhtar et al., 2019).

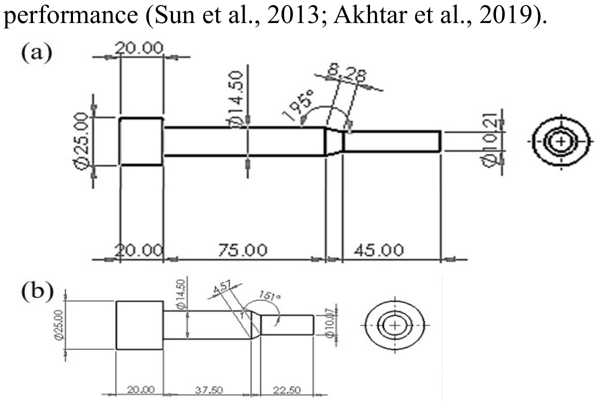


Fig. 2. Design of (a) 120 mm and (b) 60 mm long transmission shafts.

Setup for vibration signal processing

Two spur gears of the identical specifications are installed on the transmission shaft across from one another, as seen in Fig. 1 with specifications as shown in Table 1. As illustrated in Fig. 3, the rotating shaft's vibration signals were collected by the accelerometers, which were signal-conditioned by the DSA measurement device later analyzed by the FLEXDSA software installed in the laptop computer. The rotating shaft's vibration signals excited by an impact hammer were analyzed in the form of FRFs. The results are shown in "Natural frequency and FRF response of 60 mm shaft" in the following pages.

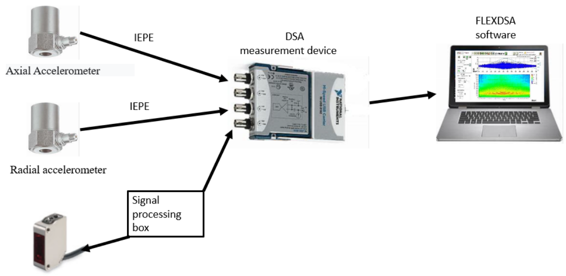


Fig. 3. Illustration of signal processing flow from accelerators to the computer with FLEXDSA program.

Table 1. Spur gear specifications.

Specification	Value
Type	Spur Gear
Number of Teeth	32
Hub Diameter	34 mm
Pitch Diameter	25 mm
Bore Diameter	10 mm
Material	S45C
Modulus	1
Pressure Angle	20^0

Vibrations of the transmission shaft

Free vibrations of stationary shafts were first examined with commercial finite element package to understand the shaft's modal contents. With one end fixed and the other end free, as exemplified by the 1st bending mode shape as shown in Fig. 4, it is apparent that the 120 mm shaft would exhibit larger lateral vibratory modal response than the 60 mm shaft.

This lateral bending mode shape can produce radial response, that is in X and Y axes, when the shaft is rotating and is excited by external vibration sources. As the length of the shaft is reduced by 50%, as shown in the Table 2. one can observe that the natural frequencies of the first three bending modes are significantly increased with 60 mm shaft as compared to the baseline 120 mm shaft. The implication of such increase in natural frequencies can be employed to decouple the shaft's vibrations from excitation frequencies as well as de-coupling from other mechanical parts. Through our investigations, the 60 mm shaft offers much higher vibratory response as opposed to 120 mm shaft as it is fully decoupled from other mechanical parts. Therefore, the following investigations are mainly focused on the results with 60 mm shaft.

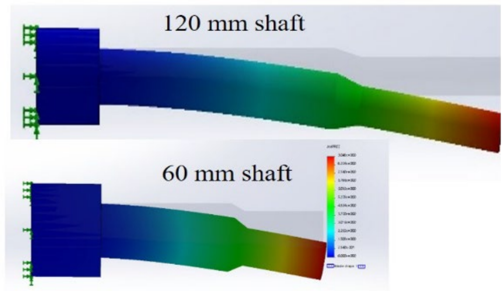


Fig. 4 Illustration of 1st bending mode shape.

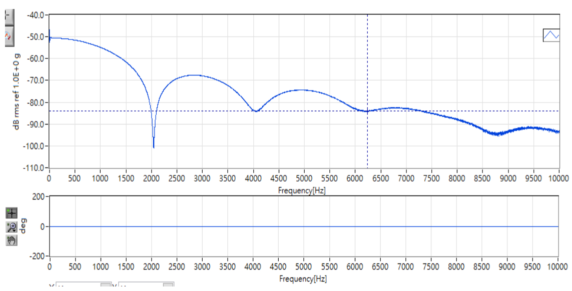
Table 2. Comparison of natural frequencies of first three bending modes of the model 120 mm and 60 mm shafts.

