ECONTECHMOD. AN INTERNATIONAL QUARTERLY JOURNAL – 2014. Vol. 1. No. 1. 127–132

Optimization of the structural components of gearwheels of cylindrical reducing gears

O. Vasyljeva[1], I. Kuzio[2]

[1]Department of exploitation of transport vehicles and fire-rescue technique, Lviv state University of Life Safety

[2] Department of Mechanics and Mechanical Engineering Automation, Lviv Polytechnic National University, 79007 Lviv, Kleparivska str., 35, e-mail: *Vassabi13@ukr.net*

Received January 22.2014: accepted February 19.2014

Abstract. For the first time the design equations for determining the structural components of cylindrical tooth gears of reduction units are improved allowing for the use of informational technologies (IT).

On the basis of the equations obtained the optimizing mathematical model of the optimal synthesis of structural components of tooth gearwheels has been developed. For solving the optimizing model the Monte-Carlo method is applied. The structural parameters (quantities), obtained by the developed optimizing mathematical model, have optimal values with probability up to 0.95 under the conditions of necessary strength and production safety.

The implementation of the optimization process is recommended at the design institutions for the design of the cylindrical reduction drives for different objects of mechanical engineering and into the educational process of higher technical educational institutions while studying theoretical matters, accompanied by the use of IT.

Key words: optimization, structural components of a tooth gearwheel, optimizing model, tooth gearwheel strength, gearwheel reliability.

SETTING THE PROBLEM

Providing the substantiated choice of the optimal parameters of a designed construction is the main problem of modern mechanical engineering.

It especially concerns constructions working under the conditions of possible high overload. Some well-known scientists were concerned with the problems of structure synthesis and parameter optimization of different constructions, namely: I.I. Artobolevski, V.L. Genkin, E.M. Gerasimov, A.F. Kirichenko, P.L. Nosko, B.I. Kindratskij, and many others. K.I. Zablonskij, A.F. Kirichenko, V.P. Shishov and others dealt with the optimizing of the tooth gears of reduction units. But the methods they developed are applied only to certain tooth gear elements and gear shafts without using IT. So, the task is to develop a methodology allowing the synthesis of main optimal structural components of the tooth gears of reduction units according to their intended purpose with the use of IT. This task is urgent and necessary to be solved at the current stage of technology development.

ANALYSIS OF RECENT ACHIEVEMENTS AND PUBLICATIONS

Many researchers at research and design institutions, as well as at higher educational institutions in Ukraine and in the world were engaged in the development of different technical objects including tooth gears for technological objects with the use of IT.

Their efforts resulted in the development and implementation to the design practice such software as **ARM WinMachine, MDesign, Kompas, T-Flex,** special software (**Deshyfr, DMCost, DMNS, Pryvod**) and other [1].

Analyzing the software mentioned above and the techniques of their applying for the computation and development of different technical constructions, it could be noticed that using them produces certain results, but they cannot determine the optimal parameters of the object being designed. This drawback is caused by their using only well-known design equations without taking into account optimizing criteria necessary for a certain construction [5]. For example, the **T-Flex** system provides the multiple-choice geometrical calculation of tooth gear construction and strength in accordance with standard techniques.

In works [2-3] the prospects of improving the tooth gearwheels by means of the synthesis of the output circuit of the gear teeth are considered. In addition, on the base of the differential equations of operating profile only the profile of gear teeth with Novikov's toothing was synthesized [4]. Examining the tooth gearwheel, one can easily notice that besides the tooth operating profile there are many other construction elements which are not considered in the optimization process.

So the task set is to eliminate this deficiency to some degree, while determining the dimensions of the structural components of the tooth gearwheels of cylindrical reduction units, which should have high maintenance reliability. Using IT, it is necessary to take into consideration the current experience of higher educational institutions from abroad, concerned with this field: Politehnica University of Bucharest (Romania), Technical Military Academy of Bucharest, Silesian Polytechnic (Poland), Technical University Magdeburg (now The Otto von Guericke University of Magdeburg, Germany), Moscow State Technical University n.a. N.E. Bauman (Russia), National Technical University of Ukraine "KPI", Donetsk National University, Donbas National Academy of Civil Engineering and Architecture, National Technical University "Kharkiv Polytechnic Institute", Donbass State Engineering Academy, Donbas State Technical University and others.

