Hindawi Publishing Corporation
Journal of Engineering
Volume 2014, Article ID 857462, 11 pages
http://dx.doi.org/10.1155/2014/857462

*Research Article*

# Experimental Investigations of Noise Control in Planetary Gear Set by Phasing

**S. H. Gawande[1] and S. N. Shaikh[2]**

[1] *Department of Mechanical Engineering, M. E. Society's College of Engineering, Pune, Maharashtra, India*
[2] *Department of Mechanical Engineering, AISSMS College of Engineering, Pune, Maharashtra, India*

Correspondence should be addressed to S. H. Gawande; shgawande@yahoo.co.in

Received 6 August 2014; Accepted 25 October 2014; Published 25 November 2014

Academic Editor: Radhey S. Jangid



Now a days reduction of gear noise and resulting vibrations has received much attention of the researchers. The internal excitation caused by the variation in tooth mesh stiffness is a key factor in causing vibration. Therefore to reduce gear noise and vibrations several techniques have been proposed in recent years. In this research the experimental work is carried out to study the effect of planet phasing on noise and subsequent resulting vibrations of Nylon-6 planetary gear drive. For this purpose experimental set-up was built and trials were conducted for two different arrangements (i.e., with phasing and without phasing) and it is observed that the noise level and resulting vibrations were reduced by planet phasing arrangement. So from the experimental results it is observed that by applying the meshing phase difference one can reduce planetary gear set noise and vibrations.

## 1. Introduction

Gears are essential parts of many precision power and torque transmitting machine such as an automobile. The major functions of a gearbox are to transform speed and torque in a given ratio and to change the axis of rotation. Planetary gears yield several advantages over conventional parallel shaft gear systems. They produce high speed reductions in compact spaces, greater load sharing, higher torque to weight ratio, diminished bearing loads, and reduced noise and vibration. They are used in automobiles, helicopters, aircraft engines, heavy machinery, and a variety of other applications. Despite their advantages, the noise induced by the vibration of planetary gear systems remains a key concern. Planetary gears have received considerably less research attention than single mesh gear pairs. There is a particular scarcity of analysis of two planetary gear systems and their dynamic response. This paper focus on the study of two PGTs with different phasing (angular positions) while keeping every individual set unchanged.

Planetary gear systems are used to perform speed reduction due to several advantages over conventional parallel shaft gear systems. Planetary gears are also used to obtain high power density, large reduction in small volume, pure torsional reactions, and multiple shafting. Another advantage of the planetary gearbox arrangement is load distribution. Because the load being transmitted is shared between multiple planets, torque capability is greatly increased. The more the planets in the system, the greater the load ability, and the higher the torque density. The planetary gearbox arrangement also creates greater stability due to the even distribution of mass and increased rotational stiffness. Despite their advantages the noise induced by vibrations of planetary is concern, particularly in automotive industry where the vehicle interior noise is a key quality metric.

Extensive research work has been carried out by many researchers on the analysis of errors, dynamic response, and noise and vibration reduction in single planetary gears. Schlegel and Mard [1] proposed one strategy of reducing planet gear vibrations which used planet phasing, where the planet configuration and tooth numbers are chosen such that self-equilibration of the mesh forces reduce the net forces and torques on the sun, ring, and carrier to reduce vibration and noise up to 11 dB. Seager [2] explained a more

detailed analysis using a static transmission error model of the dynamic excitation. Palmer and Fuehrer [3] also demonstrated the effectiveness of planet phasing and support their arguments with limited experiments. Kaharamam and Blankership [4] studied the use of planet phasing in the context of helical planetary systems in which author used the static transmission error to represent the dynamic excitation in a lumped parameter dynamic model. All of these studies focus on planetary gears with equally spaced planet gears. Parker [5] provided physical explanation for the effectiveness of planet phasing to suppress planetary gear vibration based on the physical forces acting at the sun, planet, and ring meshes. He demonstrated that phenomenon with a dynamic finite element/contact mechanics simulation. Gill-Jeong [6] analyzed a new method of reducing vibrations of spur gear. He did numerical study on reducing the vibration of spur gear pairs with phasing. This new method is based on reducing the variation in mesh stiffness by adding another pair of gears with half-pitch phasing. This reduces the variation in the mesh stiffness of the final (phasing) gear, because each gear compensates for the variation in the other's mesh stiffness. Chen and Ishibashi [7] have investigated the relationship between the meshing phase difference and torsional vibration of planet gears from the standpoint of their rotational meshing cycle. Using planetary gear sets with and without a meshing phase difference, measurements were made of their noise and vibration acceleration under various driving conditions. The method of finishing the gears, the tooth profile contact ratio, and other factors were varied in order to compare and analyze the measured data. In present work method proposed in [7] is extended to reduce the noise level and resulting vibrations of planetary gear set. Meshing of the gears in the planetary gear set that forms the ratio-changing mechanism of an automatic transmission produces gear noise over a wide range of driving conditions from low to high vehicle speed. As per the literature survey it is observed that the internal excitation caused by the variation in tooth mesh stiffness is a key factor in causing vibrations [8, 9]. In order to study the phenomenon of noise control in planetary gear set it is necessary to study the dynamic behavior of gear train.

## 2. Dynamic Models of Gear Train

To understand and control gear noise, it is necessary to have knowledge not only about the gears, but also about the dynamic behavior of the system consisting of gears, shafts, bearings, and gear train casing. While designing the gear train the noise characteristics of a gear train can be controlled at the drawing board, because all the components have an important effect on the acoustical output [10]. For relatively simple gear systems it is possible to use lumped parameter dynamic models with springs, masses, and viscous damping. For more complex models, which include, for example, the gear train casing, finite element modeling and analysis is often used. The first dynamic models were used to determine dynamic loads on gear teeth and were developed in the 1920s; the first mass-spring models were introduced in the 1950s [11].