Mode	120mm shaft	60mm shaft
1st bending mode	780 Hz	2841 Hz
2nd bending mode	3370 Hz	11120 Hz
3rd bending mode	8147 Hz	16058 Hz

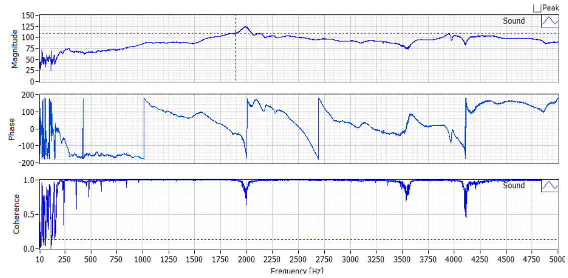
Natural frequency and FRF response of 60 mm shaft

In this vibration investigation experiment, a 60mm long, 14.5mm diameter steel shaft, coupled with a 32-tooth gear, is subjected to controlled excitation using an impact hammer, Fig. 5 and Fig. 6. A specific accelerometer captures the vibration response, while a data acquisition system records and processes the data. The experiment involves applying the impact hammer at five distinct points along the shaft, with the resulting vibrations recorded to analyze the dynamic behavior of the shaft under varying excitation conditions.

Natural frequency and FRF response of shaft without gear



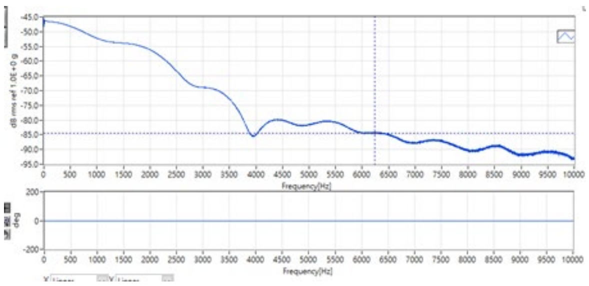
(a)



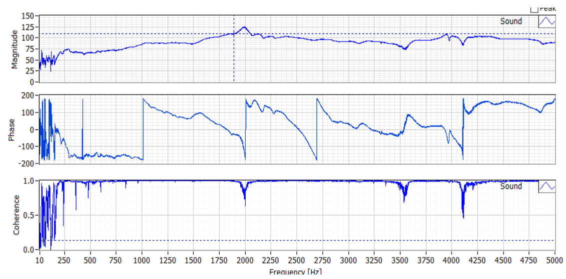
(b)

Fig. 5. (a)Cut off frequency (b) FRF Response of shaft without gear using impact hammer.

Natural frequency and FRF response of shaft with gear



(a)



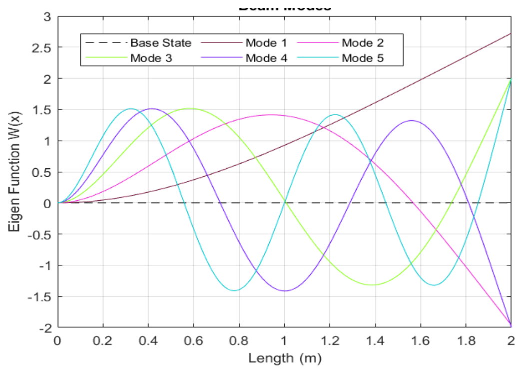
(b)

Fig. 6. (a)Cut off frequency (b) FRF Response of shaft with gear using impact hammer.

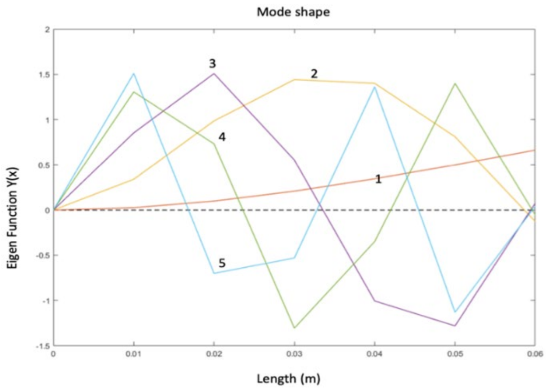
Mode shapes of 60 mm shaft with and without gear

Mode shapes represent the inherent deformation patterns a structure assumes when vibrating at its natural frequencies, illustrating how different points move relative to one another. In this study, the first five natural frequencies of a cantilevered steel shaft with and without a 32-tooth gear are shown in Fig. 7. Without the gear, the first three modes are bending, the fourth is twisting, and the fifth is a combination of bending and twisting.

