# SIMULATING RESEARCHES OF POWER TRANSMISSION SYSTEM WITH TORQUE CONVERTER IN SELECTED DRIVE CYCLES

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## Summary

In the paper it has been presented model of power train system for vehicle with diesel engine and automatic gearbox with hydraulic converter. Simulation model has been realised in MatLab Simulink R2009a programme. There has been presented its main subsystems and control system conception that have deciding influence on the driving system operation. In the proposed simulation model position of accelerating pedal is interpreted as the information according driver will of the assumed vehicle speed and way of its realisation (dynamic, economic drive), that is selection of the drive strategy. Existing ecological requirements it is the EURO 3,4,5 standards requires that proper strategies were realised ecologically. The decision of selecting the gear ratio and engine load is undertaken in the master controller, beyond the driver, finally enabling the economic and ecological or dynamic operation of the total powertrain. In the final part of the paper there have been analysed results of simulation tests.

**Keywords:** torque converter, power transmission system, engine control unit (ECU), drive cycle UITP

## 1. Introduction

The powertrain of the vehicle with automatic transmission can be presented as an object of automatic control as on Fig.1.

From many publications [2÷7], concerning hydrodynamic torque converter comes, that this transmission is self accommodating to the powertrain load changes and don't need the control, what comes also from the above picture. Engine and gearbox have their own autonomic controllers, while whole system operation is controlled by driver and master controller, which purpose is to elaborate such needs in relation to engine and gearbox, in order to their cooperation runs optimally due to assumed criterion. In the master module it is "processed" the driver will to the so called control strategies, that assign the optimal parameters of engine and gearbox ratio to the acceleration pedal position. Since over the twenty years, with development the microprocessor technique, the number of tasks for that module still arises.[8]

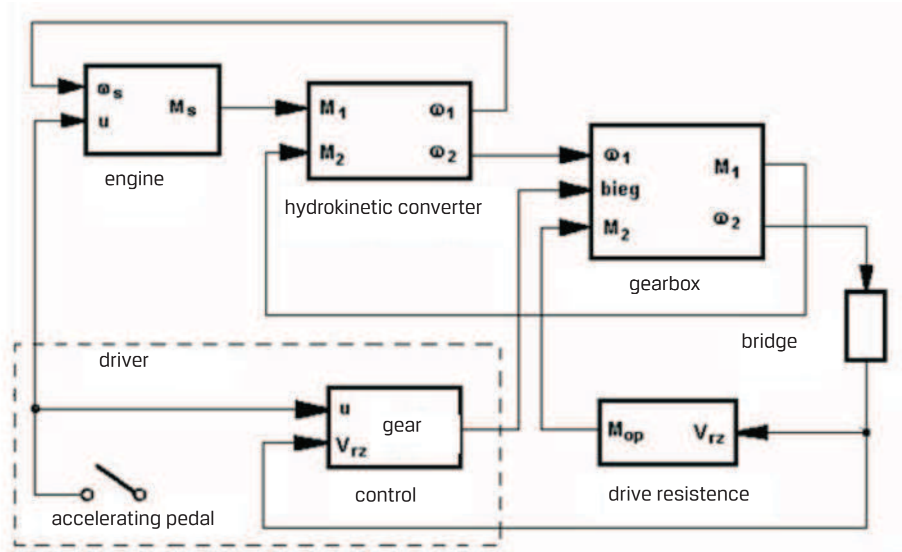


**Fig.1. Power transmission system with torque converter [1].**

## 2. Engine controller

Engine model is based on tests carried out for RABA MAN D2156 engine in steady state on the Schenck dynamometer in the Department of Vehicles and Fundamentals of Machine Design [11]. On this base have been defined three-dimensional characteristics of engine torque, specific fuel consumption, exhaust gas components of fumes: $NO_x$, $C_nH_m$, HO and solid particles PM in function of engine angular (rotational) speed, and throttle opening angle, according to the acceleration pedal position.