*2.1. Lumped Parameter Dynamic Models.* Özgüven and Houser [11] reviewed the literature on mathematical models used in gear dynamics from 1915 and up to 1986. In this work extensive literature review is carried out. They classified the models in five groups; (1) Simple dynamic factor models: this group includes most of the early studies in which a dynamic factor that can be used in gear root stress formulae is determined. These studies include empirical and semiempirical approaches as well as recent dynamic models constructed just for the determination of a dynamic factor. (2) Models with tooth compliance: there are a very large number of studies that include only the tooth stiffness as the potential energy storing element in the system. That is, the flexibility of shafts, bearings, and so forth is all neglected. In such studies the system is usually modeled as a single-degree-of-freedom spring-mass system. There is an overlap between the first group and this group since such simple models is sometimes developed for the sole purpose of determining the dynamic factor. (3) Models for gear dynamics: such models include the flexibility of the other elements as well as the tooth compliance. Of particular interest have been the torsional flexibility of shafts and the lateral flexibility of the bearings and shafts along the line of action. (4) Models for geared rotor dynamics: in some studies, the transverse vibrations of a gear carrying shaft are considered in two mutually perpendicular directions, thus allowing the shaft to whirl. In such models, the torsional vibration of the system is usually considered. (5) Models for torsional vibrations: the models in the third and fourth groups consider the flexibility of gear teeth including a constant or time varying mesh stiffness in the model. However there is also a group of studies in which the flexibility of gear teeth is neglected and a torsional model of a geared system is constructed by using torsionally flexible shafts connected with rigid gears. The studies in this group may be viewed as pure torsional vibration problems, rather than gear dynamic problems. In a study by Cheng [12], the vibrations of spur gears were simulated. The dynamics of the gears was modelled as a nonlinear time-correlated, stationary stochastic process. As excitation of the system, random and harmonic transmission error was used. The vibrations excited by random and harmonic transmission error and time varying mesh stiffness were investigated at different speeds and different loads. Optimization of gear parameters were made to avoid resonance. Kahraman and Singh [13] used a two-degree-of-freedom model of a spur gear pair with backlash, to investigate the nonlinear frequency response characteristics, for both internal and external excitations. Transmission error due to variation in mesh stiffness was used as internal excitation and low frequency torque variations were used as external excitation. Two solution methods, digital simulation technique and the method of harmonic balance, were used to develop the steady state solutions for the internal sinusoidal excitation. Analytical predictions were shown to match satisfactorily with experimental data available in the literature. A parameter study showed that the mean load determined the conditions for no impacts, single sided impacts, and double-sided impacts. A six-degree-of-freedom model of a spur gear pair was developed by Torby [14]. The gears were supported by elastic bearings with

viscous damping present. Varying mesh stiffness and friction in the gear mesh were included in the model. The equations of motion were solved by numerical integration.

*2.2. Dynamic Models of Complete Gear Trains/Gear Boxes.* Many researchers have modelled complete gearboxes in order to predict gear noise. Campell et al. [15] used finite element dynamic modelling methods to predict gear noise from a rear wheel drive automatic transmission. The model was used to investigate the effects of different component's inertia, stiffness, and resonance. The ring gear and shaft resonances and the tailstock housing stiffness were found to be significant design factors that influenced the gear-whine. Model construction issues were discussed as well as correlation of predicted gear noise traces with operating measurements. Ariga et al. [16] described a systematic approach to reduce the overall gear noise from a four-speed automatic transaxle in which the vibration characteristics were identified by finite element analysis [FEA]. A new gear train structure for effective in reducing gear noise was investigated. The effect of the modifications was verified experimentally, and the gear noise level was reduced substantially. Also changes in stiffness of the transmission case, at locations supporting the gear train bearings, were shown to affect the gear noise. Dynamic models of typical automotive gearing applications using spur, helical, bevel, hypoid, and planetary gear sets were developed by Donley et al. [17]. Basic formulations used in modelling different types of gears were discussed. These models were designed for use in finite element models of gearing systems for simulating gear-whine. A procedure for calculating the dynamic mesh force generated, and gear case response, per unit transmission error was proposed. A simplified automotive transmission was analysed to demonstrate the features of the proposed gear noise reduction technique. A basic approach to gearbox noise prediction was described by Mitchell and Daws [18]. The proposed method was dynamic modelling of the gearbox from inside out. The computational strategy for the determination of the dynamic response of the internal gearbox components was described. Force coupling between gears and dynamic coupling due to, for example, unbalance was discussed. The transfer matrix approach was used for the analysis and a benchmark example was presented to verify the calculation method. Hellinger et al. [19] used numerical methods to calculate gear noise from a transmission. They used finite element analysis to calculate natural frequencies and forced vibrations of the gearbox structure (housing). As input for the finite element calculation of forced response, they used the dynamic bearing forces of the shafts in the gearbox, calculated by multibody system software. Finite element analysis was used by Nurhadi [20] to investigate the influence of gear system parameters on noise generation. A direct time integrating method was used to predict sound generation, transmission, and radiation from mechanical structures. Naas et al. [21] optimized the gear noise from a car gearbox. They calculated transmission error and time varying stiffness of the gear mesh by using a finite element based computer program. The results were used as input to a torsional vibration model of the power train (from the engine to the wheels). The output from the torsional vibration model was mesh forces, which were transformed to the frequency domain and used as input to a finite element model of the gearbox, which was used to predict the forced response. As a result of the simulations, a modified gearbox was tested in a car and the gear noise was substantially reduced.

As per literature survey it is observed that a lot of instrument and measuring devices are required to control noise and vibration in gear boxes. Hence in order to investigate noise and vibration reduction in PGT in this paper a simple approach is proposed without the requirement of additional instruments like actuators, external power, and advanced signal processing techniques. This paper is organized as follows: Sections 1 and 2 focus on literature review and recent development in the subject, problem formulation, and objective of work is stated in Section 3; experimental set-up and measurement technique is explained in Section 4; Sections 5 and 6 focus on basics of noise and vibration in PGT and results and discussion; concluding remark is explained in Section 7.

## 3. Problem Formulation and Objective

Meshing of the gears in the planetary gear set that forms the ratio-changing mechanism of an automatic transmission produces gear noise over a wide range of driving conditions from low to high vehicle speed. As per the literature survey and industrial survey it is observed that the internal excitation caused by the variation in tooth mesh stiffness is a key factor in causing vibrations. To reduce gear vibrations, numbers of passive and active methods are reported. Many studies have been concentrated on the modification of gear teeth, but these methods have limitations on modifications. Passive methods like the use of periodic struts for gearbox support systems and periodic drive shafts are also reported to reduce gear vibrations. But these methods require additional actuators, external power, and many signal processing techniques.

Hence in order to investigate noise and vibration reduction in planetary gear train by phasing, in this work a simple approach is proposed without the requirement of additional instruments like actuators, external power, and signal processing techniques which generally results in increase in cost. To investigate noise and subsequent vibrations reduction in planetary gear set by phasing, two objectives were set such as design, development of schematic and systematic arrangement of system components to build test rig set-up and performance analysis by conducting trials with and without phasing.

## 4. Experimental Set-Up and Measurements

As per the requirement to accomplish the objectives it was necessary to develop the method to reduce PGT noise and vibrations by gear itself without requiring the additional energy, actuators, and advanced signal processing techniques. Viewing this need the method of noise reduction in planetary gears by phasing is introduced in this research work.

