# SIMULATION ANALYSIS OF PLANETARY TRANSMISSIONS IN MATLAB ENVIRONMENT

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**Abstract:** At present time three main widespread methods of planetary gearboxes analysis are used – analytical calculation, practical tests and simulations. This paper deals with verification of the Simscape driveline module of the Matlab software as a tool of vehicle transmission and driveline designing. Different parameters results of analytical computing and Simscape simulations output values are compared in this paper. Deviations of these values turned out to be insignificant and therefore Simscape module proved to be applicable, usable, cost and time reducing and very useful in transmissions designing process.

**Keywords:** Planetary gearbox; Transmission; Simulation; Simscape; Driveline.

## 1 INTRODUCTION

Planetary gearbox is a mechanism that has found wide application in a design of the both wheeled and tracked military vehicles transmission systems. A large scale of kinematic and dynamic effects appears in different designs of these mechanisms. These effects can seriously affect the dynamic and tractive effort characteristics of the vehicle and consequently can affect the applicability of the certain planetary gearbox in the vehicle powertrain. Therefore the determination or the computation of these effects and parameters values of each design at each mode is a necessity. Currently three main wide-spread computing approaches are used. Due to longer computing time and simplifications of the analytical approach and higher costs of practical gearboxes testing, different simulation tools are nowadays used more and more. Despite some cons like its simplifications, it is very effective and economically advantageous to prove applicability of certain gearbox design in vehicle powertrain by using simulation tools before practical test execution.

Purpose of this paper is to verify the Simscape Driveline module of the Matlab software as one of the possible ways of conducting planetary gearbox analysis.

Because of the impossibility of practical tests execution in our environment we have chosen the method of comparing results gained by analytical method and by simulation for verification.

We have chosen the systematical approach – from basic models and basic parameters to more complex ones. The goal is not yet to create the perfectly reality matching model, but just to verify if the Simscape module is usable for drivelines and gearboxes modeling and analyzing.

Our assumption is that the deviations of the values gained analytically and by successful models simulations should not exceed the value of 5 % deviation.

## 2 ANALYZED GEARBOX

Gearbox ZF 6HP26 is a part of the drivetrain of the armored vehicle Iveco LMV 4x4. It has three degrees of freedom, five control elements and use 7 (6+1) of 10 theoretically possible gears. [1]

Gearbox is equipped with torque converter with lock-up clutch and is composed of one simple planetary gear train (PGT) and one Ravigneaux gear set (RPGT). It has five control elements – two multiple disk brakes (B1, B2) and three multiple plate clutches (C1, C2, C3). For further analysis we need to know design of the gearbox and connections between components. This information is represented by kinematic scheme of the gearbox (Fig. 1). [1]

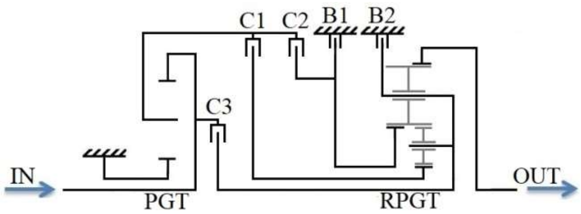


**Fig. 1** Kinematic scheme of gearbox ZF 6HP26
Source: authors.

## 3 ANALYTICAL APPROACH

The analysis of any planetary gearbox consists of six basic steps: [2] [3]

1. Planetary gear train (PGT) function analysis.
2. Kinematic parameters analysis.
3. Dynamic parameters analysis.
4. Efficiencies computing and detection of parasitic power.
5. Basic principles of PGT construction.
6. Strength calculation [4].

For first step verification of usability of the simplified powertrain model have been chosen both basic kinematic parameter (revolutions) and basic dynamic parameter (torques) of PGT main components. (steps 2. and 3. of the analysis)

If simplified powertrain model proves to be usable for further computing we will try to create more complex powertrain model. Than we would be able to verify and compare also the efficiencies gained analytically and by a simulation. (step 4. of the analysis).

### 3.1 Revolutions analysis

By accomplishment of the first point of the analysis we were able to create computing scheme and PGT angular velocities graph (Fig. 2).

