PASSENGER CAR POWERTRAIN: MODEL AND GEAR SHIFT LOGIC

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Summary

The paper describes the modelling and validation of passenger car powertrain. The model is a part of research project focused on automatisation of passenger car transmission, design of a new shift system and solutions for optimal shift strategy.

Keywords

Vehicle simulation model, gearbox model, stateflow shift strategy, E-line

1. Introduction

In last years the field of transmission has become very dynamic. The offer of transmission market increased especially for small and middle class cars, where is possible to chose among classical 5 or 6 speeds manually shifted gear-boxes, 5 to 6 speeds automatic gearboxes, automatised gearboxes, and CVT'S. The manually shifted transmissions slowly decrease its market share, although in Europe they remain to be the major transmission type. Classical manually shifted transmissions have a big advantage of very high efficiency. To adapt the shift mechanism for easy automation, and prepare shift logic conceived for automated gearboxes is very important for next transmission development. Therefore the presented article is a part of work elaborated in frame of research project which concerns automation of passenger car gearboxes.

This contribution is divided into two parts:
a) modeling and validation of simulation model of vehicle and gearbox,
b) shift strategy.

2. Drive-line model

To obtain not too complex model, which will provide necessary accuracy, the modeled vehicle driveline was decomposed to its individual components. As a modeled vehicle we choose Škoda Octavia with 5 speed manually shifted transmission and 1,9 l diesel engine, 66 kW – see Figure 1. The reason of this choice was possibility of data acquisition via VAG-COM system available in the laboratories on the author's workplace, complete engine map measured in the laboratories of author's workplace, and good position of this car as representative of middle class passenger car market segment. In the following text the model of the drive-line with validation approach will be described.

2.1 Vehicle model

The vehicle is modeled with use of Matlab + Simulink and SimMechanics as a two mass model representing torsion system with gear constrain for final drive ratio as depicted on Figure 2. All segments red bold framed represent the vehicle model. Opposite (or in some cases – in the direction) to the direction of the vehicle movement act the resistance forces (depicted in bold blue).

Figure 1: Modeled vehicle – Škoda Octavia Combi

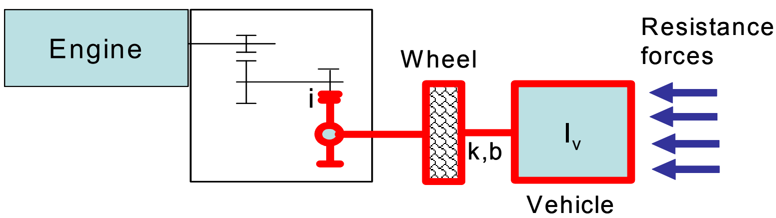

Figure 2: Simplified vehicle torsion two-mass model with torsion stiffness of tires.

The torsional tire stiffness *k* and damping *b* are included in the model. The real-life resistance characteristic of the vehicle was measured with use of VAG-COM system. The coast down method was chosen to estimate the real-life air drag, rolling resistance and linear dependency of the rolling resistance on the vehicle speed. The different extraction methods from coast-down tests are described e.g. in [2]. We chose the polynomial extraction method which includes the linear velocity dependency. The running resistance force *R [N]* was determined as shown in equation (1):

$$R = A \cdot v^2 + B \cdot v + C \quad (1)$$

The coast down test was performed on a horizontal straight road on the Vltava side between Davle and Štěchovice. Measurements were made in both directions, where the legend on the graph in Figure 3 describes the directions as follows: D->S … from Davle to Štěchovice, S->D other way round. For the data acquisition the VAG-COM diagnostic software [1] for VW concern cars was used for connecting the PC to the diagnostic terminal. The software allows registration of the actual data from the vehicle control units. Actual data are arranged into measuring blocks. For vehicle resistance validation we used two groups from ABS control unit. The following parameters were acquired:

- speed of each wheel,
- steer angle,
- lateral acceleration,
- ratio/angle.

Sampling rate reached approximately 3,5 Hz frequency for each group. The first mentioned parameter (wheel speed) is the main parameter for coast-down test. All other parameters served only as control parameters if the deviation due to the lateral movement can be neglected.

