



A NEW DESIGN OF GEARBOXES WITH REDUCED VIBRATION AND NOISE LEVELS

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Summary

The paper gives an overview of previous trends in development of silent-running gearboxes and presents solutions enabling vibration and noise reduction. Advantages and disadvantages of these solutions are discussed. In addition, the paper describes a new original gear design which enables reducing gear vibration and noise without changing gear contact conditions.

Keywords: reduced vibration, reduced noise, vibration-damping element

KONSTRUKCJA PRZEKŁADNI ZĘBATYCH O OBNIŻONYM POZIOMIE DRGAŃ I HAŁASU

Streszczenie

W pracy przedstawiono dotychczasowe kierunki rozwoju konstrukcji cichobieżnych przekładni zębatych oraz szczegółowe rozwiązania, pozwalające na obniżenie ich drgań i hałasu. Wskazano na zalety i wady tych rozwiązań. Zaproponowano nową autorską konstrukcję, umożliwiającą tłumienie drgań i hałasu przekładni zębatych bez zmiany warunków współpracy kół zębatych.

Słowa kluczowe: redukcja drgań, redukcja hałasu, wibroizolator

1. INTRODUCTION

The operation of gearboxes is characterized by vibrations and noise, both of which are undesired residual processes. There are several reasons for reducing vibration and noise levels. First of all, the problem of vibration and noise is regulated under relevant regulations concerning noise at workplace. A value exceeding 85 dB is considered to be harmful. Gearboxes are very widely used in power transmission systems of hand tools and stationary machines. A study by Lutyński [1] reports the results of noise measurement at workplaces in a hard coal processing plant where gearboxes are widely used. Noise was exceeded for the majority of workplaces. Another reason is that consumers tend to evaluate product quality intuitively when hearing noise and experiencing vibrations. One example of this are cars which usually have gearboxes with toothed wheels and final drive, or household equipment. Aside from the nuisance resulting from servicing vibration and noise-emitting gearboxes and staying in their vicinity, there is another reason why vibroacoustic activity should be reduced. Vibrations result from dynamical forces which increase stresses in the entire gear, particularly in wheels, shafts and bearings. This generates an additional load leading to lower shorter life of these elements. Gear vibrations are also transmitted on other mechanisms of the drive system. Depending on a type of joint

between the objects, this activity can be harmful to a higher or lesser extent.

The research on causes of gear vibration activity and attempts at describing it in mathematical terms to enable predicting gear vibration has been conducted for decades. In the 1930s Henry Walker found that transmission error is the main source of gear noise [2]. This finding was also acknowledged by contemporary researchers [3, 4]. Having the knowledge about the causes of vibration and noise and their propagation path, engineers have taken action to reduce these undesired residual processes. This study presents a new gear design enabling vibration and noise reduction. The solution has been applied for patent with the Patent Office of the Republic of Poland.

2. SOURCES OF GEAR NOISE AND VIBRATION

The main source of gear vibrations and noise are toothed wheels themselves. This results from wheel tooth geometry and torque transmission. The greatest amount of energy dissipates during tooth contact when wheel teeth collide due to factors which can be classified as constructional, technological and operational. The constructional factors include variable stiffness of wheel teeth resulting from a different number of tooth pairs in tooth contact as well as changes in tooth profile such

as profile modification at the tooth tip (Fig. 1) and profile crowning (Fig. 2).

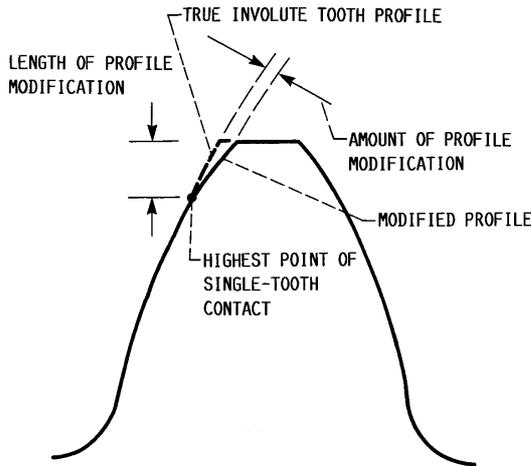


Fig. 1. Profile modification at tooth tip [5]

The technological factors include deviations of wheel tooth geometry and position from theoretical values. What has the most significant effect are errors concerning tooth pitch, profile, helix, thickness and run-out. The magnitude of profile distortion results from, among others, load and speed of wheels; in other words, from operational conditions. In addition, there occur thermal distortions and vibrations caused by lubrication and variable load (it can be generated by both the engine and the device being driven).