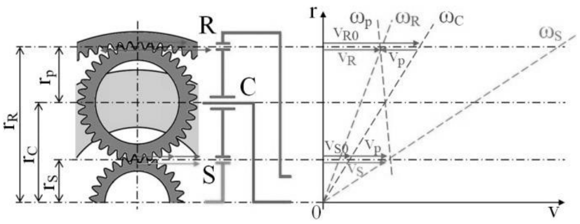


**Fig. 2** PGT angular velocities graph
Source: authors.

Further investigation of this graph, computing scheme and PGT function has led to the derivation of the basic PGT kinematic equation (1). [2] [3]

$$n_S + n_R\alpha - n_C(1+\alpha) = 0 \tag{1}$$

Where: $n_S$ - *sun gear revolutions*
$n_R$ - *ring gear revolutions*
$n_C$ - *carrier revolutions*
$\alpha$ - *PGT parameter*

Understanding and proper usage of this equation is fundamental for further computing of the revolutions (rpm) of the main PGT components.

### 3.2 Torques analysis

PGT dynamic parameters calculation is based on the basic mechanics principles and principle of the balance of forces and torques acting on the carrier. (Fig. 3) [2]

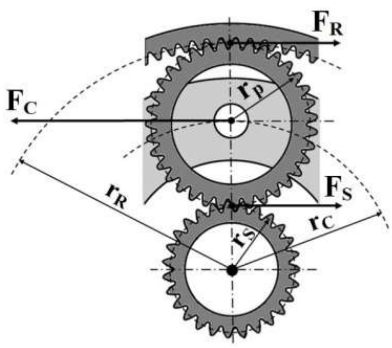


**Fig. 3** Balance of forces in basic PGT
Source: authors.

By further investigation of the balance of forces and torques inside PGT and by application of the basic torque formula (2) we can derive following PGT torque formulas. (Tab. 1) [2] [3]

$$\tau = F.r \tag{2}$$

**Tab. 1** Basic PGT torque formulas [2]

| | | |
|---|---|---|
| $\tau_S = \dfrac{\tau_R}{\alpha}$ | $\tau_R = \tau_S.\alpha$ | $\tau_C = \tau_S(1+\alpha)$ |
| $\tau_S = \tau_C\dfrac{1}{1+\alpha}$ | $\tau_C = M_0\dfrac{\alpha}{1+\alpha}$ | $\tau_C = M'\dfrac{1+\alpha}{\alpha}$ |

Where: $\tau_S$ - *sun gear torque*
$\tau_R$ - *ring gear torque*
$\tau_C$ - *carrier torque*
$\alpha$ - *PGT parameter*

### 3.3 Efficiencies analysis

Considerable losses caused mainly by friction of gears, bearings friction and oil resistance can occur in every gearbox design. Simplified calculation of the efficiencies often takes only the most significant gear friction losses into account. [2] [3]

In the PGT the power is transmitted not only by relative motion (with gear losses), but also by carrier motion (without gear losses). Therefore it is not possible to use the simple efficiency equation of fixed shaft axle transmission. We have to take to account what part of the power is transmitted by relative motion and what by carrier motion. We start with basic gear efficiency equation (3). [2] [3]

$$\eta_m = \frac{P_2}{P_1} = \frac{i_p}{i_k} \tag{3}$$

Where: $P_1$ - *input shaft power*
$P_2$ - *output shaft power*
$i_p$ - *power ratio*
$i_k$ - *kinematic ratio*

While kinematic ratio is defined as function of PGT parameters in action at particular gear (4) [2]

$$i_k = \frac{n_2}{n_1} = f(\alpha_1, \alpha_2, \dots \alpha_n) \tag{4}$$

Power ratio is defined as function of PGT parameters and efficiencies of these PGTs (5). [2]

$$i_p = \frac{\tau_2}{\tau_1} = f(\alpha_1\eta^{x1}, \alpha_2\eta^{x2}, \dots \alpha_n\eta^{xn}) \tag{5}$$

Where: $\eta$ - *PGT efficiency (0,96 – 0,98)*
$x$ - *exponent with value ± 1*

For finding +/- sign of the exponent x equation (6) has been derived. [2]

$$sngx_i = \quad sng\frac{\alpha_i}{i_i}.\frac{\partial \mathrm{i}_{ck}}{\partial \alpha_i} \qquad (6)$$

Where: $i_i$ - *kinematic gear ratio of particular gear*

Using equation (3) we can get power efficiencies of the gearbox at each gear analytically.

