

Noise Reduction in Novel Transmission with 3D Point Contact Gear System

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ABSTRACT

Increasing challenges on reducing fuel consumption has opened the new directions in Powertrain technologies. The example of such technology implemented in geared transmission is the usage of novel type of gears with 3D point system of mesh engaging which was invented, patented and developed by Alexei P. Popov. The paper identifies the milestones of R&D that were undertaken to create the new type of geared transmission. Using the Winkler hypothesis Alexei P. Popov has developed and designed new type of point contact which made possible to lower a contact stresses in comparison to the values of a contact stresses for teeth having linear contacting interaction. The new shapes of tooth were invented and the methodological tools were developed for engineers providing the formulas for design and forming geometry of tooth with high stress capacity. The patented profiles would be applied for a design of all kinds of transmissions successfully replacing the conventional vehicle gear transmissions. The results of theoretical analysis combined with experiments on prototypes illustrate the proof of developed mathematical modeling. The synthesis of new type of gearing is presented. The test results for transmission of extra heavy vehicle and for transmission of light truck are discussed. The 25% - 40% reduction of sizes and weights, and reduced noise levels up to 22 dBA are the advantages of transmissions based on invented gears having 3D point system of engaging.

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INTRODUCTION

An engine's geared transmission is loaded by external and internal forces during operation. The purpose of geared transmission is to deliver engine torque to vehicle wheels within the narrow proper range of engine's RPM. So the transmission uses gears to make the engine's torque more effective, and to keep the engine operating at an appropriate speed. Traditionally gears are designed in an assumption that gear teeth are subjected to Hertzian contact compressive stresses, deformations and elasto-hydrodynamic lubrication.

The main factors impacting on the character and intensity of teeth wear are the contact compressive stresses and deformations. Thin under-surface layers of tooth are the most stressed areas of gear material. The big value of stress gradients are monitored in these layers. The main aspects of Hertz assumptions [1], [2], [3] for contact problems would be 1. a surface of contacting bodies are perfectly smooth, isotropic, and homogeneous, 2. the deformations are elastic only, 3. a squeezing press force is normal to contact spot area, 4. the dimensions of contact spot area are very small in comparison to characteristic sizes of contacting bodies. The Hertz theory was improved by hundreds of scientists and engineers during last 100 years, see for instance [6], [7], [8], [9], [10], [11], [18], [19], [20], [21], [22], [23], [27] - [28]. Some of the researches deviated from Hertz theory and took for base of their calculations the Non-Hertzian

contact approach for explanation of new phenomena in contacting objects. Such approach could be found, for instance, in [12], [13], [14] dedicated to Non-Hertzian contact of elastic spheres. Professor Alexei P. Popov using the Winkler hypothesis [15], [16], [17] had found [4], [5] that point contact interaction of teeth could be characterized by certain values of a contact stresses which are significantly lower than the values of a contact stresses for teeth having linear contacting touch. This invention provides the opportunity for increasing of the tooth load capacity, reduction of sizes and weights, and reduction of noise and vibration. The measured numbers of overall noise reduction in tested prototypes of the truck transmissions is up to 22 dBA.

NEW MODELING APPROACH

In present times the Hertz formula (with dozens of correcting coefficients) is developed for calculations of a linear contact phenomenon in the gearings with a point contact such as hypoid gear transmissions, spiral bevel gearing transmissions and Novikov's transmissions. The Hertz approach does not reflect the physical idea of stressed tooth condition. In reality the engineer is facing the volumetric 3D stressed operation during contact interaction of teeth for such type of gearing. This lack of conformity to experimental values and the disharmony in explanation of test results regarding values of maximum strengths of materials, the research efforts for

construction of new theory of contact strength and invention of new profiles of teeth. The foundation of proof is composed on new theoretical and experimental studies of point contact in statics and dynamics on prototypes. New approach is proposed by Professor Alexei P. Popov. The developed theory [4], [5] is based on obtaining of two equally weighted functions of the contact deformations. First function of the contact deformations is determined for contacting objects taking into consideration their shapes and changes in shape **before and after** load. The Winkler hypothesis is used for determination of second function. The Winkler hypothesis was improved in [4], [5] and the area of its application was expanded and enlarged, specifically for solution of 3D contacting systems. Proposed approach [4], [5] introduces the cause - effect relationship between the function of a contact deformations and function of a contact stresses. A function of contact deformations is the cause or principle. It is reflecting the function of a contact stresses, which is the consequent result or outcome.