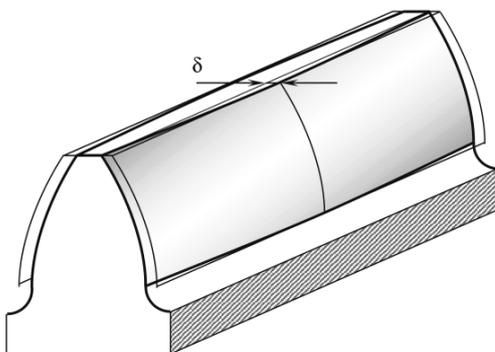


Fig. 2 Profile crowning [6]

It must be emphasized that the above-mentioned tooth profile modifications are applied in order to reduce gear vibrations and noise. In theory, the involute profile (most widely used) enables the work of a constant ratio gear. However, due to finite tooth stiffness and production errors, the constant ratio cannot be obtained. For this reason, the tooth geometry is changed on purpose to enable more silent gear work.

The effect of the above factors on vibrations and noise level can be expressed quantitatively by means of one measure described as a transmission error (kinematic deviation). The transmission error TE is defined as a difference between the real and

theoretical position of the driven wheel:

$$TE = \theta_2 - \frac{z_1}{z_2} \theta_1 \quad (1)$$

where:

θ_1 is the angular position of the pinion,

θ_2 is the angular position of the wheel,

z_1 is the number of pinion teeth,

z_2 is the number of wheel teeth.

The results demonstrate that the increase in transmission error is accompanied by an increase in noise [2, 4].

Vibrations are also generated by other moving elements of a gear. In simple design mechanisms, these parts include shafts and rolling bearings. The shaft generates vibrations resulting from unbalance. There can also occur parametric vibrations if the shaft has a variable stiffness, which is the case with joints, particularly a key joint. Rolling bearings also generate vibrations mainly due to their variable stiffness caused by the shift of rolling elements. Fig. 3 illustrates the internal causes of vibration excitation in gearboxes and a vibration propagation path.

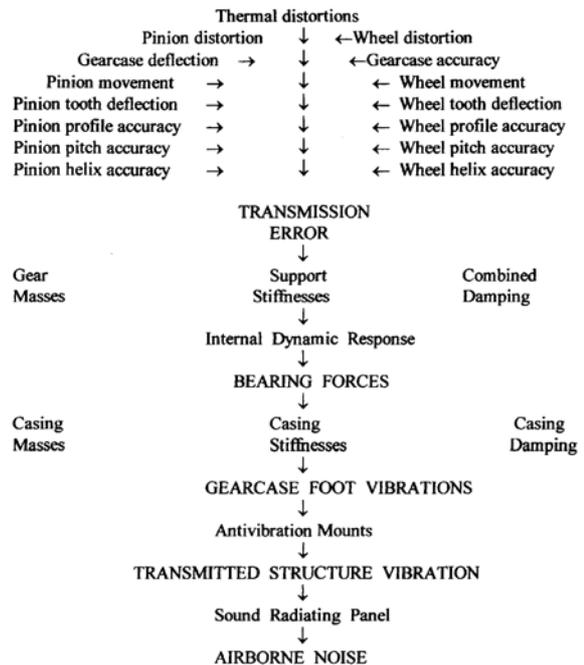


Fig. 3. Sources of vibration excitation in gearboxes and vibration propagation path [4]

Vibrations and noise can also increase due to external factors such as a change in load, assembly errors and other variable external factors.

3. METHODS FOR VIBRATION AND NOISE REDUCTION IN GEARBOXES

Whether gear work exhibits a satisfactory level of vibration and noise or not predominantly depends on its design. If errors are made at this stage, it will

be difficult to obtain a good final result by applying additional activities. The excessive vibration activity of gearboxes is predominantly blamed on manufacturing defects or - in many cases - poor gear design [4].

There are several methods for reducing noise and vibrations in gearboxes. Smith [4] divided these solutions into four groups:

- 1) solutions which decrease dynamic forces generated due to tooth contact,
- 2) solutions which damp vibrations on the way from their source (gear wheels) to the noise emitting element (e.g. casings) by the application of vibration-damping elements or by changing properties of the noise-emitting element,
- 3) solutions which consist in absorbing generated noise or putting the entire device in a soundproof casing,
- 4) solutions which employ an active noise reduction method or actuators in order to increase effectiveness of vibration-damping elements.