## 4 NUMERICAL APPROACH

The selected tool for numerical gearbox analysis is Simscape driveline – subcomponent of the Matlab software, which is designed for iterative analysis and design processes with a programing language that express matrix and array mathematics directly. [5]

Simscape driveline is a tool used especially for modeling and simulating translational and rotational mechanical systems. Its most significant advantage is its components library. This library includes basic models of drivetrain components. Therefore there is no need to design these components, but just to determine their parameters and links between them to create a model. Driveline modeling employs a physical network approach, where Simscape blocks correspond to physical components, such as engines, gears, brakes, clutches, tires, pistons and so on. The lines that connect these components represent the physical connections that transmit power. The resulting models let us describe the physical structure of a system, rather than the underlying mathematics. [4]

In general, there are more benefits that make software simulations very effective tool for gearbox analysis:

1. Low cost.
2. Calculation time – lower due to computers improving.
3. Cover multiple parameters by one calculation.
4. Possibility of parameters values monitoring at each mode by simple model editing.
5. Simple change of input parameters.
6. Simple change of model components parameters.

On the other hand, in the simulation process, there are various simplifications which make the simulation results less accurate and less reliable. Therefore it is necessary to verify these results either by practical test execution or/and by analytical calculation.

### 4.1 Basic Simulation model

To verify applicability of the Simscape driveline as a tool of the gearboxes designing the systematical approach has been chosen. We have started with basic model of the simplified drivetrain with ideal angular velocity source or ideal torque source (Fig. 3) and basic ZF 6HP26 gearbox model.

This way we were able to exclude possible mistakes and errors which could have occurred in other blocks (engine, torque converter, tires etc.).

If we could prove that the results gained by numerical and analytical approach are identical or with insignificant deviation, we would proceed to more complex models.

The simplified model (Fig. 4) consists of ideal source which is the input for the simulation and of the gearbox ZF 6HP26 itself. Outputs of the simulation are the rpms and torques of each main PGTs components measured by ideal rotational motion sensors and ideal torque sensors.

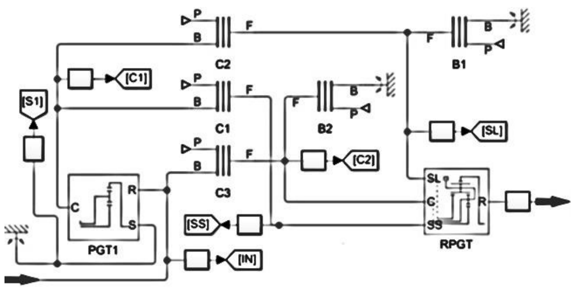


**Fig. 4** Basic ZF 6HP26 gearbox model
Source: authors.

## 5 RESULTS COMPARISON

Table 2 represents values of both rpms and torques of main components of the simple planetary gear train and Ravigneaux planetary gear train at each one of seven used gears.

Each value in the table is a relative value to the input shaft revolutions or torque. There are the values of the main components rpms and torques calculated both analytically and by simulation.

Finally there is a deviation of these two values at each gear for all gearbox main components. As we can see, the values of the deviation are zero or insignificant. Even the largest values of the deviation occur only at the third decimal place and any of them exceeds 0,03 % of the particular value. This fact has led us to assumption that the basic model of the gearbox is correct and usable for further more complex modeling and simulations.