Including the gear significantly shifts the natural frequencies: the first two bending modes shift downward, the twisting mode shifts upward, and the combination mode shifts slightly downward. These shifts result from the gear increasing the shaft's inertia and altering its cross-sectional stiffness. This has practical implications, as the lowered natural frequencies in the gear-equipped shaft make it more prone to vibrations, potentially leading to increased noise, vibration, and fatigue.



(a)



(b)

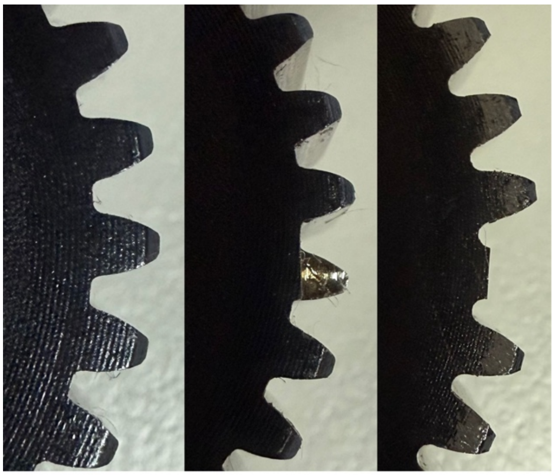
Fig. 7. (a) Mode shape of shaft without gear (b) Mode shape of shaft with gear.

Analyzing the effects of tooth damage on spur gear vibration

A 32-tooth spur gear is tested under three different conditions: a normal gear with no tooth damage, a gear with a 2mm tooth cut, and a gear with a full tooth cut, shown in Fig 8. The center distance between the gears is maintained at 0.25mm, and the gears are rotated at a constant speed of 1500 rpm. The goal is to compare the vibration characteristics of these three gear conditions under identical settings, enabling the development of techniques for detecting and diagnosing gear damage based on vibration analysis. Conditions are specified in Table 3.

Table 3. Experimental conditions for vibration analysis of spur gears

Parameter	Details
Gear Type	Spur Gear (32 teeth)
Gear Conditions Tested	- Normal gear (no tooth damage)
	- 2mm tooth cut gear
	- Full tooth cut gear
Center Distance	0.25mm
Rotational Speed	1500 rpm

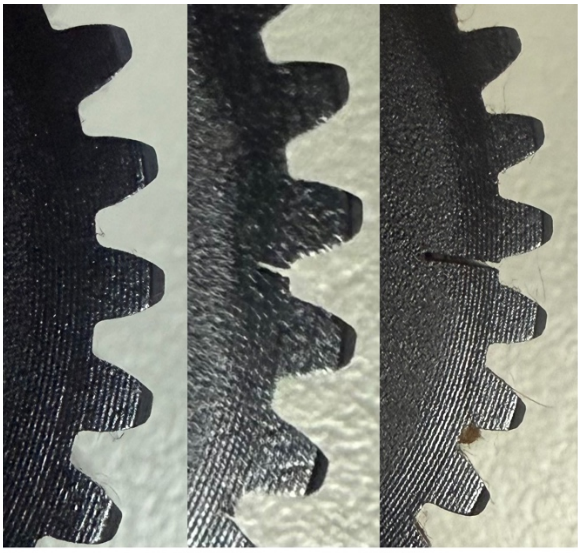


(a) (b) (c)

Fig. 8. Spur gear tooth cut (a) normal gear (b) 2 mm tooth cut gear (c) full tooth cut gear.

Analyzing the effects of tooth cracks on spur gear vibration

Three scenarios were tested: a pristine gear, one with a 1 mm crack, and another with a 3 mm crack. All cracks were 0.2 mm deep. The gears were maintained at a fixed additional center distance of 0.25 mm and rotated at 1500 rpm, shown in Fig 9.



