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# The influence of skew and deflection of the rotating shafton force factors arising in gear joints

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**Abstract.** Important and priority tasks for the industry of Ukraine are the reduction of costs for the creation of the final product, the reduction of the cost of maintenance of the construction of machines and mechanisms, the restoration of the outdated production base and the attraction of new technologies, mechanisms and equipment. That is why, increasing the efficiency, reliability and service life of machine units, which are used in conditions of misalignment of the axes of the connecting shafts, which are caused by the deflection of the rotating shaft in the mechanisms, which are connected to each other by an intermediate shaft with the help of compensating devices, and are intended for the transmission of effective power from the engine to the consumer, and, as a rule, is the most important part of any power plant, is an urgent task, and the further development of the modern machine-building industry depends on its successful solution. At the same time, the effectiveness of machine mechanisms directly depends on the productive work of gear mechanisms that are part of them, namely mechanical gears and gear couplings, which are widely reflected in almost all industries. Thus, the purpose of the work is to assess the influence of the deflection of the rotating shaft on the force factors that arise when the axes are misaligned in the gear joints of machine units. The effect of skew and deflection of a shaft rotating at a high frequency on the force factors for some toothed joints was investigated. It was found that the values of the dynamism coefficients do not exceed 1.05-1.15, and the dynamic components of these power factors can reach 30-50% of the level of their static components

**Keywords:** rotating shaft, deflection, gear clutch, transmission, misalignment of axes, power factors

## INTRODUCTION

The priority tasks for the Ukrainian industry are to reduce the costs of creating the final product, reduce the cost of construction and maintenance of machines and mechanisms, update the outdated production base and introduce new technologies and equipment. Therefore, increasing the efficiency, reliability and service

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life of machine units that are operated in conditions of misalignment of the axes of the connecting shafts, which are caused by the deflection of the rotating shaft in the mechanisms (main engine-reducer), which are connected to each other by an intermediate shaft with the help of compensating devices, and designed to transmit effective power from the engine to the consumer, and is usually the most responsible part of any power plant, is an urgent task and the further development of modern mechanical engineering depends on its successful solution. In turn, the efficiency of machine units directly depends on the reliable operation of the gear mechanisms included in their composition, namely mechanical gears and gear couplings, which have found their wide distribution in almost all branches of industry, such as: shipbuilding, aviation, mining and agriculture, as well as in rolling mills, cranes, lifting and transport mechanisms, mills, crushers, conveyors, agricultural machines and many others. Today, in the global industry, regardless of the type, gear couplings are manufactured with straight side surfaces of the sleeve and clip teeth, or with straight side surfaces of the clip and barrel-shaped (longitudinal modification) side surfaces of the sleeve teeth. As for gears, they are characterized by linear engagement of involute teeth. For a long time, it was believed that gears with linear contact of involute teeth have no alternative and cannot be manufactured with point meshing of teeth, which, on the one hand, is due to the inaccuracy of calculations that were carried out according to the Hertz formula, which is true for linear contact, but cannot to be used for point contact, and on the other hand, the difficulty of manufacturing such gears on commonly used machines (at present, the production of teeth with point contact is possible due to the use of high-precision tooth grinding machines of the German production type "Pfauter" or "Hoffler") [1-7]. However, at the moment, the influence of the deflection of the rotating shaft on the force factors arising from the misalignment of the axes in the toothed connections of machine units is not sufficiently clarified and is usually not taken into account when implementing new technology.

When the axes of the connecting shafts of machine units of power plants are misaligned, additional force factors in the form of elastic bending moments arise in gear joints. The effect of these moments is that they tend to return the bent shafts to their original position. The bending moments arising in the toothed connection overload the supports of the mechanisms connecting the shafts (intermediate, output and input shafts of the engine and gearbox, respectively), support bearings, spline and bolt connections, as well as other elements, negatively affecting on their performance, which in turn negatively affects the reliability, durability and maintainability of machine units, and can lead to their failures and unscheduled stops for repairs. In turn, the deterioration of reliability indicators leads to a decrease in reliability, an increase in additional costs and repair costs, as well as an increase in the probability of emergency situations of machine units, and as a result – power plants, in which they are operated. Thus, the problem of the negative influence and deflection of the rotating shaft on the force factors arising in gear joints is a relevant and integral part of existing scientific programs and tasks in almost all branches of modern mechanical engineering. To the greatest extent, the specified problem concerns the transmissions of machine units operated in the mining, aviation, shipbuilding and agricultural industries. The most complete research on the specified problem is presented in [1-15]. The study of cases of shifting of shaft axes due to installation inaccuracies, errors in the manufacture of power parts and operating conditions showed that when the gear half-coupling axis is skewed relative to the bearing axis, a centrifugal force acts on the rotating curved shaft, which increases its static deflection [8-13].

