Models and approaches in design and diagnostics of vehicles planetary transmissions

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Abstract. The paper presents author’s models and method for vehicle planetary transmission, which cover the stages of synthesis, kinematic, quasi-static and dynamic computations, dependability prediction and diagnostics of transmissions in operation process. The obtained data of the synthesis stage (block and kinematic diagrams, gear ratios of transmission mechanisms) are expanded at subsequent design stages with the ultimate goal of forming a complete information model (digital twin), allowing to support transmission design process and predict the behaviour of the transmission in operation. The emphasis is made on the developed original methods solving the most science intensive problems in the transmission life cycle.

1. Introduction

One of the modern trends in the synthesis of planetary transmissions is based on a complete computer search for possible structure variants [1]. The problem of transmission structure uniqueness (or isomorphism) significantly complicates the synthesis procedure, since many externally different variants of solutions describing the same structure (diagram) are involved in the process. In a number of works, this problem is solved with the use of graphs to describe the structure of planetary transmissions [2-4]. However, all these works suppose the construction of possible variants, and then the recognition of isomorphic solutions. In the paper, this problem is solved by means of forming the unique (non-isomorphic) structures in the process of constructing. The second direction in the synthesis of planetary transmissions is the use of basic modules, for example, the mechanisms of Ravigneaux, Simpson and Lepelletier [5]. Under this direction, the article presents a method for synthesizing multistage transmissions based on the modified Simpson mechanism.

In the field of kinetic, quasi-static and dynamic calculations of mechanical systems, well-known software packages that have wide universality are used [6-8]. This entails the cumbersomeness and complexity of their application to special objects, similar to the transmission. In a number of packages for the design of drives, the representation of the transmission is limited to reducers [9, 10]. Complex kinematic and other diagrams that combine gears of various types are not modeled. Special calculations for such facilities are not foreseen. Thus, there are no convenient tools for operations with basic representations of the transmission in the form of kinematic and dynamic diagrams (schemes). The development of such tools is obvious.

In predicting the reliability of systems, structural methods dominate. It is assumed [11, 12], that the prediction of system reliability is based on reliability data of components. However, the problem consists in how to obtain mentioned data. The same mechanical component, for example, a bearing or
a gear, has different lifetime in different units and machines. Hähnel et al. [13] mark dissociation of structural and engineering approaches. The first one is named as ‘system without physics’, and the second one as ‘physical without system’. Next problem of the correct reliability calculation consists in random character of initial data for operation conditions of machines and characteristics of load-carrying abilities of components. This aspect is treated as uncertainty of the initial information. The Monte Carlo method is used for passing casually chosen physical variables through fault tree of system with complex failure logic and for determining possible consequences [13]. However, the description and taking into account the dependences for components in this reliability calculation is not considered in an explicit form. In [14], degradation of dependent components is examined. It is pointed out that the degradation of one component may influence the degradation of the other component, as they are operated under the same environment and can be functionally related. Dependencies among the degradation processes of the components should be taken into account during the dynamic reliability assessment and prognostics process. The approach used by Lin and Zio [14] is based only on mathematical modelling without involving physical (mechanical) models of degradation and reproducing the common factors causing degradation processes. The simplest case, in which the dependent behaviour of components is taken into account in a frame of the mechanical model, is the consideration of the reliability of a mechanical chain as a system consisting of several identical links under the action of breaking load [15]. But this way is not admissible for the transmission containing many diverse components. Therefore, one of the problems is the calculation of transmission reliability as a system of various loaded components.

The working process of the transmission is usually accompanied by oscillations and vibrations. Therefore, vibroacoustic plays an important role. Most of the diagnostic methods developed and standard tools are effective in diagnosing machine units operating in quasi-stationary conditions. These are various gearboxes of the technological equipment and aircrafts, test and running-in stands, fans, turbines, compressors, pumps, etc. Typical publications on that subject, containing approaches and tools for diagnosing gear systems, are [16-18]. Transmission systems of cars and tractors operate under conditions of varying speeds and loads. Under such conditions, the nature of vibrations (amplitude and frequency composition) is constantly changing. The well-known equipment and techniques are not suitable for vibration monitoring of the technical state of the transmission components. Approaches that combine computing models of working processes and sensors data that provide information on the progress of these processes in a particular product are most effective for predicting individual reliability.

