Analysis of the influence of assembly electric motors in wheels on behaviour of vehicle rear suspension system

P Dukalski¹  B Będkowski¹  K Parczewski²  H Wnę³  A Urbaś³  K Augustynek³

¹Institute of Electrical Drives and Machines – Komel
²University of Bielsko-Biała, Department of Combustion Engines and Vehicles
³University of Bielsko-Biała, Department of Mechanical Engineering Fundamentals

Corresponding author e-mail: [p.dukalski, b.bedkowski]@komel.katowice.pl

Abstract. The increase in interest in electric vehicles is associated primarily with the actions of states that seek to sell new vehicles with low emissions of harmful compounds into the atmosphere. This is codified by emission standards. One of the solutions of the exhaust emission problem is the use of the electric drive (assuming that electricity comes from renewable sources). The electric drive system can be developed in several ways, depending on how the drive torque is transmitted: by gears or directly to each wheel. These constructional solutions affect the largeness of unsprung masses and require proper selection of vehicle suspensions. Series of tests and simulations has been carried out to verify the mathematical model proposed for the rear suspension system with electric motors mounted in the wheels. The simulation model has been prepared in the MSC.Adams package. This model was (tuned) verified by comparison the dynamic response of the system with the experimental results. The verified model was used to analyse the impact of the shock absorber damping characteristics on tire deflection and contact forces acting between the road and tire.

1. Introduction
It is expected that in the next few years the number of electric vehicles will increase significantly. The reason for this is the possibility of reducing fuel consumption and reduction of exhaust emissions. The basic advantages of this type of drive are: low noise of the vehicle, simplicity of the drive system, ease of control and operation, and lack of exhaust gases emitted to the atmosphere [1] and lower cost of one kilometre of mileage and low cost of vehicle servicing. Another advantage of an electric vehicle is that it can recoup energy during braking. The disadvantages of electric vehicles are still: high mass, low battery capacity, loss of battery capacity at low temperatures. The range of electric vehicles is now clearly smaller than in the case of vehicles powered by internal combustion engines and the time of their charging is clearly longer than the time of refuelling. The use of air conditioning and other appliances consuming electricity causes a significant reduction in the range of the vehicle. Despite these inconveniences, all car manufacturers try to have several electric vehicles in their range of products. Such solutions are also in the interest of companies producing electric motors.

2. Solutions of drive systems in electric vehicles
Today there are many configurations of drive systems for electric vehicles, starting with the most similar to vehicles with internal combustion engine - replacing the internal combustion engine by the electric motor, by replacing the gearbox by a gear fixed to the motors driving the individual vehicle
wheels mounted to the frame/body, while maintaining the fixed transmission and driveshaft, and ending with electric motors mounted on vehicle wheels fitted or not with fixed gearboxes. Fig. 1 shows schematically the individual solutions of drive systems.

The first solution is similar to that used in vehicles powered by internal combustion engines (Fig. 1a). The advantage of the second solution (Fig. 1b) is to give up the clutch and gearbox and use a fixed transmission, which reduces the weight of the driveline system and reduces production costs. The solution was shown in f Fig. 1c differs from the previous with transverse engine placement. Another solution (Fig. 1d) shows the driveline system in which the differential mechanism was abandoned. This allowed the use of two motors with lower power driving the right and left wheels individually. The use of this solution requires the controlling of the rotational speed of the electric motors when driving in corners or during braking [2,3]. In this case, the drive torque is transmitted to the wheels via a driveshaft and therefore the unsprung masses do not change. In the case of the solutions shown in Figs. 1e and 1f, an electric motor with or without a fixed transmission is mounted in the wheel of the vehicle[11]. This affects the increase of the unsprung mass of the vehicle. Advantage of this solution is leaving space between the wheels of the vehicle which can be used, for example for location battery pack. The modification of this solution, shown in Fig. 1f, may be the assembling of electric drive motors in rear wheel hubs.

3. Constructional solutions for electric motors designed for mounting on vehicle wheel hubs
Considered in the article, the drive is equipped with two electric traction motors built into the wheel hubs. KOMEL motors are synchronous motors, permanent magnet inductors (Fig. 2).