In works [8; 14; 15], the main attention is paid to theoretical and experimental studies of elastic bending moments that occur under static and dynamic loads, as in traditional gear couplings (that is, the teeth do not have any modification, or the teeth of the sleeve have barrel-shaped teeth of constant curvature, and the teeth of the clip are made straight), as well as gear couplings of new designs, i.e. with modified teeth.

The causes of misalignment of the axes of connecting shafts of machine units are shown in a structured form in the work [7]. The influence of the specified misalignments of the axes on the reliability parameters of machine units and power plants, in which they are operated, is shown on the example of ship power plants and highlighted in [4; 6; 7]. Ways to reduce the influence of misalignment of the axes of connecting shafts and methods of eliminating their negative impact on the operation of machines and mechanisms, in the composition of which they occur, are shown in works [3; 4; 6; 7; 8; 10]. In work [5] it is shown that increasing the efficiency of machine units with misalignment of the axes of their connecting shafts is possible by using new designs of gear couplings protected by patents of Ukraine.

However, scientific works [2; 4; 6; 7; 10; 11; 12; 13; 15] did not take into account the influence of skew and deflection of the rotating shaft on the force factors arising in gear joints, but from the analysis of publications [1; 2; 8; 9; 10], it is evidently necessary to pre-estimate the effect of the dynamics coefficients on the force factors acting in them when designing gear joints of rapidly rotating shafts.

*The purpose of the work* is to evaluate the influence of the deflection of the rotating shaft on the force factors that arise when the axes are misaligned in the gear joints of machine units.

## STUDY OF THE INFLUENCE OF SKEW AND DEFLECTION OF THE ROTATING SHAFT ON FORCE FACTORS ARISING IN GEAR JOINTS

Misalignment of the axes (misalignment) is a violation of shaft alignment, i.e. a deviation from the nominal location of the axes in any direction (Fig. 1). These deviations are called shifts, and they, in turn, are divided into longitudinal, radial, angular and combined. It should be noted that angular displacements are also called fractures.

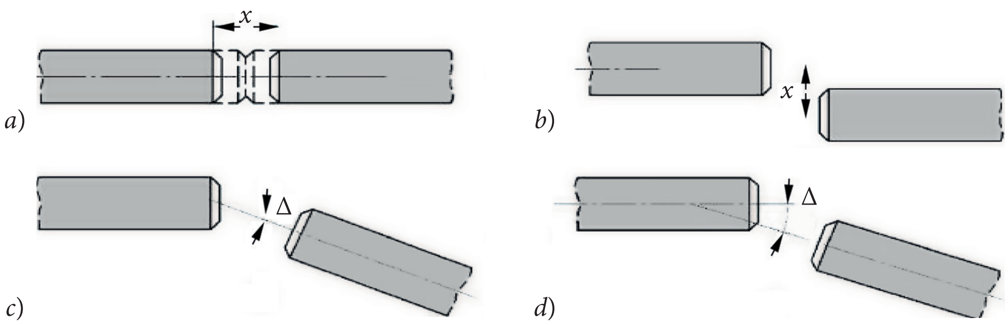


***Figure 1***. *Displacement of the axes of the connected shafts of machine units: longitudinal (a), radial (b), angular (break) (c), combined (d)*

In high-speed, heavily loaded transmissions of power plants during operation, distortions and displacements of their axes inevitably occur [1; 2; 8; 9; 3-6]. The magnitudes of axis misalignments during the periods of operation of power plants depend on a number of structural, technological and operational factors (unreliable foundation, expansion in pumping equipment, non-compliance with installation recommendations, etc.), in addition, there are combined and specific factors, but they are not considered . As a rule, misalignment of axes and their breaks are the values of misalignments caused by deviations in manufacturing technology, for example ship power plants and their elements (Fig. 2-5).

