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Investigation of Gear-Cutter Design Modifications on Decreasing Gearbox Noise

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ABSTRACT

Errors in gear & gear- cutter alignment and manufacture may lead to transmission error that is the main source of vibrations. The purpose of this paper is to present developments in gear cutter geometry & technology directed at improving contact & reducing transmission errors. Also a parabolic transmission error function that can handle misalignment during operation , is proposed. After applying modifications, corrected gears are assembled on gear box for NVH test ,that is one of the important factors in determining the quality of cars and necessary for noise control . Some sensors including accelerometers and microphones are installed in predetermined sensitive positions such as gearbox cover, gear change knob and mountings. In the end, the amplitude of vibration (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 test, the mean amplitude of noise is reduced by 35%.

KEY WORDS: noise, misalignment, transmission error, modification, gear-cutter.

1- INTRODUCTION

The purpose of this paper is presenting last modifications on gear-cutter geometry for decreasing transmission error. In this paper transmission error means the deviation between the theoretical angular position of gear-cutter and its actual position, when driving the input at a constant steady rotation.

In fact in this paper has been focused on gear cutter accuracies, tolerances, deviations from the theoretical profile and lead and pitch deviations of gear cutter.

The quality of gear cutter is the major influencing factor. The variation of the gear-cutter meshing stiffness during operation induces a deviation in the rotation-angle of the output gear from nominal transmission ratio and it will cause vibrations and noise.

In addition, some parameters may cause fluctuating in static transmission error that will cause in meshing gear pairs. Some solutions for it such as even contact between gear-cutter and gear is tested.

The evolution from linear to parabolic is discussed in terms of peak to peak transmission error.

All of these changing are consequences of tooth deflection, local contact deformation and body deformation which are the origin of transmission error.

2- The Even Contacts Method

The designing of shaving cutter is nowadays developed with the aid of computers using sophisticated soft wares based on even contact methods. In this method gear cutter and gear always have even numbers of contact. It is important to balance the forces involved. That is clear from fig 1(a) that force F pushes the cutter against the gear can be divided in 2 main components; on pushing on RH flanks and other pushing on LH ones. Each of two components is divided in 2 forces that operates always along the same line of action and that have an equal intensity. But in fig 1(b) on point 2' it is acting F2' the only one pushing against the tooth flank, thus contributing to bend it.

The effect of variation of the unbalanced forces is experientially proven, and it is tested according to fig 14-16.

If the even contacts condition is lost, in the majority of cases the gear profile obtaining is more or less as shown in fig (2).

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The even contacts situation is so desirable because it will balance the forces involved. According to fig. 2(a) force F that pushes the cutter against the gear can be divided in 2 main components : One pushing on RH flanks and the other pushing on LH ones .Each of two components is divided in 2 forces that operates always a long the same line of action and that have an equal intensity . Forces F_1 , F_2 , F_3 , F_4 acts then on point 1 , 2 , 3 , 4 in the same way and the cutting action in theses points will always be the same . But if in a determined moment of the rotation the number of points in contact is odd, like in fig. 1(b), then the situation can be schematized as fig. 2(b), It is seen that the module of vector $F3'$ is equal to the sum of vectors $F_1' + F_2'$ and in the point 3′ the cutting action will be different with reference to the situation of point 3 . In other words, the cutter pushes against point 3′ with a force double than the one exerted on points 3 & 4.

It is evident that in point 3 the metal removal will be higher and therefore we will have a profile irregularity in that point . The main reason , anyway , because we have to avoid the odd contacts situation it that we run the risk of a gear tooth distortion , due to the unbalanced forces that would tend to bend the tooth .The forces involved are quite high and that the expected variations of tooth aspect are too considered of Some microns In fig. 2(b) can be seen , first of all , that on point 2′ it is acting F_2' , the only one pushing against the tooth flank , thus contributing to bend it. But of point 3′ the force $F3'$ is considerably bigger than $F1'$, and also in this case forces are unbalanced with the risk of bending. The effect of variation of the unbalanced forces is experientially proven, if the even contacts condition is lost, in the majority of cases the gear profile is more or less as shown in fig No 3. In the central area, metal removal is bigger than required.