The calculated parameters of the electric motor operation are presented in Tab. 1.
Table 1. Parameters of electric drive system

| Parameter       | Value                           |
|-----------------|---------------------------------|
| $U_{L1}$ [V]    | 200                             |
| $P_e$ [kW]      | 42                              |
| $M_e$ [Nm]      | 400 (for $n_e = 0 \div 1000 \text{obrmin}^{-1}$) |
| $P_{max}$ [kW]  | $\sim 80$                      |
| $M_{max}$ [Nm]  | $\sim 900$ (for $n_e = 0 \div 850 \text{obrmin}^{-1}$) |

The drive system equipped with two electrical motors of this type should ensure dynamic driving of a B class car (Fiat Panda III). For vehicles with a larger weight, in order to obtain suitably high driving parameters, a drive with four engines should be used.

The overall dimensions of the developed electric motor have been matched to the 17" wheel rims. The available design space for the electric motor constructor is limited by the outside diameter, shape and weaning of the rim and the brake drum that is built into the engine (if is included in the brake system). The motor is mounted to the swingarm through the anchor disc (Fig. 3).

Figure 3. The 3D model of assembly of the SMZs200S32 electric motor from KOMEL

The components of the electric motor can be divided into fixed elements and movable elements, through which the torque is transferred directly to the wheel of the vehicle. The presented electric motor is a motor with an external rotor. The rotating element is a hull acting as a rotor, in which a magnetic core with permanent magnets is mounted. The fixed element is an anchor disc with a supporting structure, with a labyrinth type cooling system and a stator consisting of a wound magnetic core. A rotor position sensor is necessary to control the motor. The motor has an incremental encoder as standard.

4. Research on the impact of assembly of electric motors in wheels on the work of suspensions during overcoming road unevenness

The tire undergoes deformation under the influence of the force exerted on the wheel. This deflection, during a steady straight vehicle movement, on a flat horizontal road results from the magnitude of the force transmitted to the wheel axle by suspension and approximately corresponds to the stationary distribution of the loads on the wheels. When the car moves at a non-uniform speed, the tire will be additionally loaded or unloaded depending on the deceleration or acceleration of the car. Determination these loads require designation the inclination angle of the road surface and changing the height of the centre of the wheel when moving on uneven roads. The method for determining the forces acting on the wheel axle have been proposed by Zegelaar [7] is shown in Fig. 4.
Due to the shape of the tire, overcoming the road unevenness will occur from the moment of contact between the tire and the obstacle until lose contact with it after running off the obstacle. The section of the road, when the tire is in a contact with the obstacle depends on the dimensions of the tire and the height and width of the obstacle. The model approach to describe the unevenness of the road was initiated by Bandel (1988), then developed by Zegelaar and completed by Schmeitz [4,7]. Bendel found that the function describing the change in vertical reaction force during overcoming a short rectangular obstacle, consisting of increasing the load when driving into an obstacle and reducing the load during descent, can be composed of two identical basic functions, the second of which is a mirror image of the first. The basic functions are approximately independent of the initial deflection of the tire, which means that they are independent of the height of the wheel axle. The study was based on a Class B passenger car in which measuring equipment was installed, allowing for measurements of distance the wheel axle to road surface, suspension deflections, accelerations acting on the wheel axle and vehicle body directly above the wheel axle and driving speed (Fig. 5). The results were recorded on the disk of the measuring apparatus at 100Hz [5]. The tire pressures were in accordance with the vehicle manufacturer's recommendations and the same in the vehicle without masses simulating the electric motors in the wheels and with them.

Figure 4. Wheel rolling over road unevenness for a determined wheel axle load, assignation of the effective wheel motion plane

Figure 5. Test object with mounted measuring apparatus
The measurements were carried out on a flat, level road surface [6]. The triangular road unevenness (L50 profile) was chosen for comparison, which corresponds to overcoming an unevenness of the height of 38 mm with a base length of 85 mm. The obstacle was driven at speed of ~12.5 km/h⁻¹.

5. Structural model of the rear suspension system.

Fig. 7 shows the simulation model of the rear suspension system of the Fiat Panda III. The multibody formalism is applied to describe the dynamics of the system. The mathematical model is developed by means of the MSC.Adams commercial package. Solid models of the rear suspension components are prepared in the Autodesk Inventor package. The mass parameters of these components are listed in Tab. 2. Two additional masses are attached to the suspension springs in order to model the influence of the body mass on the movement of the system.
Table 2. Mass parameters of parts of the rear suspension system

