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Prediction of the Mileage Fuel Consumption of Passenger Car in the Urban Driving Cycle

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Summary: Driving cycles obtained when using chassis dynamometer (NEDC, ADAC Eco test) and under real-world driving conditions (CUEDC-P) were initially discussed in the paper. The research aimed at creating a simulation for prediction of mileage fuel consumption based on engine operational parameters, such as fuel consumption and effective power. They were determined for vehicle linear velocities (engine rotational speeds considering respective gearbox ratio) used in the UDC (this test is a sub-cycle of the NEDC test). The next parameter required for the simulation was fuel consumption in neutral gear (designated using engine test bench) and power to overcome resistance to motion. The whole algorithm for argumentation allowed determining the instantaneous fuel consumption for constant and variable velocities and, which is related to this, the simulated mileage fuel consumption. Considering measurement uncertainties, it was higher by 34.8 % to 46 % than the one given by manufacturer and vehicle users.
Key words: mileage fuel consumption; instantaneous fuel consumption; passenger car; driving cycles; ECE; UDC; EUDC; NEDC; ADAC Eco test; CUEDC-P.

INTRODUCTION

The problem of relationships between vehicle fuel consumption under real-world driving conditions and the one obtained as a result of experimental research had already been undertaken at the beginning of the 1970s of the 20th century. It was difficult to make an adequate prediction since properties of the urban traffic of cars depended on a too large number of factors (e.g. driver's driving style, vehicle's technical condition and its rated performance parameters, power transmission system efficiency [8], resistance to motion and fuel physicochemical properties [12]). Nevertheless, when assuming certain simplifications, efforts were made to accomplish this task. The European Driving Cycle, being part of the ECE vehicle regulation, was the Urban Driving Cycle [4, 13] (Fig. 1).

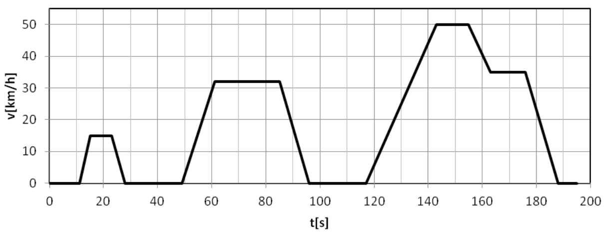

Fig. 1. Urban Driving Cycle (UDC) [13]

It is worth noticing that it includes engine operation in neutral gear and vehicle movement both with constant and variable velocities (variable accelerations or decelerations). It is completely possible to be accomplished under chassis dynamometer conditions in order to calculate the mileage fuel consumption based on measurement of the emission of toxic compounds [1,6,18] and is still being used. It is part of the NEDC cycle that consists of fourfold repeated UDC test and a single EUDC test [16].

Hardly reliable representation of fuel consumption according to the NEDC resulted in the development of a certain alternative in the form of the ADAC EcoTest cycle [1, 13]. It was designed in order to evaluate ecologically in an accurate, reliable and objective manner the performance of motor vehicles (testing of mileage fuel consumption using chassis dynamometer and exhaust gas analyser based on CO2 emission) and consisted of 3 tests [1, 13]:

- NEDC cold test (35% of total cycle) – an original is the standard NEDC test on a cold engine with real vehicle mass, instead of generally smaller test weight;
- NEDC hot test (35% of total cycle) – test conditions similar to those in the NEDC cold test but a difference consists in the testing of CO2 emission with warmed up engine and air conditioning turned on onto the set temperature of 293 K;
- ADAC motorway test (30% of total cycle) – this test is designed for a motorway ride at a velocity of 130 km/h.

ADAC EcoTest NEDC cold test allowed obtaining the CO2 emission in 2010 within the range higher by 1% than that reported by the manufacturer, whereas it was lower by 20% in relation to the data reported by users [1, 13].

ADAC EcoTest NEDC hot test showed the CO2 content in exhaust gas in 2010 to be within the range higher by 4% that that reported by the manufacturer, whereas it was 17% lower in relation to the data given by users [1, 13].