The purpose of the work is to develop the optimizing mathematical model of the synthesis of structural components of tooth gearwheels in cylindrical reduction units based on the results of the theoretical and experimental investigations with the use of IT.

SOLVING THE SET TASK

Choosing the optimization criteria. One of the main factors characterizing the quality of a certain product is its reliability. In turn, one of the main reliability indicators is the life span of every structural component and the product in general. According to State Standard of Ukraine 2860-94, one of the reliability indicators of the structural component of the product is an average life cycle $T_{сл}$. However, it should be added that increasing the product life cycle without considering its economical reliability indicators may not be reasonable.

Following the state for optimizing and synthesizing the construction parameters of the shaft tooth gears mentioned above, let's establish following criteria:

1. The economical indicator of reliability of the tooth gearwheel Q_e in UAH, which is equal to [6]:

$$Q_e = K_e \cdot T_{сл},$$
$$K_e = (Q_B + Q_E)/T_E, \qquad (1)$$
$$T_{сл} = \frac{N_0}{60 n_ц \left(\frac{s_E}{s_r}\right)^k}, \qquad (2)$$

where: K_e are expenses according to the equation (1), relating to the production and operation of the tooth gearwheel, UAH per hour; $T_{сл}$ is the average life span of the gear according to the relation (2), concerned with the maximum possible basic number of loading periods and projecting stress cycles effecting on the gear teeth, h; Q_B are production costs of the tooth gearwheel, UAH; Q_E – total operational costs, UAH; T_E is a set life span, h; N_0 is the basic number of stress cycles (for contact stresses $N_0 = N_{H\mathrm{limb}} = 30\mathrm{HB}^{2,4}$; for bend tensions $N_0 = N_{F\mathrm{limb}} = 4\cdot 10^6$; (when the hardness of tooth gearwheel is given in HRC, it is necessary to multiply hardness in HRC by 10); $n_ц$ is the frequency of stress cycles, min^{-1}; σ_E is equivalent tension (σ_H and the major of two values σ_{F1} and σ_{F2}, correspondingly), MPa; σ_r is limit endurance ($\sigma_{H\mathrm{limb}}$ = 2HB + 70 and $\sigma_{F\mathrm{limb}}$ = 1,8HB correspondingly), MPa; k is the index of bend tension hardness rate (k = 6 for wheels with homogeneous material structure and a ground fillet surface regardless of the hardness and thermal treatment of the teeth; k = 9 for nitrated tooth gearwheels, as well as for gears after carburizing and carbonitriding treatment).

2. Reliability costs Q_H, UAH:

$$Q_H = Q_П (T_E / T_{сл})^{\alpha}, \qquad (3)$$

where: $Q_П$ – reliability costs of the tooth gearwheel prototype, UAH; a – number exponent characterizing the level of production progressiveness in terms of increasing the product reliability; α = 1,3...1,5.

The choice of these criteria can be explained in such a way. $T_{сл}$ being increased, the stated criterion Q_e is gradually increasing up to the limit, when $T_{сл} = T_E$. In this case it becomes equal to $Q_B + Q_E$, determining the normal operating mode (Fig.1).

On the contrary, according to (3) the second criterion Q_H is gradually decreasing, while $T_{сл}$ being increased, at the expense of the reduction of operating costs, that is, the number of breakdowns is reduced (Fig.1).

Applying set criteria for the optimization and synthesis the constructive parameters of the tooth gearwheels of cylindrical reducers, let's present them as a two-criterion optimization surface (Fig. 2).