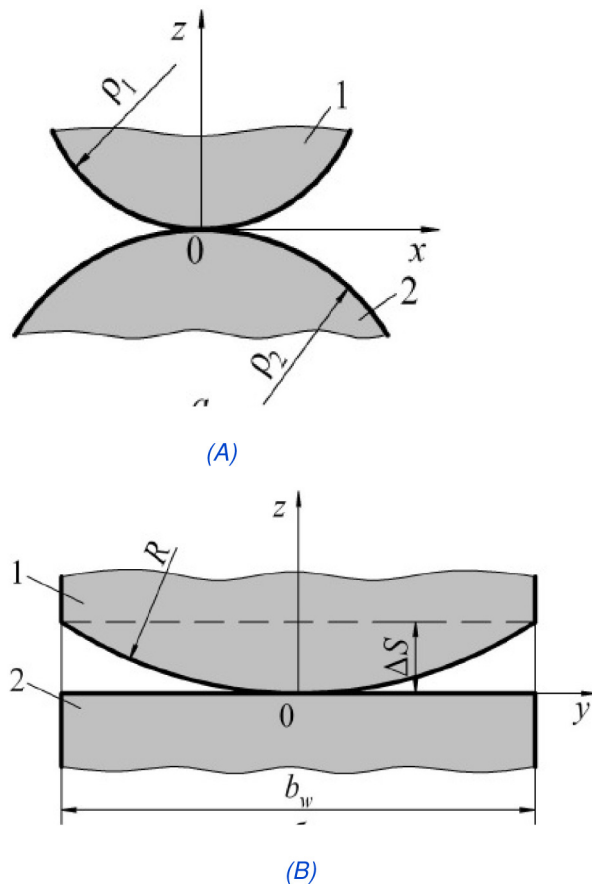


Figure 1. The model for calculations: teeth contact in plane $z0x$ (A) and contact of cylinder with plane in projection on coordinate plane $z0y$ (B); 1 - pinion tooth, 2 - gear tooth.

It is assumed in proposed model that patented gear design could lead to new modified shapes of contacting tooth. Under the load in contact area of a gear pair the observer could see a sort of “point”. This “point” looks like microscopic ellipse. Actually this fact is confirmed by experiments.

During proposed modeling the area of point contact is presented by elliptical area. According to above mentioned approach [4], [5] the equations for maximum contact stresses $\sigma_{\max}$ and size of elliptical area of a contact are obtained from a following model of a contact (see Figure 1). The proposed model is characterizing by external and internal touching contact of round cylinders having radii in ρ_1 and ρ_2 in $z0x$ plane (see Figure 1A), and by interaction of cylinder of radius R with certain plane having width of b_w in mutually perpendicular $z0y$ plane (see Figure 1B).

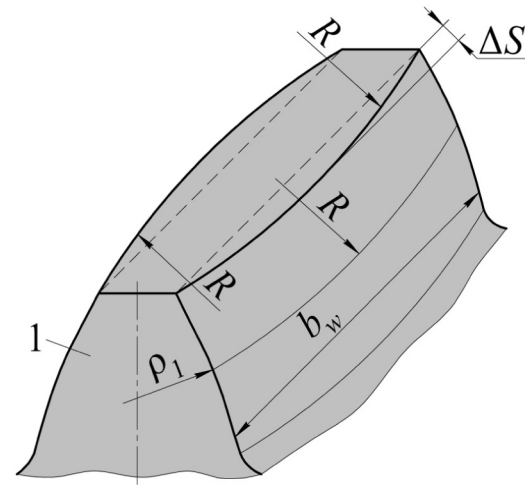


Figure 2. The general view of tooth for modified pinion.