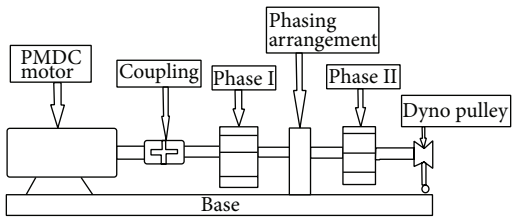


Figure 1: Schematic layout of test set-up for measurement of planetary gear set noise.

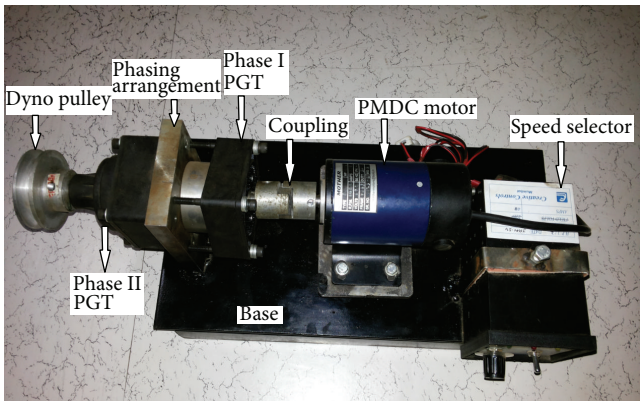


Figure 2: Experimental test rig.

In order to study the effect of phasing on noise and vibrations of planetary gear set the required experimental set-up was developed as shown in Figure 1.

Figure 1 shows schematic layout of test set-up developed for the measurement of noise level of planetary gear set by phasing. Figure 2 also shows the position of various components like motor, planetary gear sets 1 and 2, coupling, and speed regulator. The experimental work was carried out to study the effect of meshing phasing on noise level and vibrations of Nylon-6 planetary gear set. For this purpose experimental set-up was built as shown in Figure 2. Rectangular plate is placed between planetary gear sets 1 and 2 to provide meshing phase difference between ring gear of gear sets 1 and 2. Noise level is measured for two different arrangements as with phasing and without phasing. Experimental set-up shown in Figure 2 consists of different components such as PMDC motor, love joy coupling, planetary gear set, and speed selector which are explained in this section.

*4.1. Motor Selection.* The speed reduction is to be achieved using two stage reductions; hence the stage wise reduction of the system is as follows.

Stage-I

(a) Input speed = 1440 rpm.

(b) Reduction ratio = 4.

(c) Output speed of stage I = 1440/4 = 360 rpm.

Table 1: Material selection for coupling.

| Designation | Ultimate tensile strength N/mm$^2$ | Yield strength N/mm$^2$ |
|---|---|---|
| EN 9 | 600 | 480 |

Figure 3: Permanent magnet DC motor.

Stage-II

(d) Input speed = 360 rpm.

(e) Reduction ratio = 4.

(f) Output speed of stage-II = 360/4 = 90 rpm.

(g) Hence, maximum motor speed = 1500 rpm.
Power = $2 \times \pi \times 1500 \times 0.5/60 = 78.5$ W.

Hence motor of 90 watt is selected as shown in Figure 3.

(i) Motor type: fractional HP permanent magnet DC motor.

(ii) Torque: 5 kgcm.

(iii) Speed: 1500 rpm.

(iv) Input power: 90 watt.

*4.2. Lovejoy Coupling.* Lovejoy Coupling L-075 (as per specification shown in Table 1) is selected for the given application with outside diameter of hub ($D_o$) = 38 mm, inside diameter of hub ($D_i$) = 12 mm, Lovejoy Coupling L-075 considered to be a hollow shaft subjected to torsional load.

*4.3. Selection of Gear Box.* Nylon-6 planetary gear box (Table 2) is selected based on mechanical properties, wear resistance, lubrication and material availability, torque, and other parameters.

*Gear Pair-1 (Phase I).* Tables 3 and 4 show the planet gear and internal gear ring specifications.

*Gear Pair-2 (Phase I).* Tables 3 and 5 show the planet gear and sun gear specification.

*4.4. Coupler Shaft.* Coupler shaft is used to connect planetary gear sets one and two. Table 6 shows different mechanical properties of selected material EN24.

*4.5. Coupler Shaft Bearing-I.* Coupler shaft bearing is subjected to purely medium radial and axial loads; hence single row deep groove ball bearing is selected as per specifications shown in Table 7.

Table 2: Mechanical properties of Nylon-6 gear box.

| Mechanical properties | ASTM test method | Units | Nylon 6/6 | Nylon 6/6 GF30 |
|---|---|---|---|---|
| Tensile strength 73°F | D638 | psi | 12,400 | 27,000 |
| Elongation 73°F | D638 | % | 90 | 3 |
| Flexural strength, 73°F | D790 | psi | 17,000 | 39,100 |
| Flexural modulus, 73°F | D790 | psi | $4.1 \times 10^5$ | $12 \times 10^5$ |
| Izod impact strength, Notched, 73°F | D256 | — | R120-M79 | M101 |
| Rockwell hardness | D785 | ft-lbs/in. | 1.2 | 2.1 |

Table 3: Planet gear specification.

| | |
|---|---|
| Material | Nylon-6 |
| Module | 1.375 mm |
| Number of teeth | 16 |
| Addendum diameter | 24.75 mm |
| Pitch circle diameter | 22 mm |

Table 4: Internal gear ring specification.

| | |
|---|---|
| Material | Nylon-6 |
| Module | 1.375 mm |
| Number of teeth | 48 |
| Addendum diameter | 68.75 mm |
| Pitch circle diameter | 66 mm |

Table 5: Sun gear specification.

| | |
|---|---|
| Module | 1.375 mm |
| Number of teeth | 16 |
| Addendum diameter | 24.75 mm |
| Pitch circle diameter | 22 mm |

Table 6: Selection of material for coupler shaft.

| Designation | Ultimate tensile strength N/mm$^2$ | Yield strength N/mm$^2$ |
|---|---|---|
| EN 24 (40 N; 2 Cr 1 Mo 28) | 720 | 600 |

*4.6. Gear Pair-3 (Phase II).* Tables 8 and 9 show the specification of planet gear and internal gear ring.

*4.7. Gear Pair-4 (Phase II).* Tables 10 and 11 show the planet gear and internal gear ring specification.

*4.8. Sound Level Meter.* For measurement of noise level in PGT sound level meter with specifications as shown in Table 12 was used.

*4.9. FFT Analyzer.* For measurement of noise level in PGT FFT Analyzer with specifications as shown in Table 13 was used.