Some authors [7,11] determined the mileage fuel consumption based on the emission of carbon dioxide from the following relationship:

$$Q = \frac{CO_2}{k_{CO_2} \cdot 10 \cdot \rho_F}, \qquad (1)$$

where:
Q – mileage fuel consumption [dm³/100 km],
CO_2 – carbon dioxide emission in exhaust gas [g/km],
k_{CO_2} – coefficient of proportionality for total and complete combustion (= 3.15),
ρ_F – fuel density [kg/dm³].

Another solution, different from fuel consumption testing using chassis dynamometer, was to carry out the driving cycle under real-world conditions. It was called the CUEDC-P (Composite Urban Emission Driving Cycle for Petrol Vehicles), it lasted for thirty minutes and consisted of four sub-cycles: Residential, Arterial, Freeway and Congested. The adopted model assumed the determination of instantaneous fuel consumption based on theoretical formulas as well as on the driving cycle test mentioned above. Very high reliability of the estimation of instantaneous fuel consumption was demonstrated, which was barely different from the values measured in the real-world cycle CUEDC-P [2].

In the available literature, no determinations of mileage fuel consumption according to the driving cycle based on tests performed on an engine test bed have been reported. Usually, they were vehicle "mileages" obtained on the chassis dynamometer, their examples being the NEDC test, ADAC EcoTest or the real-world driving cycle test, i.e. CUEDC-P. Therefore, the author decided to take up this problem based on the standard driving cycle test, i.e. Urban Driving Cycle.

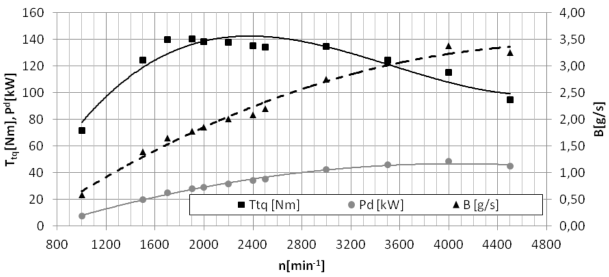

Fig. 2. External characteristic of FIAT Multijet 1.3 JTD engine where: **Ttq** – engine torque, **P^d** – engine effective power, **B** – fuel consumption, **n** – engine rotational speed

OBJECTIVE AND METHODS OF EXPERIMENTS

The objective of this study was to make the prediction of mileage fuel consumption for a passenger car equipped with a compression-ignition engine with the Common Rail fuel supply system according to the UDC (Urban Drive Cycle) based on tests. They were made using an engine test bed. Experiments were performed in conformity with the research methods of the piston combustion engine standard PN-ISO 15550 [15] and made according to the requirements specified therein.

The next aspect was to create a simulation that include determination of resistance to motion and calculation of instantaneous fuel consumption under specific motion conditions.

COURSE OF TESTS

The course of tests consisted in the creation of the external characteristics obtained for FIAT Panda with Multijet 1.3 16V JTD engine (Fig. 2). It was fed with a full dose of ON EKODIESEL fuel with the cetane number 51.1. Ambient parameters during the test were as follows:
- ambient temperature Ta = 294 K,
- ambient pressure pa = 98.5 kPa,
- relative humidity = 40%.

The measured engine torque and its effective power were corrected according to the relations comprised in the normative reference (PN-ISO 15550, [15]).

The points created for torque, effective power and fuel consumption formed characteristic curves as mean values of 4 samples. The procedure consisted in measurements when determining rotational speeds upward and then downward and was repeated twice. For each variable, a trend curve with a high value of the square of the coefficient of correlation (R^2) was matched, which was evidence of a good fitting of theoretical and real values. For further simulation, approximated equations of the trend curves for fuel consumption and engine effective power were determined. They were as follows (symbols and their descriptions as under Fig. 2):

a) fuel consumption (R^2 = 0.9976):

$$B = -9.6582 \cdot 10^{-14} n^4 + 1.0324 \cdot 10^{-9} n^3 - 4.0088 \cdot 10^{-6} n^2 + 7.5074 \cdot 10^{-3} n - 3.842, \qquad (2)$$

b) engine effective power (R^2 = 0.9937):

$$P^d = -4.13301 \cdot 10^{-6} n^2 + 0.033592345 n - 21.64131491, \qquad (3)$$

Determination of the resistance to motion and the power required to overcome it as well as the dependence of fuel consumption on

effective power for specific linear velocities required the use of vehicle details and set motion conditions (Tab.1).

