THE PURPOSEFULNESS OF APPLICATION AND CONTROL ISSUES OF FRICTION BRAKE WITHIN TEST BENCH SIMULATION REPRODUCING THE DRIVETRAIN DYNAMICS

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Summary

The paper briefly presents the implementation of a concept of a test bench simulation reproducing the drivetrain dynamics of passenger car equipped with automatic transmission. It was performed with a test bench which used the flywheel and two brakes, i.e. friction disk brake and eddy current brake, in order to simulate drag forces reduced to drive shaft. The purposefulness of friction brake application was described in more detail and some principles of parameter selection as well as control actuation system were also presented.

Some problems characteristic of friction brake torque automatic control were discussed. Problems related to long actuation delay time and variations of brake torque with variations of brake speed and temperature were discussed as well as methods to solve them with the use of suitable algorithm for open loop torque control. The algorithm was based on forecasting of the required flywheel deceleration and on predefined levels of actuation for phases of vehicle stopping. The final part of the paper presented an exemplary result of the control within the built test bench for a recorded time history of road test drive.

Keywords: experimental vehicle dynamics testing, automatic transmission, test bench simulation, friction brake, control

1. Introduction

In several previous papers co-written with J. Walkowiak [4,5,6] an idea and practical solution of test bench simulation reproducing the dynamics of drivetrain with automatic transmission in laboratory conditions were presented.

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This idea, illustrated in Fig. 1, assumes the possibility to reproduce in laboratory the working conditions of drivetrain with automatic transmission (in relation to the engine and transmission) typical of regular operating conditions. It is particularly important that it enables to reproduce the drivetrain working conditions in the typical urban driving, characterized by frequent moving off and stopping processes and gear shifting. This latter aspect is particularly important when designing a control system for gear shifting, which must take into account such issues as [3]: vehicle performance (top speed, gradeability, acceleration performance), fuel consumption, exhaust gas emission or driving comfort related to the quality of gear shifting process.

These issues have been addressed in numerous scientific papers, including those concerning testing the whole drivetrain system or components only with a test bench simulation [9,11].

J. Walkowiak and the author suggested and implemented one of many possible concepts of test bench to simulate the work of drivetrain with automatic transmission. It assumed the use of relatively inexpensive and easily available components. The implementation was described in detail in previous papers by the same authors [4,5,8].

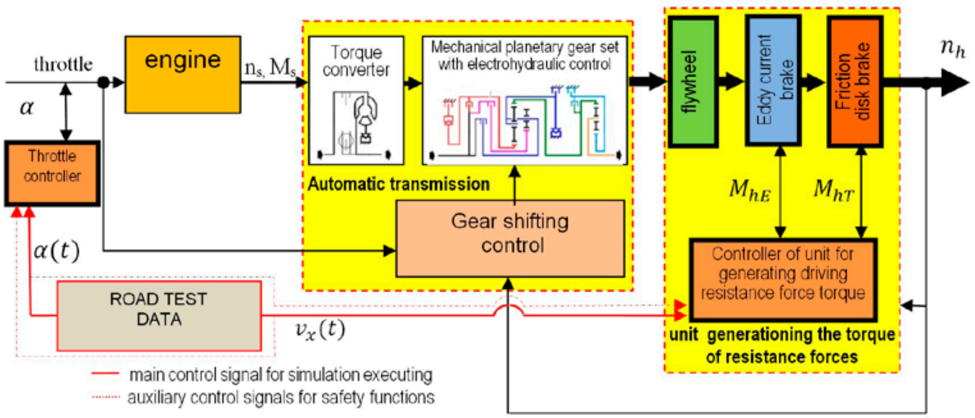

Fig. 1. Test bench control scheme for simulation reproducing drivetrain dynamics [4]

Idea of test bench simulation (Fig.1) assumes conducting simulation on the basis of data acquired during real operation of vehicle. These data concern two variables – the level of throttle opening and driveline shaft rotational velocity, which depends linearly in most cases from longitudinal vehicle velocity.

