388. COUPLING OF ENERGY FLOWS FROM INTERNAL COMBUSTION ENGINE AND ELECTRICAL MOTOR IN HYBRID VEHICLE

M. Starevičius [1, a], K. Pilkauskas [2,b] J. Sapragonas [1,c]
[1] Kaunas University of Technology, Kęstučio 27, Kaunas, LT-44312, Lithuania
[2] Kaunas University of Technology, Mickeviciaus 37, LT-44244, Lithuania
E-mail: [a] martynas.starevicius@ktu.lt, [b]kestutis.pilkauskas@ktu.lt, [c]jonas.sapragonas@ktu.lt

(Received 19 June2008; accepted 29 August 2008)

Abstract. Dynamical model of the hybrid electrical vehicle transmission with internal combustion engine and electrical motor driving the rear and the front wheels accordingly is developed. Particular elements in it are described using classical transmission elements. The model is used for the analysis of acceleration process when the engine and motor operate in non stationary regimes and transmission elements are in transition regimes of motion. Non linear models for clutch and wheel-road interaction description are used. The gear box control algorithm was developed and the vehicle prototype was constructed and tested. The experimental results proved the model validity. The difference in numerical and experimental results for the acceleration at 400 m. distance does not exceed 10 %. The model could be the background for the simplified control algorithm development with no mechanical link between the transmissions. Numerical modeling and experimental research show the functionality of the algorithm. The performed numerical modeling revealed that different from commonly used criteria for velocity ratios selection in the gear box are necessary

Keywords: hybrid vehicle, transmission, tyre road interaction.

Introduction

The tendency of expanding hybrid drives application in automotive engineering is observed. Though different technologies exist in the majority of hybrid vehicles the combination of internal combustion engine and electric motor is used. Started to be used for small city type cars as a means of fuel economy and ecological requirement assurance later this technology was begun to be used for minivans, high-class sedans, off-road and sport utility vehicles [1-5]. The spread of hybrid drives application is preconditioned by the possibility to reduce the power of internal combustion engine because only in rear cases its full power operation is necessary at common regimes of the vehicle exploitation. Thus at additional parallel operation of electric motor the internal combustion engine for a longer period operates at a more efficient and with lower emissions regime what is useful for all types of the vehicles. Nevertheless the application of hybrid drives is inevitably related to the increase of the vehicle mass what in turn increases fuel consumption and lowers the efficiency of internal combustion engine when it operates alone. The experience of batch production hybrid vehicles shows that these vehicles can have advantages only when their structure (power ratio of internal combustion engine and electric motor, battery type and capacity, generator's power) and optimal control algorithms are properly selected. These problems are mutually related.

The configuration of hybrid drives is not jet settled, the following four types [3, 4] can be distinguished: in series, parallel, series-parallel, combined. In parallel configuration hybrid drives it is necessary to use electric motors generators, because common electric motors usually are used for energy regeneration at braking. The greatest possibilities to conduct experimental research can be when developing combined configuration systems as the advantages of separate systems can be used (at the expense of the system complexity). Such system was selected as an object for the investigation.

After the analysis of possible configuration schemes of the transmission the decision was made to use 4x4 transmission scheme with mechanically non linked internal combustion engine and electric motor transmissions. This makes it possible in the experimental vehicle-prototype to change its units in a simpler way and to use different combinations of them. The research was aimed at a single vehicle performance regime – its acceleration. The aim of this research is to construct dynamical model for experimental hybrid electric sport utility vehicle as closely as possible representing the real vehicle and enable both optimal combining of the vehicle units and construction of simple control algorithms for internal combustion engine and electrical motor.

Vehicle Characteristics

4x4 type vehicle configuration when the engine and the motor drive separate wheels allows simplifying the transmission when changing the mechanical drive into the electric one on one of the axles. After the analysis of the possible power train configurations, the following configuration was chosen: internal combustion engine at front of the vehicle, and electric motor at the rear (Fig.1.).

The main characteristics of the vehicle are presented in Table 1. Though the electrical motor operates at characteristic short period regime the additional fluidic cooling system for it and its controller had to be installed what led to the increase of the total vehicle mass.

