The machinery and aggregates fundamental improvements reserves

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Abstract. The machinery and aggregates characteristics fundamental improvements possibilities are shown by solving acute (which are not solved by traditional methods) problems of improving the power contact units, (gears) performance. The gears advanced import substitution examples are given.

Introduction
The aim of this work is to substantiate the machinery and aggregates characteristics fundamental improvements possibility, the design of which contains power contact units (e.g. gears), largely determining by its size, weight and other layout characteristics, the lifting, resource-strength and machinery and aggregates speed performance. Therefore, three tasks will be solved in this paper: 1) to show that the power gear mesh geometry development traditional methods, based on the Hertz-Belyaev’s contact problems classical solutions use, are not confirmed by practice; 2) to explain the reason for the traditional methods’ low correlation and their application practices; 3) to justify the refined physical basis for creating real opportunities for the gear mesh fundamental improvements and as a consequence the performance of machinery and aggregates.

The relevance of solving these problems is due to the fact that the world gear construction development and competitive pressures main way is traditionally associated only with the best choice of gear material, with the processing technology (the types of modification of the working surface of the gearing, finishing methods, etc.) improvement (and also complication) and with the most advanced technological equipment acquisition. And all this — at a time when there was a mesh promising system of E.Buckingham (the first half of the twentieth century) and M. L. Novikov (the second half of the twentieth century). Moreover, it is now believed that the gearboxes of the world’s leading manufacturers are the “works of art” for their technological processin development. Materials and technological processes should be improved, but the high-cost way of creating the “works of art” is not very effective competitively. The solution of the tasks set here will show that the main tool for improving such a complex contact unit as the gear mesh can be its geometric parameters.

Discussion
The traditional physical basis for the gears geometry development (based on the Hertz plane contact problem solution) is given in the two main factors straight-linear relationship contact form (the permissible force $F_p$ and the reduced radius of curvature):
The scope applicability classical solutions contact problems estimates are strictly limited by the indicators [1, 2], and if there is no compliance with these indicators the solutions of other (non-classical) problems should be used [1]. A particularly exceptional position, which is occupied by the classical solutions of Hertz-Belyaev [1, 2] in the world science, is traditionally explained by the prevailing idea of their high accuracy not only within the canonical form contacting bodies framework, but also in relation to a wide (at least from rolling bearings to involute gears) calculating real contact nodes engineering practice [1, 2].

However, since the middle of the XX-th century the involute gears contact calculations practice traditional theory unexpected (“inexplicable”) contradictions descriptions accumulation starts in the world scientific literature.

A large negative role in the development of the complex contact nodes shape geometry (and, as a result, of their resource-strength calculation theory and practice) played and keeps playing a literally sacred attitude to any doubts about the classical solutions accuracy. It is believed that the errors are caused by each gear manufacturer technological features influence, and they can always be compensated by some empirical (without analytical evaluation) solution.

Further design scheme refinement, as a rule, are being looked for in terms of Hertz model various assumptions impact assessing: the dynamic phenomena consideration lack and the intermediate oil layer, the roughness and friction of surfaces in contact, as well as deviations of the compressible bodies materials properties from isotropy, homogeneity and ideal elasticity.

Moreover, since the mid-twentieth century the results of equally reliable involute gears bench tests are periodically published in different countries: “inexplicable” (with a completely non-Hertz influence of the mesh angle on its contact strength) results and controls, made to check the "unexplained" results by other authors and on other stands. Control tests, which showed full compliance with traditional methods, became an additional support for the wide use of traditional methods.

The ways of improving based on the physical foundations (1), on the gear mesh kinematics were searched for in the USSR (N.I. Kolchin, Y.N. Budyka et al.) and in the United States (E.Buckingham, S. Bramley-Moore et al.). E.Buckingham had the most interesting solutions – he proposed the new gear mesh, and it (mixed gear mesh) was even standardized in the USA and in the USSR. But practice has not confirmed the theoretical conclusions and those standards have been eliminated.