There is a model of proposed pinion tooth having the side surfaces designed with help of curve lines in Figure 2.

The solutions for teeth contact problems for linear and nonlinear relationship between elastic deformations of teeth and outcome stresses are obtained. Following the formulas derived in [5], the maximum point contact stress would be determined as

$$\sigma_{\max} = 0.33 Z_j (K_0 K_E^{3/2} K_j^{3/2} \alpha (\alpha + \nu)^2 E^2 T_o / (\rho_w^2 R_p))^{1/3} \quad (1)$$

where (from Figure 1 and Figure 2)

$$\alpha = (\rho/R)^{1/2}$$

$$R = b_w^2 / 8 \Delta S$$

and according to [5]: R_p - pitch radius, K_0 is the contact stress coefficient, K_E is the coefficient reflecting the influence of type of engine on the contact tooth strength, Z_j and K_j are the combination of coefficients determined in [5]. Linear or nonlinear contact properties are modeled by the geometrical deviation parameter ΔS , which could be estimated using the linear or nonlinear assumptions. The following example of calculations was formed for an illustrative purpose. The prototype of geared reducer has several shifts. The considered example is for 1st shift [5] of transmission having point mesh. Parameters of 1st shift are $z_1 = 33$; $z_2 = 127$ (ratio is 3.85); $m = 4$ mm; $\alpha_w = 20^\circ$; $b_w = 77$ mm; $\rho_w = 17,916$ mm; $\nu = 0,3$; $E = 2,1 \cdot 105$ MPa. The allowable compressive stress for selected material and hardness is calculated based on recommendation of [6] and would be

1282 MPa. The estimates are done for a model of linear or nonlinear contact properties. A direct calculations were used for evaluation of contact stresses (1) for a case of linear contact properties, and final element technique was used for a case of nonlinear contact properties. A graphical interpretation of analysis is presented in Figure 3A, 3B, 3C.

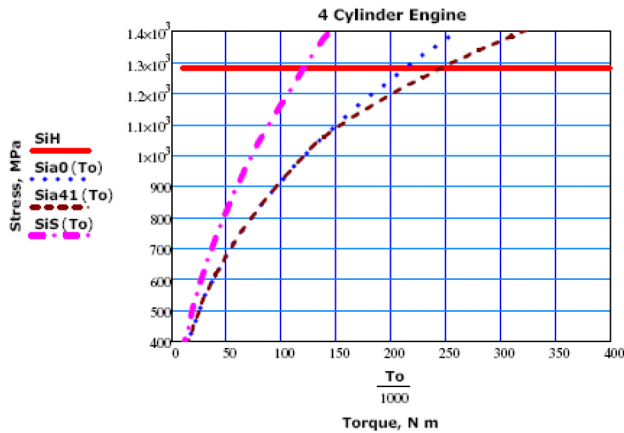


Figure 3A.

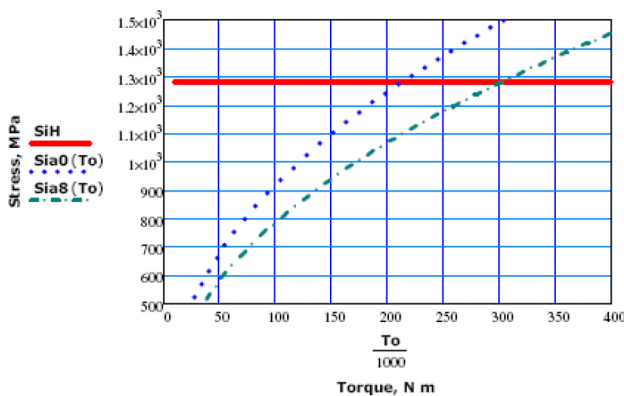


Figure 3B.

