A MATHEMATICAL MODEL OF OPERATION OF A SEMI-TRAILER TRACTOR POWERTRAIN

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Resume

The mathematical modelling is an inseparable part of mechanical engineers' activities. It is applied in design, optimization, verifying, as well as modifying of products. The approach of creating virtual models and their analyzing without having real products is applied in several domains. A mathematical model is given by equations of motion. The main objective of this work is to present the way how a mathematical model of a car powertrain operation is derived and applied in practical applications. A semi-trailer tractor with a standard powertrain was chosen as a reference car. The derived equations of motion are written in a matrix form and subsequently they are solved by means of a technical programming language Matlab. Parameters of the solved tractor mechanical system come from the real data and they are supplemented by the empirical data.

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1 Introduction

Presently, the lorry transport belongs to the most common transport of goods in Europe. Lorry transport is provided by the combination vehicles. In the most European countries, road combination vehicles usually consist of a tractor and one semitrailer or a lorry with one trailer. Although these two types of combination vehicles differ by the maximal permissible length, their maximal permissible weight is of 40 t (Figure 1a). The exception are Scandinavian countries (Norway, Finland, Sweden, Denmark). In these countries, the standard maximal weight is of 60 t and combination vehicles can comprise more trailers (or semitrailers) (Figure 1b, c). The maximal permissible weight of combination vehicles in UK is also slightly higher than in the rest of European countries, namely of 48 t. As the Europe countries have different geographic and climatic conditions, topology of roads, roads networks and other factors and the goods is very often transported through many countries (e.g. from the north to the south and from the east to the west), it is needed to ensure fast enough and reliable movement of goods on roads. One of the key factors is sufficient performance of a powertrain vehicle [1-3].

The data mentioned above lead to the fact, that current tractors of combination vehicles the most often use engines with power of 300 kW to 400 kW and with the torque of 2000 Nm to 2700 Nm [4]. The power is transmitted to the driving axle from an engine through a gearbox, either manual or automatic. Thus, in principle, a powertrain of a standard two-axle semi-trailer tractor consists of a combustion engine, a gearbox, optionally an additional gearbox, further the drive shaft, a differential and the drive axle (the rear axle).

The objective of engineers and designers of lorries is to reach optimal driving characteristics of powertrain and thus the entire combination vehicles. The combustion engine works efficiently at certain range of operational conditions, which include engine speed, engine load and others. These parameters are influenced by a road topology, i.e. whether a vehicle moves on a plane road, in climbs, on highways, in urban environment etc. [5-7]. Therefore, tractors are equipped by engines with various maximal powers to reach sufficient speed at low fuel consumption. In order to be able to predict the waveforms of important powertrain parameters in the design process, it is necessary to create a suitable model [8-9].

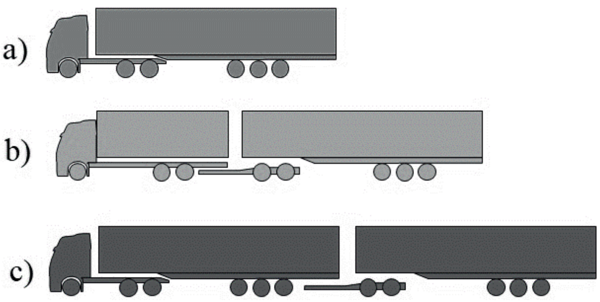

Figure 1 Possible configurations of combination vehicles in Europe [10]

Figure 2 An illustration of the solved semitrailer tractor [11]

Table 1 Parameters of the solved vehicle

Parameter	Designation	Value	Unit
Curb weight	m_c	7250	kg
Total weight of a combination vehicle	m_{tot}	40000	kg
Maximal power	P_{max}	337 (460)	kW (HP)
Maximal torque	M_{max}	2600	Nm
Engine displacement	V	13	l

This work is focused on derivation of a mathematical model of powertrain operation, which belongs to a chosen semi-trailer tractor. Following sections include description of a selected tractor, the derivation of a drive-train mathematical model by means of the Lagrange's equations of motion of the second kind method, its solution using the Matlab software and presentation of the reached results.

