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Optimizing Helical Gear Profile for Decreasing Gearbox Noise

Reihaneh Mohammadkhani[1], Danial Nemati[2], Behzad Babaei[3]

[1]MS. Mechanical Engineering, R&D Department, NMI co, Qazvin, Iran.
[2]MS. Mechanical Engineering, Department of Mechanical Engineering, Imam Khomeini Technical College of Behshahr, Mazandaran, Iran.
[3]MS. Mechanical Engineering, R&D Department, NMI co, Qazvin, Iran.

ABSTRACT

Inspection of NVH is one of important factors in determining the quality of cars and necessary for noise control. Relation between design of gears and noise is focused in this paper. In this paper, new developments in the theory of gearing and modifications in gear geometry, which are necessary to improve the condition of meshing, are presented which will cause a low noise gears that have a stable contact during meshing. In fact a section of an involute profile is divided by 3 parts: 1- Tip of the tooth that the maximum corrosions and sliding are in this part. 2- Pitch line of tooth that the maximum hertz stresses are in this part. 3- Root of tooth that the maximum bending stress is in this part and also is so damageable against fatigue loading [10].

The noise starts of gears and will cause to twisting and bending vibrations in the shaft so different bearing forces create in bearing seat and lead to whine of gearbox cover [8]. After applying profile corrections, corrected gears are assembled on a gearbox and sensors including accelerometer and microphones are installed in predetermined sensitive positions such as gearbox cover, gear change knob and mountings. In the end, amplitude of vibrations (m/s^2) and intensity of noise (db) are compared before and after gear corrections. It is shown that according to graph results of NVH tests, the mean amplitude of noise is reduced by 20% to 40%.

KEY WORDS: noise, gear profile, transmission error, correction, vibration.

1- INTRODUCTION

Errors in gear alignment and manufacture may shift the bearing contact turn it into edge contact and cause transmission error that is the main source of vibration [16]. The purpose of this paper is to present developments in gear geometry and technology, directed at improving bearing contact and reducing transmission errors [17]. The main errors of alignment and manufacture are as fallows: The error of shaft angle, the shortest center distance, the leads in the case of helical gears, The errors in machine-tool setting (errors of orientation and location of the tool with respect to the gear being generated), and errors of circular pitches. In addition, it has to take into account the deflection of the teeth and shafts under load. To avoid or at least to reduce such defects, it becomes necessary to substitute the line contact of the gear tooth surfaces by the point contact and then, in addition, to modify the gear tooth surfaces. The modification of gear geometry is based on the proper deviation of the gear tooth surfaces from the theoretical ones [11]. The surface deviation can be provided in the longitudinal direction with the contact path in the profile direction (the direction across the tooth surface) and in the profile direction with the longitudinal direction of the contact path [5], [6] . In same cases both types of deviation must be provided simultaneously, but one of them must be the dominant. The desired modification of gear geometry becomes possible by applying inventive methods of gear technology such as the mismatch of tool surfaces for the generation of gears.

2-Modification of geometry of involute spur gear:

Spur gears are very sensitive to the misalignment of their axes, which causes an edge contact [12]. The localization of the contact can be achieved by plunging the grinding disk in the generation of the pinion by form-grinding. (fig.1)

The mating gear is generated as a conventional involute gear. The plunging means that during the pinion generation the distance between the axes of grinding disk and the pinion will satisfy the equation. (fig.1)

***Corresponding Author:** Reihaneh Mohammadkhani, MS. Mechanical Engineering, R&D Department, NMI co, Qazvin, Iran. reihaneh.mohmmadkhani@gmail.com

