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MATHEMATICAL MODEL AND OPTIMIZATION ALGORITHM BY MASS OF TRACKED LOAD-CARRIER/PRIME MOVER MT-LB TRANSMISSION

USTYNENKO Oleksandr[1], BONDARENKO Oleksij[1], KLOCHKOV Illia[1], PROTASOV Roman[2*], BOŠANSKÝ Miroslav[2], ANDRIENKO Sergij[3]

[1] National Technical University "Kharkiv Polytechnic Institute", Institute of Education and Science in Mechanical Engineering and Transport, Department of Theory and Computer-Aided Design of Mechanisms and Machines, Kyrpychova str. 2, 61002 Kharkiv, Ukraine
[2] Slovak University of Technology in Bratislava, Faculty of Mechanical Engineering, Institute of Transport Technology and Engineering Design, Nám. Slobody 17, 812 31 Bratislava, Slovakia,
e – mail: roman.protasov@stuba.sk
[3] Kharkiv National Automobile and Highway University, Mechanical Faculty, Department of Engineering and Computer Graphics, Yaroslava Mudrogo str. 25, 61002 Kharkiv, Ukraine

Abstract: Solving the problem of optimization for transmission load-carrier/prime mover MT-LB improves the mass characteristics of the machine. Mathematical model of optimization was constructed, namely: optimization objective function was built; variables planning are defined; a system of constraints was constructed. The solution is based on probing the parameter space, where points of the *LPτ*-sequence are used as test points in a single multidimensional cube. Also, the algorithm for optimal design of the transmission has been developed.

KEYWORDS: optimization, multipurpose load-carrier/prime mover MT-LB, transmission, objective function, variables planning, constraints on the variables planning, *LPτ*-sequence, algorithm.

1 Introduction

Modern transport engineering makes more and more stringent requirements for the mass and size characteristics of machines. Of course, the weight of the machine as a whole is significantly affected by the weight of its individual units, such as the transmission. Therefore, the maximum possible reduction in the mass of the latter is an urgent scientific and technical task.

One of the promising ways in this direction is to solve the problem of optimal design of transmissions [1] by the criterion of minimum mass. This problem occurs in two cases:

1. Creation of new machines and, accordingly, a new transmission. In this case, the optimal design is limited by technical and technological capabilities of production and transmission requirements.

2. Modernization of the existing machine, for example, when replacing the engine with a more powerful one, increasing the mass and (or) maximum speed, etc. In this case, the optimal design is further limited by the existing dimensions of the engine-transmission compartment of the machine.

At present, the task of modernization is especially important for military tracked and wheeled vehicles of the Armed Forces of Ukraine. This is due to the high cost of production of new equipment (modernization of existing ones is much cheaper) and long production cycle [2]. And the current situation in eastern Ukraine, caused by Russian aggression, requires

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the supply of armored vehicles to the troops as soon as possible, which meets the latest requirements.

The majority of the caterpillar equipment of the Armed Forces of Ukraine consists of lightly armored tracked vehicles. Mechanical double-flow transmission and turning mechanism (TTM) are widely used in tracked vehicles transmissions to control the gear ratio and rotation during machine movement.

One of the most common in Ukraine and other countries lightly armored tracked vehicles with mechanical double-flow TTM – light multi-purpose tracked load-carrier/prime mover MT-LB. It was adopted in 1964 and produced (by Kharkiv Tractor Plant, in Poland and Bulgaria) in the amount of approximately 55,000 machines, of which approximately 7,500 are still in operation [3]. Today it no longer meets modern tactical and technical characteristics in terms of engine power and average speeds. In this regard, in recent years, many options have been proposed for its modernization, most of which is to replace the engine with a more powerful one. At the same time there is a problem of overload of other units, primarily transmission. Simply increasing its load capacity by increasing the size is almost impossible, which is due to the above-mentioned limitations of the dimensions of the engine-transmission compartment.

The way out of this situation is to optimally design a new transmission while ensuring its load capacity, durability and requirements for placement in the existing engine-transmission compartment.

General optimization problems in engineering and for transmissions are considered, for example, in [1, 4, 5]. Similar issues of optimization of reducers, transmissions and cars gearboxes were studied in [6, 7]. Of particular note is the work [7], which solved the problem of optimal and rational design of coaxial stepped mechanical drive on the example of a three-shaft gearbox of the car. The solution was performed on the basis of probing the parameter space, where $LP\tau$-sequence points are used as test points in a single multidimensional cube [8]. This approach was chosen by the authors for use in further research.