Along the axe X (Fig. 2) the operating span of the tooth gear is shown, which is proportional to the economical reliability indicator of the tooth gearwheel Q_e, and the axe Y represents the operating span of the tooth gearwheel, proportional to the reliability cost $-Q_H$. Then the general optimization criterion can be shown as:

$$\left| Q_e - Q_H \right| \Rightarrow \min .$$

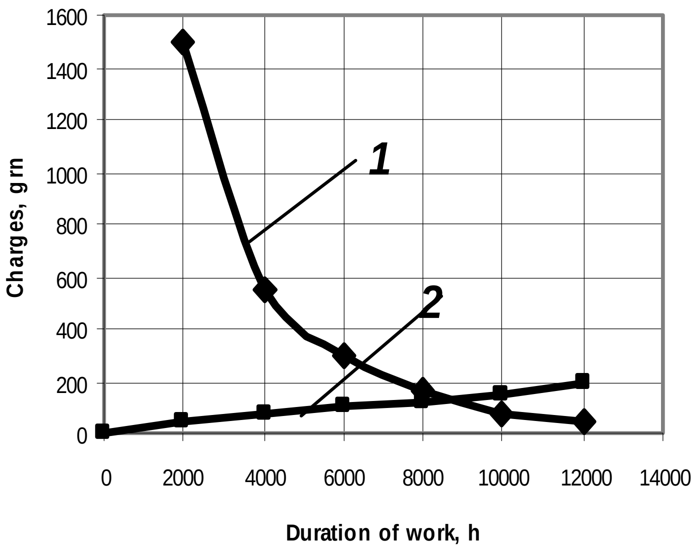

Fig. 1. Criterion relations: *1* – Q_H; *2* – Q_e

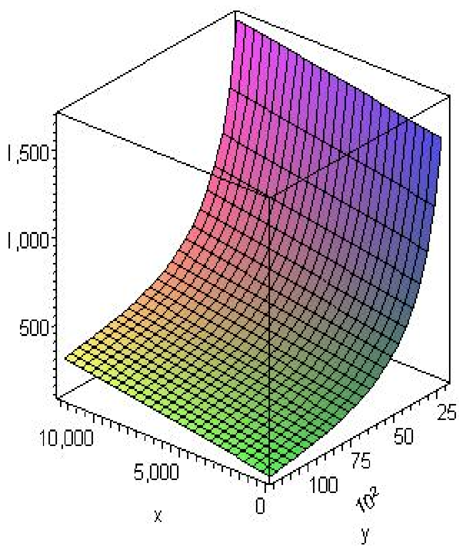

Fig. 2. Two-criterion optimization surface

The constructions of the tooth gears of cylindrical reducers. For the cylindrical reducers, three construction types of tooth gearwheels are mainly used: 1) single-crown gearwheels with sufficient length l and diameter d_o of a datum bore (a hub), so that $l/d_o > 1$; 2) single-crown gearwheels of disc type with $l/d_o \leq 1$; 3) tooth wheel-shafts, made as a whole, where the shaft is significantly long comparing the length of a toothed ring.

Determining the dimensions of structural components of the tooth gearwheels according to their mode of deformation. While operating under the load the deformation mode of the every tooth gearwheel appears, which is affected by the *thrust force* F_t, *axial force* F_a and *radial force* F_r. The values of these forces are determined by the equations:

$$F_t = \frac{2T_1 \cdot 10^3}{d_1},\ F_a = F_t tg\beta,\ F_r = \frac{F_t tg a_n}{\cos b},$$

where: T_1 is a torque, Н·м; d_1 is the diameter of the index gear of a driving tooth gearwheel, mm; β denotes the gradient angle of a gearwheel; α_n is a pressure angle in the normal section of the gearwheel tooth.

The main structural components of the cylindrical tooth gearwheel are: the top of tooth diameter (the outer diameter of the tooth gear) d_a; face width b; the datum bore of the hub d_o; a key groove in the datum bore of the hub $l_ш$ long, $b_ш$ wide, t_2 deep, and with the depth in the shaft t_1; the length of the hub l; plate thickness between the hub and the toothed ring $\delta_д$; the internal diameter of the toothed ring $d_в$.