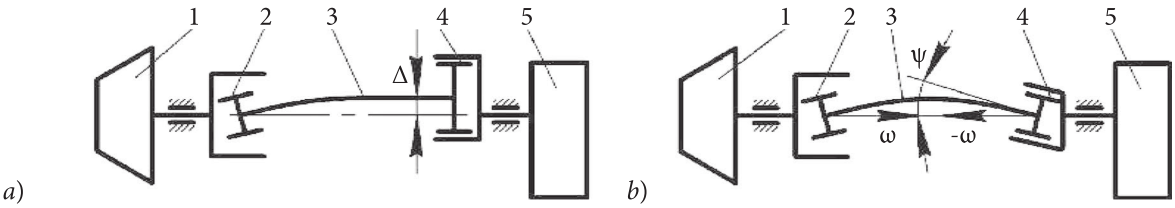


***Figure 2***. *Beating (a) and misalignment (b) of the teeth of the gear clutch: 1 – propeller turbine; 2, 4 – toothed bushings; 3 – shaft; 5 – reducer*

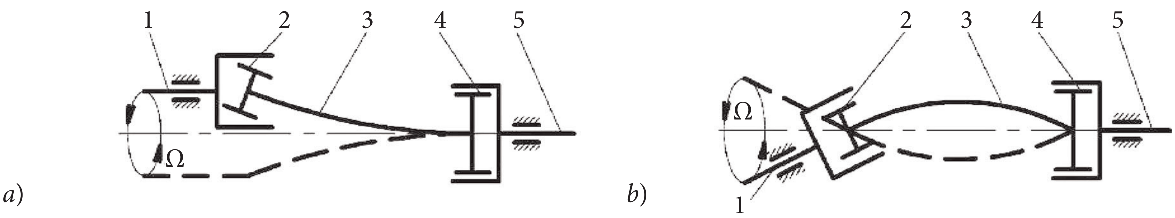


***Figure 3***. *Runout (a) and misalignment (b) of the axis of the gear bushing: 1 – shaft; 2, 4 – toothed bushings; 3 – spring; 5 – drive mechanism shaft*

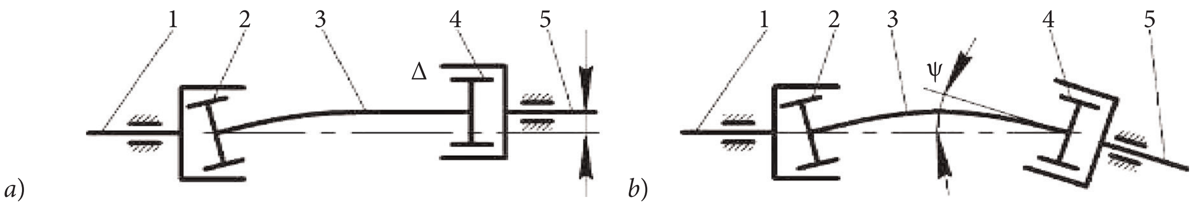


***Figure 4***. *Displacement (a) and misalignment (b) of the axis of the gear mechanism: 1 – motor shaft; 2, 4 – toothed bushings; 3 – spring; 5 – drive mechanism shaft*

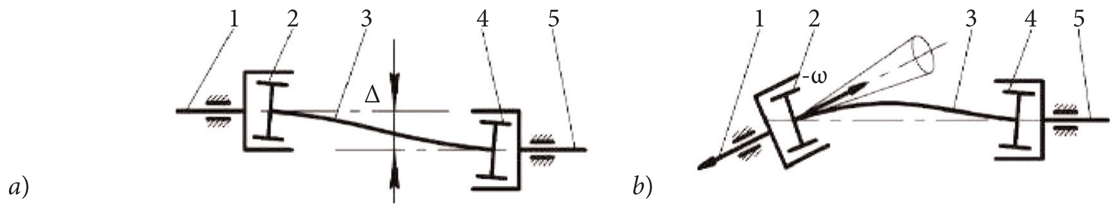


***Figure 5***. *Shift (a) and misalignment (b) of the engine axis:*
*1 – engine shaft; 2, 4 – toothed bushings; 3 – spring; 5 – drive mechanism shaft*

In order to eliminate the negative impact of misalignment of the axes of the connecting shafts of machine units and, as a result, to ensure the performance of the specified functions, the installed equipment of the installations provides for the control of alignment, i.e. checking the alignment of the axes of the connecting shafts of engines and drive mechanisms. This method is associated with a complete shutdown of the power plant, and the re-centering itself is quite time-consuming and expensive. Another method involves compensating the harmful effects of the specified misalignments of the axes, due to the use of compensating devices, but today, the influence of the static deflection of the rotating shaft on the force factors arising in their toothed joints, as already indicated, is almost not taken into account.