According to the generalized (Y.N. Budyka) kinematic Euler-Savary’s theorem on the conjugate gear profiles curvatures obtained by the plane four-link mechanism with lower kinematic pairs analysis, the curvature radii of the interacting gear profiles are determined by the equations:

\[ R_1 = \frac{r_{w1}\sin\alpha_{tw}}{1 + \frac{r_{w1}\cos\alpha_{tw}\tan\gamma_p}{l_p}} + l_p; \]
\[ R_2 = \frac{r_{w2}\sin\alpha_{tw}}{1 - \frac{r_{w2}\cos\alpha_{tw}\tan\gamma_p}{l_p}} - l_p; \]

where \( r_{w1,2} \) are the radii of the curvature of centroids; \( l_p \) – is the distance of the contact point from the gearing pole; \( \gamma_p \) – is the angle between the normal line to the gear profiles at the point of contact and the tangent to the gearing line at this point.

Based on the works of E.Buckingham and Y.N. Budyka, M.L. Novikov [3] has developed a practically important variant of the reserves implementation of the out-of-pole gear outside the plane gear – with a spatial point mesh. His notable step was the conditions rejection for the end overlap of the gear and the spatial point mesh geometric theory creation using the angle \( \gamma \) ranges \( \gamma_p \) to its value at which \( R_1 = \infty \). The advantages of such mesh seemed so huge that it allowed to conclude a significant reduction in contact stresses, exclusively in the point out-of-pole mesh. Moreover, M.L. Novikov directly pointed out in his book [3] that the new mesh would be a complete alternative to
involute mesh in power gears. In some cases of a serious restrictions number strict combination (in wide crown and non-spur gear pairs with “soft” gear and at the same time low-loaded and low-speed ones), these advantages of Novikov’s gears were confirmed, but such application scope is completely unimportant. The real reserve for increasing the contact strength of Novikov’s mesh was significantly lower than that required to replace the most common “hard” wheels, spur gear pairs and/or narrow crown gears (and in general – heavy-loaded gears) to compensate for the point mesh shortcomings in bending strength and axial forces [4].

M.L. Novikov proved the replacing the conditions possibility on the face overlap with the condition on the axial overlap and was able to improve the gear transverse profiles curvature. But as a result of the physical foundations underlying Novikov’s mesh, such an improvement (at the cost of transition to point mesh and deterioration of curvature along the gear line) gave a limited result of competitiveness due to a significant underestimation (and the traditional denial of development reserves) of the meshes with the initial-linear contact and, conversely, excessively high overestimation of Novikov’s mesh. With the geometric theory complete perfection, Novikov’s mesh turned out to be erroneous in terms of its physical foundations. As a result, it is significantly inferior to the involute mesh in the resource-strength and other structural indicators, especially for the transmissions with “hard” gear. That is why the Novikov’s force mesh hasn’t become an alternative to the involute mesh and, despite the high theoretical prerequisites, has a very small sphere of rational application and only in two countries – in Russia and China.

Therefore, all theoretically very promising (based on traditional physical basics) options for the gears kinematics development (including the meshes of E.Buckingham and M.L. Novikov) were not confirmed by practice. And there are no explanations for this and many other (for example – “inexplicable” experiments) inconsistencies of the contact problems classical solutions traditional use (more precisely – obtained on the traditional restrictions basis) in the contact nodes calculations (for example, the gear type) and in the ways choice to improve them.

During the whole period (since the beginning of the twentieth century to the present days) the classical solutions use of contact problems not even one really promising gear system has not been created. The serious restrictions were imposed instead on the involute mesh improvement and even on the possibility of mesh kinematics developing. It was involute mesh that received (within the traditional restrictions framework) a significant development [5] of the geometry and technology theory. The work developed by L. Euler [6] (during his work in Russia) long before the classical solutions of contact problems has great reserves nowadays [7, 8, 9] beyond the traditional restrictions.