**Tab. 2** Comparison of analytical and simulation rpms and torques results

| Gear | Calculation | Revolutions of the main Components | | | | | | | Torques of the main Components | | | | | | | i |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | S1 | C1 | R1 | SS | SL | C2 | R2 | S1 | C1 | R1 | SS | SL | C2 | R2 | |
| 1st | Analytical | 0,0000 | 0,6575 | 1,0000 | 0,6575 | -0,5361 | 0,0000 | 0,2397 | -0,5208 | 1,5210 | 1,0000 | 1,5210 | 0,0000 | 2,6460 | 4,1670 | 4,1670 |
| | Simulation | 0,0000 | 0,6575 | 1,0000 | 0,6575 | -0,5375 | 0,0000 | 0,2400 | -0,5210 | 1,5210 | 1,0000 | 1,5210 | 0,0000 | 2,6508 | 4,1717 | 4,1717 |
| | Deviation | 0,0000 | 0,0000 | 0,0000 | 0,0000 | 0,0014 | 0,0000 | 0,0003 | 0,0002 | 0,0000 | 0,0000 | 0,0000 | 0,0000 | **0,0048** | **0,0047** | **0,0047** |
| 2nd | Analytical | 0,0000 | 0,6575 | 1,0000 | 0,6575 | 0,0000 | 0,2953 | 0,4273 | -0,5208 | 1,5210 | 1,0000 | 1,5210 | 0,8167 | 0,0000 | 2,3380 | 2,3380 |
| | Simulation | 0,0000 | 0,6575 | 1,0000 | 0,6575 | 0,0000 | 0,2958 | 0,4275 | -0,5210 | 1,5210 | 1,0000 | 1,5210 | 0,8191 | 0,0000 | 2,3400 | 2,3400 |
| | Deviation | 0,0000 | 0,0000 | 0,0000 | 0,0000 | 0,0000 | 0,0005 | 0,0002 | 0,0002 | 0,0000 | 0,0000 | 0,0000 | 0,0024 | 0,0000 | 0,0020 | 0,0020 |
| 3rd | Analytical | 0,0000 | 0,6575 | 1,0000 | 0,6575 | 0,6575 | 0,6575 | 0,6575 | -0,5208 | 1,5210 | 1,0000 | 0,9895 | 0,5314 | 0,0000 | 1,5210 | 1,5210 |
| | Simulation | 0,0000 | 0,6575 | 1,0000 | 0,6575 | 0,6575 | 0,6575 | 0,6575 | -0,5210 | 1,5210 | 1,0000 | 0,9886 | 0,5324 | 0,0000 | 1,5210 | 1,5210 |
| | Deviation | 0,0000 | 0,0000 | 0,0000 | 0,0000 | 0,0000 | 0,0000 | 0,0000 | 0,0002 | 0,0000 | 0,0000 | 0,0009 | 0,0010 | 0,0000 | 0,0000 | 0,0000 |
| 4th | Analytical | 0,0000 | 0,6575 | 1,0000 | 0,6575 | 1,2793 | 1,0000 | 0,8751 | -0,1428 | 0,4171 | 0,2743 | 0,4171 | 0,0000 | 0,7257 | 1,1430 | 1,1430 |
| | Simulation | 0,0000 | 0,6575 | 1,0000 | 0,6575 | 1,2800 | 1,0000 | 0,8750 | -0,1427 | 0,4166 | 0,2739 | 0,4166 | 0,0000 | 0,7261 | 1,1427 | 1,1427 |
| | Deviation | 0,0000 | 0,0000 | 0,0000 | 0,0000 | 0,0007 | 0,0000 | 0,0001 | 0,0001 | 0,0005 | 0,0004 | 0,0005 | 0,0000 | 0,0004 | 0,0003 | 0,0003 |
| 5th | Analytical | 0,0000 | 0,6575 | 1,0000 | 1,4201 | 0,6575 | 1,0000 | 1,1532 | 0,1326 | -0,3872 | -0,2546 | 0,0000 | -0,3872 | 1,2550 | 0,8674 | 0,8677 |
| | Simulation | 0,0000 | 0,6575 | 1,0000 | 1,4190 | 0,6575 | 1,0000 | 1,1530 | 0,1328 | -0,3878 | -0,2550 | 0,0000 | -0,3878 | 1,2550 | 0,8672 | 0,8672 |
| | Deviation | 0,0000 | 0,0000 | 0,0000 | 0,0011 | 0,0000 | 0,0000 | 0,0002 | 0,0002 | 0,0006 | 0,0004 | 0,0000 | 0,0006 | 0,0000 | 0,0002 | 0,0005 |
| 6th | Analytical | 0,0000 | 0,6575 | 1,0000 | 2,2265 | 0,0000 | 1,0000 | 1,4472 | 0,0000 | 0,0000 | 0,0000 | 0,0000 | -0,3086 | 1,0000 | 0,6914 | 0,6914 |
| | Simulation | 0,0000 | 0,6575 | 1,0000 | 2,2230 | 0,0000 | 1,0000 | 1,4460 | 0,0000 | 0,0000 | 0,0000 | 0,0000 | -0,3090 | 1,0000 | 0,6910 | 0,6910 |
| | Deviation | 0,0000 | 0,0000 | 0,0000 | **0,0035** | 0,0000 | 0,0000 | 0,0012 | 0,0000 | 0,0000 | 0,0000 | 0,0000 | 0,0004 | 0,0000 | 0,0004 | 0,0004 |
| R | Analytical | 0,0000 | 0,6575 | 1,0000 | -0,8064 | 0,6575 | 0,0000 | -0,2940 | -0,5208 | 1,5210 | 1,0000 | 0,0000 | 1,5210 | -4,9280 | -3,4070 | -3,4070 |
| | Simulation | 0,0000 | 0,6575 | 1,0000 | -0,8043 | 0,6575 | 0,0000 | -0,2935 | -0,5210 | 1,5210 | 1,0000 | 0,0000 | 1,5210 | -4,9223 | -3,4013 | -3,4013 |
| | Deviation | 0,0000 | 0,0000 | 0,0000 | **0,0021** | 0,0000 | 0,0000 | 0,0005 | 0,0002 | 0,0000 | 0,0000 | 0,0000 | 0,0000 | **0,0057** | **0,0057** | **0,0057** |