For example on the fig. 2 has been presented engine torque characteristic, while on fig 3 contents of CnHm in exhaust gas. There are seen specific characteristics by partially opened throttle and turbo load. In the angle of throttle opening range 0÷30%, fig. 2 it exists area of low, almost constant engine torque values. This feature of engine is by many firm considered as out of control area on the gear shift diagrams [1,9,10]

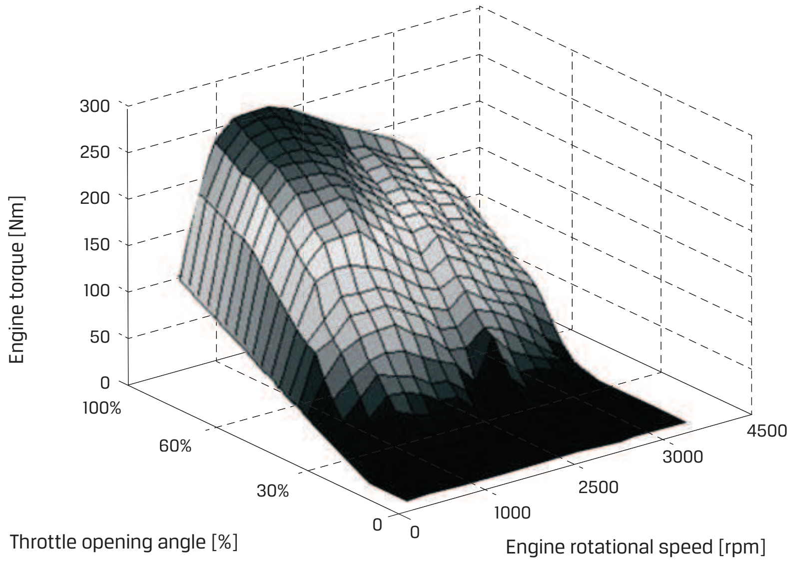


**Fig.2. Engine characteristic [10].**

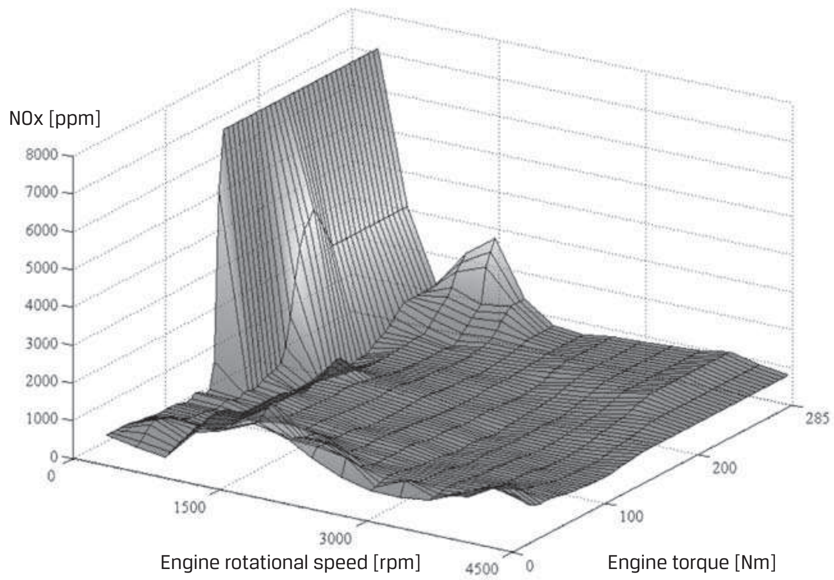


**Fig.3. Emission of nitrogen oxides NOx [1].**

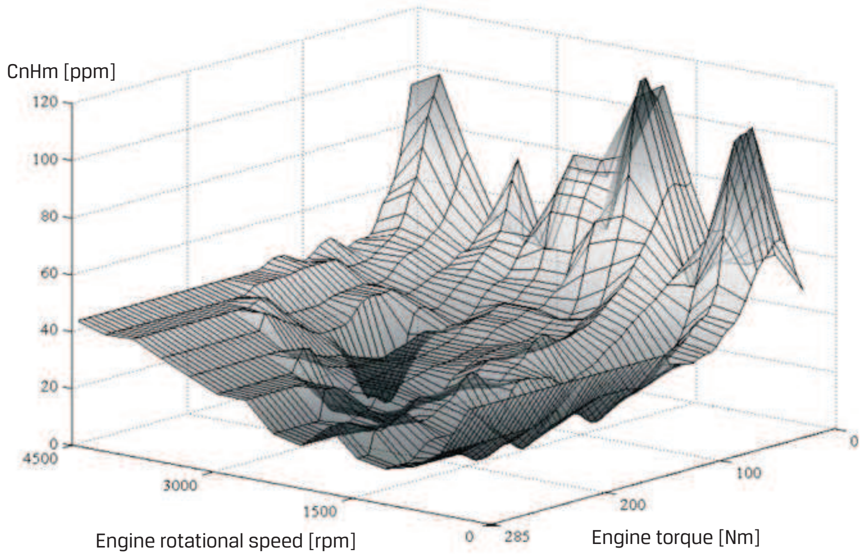


**Fig.4. Emission of hydrocarbons $C_nH_m$ [10].**

For each of mentioned characteristics, it is possible to designate so called optimal lines. It is easy to see that each of this "optimal" lines is another function of torque and rotational speed. Thus defining the one engine control line, that fulfils the partial criteria is impossible. They can be weighted average of some optimal lines, while the weights for individual lines can either be changeable. For the weights distribution in individual criterions, as toxicity, economy, have additionally influence another parameters such as:

- cooling system temperature (by the lower engine temperature change proportions between the exhaust gas toxic components).
- catalyst temperature.

Engine control line can be defined in different way dependent from defined criterion [8,9,10,12]. One of the engine control method is for instance Pareto method, which enables define for example minimum the following optimised function

$$K = w_{ge}K_{ge} + w_{CO}K_{CO} + w_{NOx}K_{NOx} + w_{CnHm}K_{CnHm} + w_{PM}K_{PM} + \dots \quad (1)$$

Where: $W_{xy}$ – weight of "xy" parameter; connecting the weight value with actually "serviced" power it is possible to take into account different "boundary" conditions.

$$w_{ge} + w_{CO} + w_{NOx} + w_{CnHm} + w_{PM} + \dots = 1. \quad (2)$$