For the determining the optimal dimensions of structural components of the tooth gearwheels depending to their deformation mode it is necessary to obtain the dependencies allowing the use of relevant IT methods. In this case the list of input data which are useful for obtaining design equations should be compiled.

The input data for obtaining design equations and the optimization of structural components of the tooth gearwheel are as foolows:

- material of the tooth gearwheel, with the hardness of HB or HRC in accordance with the grade of material and the kind of thermal treatment applied;
- liquid limit of the material of the tooth gearwheel $\sigma_т$, MPa;
- reduction ratio of the tooth gearwheel u_{12};
- given life span of the tooth gearwheel is considered to be up to 30000 hours (according to routine repair tips the average duration of a repair cycle from the first putting into operation equals to 30000 hours[8]);
- trouble-free life T_B, hours;
- circular velocity of the driving gearwheel V_1, m/s;
- driving torque on the driving gearwheel T_1, N·m;
- possible short-time overloads k_n, times;
- face width index $\psi_{ba} = 0{,}2\ldots0{,}5$;
- allowable contact stress $[\sigma]_H$, MPa;
- allowable bend tension of the driving gearwheel tooth $[\sigma]_{F1}$, MPa;
- allowable bend tension of the driven gearwheel $[\sigma]_{F2}$, MPa;
- safety margin index S_F;
- index taking into account the transmission parameters K_a, MPa$^{1/3}$;
- accuracy grade of gearwheels n_{cm};
- acceptable sound volume L, dBA;
- reliability cost of the tooth gearwheel prototype $Q_П$, UAH;
- production cost of the tooth gearwheel Q_B, UAH;
- total operational costs Q_E, UAH;
- index of the degree of a stability curve k;
- number exponent characterizing the level of production progressiveness in terms of increasing the product reliability α;
- allowable bearing stress for a key joint $[\sigma]_{зм}$, MPa;
- allowable tensile stress for steel $[\sigma]_p$, MPa;

- the diameters of a driveshaft $d_{в1}$ and a driven shaft $d_{в2}$, on which tooth gearwheels are located.

Design equations.

1. Determining the number of the teeth of the driving gearwheel z_1 and the driven gearwheel z_2 according to the given circular velocity V_1:

$$z_1 = -0,0004V_1^2 + 0,1477V_1 + 17,728, \qquad (6)$$

$$z_2 = u_{12} z_1 .$$

The obtained numbers of the teeth are rounded off to the nearest integer.

2. The minimum distance between axes for the externally toothed tooth gearwheel, mm [5]:

$$a_{w\min} = K_a (u_{12} + 1) \sqrt[3]{\frac{T_1 K_{Hb}}{u_{12} y_{ba} [s]_H^2}} ,$$

where: $K_{H\beta}$ is a tentative rough value of the index which takes into account the irregularity of load distribution along the width of toothed ring:

when HB≤350

$$K_{Hb} = 0,994 \exp[0,0486 y_{ba} (u_{12} + 1)] ,$$

when HB>350,

$$K_{Hb} = 0,984 \exp[0,1057 y_{ba} (u_{12} + 1)] .$$

The determined value $a_{w\min}$ is rounded off to the nearest integer a_w.

3. Tooth gradient angle β, degr:

$$\beta = 0,0183L^2 - 3,0583L + 127,38,$$

where: L is the acceptable sound volume, dBA (according to the Ukrainian sanitary code CH 2.2.4/2.1.8.562-96 L = 50…85 dBA).

4. The module of the tooth gearwheel m_n, mm, is obtained as follows:

$$m_n = \frac{2a_w \cos b}{z_1 + z_2} .$$

The determined module value is agreed with a standard value and the real value of the distance between axes a_w of the tooth gearwheel is calculated.