| Name of the component | mass [kg]  | mass moment of inertia [kgm^2] |
|-----------------------|-----------|---------------------------------|
|                       | \(m\)     | \(I_x\)                         | \(I_y\) | \(I_z\) |
| suspension beam       | 20.2      | 4.7725                          | 4.4934  | 0.3407  |
| brake drum            | 5.3       | \(5.3173 \cdot 10^{-2}\)        | 2.8430 \cdot 10^{-2} | 2.8430 \cdot 10^{-2} |
| tyre                  | 8.9       | 0.2822                          | 0.1681  | 0.1676  |
| tire                  | 7         | 1.1403                          | 0.6383  | 0.6382  |
| rotor                 | 12        | 0.3694                          | 0.1989  | 0.1988  |
| wheel disk            | 1.9       | \(1.7616 \cdot 10^{-2}\)        | 9.1581 \cdot 10^{-3} | 9.1406 \cdot 10^{-3} |
| axle shaft            | 1.4       | \(1.4663 \cdot 10^{-3}\)        | 1.2233 \cdot 10^{-3} | 1.1155 \cdot 10^{-3} |
| stator                | 24        | 0.5418                          | 0.2999  | 0.2997  |

The last elements used in the simulation model are spring-damping elements, which are used to model the spring suspension and shock absorber. Parameters of these elements adopted in simulations are presented in Tab. 3.

Table 3. Parameters of springs and dampers applied in simulations

| springs | dampers |
|---------|---------|
| stiffness coefficients \(k_s\) [Nm^{-1}] | length at preload [m] | damping coefficients \(c_s\) [Nsm^{-1}] |
| 18000   | 0.35    | 1200 |
|         |         | 2400 |
|         |         | 3600 |

A standard contact element is applied to model the contact between the tire and the road, which is part of the MSC.Adams/View module. Its parameters are selected to be compatible with real vehicle conditions (Tab. 4).

Table 4. Parameters of the contact tire-road

| Contact parameters | Contact parameters |
|--------------------|--------------------|
| stiffness coefficients \(k_p\), Nm^{-1} | 5.2 \cdot 10^3 |
| force exponent \(\delta\)                  | 1.1               |
| damping coefficients \(c_p\), Nsm^{-1}    | 4.8 \cdot 10^3    |
| static friction coefficient \(\mu_s\)     | 0.8               |
| dynamic friction coefficient \(\mu_d\)    | 0.6               |
| stiction transition velocity \(v_s\), ms^{-1} | 0.1 |
| friction transition velocity \(v_f\), ms^{-1} | 1 |

Presented the contact model is acceptable due to the low speed of the vehicle while overcoming an obstacle.

6. Measurement results and numerical calculations

Experimental tests are carried out, in order to verify the mathematical model of the rear suspension system. The results from the experiment are compared with those from computer simulations. Fig. 8 presents the vertical displacements of the wheel centre obtained for the standard vehicle (Fig. 8a) and the one equipped with electric motors in wheels (Fig. 8b).
It can be seen good agreement of the courses obtained from the simulation with ones taken from the experiment when the vehicle overcomes the obstacle. The Pearson correlation coefficient ($r_{EC}$) and the integral relative error ($I_{EC}$) are used to analyze the compatibility of these courses. In the case when the masses of the electric motors are omitted then the values of these factors are as follows: $r_{EC} = 0.96$ and $I_{EC} = 22.6\%$. If the masses of the electric motors are taken into account then obtained the values are $r_{EC} = 0.92$ and $I_{EC} = 8.7\%$. According to the authors, the obtained results are acceptable. After overcoming the differences between courses arise with time. It can be explained by the simplifications used in the model of the influence of the body on the rear suspension system, assumed constant damping coefficient of shock absorber (for jounce and rebound) and the change of the tire stiffness caused by a smaller tire-road contact area while the vehicle overcomes the obstacle.

The simulation model well shows the test results, the differences occur due to the adopted shock absorber damping model (constant value of the damping coefficient), and the assumption of a constant stiffness value of the tire, which also decreases if the contact surface is reduced.

In the further part of the paper, the influence of the damping coefficient of the shock absorber on the vehicle motion is analysed. Figs. 9 and 10 shows, respectively, the course of vertical displacements of the centre of the wheel and the contact force acting between the road and tire obtained for the standard vehicle (a) and the system with electric motors embedded in the wheels (b).
Analysing the results obtained, it can be seen that applied damping coefficient of the shock absorber has a negligible influence on values of the contact force and vertical displacement of the centre of the wheel. These ones are significantly larger when an additional unsprung mass resulting from the electric engines embedded into the wheels is applied.