The study of the cases of shifting of shaft axes [1-8; 9; 14], caused by installation inaccuracies, errors in the manufacture of power parts and operating conditions, showed that when the gear half-coupling axis is skewed relative to the bearing axis by an angle $\theta$b (Fig. 6*a*) on a rotating curved shaft there is a centrifugal force that increases its static deflection.

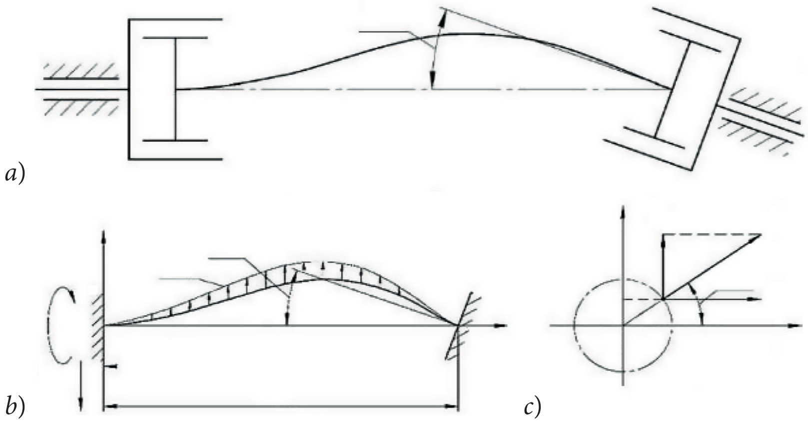


***Figure 6***. *Scheme of shaft deflection: a – at the angle $\theta$b; b, c – under the action of centrifugal force*

In the article, an attempt is made to take into account the influence of the specified deflection of the rotating shaft on the force factors that occur when the axes are misaligned in gear joints. When determining the dynamic shape of the deflection of the rotating shaft, which is under the action of centrifugal forces Pц (Fig. 6 *b*, *c*), the theory of constant forced bending vibrations of the beam, developed by O.M. Winged

Centrifugal force acts on the rotating bent shaft, which increases its static deflection. At the same time, the projections of the centrifugal force Pц acting on each element of the curved shaft are equal to:

$$P_y = P_{ц} sin\omega t; \quad P_z = P_{ц} cos\omega t,$$

where $\omega$ is the angular speed of the shaft.

The indicated projections Py and Pz cause forced oscillations of the shaft in the plane of action $y_0z$.

The movement of the shaft cross-section at any moment in time can be expressed by the following dependencies:

$$Y = Y_{дин} in\omega t; \quad Z = Y_{дин} cos\omega t,$$

and the resulting movement of the shaft cross-section represents movement along a circle, the radius of which is equal to:

$$Y = Y_{ст} + Y_{дин}. \tag{1}$$

The output static deflection of the shaft in this scheme (Fig. 1 *a*, *b*) has the form:

$$YCT(x) = \theta BL\left[\left(\frac{X}{L}\right)^2 - \left(\frac{X}{L}\right)^3\right]. \tag{2}$$

Thus, the rotating shaft will be acted upon by a disturbing (centrifugal) force determined from the expression

$$P(x,t) = \frac{\gamma F}{g}\,\omega^2 Y_{CT(x)}\,sin\,\omega\,t \tag{3}$$

Where $\gamma$ is the specific weight of the shaft material, $H/m_3$; $F$ – cross-sectional area of the shaft, $m_2$; $g$ – acceleration of gravity, $m/s_2$; $L$ – shaft length, m;

$\theta_B$ – angle of misalignment of the axis of the toothed semi-coupling with respect to the axis of the bearing, rad.

The equation of forced bending vibrations of a beam of constant cross-section will have the form:

$$EJ\frac{\partial^4 Y_{(x,t)}}{\partial X^4}+\frac{\gamma F\partial^2 Y_{(x,t)}}{\partial t^2}=P_{(x,t)} \quad (4)$$

where $E$ is the modulus of elasticity of the shaft material, Pa; $J$ is the moment of inertia of the shaft section, kg.m$_2$

Since the minimum critical revolutions for a number of analyzed structures are significantly higher than the operating revolutions, to solve this problem it is sufficient to consider only the steady forced oscillations of the shaft with the frequency $\omega$. Then we look for the solution of equation (4) in the form

$$Y(x,t)=Y(x)\sin\omega t, \quad (5)$$

where $Y(x)$ is the shape of the dynamic deflection of the rotating shaft.