The paper objective is to present the complex of models and methods for designing and monitoring the operation of vehicles planetary transmissions developed at Joint Institute of Mechanical Engineering of NASB within the framework of scientific direction ‘Lifetime mechanics of machines’. Paper includes: questions of synthesis of planetary transmissions (Section 2), their kinematic, quasi-static and dynamic calculations (Section 3), predicting dependability (Section 4), monitoring dependability in operation (Section 5), and Conclusion (Section 6).

2. Synthesis of planetary transmission

The stage of synthesis is initial in the transmission life cycle. This stage is completed by the development of several promising variants of the transmission (for further developmental work), presented in the form of kinematic diagrams and their parameters (gear ratios and so on). Kinematic diagram is used as a basis for layout design, rated calculations, as well as for determining the loads for verification calculations of the transmission lifetime.

2.1. Synthesis of non-isomorphic structures

Known approaches to synthesis of planetary transmissions suggest the construction of all possible variants of structures, and then the selection of non-isomorphic ones. The proposed approach is based on the construction of obviously non-isomorphic structures. The structure is described by a canonical matrix, formed under the special rules, which makes it possible to avoid isomorphism.
The mechanism \((U_j, j=1, \ldots, N_D)\) having three parts (links) \((V_i, i=k, l, m)\) and two degree of freedom \((W=2)\) is a basic element of transmission structure, where \(N_D\) is a general number of three-part transmission mechanisms. Part \(V_i\) can enter into \(k\) mechanisms. The number \(k\) is parameter named as part degree (deg\(V_i\)). All variants of distribution deg\(V_i\) for parts are created beforehand. This is not a problem.

The structure is shaped by means of parts junction by the resolved modes. When structures are constructed, the incidence matrixes are used as well as a set of rules which allow receiving only original (non-isomorphic) structures. Principal rule is ‘use for each mechanism parts with the smallest numbers’. An additional rule is ‘use of all various variants of selecting parts taking into account their entrance in certain number of mechanisms’. Besides, conditions of existence and workability for the mechanism are checked. These are: 1) integrity, and 2) lack of interlocking. Incidence matrixes generated by such a way are named ‘canonical matrixes’ [19]. Figure 1 shows fragments of computer synthesis for structures with \(W=3\) and \(N_D = 4\). Variants of the deg\(V_i\) distribution by links are on the left, and a fragment of variants of constructing canonical matrices for distribution 2 (variant 1) and 3 (variants 2 and 3) are on the right.

![Figure 1. Fragments of computer synthesis: deg\(V_i\) distribution (left) and canonical matrices (right).](image1.png)

2.2. Synthesis of transmissions based on modified Simpson mechanism

The developed synthesis procedure refers to transmissions consisting of two modules: the main gearbox (MG) and an additional (range) reducer (AR). The MG contains a basic mechanism (BM) in the form of a modified Simpson mechanism, and the attached mechanism (AM). Simpson mechanism has a high load-bearing capacity due to the distribution of the input power flow among several parallel branches and the summation of these flows on the output shaft. MG, containing such a mechanism, can be placed not only at the output, but also at the input to the transmission, ensuring a sufficiently uniform loading for the planetary gear sets of MG and AR.

The block diagram of the transmission is shown in Figure 2. In the first execution case (input element 7, output element 8), the MG has three drive-down gears. In the second execution case (input element 8, output element 7), MG has three overdrive gears. The characteristic result of the synthesis for the transmission with \(W=3\) is shown in Figure 3, and the possible gear ratios are presented in Table 1.

![Figure 2. Transmission block diagram.](image2.png)

![Figure 3. One of the synthesized transmission.](image3.png)
3. Transmission kinematic, quasi-static and dynamic calculations

3.1. Kinematic and quasi-static calculations based on symbolical representation devices and structurally-distributive matrix

Transmission devices, which have the same mathematical structure and differ only in parameters, can be represented in a generalized (symbolic) form (Figures 4). To describe the structure and distribution of the internal torques in devices, a structurally-distributive matrix (SDM) is introduced. Each mentioned device is presented at such a matrix in the form of a column, as it is shown in the Table 2.