(a) (b) (c)

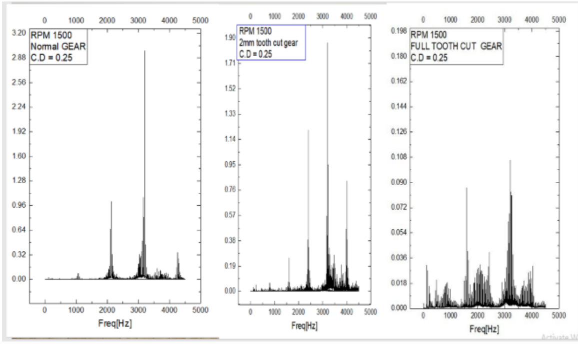
Fig. 9. Spur gear tooth crack (a) Normal gear
(b) 1mm tooth crack gear (c) 3mm tooth crack gear

RESPONSE RESULTS AND DISCUSSIONS

Data processing and analysis

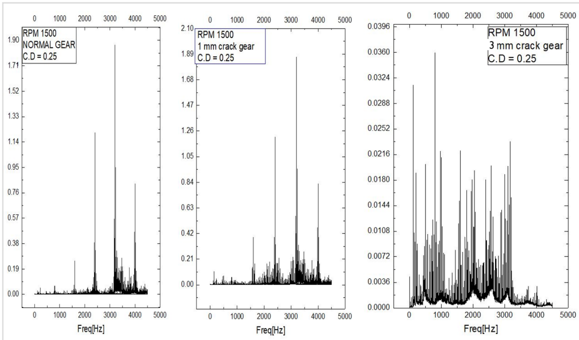
The raw vibration data from each test run is processed by converting it into the frequency domain using FFT techniques. This transformation generates a power spectrum, which highlights how vibration energy is distributed across various frequencies. By comparing the power spectra of healthy and damaged gears, such as normal and full tooth cut gears, specific damage features can be identified, Fig 5 and Fig 6. These features often manifest as distinct peaks or additional frequency components in the spectra of the damaged gears.

Further analysis involves comparing the damage signatures between normal and damaged gears and full-tooth cut damaged gears, allowing for the identification of damage-specific patterns related to different tooth profiles, Fig 10, and Fig 11.



(a) (b) (c)

Fig. 10. Fast Fourier Transform (FFT) data analysis (a) normal gear (b)2 mm tooth cut gear (c) full tooth cut gear.



(a) (b) (c)

Fig. 11. Fast Fourier Transform (FFT) data analysis (a) normal gear (b)1 mm tooth crack gear (c) 3 mm tooth crack gear

Area under the curve using mathematical equation and MATLAB.

A Mathematical concept called definite integration is employed:

Define the function:

f(x) represents the function that describes the power spectrum density (PSD) (g-RMS vs frequency). In this study, f(x) would be the mathematical formula representing the curve in the PSD graph.

Set the limits of integration:

$a = 3250$ Hz and $b = 3750$ Hz

Definite integral:

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∫ from a to b f(x) dx (1)

The integral represents the sum of infinitely many infinitesimal rectangular areas under the curve f(x) between x = a and x = b.

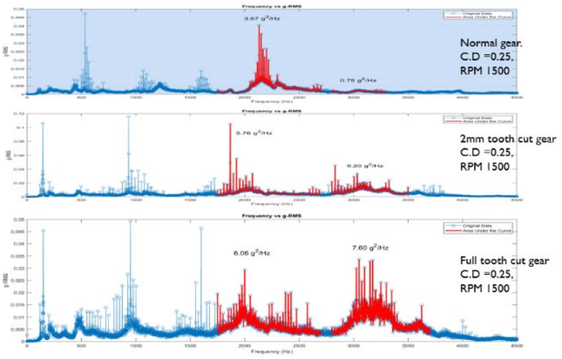

Fig. 12. Variation of area under the curve for normal gear,2 mm tooth cut gear and full tooth cut gear.

The Fig. 12 Presents three PSD plots, showing the vibration energy distribution of a spur gear under different tooth conditions: normal, 2mm tooth cut, and full tooth cut. The x-axis represents frequency, and the y-axis shows the root mean square (RMS) acceleration.

- **Normal Gear**: The PSD plot shows low vibration levels and minimal peaks, indicating a healthy gear.
- **2mm Tooth Cut Gear**: Vibration levels increase with more pronounced peaks, suggesting that the 2mm tooth cut introduces additional vibration.
- **Full Tooth Cut Gear**: The most significant increase in vibration and peaks is observed, indicating severe tooth damage.

The PSD plots demonstrate a clear relationship between tooth damage severity and increased vibration energy.