5. Reference diameters of the driving gearwheel d_1 and the driven gearwheel d_2, as well as gear faces b_1 and b_2, mm, are calculated according to:

$$d_1 = \frac{m_n z_1}{\cos b}, \quad d_2 = \frac{m_n z_2}{\cos b},$$

$$b_2 = y_{ba} a_w ; \; b_1 = b_2 + 2 .$$

6. The adjustment of the diameter of the wheel slot d_{f1} for the tooth wheel-shafts with the shaft external diameter $d_в$, bearing the driving gearwheel:

$$d_{f1} = d_1 - 2,5m_n \geq d_{в1} . \qquad (4)$$

In the case, when according to the dependence (4) $d_{f1} < d_{в1}$, it is necessary to perform a recomputation, increasing module m_n or the number of teeth z_1.

7. Circular velocity in gears mesh, N:

$$F_t = \frac{2T_1 10^3}{d_1} .$$

8. Indexes taken into account during the calculations of contact hardiness and bend hardiness of gear teeth respectively: load distribution between the teeth $K_{H\alpha}$, $K_{F\alpha}$; adjusted load distribution along the face width $K_{H\beta}$, $K_{F\beta}$; tooth dynamic load K_{Hv}, K_{Fv}:

$$K_{Ha} = 0,53 n_{cm}^{0,33} V_1^{0,04} ,$$

$$K_{Fa} = \frac{4 + \left(\left[1,88 - 3,2\left(\frac{1}{z_1} + \frac{1}{z_2} \right) \right] \cos b - 1 \right)(n_{cm} - 5)}{4\left[1,88 - 3,2\left(\frac{1}{z_1} + \frac{1}{z_2} \right) \right] \cos b} ,$$

where: n_{cm} is the accuracy grade of the tooth gearwheel (n_{cm} = 5…9); V_1 is the circular velocity of the driving tooth gearwheel between 2,5…25 m/s:

$$K_{Hb} = 1 + \frac{7 \cdot 10^{-8} HB^{2,432} b_2}{d_1} ,$$

$$K_{Fb} = 1 + \frac{7,5 \cdot 10^{-8} HB^{2,432} b_2}{d_1} ,$$

$$K_{Hv} = 1,56 n_{cm}^{0,064} V_1^{0,029} HB^{-0,1} ,$$

$$K_{Fv} = 5,26 n_{cm}^{0,12} V_1^{0,11} HB^{-0,35} .$$

9. Specific design thrust forces, while the contact hardiness and strength of the tooth surface w_{Ht} and bend hardiness and strength w_{Ft} are calculated in accordance with the equation:

$$w_{Ht} = \frac{F_t}{b_2} K_{Ha} K_{Hb} K_{Hv} ,$$

$$w_{Ft} = \frac{F_t}{b_2} K_{Fa} K_{Fb} K_{Fv} .$$

10. Contact hardiness and strength of the teeth of the steel tooth gearwheels externally toothed at α_n = 20° is obtained as follows:

$$s_H = 487 \cos b \sqrt{\frac{w_{Ht}(u_{12} + 1)}{d_2 \left[1,88 - 3,2\left(\frac{1}{z_1} + \frac{1}{z_2} \right) \right] \cos b}} \leq [s]_H .$$

Designed contact hardiness is allowed in the range:

$$1,05\,[\sigma]_H \geq \sigma_H \geq 0,9\,[\sigma]_H.$$

In the case of the failure to meet this requirement, it is necessary to change the face width b_2 or the distance between axes a_w etc.

Maximum contact stresses are calculated as shown:

$$s_{H\max} = s_H \sqrt{k_n} \leq [s]_{H\max} = 2,8 s_T .$$

11. Bend hardness and strength of the teeth of the steel tooth gearwheels is obtained as follows:

- for the driving gearwheel:

$$s_{F1} = \frac{5,32\left(1 - \frac{b}{140^0} \right) w_{Ft}}{m_n \left(\frac{z_1}{\cos^3 b} \right)^{0,092}} \leq [s]_{F1} ,$$

- for the driven gearwheel σ_{F2} is determined under the condition of z_2 in comparison with $[\sigma]_{F2}$.

Maximum bend tensions are calculated by the major value σ_{F1} or σ_{F2}:

$$s_{F\max} = s_{Fi} k_n \le [s]_{F\max} = \frac{4,8HB}{S_F},$$

where: S_F is a safety margin (S_F = 1,4…1,7).