The purpose of this study is to develop a mathematical model of optimization of the transmission mass for a light multi-purpose tracked load-carrier/prime mover MT-LB, as well as to build an algorithm for software implementation of this problem.

2 The sequence of solving the problem

As you know [4], the solution of any optimization problem consists of the following stages:

1. Statement of the optimization problem, which consists of the construction of the objective function, the definition of variables planning and the formation of a system of constraints.

2. Selection or development of methods for solving the optimization problem.

3. Construction of methods and algorithms for solving the problem.

4. Execution of test and verification calculations to confirm and evaluate the obtained theoretical results.

The kinematic scheme of the transmission is shown in Fig. 1 [9]. It is made according to a double-flow scheme, the gearbox is combined with the turning mechanism into a single TTM. Onboard gears – planetary single-stage.

In gearbox, additional gear, summary planetary gear and onboard gear used straight teeth. Main gear is bevel with spiral teeth.

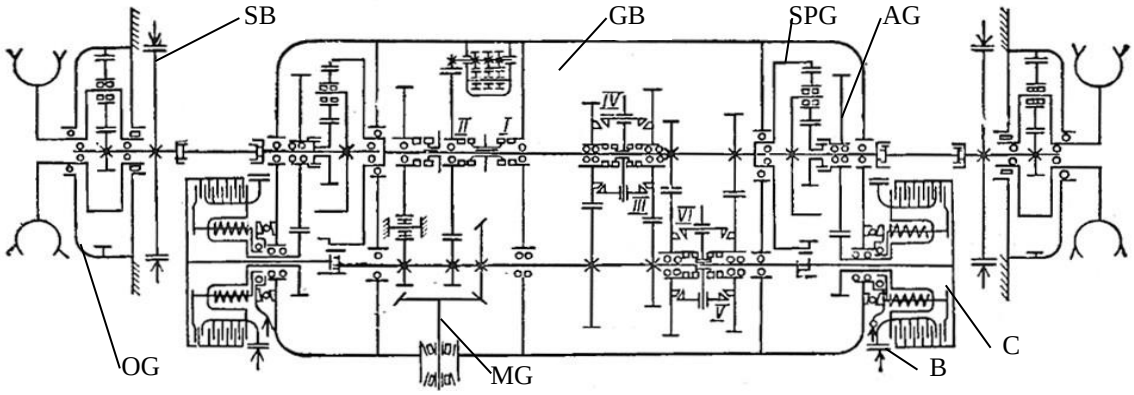


Fig. 1 Kinematic scheme of the MT-LB load-carrier/prime mover transmission:
MG – main gear; GB – gearbox; AG – additional gear;
C – TTM clutch; B – TTM brake; SPG – summary planetary gear;
SB – stop brake; OG – onboard gear

3 Objective function and variables planning

Let's construction of the objective function. The criterion of optimality for our problem will be formulated quite simply: the mass of the transmission must take the minimum possible value to take into account all constraints. Then the objective function writing as:

$$F_M = \sum M \to \min, \tag{1}$$

where $\sum M$ – is the total mass of the main elements of the transmission, kg,

$$\begin{aligned}\sum M = &\sum_{i=1}^{2} M_{MGWi} + M_{MGS} + \sum_{i=1}^{n_{GBW}} M_{GBWi} + \sum_{i=1}^{2} M_{GBSi} + \sum_{i=1}^{n_{GBSc}} M_{GBSci} + 2\sum_{i=1}^{2} M_{AGWi} + \\ &+2M_{CS} + 2M_{CB} + 2\sum_{i=1}^{n_{SPGW}} M_{SPGWi} + 2\sum_{i=1}^{n_{SPGS}} M_{SPGSi} + 2M_{SPGC} + 2M_{CS} + \\ &+2M_{SB} + 2\sum_{i=1}^{n_{OGW}} M_{OGWi} + 2\sum_{i=1}^{n_{OGS}} M_{OGSi} + 2M_{OGC} + \sum_{i=1}^{n_{Br}} M_{Bri} + M_{TTMCs} + 2M_{OGCs},\end{aligned} \tag{2}$$