$K_{xy}$ – Relative value of parameter „xy" in relation to minimal value for example from the exhaust gas toxicity standard, calculated from formula beneath e.g. for specific fuel consumption "ge":

$$K_{ge} = \frac{ge_{pcel} - \frac{\sum_{i=1}^{N} ge_i}{N}}{ge_{pcel}} , \tag{3}$$

Where: N – Number of calculation points for given curve, $ge_{pcel}$ – aimed value of specific fuel consumption (minimum), $ge_i$ – value of specific fuel consumption for engine operation point i

Values of relation xypcel parameters taken to analysis create coordinates of target point of optimized function. The use of relative values enables "to standardise" components and their equivalence is achieved when their relative values are of the same order. The presented above proposition enables to enlarge the number of considered parameters and treat them in the equivalent way in respect to each other. It is obvious creation another optimisation function.

It should be emphasized, that the engine control function shape should take into account fulfilment actually effective exhaust gas standards and generate for this conditions optimal fuel consumption line. Such engine control philosophy enables fulfilment of stricter exhaust gas standards by means of programming, without the needs of design changes.

## 3. Progressive gearbox controller

In automatic transmission decision about shift comes not directly from driver, but is elaborated by control system and shifts logic determines which gear is switched and when it will be shifted. In the same time the transmission should behave so as it is expected by driver due to actual traffic situation.

The shift decision base is created by so called basic shifts program, which consists of shift lines by acceleration and by reduction, for all forward gears. For instance, fig. 5 presents the basic shifts program for the vehicle and selected control strategy. From the input values: throttle opening angle and vehicle speed, results to applied in the time shifts characteristic and gear selection.