The control system of test bench with follow-up method controls the throttle opening angle (based on the acquired data) and simultaneously controls the torque level of driving resistance so as to obtain equal values of actual velocity changes and velocity changes recorded during road tests of driveline shaft rotational velocity changes , which in fact is a follow-up control of driveline shaft rotational velocity.

The torque of resistance forces is generated with three components of the unit generating driving resistance force torque (Fig.2.):

- the flywheel which physically simulates a car inertia; its value was calculated by reducing car mass to the axis of drive shaft assuming that kinetic energy of the moving car and the flywheel at test bench are equal,
- Eddy current brake smoothly actuated the pulse width modulation of control signal,
- friction disk brake enabling to stop the flywheel and to generate reaction torque to engine residual drive torque at idle speed amplified by torque converter.

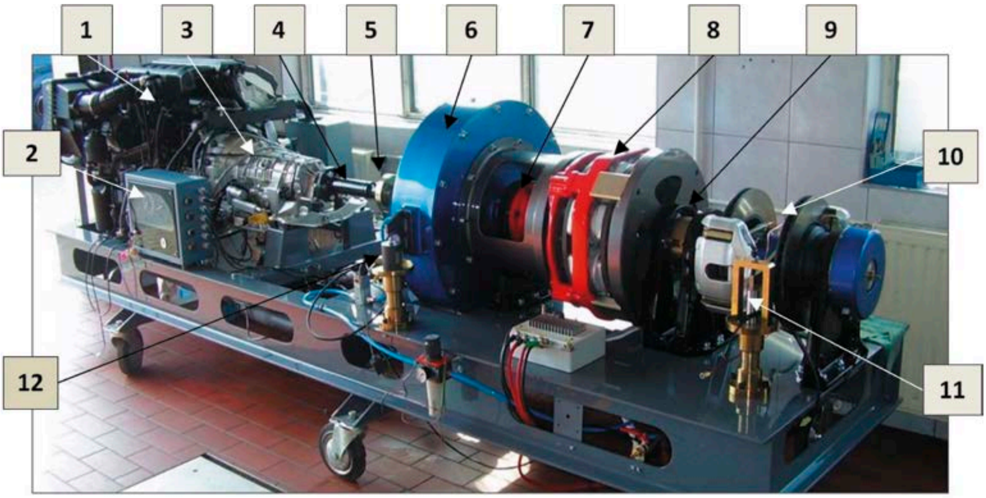

Fig. 2. A view of drive-train test bench: 1 – engine, 2 – control unit (instrument panel), 3 – automatic transmission, 4 – driveline shaft, 5 – driveline shaft torque sensor, 6 – flywheel with cover, 7 – reduction planetary gearset, 8 – Eddy current brake, 9 – safety clutch, 10 – unit of friction brakes, 11 – friction brake torque sensor, 12 – Eddy current brake torque sensor

The paper highlights the application of friction disk brake in generating a portion of torque of vehicle resistance forces, which is a problematic solution in some control aspects. Therefore the purposefulness of application of such brake in the test bench was discussed, and problems resulting from its application were indicated. The last section of the paper presents the implemented solutions of discussed problems.

2. The purposefulness of friction brake application

During the research preceding the creation of test bench, numerous experimental tests were conducted with a view to qualitative and quantitative analysis of drivetrain loads with torque of vehicle resistance forces [7].

The analyses involved an assessment of torque values occurrence frequency within intervals from -200 to 1000 Nm within intervals of operational rotational velocities f driveshaft (Fig 3.).

A limited rotational speed of driveshaft is characteristic for vehicle operation in urban driving conditions. The speed does not exceed 2000 rpm and in most cases it is below 1000 rpm. A frequent torque load of driveshaft about 150 Nm at vehicle standstill is also typical.

The reason for such a torque load is the torque converter characteristics. The torque converter amplifies the output torque in proportion to torque ratio, which is indirectly dependent on speed ratio:

$$ik = \frac{n_t}{n_p} = \frac{\omega_t}{\omega_p} \tag{1}$$

where:
ω_p – angular velocity of converter pump equal to engine crankshaft angular velocity [rad/s],
ω_t – angular velocity of converter turbine equal to input gear of planetary gearset [rad/s].