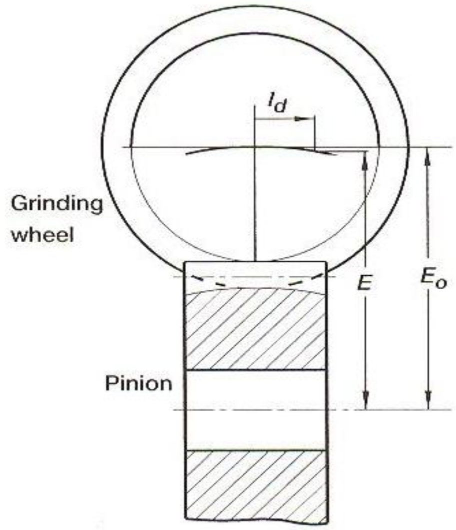

Fig.1. Form grinding of pinion by a plunging grinding wheel

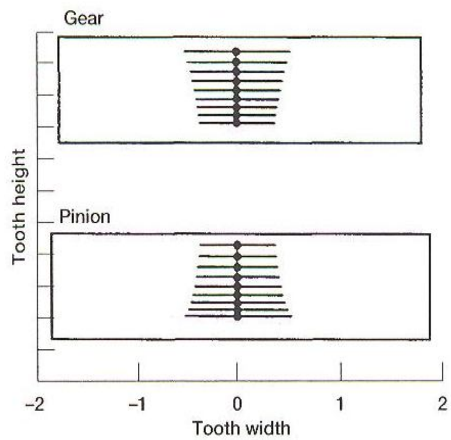

Fig.2. Path of contact and bearing contact of aligned gear drive with modified geometry

$E = Eo - ad.Ld^2$ **(1)**

Where E And Eo are the current and nominal values of the shortest distance, ad is the parabola coefficient of function E (Ld) and Ld is the direction of pinion axis. The localized bearing contact is shown in figure 2. The same results can be obtained by varying the shortest distance between the hob (grinding worm) and the generated spur or helical pinion [4], [15]. This variation is based on the application of an equation that is similar to equation (1)

3. Transformation and reduction of transmission errors.

The profile of pinion cross section for Ld=0 coincides with the axial profile of the disk. If the axial profile of the disk is designed as the involute profile of the conventional pinion the gear drive is still sensitive to misalignment $\Delta\alpha$ and the error of tooth distance which will cause an almost linear discontinuous function of the transmission on errors with the period of the meshing cycle
$ø = 2\pi / N1$ (fig.2)

Applying a predesigned parabolic function enables the absorption of the transmission error linear functions shown in fig 3. The redesign of the parabolic function of the transmission error is based on the following alternative approaches: 1- changing the curvature of the pinion or the gear 2- executing a nonlinear function that relates the rotational motion of gear (or the pinion) and the translation of the imaginary rock – cutter used to generate the gear (or the pinion.) Changing the curvature of the pinion can be achieved by substituting the tooth involute profile that corresponds to the theoretical base circle radius rbl by an involute profile of radius r'bl (fig4).

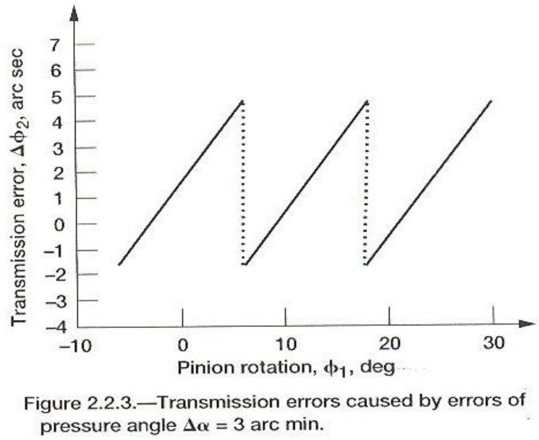

Fig.3. Transmission errors caused by errors of pressure angle $\Delta\alpha = 3$ arc min.

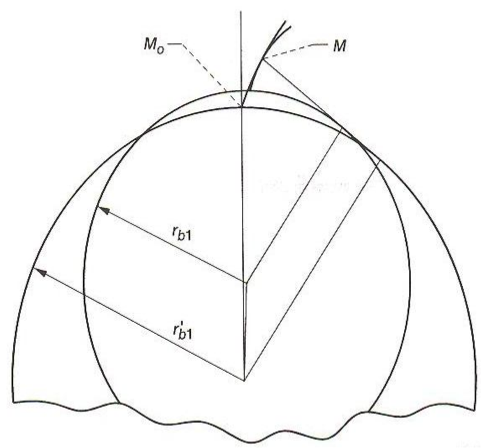

Fig.4. Theoretical and modified profiles of spur pinion.