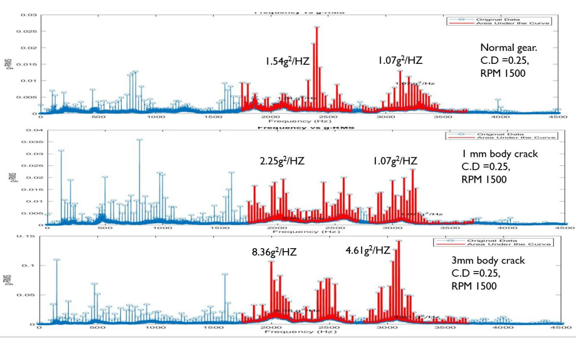

Fig. 13. Variation of area under the curve for normal gear (b)1 mm tooth crack gear (c) 3 mm tooth crack gear.

Fig 13 shows spur gear under three conditions: normal, 1mm crack, and 3mm crack. The x-axis represents frequency, and the y-axis shows g-RMS.

- **Normal Gear**: Low amplitude and random pattern, indicating minimal vibration and a healthy gear.
- **1mm Crack**: Increased amplitude with a more regular pattern, suggesting the crack introduces a dominant vibration frequency.
- **3mm Crack**: Highest amplitude and a pronounced periodic pattern, indicating significant vibration due to the severe crack.

The analysis reveals a clear relationship between tooth cut/crack severity and gear vibration. As the extent of damage increases, both the amplitude and periodic nature of the vibrations become more pronounced. This behavior highlights the potential of using novel method for calculating area under the curve to detect and assess the severity of gear tooth damage.

In order to assess the effectiveness of the AUC-based approach presented, a comparison with conventional techniques such as FFT peak analysis, crest factor, and kurtosis is made. The conventional techniques rely on linearity and stationary assumptions of the signal and do not have the sensitivity during transient conditions of the system. In contrast, the AUC method gives a quantitative assessment of energy distribution in a specified frequency band, enabling the detection of incipient faults of 2 mm tooth cuts and 1 mm cracks to be made accurately. This indicates better sensitivity and reliability in diagnosis, thus suggesting its application in continuous gear condition monitoring.

CONCLUSIONS

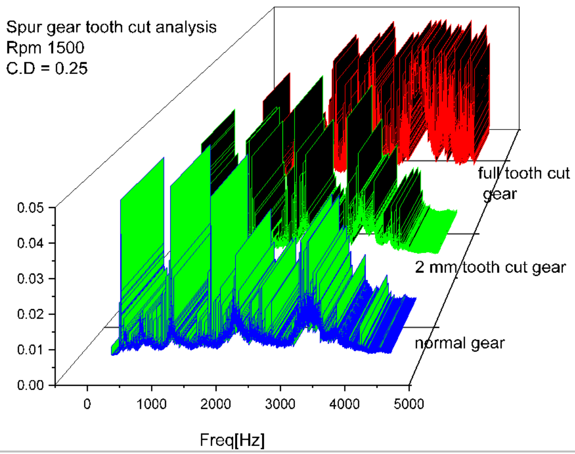

Fig. 14. Spur gear tooth cut analysis.

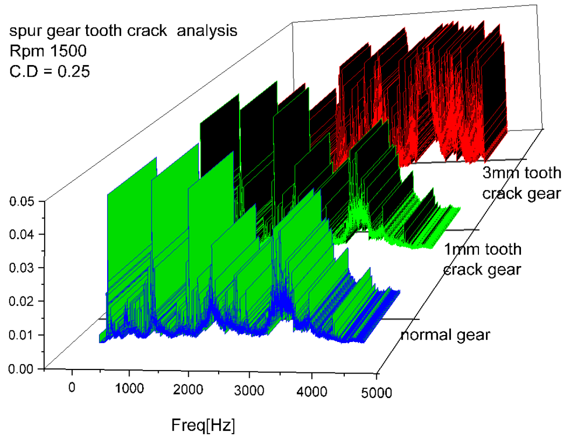

Fig. 15. Spur gear tooth crack analysis.

The PSD analysis of spur gears with artificial defects including tooth cuts and cracks demonstrates a direct relationship between damage severity and dynamic response characteristics shown in Fig. 14 and Fig. 15, Progressive damage conditions result in increased amplitude magnitudes and distinct frequency peaks, reflecting elevated vibrational energy and periodic excitation. This trend confirms that integrating definite-integral based metrics, such as the area under the PSD curve (AUC), serves as a robust quantitative indicator for detecting and evaluating gear tooth degradation. The proposed AUC-based method thus enhances the sensitivity and accuracy of vibration-based diagnostic strategies for gear fault identification.

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