Entering the dimensionless coordinate $\xi=X/L$ and substituting expressions (3) and (5) into equation (4), we get the equation for the shape of the dynamic deflection of a rotating shaft:

$$Y\frac{IV}{\xi}-\alpha 4Y(\pounds)=\alpha 4\theta_B L(\xi 3-\xi 2) \quad (6)$$

where

$$\alpha 4=\frac{\gamma F}{g}\omega^2\frac{L^4}{EJ} \quad (7)$$

We solve equation (6):

$$Y_{(\xi)}=\overline{Y}_{(\xi)}+\overline{Y}_{(\xi)}, \quad (8)$$

where $\overline{Y}(\xi)$ is the solution of equation (6) without the right-hand side; $\overline{Y}(\xi)$ is a private solution, which is determined by the form of the right-hand side by the method of undetermined coefficients.

Equation for $Y(\xi)$:

$$\overline{Y}(\xi)=C_1V_{1(\alpha\xi)}+C_2V_{2(\alpha\xi)}+C_3V_{3(\alpha\xi)}+C_4V_{4(\alpha\xi)} \quad (9)$$

where $V_{1(\alpha\xi)}$, $V_{2(\alpha\xi)}$, $V_{3(\alpha\xi)}$, $V_{4(\alpha\xi)}$ – Krylov's functions, which for the calculated value of $\alpha$ according to (7), are selected according to [2, 9].

The final solution is equal (8) to the improvements (9) in the future will look:

$$Y_{(\xi)}=C_1V_{1(\alpha\xi)}+C_2V_{2(\alpha\xi)}+C_3V_{3(\alpha\xi)}+C_4V_{4(\alpha\xi)}-\theta_B L(\xi^3-\xi^2) \quad (10)$$

The constants $C_1$, $C_2$, $C_3$ and $C_4$ are determined from the boundary conditions according to the scheme of laying the shaft.

According to the diagram (Fig. 1*b*), it is $\xi=0$, $Y_{(0)}=0$; $Y'_{(0)}=0$, and at $\xi=1$, $Y_{(\xi)}=0$; $Y'_{(\xi)}=0$

Then, subjecting equation (10) to the written boundary conditions, it is found $C_1=C_2=0$; $C_3V_{3(\alpha\xi)}+$ $+C_4V_{4(\alpha\xi)}=0$; $C_3V_{2(\alpha)}+C_4V_{3(\alpha)}=0$, where

$$C3=-\theta B\frac{L}{\alpha}\frac{V_{4(\alpha)}}{V_{3(\alpha)}^2-V_{2(\alpha)}V_{4(\alpha)}}; \quad (11)$$

$$C4=-\theta B\frac{L}{\alpha}\frac{V_{3(\alpha)}}{V_{3(\alpha)}^2-V_{2(\alpha)}V_{4(\alpha)}}. \quad (12)$$

At the same time, the dynamic shape of the deflection of the rotating shaft according to expression (10) is equal to:

$$Y=\theta B\frac{L}{\alpha}\left[\frac{V_{3(\alpha)}V_{4(\alpha\chi)}}{V_{3(\alpha)}^2-V_{2(\alpha)}V_{4(\alpha)}}-\right.$$
$$\left.-\frac{V_{4(\alpha)}V_{3(\alpha\chi)}}{V_{3(\alpha)}^2-V_{2(\alpha)}V_{4(\alpha)}}-\alpha(\xi^3-\xi^2\right] \quad (13)$$

The total shape of the deflection of the rotating shaft, taking into account (1), will take the form:

$$Y\Sigma=\theta B\frac{L}{\alpha}\left[\frac{V_{3(\alpha)}V_{4(\alpha\xi)}}{V_3^2-V_2V_4}-\frac{V_{4(\alpha)}V_{3(\alpha\xi)}}{V_3^2-V_2V_4}\right] \quad (14)$$