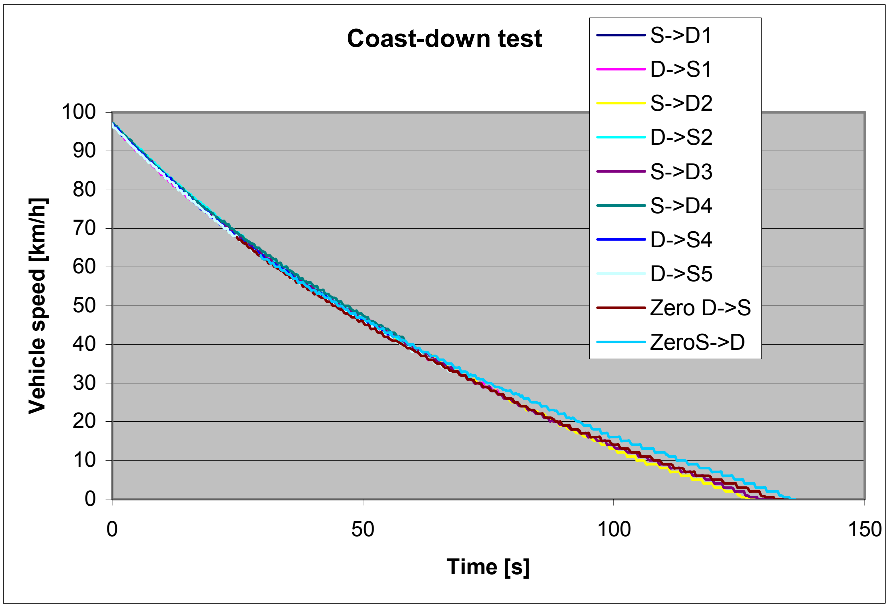

Figure 3: Graph represents all measured values during coast-down tests. Five drives in each direction were acquired. Only the results of three run times where the statistical error remained within the required 2 % of the coast-down test were taken into account.

From the depicted values the equation of running resistance *R* on the straight horizontal road can be determined as:

$$R = 0{,}4428 \cdot v^2 + 1{,}3796 \cdot v + 235{,}7887 \quad (2)$$

The tests were made in calm weather, with temperature of 10,5°C. The air pressure from the data published by hydrometeorological institute was 1014 hPa. The vehicle is equipped with summer tires of size 195/65 R 15, where the dynamic wheel radius obtained from [3] is r_{dyn} = 0.308. From these parameters the following vehicle resistance parameters were determined (*v* corresponds to the vehicle speed in km/h):

$$c_x = 0{,}3392 \quad \text{Air drag coefficient} \quad (3)$$

$$f = 0{,}0157 + 9{,}0266 \cdot 10^{-5} \cdot v \quad \text{Rolling resistance coefficient} \quad (4)$$

The graph on Figure 4 shows the match of simulated and measured vehicle coast-down curves after the fit of resistance coefficients in the vehicle model.

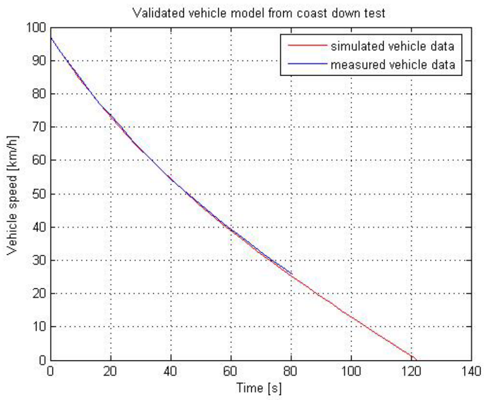

Figure 4: Comparison of simulated and measured vehicle data when driving the coast-down test after a fit of resistance coefficient in the simulation model.

The inertia of the wheel and tyre was measured in the laboratory of author's workplace. The stiffness of drive shaft was determined by calculation.