3. Transition From Surface Line Contact to Point Contact

Instantaneous line contact of gear tooth surfaces may exist only in ideal gear drives without misalignment and manufacturing errors. Such errors cause the gear tooth surfaces to contact each other at a point at every instant instead of on a line. The set of contact points on the gear tooth surface forms the contact path. A current point of the contact path indicates the location of the center of the instantaneous contact ellipse (Recall that because of the elastic deformation of the teeth, the contact is spread over an elliptical area.). The set of contact ellipses represents the bearing contact, which covers only a certain part of the tooth surface instead of the entire working tooth surface (in the case of the line contact of an ideal gear drive). Our goals are to determine the following: (1) the contact path for a misaligned gear drive and (2) the transmission errors caused by misalignment. Such problems are important for those gear drives whose gear tooth surfaces are designed as mutually enveloping. Typical examples are a worm – gear drive with a cylindrical worm and involute helical gears with parallel axes.

Figure 4(a) shows two neighboring contact lines L1 (ϕ) and L2 ($\phi + d\phi$) on surface $\sum_1$ of an ideal gear drive without misalignment. Surface $\sum_1$ is in instantaneous line contact with $\sum_2$; parameter ϕ is the generalized parameter of motion , and point M is a current point of contact line $L_1(\phi)$. The displacement from M to any point on L2 ($\phi + d\phi$) can be performed in any direction if it differs from the tangent to line $L_1(\phi)$ at M . However in the case of a misaligned gear drive with a point contact of surfaces $\sum_1$ and $\sum_2$, we have to determine the transition point P on the contact line $L_1(\phi)$ (fig 4(b)) (the transfer from line contact of the surfaces to point contact will occur in the neighborhood of P) and (2) the current point P^* of the real contact path. The direct determination of P^* is impossible because the Jacobean Δ_5 of the system of surface tangency is equal to zero . Therefore, it becomes necessary to determine an intermediate point K in the neighborhood of P (fig 4(b)). The determination of point K is based on the fact that vector PK is collinear to the vector that passes through two neighborhood transition points . The Jacobean Δ_5 at point K differs from zero , and we can start the procedure of simulating the meshing of two surfaces in point contact . Fig 5 shows the shift in the bearing contact in a misaligned worm – gear drive. Transmission errors due to misalignment will occur and may cause noise and vibration .

4. Design and Generation of Gear Drives with Compensated Transmission Errors (Influence of Transmission Errors on Condition for Transfer of Meshing)

Experimental tests show that the level of noise and vibration depends on the level and shape of transmission errors caused by gear misalignment . Henceforth , we will assume that the gear tooth surfaces are mismatched and that they contact each other at every instant at a point . This precondition is important when designing low noise gear drives , but it must be complemented with the requirement that one apply the predesigned parabolic function of transmission errors , which is represented as :

$$\Delta\phi_2(\phi_1) = -a\phi_1^2 \qquad \textbf{(1)}$$

It will be now shown that the application of such a function allows one to absorb transmission errors caused by gear misalignment , to avoid edge contact , and to improve the conditions for the transfer of meshing . Edge contact means curve-to-surface contact that may occur instead of surface-to-surface contact . In such a case , the curve is the edge of the gear tooth surface of one of the mating gears that is in mesh with the tooth surface of the mating gear . The transfer of meshing means that the continuous transformation of motions by a gear drive requires that a pair of teeth in mesh be changed for another pair. Fig 6(a) shows that the transmission function $\phi_2(\phi_1)$ for an ideal gear drive is linear and is represented as :

$\Phi_2(\Phi_1) = (N1/N2)\ \phi_1$ **(2)**

Where N_1 and N2 are the gear tooth numbers. The contact ratio (the number of teeth being in mesh simultaneously) may be larger than 1 in an ideal gear dive. In reality , ideal gear drives do not exist because alignment errors cause transmission errors that substantially worsen the conditions for the transfer of motion.

Fig 6(b) shows the transmission function $\phi_2(\phi_1)$ for a misaligned gear drive that is a piecewise nonlinear function for each cycle of meshing with worsened conditions for the transfer of meshing. The cycle of meshing is determined with the angles of rotation of the driving and driven gear represented as $\phi_1 = (2\pi/N_1)$ and $\phi_2 = (2\pi/N_2)$. They found that the function of transmission errors $\Delta\phi_2(\phi_1)$ for misaligned gear drives usually has the shape shown in Fig 7(a) .

The linear part of $\Delta\phi_2(\phi_1)$ is caused by gear misalignment ; the nonlinear dashed part of $\Delta\phi_2(\phi_1)$ corresponds to the portion of the meshing cycle when the edge contact occurs. The second derivative of $\Delta\phi_2(\phi_1)$, and therefore the acceleration of the driven gear , makes a big jump at the transfer point A of the meshing cycle.