Table 1. Transmission gear ratios.

| AR clutch/brake | MG clutch/brake |
|-----------------|-----------------|
| 10              | 20              | 30 | 14 | 320 | 330 | 310 |
| 64              | 11.53           | 8.700 | 6.554 | 4.973 | 3.825 | 2.942 | –9.597 |
| 50              | 2.318           | 1.749 | 1.318 | 1.000 | 0.769 | 0.592 | –1.930 |

Table 2. Devices representation and distribution of internal torques among parts of devices.

| Index for device part | Differential D | Train P | Shaft S | Frame R | Clutch F | Brake T |
|-----------------------|----------------|---------|---------|---------|----------|---------|
| 1 (i)                 | 1              | 1       | 1       | 1       | 1        | 1       |
| 2 (j)                 | –u             | –u      | –1 (u=1)| 0 (u=0) | –1       | 0       |
| 3 (k)                 | –(1–u)         | —       | —       | —       | —        | —       |

The meanings of non-zero elements of kth column ($A_{nk}$) describe the distribution of torques in the kinematic unit ($A_n=M_n/M_1$). The equations for its kinematic and quasi-static calculations are automatically formed and the speeds of the links, torques in devices and the efficiency for transmission gears are determined [20]. Kinematic gear ratio $i_k$ is replaced by the power gear ratio $\dot{u}_k = u_k \eta_k$, for each device, where $x$ is equal +1 or –1, depending on a direction of power in the device.

3.2. Concept of regular mechanical system. Dynamic computing

A mechanical system with elementary mechanical components (the concentrated masses and the massless joining links) is essential idealization which imposes certain restrictions on possible combinations of joints (connections) for mentioned items.
The concept of regular mechanical system (RMS) considers that mechanical system consists of the concentrated masses (inertial components) and massless (non-inertial) devices-connectors: shafts, clutches, brakes, gears, motionless links, and other devices imposing kinematic connections for masses (Figure 5). Masses can be in contact interaction. For connecting device, the direct contact (not through inertial component) is prohibited. This is a principle of regularity, which is used for the representation of a real object. Its violation leads to wrong schematizations and errors in calculations or impossibility of mathematical model realization by the computer [21].

Clutches and brakes are the typical devices having variables states (Figure 6). For getting all-purpose mathematical model of their dynamics, a method of internal torques is used. This one is based on logical variables \( \lambda \), named indicators states, which describe states of clutches/brakes. The equations corresponding to the clutches are as follows:

\[
\omega_1 = \frac{\dot{\lambda}_1}{J_1} = \frac{[M_1 - (1 - \lambda_p)M_p - \lambda_p M_{12}]}{J_1}
\]

\[
\omega_2 = \frac{\dot{\lambda}_2}{J_2} = \frac{[(1 - \lambda_p)M_p - \lambda_p M_{12} - M_2]}{J_2}
\]

where \( \dot{\lambda}_p = 0 \) under locking, and \( \dot{\lambda}_p = 1 \) under slippage (unlocking) of the clutch; \( M_i \) is the known function describing a friction torque during slipping friction clutch parts.

In order to use all-purpose equations a special procedure is developed for finding the internal torques (like \( M_{12} \)) which acting in rigid devices. Besides, conditions of change for friction clutches states should be described during the solving the differential equations. In a case under review the internal torque can be calculated by formula:

\[
M_{12} = \frac{(J_1 M_1 + J_2 M_2)}{(J_1 + J_2)}
\]

Generally, it is necessary to consider spasmodic change of a friction torque at transition from slippage to friction clutch locking and inversely. Generalization of these situations for computing leads to the formulation of following principle: if a new condition of contacting inertial parts (for example, slippage or locking of friction clutch/brake) becomes possible then this condition should be necessarily presented at next step of computing process [21].

In general case the RMS can be stiff-elastic object that contains rigid and elastic devices. As typical example, RMS with rigid and elastic devices is presented in Figure 7. Dynamic diagrams related to this RMS are shown in Figure 8. During computing stiff-elastic object, it is necessary to solve a system of differential equations, and this is accompanied by solution of a system of algebraic equations to determine the internal torques in rigid devices. To avoid solving algebraic equations, the dynamical system can be normalized by replacing rigid devices with elastic ones having elasticities \( E_i > 0 \) (Figure 8, for brakes external links like \( E_{22} \) may be rigid). In the normalized system, masses and elastic links alternate. To determine the internal torques of closed clutches/brakes, a simple formula (3) is used instead of solving the system of algebraic equations. This approach simplifies forming mathematical models but increases a number of differential equations.
4. Calculating transmission dependability

The general approach reproduces the probabilistic nature of components properties and effects of their dependent behaviour in a system. This approach [22, 23] has general character and it can be applied to any technically complicated item, like a transmission.