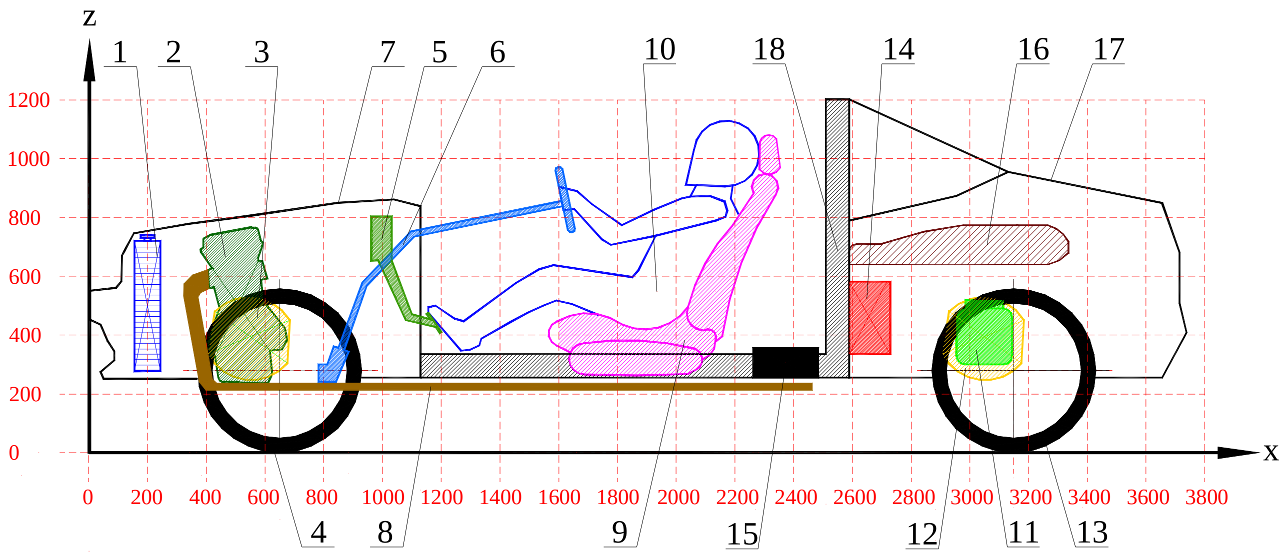

Fig. 1. Configuration of hybrid electric sport utility vehicle prototype: 1 – radiator, 2 – internal combustion engine, 3 – front gear box, 4 – front wheels, 5 – pedal unit, 6 – steering mechanism, 7 – front part of the vehicle body, 8 – exhaust system, 9 – seats, 10 – driver, 11 – electrical motor, 12 – rear gear box, 13 – rear wheels, 14 –inverter, 15 – batteries, 16 – fuel tank, 17 – rear part of the vehicle body, 18 – central part of the vehicle body

Dynamical model of the vehicle consists of the description of internal combustion engine and electrical motor output characteristics, transmission dynamical model, tyre – pavement interaction model.

Table 1. Main technical characteristics of the vehicle

Internal combustion engine Honda1.6 L, 16 valve Fuel Power, kW / rpm Torque, Nm / rpm Gearbox	 Gasoline 77 / 6500 133 / 4500 5 speed manual	***Generator*** Voltage, V Current, A Power, kW	G290B 24V 150A 24 150 3.6
Electric motor **Siemens**1PV5105 WS12 Type Power (peak / rated), kW Peak torque, Nm / rpm Time sustaining max ratings, min Transmission		 AC induction motor 78.4 / 18 125 / 1000 1.5 2 speed reduction gear, based on Honda Civic final drive	
Vehicle mass, kg		1140	
Load distribution between axles (front / rear), %		51 / 49	

Descriptions of Internal Combustion Engine and Electrical Motor

Making analysis of control algorithms effectiveness, i.e. taking into account that control system will seek to adjust the operation of internal combustion engine and electrical motor one to another the output characteristic description of the engine and motor at full power operation are not enough. [6]. The full internal combustion engine characteristics was modeled using the method of analogies [7]. The characteristics of electrical motor are described using the manufacturer data [8]. The characteristics of the engine and the motor power unit are presented in Fig. 2.

The case of synchronous operation of both the engine and the motor is presented in Fig. 2. At synchronous operation of both the engine and the motor the output torque characteristics is strongly distorted and the problem of optimal selection of velocity ratios for the transmissions appears. The problem was more precisely analysed after construction of the dynamical model for the vehicle transmission.