The approach is directed at improving the conditions for the transfer of meshing and is based on the application of a predesigned parabolic function of transmission errors. Such a function is provided by the proper modification of gear tooth surfaces or by the stipulation of specific relations between the motions of the tool and the generating gear in the generation process. It will be shown , that the simultaneous action of both transmission error functions , the predesigned one and that caused by misalignment (in fig 7(a)) , causes a resulting function of transmission errors that is again a parabolic function having the same slope as the initially predesigned parabolic function. The magnitude $\Delta\phi_{2max}$ of the resulting maximal transmission errors (caused by the interaction of both functions shown in fig 7(b) can be substantially reduced. The level of the driven gear accelerations is reduced as well, and an edge contact, as a rule, can be avoided.

The transmission function for the gear drive, when the predesigned parabolic Function of transmission errors is provided, is shown in fig 8(a) . The predesigned parabolic function is shown in fig 8(b). It is important to recognize that the contact ratio for a misaligned gear drive with rigid teeth is equal to 1. However, the real contact ratio is larger than 1 because of the elastic deformation of the teeth. While investigating the correlation between the predesigned function of transmission errors and the elastic deformation of teeth, we have to consider the variation in the elastic deformation of the teeth during the meshing process, but not the whole value of the elastic deformation. It is assumed that the variation in elastic deformation is comparable to the level of compensated transmission errors.

5. Interaction of Parabolic and Linear Functions of Transmission Errors

Fig 9(a) shows the interaction of two functions: (1) the linear function $\Delta\phi_2^{(1)} = b\phi_1$ caused by gear misalignment and (2) the predesigned parabolic function $\Delta\phi_2^{(2)} = -a\phi_1^2$ provided by the modification of the contacting gear tooth surfaces. Our goal is to prove that the linear function $\Delta\phi_2^{(1)}(\phi_1)$ will be absorbed because of the existence of the parabolic function $\Delta\phi_2^{(2)} = -a\phi_1^2$. To prove it, we consider the resulting function of transmission errors to be:

$\Delta\phi_2 (\phi_1) = \Delta\phi_2^{(1)} (\phi_1) + \Delta\phi_2^{(2)} (\phi_1) = b\phi_1 - a\phi_1^2$ **(3)**

The proof is based on the consideration that equation (3) represents in a new coordinate system with axes $(\Delta\Psi_2, \Psi_1)$ (fig4 (a)) the parabolic function

$\Delta\Psi_2=-a\Psi_1^2$ **(4)**

The axes of coordinate systems $(\Delta\Psi_2, \Psi_1)$ and $(\Delta\phi_2, \phi_1)$ are parallel but their origins are different. The coordinate transformation between the coordinate system above is represented by

$\Delta\Psi_2 = \Delta\phi_2 - b^2/4a, \quad \Psi_1 = \phi_1 - b/2a$ **(5)**

Equations (3) and (5) , considered simultaneously , yield equation (4) . Thus, the linear function $\Delta\phi_2^{(1)}$ (ϕ_1) is indeed absorbed because of its interaction with the predesigned parabolic function $\Delta\phi_2^{(2)} (\phi_1)$. This statement is in agreement with the transformation of equations of second-order curves discussed in the mathematics literature.

The difference between the predesigned parabolic function $\Delta\phi_2^{(2)} (\phi_1)$ and the resulting parabolic function $\Delta\Psi_2(\Psi_1)$ is the location of points (A* , B*) in comparison with (A ,B). Fig 9(a) shows that the symmetrical location of (A, B) is turned into the asymmetrical location of (A*, B*). However, the interaction of several functions $\Delta\Psi_2 (\Psi_1)$, determined for several neighboring tooth surfaces, provides a symmetrical parabolic function of transmission errors $\Delta\Psi_2$ (Ψ_1) as shown in Fig 9(b). (The neighboring tooth surfaces enter into mesh in sequence.) The symmetrical shape of function $\Delta\Psi_2$ (Ψ_1) determined for several cycles of meshing can be achieved if the parabolic function $\Delta\phi 2^{(2)}$ (ϕ_1) is predesigned in the area (fig 9(a)).

$\phi_1(B)–\phi_2(A)\geq 2\pi/N1+b/a$ **(6)**

The requirement (6), if observed, provides a continuous symmetrical function $\Delta\Psi_2$ (Ψ_1) for the range of the meshing cycle $\phi_1 = 2\pi/N_1$.