7. Summary and Conclusions

The research carried out by the project team is aimed at determining the influence of the electric motors embedded in the wheel hub on the motion of the vehicle. The results will be used to determine the possible mass of the electric motor which can be appropriate for this type of the electric drives. The extra unsprung mass is one of the main delimiters for the constructors who need to achieve very high torque over a wide speed range with high motor efficiency.

The article discusses the solution of the construction of a prototype wheel hub motor, developed and manufactured by the Institute of Electrical Drives and Electrical Machines KOMEL. Problems resulting from the increase of unsprung masses are shown on the example of research and simulations carried out at University of Bielsko-Biała. The presented analysis shows that the assembling electric motors at the rear wheels affects the size of the unsprung weight of the rear suspension of vehicle. This increases the amplitudes of the rear suspension vibrations and increases the time of their suppression. The simplified suspension simulation model was used for analyses, which will be extended in later works. Despite this the simulations carried out in this paper allow us to indicate the direction of changes in suspension damping characteristics. Applying electric drive motors in the wheels causes that the change of damping coefficients has a stronger effect on the amplitude values of vibrations, both during the entrance to the obstacle and after the drove through the obstacle. It also increases the value of forces at the tire-road interaction. The analysis of the impact of vibrations of the rear suspension system with and without electric drives will be the subject of the further research. The tests presented in this publication should be carried out for individual wheel hub motor applications that differ in the suspension system and the mass of the motor meeting the requirements of the drive parameters.

The project aims to develop a number of constructional solutions that will allow the development of motors for installation in wheel hubs for various types of applications and for different requirements of individual customers. Motors of this type offer very good benefits in the automotive industry so they can be widely used in the electric vehicle industry for a variety of applications: from small city cars, by personal and family cars, for vans and buses. It should be noted that motors built into wheel hubs can also be used to support other types of drive, including combustion engines, for example when starting, when the highest torque values or manoeuvres are required from the drive, increasing the driving dynamics.
The project "Innovative Solutions for Direct Drive of Electric Vehicles", co-financed by National Centre for Research and Development under the LIDER VII program, in accordance with the agreement: LIDER / 24/0082 / L-7/15 / NCBR / 2016 (Poland)

Literatura
[1] Merkisz J., Pielecha I.: 2015, Układy elektryczne pojazdów hybrydowych, Wydawnictwo Politechniki Poznańskiej, Poznań, pp.12-19, 139-154
[2] Dzida J.: 2017, Porównanie różnych sposobów kierunkowego napędzania pojazdów silnikami elektrycznymi, Napędy i Sterowanie, 2, Racibórz, pp.50-55
[3] Ehsani M., Gao Y., Gay S. E., Emadi A.: 2004, Modern Electric, Hybrid Electric and Fuel Cells Vehicles, Fundamentals, Theory and Design, CRC Press, London, pp.99-116
[4] Pacejka H. B.: 2006, Tire and vehicle dynamics, SAE, Warrendale pp.484-512
[5] Grążiewicz K., Pokorski J.: 2015, Układ pomiarowy AD-32 Grapol Electronic, Warszawa,
[6] Parczewski K., Wnęk H.: 2017, Impact of tire inflation pressure during overcoming of road unevenness. Proceedings of 21st International Conference Transport Means. Kaunas, Part 1, pp.154-157
[7] Zegelaar P W A.: 1998, The dynamic response of tyres to brake torque variations and road unevennesses, PhD Thesis Delft University of Technology, Delft,
[8] Gawron S., Bernatt J.: 2017, Doświadczenia z eksploatacji samochodów elektrycznych w działalności gospodarczej, Maszyny Elektryczne Zeszyty Problemowe, Katowice 2(102)
[9] Dukalski P., Będkowski B., Wolnik T., Urbaś A., Augustyniek K.: 2017, Założenia projektu silnika do zabudowy w piaście koła samochodu elektrycznego, Maszyny Elektryczne - Zeszyty Problemowe, Katowice 2(102)
[10] Król E., 2014, Silniki Elektryczne w Napędach Pojazdów Sportowo-Rekreacyjnych, Maszyny Elektryczne Zeszyty Problemowe, Katowice, 2(102)
[11] Ślaski G, Gudra A, Borowicz A 2014 Analysis of the influence of additional unsprung mass of in-wheel motors on the comfort and safety of a passenger car, Archives of automotive engineering, 3 (65), pp.51-64