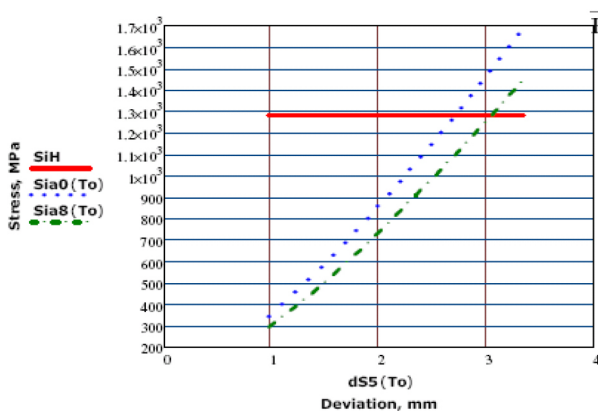


Figure 3C.

An ordinate axis is standing for compressive stress numbers in these graphs. An abscissa axis is for torque magnitudes T_o in Figure 3A and Figure 3B. The deviation numbers ΔS are given in the abscissa axis in Figure 3C. Red solid straight line is reflected the value of allowable compressive stress in this gear pair. The Figure 3A is the graphical illustration of contact stress calculations for transmission

used for 4 cylinder engine. There are three curves are shown in Figure 3A. Brown dashed curve is reflecting the relationship between the applied torque and the contact compressive stresses for linear considered model, and blue dotted curve is showing the dependence of the contact compressive stresses from applied torque for nonlinear considered model. The dash - dotted pink curve is built using the standard formulas (for instance see [6], [20], [21]) for stress calculations in traditional contact case - a case with "line of contact". The comparison of calculations for "line of contact" case (a dash - dotted pink curve) and "point of contact" case (a blue dotted curve) demonstrates that for low loads (low values of torque) both curves are practically coincided, but when the value of applied torque is increasing the load capacity of gear is higher for case of point contact. In given example the maximum stress would be reached at torque of 120 N m for "line of contact" case, and maximum stress would be reached at torque of 220 N m for "point of contact" case. Further analysis shows that for the lower values of torques the both curves reflecting the point contact case are practically coincided up to value of torque of 125 N m. Then using the nonlinear model the maximum value of stress could be reached at torque of 220 N m, while the maximum value of stress for linear model would be at torque of 250 N m. The received result could help for formulation of the recommendation - use the nonlinear model for having conservative safety factor. The influence of engine type (4 cylinder engine vs. 8 cylinder engine) is shown in Figure 3B. The blue dotted curve still is reflecting the relationship between the applied torque and the contact compressive stresses for transmission used with 4 cylinder engine (nonlinear model), and green dotted - dashed curve is reflecting the relationship between the applied torque and the contact compressive stresses for transmission used with 8 cylinder engine (similar nonlinear model). The replacement of 4 cylinder engine to 8 cylinder engine allows increasing of torque (in considered case from 220 N m to 300 N m) keeping the same level of contact stress in gear pair. Figure 3C gives the idea of relationship between parameter ΔS (torque dependent) and contact stresses (nonlinear model is applied). The blue dotted curve is reflecting the relationship between the applied torque and the contact compressive stresses for transmission used with 4 cylinder engine, and green dotted - dashed curve is reflecting the relationship between the applied torque and the contact compressive stresses for transmission used with 8 cylinder engine. Analysis of curves demonstrates that the magnitude of a parameter ΔS is increasing with torque rise and simultaneously this increase is forcing to increase of contact stresses. The maximum contact stress (in case of 4 cylinder engine) corresponds to value of deviation ΔS of 2.7 mm, while the maximum contact stress (in case of 8 cylinder engine) corresponds to value of deviation ΔS of 3.1 mm.

STATIC TESTS

In order to identify the applicability of introduced theory and to make its verification, a series of experimental tests were employed. First of all the static tests were executed [5]. There was contact compressive imitation of two cylinders loaded by force F_n ; circle cylinder with radius $r_1 = 60$ mm was contacting the barrel - shaped cylinder with maximal radius $r_2 = 40$ mm. The radius of barrel shape was $R = 41.667$ m. The length of cylinders was $b_w = 100$ mm. The force F was applied several times as step load in range of 5,000 N - 30,000 N, step equal to 5,000 N.