Table 1. FIAT Panda – basic details and vehicle motion conditions

variable	value	unit	where:
M_V	1549	kg	total vehicle mass
η_{PT}	0.9	-	power transmission system efficiency
r_k	0.270	m	wheel kinematic radius
c_x	0.3	-	dimensionless air resistance coefficient
γ_P	0.9	-	fill factor
b_k	1.578	m	wheel track
H	1.540	m	vehicle overall height
F	2.190	m²	vehicle frontal area
f_R^0	0.012	-	basic rolling resistance coefficient
A	0.00005	s²/m²	additional rolling resistance coefficient
i_{GI}	3.909	-	first gear ratio
i_{GII}	2.158	-	second gear ratio
i_{GIII}	1.345	-	third gear ratio
i_{FD}	3.438	-	final drive ratio

Basic assumptions referring to the selection of values for vehicle details are as follows [3, 16]:

- vehicle was loaded to its permitted gross mass (*MV*),
- power transmission system power efficiency value was adopted as for a passenger car,
- wheel kinematic radius resulted from tyre size (tyre inflated to the pressure being given by manufacturer) and wheel rim taking into consideration static loads during motion,
- fill factor γ_P value was adopted as for a passenger car,
- vehicle frontal area was calculated based on the following relationship:

$$F=\gamma_P\, b_k H,$$

- basic rolling resistance coefficient fR^0 value was adopted as for smooth asphalt road pavement,
- additional rolling resistance coefficient *A* value was adopted as for most road pavements being applied.

Based on the external characteristics, the values of fuel consumption and engine effective power were specified for the vehicle velocity equal to 15, 32, 35 and 50 km/h used in the UDC. Engine rotational speeds were calculated from the following relationship for vehicle linear velocity [3,16]:

$$v=\omega_w r_k=\frac{2\pi r_k n_w}{60}=\frac{2\pi\cdot r_k\cdot n}{60 i_G i_{FD}}, \tag{4}$$

where:
v – vehicle linear velocity [m/s],
ww- wheel angular velocity [1/min],
rk -wheel kinematic radius [m],
nw – wheel rotational speed [min⁻¹],
n – engine rotational speed [min⁻¹],
iG - gearbox ratio (selectable),
iFD – final drive ratio (constant).

After transformation, it was as follows:

$$n=\frac{60\cdot v\cdot i_G\cdot i_{FD}}{2\pi r_k}, \tag{5}$$

Total power transmission system ratio is as follows [3,16]:

$$i_{PT}=i_G\, i_{FD}. \tag{6}$$

Engine rotational speeds and fuel consumption and effective power values were calculated from Equations (5), (2) and (3).

Table 2. Rotational speeds (*n*), fuel consumption (*B*) and effective power (P^d)

gear	v	v	i_{PT}	n	n	B	P^d
	km/h	m/s	-	s⁻¹	min⁻¹	g/s	kW
I	15	4.17	13.439	33.01	1980	1.84	28.7
II	32	8.89	7.419	38.87	2332	2.10	34.2
III	35	9.72	4.624	26.50	1590	1.49	21.3
III	50	13.89	4.624	37.86	2271	2.06	33.3

PREDICTION OF MILEAGE FUEL CONSUMPTION AT CONSTANT VELOCITIES

In order to extrapolate the instantaneous fuel consumption, vehicle motion resistances (rolling resistance and air resistance) had to be determined. The first one was determined in the following manner [3,16]:

$$F_R=f_R M_V\cdot 9.81=f_R^0(1+Av^2)\,M_V\cdot 9.81, \tag{7}$$

where:
F_R – rolling resistance [N],
f_R – rolling resistance coefficient,
M_V – vehicle mass [kg],
f_R^0 – basic rolling resistance coefficient,
A – additional rolling resistance coefficient [s²/m²],
v – vehicle linear velocity [m/s].