*4.10. Phasing Arrangement.* Figure 4 shows planetary gear set used in experimental set-up. This planetary gear set is used to calculate the angle of indexing. Phase difference is provided in between ring gear of planetary gear sets 1 and 2. Figures 5(b) and 5(c) show without phasing and with phasing arrangement. Phase difference is provided as shown in Figure 5(a).

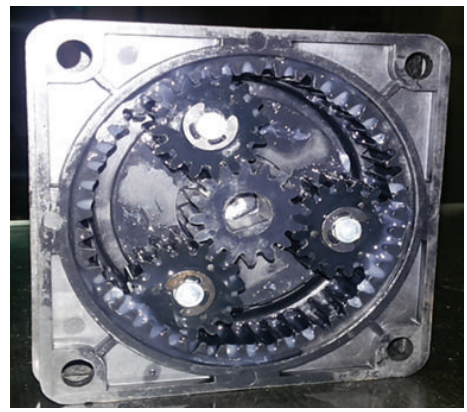

Figure 4: Planetary gear set.

No of teeth of ring gear = 48.

Angle of pitch = 360/48 = 7.5° = 7.5/2 = 3.75°.

Angle of indexing = 5 × angle between two teeth = 5 × 3.75 = 18.75°.

## 5. Noise Measurement in PGT

Noise measurement and signal analysis are important tools when experimentally investigating gear noise. Gears create noise at specific frequencies, related to the rotational speed and number of teeth of the gear. It is also possible to detect different errors like for example, run out (eccentricity) due to side-band generation [22]. Closely related is also vibration measurement and signal analysis for the purpose of gear fault detection, used in machine diagnostics in order to detect gear failures before catastrophic failure occurs. Middelton [23] discussed noise testing of gearboxes in the production line in which a noise testing equipment, utilizing low cost digital analysis and control techniques, was described. For each gear of the gearbox, the speed was ramped up while measuring noise with three microphones. For each order of interest (gear mesh frequency and its harmonics) the pass/fail target levels were defined by testing a selection of gearboxes which had noise characteristics regarded as just acceptable.

Table 7: Selection of coupler shaft bearing series 60.

| Bearing of basic design number (SKF) | $D$ | $D_1$ | $D$ | $D_2$ | $B$ | Basic capacity | |
|---|---|---|---|---|---|---|---|
| 6003 | 17 | 19 | 35 | 33 | 10 | 2850 | 4650 |

Table 8: Planet gear specification.

| | |
|---|---|
| Material | Nylon-6 |
| Module | 1.375 mm |
| Number of teeth | 16 |
| Addendum diameter | 24.75 mm |
| Pitch circle diameter | 22 mm |

Table 9: Internal gear specification.

| | |
|---|---|
| Material | Nylon-6 |
| Module | 1.375 |
| Number of teeth | 48 |
| Addendum diameter | 68.75 |
| Pitch circle diameter | 66 |

Table 10: Planet gear specification.

| | |
|---|---|
| Material | Nylon-6 |
| Module | 1.375 mm |
| Number of teeth | 16 |
| Addendum diameter | 24.75 |
| Pitch circle diameter | 22 |

Table 11: Sun gear specification.

| | |
|---|---|
| Material | Nylon-6 |
| Module | 1.375 mm |
| Number of teeth | 16 |
| Addendum diameter | 24.75 |
| Pitch circle diameter | 22 |

Table 12: Specifications of sound level meter.

| | |
|---|---|
| Frequency range | 31.5 Hz~8 KHz |
| Display | LCD |
| Measuring level range | 35~130 dB |
| Accuracy | ±1.5 dB (under reference conditions) |
| Dynamic range | 65 dB |
| Power supply | One 9 V battery, 006P or IEC 6F22 or NEDA 1604 |
| Calibration | Electrical calibration with the internal oscillator (1 kHz sine wave) |

A test rig was developed by Gielisch and Heitmann [24] to investigate gear noise from a car rear axle, without the need for a complete vehicle. Vibrations were measured on the final drive casing and the corresponding forces and torques in the gearing were calculated. The investigation gave information about the dynamics of the driving gear and the possibility to make comparisons between different driving gears. Oswald et al. [25] investigated the influence of gear design on gearbox radiated noise in which nine different spur and helical gear designs were tested in a gear noise test rig to compare the noise radiated from the gearbox top for the various gear design and the results were summarized as follows.

Table 13: Specifications of FFT Analyzer.

| | |
|---|---|
| Dynamic signal analyzer | Type PHOTON+ |
| Electronics | Electronics differential amplifier, programmable gain amplifier, |
| Frequency range | Up to 84 kHz analysis frequency (192 k samples per second) |
| Voltage ranges | ±0.01, ±0.1, ±1.0, ±10 V |
| Resolution | 24-bit |
| Dynamic range | 115 dBfs two-tone test, 100 linear averages |
| Accuracy | ±0.04 dB (1 kHz sine at full scale) |

(i) The total contact ratio was the most significant factor for reducing noise, increasing either the profile or face contact ratio reduced the noise.

(ii) The noninvolute spur gears were 3-4 dB noisier than involute spur gears.

(iii) High contact ratio spur gears showed a noise reduction of about 2 dB over standard spur gears.

(iv) The noise level of double helical gears averaged about 4 dB higher than otherwise similar single helical gears.

In this paper method proposed in [7] is extended by applying novelty of phasing concept to reduce the noise level and resulting vibrations of planetary gear set without the requirement of additional instruments like actuators, external power, and signal processing techniques which generally results in increase in cost.

Figure 6 shows experimental set-up developed to measure noise of planetary gear set for with and without phasing arrangement. For development of test ring as shown in Figure 6 exhaustive literature review was carried out [22–26]. Specific motor speed (e.g., 1200 rpm) is selected with the help of speed regulator and gear set is rotated; then sound meter [26] and FFT Analyzer were used to measure noise level of gear set. In noise reduction tests, variation due to unintended effects, such as testing different part specimens or even reassembly with the same parts, may be of the same order of magnitude as the effect of deliberate design changes.