2 Derivation of mathematical model of a tractor powertrain

2.1 Description of the semi-trailer tractor

An analysed semitrailer tractor is intended for towing semitrailers with maximal weight of 24 tons. The total weight of the combination vehicles is considered of 40 tons.

It is a two-axle tractor with chassis configuration 4x2, i.e. the rear axle is a drive axle and the front axle is a steering axle. It is considered that the tractor is powered by a diesel combustion engine with the swept volume of 13 litres. The maximum toque is of 2600 Nm and the maximum power is of 337 kW (460 HP). The torques is transmitted to the rear axle by an automatic gearbox. It has installed a hydrodynamic converter of torque [11].

The curb weight of a solo tractor is considered of 7250 kg. All the needed parameters of the tractor are listed in Table 1. An illustration of the solved semitrailer tractor is depicted in Figure 2.

2.2 A mathematical model

A powertrain of the solved semitrailer tractor is a mechanical system with certain typical properties [12-14]. In this case, the work is focused on investigation of running-up the powertrain. For this, it is necessary to set-up a mathematical model. It will consist of equations of motion and a suitable method should be applied for their derivation. The Lagrange's equations of motion of the second kind method was chosen. It is known and widely applied method for derivation of dynamical models of various mechanical systems [15-17]. It is suitable for mechanical systems performing translational, rotational or combined motions. Its general form is:

$$\frac{d}{dt}\frac{\partial E_K}{\partial \dot{q}_j} - \frac{\partial E_K}{\partial q_j} + \frac{\partial E_D}{\partial \dot{q}_j} + \frac{\partial E_P}{\partial q_j} = Q_j, \quad j = 1, 2, \ldots, n, \tag{1}$$

where E_K, E_D and E_P are kinetic, dissipative and potential energy of the system, respectively, q_j, $\dot{q}_j$ are generalized coordinates and their time derivatives of the system, respectively, Q_j is a vector of external loads and n determines number of degrees of freedom. Based on Equation (1), it is certain that number of equations of motion depends on degrees of freedom of a mechanical system. Moreover, using of this method requires to determine proper values of individual energies [18-21].

Therefore, the first step for application of the method is creation of a dynamical model. It comes from a scheme depicted in Figure 3. The solved powertrain of the tractor composes of several components marked

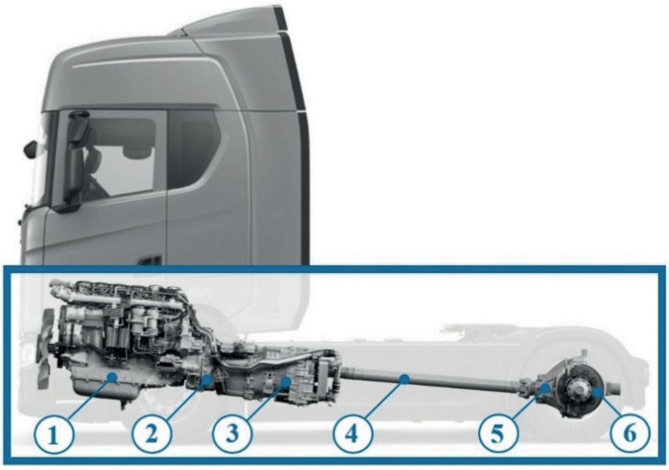

Figure 3 *Components of the solved tractor powertrain: 1 - engine, 2 - hydrodynamic converter, 3 - gearbox, 4 - propeller shaft, 5 - differential, 6 - drive wheels [22]*

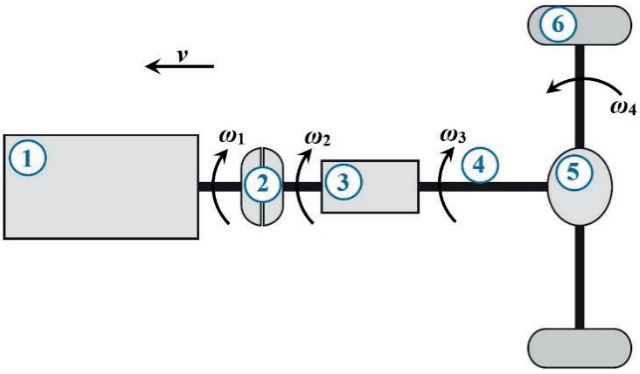

Figure 4 *Scheme of the powertrain with marking of velocities of individual components [5]*

by numbers 1 to 6: 1 - an engine, 2 - a hydrodynamic converter, 3 - a gearbox, 4 - a propeller shaft, 5 - a differential, 6 - drive wheels.