Air resistance was described by the following equation [16]:

$$F_A=c_x F p_d=c_x F\frac{\rho v^2}{2}, \tag{8}$$

where:
F_A – air resistance[N],
c_x – air resistance coefficient,
F – vehicle frontal area [m²],
p_d – dynamic pressure [N/m²].
ρ – air density [kg/m³],
v – vehicle and air (wind) relative velocity [m/s].

Air density was determined from the following relationship [3]:

$$\rho=\frac{0.46 p_b}{T}, \tag{9}$$

where:
ρ – air density [N s² m⁻⁴],
p_b – barometric pressure [1 mm Hg = 133.33 Pa],
T – air temperature [K].

When bringing the air density to reference conditions (pressure *pt* = 100 kPa = 750 mm Hg, temperature *Tr* = 298 K) the following value was obtained:

$$\rho = \frac{0.46 \cdot p_b}{T} = \frac{0.46 \cdot 750}{298} = 1.16\,[\mathrm{kg/m^3}], \qquad (10)$$

Thus, equation (8) took the following form:

$$F_A = 0.579 c_x F v^2, \qquad (11)$$

Prediction of the mileage fuel consumption was determined based on the value of instantaneous fuel consumption. To determine this value, knowing the value of fuel consumption in neutral gear (Multijet 1.3 JTD engine), fuel efficiency rate and power required to overcome resistance to vehicle motion (FIAT Panda) was required. The process of calculating these functions is presented below.

For example, for the velocity of 15 km/h, it is as follows:

a) fuel consumption in neutral gear a (constant value) was determined from the following relationship [2]:

$$\alpha = B_v = \frac{B_m}{\rho_F}, \qquad (12)$$

where:
B_v – volumetric fuel consumption [dm³/s],
B_m – mass fuel consumption [g/s],
ρ_F – fuel density [g/dm³].

In Table 3 below, fuel consumption in neutral gear was determined.

Table 3. Fuel consumption in neutral gear

n_{NG}	B_m	ρ_F	B_v	$\alpha = B_v$
min⁻¹	g/s	g/dm³	dm³/s	mdm³/s
800	0.06	820.1	0.00007	**0.0731**

where:
n_{NG} – engine rotational speed in neutral gear.

b) fuel efficiency rate β was determined by the following function [2]:

$$\beta = \frac{B_{Cv}}{P_C^d}, \qquad (13)$$

where:
B_{Cv} – volumetric instantaneous fuel consumption [mdm³/s],
P_C^d – engine power corresponding to volumetric instantaneous fuel consumption [kW].

In Table 4, fuel efficiency rate was determined, taking into consideration the data from Table 3.

Table 4. Fuel efficiency rate (gear I)

n	B_m	ρ_F	B_v	B_{CV}	P_C^d	β
min⁻¹	kg/s	kg/dm³	dm³/s	mdm³/s	kW	mdm³/kJ
1980	0.00184	0.8201	0.002239	2.238782	28.7	**0.0781**

c) power required to overcome rolling resistance and air resistance (determined as an example for the velocity of 15 km/h) [3,16]:

$$P_T = P_R + P_A = F_R \cdot v + F_A \cdot v = (F_R + F_A) \cdot v, \qquad (14)$$

where:
P_T – total power required to overcome rolling resistance and air resistance by vehicle [kW],
P_R – power required to overcome rolling resistance by vehicle [kW],
P_A – power required to overcome air resistance by vehicle [kW],
F_R – rolling resistance [kN],
F_A – air resistance [kN],
v – vehicle linear velocity [m/s].

In Table 5 below, the power required to overcome rolling resistance and air resistance was calculated (resistance to motion was determined based on relationships (7) and (11)).