4. Modification of involute helical gears (computerized investigation of misaligned conventional involute helical gears)

Aligned involute helical gears are in line contact at every instant as shown in figure 5. Misalignment caused by changes in the shaft angle, the lead and the normal profile angle of one of the mating gears causes edge contact instead of surface to surface tangency [9] , [7] . Edge contact means tangency of the edge of one gear with the tooth surface of the mating gear. The edge and the surface are in mesh at every instant at a point instead of on a line. An example of an edge contact caused by the change of shaft angle $\Delta\gamma$ or the change of the pinion lead angle $\Delta\lambda p1$ is shown in figure 6. The edge contact caused by $\Delta\gamma$ or $\Delta\lambda p1$ is also accompanied by transmission errors as shown in fig 6. However the change in the normal profile angle does not cause transmission errors, only an edge contact. In the case of a change in the center distance E, the gear tooth surfaces are still in a line contact similar to those shown in fig 5.

However, an error in ΔE causes a change in the backlash and in the pressure angle of the gear drive. [3]. There is a mistaken impression that a change in the lead is sufficient to shift the bearing contact from the edge to the central position and avoid transmission errors [14] . Our investigation shows that a combination of $\Delta\gamma$ and $\Delta\lambda p1$ errors will enable one to avoid transmission errors and obtain the favorable contact path shown in figure 8 if and only if :

$$\Delta\gamma = |\Delta\lambda p1| \qquad (2)$$

The signs of $\Delta\gamma$ and $\Delta\lambda p1$ depends on the hand of helix. Equation 2 must be observed with great precision because even a small difference between $\Delta\gamma$ and $|\Delta\lambda p1|$ (or $|\Delta\lambda p2|$) will cause an edge contact. Therefore, the change of the lead, if it is not accompanied by applying a predesigned parabolic function of transmission errors, is not a convenient way to avoid an edge contact. In conclusion it is emphasized that the computerized investigation of misaligned helical gears is a complex mathematical problem because the Jacobean of the system of equations that relate the surface parameters and parameters of motion is close to zero that is required for computerized investigation of a misaligned gear drive is a determination of the proper initial guess.

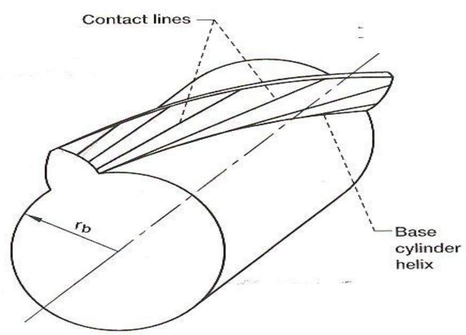

Fig.5. Contact lines on tooth surfaces of helical gear.

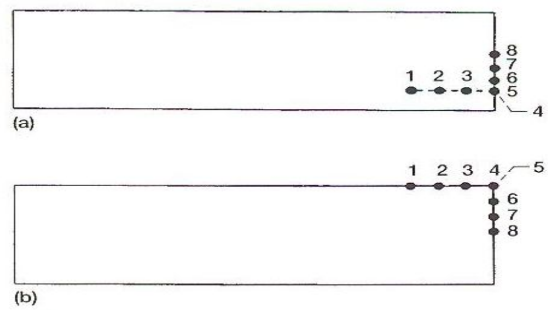

Fig.6. Edge contact caused by Δγ or Δλp1=3 arc min. (a) Pinion tooth surface (b) Gear tooth surface.

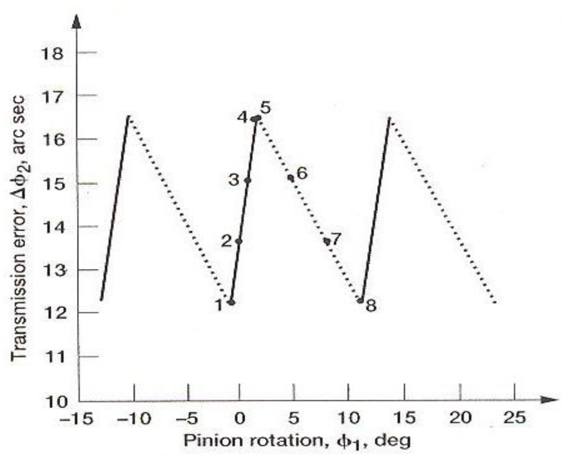

Fig.7. Function of transmission errors caused by Δγ or Δλp1= 3 arc min.