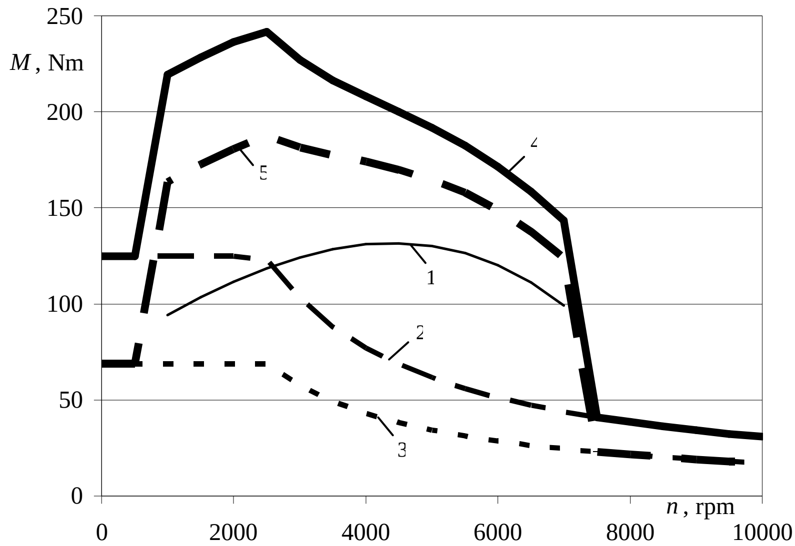

Fig. 2. Torque generated by the power unit under investigation versus rotation frequency: Internal combustion engine, 2 – maximal torque of electrical motor at 144 V DC feeding voltage, 3 – nominal torque of electrical motor at 144 V DC feeding voltage, 4 – resultant maximal torque of hybrid power unit, 5 – resultant torque of hybrid power unit

Dynamical Model of the Vehicle Transmission

Dynamics of this type transmission is already analysed [9, 10] using typical drive elements (Fig. 3). Précised characteristics the drives are given in the publication [11]. Using the constructed numerical model operation of the transmissions of the motor and the engine were analysed at typical acceleration regimes – start up motion throwing in the clutch (for electrical motor the case when it operates with no clutch was analysed), acceleration with fully thrown in clutch, gear shift process which consists of three stages – gear switching of by throwing out the clutch, synchronization of primary and secondary shafts in the gear box, gear change and the clutch throwing in. The analysis of these regimes is already performed and presented in publications [9, 10] when the possibilities to use the motor and engine for synchronization speed increase of the primary and secondary shafts were researched. That's why the algorithms obtained were used for further research. There was the need to modify the model not taking into account the influence of generator. In general the model allows the analysis of internal combustion engine – generator interaction but the generator applied was of comparatively low power. An electromagnetic clutch is incorporated into the structure of generator's drive and with its help the generator is disconnected at the acceleration phase. Numerical modeling revealed that there is no sense to connect generator at the periods when the internal combustion engine is unloaded (e.g. during gear shift) because this will cause additional dynamical loads but has no essential influence on energy consumption.

The constructed dynamical model of the hybrid electric vehicle transmission is presented in Fig.3. In it the following designations are used: I_1, I_2, I_3, I_4, I_5, I_6, I_7, I_8, I_{10}, I_9 and I_{11} are accordingly moments of inertia of internal combustion engine with a flywheel, the generator, the clutch driven disk of the engine, components of the engine transmission, stocks of the rear axle wheels, external rubber rings of the rear axle tires, electric motor with a clutch drive disk, the clutch driven disk, components of the motor transmission, stocks of the front axle wheels, external rubber rings of the front axle tires; c_{12}, c_{13}, c_{34}, c_{45}, c_{56}, c_{78}, c_{89}, c_{910} and c_{1011} are accordingly torsional stiffness coefficients of the belt drive, the engine clutch, components of the engine transmission, axle shafts of the rear axles, rear tyres, the motor clutch, components of the motor transmission, axle shafts of the front axles and the front tyres; k_{12}, k_{13}, k_{34}, k_{45}, k_{56}, k_{78}, k_{89}, k_{910} and k_{1011} are accordingly viscous damping coefficients of the belt drive, the engine clutch, components of the engine transmission, axle shafts of the rear axles, rear tyres, the motor clutch, components of the motor transmission, axle shafts of the front axles and the front tyres; M_{VDV}, M_{EV} are torques of internal combustion engine and electric motor, M_{Gen} is the generator torque; M_{fS} and M_{fSE} are frictional torques of the clutches of internal combustion engine and electric motor; m_a is the vehicle mass; $\sum R_{xj}$ is total traction force; $\sum F_P$ is total resistance force.