where $\sum_{i=1}^{2} M_{MGWi}$ – total mass of the MG wheels; M_{MGS} – mass of the MG input shaft; $\sum_{i=1}^{n_{GBW}} M_{GBWi}$ – total mass of the GB wheels, n_{GBW} – number of the GB wheels ($n_{GBW} = 2(k-1)+1$, where k – number of forward and reverse gears, the unit takes into account the parasitic reverse wheel, and –1 takes into account the absence of gear wheels of the first gear, which is realized by stopping the SPG ring wheels); $\sum_{i=1}^{2} M_{GBSi}$ – total mass of the GB shafts; $\sum_{i=1}^{n_{GBSc}} M_{GBSci}$ – total mass of the GB synchronizers, n_{GBSc} – number of the synchronizer; $2\sum_{i=1}^{2} M_{AGWi}$ – total mass of the AG wheels; $2M_{CS}$ – total mass of the clutch

shafts; $2M_{CB}$ – total mass of the clutch and brakes; $2\sum_{i=1}^{n_{SPGW}} M_{SPGWi}$ – total mass of the SPG wheels, n_{SPGW} – number of the SPG wheels; $2\sum_{i=1}^{n_{SPGS}} M_{SPGSi}$ – total mass of the SPG shafts and axis, n_{SPGS} – number of the SPG shafts and axis; $2M_{SPGC}$ – total mass of the SPG carriers; $2M_{CS}$ – total mass of the cardan shafts; $2M_{SB}$ – total mass of the stop brakes; $2\sum_{i=1}^{n_{OGW}} M_{OGWi}$ – total mass of the OG wheels, n_{OGW} – number of the OG wheels; $2\sum_{i=1}^{n_{OGS}} M_{OGSi}$ – total mass of the OG shafts and axis, n_{OGS} – number of the OG shafts and axis; $2M_{OGC}$ – total mass of the OG carriers; $\sum_{i=1}^{n_{Br}} M_{Bri}$ – total mass of the transmission bearings, n_{Br} – number of the transmission bearings; M_{TTMCs} – mass of the TTM crankcase; $2M_{OGCs}$ – total mass of the OG crankcase.

The masses of the wheels are calculated directly through the values of the variables planning to take into account the system of constraints. Masses of other parts and assemblies – according to calculations for strength, durability and (or) rigidity.

Geometric parameters of gears are chosen as variables planning [7, 10] – gear modules m, teeth numbers z and helix angle of teeth β, namely: for MG – m_{nMG}, z_{MG1}, z_{MG2}, β_{mMG}; for GB – $m_{GB\mu}$, $z_{GB\mu,1}$ ($\mu = 2,\ldots,k-1$ – forward gear serial number); for AG – m_{AR}, z_{AG1}, z_{AG2}; for SPG – m_{SPG}, z_{SPGs}, z_{SPGr}; for OG – m_{OG}, z_{OGs}, z_{OGr}.

In order to reduce the dimension of the problem, the teeth numbers of the GB driven wheels $z_{GB\mu,2}$ are not variables planning, but are calculated through the teeth numbers of the drive wheels $z_{GB\mu,1}$ and preset transmission gear ratios $i_{T\mu}$.

4 Constraints on variables planning

The following is a sequence of constraints adopted in ascending order of the approximate volume of mathematical calculations, and it is noted that some constraints can be tested only after others [10].

1) Gear modules for cylindrical gears are standardized, they are selected from the following series:

$$m_n = 2;\ 2.25;\ 2.5;\ 2.75;\ 3;\ 3.5;\ 4;\ 4.5;\ 5;\ 5.5;\ 6;\ldots;\ m_{n\max}. \tag{3}$$

In a bevel gear, the normal module can be a non-standard value, so for her:

$$m_{n\min} \le m_n \le m_{n\max}. \tag{4}$$

2) The teeth numbers of the wheels must take integer values (must be natural – N), also limited to the upper $z_{\max}$ and lower values $z_{\min}$. Then:

$$\begin{gathered}\{z_{MG1},\ z_{MG2},\ z_{GB2,1},\ldots,z_{GBk-1,1},\ z_{GB2,2},\ldots,z_{GBk-1,2},\\ z_{AG1},\ z_{AG2},\ z_{SPGs},\ z_{SPGp},\ z_{SPGr},\ z_{OGs},\ z_{OGp},\ z_{OGr}\} \in N;\\ z_{\min} \le \{z_{MG1},\ z_{MG2},\ z_{GB2,1},\ldots,z_{GBk-1,1},\ z_{GB2,2},\ldots,z_{GBk-1,2},\\ z_{AG1},\ z_{AG2},\ z_{SPGs},\ z_{SPGp},\ z_{SPGr},\ z_{OGs},\ z_{OGp},\ z_{OGr}\} \le z_{\max}.\end{gathered} \tag{5}$$