The hydrodynamic converter has two main moving rotating parts - a pump and a turbine. Angular velocity of a pump is ω_P and angular velocity of a turbine is ω_T. Other angular velocities are marked as following: ω_1 - angular velocity of an engine crankshaft, ω_2 - angular velocity of an input shaft to a gearbox, ω_3 - angular velocity of an output shaft of a gearbox (a propeller shaft), ω_4 - angular velocity of a drive axle shaft. When the angular velocities are expressed using the angular coordinates, one gets $\omega_1 = \dot{\varphi}_1, \omega_2 = \dot{\varphi}_2, \omega_3 = \dot{\varphi}_3$ and finally $\omega_4 = \dot{\varphi}_4$.

It is considered that all the connecting shafts of the mechanical system are rigid. This means, that $\omega_P = \omega_1$ and $\omega_T = \omega_2$ or that $\varphi_P = \varphi_1$ and $\varphi_T = \varphi_2$. Angular velocities of individual rotating components are shown in Figure 4.

Thus, it is supposed that the mechanical system has two degrees of freedom (2 DOF) and generalized coordinates are φ_1 and φ_2, i.e. angular deviation of the input shaft to the gearbox (the same with the engine crankshaft and the pump shaft) and angular deviation of the output shaft from the gearbox (the same with the turbine shaft).

Kinetic energy of the system is:

$$E_K = \frac{1}{2} \cdot I_E \cdot \omega_1^2 + \frac{1}{2} \cdot I_P \cdot \omega_1^2 + \frac{1}{2} \cdot I_T \cdot \omega_2^2 + \frac{1}{2} \cdot I_4 \cdot \omega_3^2 + \frac{1}{2} \cdot I_5 \cdot \omega_4^2 + \frac{1}{2} \cdot m_{tot} \cdot v^2, \quad (2)$$

where I_E - moment of inertia of the engine crankshaft, I_P - moment of inertia of the pump, I_T - moment of inertia of the turbine, I_4 - moment of inertia of the shaft between the gearbox and the rear axle differential (a propeller shaft), I_5 - moment of inertia of the drive axle shafts and v - tractor velocity. As can be seen, the kinetic energy includes besides the generalized coordinates φ_1 and φ_2 other coordinates, as well. Hence, the following relations are considered:

$$\omega_3 = \frac{\omega_2}{i_G}, \omega_4 = \frac{\omega_3}{i_C} = \frac{\omega_3}{i_G \cdot i_D}, v = R \cdot \omega_4 = \frac{R \cdot \omega_2}{i_G \cdot i_D}, \quad (3)$$

where i_G - the gear ratio of the activated speed gear, i_D - the permanent gear ratio of the differential and R - the drive wheel radius, which leads to the modified form of the kinetic energy:

$$E_K = \frac{1}{2} \cdot (I_M + I_P) \cdot \omega_1^2 + \frac{1}{2} \cdot \left(I_T + \frac{I_4}{i_G^2} + \frac{I_5}{(i_G \cdot i_D)^2} + \frac{m_{tot} \cdot R^2}{(i_G \cdot i_D)^2} \right) \cdot \omega_2^2, \quad (4)$$

Oror

$$E_K = \frac{1}{2} \cdot I_{1red} \cdot \omega_1^2 + \frac{1}{2} I_{2red} \cdot \omega_2^2 , \tag{5}$$

where I_{1red} and I_{2red} are moments of inertia of rotational components reduced to the pump shaft and the turbine shaft, respectively.