The goal of this research is, in accordance with dynamical model of the hybrid electric vehicle transmission, to calculate dynamical characteristics of the vehicle acceleration when driving conditions and tyre – road contact models are different.

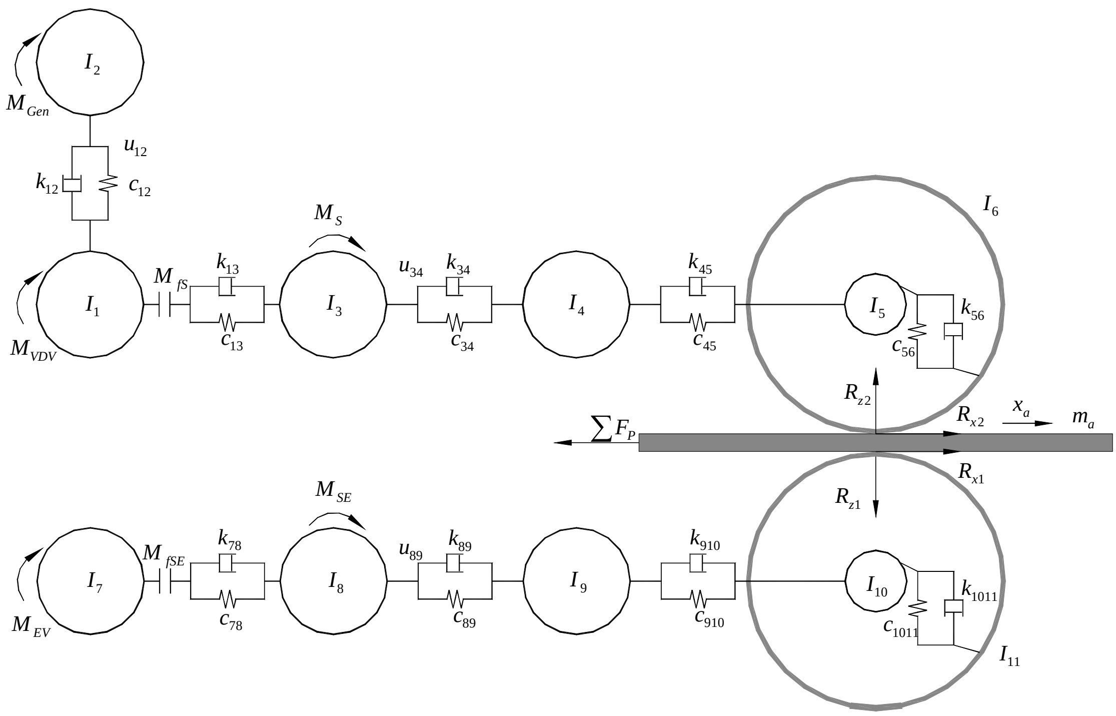

Fig. 3. Transmission model

Tyre road interaction

As it can be seen from dynamical model presented in Fig. 3 traction effort of the vehicle is caused by friction forces appearing at the interaction places of the road and both front and rear wheels:

For the description of friction phenomena at tyre – road contact the model – "magic formula" - given in [12] was used:

$$R_{xj} = R_{zj}\mu_{x0}$$
$$\mu_{x0} = D\sin\left[C\arctan\left\{B\chi - E\left(B\chi - \arctan B\chi\right)\right\}\right] \quad (1)$$
$$\chi = \frac{\dot{x}_a - \dot{\varphi}_j r_{dj}}{\dot{x}_a}$$

Here R_{xj} – traction force, R_{zj} – pavement reaction force (see Fig. 3), μ_{x0} - friction coefficient, χ - relative sliding factor, $\dot{x}_a$ - vehicle velocity, $\dot{\varphi}_j$ - angular velocity of a tyre, r_{dj} – dynamical radius of the wheel.

The dependency of friction coefficient on relative sliding factor is presented in Fig. 4. Here line 1 is the graph of μ_{x0} obtained according "magic formula". It is used for the evaluation of tyre road interaction in the transmission control algorithm.