2.2 Gear-box model

The gearbox is a typical passenger car transversally mounted five speed gear-box. The disposition is two stage, with gears shifted on both shafts. The scheme is depicted on Figure 5a, which shows that the IIIrd, VIth and Vth gears are shifted on the input shaft, and the 1st and 2nd gears on the counter shaft. The simulation model corresponds to the disposition and gear shift placement. The model is created with use of SimMechanics (see Figure 5b) and includes the inertia of gearwheels, synchronizers and gearbox shafts. The bold blue arrow indicates the equivalent of 1st gear free wheel in the SimMechanics model, the green arrow points out the model of input shaft. All inertias were calculated, measured or determined from 3D CAD model. Final drive and differential take part in the vehicle block – described in section 2.1.

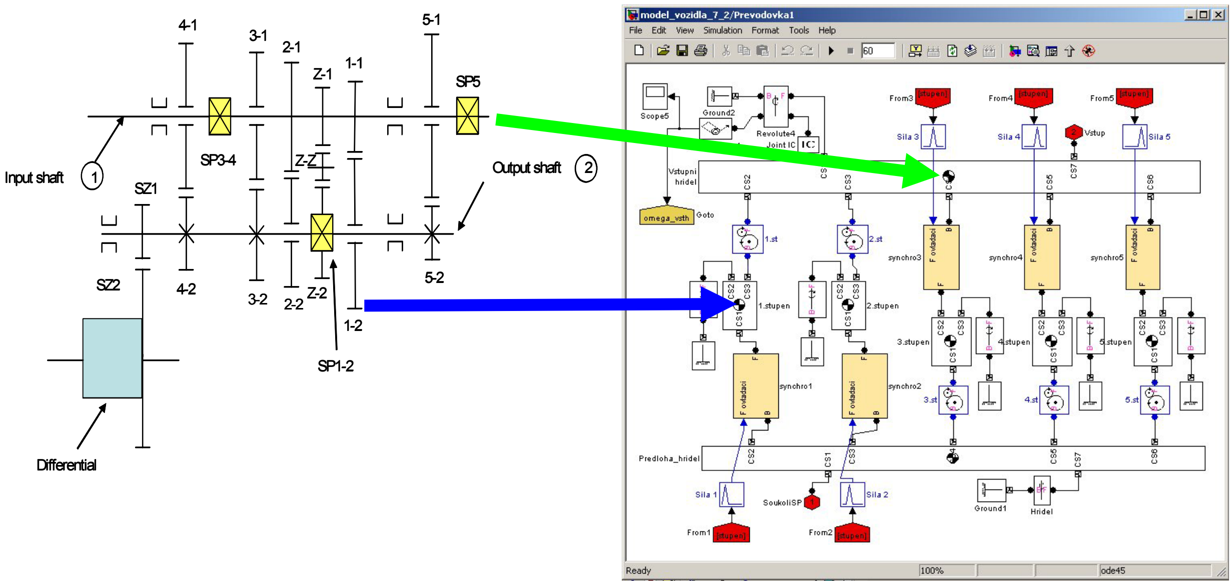

Figure 5a: The scheme of five speed Škoda Octavia gear-box type MQ200

Figure 5b: Model of Škoda Octavia MQ200 gear-box prepared with use of SimMechanics

2.2.1 Gear-shift modeling

The gear-box is equipped with cone synchronizers. The SimMechanics Stiction joint model was used with control by two external forces. The first force corresponds to the frictional force acting on the radius, with estimated coefficient of friction. At the same time the indexing moment counteracts on the blocking chamfers, wherefrom the counteracting force is calculated. More detailed description of synchronizer was described in [4].

2.3 Drive-away Clutch model

The drive-away clutch is likewise the synchronizer clutch modeled with help of SimMechanics Stiction Joint Model. The control strategy is not yet completely finished, but will be determined from the measurements of clutch pedal during different riding and drive-away situation when controlled by different drivers. The average control strategy will be taken into account. From the lift will be determined the clamp force function, which will serve as input in the prepared clutch model. The vehicle behavior during drive-away or gear-shift will be validated by measurements of vehicle longitudinal acceleration during the mentioned maneuvers.