Table 5. Power required to overcome rolling resistance and air resistance

v	F_R	P_R	F_A	P_A	F_R+F_A	P_R+P_A
m/s	kN	kW	kN	kW	kN	kW
4.2	0.179	0.75	0.01	0.028	0.2	**0.77**

d) instantaneous fuel consumption fc when driving with constant velocity (for v = 15 km/h) was obtained from the following relationship [2]:

$$f_c = \alpha + \beta \cdot P_T = 0.073 + 0.078 \cdot 0.7 = 0.133\,[\mathrm{mdm^3/s}], (15)$$

e) mileage fuel consumption Q (v = 15 km/h = 4.2 m/s):

$$Q = \frac{f_c}{v} = \frac{0.134}{4.2} = 0.0319\ [\mathrm{mdm^3/m}], \qquad (16)$$

After changing the units from [mdm³/m] to [dm³/100km], the mileage fuel consumption for the constant velocity of 15 km/h amounted to **3.2 [dm³/100 km].**

PREDICTION OF MILEAGE FUEL CONSUMPTION AT VARIABLE VELOCITIES

The author of this paper has found out that determination of the simulated mileage fuel consumption both for constant velocities (similarly as in section 4) and variable ones, taking into consideration vehicle inertia resistance, was possible.

Instantaneous fuel consumption at variable velocities 15, 32, 35 and 50 km/h was defined in a similar manner as in the case of predicting the mileage fuel consumption at constant velocities, taking into account the value of vehicle accelerations and, which is related to it, vehicle inertia resistance. At the time when vehicle decelerated (negative acceleration value – delay), engine braking (instantaneous fuel consumption equal to zero) and engine running time in neutral gear (after pressing the clutch pedal) were taken into consideration.

Instantaneous fuel consumption at variable velocity (for example for v = 15 km/h = 4.2 m/s and a = 1.04 m/s²) was calculated in the following manner [2]:

$$f_c = \alpha + \beta \cdot (P_T + P_I) + \beta \cdot P_I, \qquad (17)$$

where:
f_c – instantaneous fuel consumption [mdm³/s],
α – fuel consumption in neutral gear [mdm³/s],
β – fuel efficiency rate [mdm³/kJ],
P_T – power required to overcome rolling resistance and air resistance by vehicle [kW],
P_I – power required to overcome inertia resistance by vehicle [kW].

Fuel consumption in neutral gear and fuel efficiency rate were determined based on the above-presented relationships in section prediction of mileage fuel consumption at constant velocities a and b, while the power required to overcome rolling resistance and air resistance by vehicle was determined based on Equation (14).

The power required to overcome inertia resistance was determined from relationship [2]:

$$P_I = \frac{M_V \cdot a \cdot v}{1000}, \tag{18}$$

where:
P_I – power required to overcome inertia resistance [kW],
M_V – vehicle mass = 1520 [kg],
a – vehicle acceleration [m/s²],
v – vehicle linear velocity [m/s].

For example for FIAT Panda, it was as follows (a=1.04 m/s², v= 15 km/h= 4.2 m/s):

$$P_I = \frac{1520 \cdot 1.04 \cdot 4.2}{1000} = 6.59[kW]. \tag{19}$$

The value of function (18) allowed determining the instantaneous fuel consumption according to formula (17):

$$\begin{aligned} f_c &= \alpha + \beta \cdot (P_T + P_I) + \beta \cdot P_I = \\ &0.073 + 0.078 \cdot (0.77 + 6.59) + 0.078 \cdot 6.59 \\ &= 1.162\ [\text{mdm}^3/\text{s}]. \end{aligned} \tag{20}$$

Variable instantaneous fuel consumption for variable velocities 15, 32, 35 and 50 km/h at specific vehicle accelerations was determined in a similar manner. In Table 6 above, a simulated instantaneous fuel consumption in the UDC was calculated for respective vehicle motion phase. This was the basis for calculation of aggregate simulated instantaneous fuel consumption (taking into account all phases).