For the control of output torques developed by internal combustion engine and electrical motor the simplified friction coefficient dependency on relative sliding is used (Fig. 4, line 2). The essential idea of such dependency is that when tyre – road sliding is below critical value the friction coefficient is taken equal to its critical value and when the sliding exceeds critical value the friction coefficient is taken equal to the value of non stable sliding.

The block diagram of the algorithm for tyre road interaction evaluation is presented in Fig. 5.

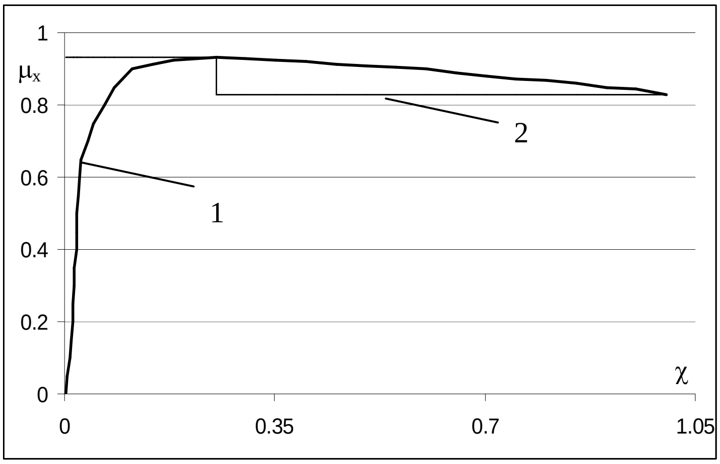

Fig. 4. Dependency of friction coefficient μ_x on relative sliding factor χ

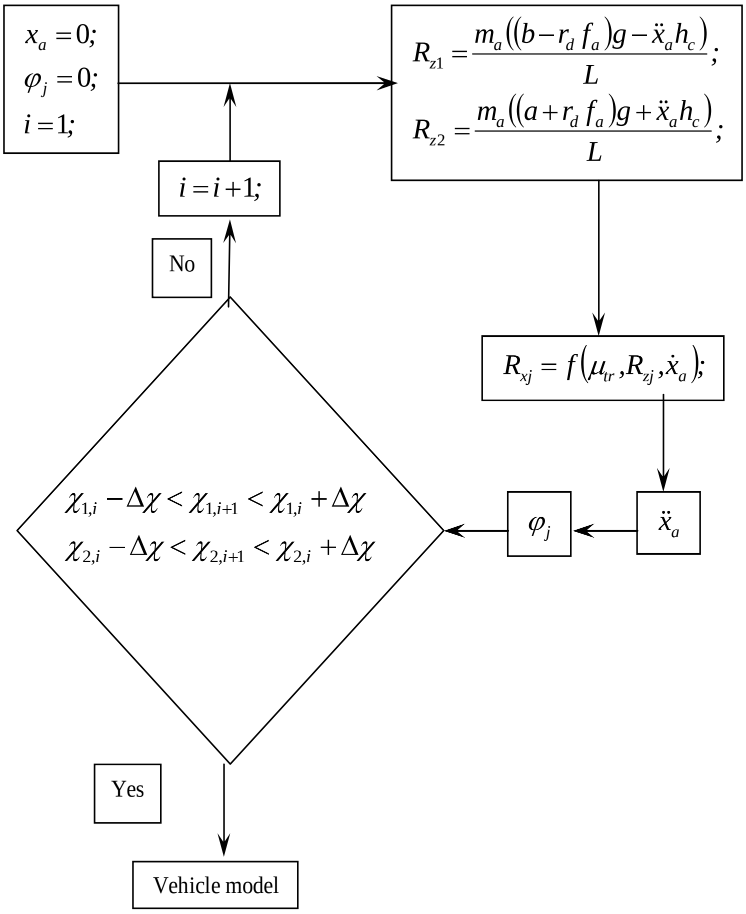

Fig. 5. Block diagram for tyre road interaction evaluation: x_a – vehicle displacement, φ_j – rotation angle(angular coordinate) of the rubber ring of the tyre, j=1,2 – front and rear wheels accordingly, R_{z1} and R_{z2} – vertical reactions of the front and rear wheels accordingly, m_a - vehicle mass, a, b and h – mass center position parameters distances from the front and rear axles and the height with respect to the road, L – vehicle base, f_a – rolling resistance coefficient, g – acceleration of gravity, r_{d1}, r_{d2} – dynamical radii of the front and rear wheels accordingly, R_{xj} – vehicle traction force, μ_{tr} – friction coefficient at road tyre contact.