Dissipative energy expresses viscous losses in the system and it is given as:

$$E_D = \frac{1}{2} \cdot b_1 \cdot \dot{\varphi}_1^2 + \frac{1}{2} \cdot b_2 \cdot \dot{\varphi}_2^2 , \tag{6}$$

where b_1 and b_2 are the viscous losses coefficients.

It should be note, that in the solved task, the tractor moves on a straight road without a climb. Therefore, the potential energy is not being changed, i.e. $E_P = 0$.

Further, in the mechanical system of the powertrain the load moments act, namely the drive moment of the engine M_E and moment of the driving resistance M_R. Besides those, moments of the pump M_P and the moment of the turbine M_T, are considered as well. These moments have also to be reduced to the corresponding shafts, i.e. to the pump shaft and to the turbine shaft.

For coordinate φ_1:

$$M_{1red} \cdot \omega_1 = M_E \cdot \omega_1 - M_P \cdot \omega_P . \tag{7}$$

Thus

$$M_{1red} = M_E - M_P . \tag{8}$$

For coordinate φ_2 is considered a simplified situation, namely, the tractor runs-up from the zero speed, i.e. the drag can be neglected and only the weight of the tractor is considered:

$$\begin{aligned} &M_{2red} \cdot \omega_2 = M_T \cdot \omega_2 - m_{tot} \cdot g \cdot \omega_4 \cdot R = \\ &\left(M_T - m_{tot} \cdot g \cdot \frac{R}{i_G \cdot i_D} \right) \cdot \omega_2 . \end{aligned} \tag{9}$$

Thus

$$M_{2red} = M_T - M_R . \tag{10}$$

Now, one can perform the partial derivations of kinetic and dissipative energies with respect to the generalized coordinates φ_1 and φ_2; the right-hand sides of the equations of motion include reduced moments described above.

Hence, one gets the equations of motion of the semitrailer tractor powertrain in the following form:

$$\begin{aligned} &(I_M + I_P) \cdot \ddot{\varphi}_1 + b_1 \cdot \dot{\varphi}_1 = M_E - M_P \\ &\left(I_T + \frac{I_4}{i_G^2} + \frac{I_5}{(i_G \cdot i_D)^2} + \frac{m_{tot} \cdot R^2}{(i_G \cdot i_D)^2} \right) \cdot \ddot{\varphi}_2 + b_2 \cdot \dot{\varphi}_2 = \\ &M_T - M_R , \end{aligned} \tag{11}$$

or in a shortened form:

$$\begin{aligned} &I_{1red} \cdot \ddot{\varphi}_1 + b_1 \cdot \dot{\varphi}_1 = M_{1red} \\ &I_{2red} \cdot \ddot{\varphi}_2 + b_2 \cdot \dot{\varphi}_2 = M_{2red} , \end{aligned} \tag{12}$$

where I_{1red} and I_{2red} are the moments of inertia reduced to the shaft rotating at the angular velocity $\dot{\varphi}_1$ and to the shaft rotating at the angular velocity $\dot{\varphi}_2$, respectively. They are:

$$\begin{aligned} &I_{1red} = (I_M + I_P) \\ &I_{2red} = \left(I_T + \frac{I_4}{i_G^2} + \frac{I_5}{(i_G \cdot i_D)^2} + \frac{m_{tot} \cdot R^2}{(i_G \cdot i_D)^2} \right). \end{aligned} \tag{13}$$

Matrix forms of Equations (11) and (12) are:

$$\begin{bmatrix} I_M + I_P & 0 \\ 0 & I_T + \frac{I_4}{i_G^2} + \frac{I_5}{(i_G \cdot i_D)^2} + \frac{m_{tot} \cdot R^2}{(i_G \cdot i_D)^2} \end{bmatrix} \cdot \begin{bmatrix} \ddot{\varphi}_1 \\ \ddot{\varphi}_2 \end{bmatrix} + \begin{bmatrix} b_1 & 0 \\ 0 & b_2 \end{bmatrix} \cdot \begin{bmatrix} \dot{\varphi}_1 \\ \dot{\varphi}_2 \end{bmatrix} = \begin{bmatrix} M_E - M_P \\ M_T - M_R \end{bmatrix} \tag{14}$$