Total simulated mileage fuel consumption was determined from the following relationship:

$$Q = \frac{\sum (f_c \cdot t)}{l} = \frac{7\ .682}{1017} = 0.0764, \tag{21}$$

Table 6. Simulated instantaneous fuel consumption according to the UDC [13]

Phase No.	Action	v_i	v_f	t	t_{sum}	a	f_c	$f_c\, t$
		km/h	km/h	s	s	m/s²	mdm³/s	mdm³
1	neutral gear			11	11		0.073	0.805
2	speeding up	0	15	4	15	1.04	1.162	4.649
3	constant velocity	15	15	9	23		0.134	1.202
4	braking	15	10	2	25	-0.69	0.000	0.000
5	braking, clutch disengaged	10	0	3	28	-0.92	0.073	0.219
6	neutral gear			21	49		0.073	1.536
7	speeding up	0	15	5	54	0.83	0.955	4.773
8	gear change			2	56		0.073	0.146
9	speeding up	15	32	5	61	0.94	2.114	10.572
10	constant velocity	32	32	24	85		0.213	5.105
11	braking	32	10	8	93	-0.75	0.000	0.000
12	braking, clutch disengaged	10	0	3	96	-0.92	0.073	0.219
13	neutral gear			21	117		0.073	1.536
14	speeding up	0	15	5	122	0.83	1.003	5.017
15	gear change			2	124		0.073	0.146
16	speeding up	15	35	9	133	0.62	1.817	16.349
17	gear change			2	135		0.073	0.146
18	speeding up	35	50	8	143	0.52	2.046	16.371
19	constant velocity	50	50	12	155		0.394	4.732
20	braking	50	35	8	163	-0.52	0.000	0.000
21	constant velocity	35	35	13	176		0.252	3.279
22	gear change			2	178		0.073	0.146
23	braking	32	10	7	185	-0.86	0.000	0.000
24	braking, clutch disengaged	10	0	3	188	-0.92	0.073	0.219
25	neutral gear			7	195		0.073	0.512
							Total	**77.682**

where:
vi – initial velocity, *vf* – final velocity, *t* – phase length, *tsum* – total time of respective phases, *a* – vehicle acceleration, *fc* – instantaneous fuel consumption

where:
Q – mileage fuel consumption [mdm³/m],
l – distance “covered“ during the cycle = 1017. [m].

For FIAT Panda, total simulated mileage fuel consumption after changing the units from [mdm³/m] to [dm³/100km] amounted to **7.64** [dm³/100km] and was higher by 41.45 % than real-world mileage fuel consumption (analysis of measurement uncertainties see section 7).

Real-world mileage fuel consumption under determination was based on the type-approval tests according to the guidelines of European Commission Directive 1999/100/EC amounted to 5.4 dm³/100 km in the urban driving cycle [4, 5, 19]. Opinions of the users of vehicles equipped with Multijet 1.3 JTD engine report the same average fuel consumption under urban traffic conditions [14, 17].

ANALYSIS OF MEASUREMENT UNCERTAINTIES

Table 7 (next page) shows the values of measurement uncertainties (A type standard uncertainty, B type standard uncertainty, expanded standard measurement uncertainty) for the values of fuel consumption obtained in the external characteristics (Fig. 2). They were calculated based on the relationships described in the Guide to the Expression of Uncertainty in Measurement [9]. Expanded standard measurement uncertainties for engine effective power (external characteristics in Fig. 2) were calculated in a similar manner. Table 8 presents the percentage deviations for the observed values. The values of measurement uncertainties for fuel consumption and effective power for the engine rotational speed obtained by approximations used in the UDC are presented below (Tab. 9). Owing to the fact that the values of fuel consumption and engine effective power for the UDC rotational speeds were determined based on the trend curve bit and not measurements, the author of this paper found out that the maximum measurement deviation resulting from two measurement deviations of adjacent rotational speeds had to be adopted for each value. For example, for the rotational speed of 1980 min^{-1}, adjacent speeds were 1900 and 2000 min^{-1} with the value of deviations being respectively 1.6 and 1.1 % for fuel consumption and 1.1 and 0.7 % for effective power. Then, a deviation of 1.6% was chosen for fuel consumption and 1.1% for effective power. This was reflected in the ranges of measurement uncertainties for fuel consumption equal to 1.84 ± 0,03 g/s and engine effective power equal to 28.7 ± 0.4 kW.

When assuming the lower value of measurement, mileage fuel consumption was equal to 7.28 dm³/100 km (higher by 34.8 % than real-world mileage fuel consumption), whereas for the upper value of measurement the mileage fuel consumption amounted to 7.88 dm³/100 km (higher by 46 % than real-world mileage fuel consumption).