3) The teeth numbers of the wheels for the planetary gears must meet the conditions of alignment, assembly and proximity, taking into account the number of planetary pinions n_p,

$$z_p = 0.5\left(z_r - z_s\right); \quad \left(z_s + z_r\right)/n_p = \text{integer}; \quad \left(z_s + z_p\right)\sin\left(\pi/n_p\right) > \left(z_p + 2\right). \tag{6}$$

4) The helix angle of teeth for the bevel wheels must be within from $\beta_{m\min}$ to $\beta_{m\max}$, then

$$\beta_{m\min} \le \beta_m \le \beta_{m\max}. \tag{7}$$

5) The axis distances of the wheels for the GB forward gears and the AG must be equal, i.e. taking into account the variables planning for each gear:

$$0.5m_{GB2}\left(z_{GB2,1} + z_{GB2,2}\right) = ... = 0.5m_{GBk-1}\left(z_{GBk-1,1} + z_{GBk-1,2}\right) = 0.5m_{AG}\left(z_{AG1} + z_{AG2}\right). \tag{8}$$

6) Estimated transmission gear ratios $i^e_{T\mu}$ must satisfy the condition associated with integer values of the teeth number,

$$i_{T\mu}\left(1 - \Delta i_T/100\right) \le i^e_{T\mu} \le i_{T\mu}\left(1 + \Delta i_T/100\right), \quad \mu = 1,...,k-1, \tag{9}$$

where Δi_T – error of the transmission gear ratio specified by the designer;

$$i^e_{T1} = i_{MG}\cdot i_{AG}\cdot i_{OG}\left(1 + \left|p_{SPG}\right|\right) = \frac{z_{MG2}}{z_{MG1}}\cdot\frac{z_{AG2}}{z_{AG1}}\cdot\left(1 + \frac{z_{OGr}}{z_{OGs}}\right)\left(1 + \frac{z_{SPGr}}{z_{SPGs}}\right);$$

$$i^e_{T\mu} = \frac{i_{MG}\cdot i_{GB\mu}\cdot i_{AG}\cdot i_{OG}\left(1 + \left|p_{SPG}\right|\right)}{\left|p_{SPG}\right|\cdot i_{AG} + i_{GB\mu}} = \frac{\frac{z_{MG2}}{z_{MG1}}\cdot\frac{z_{GB\mu 2}}{z_{GB\mu 1}}\cdot\frac{z_{AG2}}{z_{AG1}}\cdot\left(1 + \frac{z_{OGr}}{z_{OGs}}\right)\left(1 + \frac{z_{SPGr}}{z_{SPGs}}\right)}{\frac{z_{SPGr}}{z_{SPGs}}\cdot\frac{z_{AG2}}{z_{AG1}} + \frac{z_{GB\mu 2}}{z_{GB\mu 1}}}, \tag{10}$$

$$\mu = 2,...,k-1.$$

7) Constraints on gear ratios for transmission teeth pairs. According to the requirements of the overall ratio for wheels we have:

$$\left|i_{\min}\right| \le \left|i\right| = \left|\omega_1/\omega_2\right| = z_2/z_1 \le \left|i_{\max}\right|. \tag{11}$$

For planetary gears, the constraint (11) is written through the internal gear ratio p,

$$\left|p_{\min}\right| \le \left|p\right| = z_r/z_s \le \left|p_{\max}\right|. \tag{12}$$

8) Providing for the teeth of contact and bending endurance [7, 11], as well as overload strength, namely,

a) for MG bevel gear:

$$\begin{gathered}\sigma_{H\,MG} \le \sigma_{HP\,MG},\ \sigma_{HM\,MG} \le \sigma_{HPM\,MG};\\ \sigma_{F\,MG1(2)} \le \sigma_{FP\,MG1(2)},\ \sigma_{FM\,MG1(2)} \le \sigma_{FPM\,MG1(2)};\end{gathered} \tag{13}$$

b) for GB and AG cylindrical gears:

$$\begin{gathered}\sigma_{H\,GB\mu} \le \sigma_{HP\,GB\mu},\ \sigma_{HM\,GB\mu} \le \sigma_{HPM\,GB\mu};\\ \sigma_{F\,GB\mu 1(2)} \le \sigma_{FP\,GB\mu 1(2)},\ \sigma_{FM\,GB\mu 1(2)} \le \sigma_{FPM\,GB\mu 1(2)};\\ \sigma_{H\,AG} \le \sigma_{HP\,AG},\ \sigma_{HM\,AG} \le \sigma_{HPM\,AG};\\ \sigma_{F\,AG1(2)} \le \sigma_{FP\,AG1(2)},\ \sigma_{FM\,AG1(2)} \le \sigma_{FPM\,AG1(2)};\end{gathered} \tag{14}$$

c) for SPG and OG cylindrical planetary gears:

$$\begin{gathered}
\sigma_{H\,SPGs\text{-}p} \le \sigma_{HP\,SPGs\text{-}p},\ \sigma_{H\,SPGp\text{-}r} \le \sigma_{HP\,SPGp\text{-}r},\\
\sigma_{HM\,SPGs\text{-}p} \le \sigma_{HPM\,SPGs\text{-}p},\ \sigma_{HM\,SPGp\text{-}r} \le \sigma_{HPM\,SPGp\text{-}r};\\
\sigma_{H\,OGs\text{-}p} \le \sigma_{HP\,OGs\text{-}p},\ \sigma_{H\,OGp\text{-}r} \le \sigma_{HP\,OGp\text{-}r},\\
\sigma_{HM\,OGs\text{-}p} \le \sigma_{HPM\,OGs\text{-}p},\ \sigma_{HM\,OGp\text{-}r} \le \sigma_{HPM\,OGp\text{-}r};\\
\sigma_{F\,SPGs} \le \sigma_{FP\,SPGs},\ \sigma_{F\,SPGp} \le \sigma_{FP\,SPGp},\ \sigma_{F\,SPGr} \le \sigma_{FP\,SPGr},\\
\sigma_{FM\,SPGs} \le \sigma_{FPM\,SPGs},\ \sigma_{FM\,SPGp} \le \sigma_{FPM\,SPGp},\ \sigma_{FM\,SPGr} \le \sigma_{FPM\,SPGr};\\
\sigma_{F\,OGs} \le \sigma_{FP\,OGs},\ \sigma_{F\,OGp} \le \sigma_{FP\,OGp},\ \sigma_{F\,OGr} \le \sigma_{FP\,OGr},\\
\sigma_{FM\,OGs} \le \sigma_{FPM\,OGs},\ \sigma_{FM\,OGp} \le \sigma_{FPM\,OGp},\ \sigma_{FM\,OGr} \le \sigma_{FPM\,OGr},
\end{gathered} \tag{15}$$

where σ_H, σ_F and σ_{HM}, σ_{FM} – contact and bending stresses under nominal and maximum loads; σ_{HP}, σ_{FP} and σ_{HPM}, σ_{FPM} – the corresponding permissible stresses.

10) Gear width factors $\psi_{bd} = b_w / d_1$ for cylindrical gears, and $\psi_m = b_w / R_m$ for bevel gears should not exceed the maximum values,

$$\psi_{bd} \le \psi_{bd\,\max};\quad \psi_m \le \psi_{m\max}. \tag{16}$$

Effective face width b_w is determined from the contact endurance condition.

11) Dimensional constraints for crankcase of TTM and OG:

$$L_{TTM} \le L_{TTM\,\max},\ B_{TTM} \le B_{TTM\,\max},\ H_{TTM} \le H_{TTM\,\max};\ D_{OG} \le D_{OG\,\max},\ B_{OG} \le B_{OG\,\max}, \tag{17}$$

where $L_{TTM\,\max}$, $B_{TTM\,\max}$, $H_{TTM\,\max}$, $D_{OG\,\max}$, $B_{OG\,\max}$ – the corresponding maximum possible dimensions from the condition of TTM and OG placement in the existing engine-transmission compartment.

5 Method of solving the optimal design problem

As suggested, a method of probing the parameter space, where the test points in a single multidimensional cube are *LPτ*-sequence points [8], was chosen. The method allows to operate with a significant number of parameters (up to 51) and provides a sufficiently large number of evenly distributed test points (up to 2^{20}). Another important argument is that *LPτ*-sequences are the best evenly distributed sequences known today [8].

Here are the main calculated dependences [8] concerning uniformly distributed sequences.