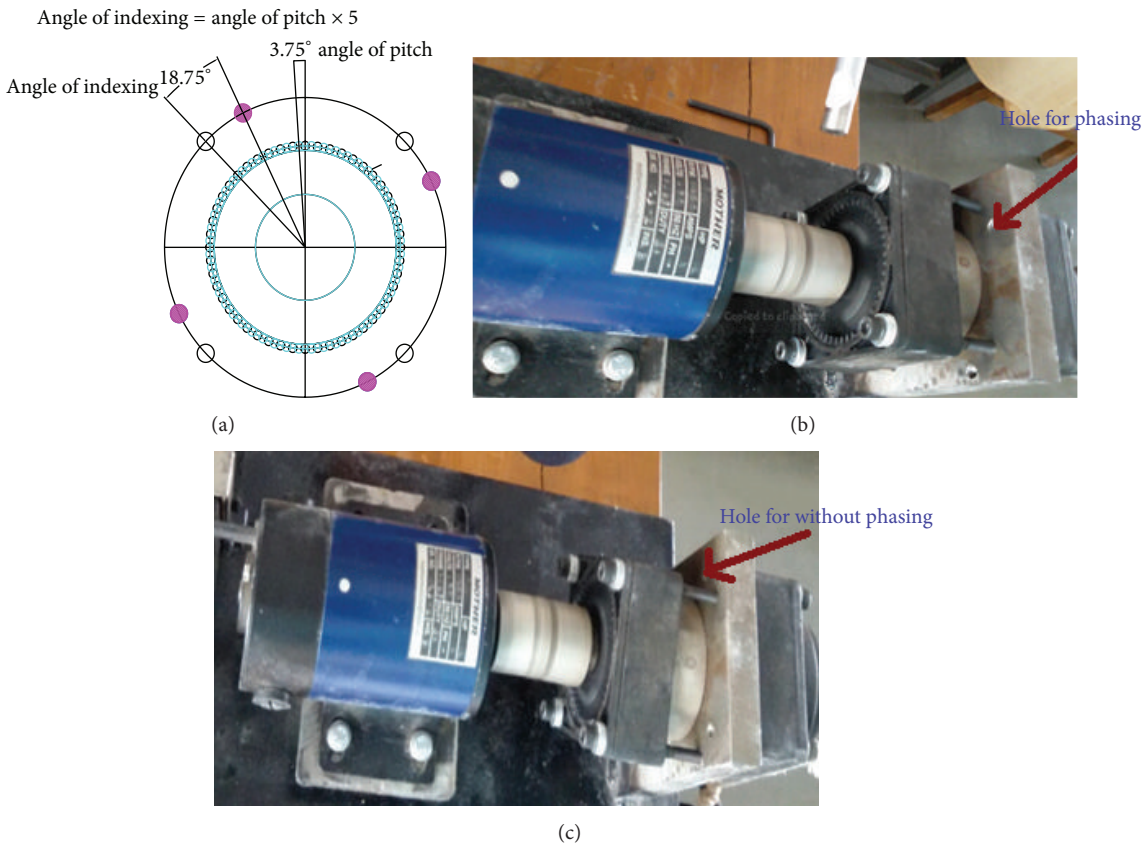


Figure 5: (a) Angle of Indexing, (b) gear arrangement without phasing, and (c) gear arrangement with phasing (normal gears).

Table 14: Observations without phasing arrangement.

| Load (Kg) | Speed (rpm) | Torque (Nm) | Power (W) | Efficiency (%) | Noise |
|---|---|---|---|---|---|
| 1 | 74 | 1.56 | 12.00 | 17.14 | 78 |
| 2 | 73 | 3.13 | 23.35 | 33.31 | 80 |
| 3 | 72 | 4.70 | 35.52 | 50.74 | 83 |
| 4 | 71 | 6.27 | 46.71 | 66.73 | 86 |
| 5 | 71 | 7.84 | 58.29 | 83.93 | 89 |
| 6 | 70 | 9.41 | 69.03 | 98.90 | 91 |

## 6. Results and Discussion

To validate the experimental findings number of trials were conducted by varying load from 1 Kg to 6 Kg and changes in torque, power, and resulting noise level are noted. Results obtained by without phasing and with phasing arrangement are shown in Tables 14 and 15. From Tables 14 and 15 relations between speed, measured torque, power, mechanical efficiency, and noise are plotted as shown in Figures 7, 8, and 9 for with and without phasing arrangement. The results were remarkably consistent with experimental measurements including excellent predictions of noise level by using sound level meter and FFT Analyzer.

From Figure 7 it is observed that as speed increases efficiency decreases. Efficiency of with phasing arrangement is greater as compared to without phasing arrangement at motor speed of 1200 rpm. Efficiency is maximum with 98.90% value at 70 rpm output speed at dyno pulley for without phasing arrangement, whereas efficiency is maximum as 99.45% value at 71 rpm output speed at dyno pulley with phasing arrangement. This shows efficiency is improved by using with phasing arrangement.

From Figure 8 it is observed that as speed increases, noise level decreases because speed decreases with increase in load. Noise level in without phasing arrangement (i.e., 78 dB at 74 rpm) is greater as compared to with phasing arrangement (i.e., 74 dB at 74 rpm) at motor speed of 1200 rpm. This shows noise level decreases with 4 dB by using with phasing arrangement. From Figure 8 it is concluded that the level of noise decreases when phase difference is provided. Figure 9 shows the effect of the meshing phase difference on the measured noise level. From Figure 9 it is seen that

Table 15: Observations with phasing arrangement.

| Load (Kg) | Speed (rpm) | Torque (Nm) | Power (W) | Efficiency (%) | Noise (dB) |
|---|---|---|---|---|---|
| 1 | 75 | 1.56 | 12.25 | 17.50 | 72 |
| 2 | 74 | 3.13 | 24.25 | 34.64 | 74 |
| 3 | 73.5 | 4.70 | 36.17 | 51.67 | 76 |
| 4 | 72.5 | 6.27 | 47.60 | 68.00 | 79 |
| 5 | 72 | 7.84 | 59.11 | 84.44 | 82 |
| 6 | 71 | 9.41 | 69.96 | 99.45 | 85 |

Figure 6: Recording results on sound meter and FFT spectrum analyzer.

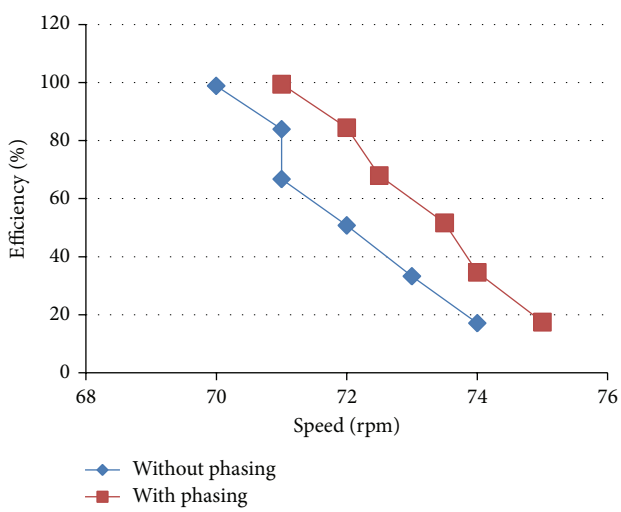


Figure 7: Comparison of speed (rpm) with efficiency (%) without and with phasing.