Modelled gearbox has been patterned to the gearbox assembled in the town bus B12BLE VOLVO [20] The gear ratio of planetary gearbox were:

$$i_1=3{,}43,\ i_2=2{,}01,\ i_3=1{,}42,\ i_4=1{,}00,\ i_5=0{,}83,$$

and final drive ratio

$$i_g=5{,}63.$$

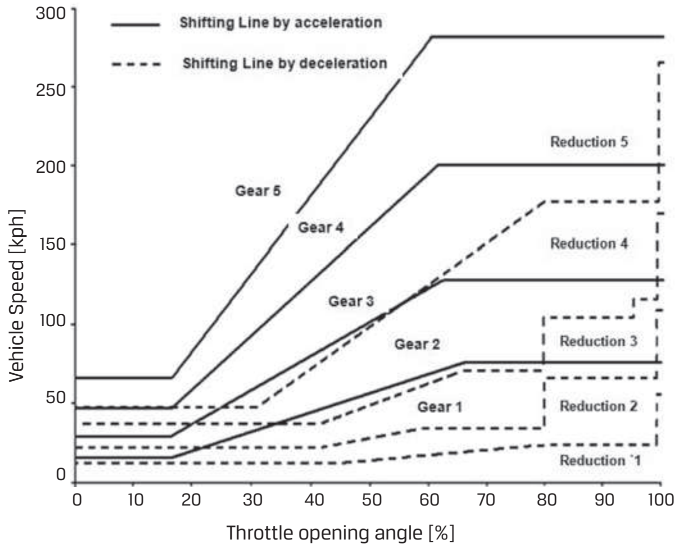


**Fig.5. Basic characteristic of shifts [8].**

## 4. Power transmission simulation model at Matlab/Simulink environment

The proposed model of the propulsion system based on the above assumptions for the city bus illustrated Figure 6

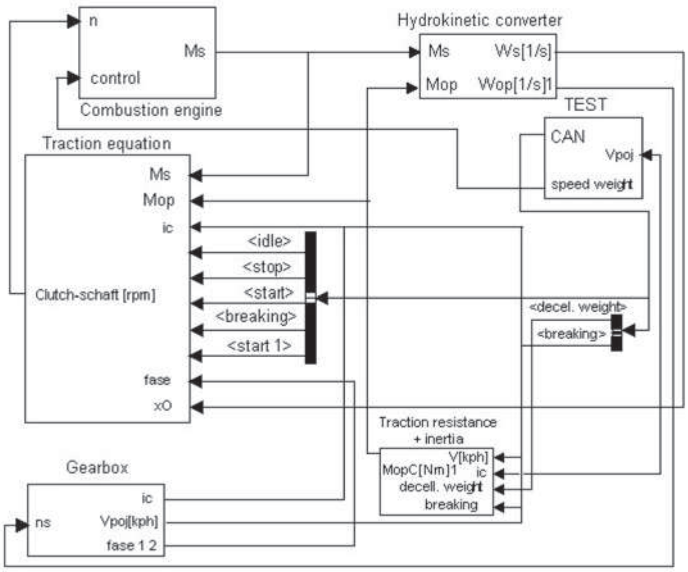


**Fig.6. Power transmission system at Matlab/Simulink environment [8].**

Nonlinear model of the hydrodynamic torque converter (fig.7) taking into account coefficient of friction lose and variable knock lose coefficient has been presented in the works [1,4,13,14] Model has been verified on the base of the tests carried out on the test stand in the Department of Vehicles an Fundamentals of Machine Design.