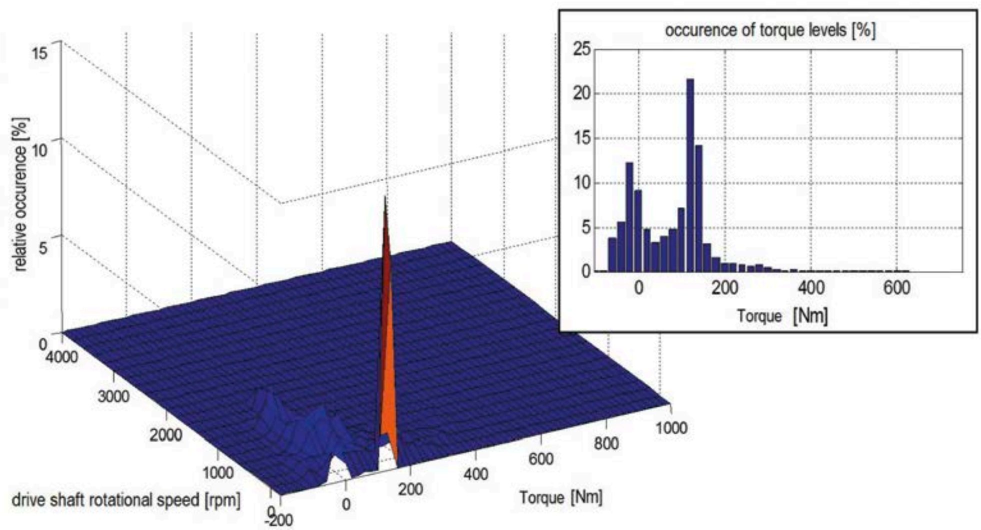

Fig. 3. The spectrum of drivetrain loads and frequency of occurrence of respective torque levels [7]

During a forced vehicle standstill the angular velocity of torque converter turbine slows down to zero, due to the fact that in D range (forward gear selected) in most transmission types there is a state of connection between the turbine and transmission output shaft via planetary gearset and clutches. This results in a drop of speed ratio and thereby a buildup of torque ratio i_d, which reaches its greatest value for speed ratio i_k = 0.

An additional analysis of Eddy current brake characteristics has shown that it can ensure braking torque with a maximum level of about 1500 Nm, which would enable to reach maximum angular deceleration for the assumed flywheel moment of inertia at a level corresponding to vehicle braking with deceleration about 0.3 g.

Therefore in order to reproduce a car braking process with deceleration higher than that it was necessary to apply also a brake with a higher braking torque than the one which could be obtained with Eddy current brake.

To sum up the relations observed in tests, the following functional requirements were formulated for friction disk brake, which could not be obtained with only Eddy current brake in test bench:

- a possibility to maintain flywheel in standstill in spite of engine idling and transmission working with D range engaged,
- a possibility to brake the flywheel with deceleration corresponding to vehicle braking with deceleration 1 g.

3. Friction brake selection

Minimal requirement for friction brake in the test stand is to generate 170 Nm braking torque – the torque bigger than engine residual drive torque at idle speed amplified by torque converter. Another requirement, i.e. generating a braking torque able to slow the flywheel with angular deceleration $\varepsilon = 28{,}1 \left[\frac{\mathrm{rad}}{\mathrm{s}^2}\right]$, corresponding to vehicle linear deceleration equal to 1 g, results in the required torque value of about 4,500 Nm.

After calculations the system of two disc brakes working together was applied in the test bench The brakes were from a passenger car which could reach the maximum speed of 250 km/h, the equivalent of 69,4 m/s. The speed became the basis for the assumed safe rotational velocity limit of brake discs. Assuming dynamic wheel radius r_d=323 mm, maximum disks rotational velocity was estimated at 2,052 rpm. In order not to exceed this value the system of disc brakes is placed behinds the reduction gear, which prevents exceeding this value to rotational velocity value of flywheel equal to about 5580 rpm.