The first group comprises solutions aimed at reducing transmission error. They mainly focus on wheel teeth with respect to:

- Profile modification. Under load the wheel tooth undergoes elastic deformation; as a result, in tooth contact the addendum flank collides with the dedendum flank of another tooth. In addition, manufacturing and assembly errors add up to this phenomenon. To remove the effect of manufacturing defects and finite stiffness of wheel teeth, involute profile modifications are applied. In most cases, when the pinion is the only element to be modified, the material is removed from the dedendum flank and addendum flank. If, however, both toothed wheels undergo modification, it is the involute of the addendum flank that gets modified. This method is the most effective for spur gears and a specified load range.
- Longitudinal profile modification – crowning. With a spur gear, we obtain a curved tooth profile (Fig. 2). As a consequence, tooth contact takes place in the central region of the tooth despite the incorrect position of the axis of toothed wheels due to production errors and deformation.
- Higher production accuracy. Decreasing production errors with respect to tooth pitch, profile, roughness etc. leads to smaller gear vibration and noise. The result greatly depends on initial accuracy.
- Increasing contact ratio. The operation of gearboxes often involves contact between, first, one pair of teeth and, then, two pairs of teeth. This leads to variable stiffness of teeth causing vibrations. Two pairs of teeth in contact cause much smaller tooth deformations. For this reason, it is good to ensure constant contact between two pairs of teeth. Nonetheless, such a high contact ratio is often impossible to obtain due to different reasons.

- Avoiding mesh frequency ranging from 500 Hz to 1000 Hz. In this frequency range human ear is the most sensitive to noise; there are also many other loud structures within this range [4].
- Straight teeth are changed to curved teeth. As a result, gear work is more silent because both the contact ratio and the tooth contact increase. At the same time, the tooth does not work over all its entire width; it gradually enters and leaves tooth contact.

The second group includes solutions predominantly involving changes in toothed wheels and gear casing. There are different designs of toothed wheels which are based on the use of vibration-damping elements. They are located between a toothed ring and a wheel hub. Examples of these solutions are shown in Fig. 4.

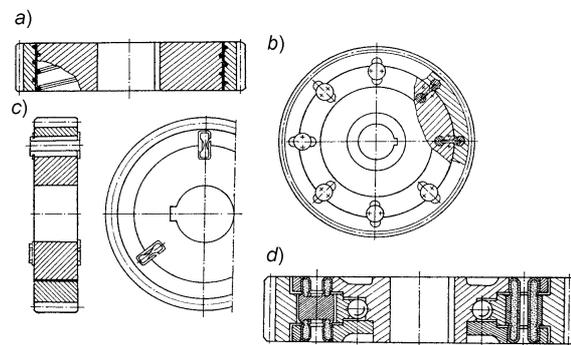


Fig. 4. Examples of the solutions for gearboxes with vibration-damping elements [7]

As far as the problem of gear casing is concerned, the studies mainly focus on changing casing stiffness and developing its optimal shape. It can be observed that an increase in stiffness leads to an increase in resonance frequency and to a decrease in the amplitude of vibration. The simplest way to increase the stiffness of a casing is to increase wall thickness. This solution is not optimal because it leads to a much higher increase in mass of the entire gear. A more sophisticated approach is to design suitable ribbing. Such studies are described in the works [8, 9]. Their authors investigated the effect of varying rib spacing and different rib thickness and height on vibration activity of gearboxes. A different approach was presented in the study [10]. Here, the gear casing was made of plates. The shape of the plates was optimized such to reduce vibrations while maintaining their mass constant (Fig. 5).

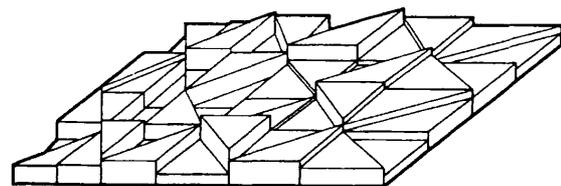


Fig. 5. Optimal shape of the upper plate [10]

With flat casing plates it is more practical to add a rib in the plate's centre in order to change the mode shape of vibration such that the vibrations undergo a phase shift of 180° . This will lead to the desired noise reduction [4] (Fig. 6).