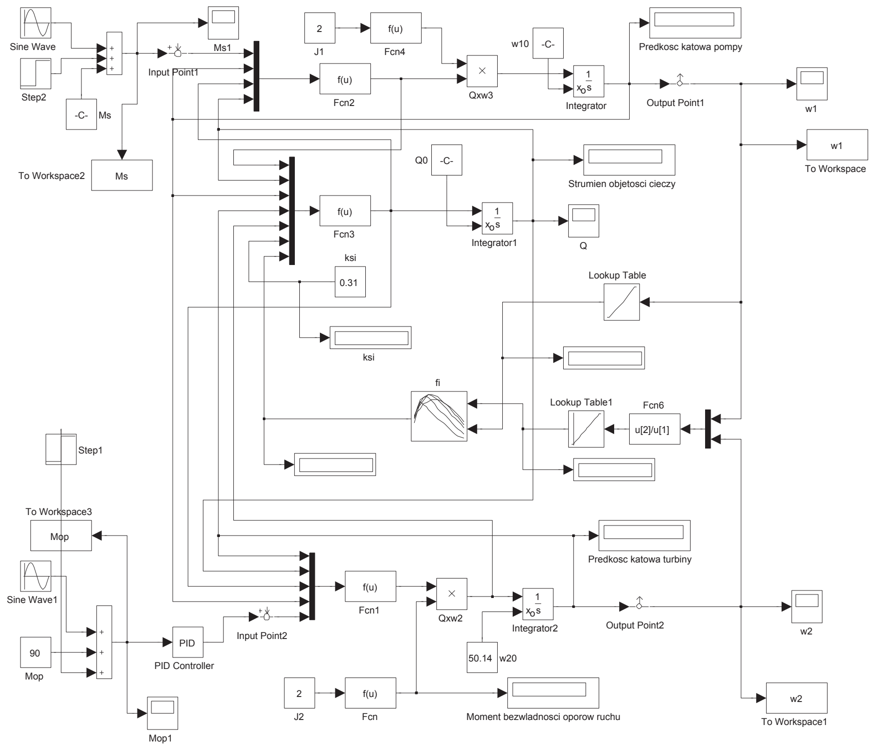


**Fig. 7. Torque converter ZM130 model at Simulink environment [3,5].**

## 5. The concept of the power transmission control system in the simulation model

For the proposed control system the input and output values, Fig.6, can be characterized as:

a) $v_{zad}$ – assumed vehicle speed, results from assumption of driving cycle, it can be chosen the exhaust gas toxicity test or it's part.

b) $T_{op}$ – resistance of motion, constitute the driving system load, resulting mainlyfrom air resistance, elevation and inertial forces.

c) $V_{poj}$ – vehicle speed.

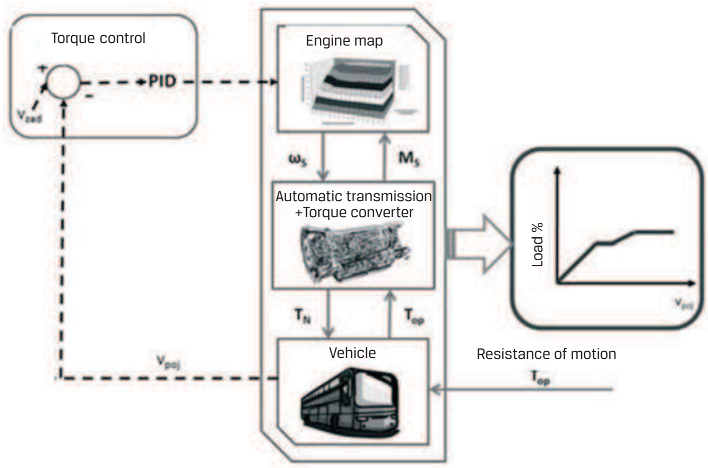


**Fig.8. Powertrain control schema used for modeling with MatlabR2008a/Simulink.**

In the control system shown in fig. 8, with feedback has been used PID controller. The controller parameters have been selected by successive approximation method, in order to the control system was universal for given different speed passes. The controller parameters should assure that the passage of vehicle will assure following the speed with fault not exceeding 5%.

Increase the number of gears, e.g. from 5 to 10 as in fig.9, enabled realize the gearbox control according to different parameters, for instance economic or dynamic drive. By the economic drive the combustion engine operation should be narrowed to the surroundings of angular speed by maximal torque (dotted line), which usually responds to the engine operation with maximal efficiency. Dynamic drive responds engine operation in the surrounding of maximal power (upper part in fig. 9 – solid line).[1]