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 10,11,12,13. The amplitude of vibrations (m/s ^{2}) and the intensity of noise (db) are measured by sensors. 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 14 shows the increasing of motor speed in gear 5 during test.

As is shown in fig 15 before modification, there are 2 waves of noise, one of them (red color, order 22.93) is for pinion - crown wheel in differential and another one (green color, order 47) is for gear 5, both of them are at the same time .The sensors are installed on the gearbox mounting and are registered the amplitude in 2 directions. After modification the test is done again and it is shown fig 16 the amplitude of vibration is decreased, both, for pinion - crown wheel & gear 5.

The intensity of noise is decreased as shown in fig 18. As is shown in fig 19, 20 the sensor is installed on the gearbox covering & the test is repeated. According the above tests the amplitude of vibration and the intensity of noise are decreased after gear modifications.

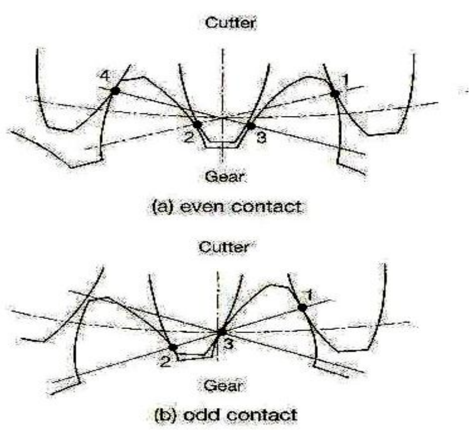

Fig. 1. Indication of the number of point of contact between gear and cutter

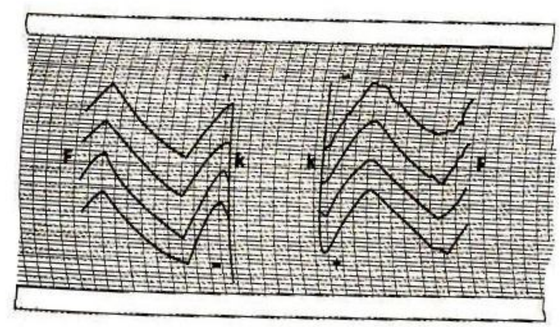

Fig. 3. Typical profile error generated by the non fulfilment of even contacts condition

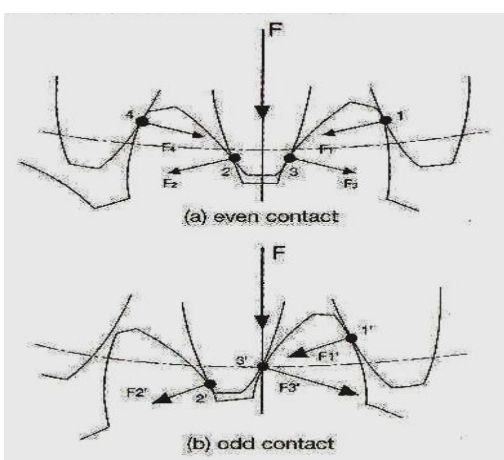

Fig. 2. Indication of the forces acting on each single point of contact

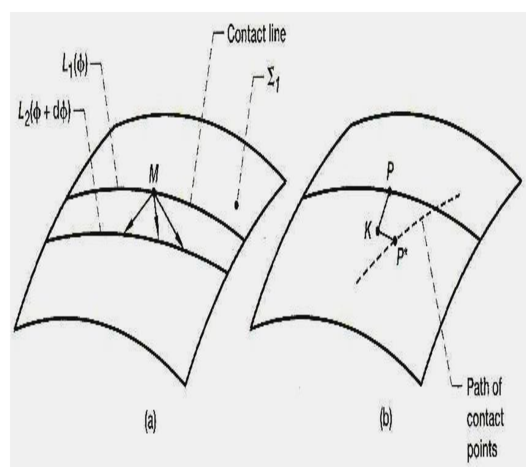

Fig. 4. For derivation of transition point. (a)Representation of two neighboring contact lines. (b) Transition from surface point P to P* via K.