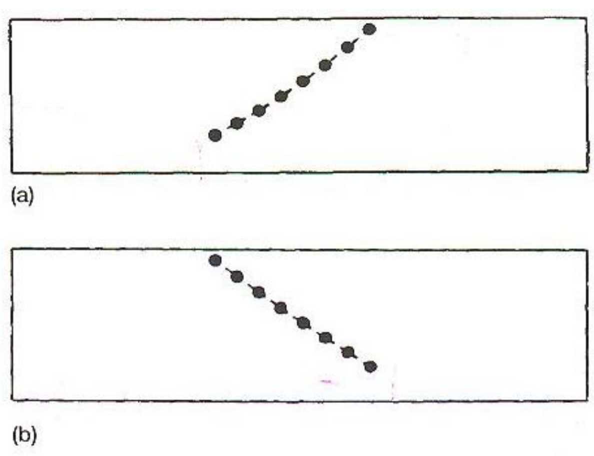

Fig.8. Path of contact caused by Δγ =3 arc min and Δλp1= -3 arc min. (a) Pinion tooth surface (b) Gear tooth surface.

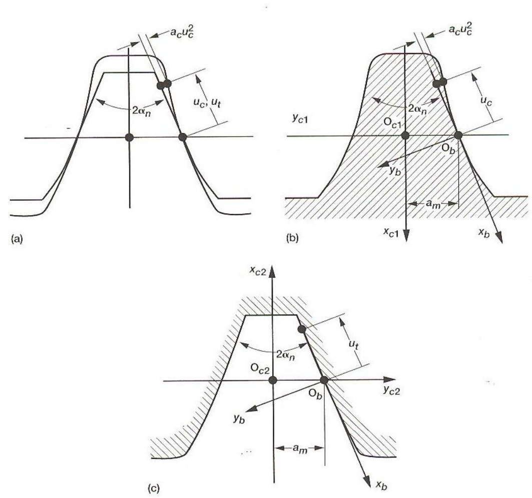

Fig.9. Normal section of rack-cutters. (a) Rack-cutters for gear and pinion generation (b) Pinion rack-cutter (c) Gear rack-cutter.

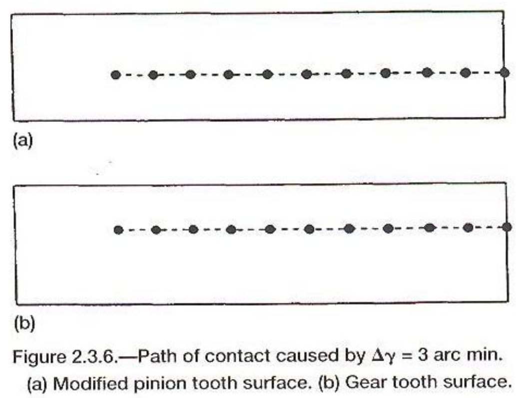

Fig.10. Path of contact caused by Δγ =3 arc min (a) Modified pinion tooth surface (b) Gear tooth surface.

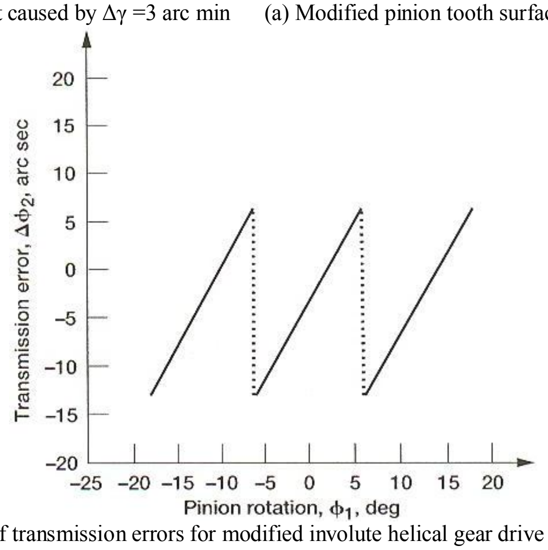

Fig.11. Function of transmission errors for modified involute helical gear drive when Δγ =3 arc min.

5. Profile modification

The sensitivity of helical gears to misalignment has led designers and manufacturers to crown tooth surfaces [1]. The most common gears (usually of the pinion) is deviated from the conventional involute profile. This can be achieved by the application of two imaginary rack-cutters shown in fig 9.