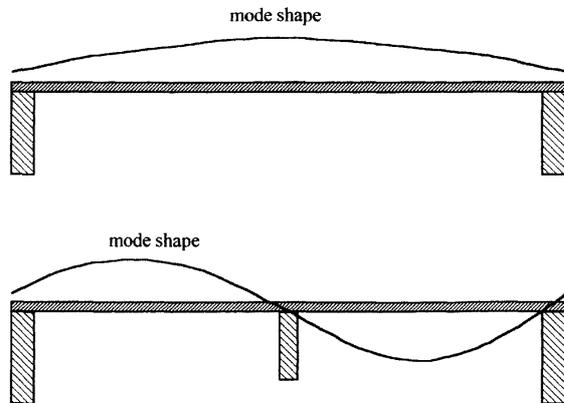


Fig. 6. Changing the mode shape of vibrations by adding a rib to the plate [4]

Another interesting yet hardly effective attempt involved placing a material characterized by higher damping properties than steel (e.g. synthetic granite) in the places of contact between gear casings [4].

The third group comprises methods based on the use of damping casings of entire devices and acoustic screens to reduce noise. One example of this solution is a German patent DE no. 3217180. In this invention, the protective part consists of many coatings which are mounted on the main part of the body by means of elastic clamping elements (Fig. 7).

The last group contains active methods. These methods are more seldom used and they complement the above-mentioned passive methods. The idea of

an active noise protection method is based on creating a copy of undesired sound, processing it and emitting in order to remove noise [12]. On the other hand, vibrations are reduced by means of supports or other actuators. An interesting example of this is the solution implemented to improve acoustic comfort of passengers on a S-76 helicopter [13]. The main problem involved here vibrations generated by gear wheels, at a frequency exceeding 700 Hz. The actuators were used to exert point forces located in the vicinity of the gear mounting (Fig. 8). The objective was to reduce vibrations generated by teeth in mesh before their spread to the helicopter's frame. As a result, the noise generated by the mesh frequency ranging from 10 to 20 dB was reduced.

The methods in first group are widely used because they are practically free from any shortcomings. They enable reducing dynamic forces generated in teeth mesh, which leads to reduced vibrations and noise as well as to reduced load on gear elements. The changes mainly pertain to gear teeth. As for the second group methods, the ones that most often used in standard gearboxes involve optimization of shape and body stiffness. Vibration and noise-damping casings of gearboxes (third group of methods) or of entire devices containing a gear are seldom to be found. Solutions of this kind have two basic disadvantages: they are expensive to implement and take too much space. In addition, they can decrease airflow, which leads to impaired cooling; they are rather impractical and "fragile." The fourth group of methods is still being developed. So far no standard has been established with respect to gearboxes. These methods are typically applied as active hearing protectors and noise damping actuators in installations which use gas as a working medium.

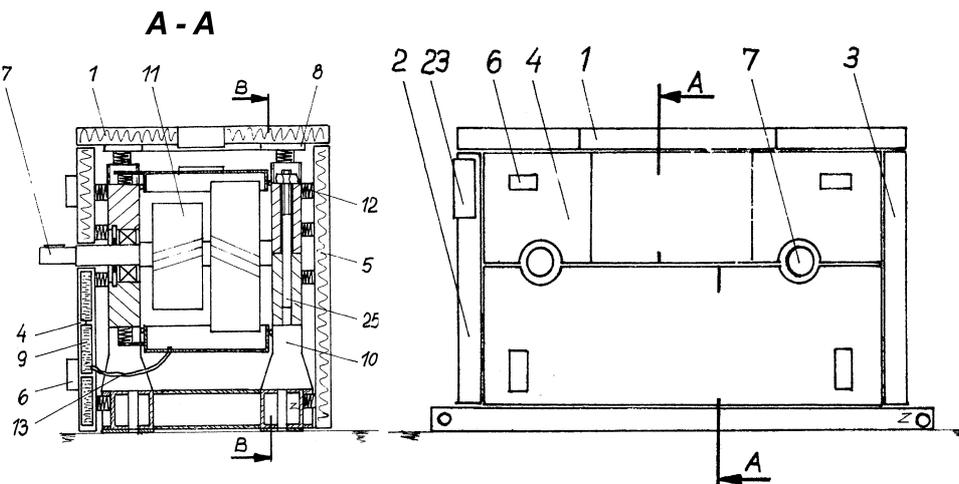


Fig. 7. Sound-damping gear casing [11]