12. The dimentions of the key joint of gearwheels with shafts $d_{в1}$ and $d_{в2}$, mm are calculated in accordance with the formula:

$$l_i \ge \frac{2\cdot 10^3 T_1}{(0,042 d_{вi}^2 + 1,574 d_{вi})[s]_{зм}} + 0,271 d_{вi} + 0,93.$$

Determined dowel lengths l_1 and l_2 are agreed with standard boundary lengths according to the shaft diameter $d_{вi}$.

Groove depth in the datum bore in the hub t_{2i} is determined by the equation:

$$t_{2i} = 0,044 d_{вi} + 1,822.$$

Dowel width $b_{шi}$ is calculated by:

$$b_{шi} = 0,271 d_{вi} + 0,9301.$$

The value $b_{шi}$ is to be rounded up to the integer.

13. Gearwheel hub lengths $l_{м1}$ and $l_{м2}$ depend on the dowel lengths:

$$l_{м1} = l_1;\ l_{м2} = l_2.$$

14. Hub external diameter $d_{мi}$ is:

$$d_{мi} = \frac{4\cdot 10^3 T_1}{d_{вi} l_{мi} [s]_p} + d_{вi} + 2(0,044 d_{вi} + 1,822).$$

The value $d_{мi}$ is to be rounded up to the integer.

15. The internal diameter of the toothed ring $d_{з.в.i}$ is determined as:

$$d_{з.в.i} = 0,8 d_{fi}.$$

16. Plate thickness $b_д$ between the hub and the toothed ring is found as:

$$b_{дi} \ge \frac{10^3 T_1}{6 p\, d_{мi}^2}.$$

The value $b_{дi}$ is to be rounded up to the integer.

17. The weight of the steel tooth gearwheel M, kg is calculated as:

$$M_i = \frac{7,8\cdot 10^{-6} p}{4}\left(\begin{array}{l} b_i[0,5(d_{ai}^2 - d_{fi}^2) + (d_{fi}^2 - d_{з.в.i}^2)] + \\ +(d_{з.в.i}^2 - d_{мi}^2) b_{дi} + (d_{мi}^2 - d_{вi}^2) l_{мi} \end{array} \right),$$

$$d_{ai} = d_i + 2m_n;\ d_{fi} = d_i - 2,5m_n.$$

In the case when $0,5(d_{з.в.i} - d_{вi}) \le 20$, it is necessary to accept the condition $b_{дi} = b_i$.

18. Reliability costs of the gearwheel prototype $Q_П$ in UAH are determined according the equation:

$$Q_П = 1,97\cdot 10^{-4} M_i^{1,11} n_{cm}^{-1,,05} d_i^{1,01} l^{-0,9}, \quad (5)$$

where: λ is the failure rate, h^{-1}:

$$l = \frac{1}{T_B}.$$

T_B is the trouble-free life, h (for the tooth gear of the seventh accuracy grade T_B = 3600 hours [7]).

19. Production costs for the tooth gearwheel Q_B, UAH. The value Q_B is determined by the equation (5), when the values obtained after the calculations of the structural components of the tooth gearwheel, namely, the reference diameter, weight, set accuracy degree and failure rates are substituted in it.

20. Total operational costs Q_E in UAH are determined according to the equation:

$$Q_E = 6\cdot 10^{-7} T_E^2 + 0,01 T_E - 2,381,$$

where: T_E – set life span of the tooth gearwheel, h.

21. Cycle friquency $n_ц$, min^{-1}:

$$n_{цi} = \frac{6\cdot 10^4 V_1}{p\, d_i}.$$

Optimization mathematical model of the determining the structural components of the tooth gearwheel. The task set for developing the optimization mathematical model refers to the discrete programming. The Monte-Carlo method is suitable for its solving. Constraint region, defined with the variable factors and optimization criteria, is enclosed into the *n*-dimentional parallelepiped, where the investigations are carried out.