The rack - cutters are rigidly connected and generate, the pinion and the gear separately. Fig 9 (a) shows both rack - cutter that generates the pinion space is shown in fig 9 (b), and the normal section of the gear rack cutter that generates the gear tooth is shown in fig 9(c)

The deviation of the pinion rack - cutter profile from the gear rack-cutter profile is represented by a parabolic function with the parabola coefficient ac.

The tooth surfaces in the case of profile modification are in point contact and the path of contact is a helix as shown in fig (1).

It can be easily verified that the profile modification enables one to localize the bearing contact and to avoid an edge contact that might be caused by gear misalignment [13] . However, the discussed modification doesn't allow the elimination of transmission errors caused by misalignment as shown in fig 11. Therefore to reduce the level of noise and vibration it is necessary to provide predesigned parabolic function of transmission errors in addition to profile modification.

6. RESULT AND DISCUSSION

After applying modifications, corrected gears are assembled on the gearbox and sensors including accelerometer and microphones are installed in predetermined sensitive positions such as gearbox cover , gear change knob , and mountings as shown in figures 12,13,14,15 . The amplitude of vibrations (m/s^2) and the intensity of noise (db) are measured by sensors [2]. The results are compared before and after gear corrections. The test is done on the XU7 motor & BE-5N 17×77 gearbox in R&D center of Nirou Moharrekeh

Fig.12. The place of sensors(on the knob & steer)

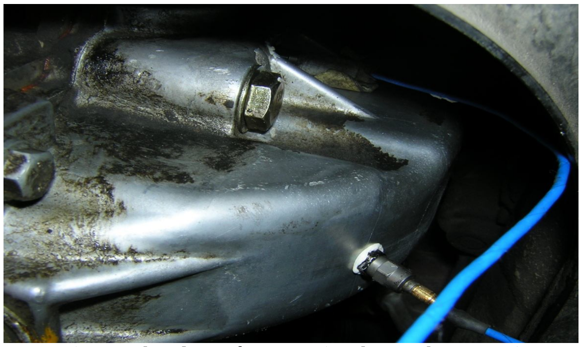

Fig.13. The place of sensors (on the gearbox cover)

Fig.14. The place of sensors (on the driver seat)

Fig.15. The place of microphone.

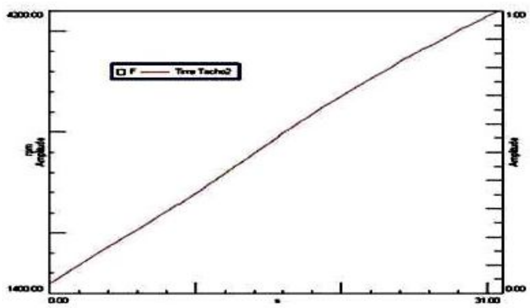

Fig.16. Increasing of motor speed in gear 5 during test.

Fig 16 shows the increasing of motor speed in gear 5 during test.

As is shown in fig 17 before modification, there are 2 waves of noise, one of them (green color , order 22.93) is for pinion - crownwheel in differential and another one (red color , order 47) is for gear 5 , both of them are at the same time .The sensors are installed on the gearbox cover and are registered the amplitude in 2 directions . After modification the test is done again and it is shown fig 18 the amplitude of vibration is decreased, both, for pinion - crownwheel & gear 5.

The intensity of noise is decreased as shown in fig 20. As is shown in fig 20, 21 the sensor is installed on the knob of gearbox & the test is repeated [18]. According the above tests the amplitude of vibration and the intensity of noise are decreased after gear modifications.

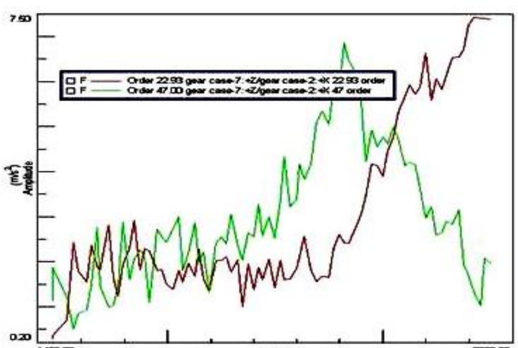

Fig.17. Amplitude of noise in gear 5 (gear 5 + pinion crownwheel) before gear modification in direction z on gearbox covering