**Results of the research**

In the works [4, 7……12] the contact problems classical solutions inconsistencies with geometric forms of contacting (including canonical [9…12]) bodies are shown. Nevertheless, their correspondence is generally recognized. The contact main factors interconnections, quantifying the classical solutions applicability limitations (under the contact area size different conditions, depending on the bodies shape and their contact elements curvature) are obtained via analytical (for canonical bodies) and approximate-analytical (for complex shape bodies) contactless methods. Also, the contradictions explanation between the contact calculations theory and the practice is given. The canonical bodies calculation methods are constructed using the theoretical foundations of N.I. Muskhelishvili [13].

As a result, there is a good agreement between the various formula results: the contact area ellipse eccentricity value decreases, the smaller reduced radius growth effect on the contact stresses reduction increases in accordance with the exponential dependence [4, 7, 8]:

\[ F_p = c^k, \quad k \geq 1, \]

where the exponent takes into account the change of the curvature radius second reduced. This dependence formed the basis for identifying the contact curvature effects – the effects of a much larger
(with a relevance to traditional representations) influence of the contacting bodies curvature on the maximum normal stresses and the permissible contact load. The contact curvature effects are determined by the deformable elastic bodies’ curvature role manifestation selective mechanism as one of their main geometric parameters.

When increasing the contact area localization (for example, typical for involute gear), the classical solution overestimates the results of calculating the maximum contact stresses by about 50%. The reduction of contact localization leads to this error reduction up to asymptotic approximation (in the case of initial-linear contact) to classical values. It is shown that not only an inflated size (smallness condition) of the contact area, but also its understated size depending on the contacting bodies shape and the curvature of their contact elements are related with the discrepancy indicators between the Hertz-Belyaev contact problems classical solutions and the canonical and many real bodies.

Traditional ideas do not reflect the real relationship of the main factors characterizing the gears bearing capacity and have narrowly applicable private character. Involute mesh with a gear angle \( \alpha_{ow} < 20^\circ \) doesn’t take into account the opportunity of a much greater (in comparison with the influence of the friction force) effect (which is opposite in sign) of increasing the total rolling velocity and a given radius of curvature. Involute gearing (with hypothetical accounting of the hydrodynamic effect) with angles \( \alpha_{ow} > 20^\circ \) doesn’t take into account the hydrodynamic effect absence zone existence and, most importantly, the hydrodynamic effect noticeable effect absence on the permissible contact gears stresses. The gear contact interaction mechanism found description use (2) creates an effective tool for improving the mesh geometry. Thus, the mesh angle rational values choice regularities are given for involute gears on the initial-point contact concept basis (Patent of the USSR №1710889 from 10.08.1991) taking into account the deformable state of the contacting gears. For example:

\[
6.0 \leq \left( \alpha_{mw} - 11 \right) \leq \frac{8 \left( K_H^3 - 0.2 \right)}{\left( 3 - K_H^{-3} \right)},
\]

where \( K_{HE} \) is the rate defining the concentration of load along the gear line.

This shows some universal absence (or inherent in a particular branch of technology), invariant to the contact parameters of the mesh real gear optimal angle value, for example 25° or extremely possible (structurally maximum or technologically minimum) angles.

The expression of physical bases (2) shows (in contrast to traditional representations) the possibility of reducing contact stresses (regardless of the mesh system) in the phase of pole mesh. The expression (3) allows the choice, justified by the involute gear mesh angle value condition (2). Its rational values for the mesh pole point in a wider representation of the various meshes geometry and the contact conditions of the gear depend on the real contact parameters and often lie in a wide range: \( \alpha_{recopt} = \alpha_{remin} = 3.3^\circ \), where \( \alpha_{remin} \) is the minimum possible (according to technological or constructive constraints) mesh angle value. The increase in the mesh angle obtained under the condition (3) also means an increase in the reserves for improving the involute gear.