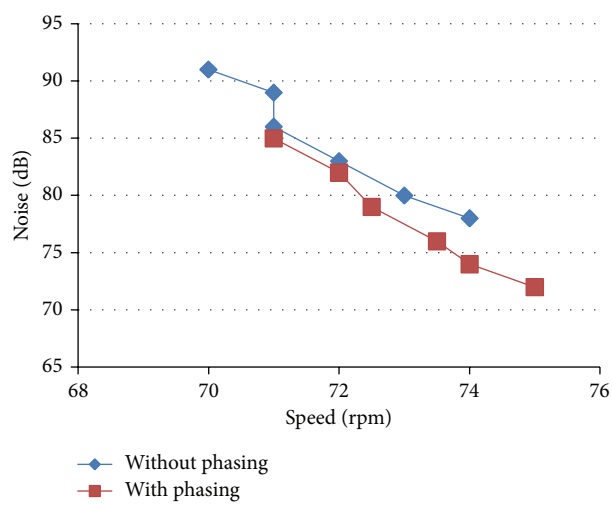


Figure 8: Comparison of speed (rpm) with noise (dB) without and with phasing.

as load increases noise level increases drastically in case without phasing arrangement as compared to with phasing arrangement at motor speed of 1200 rpm.

Figures 10 and 11 show the variation of noise level with frequency at 1200 rpm for 6 Kg load without and with phasing obtained by FFT Analyzer. By comparing Figures 10 and 11 it is seen that the average noise level is reduced by 6 dB to 7 dB. By using FFT Analyzer similar plots were obtained for load varying from 1 Kg to 5 Kg and resulting changes in noise level were recorded for required frequencies. For 1 kg load, from Figure 12 noise level with magnitude of 84 dB at frequency of 2000 Hz for without phasing arrangement is seen, while in with phasing arrangement noise measured found to be 77 dB. This indicates that the level of noise was reduced by 7 dB with phasing arrangement. For 2 kg load, from Figure 13 noise level with magnitude of 84.53 dB at frequency of 2000 Hz for without phasing arrangement is

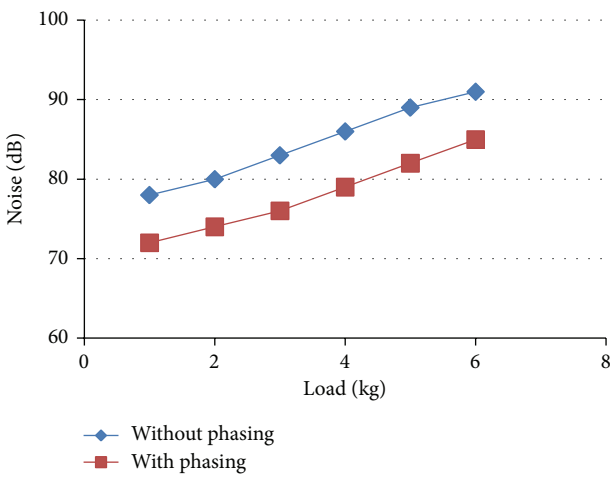


FIGURE 9: Comparison of load (kg) with noise (dB) without and with phasing.

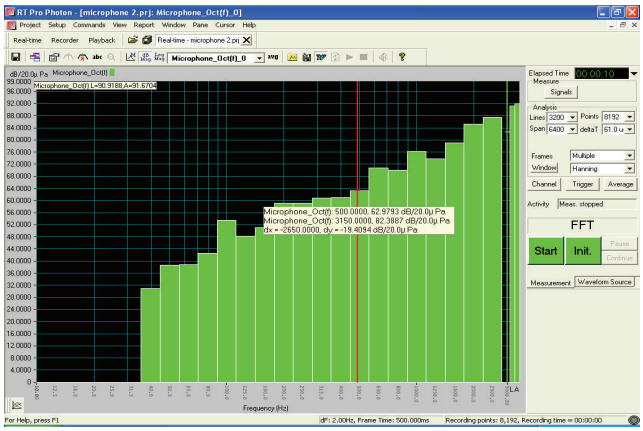


FIGURE 10: Variation of noise (dB) with frequency (Hz) at 1200 rpm for 6 Kg load without phasing.

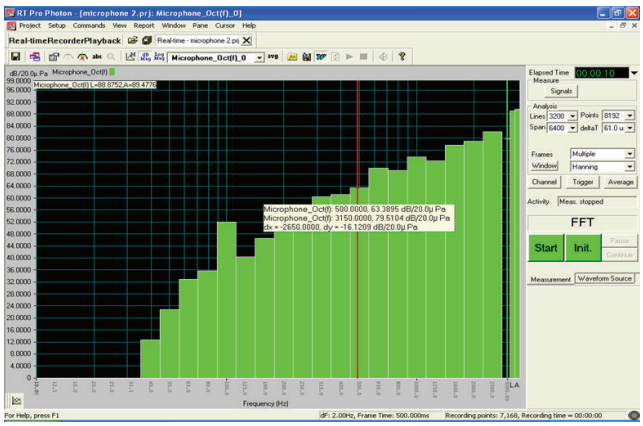


FIGURE 11: Variation of noise (dB) with frequency (Hz) at 1200 rpm for 6 Kg load with phasing.

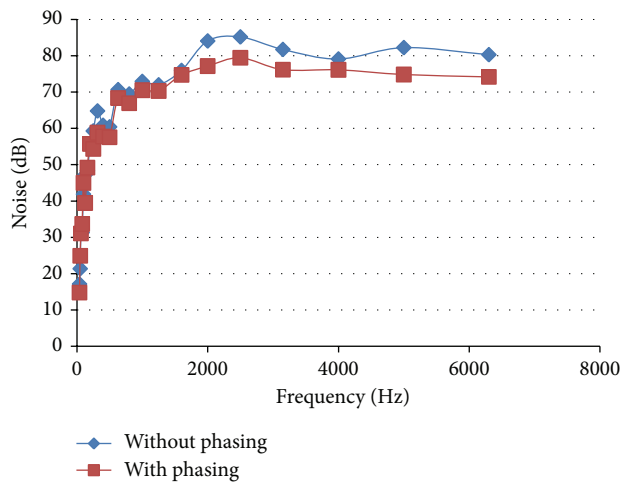


FIGURE 12: Variation of noise (dB) with frequency (Hz) at 1200 rpm 1 Kg load.