If the points Q_i with cartesian coordinates $(q_{i,1},...,q_{i,n})$ is a uniformly distributed sequence in K^n (single *n*-dimensional cube), then points A_i with cartesian coordinates $(\alpha_{i,1},...,\alpha_{i,n})$, where if $j = 1, 2,\ldots, n$ (*n* – number of variables planning),

$$\alpha_{i,j} = a_j + \left(b_j - a_j\right)\cdot q_{i,j} \tag{18}$$

is a uniformly distributed sequence in a parallelepiped *P* (*n*-dimensional parallelepiped) consisting of points $(\alpha_1,...,\alpha_n)$, the coordinates of which satisfy the inequalities $a_j \le \alpha_j \le b_j$.

Cartesian coordinates $q_{i,j}$ for *LPτ*-sequence calculated by an arithmetic algorithm. This algorithm is based directly on the calculations of the numerators table $r_j^{(l)}$. For a given number for *i*-th point first calculate the auxiliary value *m*, and then for each *j*-th parameter (serial number of variable planning) calculate the coordinate $q_{i,j}$,

$$q_{i,j} = \sum_{k=1}^{m} 2^{-k+1} \left\{ \frac{1}{2} \sum_{l=k}^{m} \left[2\left\{ i^{-l} \right\} \right] \cdot \left[2\left\{ r_j^{(l)} 2^{k-1-l} \right\} \right] \right\}, \qquad (19)$$

where $m = 1 + [\ln i / \ln 2]$; $[\ldots]$ – integer part, $\{\ldots\}$ – fractional part of the corresponding value.

6 Algorithm of optimal design

Let's combine the above methods with the data input-output, compiling a table of possible parameters combinations, finding the best option (test point that corresponds to the minimum objective function value) and combine them to a single algorithm, which is presented in Fig. 2.