2.4 Engine model

The engine in the model is included only as Look-up tables covering the complete engine map measured in the internal combustion engine laboratories of author's workplace. The inertia of the engine is taken into account. An important engine parameter which is not possible to measure on the test-stands is engine braking force with different shifted gears. The engine braking forces will be determined with help of VAG-COM acquisition system.

2.5 Complete vehicle model

Figure 6 depicts the complete model of vehicle drive-line. The two blocks with blue background present the SimMechanics models of gear-box and vehicle. The yellow background color block is Joint Stiction model for dry friction modelling of drive-away clutch. In the bottom part of the model are two blocks representing the shift-strategy of gear change. The gas pedal position is in the picture defined with repeating sequence block, but mostly are used data measured via VAG-COM software, where the gas pedal position is included in the engine control unit. Data acquisition during drive with VAG-COM system, with some examples of data acquired from engine ECU are described and shown in [5].

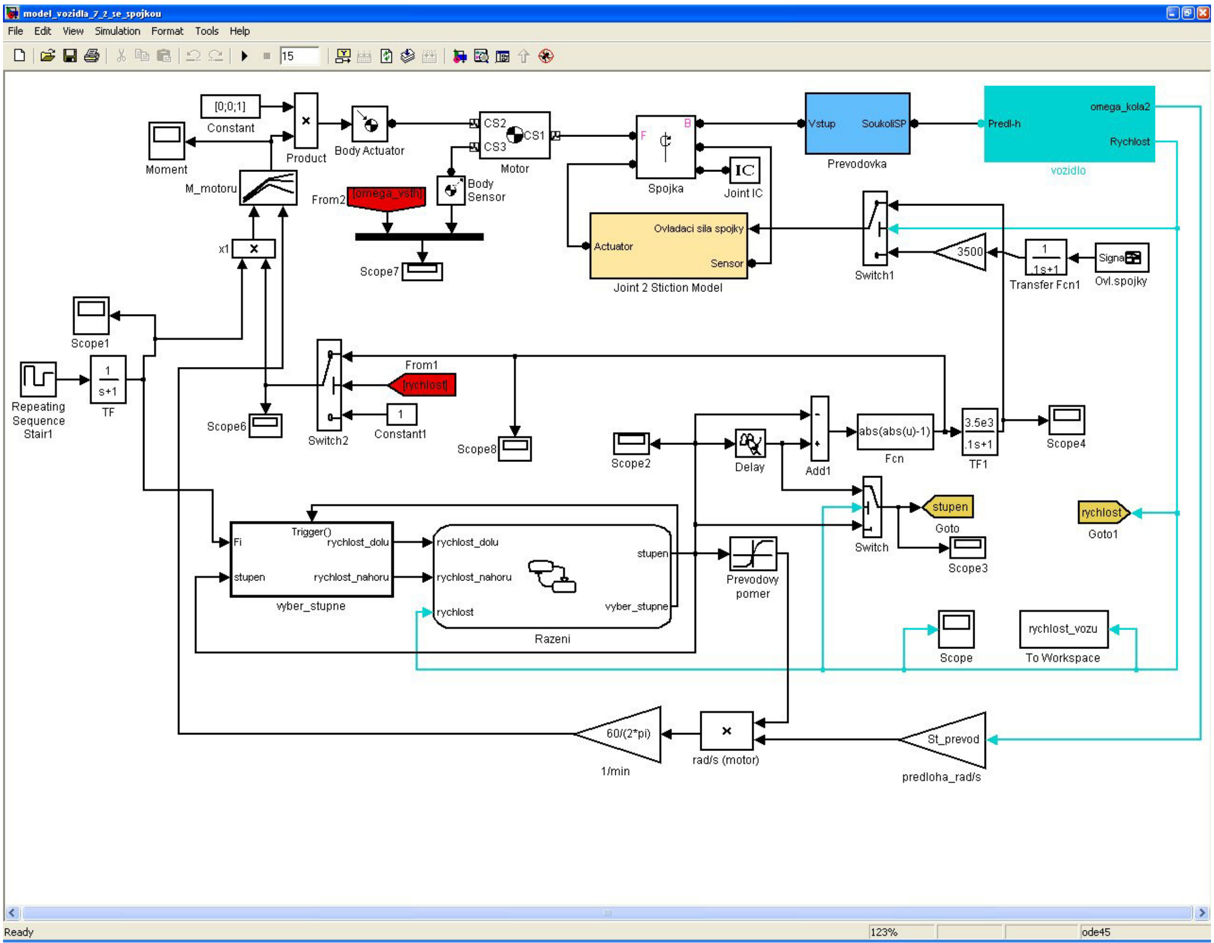

Figure 6: Complete model of Škoda Octavia drive-line with gas pedal position, clutch actuation and gear-shift strategy

3. Gear-shift strategy

The gear-shift strategy is prepared as Stateflow control. Two parameters are decisive for the gear-speed choice:

- vehicle speed,
- gas pedal position.

The relation between vehicle speed and gas pedal position form the gear-shift law. The gear-shift lines depicted on Figure 7 were determined from the engine map. The complete engine map (dependency engine rotations versus engine torque) was completed with isocurves of constant specific fuel consumption. Further we amended the