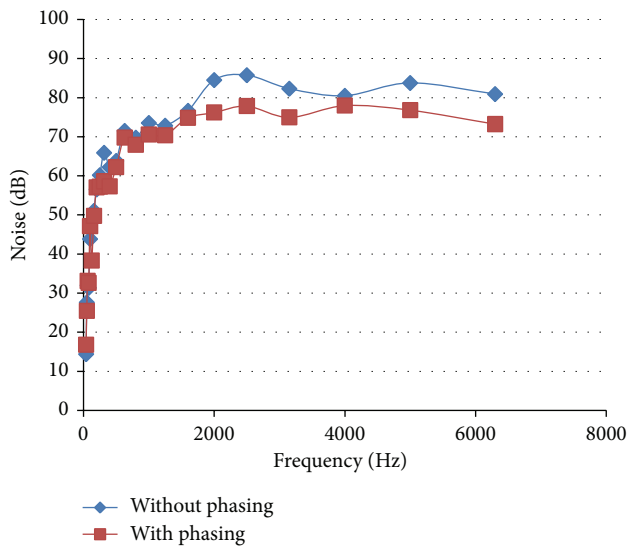


FIGURE 13: Variation of noise (dB) with frequency (Hz) at 1200 rpm at 2 kg load.

seen, while in with phasing arrangement noise measured found to be 76 dB. This indicates that the level of noise was reduced by 8.53 dB with phasing arrangement. For 3 kg load, from Figure 14 noise level with magnitude of 80 dB at frequency of 2000 Hz for without phasing arrangement is seen, while in with phasing arrangement noise measured found to be 76 dB. This indicates that the level of noise was reduced by 4 dB with phasing arrangement. For 4 kg load, from Figure 15 noise level with magnitude of 85 dB at frequency of 2000 Hz for without phasing arrangement is seen, while in with phasing arrangement noise level measured found to be 78 dB. This indicates that the level of noise was reduced by 7 dB with phasing arrangement. For 5 kg load, from Figure 16 noise level with magnitude of 84 dB at frequency of 2000 Hz for without phasing arrangement is seen, while in with phasing arrangement noise measured found to be 77 dB. This indicates that the level of noise

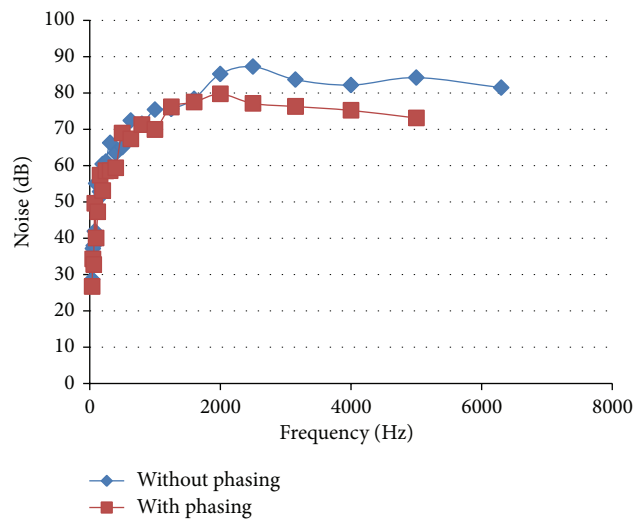


Figure 14: Variation of noise (dB) with frequency (Hz) at 1200 rpm at 3 kg load.

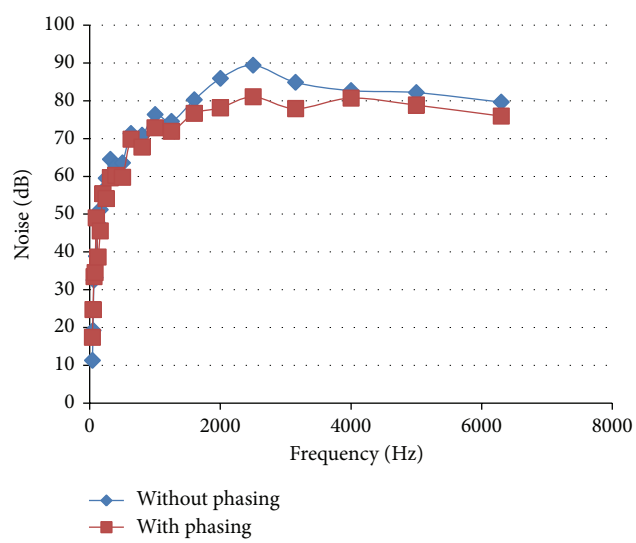


Figure 15: Variation of noise (dB) with frequency (Hz) at 1200 rpm at 4 kg load.

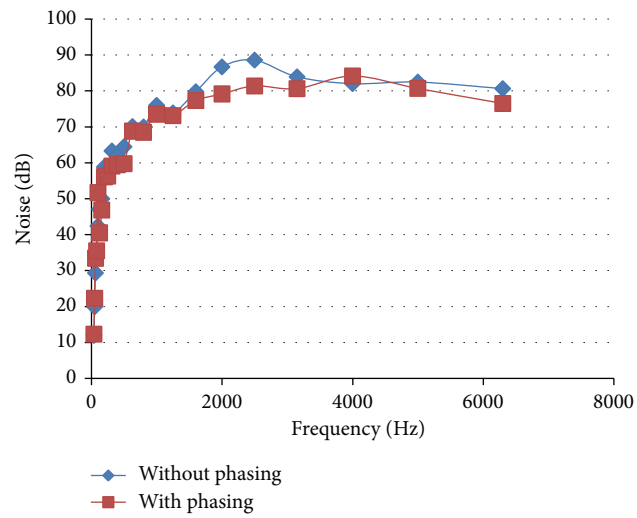


Figure 16: Variation of noise (dB) with frequency (Hz) at 1200 rpm at 5 kg load.

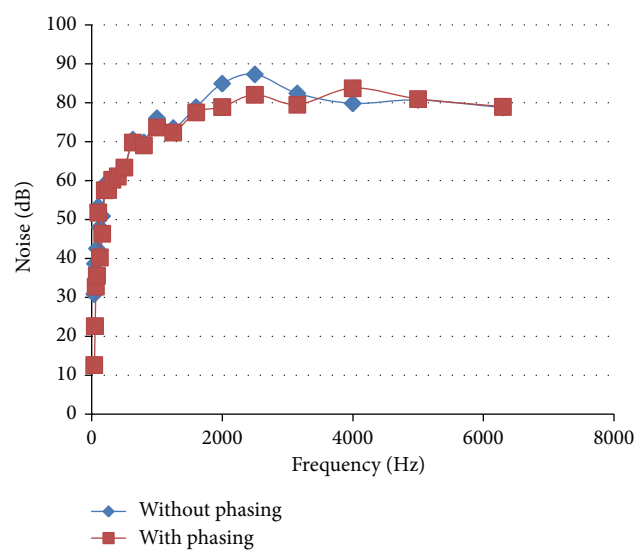


Figure 17: Variation of noise (dB) with frequency (Hz) at 1200 rpm at 6 kg load.

was reduced by 7 dB with phasing arrangement. For 6 kg load, from Figure 17 noise level with magnitude of 85 dB at frequency of 2000 Hz for without phasing arrangement is seen, while in with phasing arrangement noise measured found to be 79 dB. This indicates that the level of noise was reduced by 6 dB with phasing arrangement.