E. Buckingham’s mixed mesh systems are based on the gear geometric shape additivity conditions (with a smooth interface between the gear profile sections with different kinematics), which leads to the gear profile different sections geometric parameters choice complete interdependence, and, as a result, to the structural flexibility reduction and resource-strength indicators of gear. The feasibility of increasing, according to the involute section mesh angle physical principles (2) led to the plane mixed establishment (involute-point) mesh – “IP-S” (Patent of the USSR № 1571330 from 04.25.1988; Europatent No, 1908992 from 05.05.2010; Eurasian patent No. 011706 from 04.28.2009), in which the near-pole involute sections and the out-of-pole convexo-concave sections are easily integrated, and the contact points of the out-of-pole sections coincide with the involute section profile terminal point. Thanks to the mixed mesh “IP-S” use in spur on-board gear pairs (which used to limit severely the crawler tractor “VT-100” competitiveness), the unsolvable problem of replacing the engine (and even with the gear shaving operation refusal) of the new tractor “VT-150” was solved. Ten-year comparative tests showed that the load capacity was increased by 1.5-2.0 times and the cost of the “IP-
S" gear with milled gear was reduced to \([a_w = 276.25 \text{ mm}, m = 6.3 \text{ mm}, \text{ cementation, hardening, HRC (56...60)}]\).

The service-life of the “IP-S" gear (with no failures before the end of the trial) amounted to 4885 operating hours, and involute gear (with shaved gear and a reduced load) – 2500-3000 operating hours before failing.

Comparative tests (Figure 1) of low noise spur, helical “IP-S" gears (with milled gear) and involute gears with shaved (2) gear were performed for other objects.

![Figure 1](image)

**Figure 1.** To the evaluation of low-noise spur and helical “IP-S" gears. The noise level of cylindrical “IP-S" gears under partial load (microphone – 1000 mm from the gear ends): 1 – “IP-S" gears with milled gear (Rz = 30 microns); 2.3 – involute gears with generated teeth (2) (Rz=10-15 microns) and milled (3 – indicative data) teeth (Rz = 20-30 microns) of a special initial contour;

- the experiments for the spur gears \(z_{1,2} = 14/38; \beta = 0; m = 4.25 \text{ mm} \) (gear reduction rate); \(v_o = 9.3 \text{ m/s}\); • the experiments for the helical gears \(z_{1,2} =38/29; \beta = 28.91^\circ; m = 2.25 \text{ mm} \) (multiplication); \(v_o = 15.7 \text{ m/s}\).

**Summary**

An even greater manifestation of the stress levels reducing effect and the heat release in the gear contact was achieved in two different spatial mixed systems (involute-point, with areas of mesh – involute and Novikov’s) “IP-B" and “IP-Baf” meshes (European Patent №0293473 from 06.29.1992). The gear profiles of the “IP-B" and “IP-Baf” wheels consist of kinematically independent sections made according to the previous laws, which removes the restrictions (imposed by a single law of the profile description) on the radius \(\rho_2\) values choice in different mesh phases. These spatial mixed mesh systems are constructed on the basis of the previously unknown effect of mechanics – the geometric concentrator’s curvature effect (GCCE) [14], revealed by G. A. Zhuravlev. The essence of the GCCE lies in the fact that under certain (created in the gear of “IP-B" and “IP-Baf") conditions of the geometric concentrator use in the bodies of a complex shape, the maximum bending stresses in the geometric concentrator section are reduced as the concentrator curvature radius decreases (and not increases – as shown by the stress concentration theory). The GCCE use allowed to solve four problems at once – to strengthen the advantages and to eliminate the main drawbacks (for example, to implement the out-of-pole involute mesh and the mesh of Novikov with an unlimited flexural strength, at least at the gear head) of each of these gearing, and, therefore, to increase the competitiveness of mixed meshes and other meshes in their kinematics for the first time (during the involute mesh and the mesh of Novikov existence). The testing “IP-B" and “IP-Baf” gears experience and mastering “IP-B" gear (for example, in the trolleybus traction mechanism in the advanced import substitution format) gives reason to believe that in this way Novikov’s mesh will find a good field of application in mechanical engineering.

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