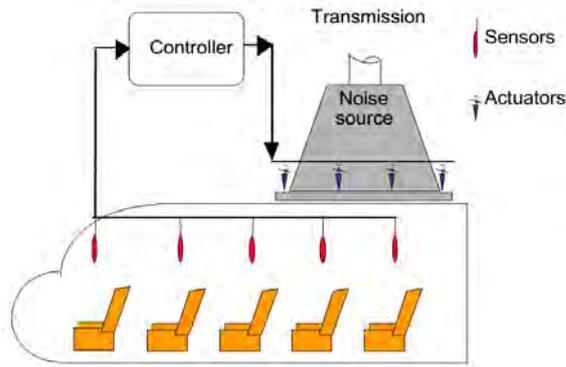


Fig. 8. Schematic design of active reduction of noise resulting from a gear mesh [13]

4. NEW SOLUTION OF A GEAR WITH REDUCED VIBROACOUSTIC ACTIVITY

The proposed solution belongs to the second group of methods and it consists in reducing vibrations on a path from gears 2, 3 to casing 6 (Fig. 9) [14]. One additional element is applied, namely - yoke 5 in which shafts 1 and 4 are mounted. Metal and rubber tube-shaped vibration-damping elements 7 are mounted on the yoke 5 between the bearings 8 and the casing 6. The role of the yoke is to ensure that the axis of the shaft is maintained in a constant position such that tooth contact conditions are maintained the same as in a standard gear.

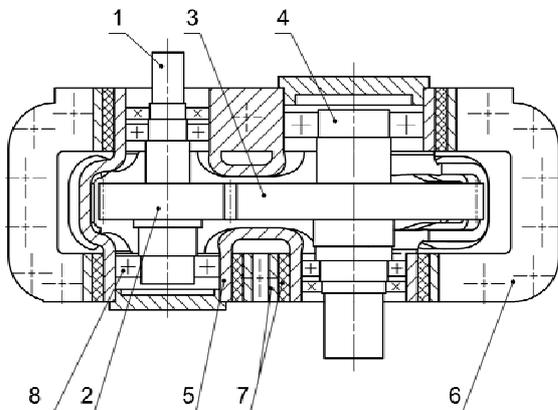


Fig. 9. A design of a gearbox with reduced vibrations and noise [14]

The yoke can be an indivisible part with vibration-damping elements (Fig. 10a) or vibration-damping elements can be assembled separately (Fig. 10b).

5. SUMMARY

This paper presented the causes of gear vibration and noise, vibration propagation path and different possibilities of reducing vibroacoustic activity in gearboxes. The methods in the four discussed groups are often used together in different combinations.

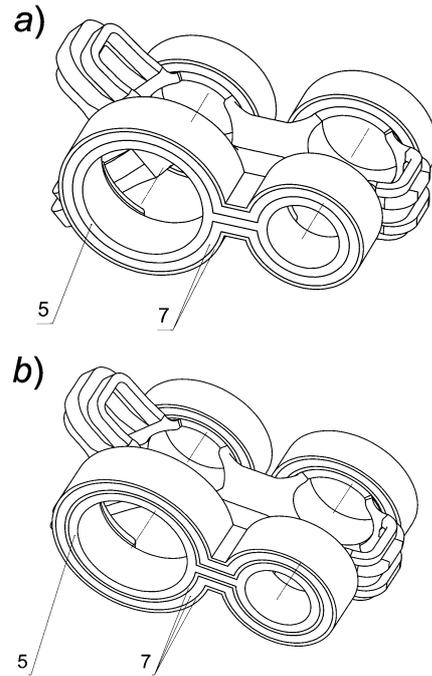


Fig. 10. Two exemplary variants of producing a yoke: a) an indivisible part, b) a part with vibration-damping elements assembled separately [14]

The solution proposed by the author of this paper is based on the use of a yoke equipped with elements for reducing vibrations generated by the gear's moving parts (mainly toothed wheels). Like any other solution, the proposed method has its advantages and disadvantages. Given the fact that the method has not been experimentally tested yet, the significance of its individual positive and negative features cannot be assessed. The potential advantage of the presented method is that it has no effect on teeth contact conditions and that it can be used to dampen vibrations generated by gears, shafts and bearings. The disadvantages include the necessity of using clutches with angular, radial and axial compensation, a higher total cost of the gear as well as higher gear mass and dimensions.

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