and

$$\begin{bmatrix} I_{1red} & 0 \\ 0 & I_{2red} \end{bmatrix} \cdot \begin{bmatrix} \ddot{\varphi}_1 \\ \ddot{\varphi}_2 \end{bmatrix} + \begin{bmatrix} b_1 & 0 \\ 0 & b_2 \end{bmatrix} \cdot \begin{bmatrix} \dot{\varphi}_1 \\ \dot{\varphi}_2 \end{bmatrix} = \begin{bmatrix} M_{1red} \\ M_{2red} \end{bmatrix}. \tag{15}$$

3 Obtaining the calculations' results

Equations of motion derived in the previous section (Equations (11), or (12)) have been solved in the Matlab software. However, these equations of motion should be adapted for demands of this software. Firstly, accelerations (in this case angular accelerations) must be independent. Thus, one gets:

$$\begin{aligned} &\ddot{\varphi}_1 = \frac{1}{I_{1red}} \cdot (M_{1red} - b_1 \cdot \dot{\varphi}_1) \\ &\ddot{\varphi}_2 = \frac{1}{I_{2red}} \cdot (M_{2red} - b_2 \cdot \dot{\varphi}_2). \end{aligned} \tag{16}$$

The next modification relates to the fact, that the used software is not able to solve differential equations of the second order. Therefore, Equations (16) must be substituted by the differential equations of the first order.

For coordinate φ_1, we consider the following substitution:

$$\theta_1 = \dot{\varphi}_1, \quad \theta_2 = \varphi_1 . \tag{17}$$

Then, their derivations are:

$$\theta_1' = \ddot{\varphi}_1, \quad \theta_2' = \dot{\varphi}_1 = \theta_1 \tag{18}$$

It is similar for the coordinate φ_2:

$$\theta_3 = \dot{\varphi}_2, \quad \theta_4 = \varphi_2 , \tag{19}$$

$$\theta'_3 = \ddot{\varphi}_2, \quad \theta'_4 = \dot{\varphi}_2 = \theta_3\,. \qquad (20)$$

When the derived equations of motion in Equation (11) of the tractor powertrain are rewritten by considering formulations in Equations (17) to (20), one obtains four differential equations of the first order:

$$\begin{aligned}
\theta'_1 &= \frac{1}{(I_M + I_P)} \cdot (M_E - M_P - b_1 \cdot \theta_1) \\
\theta'_2 &= \theta_1 \\
\theta'_3 &= \frac{1}{\left(I_T + \frac{I_4}{i_G^2} + \frac{I_2}{(i_G \cdot i_D)^2} + \frac{m_{tot} \cdot R^2}{(i_G \cdot i_D)^2}\right)} \cdot \\
&(M_T - M_R - b_2 \cdot \theta_3) \\
\theta'_4 &= \theta_3 .
\end{aligned} \qquad (21)$$

Figure 5 shows the obtained results of calculation of Equation (21). The output values are calculated in the time interval of 15 s. Figure 6 additionally depicts details of the results obtained in the time interval of 2.5 s.

Figure 5 includes two parts. Torque of the used combustion engine (M_E) and moments of the hydrodynamic converter for the pump part (M_P) and the turbine part (M_T) are in the upper part of the figure.

Rotational speed of the input shaft (n_P) and of the output shaft (n_T) together with tractor speed (v) are shown in the lower part of Figure 5.

After starting the combustion engine and reaching the sufficient torque, the pump of the converter begins to increase; thus the torque of the turbine also begins to increase. In the turbine torque, the moment of loads (i.e. driving resistance) is included. After certain time interval, in this case approx. after 2.8 s, the equilibrium of the torques is reached and then the torques are practically identical.

A similar situation is detected for the rotational speeds of the pump shaft and the turbine shaft. Rotational speed of the pump shaft n_P begins to increase with the engine torque M_E. The rotational speed of the turbine shaft n_T is lower than the n_P, because the hydrodynamic converter causes certain decreasing of the rotational speed due to losses. The tractor velocity v is stabilized at the value of 26.64 km/h (a purple curve).