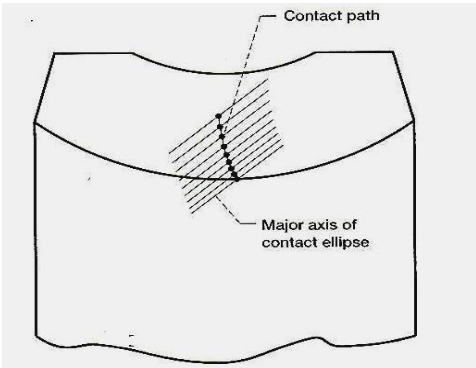

Fig. 5. Contact path of misalignment of worm-gear drive. Change of center distance, ΔE=0.5 mm; change of shaft angle, Δγ = 5'

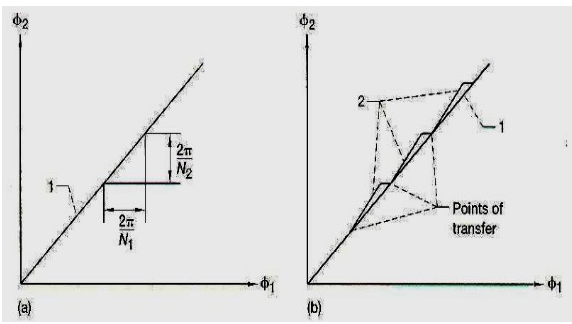

Fig. 6. Transmission functions of ideal and misaligned gear drives. (a) Ideal . (b) Misaligned.

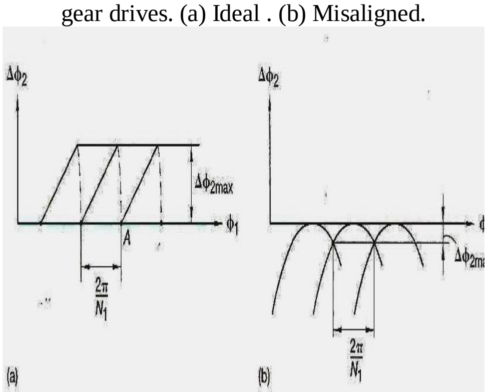

Fig. 7. Function of transmission errors for gears (a) Existing geometry (b) Modified geometry

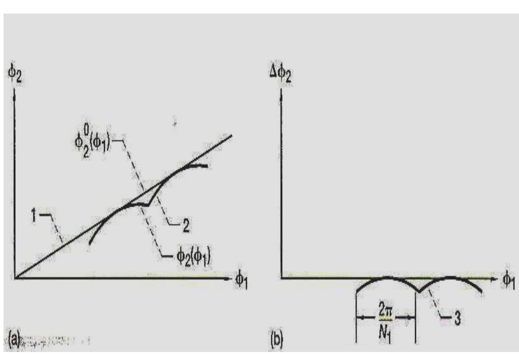

Fig. 8. Transmission function and function of transmission errors for misaligned gear drive. (a) 1,Ideal transmission function; 2, transmission function for gears with modified geometry, (b) 3, predesigned parabolic function of transmission errors.

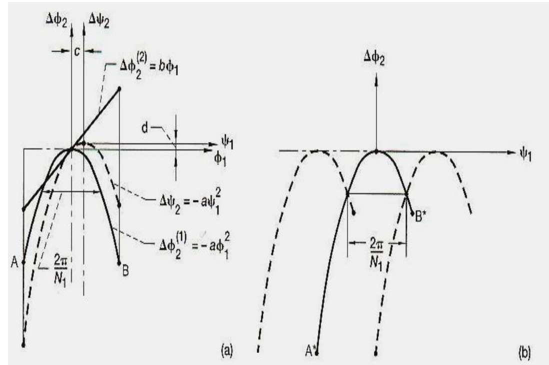

Fig. 9 . Interaction of parabolic and linear functions. (a) Linear and parabolic functions of transmission errors. (b)Resulting function of transmission errors.

Fig.10. The place of sensors (on the knob & steer)

Fig.11. The place of sensors (on the gearbox cover)

Fig.12. The place of sensors (on the driver seat)

Fig.13. The place of microphone.

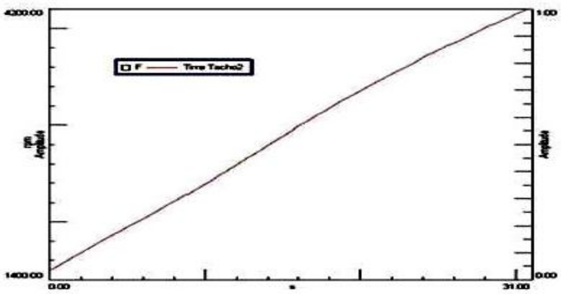

Fig.14. Increasing of motor speed in gear 5 during test.