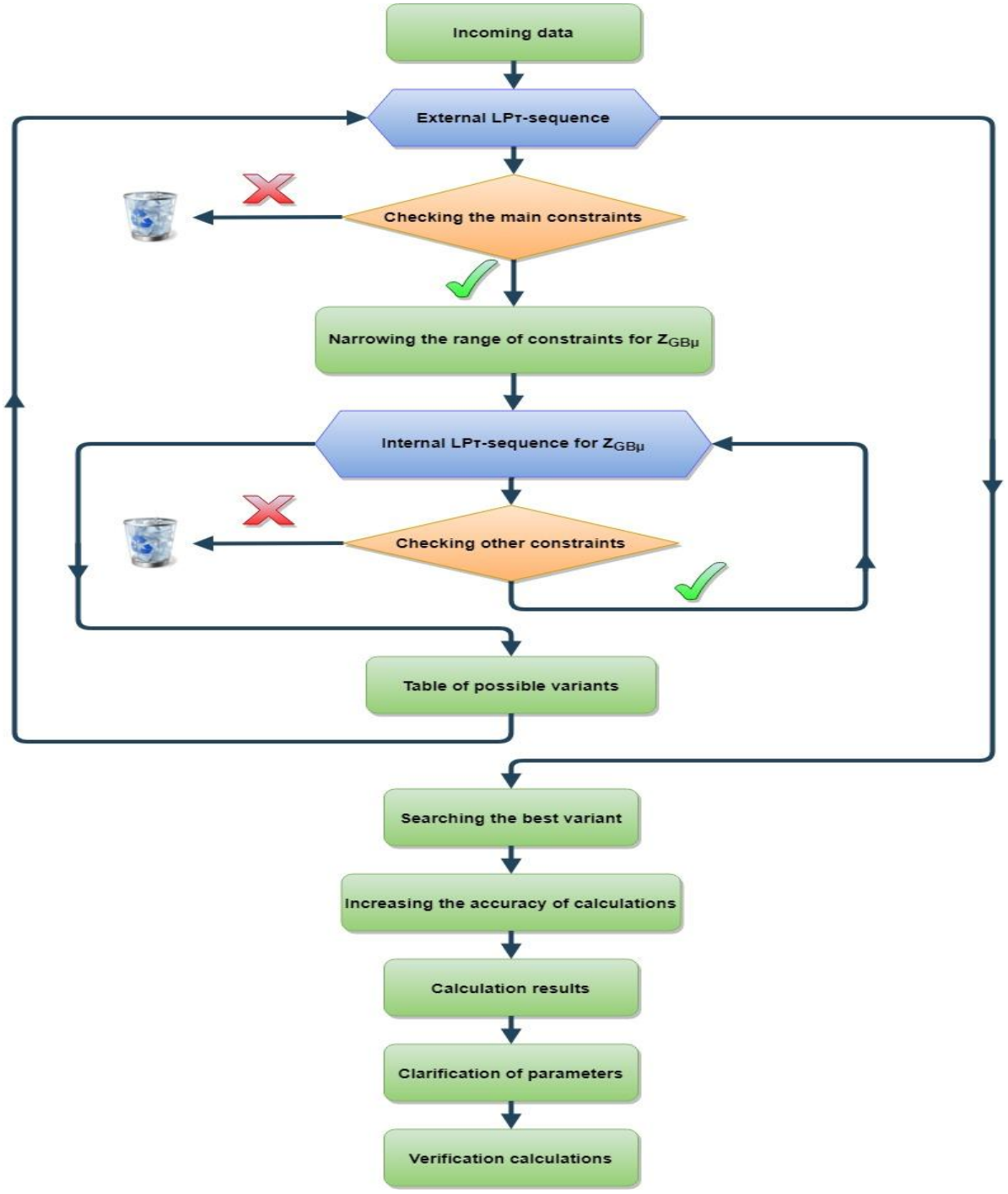


Fig. 2 Scheme of the optimal design algorithm
for the MT-LB load-carrier/prime mover transmission

A feature of this algorithm is the presence of external and internal *LPτ*-sequences. In the outer loop, an *LPτ*-sequences are generated for all variables planning except $z_{GB\mu,1}$. In inner – *LPτ*-sequences are generated for $z_{GB\mu,1}$ considering the narrowing of the ranges $z_{GB\mu,1\min}$, $z_{GB\mu,1\max}$. To do this, after checking the basic restrictions [12]:

1. GB gear ratios $i_{GB\mu}$ are calculated at the test point,

$$\left|i_{GB\mu}\right| = \frac{z_{GB\mu,2}}{z_{GB\mu,1}} = \left| \frac{i_{T\mu} \cdot z_{AG2} \cdot z_{MG1} \cdot z_{OGs} \cdot z_{SPGe}}{z_{AG2} \cdot z_{MG2} \left(z_{SPGs} + z_{SPGr}\right)\left(z_{OGs} + z_{OGr}\right) - i_{T\mu} \cdot z_{AG1} \cdot z_{MG1} \cdot z_{OGs} \cdot z_{SPGs}} \right|. \quad (20)$$
$$\mu = 2, ..., k-1.$$

2. Limit of teeth numbers $z_{GB\mu,1\min}$ and $z_{GB\mu,1\max}$ are defined starting with the second gear. If $|i_{GB\mu}| \geq 1$,

$$z_{GB\mu,1\min} = z_{\min}, \; z_{GB\mu,1\max} = z_{\max} / |\, i_{GB\mu} \,|; \quad (21)$$

if $|i_{GB\mu}| < 1$,

$$z_{GB\mu,1\min} = z_{\min} / |\, i_{GB\mu} \,|, \; z_{GB\mu,1\max} = z_{\max}. \quad (22)$$

3. An internal *LPτ*-sequences for $z_{GB\mu,1}$ are generated, and are calculated $z_{GB\mu,2}$,

$$z_{GB\mu,2} = z_{GB\mu,1} \cdot |\, i_{GB\mu} \,|. \quad (23)$$

This construction of the algorithm reduces the calculation time by narrowing the constraints for the teeth numbers of the GB wheels $z_{GB\mu,1}$.

If the test point passes all the conditions, it falls into the table of possible options. The table is a two-dimensional array in which each point number corresponds to a set of variables planning and values of the objective function. The search for the best option is carried out by sorting, for example, the method of insertion.

In the process of increasing the accuracy of calculations, gradually reduce the errors of equality for the axis distances of the GB and AG gears. If necessary, it is also possible to reduce transmission gear ratio error.

At the stage of refining the parameters it is possible to ensure the exact equality for the axis distances of the GB and AR gears by calculating the required total addendum modification coefficient $x_\Sigma = x_1 \pm x_2$ for each gear pair.

7 Results and discussions

Based on the proposed algorithm, a software for optimal design was developed for of load-carrier/prime mover MT-LB transmission. Its implementation was carried out in the software environment of PascalABC.NET.

Calculations were performed for the basic engine (240 hp) and for 4 engines with 300 to 448 hp. For example, approximate crankcase contours of the optimized transmission for the basic engine and the engine DEUTZ AG BF6M 1015 Typ-16 presented in Fig. 3. The results of calculations for all engines are given in Table 1.