4. Actuation system of friction brake

As an actuation system of friction brake a simplified vehicle brake system was used with a generation of brake pedal force along with a pneumatic actuator, supplied with air at a pressure controlled by proportional valve controlled by electric signal (Fig. 4).

The complex actuation system, albeit simple in design, is difficult to control. This results from the influence of all serially connected elements characteristic of this system on the actuation system response to control signal and includes the characteristics of proportional valve, pneumatic actuator, vacuum booster valve and master cylinder and wheel brakes characteristics.

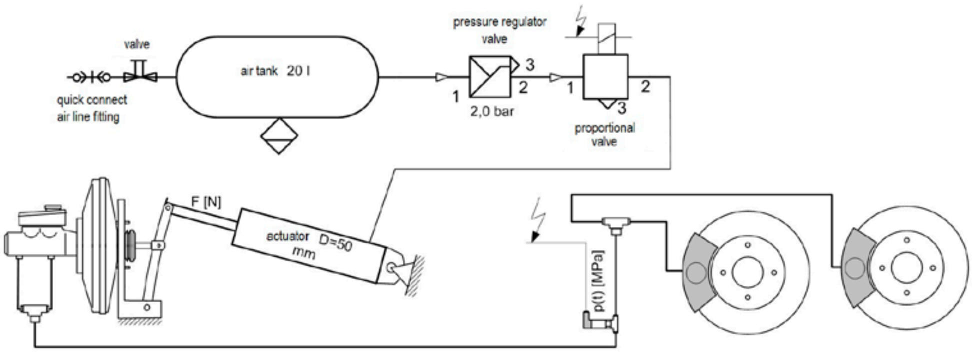

Fig. 4. Pneumatic and hydraulic actuation system of disk brakes

When controlling the torque in test bench the crucial element is a step response of actuation system, which indicates braking force dead time of about 0.3 s in relation to control signal (Fig. 6). This absolutely excludes a closed loop control. Therefore it was decided to use open loop control, the implementation of which is described in the next section.

5. Disc brake torque variations

Another factor which renders disc brake control difficult even which open loop control are variations of friction coefficient between brake pads and brake disk, which generates variation of brake torque at a constant level of pressure of hydraulic fluid resulting from constant value of control signal.

This issue is related to the variations of friction coefficient related to changes in friction speeds of brake pads and disk and also to temperature changes of both elements. It is discussed in numerous scientific papers and constitutes a major control problem.

An example can be paper [1], which presents the characteristics of variations of friction coefficient versus fiction speed, according to which a change in friction coefficient between friction speed 60 m/s and 5 m/s was linearly rising from value about 0,4 to 0, 58.

On the other hand paper [2] presents the variation of friction coefficient between brake pads and disk versus temperature of friction surfaces. When temperature changed from about 25 ^{0}C to about 350 ^{0}C, the friction coefficient fell linearly from 0.48 to 0.37, as is is illustrated in Fig. 5.

The relations between friction coefficient and temperature and friction speed depend on specific properties of friction pair elements and the above-presented examples are of only qualitative in relation to the tested system of disk brakes.

During the test of step excitation applied to the input of friction brake control system

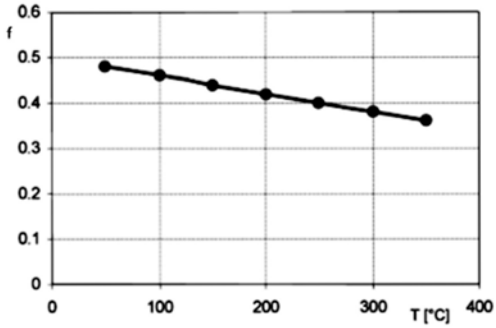

Fig. 5. Exemplary variations of friction coefficient versus temperature [2]