Waveforms of the outputs are calculated for one particular activated gear ratio in the gearbox. In the real operation of the tractor, a driver (either manually of automatically) would change the gear ratio, values of torques would decrease (in the case of driving on the

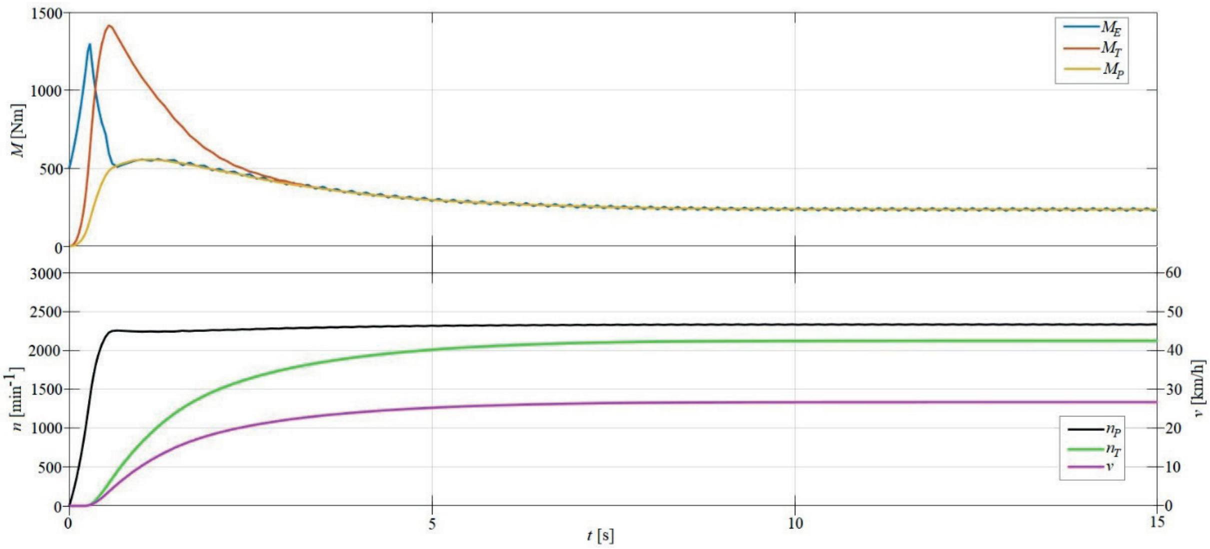

Figure 5 *The results of calculation of the tractor powertrain mathematical model, the time interval 0 to 15 s*

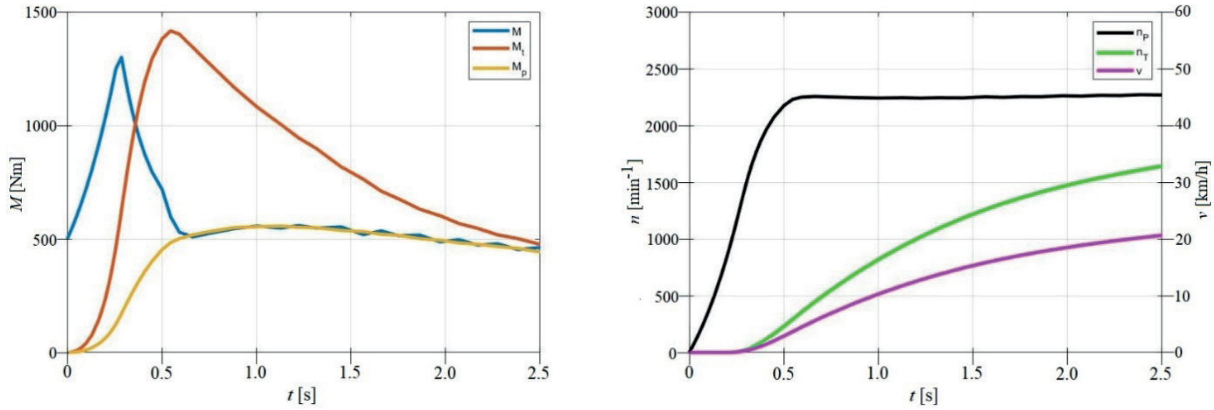

Figure 6 *The detail of waveforms of the obtained results, the time interval 0 to 2.5 s*