Analysis of the results in Table 1 allows us to conclude that a significant reduction of the weight and dimensions for the transmission has been achieved compared to the basic one. At the same time, the specific power of the load-carrier/prime mover increases respectively on 14.6%, 9%, 0.3%, 1.1%, as well as on 16.4% for the basic engine in 240 hp.

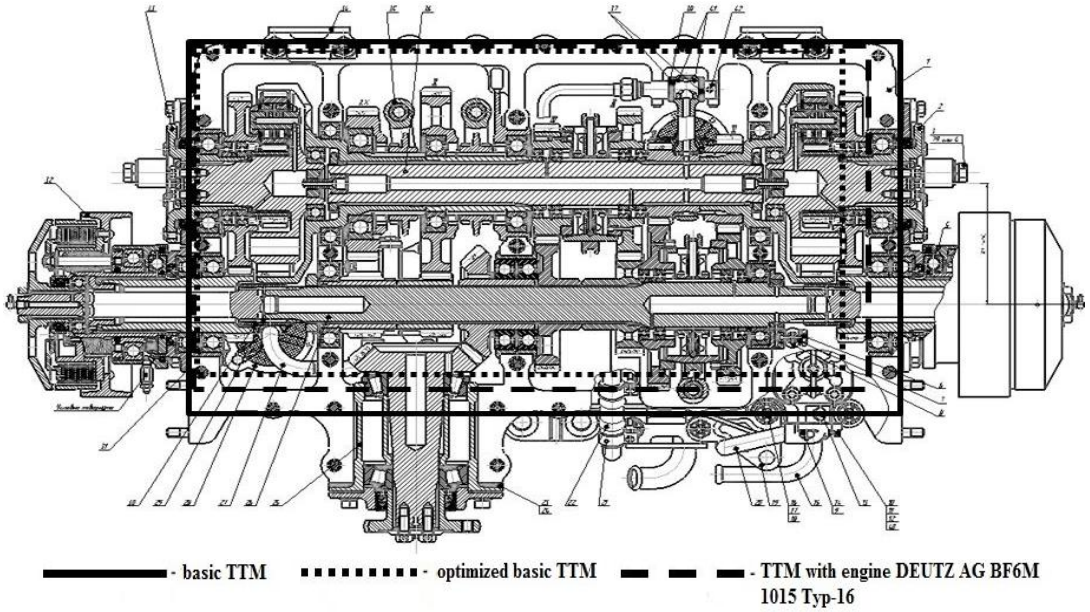


Fig. 3 Approximate crankcase contours of the basic and the optimized transmission

Table 1. The results of a numerical experiment

Engine (Power, hp)	Specific Power, hp / t	Transmission Mass *M*, kg	TTM Length *L*, mm	TTM Width *B*, mm	TTM Height *H*, mm
JaMZ-238 basic engine with basic transmission (240)	*24.74*	*533.2*	*1285*	*692*	*412*
JaMZ-238 (240)	24.97	445.5	1257	602	356
JaMZ-238 turbocharged (300)	31.18	455.3	1264	603	356
DEUTZ AG BF6M 1015 Typ-16 (326)	33.78	485.4	1264	645	383
DEUTZ AG BF6M 1015C (408)	42.07	531.5	1268	677	400
DEUTZ AG BF6M 1015CP (448)	46.21	527.3	1265	677	400

CONCLUSION

1. Optimization of MT-LB load-carrier/prime mover transmission is a promising scientific and applied problem, the solution of which allows to improve the mass characteristics of the machine, as well as to ensure the load capacity and durability of the transmission during modernization.

2. The mathematical model of transmission optimization by weight is constructed, namely, the objective function taking into account geometry, durability and mass-dimensional properties of the basic details and assemblies is considered; defined variables planning; a system of constraints has been formed.

3. The algorithm of optimum design for transmission is developed. It takes into account its main features, reduces the time of calculations by narrowing the limits for teeth numbers of

the GB wheels $z_{GB\mu,1}$, improves the accuracy of calculations by managing errors in calculating gear ratios and equality of axis distances.

4. The calculations results confirm the adequacy of the mathematical model for solving the optimization problem:

– achieved the placement of new transmission variants in the existing engine-transmission compartment instead of the basic while increasing engine power from 240 to 300, 326, 408, 448 hp;

– achieved a mass reduction of new transmission variants compared to the basic 533.2 kg, respectively, by 14.6%, 9%, 0.3%, 1.1%, as well as 16.4% for the basic engine of 240 hp.

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