## 6 COMPLEX DRIVETRAIN MODEL

The comparison of the values of rpms and torques of the main ZF 6HP26 gearbox components gained both analytically and by simulation proved that this gearbox model can be integrated into the more complex simulation models.

In these simulations, instead of being the main part, the gearbox is just a subcomponent of the whole vehicle drivetrain model.

Modeled vehicle is the Iveco LMV 4x4 which is used in Slovak military forces and which actually uses ZF 6HP26 gearbox. We have tried to model all of the vehicle drivetrain subcomponents as realistic as we have been able to. The level of matching the reality is mostly based on the availability of the components parameters.

The purpose of this particular model is not to completely match the reality yet, but just to show the way we can observe the function of the vehicle drivetrain in much easier and much more intuitive way.

The more complex model of the Iveco LMV 4x4 drivetrain model consists of seven main subcomponents: (Fig. 5)

1. **Driver input** – which represents position of the accelerator pedal and is also an input for the engine.
2. **Engine** – engine model characterized by basic parameters.
3. **Torque converter** – generic model.
4. **Gearbox** – analyzed ZF 6HP26 model.
5. **TCU** – transmission control unit shifting gears and working with shift map based on vehicle speed and accelerator pedal position.
6. **Additional gearbox** – reduction and transfer gearbox.
7. **Vehicle body** – consists of chassis components (differentials, final gears, wheels etc.) and vehicle body represented by vehicle dimensions.

In these simulations, instead of being the main part, the gearbox is just a subcomponent of the whole vehicle drivetrain model.

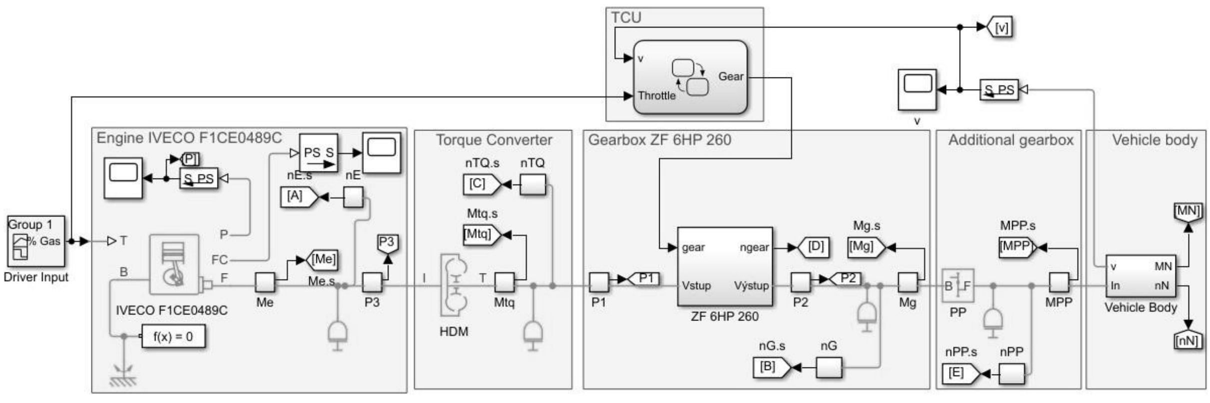


**Fig. 5** Complex IVECO LMV 4x4 drivetrain model
Source: authors.

### 6.1 Complex drivetrain verification

For first step verification of the model applicability the fourth step of gearbox analysis has been chosen – comparison of the values of efficiencies gained both analytically and by the complex model simulation (part 3).

The complex model simulations are able to provide great number of different outputs and results. We are not able to reliable compute these values analytically, especially with large number of unknown parameters influencing the results.