To find the force and moment in the front gear coupling, the equations for $Y''_{\Sigma(0)}$ and $Y'''_{\Sigma(0)}$ are determined when using (14):

$$M\Sigma(0)=EJY''_{\Sigma(0)}=\theta B\frac{EJ}{L}\frac{\alpha V_{4(\alpha)}}{V_{3(\alpha)}^2-V_{2(\alpha)}V_{4(\alpha)}} \quad (15)$$

$$Q\Sigma(0)=EJY'''_{\Sigma(0)}=\theta B\frac{EJ}{L}\frac{\alpha^2 V_{3(\alpha)}}{V_{3(\alpha)}^2-V_{2(\alpha)}V_{4(\alpha)}} \quad (16)$$

On the other hand, $M_{\Sigma(0)}$ and $Q_{\Sigma(0)}$ can be represented as $M_{\Sigma(0)}=K_mM_{ст(0)}$; $Q_{\Sigma(0)}=K_QQ_{ст(0)}$. Dependencies for $M_{ст(0)}$ and $Q_{ст(0)}$ based on the form of static deflection (2) have the form:

$$M_{ст}(0)=-\frac{2EJ}{L}\theta B, \quad (17)$$

$$Q_{ст}(0)=-\frac{6EJ}{L^2}\theta B, \quad (18)$$

After considering equations (15), (16) and (17), (18), we will obtain the final form of the formulas for the dynamism coefficients:

$$KM=\frac{\alpha V_{4(\alpha)}}{2\left[V_{3(\alpha)}^2-V_{2(\alpha)}V_{4(\alpha)}\right]} \quad (19)$$

$$KQ=\frac{\alpha^2 V_{3(\alpha)}}{6\left[V_{3(\alpha)}^2-V_{2(\alpha)}V_{4(\alpha)}\right]}. \quad (20)$$

On the basis of the obtained dependencies (19) and (20), an analysis was carried out for a number of toothed connections [4, 5, 9], which showed that the values of the dynamics coefficients do not exceed 1.05-1.15, and the dynamic components of the force factors can reach 30-50% of the values of the static components of the shaft rotating at high revolutions.

## CONCLUSIONS

1. The analysis of the obtained solutions for toothed joints showed that the values of the dynamic coefficients do not exceed 1.05-1.15, thus, it was established that the dynamic components of the force factors are approximately an order of magnitude lower

than the static loads. 2. It was established that the dynamic components of power factors can reach 30-50% of the values of the static components of a rotating shaft with high revolutions due to an unfavorable combination of dimensions, wear of contacting surfaces, manufacturing errors and other reasons. 3. When designing toothed connections of rapidly rotating shafts, it is necessary to preliminarily assess the influence of dynamic coefficients on the force factors acting in them.

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## Вплив перекосу і прогину обертового вала на силові фактори, що виникають у зубчастих з’єднаннях

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**Анотація.** Важливими та пріоритетними завданнями для галузі промисловості України є зменшення витрат на створення кінцевого продукту, зменшення собівартості обслуговування побудови машин та механізмів, реставрації застарілої виробничої бази та залучення нових технологій, механізмів та устаткування. Саме тому, підвищення працездатності, надійності і терміну служби машинних агрегатів, які застосовуються в умовах перекосів осей з'єднувальних валів, що обумовлені прогином обертового вала у механізмах, які з'єднані між собою проміжним валом за допомогою компенсувальних пристроїв, та призначені для передачі ефективної потужності від двигуна до споживача, і, як правило, є найбільш важливою частиною будь-якої енергетичної установки, є актуальною задачею і від її вдалого рішення залежить наступний розвиток сучасної машинобудівної сфери. Водночас, дієвість машинних механізмів безпосередньо залежить від продуктивної роботи зубчастих механізмів, які входять до їх складу, а саме механічних передач та зубчастих муфт, які знайшли своє широке відображення майже у всіх галузях промисловості. Таким чином, метою роботи є оцінка впливу прогину валу, що обертається, на силові фактори, які виникають при перекосі осей в зубчастих з'єднаннях машинних агрегатів. Досліджено вплив перекосу і прогину валу, що обертається з великою частотою, на силові фактори для деяких зубчастих з’єднань. З’ясовано, що значення коефіцієнтів динамічності при цьому не перевищують 1,05-1,15, а динамічні складові цих силових факторів можуть досягати 30-50% від величин рівня їх статичних складових

**Ключові слова:** обертовий вал, прогин, зубчаста муфта, трансмісія, перекіс осей, силові фактори