Load modes are various for different machine parts. But all load modes are determined by the operation conditions. Therefore, the operation conditions are described by probabilistic manner in a form of the relative durations for the commonly accepted typical conditions (Figure 9, left).

To rate the loaded systems in the general case, the calculation can be performed under the scheme ‘operation conditions — lifetime’. For this purpose, lifetime-strength curves are introduced (Figure 9, right). These curves are preliminary calculated for every component by methods of mechanics.

The limiting state of a complex item is usually determined by a complicated way, based on combination of limiting states of its components. To describe and calculate such states, the scheme of limiting states (SLS) can be used. It describes the limiting states of a complex item much easier than the known tools (Failure tree and Reliability block diagram). In addition, SLS is convenient for use in statistical modelling procedures, where it is presented in the form of simple records. The SLS consists of a hierarchical structural scheme and records describing limiting states for every scheme object (except lowest object). The objects of the lower and intermediate levels are endowed with the type: the first type, the second type, and so on. Objects whose limiting states have the same significance for an object of a higher level are classed as the same type. The object type corresponds to its position (first, second, etc.) in the schematic record, which describes the criterion of the limiting state.

The record (X1, X2, X3, etc.) means that the limiting state of the machine (assembly, unit …) occurs if the limiting states are reached with its X1 parts of the first type (here X1 is the number standing in the first position), its X2 parts of the second type (here X2 is the number standing in the second position), etc. The unit, assembly, machine can have some the SLS. For example, a mechanical gearbox (Figure 10) has the following SLS: (1,0,0,0) (0,3,0,0) (0,0,1,2).

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**Figure 7.** Stiff-elastic RMS.  
**Figure 8.** Stiff-elastic (left) and normalized (right) diagrams.

**Figure 9.** Representation of operation conditions (left) and lifetime-strength curves for a component (right).

**Figure 10.** SLS for the mechanical gearbox.
In the general case, there is a multilevel SLS that reproduces the following levels: 1 = the machine (for example, a car); 2 = aggregates and systems (e.g. transmission, carrier system); 3 = units and subsystems (e.g. gearbox, drive axle); 4 = parts (e.g. gear), typical component parts (e.g. ball bearing), joints (e.g. splined connection); 5 = constructional elements (e.g. gear teeth); 6 = the simplest components (e.g. local area of the surface layer of the gear teeth). Mechanics methods are used for calculations of limiting states at levels 6, 5, and 4 (in some cases), and structural ones are used at levels (from 4 or 5 to 1).

The life cycle forecasting is based on the Monte Carlo method that has the following features. The relative durations of operation conditions and the load-carrying ability of elements are considered as the random variables. They are randomly selected in the process of statistical modelling and describes the concrete item and its operation condition. Every simulation cycle is supplemented by the SLS analysis and lifetime determination for objects of higher levels. When reproducing processes and states related to mechanical levels, factors and effects leading to dependent behaviours of the mechanical components are realized, for example, the general loading levels of components in the particular transmission device.

5. Monitoring transmission dependability in operation

The main feature of the developed diagnostic method is using meaningful modeling the oscillating process for the gear drive and the propagation of vibrations in the transmission. For this aim, one more type (two-way) of the transmission model is used. It is advisable to applicate together integral diagnostic models, describing vibroactivity of each gear, and predictive ones based on damage accumulation. Such a ‘two-coordinate’ approach (two points of view) ensures a higher veracity of the individual lifetime forecast. The method is demonstrated using the example for a reducer of a mining dump truck [24].

5.1. System for vibrodiagnostics of the RMW

The reducer of a motor-wheel (RMW) with installed sensors is shown in Figure 11. The mean square value and the mean amplitude of the first seven harmonics for the tooth frequency are adopted as indicators of the technical state of the gears. The operation of the system (Figure 12, block A1) includes: 1) analysis of the parameters for vibroimpulses synchronized with the angle of rotation of the diagnosed tooth gear; 2) identification of vibroimpulse harmonic components, which multiple of the tooth frequency, locate in the region of resonance frequencies of the mechanism and excite the most intense oscillations in the system; 3) determination of the technical state of gears under variable load-speed regimes through analysing the changing the parameters of vibroimpulses. The diagnostic system periodically interrogates the sensors, processes the diagnostic information, evaluates the technical state

Figure 11. RMW with sensors.  
Figure 12. Processes in the RMW monitoring system.
of the reducer, comparing the received root mean square (RMS) values of vibration acceleration with
the maximum permissible values for each of the states of the reducer.