CONCLUSIONS

The performed extrapolation of mileage fuel consumption allowed for the following conclusions:

a) mileage fuel consumption determined by simulation according to the reported scheme ranges from 7.28 to 7.88 dm³/100 km (value given be vehicle’s manufactures is to 5.4 dm³/100 km);
b) the method included calculation of fuel consumption for the permitted gross vehicle mass; in the chassis dynamometer test, this weight could be lower, hence the lower value of mileage fuel consumption given by manufacturer;
c) the real-world mileage fuel consumption is affected by a great deal of factors not included in this paper or simplified (e.g. variable fuel density, driving conditions – external pressure and temperature, rolling resistance coefficient, tyre inflation pressure, elevations, degree of engine warming up, power transmission system efficiency resulting from changes in the viscosity of gear oils, etc.);
d) fuel consumption according to the type-approval tests carried out when using chassis dynamometer is also loaded by its own error (resulting for instance from inaccuracies of the measuring instruments for emissions of toxic compounds), not given by manufacturer;
e) the UDC is not the best cycle that reflects the simulated mileage fuel consumption in the urban driving cycle for passenger cars being equipped with a compression-ignition engine with the Common Rail fuel supply system;
f) inconsistency of the cycle with the method being assumed results, among other, from high values of accelerations that increase the instantaneous fuel consumption even nine-fold (for example for FIAT Panda: constant v = 15 km/h, constant f_c = 0.134, variable v = 15 km/h, a = 1.04 m/s², f_c = 1.162);
g) the arguments being mentioned allow treating the prediction as an initial one and moving in this research direction in order to possibly modify this method and apply another test than the UDC (or apply its modified version) that will allow reflecting the real-world mileage fuel consumption in the best manner.

Table 7. Values of measurement uncertainties for particular engine rotational speeds

n	*B*	$u_A(B)$	$u_B(B)$	*U(B)*	B+-U(B)
min^{-1}	g/s	g/s	g/s	g/s	g/s
1000	0.59	0.006	0.006	0.0163	0.59 ± 0.02
1500	1.39	0.007		0.0183	1.39 ± 0.02
1700	1.65	0.016		0.0337	1.65 ± 0.04
1900	1.78	0.009		0.0208	1.78 ± 0.03
2000	1.85	0.005		0.0153	1.85 ± 0.02
2200	2.01	0.003		0.0129	2.01 ± 0.02
2400	2.08	0.046		0.0922	2.08 ± 0.10
2500	2.20	0.007		0.0183	2.20 ± 0.02
3000	2.75	0.009		0.0208	2.75 ± 0.03
3500	3.08	0.012		0.0271	3.08 ± 0.03
4000	3.38	0.004		0.0141	3.38 ± 0.02
4500	3.25	0.010		0.0224	3.25 ± 0.03

where:
B – engine fuel consumption (mean value of 4 measurements), *uA* – A type standard uncertainty, *uB* – B type standard uncertainty, *U* – expanded standard measurement uncertainty

Table 8. Deviations for fuel consumption and engine effective power

n	B+- U(B)	*(U(B)/B) *100 %*	P^d+-U(P^d)	*(U(P^d)/P^d) *100 %*
min^{-1}	g/s	%	kW	%
1000	0.59 ± 0.02	3.3	7.5 ± 0.2	2.7
1500	1.39 ± 0.02	1.4	19.5 ± 0.3	1.5
1700	1.65 ± 0.04	2.4	24.9 ± 0.3	1.2
1900	1.78 ± 0.03	1.6	27.9 ± 0.3	1.1
2000	1.85 ± 0.02	1.1	29.0 ± 0.2	0.7
2200	2.01 ± 0.02	1.1	31.7 ± 0.2	0.7
2400	2.08 ± 0.10	4.8	34.1 ± 0.2	0.6
2500	2.20 ± 0.02	1.0	35.2 ± 0.2	0.6
3000	2.75 ± 0.03	1.1	42.3 ± 0.3	0.7
3500	3.08 ± 0.03	1.0	45.6 ± 0.2	0.4
4000	3.38 ± 0.02	0.6	48.3 ± 0.2	0.4
4500	3.25 ± 0.03	1.0	44.7 ± 0.3	0.7