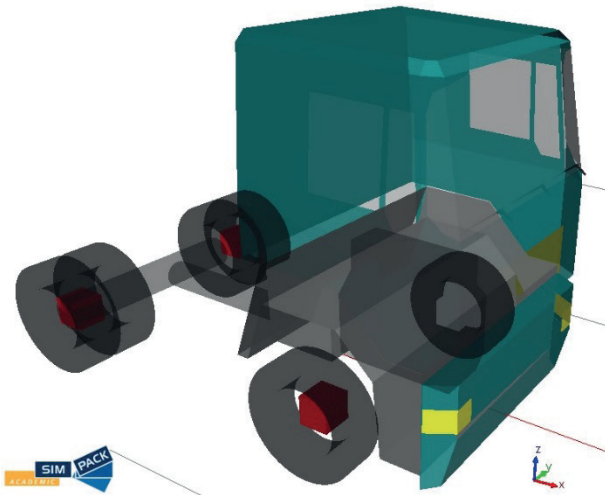

Figure 7 *An illustration of a semi-trailer tractor trailer created in the Simpack software*

flat road, without climb) and the driving speed would be higher. In the case of driving at the higher speed, the drag should be already considered in the mathematical model and it would be incorporated within the moment of driving resistance M_R.

The future research will be focused on modelling the operation of the semitrailer tractor powertrain. The modified mathematical model will be more difficult. It will include other parts for simulation of all the driving resistances, driving in a climb, comparison of outputs values for driving with fully loaded semitrailer etc. As the other step of the future research, challenge for the authors will be to set-up a multibody model (MBS model) of the considered combination vehicle. On one hand, the modelling of the multibody model in the MBS software does not need to derive equations of motion, they are created automatically by a software. On the other hand, it is not possible to check and compare the equations of motion derived by the researcher (let say "manually") with equations of motion derived by the software during the modelling process. Figure 7 shows an ongoing state of the semi-trailer tractor model created in the Simpack software. The research will continue in order to reach a representative MBS model of the tractor and it will serve for performing the simulation computations and evaluation of wanted quantities.

4 Conclusion

The presented research has brought an overview of a creation of the mathematical model for investigation of dynamical properties of a semi-trailer tractor powertrain. This particular tractor uses the hydrodynamic converter of torque. The mathematical model was derived by means of the Lagrange's equations of motion of the second kind method. Calculated values have included output torques and rotational speed of shafts of the powertrain. The achieved results show their graphical waveforms, which are essential for evaluation of the tractor behavior during operation. Tractor's acceleration leads to increase of torque on the drive shaft. After certain time, the mechanical system reaches the equilibrium. It means that the torques of the drive shaft and the driven shaft have the same characteristics in terms of their waveforms. Hence, values of torques are identical. Further change of torques, rotational speeds of driving speed ,will happen after a change of the gear ratio of a change of the external loads (e.g. a change of driving resistance). These driving resistances include mainly the drag at higher driving speeds, inertia resistance during accelerations and the resistance due to climbs. Another important driving resistance is the rolling resistance. It appears at any driving speed and depends on the rolling friction coefficient and on the total weight of the tractor (or of the entire combination vehicle).

The main objective of the performed researchs activities has been to create a basic mathematical model of a semi-trailer tractor powertrain. The further step in the research is to set-up a more complex model, which will also include effects of other phenomena, like flexibility of shafts, a change of weight of a vehicle, driving in climbs and others. An intention of researchers is, besides the mathematical model solved in the Matlab software to set-up a multibody model of the tractor or the entire combination vehicles. The advantage of having two independent mathematical models of a mechanical system in two different software is the possibility to compare the obtained results to each other. Then, it will be possible to investigate the operational properties of the vehicle (with various parameters) and to reveal possibilities to improve some parameters, mainly in terms of the required power during the operation, fuel consumption and others, which influence effective and economical operation of a vehicle.

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