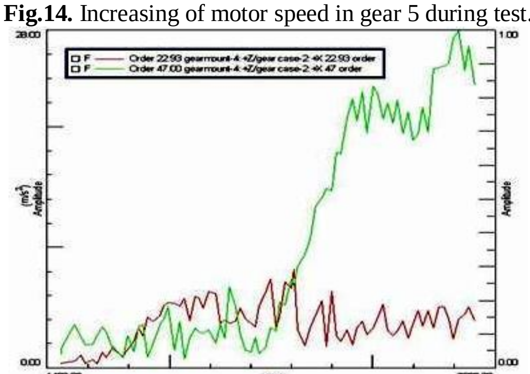

Fig.15. Amplitude of noise in gear 5 (gear 5 + pinion crownwheel) before gear modification in direction z on gearbox mounting.

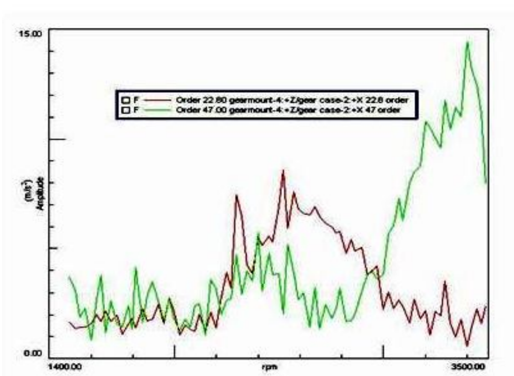

Fig.16. Amplitude of noise in gear 5 (gear 5 + pinion crownwheel) after gear modification in direction z on gearbox mounting

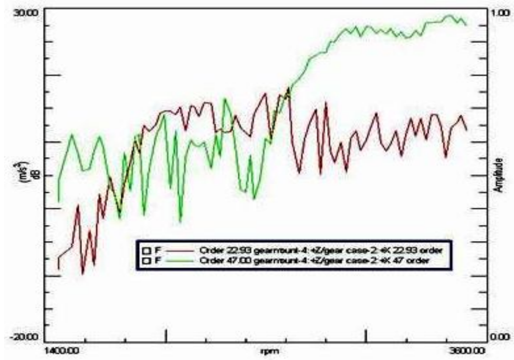

Fig.17. Intensity of noise in gear 5 (gear 5 + pinion crownwheel) before gear modification in direction z on gearbox mounting.

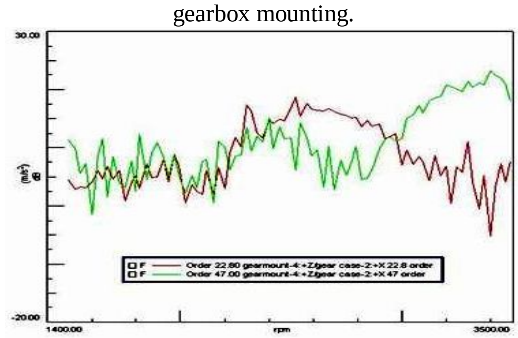

Fig.18. Intensity of noise in gear 5 (gear 5 + pinion crownwheel) after gear modification in direction z on gearbox mounting.

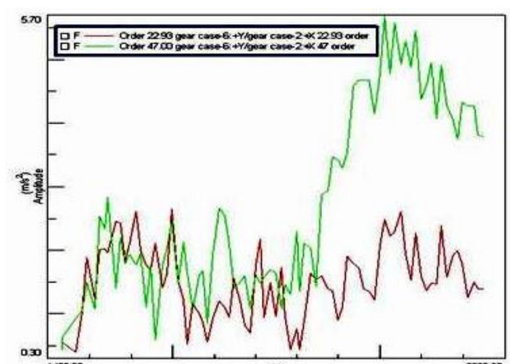

Fig. 19. Amplitude of noise in gear 5 (gear 5 + pinion crownwheel) before gear modification in direction Y on the gearbox covering

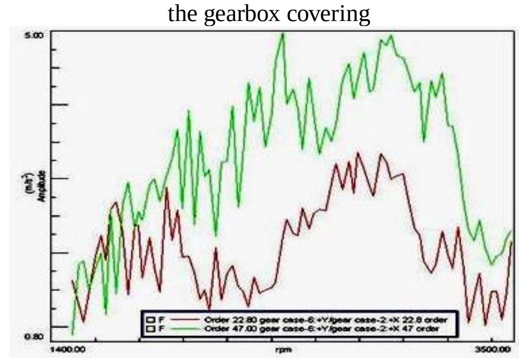

Fig.20. Amplitude of noise in gear 5 (gear 5 + pinion crownwheel) after gear modification in direction Y on the gearbox covering

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 about 35%

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