The optimization mathematical model can be represented as follows:

aim function:

$$Q_{Bi} \Rightarrow \min, \quad (6)$$

by the criterion:

$$|Q_{ei} - Q_{Hi}| \Rightarrow \min, \quad (7)$$

limited by:

$$\left. \begin{array}{c} a_1 \le y_{ba} \le b_1; \\ a_2 \le L \le b_2; \\ a_3 \le T_B \le b_3; \\ a_4 \le T_E \le b_4; \\ a_5 \le V_1 \le b_5; \end{array} \right\}, \quad (8)$$

$$p \ge [p]. \quad (9)$$

where: a_1, a_2, …, a_5 are minimal values of operational and structural factors, which are determined at the stage of developing the technical development task; b_1, b_2, …, b_5 are the maximum values of the operational and structural factors; p is the probability of hitting the investigation point into the range of the feasible solutions, surrounded by the *n*-dimentional parallelepiped determined by the limitations (8) and the criterion (7); $[p]$ is the allowable value of the probability of hitting the investigation point into the range of the feasible solutions.

While using the Monte-Carlo method for solving the task, it is necessary to create the sequence of pseudorandom numbers m_i in the range from 0 to 1,0 using special software. Pseudorandom numbers should be transformed into the factor values (8) according to the equation:

$$x_i = a_j + m_i (b_j - a_j),$$

where: x_i is the factor value at the *i*-stage of the task solving process; m_i is the pseudorandom number at this stage; a_j, b_j are correspondingly minimum and maximum values of *j*- limitation according to the equations (8).

For solving this optimization task an algorithm has been developed including all necessary calculations for the tooth gear and the simultaneous definition of the optimization criteria: 1) design algorithm determining the distance between axes a_w, module value m_n and design parameters of the toothed ring; 2) testing active tooth surfaces for their contact hardiness; 3) testing active tooth surfaces for their contact strength; 4) testing teeth for the bend hardiness; 5) testing teeth for the bend strength; 6) structural dimensions of the key joint, the hub, the internal diameter of the toothed ring and plate thickness; 7) weight of the tooth gearwheel.

It is necessary to include into the algorithm the determinations:

$$T_{сл1} = f(\sigma_H),\ T_{сл2} = f(\sigma_{F1})\ \text{i}\ T_{сл3} = f(\sigma_{F2}),$$

taking the least of values obtained and taking it into account during the calculations of optimization criteria.

On the basis of the algorithm an application package has been developed which has contributed to the solving the task set in the range of the feasible solutions, limited by the aim function (6), the criterion (7) and the limitations (8). During running the program the probability p of hitting the investigation point into the range of the feasible solutions is determined as:

$$p = \frac{w}{N},$$

where: w is the general number of computer operational cycles, having hit into the range of the feasible solutions; N is the general number of computer operational cycles.

CONCLUSIONS

1. The method of multiparametric synthesis of the basic structural components of the cylindrical tooth gear using the optimization multicriterion mathematical model, which makes it possible to reduce many times the period of designing the production stage. With the use of the computer the duration of the synthesis of the parameters of the structural components of the tooth gearwheel does not exceed 20…40 s.

2. For the realization of the method of multiparametric synthesis of the structural components the Monte-Carlo method has been used, which considerably simplifies the optimization process due to the use of the pseudorandom numbers.