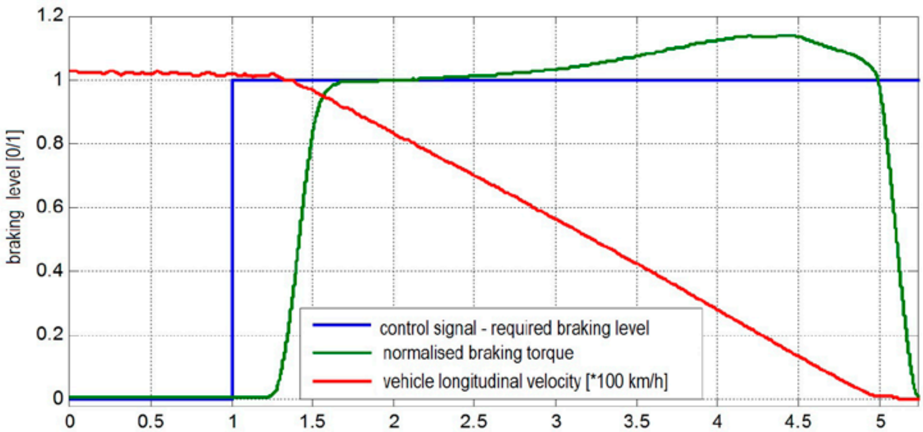

Fig. 6. Change of the braking torque at the step response test

a response obtained was a variation of braking torque as presented in Fig. 6. It shows a long response time delay to control and about 15% increase of braking torque along with friction speed decrease and increase of brake discs temperature.

6. Control of friction brake

To control the friction brake a control strategy was developed, whose algorithm is shown as a diagram of a disc brake control subsystem – Fig. 7.

The subsystem includes modules generating the signal of vehicle speed acquired during road tests. Their task is to feed the values of road test speed, varying with simulation time recorded for subsequent time steps, to control algorithm.

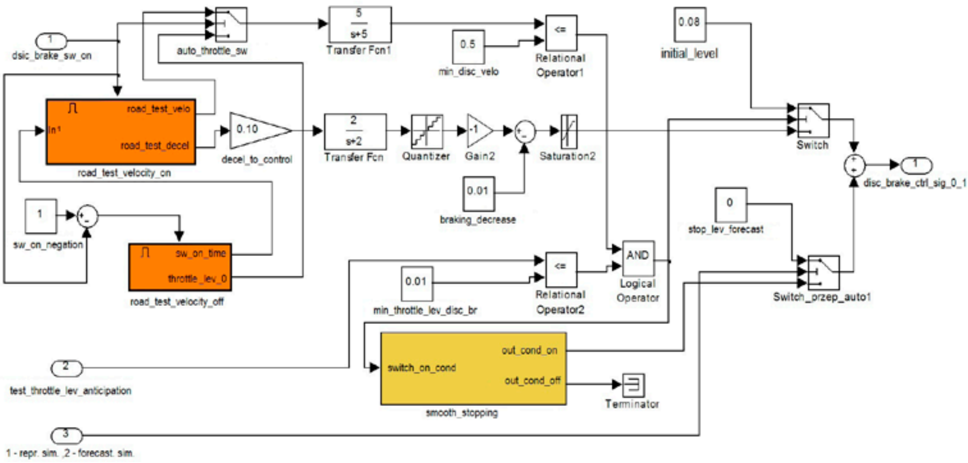

Fig. 7. Disc brake control subsystem

However, besides generating vehicle speed signal, a time signal of deceleration is generated with some anticipation and calculated as a ratio of speed change and specified calculation time.

Blocks "road_test_velocity_on" and " road_test_velocity_off" are responsible for this task. The time value at the moment of triggering " road_test_velocity_on" block is subtracted from current simulation time. The moment of triggering "road_test_velocity_on" is equal to moment of deactivating " road_test_velocity_off" block. There are two additional blocks used within "road_test_velocity_on" block, allowing to read out from vector of vehicle test speed values of speed with some anticipation time in relation to current simulation time.

The first anticipation time is used to neutralize dead time response of disk brake actuation system. The second one, the internal anticipating block, is used to calculate speed change in time period used to third reading with time advance of vehicle test speed. It allows to calculate the vehicle deceleration value, averaged over period of time advance. This deceleration is afterwards the base to calculate the level of control signal used to control disc brake - signal „decel_to_control".