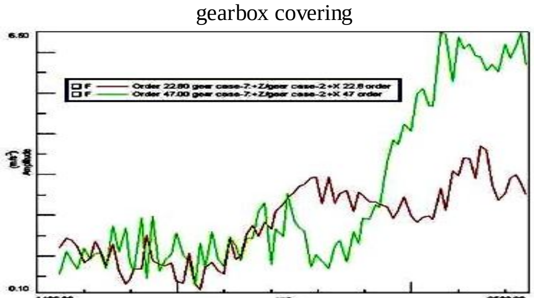

Fig.18. Amplitude of noise in gear 5 (gear 5 + pinion crownwheel) after gear modification in direction z on gearbox covering

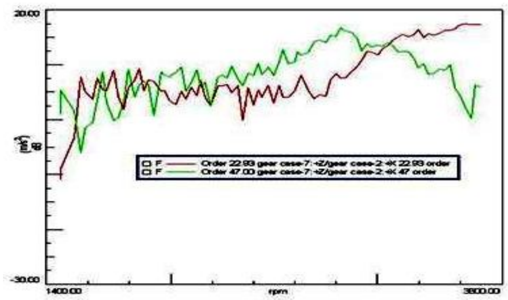

Fig.19. Intensity of noise in gear 5 (gear 5 + pinion crownwheel) before gear modification in direction z on gearbox covering

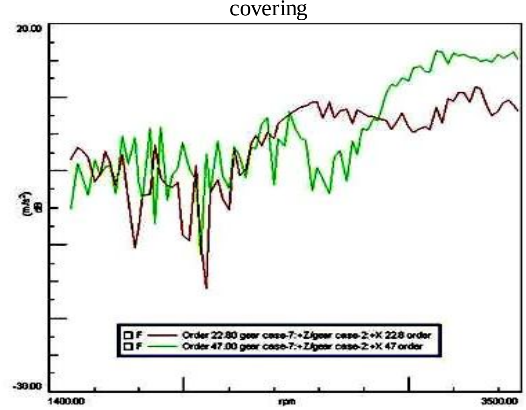

Fig.20. Intensity of noise in gear 5 (gear 5 + pinion crownwheel) after gear modification in direction z on gearbox covering

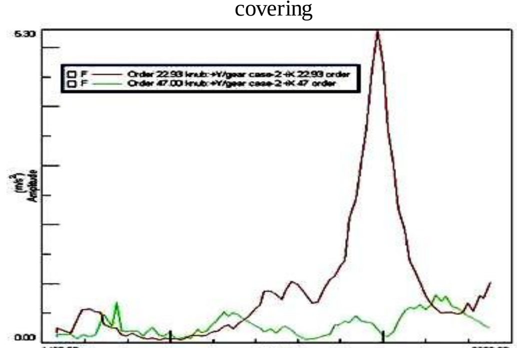

Fig.21. Amplitude of noise in gear 5 (gear 5 + pinion crownwheel) before gear modification in direction Y on the gearbox knob

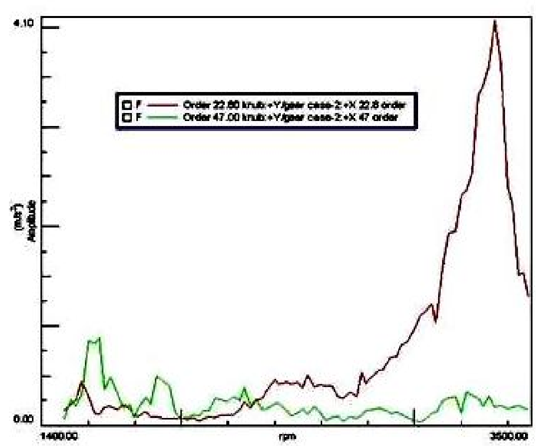

Fig.22. Amplitude of noise in gear 5 (gear 5 + pinion crownwheel) after gear modification in direction Y on the gearbox knob

7. Conclusion

NVH tests have been done for inspection of gearbox noise. It is done for gearbox in gear 5 and in real condition in road and sensors including accelerometer and microphone are installed in sensitive positions and amplitude of vibration and intensity of noise are compared before and after modification of gear profiles. Experimental results showed that by modifying the design of gears. The mean amplitude of noise is reduced by 20% to 40%.

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