3. Using the set criteria allows ensuring the given life span duration of the designed cylindrical tooth gear.

REFERENCES

1. **Nechepaev V.G. 2011.** Development of and introduction of innovative model of general engineering preparation of specialists for the mountain metallurgical complex of the Donetsk of region / Engineering of Science/ Materials of 13th regional scientifically - methodical conference. it is of Donetsk : DonNTU, 6-10. (Ukraine)
2. **Kirichenko A.F. and Berejnoy V.A. 2004**. Prospects of improvement of work of evolventen transmissions. «Collection of the National technical university "Kharkivs polytechnic institute"». - Kharkiv: NTU "KPI". - Prod. 31, 82-88. (Ukraine)
3. **Shishov V.P., and Tkach P.N., and Revjakina O. A. and Muhovaty A.A. 2002**. Synthesis of the cylindrical gearings with high loading ability. «Collection of the East Ukrainian National university the name of V. Dals», №10(56), Luhansk: ENU the name of. B. of V. Dals, 247-254. (Ukraine)
4. **Shishov V.P. and Tkach P.N. 2003.** Differential equalizations for the synthesis of gearings with the biconvex-concave points «Lifting-transport technique», №4,25-32. (Ukraine)
5. **Hulida E.M., and Dsjuba L.F. and Olkhovyji I.M. 2007.** Applied mechanics. – Lviv: Svit. – 284. (Ukraine)
6. **Hulida E.M. 1983**. Management reliability of cylindrical gear-wheels. it is. – Lviv: Higher school. – 136. (Ukraine)
7. **Anurjev V.I. 1980**. Reference book of machine designer-builder: In 3 т. of T. 3. it is a 5th publ., proces. and addit. – M.: Engineer. – 557. (Russia)
8. **Push V.E., Beljaev V.H., Havrjushyn A.A. 1985.** Metal-cutting machine tools. - M.: Engineer. - 256. (Russia)
9. **Hulida E.M. 1977**. Technology of finishings operations of treatment of teeth of cylindrical wheels. – Lviv: Vyshcha shkola, 168. (Ukraine)
10. **Kirichenko A.F. and Berejnoy V.A. 2004**. Adjusting of inflexibility of evolventy of direct teeth transmissions. Collection of the National technical university "Kharkivs polytechnic institute". - Kharkiv: NTU "KhPI". - Prod. 28, 142-149. (Ukraine)
11. **Kirichenko A.F. and Berejnoy V.A. 2005**. Influence of modification as проточки along the top of tooth on his flexural inflexibility. Collection of the National technical university "Kharkivs polytechnic institute". - Kharkiv: NTU "KPI". - Prod. 40, 131-134. (Ukraine)
12. **Shishov V.P. and Tkach P.N. and Pymchenko A.V. 2002.** Generalization of geometry of teeth of cylindrical wheels, cut a lath instrument. Collection of the East Ukrainian National university the name of V. Dals, №7(53). – Luhansk: ENU the name of. B. of V. Dals, 117-122. (Ukraine)
13. **Shishov V.P. and Tkach P.N. 2004.** A choice of rational parameters of transmissions is with two protuberant and outbowed teeth. Engineering science, № 2, 25-32. (Ukraine)
14. **Pavlyshe V.T. 2003.** Bases of constructing and calculation of details of machines. it is. Lvov: Playbill, 560. (Ukraine)
15. **Pukach P. and Kuzio I. and Nytrebych Z. 2013.** Influence of some speed parameters on dynamics of nonlinear flexural vibrations of a drill column. Econtechmod an international quarterly journal. Vol. 2, № 4, 61-66. (Poland)
16. **Hulida E.M. 1980**. Technological government bases reliability of cylindrical gear-wheels. Announcer of the Lvov polytechnic institute, № 146, 30-33. (Ukraine)
17. **Shishov V.P. and Tkach P.N. and Skljar J.A. 2009.** Geometrical kinematics criterion for the synthesis of the highly loaded cylindrical direct teeth transmissions. Lifting-transport technique, № 3 (31), 109-114. (Ukraine)
18. **Kuzio I. V. and Vasyljeva O.E. 1999.**
19. Influence of construction factors is on durability of gearings. Engineering science, № 6, 50-53. (Ukraine)
20. **Kuzio I. V. and Vasyljeva O.E. 2000.** Influence of dynamic overloads of gearings is on their durability. Announcer. Optimization of production processes and technical control is in an engineer and instrument-making, № 394, 75-80. (Ukraine)
21. **Shishov V.P. and Tkach P.N. and Revjakina O.A. and Skljar J.A. 2009.** About tensions of bend at foundation of зубьев with the generalized initial contour. Vesnik of national technical university of «KhPI», № 19, 160-166.(Ukraine)