Therefore we have chosen the efficiencies values as the first step of the model verification. The analytical computing has been conducted using method stated in part 3.3. the gearbox in complex model (Fig. 5) is equipped with power sensors P1 at input shaft and P2 at output shaft. By comparison of these two power values at each gear we have got the following graph. (Fig. 6)

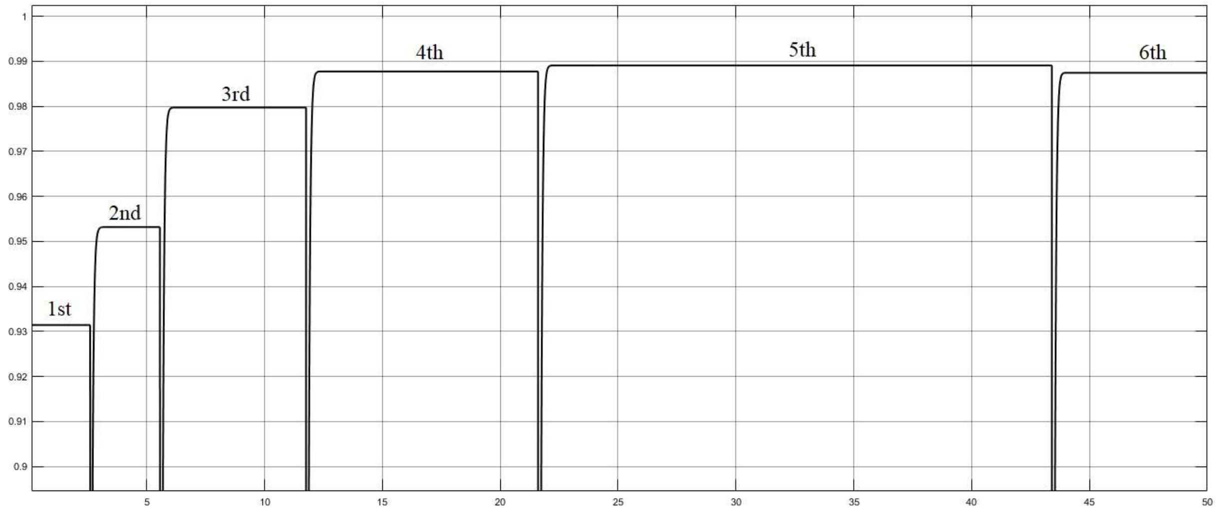


**Fig. 6** Efficiencies of the ZF 6HP26 gearbox at each gear gained by simulation
Source: authors.

The following table (Tab. 3) represents the comparison of the efficiencies values gained both analytically and by the complex model simulation.

As we can observe from the table, the results are very similar with insignificant deviation which does not exceed 1 % of the efficiencies values.

**Tab. 3** Comparison of analytical and simulation efficiencies results

| Gear | 1st | 2nd | 3rd | 4th | 5th | 6th |
|---|---|---|---|---|---|---|
| **Analytical** | 0,9245 | 0,9606 | 0,9887 | 0,9880 | 0,9925 | 0,9896 |
| **Simulation** | 0,9314 | 0,9532 | 0,9797 | 0,9877 | 0,9891 | 0,9874 |
| **Deviation** | 0,0069 | 0,0074 | 0,0090 | 0,0003 | 0,0034 | 0,0022 |
| **%** | 0,7442 | 0,7698 | 0,9100 | 0,0259 | 0,3404 | 0,2188 |

## 7 CONCLUSION

This paper deals with the verifying of the usability and applicability of the Simscape driveline software as a tool of the modeling and designing of the transmission mechanisms. To verify applicability of the software, the systematical approach has been chosen – from the simplified models to the more complex models and simulations. For this purpose the results of both analytical calculations and software simulations of the planetary gearbox ZF 6HP26 have been compared. We have created both the simplified and complex model of the vehicle drivetrain and have conducted a simulation. We have also made an analytical calculation of three major parameters at each gear.

In first step we have compared values of revolutions and torques of each main gearbox component at each gear with values gained by simple model simulation. The largest deviation has been 0.03 % of the original value. We have found this deviation acceptable for further modeling of the more complex model.

In next step we have conducted comparison of efficiencies values at each gear gained by analytical calculation and by the complex model simulation. The largest deviation has been 0.91 % of original value.

Our assumption had been that the deviation between successful model values and analytical results should not exceed 5 % value.

Therefore we can confirm our hypothesis and we can claim that the Simscape driveline model is one of the possible ways of conducting transmissions and drivelines analysis. This tool proved to be applicable, usable, cost and time reducing and very useful in transmissions designing process.

This paper also indicates the possible path of further effort – to investigate another ways and parameters which could be observed, verified and analyzed in simulation and either analytical or real conditions, what could lead to more reliable verification of the Simscape driveline module.

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