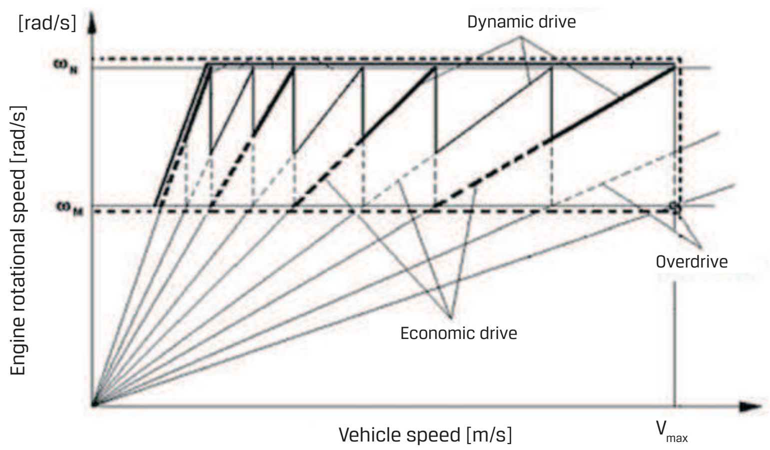


**Fig. 9. Sawing graph for step transmission during economic and dynamic drive [1].**

In the simulation model it was taken into account the control philosophy shown in fig. 9. It have been increased two times the number of gears, which have been calculated as double geometric series with exponents two times lesser than engine flexibility factor. It have been obtained following values

$i_1$=3,43, $i_2$=2,23, $i_3$=2,01, $i_4$=1,54, $i_5$=1,42, $i_6$=1,065, $i_7$=1, $i_8$=0,83, $i_9$=0,778, $i_{10}$=0,59.

Final drive ratio remained without change.

## 6. Results of simulation tests

In the fig 10 it has been presented comparison the vehicle acceleration for dynamic and economical drive.

In both cases the number of gears has been increased to 10. It can be seen the difference 4 kph in the maximal vehicle speed for the advantage of dynamic drive.

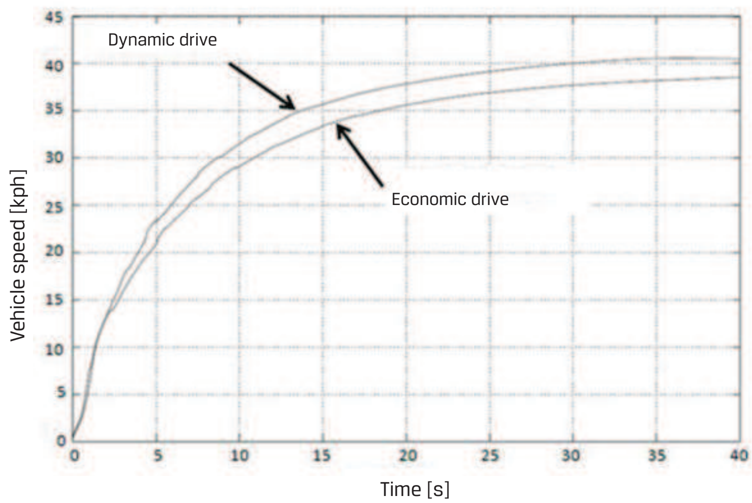


**Fig.10. Accelerating city buss for dynamic and economic drive.**

**Table 1. Toxic emission for selected cycle driving.**

| Version | $g_{eon}$ $g_e \left[\frac{g}{kWh}\right]$ | $G_{100}$ $\left[\frac{dcm^3}{100km}\right]$ | NOx *[ppm]* | CO *[ppm]* | CH *[ppm]* |
|---|---|---|---|---|---|
| ECO (5 gears) | 195,8 | 55,47 | 2471,2 | 892,39 | 14,23 |
| DYNAMIC(5 gears) | 198,12 | 56,93 | 2491,13 | 895,66 | 14,46 |
| ECO (10 gears) | 194,4 | 54,81 | 2467,71 | 890,35 | 14,18 |
| SPROT (10 gears) | 196,15 | 56,15 | 2490,42 | 894,35 | 14,35 |

## 7. Simulation tests of the modelled driving system for selected driving cycles

For evaluation the fuel consumption end exhaust gas components emission by buses had been elaborated many driving tests

During the simulation investigations have been used two driving tests:
- UITP Easy Urban,
- UITP Heavy Urban.