5.2. Forecasting the lifetime expense
The method of forecasting the lifetime expense (consumption) of the responsible elements of the drive
mechanisms of machines under operation conditions is based on the determination of the shock pulse
in a meshing according to the results of vibration monitoring. Then a discrete spectrum of oscillations
of a periodically acting shock pulse is formed, and, accordingly, a set of harmonic oscillations
determining the load in the meshing. From these data, the actual circumferential force and contact
stresses in the meshing are calculated. The lifetime expense of the gear is determined for each fixed
interval of the running time \( \Delta S \) by the formula:

\[
\Delta Q = \sigma_{Hi}^\prime N_i
\]

where \( \sigma_{Hi} \) = contact stress; \( q_H \) = the exponent of the fatigue curve under calculating the gear for the
contact endurance; \( N_i \) = the number of loading cycles of the gear tooth.

Figure 13. Harmonic spectra of vibrations generated by the sun gear (left, 21 teeth) and satellites (right,
47 teeth) of the RMW planetary row.

5.3. Application example
Estimating the state of the gearing ‘sun gear/satellite’ of the second planetary row of RMW is
presented. When the technical state is evaluated via integral assessment, the change in RMS values of
the vibrational acceleration is monitored. The change in this indicator is shown in Figure 14. When the
dump truck mileage is less 200,000 km, this value remains practically constant. Further it increases, at
the same time the peak value of vibration acceleration begins to increase. When lifetime expense is

Figure 14. Dependence of RMS values of vibration acceleration of RMW on the mileage of a dump truck.

Figure 15. Damaged working surfaces of the sun gear teeth.
calculated using the diagnostics current data, it is taken into account that a linear relationship exists between the amplitude of the shock pulse and the peak value of the vibration acceleration. The growth of peak values means an increase in the dynamic factor $K_{dh}$, which is used in the calculation of contact stresses $\sigma_{H}$. With a mileage of 238,100 km, the operation of the dump truck was stopped, as the vibration monitoring system showed that the vibration level of the reducer reached the critical range. Reducer disassembly confirmed the result of the calculation and the data of the monitoring system. The reinforced layers of four teeth of sun gear were almost completely destroyed (Figure 15).

6. Conclusion
The presented models and methods are the basis for computation and information support of the life cycle of vehicle planetary transmissions. The developed models are based on five basic representations of the transmission: structural (block), kinematic and dynamic diagrams, a model of general reliability, including scheme of limiting state, and an individual diagnostic model of operational reliability that combines aspects of dynamics and durability.

Under computer synthesis, the way for construction of non-isomorphic structures for planetary transmissions is used. This allows to avoid consideration of set of identical structural variants. Another way is realised within the framework of the modular approach: a basic module is proposed in the form of a modified Simpson mechanism with reduced loading the elements.

The kinematic and quasi-static computations of transmission are based on the representation of its mechanisms in a generalized form and using a structure-distributive matrix describing the structural relationships and distribution of the torques between the transmission elements. This makes it possible to calculate of transmission of arbitrary configuration by its kinematic diagram with various types of planetary and non-planetary mechanisms.

Dynamic computation is based on the developed concept of a regular mechanical system. This ensures the correct schematization of transmission for modelling, the use of universal dynamic models, taking into account the change in the states of devices with variable structure (clutches and brakes), as well as changes in the direction of power flows passing through the transmission mechanisms at various gears and modes.

The developed models and methods for calculating dependability are of universal significance. The general approach reproduces the probabilistic nature of components properties and effects of their dependent behaviour in a system and is based on the following features: 1) probabilistic model of the variation in the operation conditions (variability ‘from vehicle to vehicle’), 2) the reproduction of dependencies in the load modes of various elements under calculating the reliability of the transmission as a system, 3) the use of a scheme of limiting state for computing the durability of a multicomponent mechanical system with different component lifetimes. These features provide prediction of real dependability, using mechanical, and then structural models for complex multi-level systems.

Diagnosis of transmission in operation provides for the use of two models of various types, which increases its authenticity. The first is integral diagnostic models, describing vibration activity of each gear. The second model realizes lifetime expense that is calculated using the diagnostic current data. The presented complex of models, methods and data, accompanying them, forms the information model (digital twin) of the transmission. These methods and approaches developed within the framework of scientific direction ‘Lifetime mechanics of machines’ are methodologically superior to other known solutions and fully correspond to Industry 4.0 concept.

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