map with hyperbolas of constant power. From the intersection of power hyperbolas with the lowest specific fuel consumption the E-line (most economic curve) was determined. The primary rule for shift limits determination was to maintain the engine in the most economical point, i.e. to maximally follow the E-line.

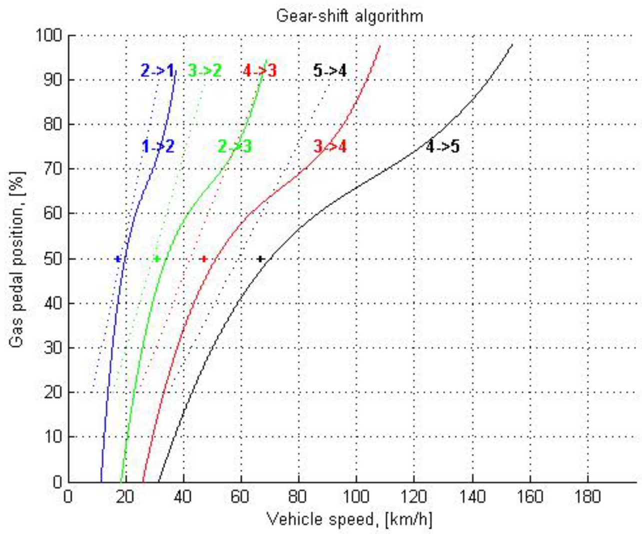

Figure 7: Gear-shift algorithm for five speed transmission used in Stateflow logic, determined based on engine E-line.

Furthermore the fuzzy logic control strategy is in preparation, where more parameters such as longitudinal acceleration, speed of gas pedal actuation, and lateral acceleration are included in the decision strategy. The fuzzy logic toolbox is used. In the future work the fuzzy logic controller will be optimised with use of neural network.

6. Conclusion and software use

The vehicle parameters, determination of E-line as well as extraction of vehicle resistance parameters were programmed in Matlab m-files. The vehicle model is built in Simulink and SimMechanics. For the “classical” control strategy resulting from shift limits based on vehicle speed and gas pedal position is used Stateflow. In the future this shift algorithm will be compared with the fuzzy and neural network strategies.

References

[1] Ross: VAG-COM Diagnostic Software, http://www.ross-tech.com

[2] Takagi H., *Real-Life Coefficient of Drag – a Simple Extraction Method*, In: AutoTechnology Vol.4, August 2005, pp. 52-56

[3] Lechner G., Naunheimer H., *Automotive Transmissions*, Springer Verlag, 1999, ISBN 3-540-65903-X

[4] Achtenová G., *Simulation model of Automated Gear-box*, In: Proceedings of AED Conference, 2004, Glasgow.

[5] Achtenová G., Vávra J., *New Control Algorithm for Automatic and Automated Gear-boxes,* In: Proceedings of AVEC conference, 2004, Arnhem, The Netherlands.