Table 9. Values of measurement uncertainties

n	n_1	n_2	d_1	d_2	$d=max(d_1,d_2)$	*B*	*U(B)=Bd*	*B-U(B)*	*B+U(B)*
[min^{-1}]	[min^{-1}]	[min^{-1}]	[%]	[%]	[%]	[g/s]	[g/s]	[g/s]	[g/s]
800	1000	1000	3.3	3.3	3.3	0.06	0.01	0.05	0.07
1980	1900	2000	1.6	1.1	1.6	1.84	0.03	1.81	1.87
2332	2200	2400	1.1	4.8	4.8	2.10	0.10	2.00	2.20
1590	1500	1700	1.4	2.4	2.4	1.49	0.04	1.45	1.53
2271	2200	2400	1.1	4.8	4.8	2.06	0.10	1.96	2.16
n	n_1	n_2	d_1	d_2	$d=max(d_1,d_2)$	P^d	$U(P^d)=P^d d$	P^d-$U(P^d)$	P^d+$U(P^d)$
[min^{-1}]	[min^{-1}]	[min^{-1}]	[%]	[%]	[%]	[kW]	[kW]	[kW]	[kW]
1980	1900	2000	1.1	0.7	1.1	28.7	0.4	28.3	31.1
2332	2200	2400	0.7	0.6	0.7	34.2	0.3	33.9	34.5
1590	1500	1700	1.5	1.2	1.5	21.3	0.4	20.9	21.7
2271	2200	2400	0.7	0.6	0.7	33.3	0.3	33.0	33.6

where:
n_1 – adjacent lower engine rotational speed for which measurements were made, n_2 – adjacent higher engine rotational speed for which measurements were made, d_1 – measurement deviation for speed n_1, d_2 – measurement deviation for speed n_2, d – maximum deviation selected from among d_1 and d_2.

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PROGNOZOWANIE PRZEBIEGOWEGO ZUŻYCIA PALIWA SAMOCHODU OSOBOWEGO W CYKLU MIEJSKIM

Streszczenie. W artykule wstępnie omówiono cykle jazdy uzyskiwane przy wykorzystaniu hamowni podwoziowej (NEDC, ADAC EcoTest) oraz w warunkach rzeczywistych (CUEDC-P). Celem publikacji było stworzenie symulacji prognozującej przebiegowe zużycie paliwa na podstawie parametrów operacyjnych silnika takich jak: zużycie paliwa oraz moc użyteczna. Wartości tych wielkości zostały wyliczone przy użyciu równań krzywych trendu dobranych do punktów pomiarowych charakterystyki zewnętrznej silnika. Zostały wyznaczone dla prędkości liniowych pojazdu (prędkości obrotowych silnika uwzględniając określone przełożenia skrzyni biegów) wykorzystywanych w cyklu UDC (test ten jest podcyklem testu NEDC). Pozwoliło to na uzyskanie wskaźnika efektywności paliwa dla prędkości 15,32,35, i 50 km/h. Kolejnym parametrem niezbędnym do symulacji było zużycie paliwa na biegu jałowym (wyznaczone przy użyciu hamowni silnikowej) oraz moc potrzebna na pokonanie oporów ruchu. Opory toczenia i powietrza sprecyzowano relacjami wykorzystując dane pojazdu oraz warunki ruchu. Cały algorytm rozumowania pozwolił na wyznaczenie chwilowego zużycia paliwa dla stałych i zmiennych prędkości, a co jest z tym związane, symulacyjnego przebiegowego zużycia paliwa. Uwzględniając niepewności pomiarowe było one od 34,8 % do 46 % wyższe niż to podane przez producenta i użytkowników pojazdu.

Słowa kluczowe: przebiegowe zużycie paliwa; chwilowe zużycie paliwa; samochód osobowy; cykle jezdne; ECE; UDC; EUDC; NEDC; ADAC Eco test; CUEDC-P.