From these observations it is observed that the results obtained by sound level meter and FFT Analyzer are remarkably consistent with experimental measurements including excellent predictions of noise level as per literature survey.

## 7. Conclusion

The primary objective of this research work was to investigate noise reduction in planetary gear set by phasing. This objective was achieved with the help of extensive experimental investigations. Comparing results obtained by with and without phasing arrangement at 1200 rpm, it is observed that the level of noise in planetary gear set with meshing phase difference was approximately 6 dB to 7 dB lower than that of planetary gear set without a meshing phase. From the observations it is seen that the results obtained by sound level meter and FFT Analyzer are remarkably consistent with experimental measurements. So from the experimental results obtained by sound level meter and FFT Analyzer it is observed that by applying the meshing phase difference one can reduce planetary gear set noise and resulting vibrations.

## Conflict of Interests

The authors declare that there is no conflict of interests regarding the publication of this paper.

## References


[1] R. G. Schlegel and K. C. Mard, "Transmission noise control-approaches in helicopter design," in *Proceedings of the ASME Design Engineering Conference*, ASME paper 67-DE-58, New York, NY, USA, 1967.

[2] D. L. Seager, "Conditions for the neutralization of excitation by the teeth in Epicyclic Gearing," *Journal of Mechanical Engineering Science*, vol. 17, no. 5, pp. 293–298, 1975.

[3] W. E. Palmer and R. R. Fuehrer, "Noise control in planetary transmissions," SAE Technical Paper 770561, 1977.

[4] A. Kaharamam and G. W. Blankership, "Planet mesh phasing in Epicyclic gear sets," in *Proceedings of the International Gearing Conference*, pp. 99–104, Newcastle, Wash, USA, 1994.

[5] R. G. Parker, "Physical explanation for the effectiveness of planet phasing to suppress planetary gear vibration," *Journal of Sound and Vibration*, vol. 236, no. 4, pp. 561–573, 2000.

[6] C. Gill-Jeong, "Numerical study on reducing the vibration of spur gear pairs with phasing," *Journal of Sound and Vibration*, vol. 329, no. 19, pp. 3915–3927, 2010.

[7] Y. Chen and A. Ishibashi, "Investigation of noise and vibration of planetary gear drives," *Gear Technology*, vol. 23, no. 1, pp. 48–55, 2006.

[8] R. G. Parker, V. Agashe, and S. M. Vijayakar, "Dynamic response of a planetary gear system using a finite element/contact mechanics model," *Journal of Mechanical Design, Transactions of the ASME*, vol. 122, no. 3, pp. 304–310, 2000.

[9] P. Velex and L. Flamand, "Dynamic response of planetary trains to mesh parametric excitations," *Journal of Mechanical Design*, vol. 118, no. 1, pp. 7–14, 1996.

[10] R. J. Drago, "How to design quiet transmissions," *Machine Design*, vol. 52, no. 28, pp. 175–181, 1980.

[11] H. N. Özgüven and D. R. Houser, "Mathematical models used in gear dynamics—a review," *Journal of Sound and Vibration*, vol. 121, no. 3, pp. 383–411, 1988.

[12] W. Cheng, "Simulation of the stochastic vibration of spur gears," in *Proceedings of the 16th International Conference on Computer Aided Production Engineering*, Edinburgh, UK, November 1988.

[13] A. Kahraman and R. Singh, "Non-linear dynamics of a spur gear pair," *Journal of Sound and Vibration*, vol. 142, no. 1, pp. 49–75, 1990.

[14] B. Torby, *Spur-Gear Dynamics*, vol. 13 of *TRITA-MMK*, Royal Institute of Technology, Stockholm, Sweden, 1995.

[15] B. Campell, W. Stokes, G. Steyer, M. Clapper, R. Krishnaswami, and N. Gagnon, "Gear noise reduction of an automatic transmission trough finite element dynamic simulation," SAE Technical Paper 971966, 1966.

[16] K. Ariga, T. Abe, Y. Yokoyama, and Y. Enomoto, "Reduction of transaxle gear noise by gear train modification," SAE Technical Paper 922108, 1992.

[17] M. G. Donley, T. C. Lim, and G. C. Steyer, "Dynamic analysis of automotive gearing systems," SAE Technical Paper 920762, 1992.

[18] L. D. Mitchell and J. Daws, "A basic approach to gearbox noise prediction," SAE Technical Paper 821065, 1982.

[19] W. Hellinger, H. Raffel, and G. Rainer, "Numerical methods to calculate gear transmission noise," SAE Technical Paper 971965, 1965.

[20] I. Nurhadi, *Investigation of the influence of gear system parameters on noise generation [Ph.D. thesis]*, The University of Wisconsin, Madison, Wis, USA, 1985.

[21] J. Naas, H. Stoffels, and A. Troska, "Integrierte berechnung zur optimierung des verzahnungsgeräusches eines Pkw-handschaltgetriebes," *ATZ Automobiltechnische Zeitschrift*, vol. 103, no. 1, pp. 16–17, 2001.

[22] P. J. Sweeney, *Transmission error measurement and analysis [Ph.D. thesis]*, University of New South Wales, New South Wales, Australia, 1995.

[23] A. H. Middelton, "Noise testing of gearboxes and transmissions using low cost digital analysis and control techniques," SAE Technical Paper 861284, 1986.

[24] F. Gielisch and F. T. Heitmann, "Ermittlung der Geräuschanregungaus der Achsgetriebeverzahnungdurch Prüfstandsversuche," *ATZ Automobiltechnische Zeitschrift*, vol. 100, no. 4, pp. 282–286, 1998.

[25] F. B. Oswald, D. P. Townsend, M. J. Valco, R. H. Spencer, R. J. Drago, and J. W. Lenski Jr., "Influence of gear design on gearbox radiated noise," *Gear Technology*, vol. 15, no. 1, pp. 10–15, 1998.

[26] S. H. Gawande, S. N. Shaikh, R. N. Yerrawar, and K. A. Mahajan, "Noise level reduction in planetary gear set," *Journal of Mechanical Design & Vibration*, vol. 2, no. 3, pp. 60–62, 2014.