Results

The vehicle acceleration regime was simulated by numerical methods and investigated experimentally. For the simulation the dynamical model presented in Fig. 3 was used assuming that at initial instant of time the vehicle was stationary – all its motion parameters (acceleration, velocity, displacement) are equal to zero and then it moves with maximal possible acceleration operating only internal combustion engine, only electrical motor or both the motor and the engine.

For experimental research the vehicle prototype presented in Fig. 1 was used. It was tested on horizontal straight track with concrete pavement. The test drive was performed in the following way: from initial stationary position (acceleration, velocity, displacement are equal to zero) the vehicle with maximal possible accelerating rate was driven 400 m. distance operating only internal combustion engine, only electrical motor or both the motor and the engine. Its motion parameters were registered.

The obtained experimental and theoretical results are presented in Fig. 6, Fig. 7.

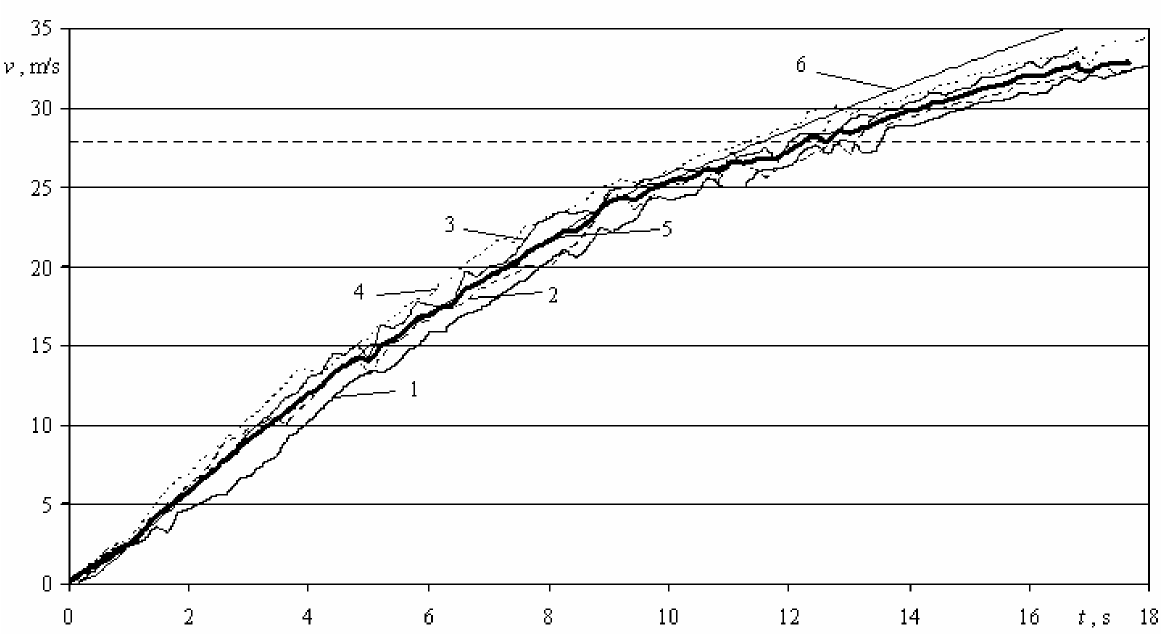

Fig. 6. Dependency of vehicle velocity on time operating only internal combustion engine: 1-4 – test drives, 5 – mean velocity of test drives, 6 – simulation results