The calculated deceleration value is converted to control signal level based on the characteristics of limits of maximum braking torque to the level allowing to brake the flywheel with deceleration corresponding to the value of 1 g (9,8 m/s^2).

Then the calculated deceleration values can be simply converted into a level of control signal. Due to considerable time delays of actuation system of disk brakes, the calculated signal is quantized with 20 levels to decrease slight fluctuations of brake torque, superfluous for control process.

This tenet was combined with the principle of using disc brake with braking torque lower

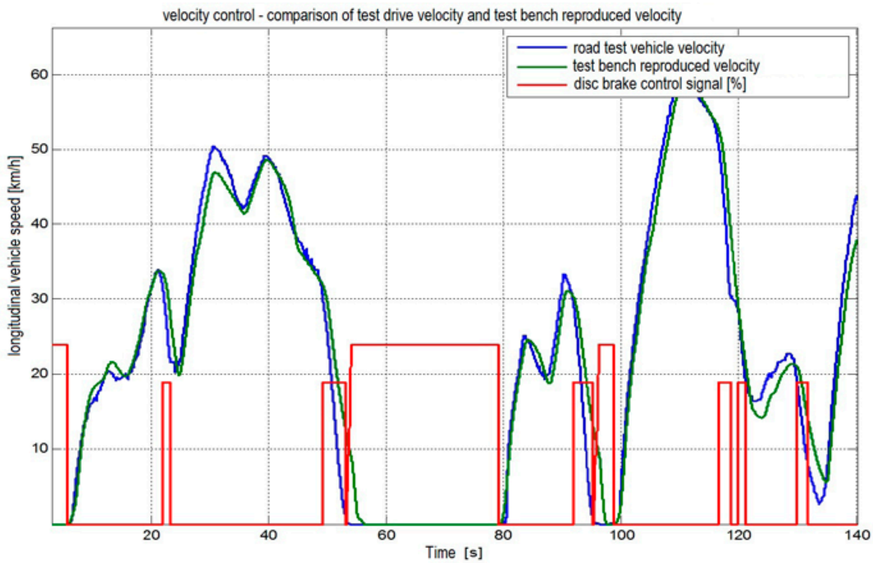

Fig. 8. Result of vehicle test speed profile reproducing and generated control signal levels

than required level and to complete braking torque with torque generated by Eddy current brake. A constant value subtracted from calculated level of braking torque was placed inside the block and called "braking_decrease".

Another disc brake working mode is "smooth stopping", understood as gradual increase of braking torque until stopping the flywheel rotation, sustained by engine residual drive torque at idle speed, amplified by torque converter.

This function was implemented by defining conditions of "smooth stopping" function activation. These conditions are throttle closing with simultaneous determination of vehicle speed lower than the specified triggering level. When such conditions occur, the level of braking torque control signal of disk brake subsystem results from value named as a "initial level" (this value neutralize brake pedal freeplay) and from value generated by "smooth stopping" block, which after activating cause brake torque control signal rising over time. Utilization of two signals sum - constant and rising over time allowed to smooth torque rise, and thereby smooth stopping of flywheel with less loads of mechanical system of test bench.

The result of vehicle test speed profile reproducing by torque control is presented in Fig. 8. The phases of use friction brake to smooth stopping function (approx. 22 second and 50, 90 or 120) and in the vehicle standstill phases (from 50 to 80 seconds) are shown.

6. Recapitulation

The presented and implemented idea of friction brake control was related to necessity for providing ability to stop the flywheel rotation and additionally for complementing the braking torque value during intensive braking.

The conducted introductory tests and later also validation tests showed that automatic control of friction disc brake is difficult and needs to take advantage of methods using to control estimation of required torque and controlling in open loop from torque value point of view.

However, in addition of the use of Eddy current brake as a main controlled brake and friction brake as complementary brake, it is also possible to effectively use a friction brake in automatic control.

An issue similar to the one presented in this paper occurs in braking torque control system of vehicles using recuperative braking in connection with braking using friction brake [10].

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