### 7.1. Demanded and realized vehicle speed

The course of demanded speed vs. time in the UITP Heavy Urban cycle has been presented in Fig.11. On the background of demanded vehicle speed has been laid down realized speed. On the smaller drawing (Fig.11) it has been presented more exactly segment (from the time 26s to 32s) of analyzed test. Fault of realization demanded speed, defined as difference between demanded and realized speed maximally don't overcome 2.2 kph. Fig. 12 presents the bus drive for the UITP Easy Urban drive, for which the fault of realization was maximally 1.9 kph.

### 7.2 Vehicle acceleration and selection the parameters of PID controller

The selection the parameters of PID controller for presented model have been preceded by great number of simulation tests. It is worth to notice, that for both cycles has been successfully chosen the controller with similar parameters, which assured good preset and realized speed compatibility.

**Tab.2. Values of PID parameters for individual driver cycles.[8]**

| | PID kontroler parameters value | | |
|---|---|---|---|
| Driving cycle | Proportional term | Integral term | Derivative term |
| Easy Urban | 6 | 0,11 | 0,1 |
| Heavy Urban | 4 | 0,11 | 0,1 |

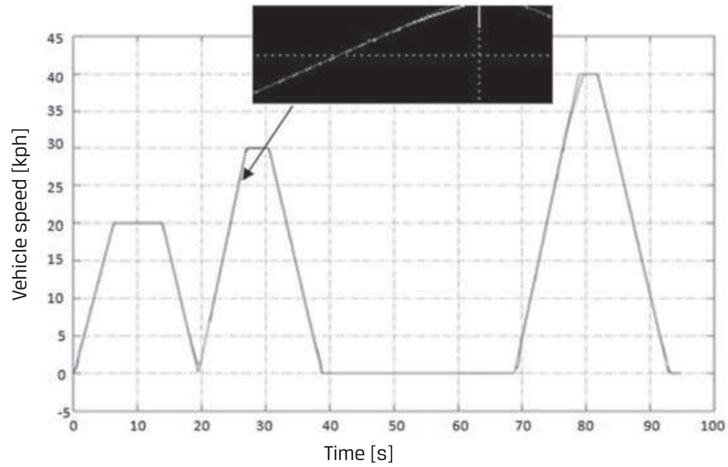


**Fig11. Demanded and realized vehicle speed at UITP Heavy Urban cycle [8].**

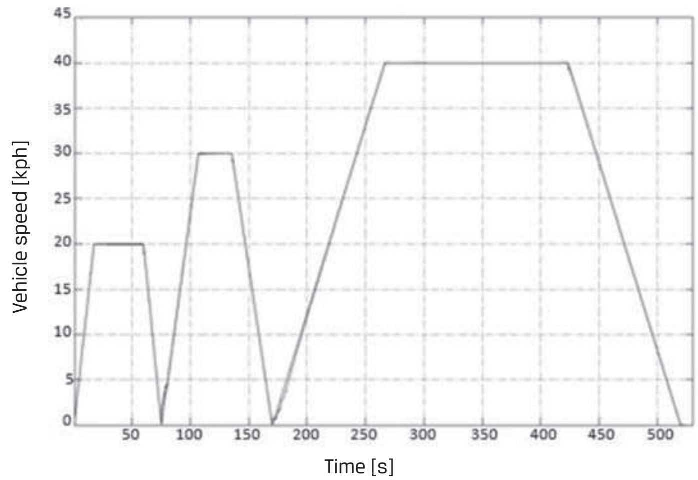


**Fig. 12. Demanded and realized vehicle speed at UITP Easy Urban cycle [8].**

## 8. Statements

- In spite of nonlinear engine characteristic and nonlinear characteristic of hydrokinetic converter, the PID controller with constant parameters assured the realisation of analysed driving cycles for different control mode: economic drive and dynamic drive.
- The presented model of control system can be considered correct, as simulation test results follow substantially exact analysed UITP driving cycles.

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