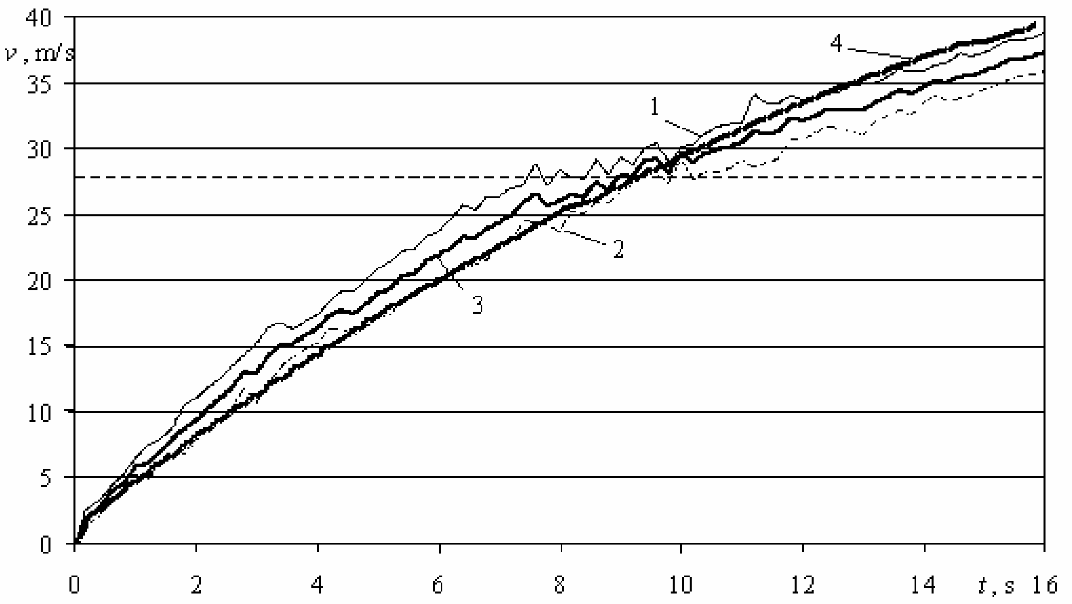

Fig. 7. Dependency of vehicle velocity on time operating both internal combustion engine and electrical motor with the third gear of electrical motor transmission: 1-2 – test drives, 3 – mean velocity of test drives, 4 – simulation results

Conclusions

1. The constructed dynamical model of the hybrid electric vehicle transmission with no mechanical link between internal combustion engine and electric motor transmissions allows simulating the behavior of its elements and the vehicle motion under different conditions. This model serves as a background for the system control strategies and control algorithms development.

2. The constructed vehicle prototype with identical compartments for internal combustion engine and electrical motor drives in the front and the rear of the vehicle enables experimental testing of several different prototype configurations.

3. The obtained simulation and experimental results prove the validity of simple structure algorithm for internal combustion engine and electrical motor control when driving the hybrid electric sport utility vehicle on dry concrete pavement or the pavement with similar characteristics.

References

1. **Takoka T., Ueda T.** Study of the Optimization of the Engine in the Hybrid Vehicle .- Auto Technology, 2004, No 3., 50-53p.
2. **Jost K.** Chrysler's hybrid concept. – Automotive engineering, 1996, No 5, 110-113p.
3. **Hodkinson R.**, **Fenton J.** Lightweight Electric/Hybrid vehicle design.- Oxford: Elselvier Butterworth – Heinemann, 2001. –280p.
4. **Iqbal H.** Electric and hybrid vehicles: design fundamentals. Boca Raton: CRS Press, 2003. 270p.
5. **Kimura A.**, **Abe T., Sasaki S.** Drive Force Control of a Parallel – series Hybrid System, JSAE Review, 1999, No 20. 337-341p
6. Honda Civic , Finland Esan Kirjapaino Oy, 1990. –28p.
7. **A. Stonys, J. Sapragonas, S. Mockus.** The Analogy Method for the Description of External Characteristic of Inner Combustion Engines. – Transport. ISSN 1648-4142. 2004, Vol. XIX, No. 5, p. 214-218.
8. Siemens Power Inverters, Installation Manual, Automobiltechnik, Siemens Corporation, Germany, 2003. 32p.
9. **Starevičius M., Sapragonas J.** Dynamical Model of Hybrid Electric Vehicle Transmission.-Mechanika, 2004, Nr.4(48), p. 33-38.
10. **Sapragonas J. , Starevičius M., Markšaitis D., Ilgakojis P.** Dynamical Model of Hybrid Electric Vehicle Transmission. EAEC 2005 European Automotive Congress, Beograd, Serbia and Montenegro, May 30th – June 1st, 2005. ISBN 86-80941-30-1. Belgrade, 2005, [p. 1-10]
11. **Starevičius M. Ilgakojis P., Stepanov D.** Dynamical Characteristics of the Transmission Elements. Transport Means - 2005: Proceedings of the 9th International Conference, Kaunas, Lithuania, October 20-21, 2005. ISSN 1822-296-X, p. 307-311.
12. **Pacejka, H.B.** Tyre and Vehicle Dynamics.-Oxford. Butterworth-Heinemann, 2002.-627p.