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COMPARISON OF THE INSTANTANEOUS FUEL CONSUMPTION OF VEHICLES WITH A DIFFERENT TYPE OF PROPULSION SYSTEM AT CONSTANT VELOCITY

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Abstract

The paper presents a comparative analysis of the instantaneous fuel consumption of a FIAT Panda vehicle equipped with a 1.3 JTD MultiJet compression ignition engine with Common Rail fuel system. Different types of propulsion system were taken into consideration (engine positioned front-lengthwise to the direction of travel and rear-wheel drive, engine positioned front-transversely to the direction of travel and front-wheel drive, and all-wheel drive). The method for determining the instantaneous fuel consumption was based on an experimental part where the load characteristics were established (relationship between specific fuel consumption and engine torque). It was carried out for the steady states of these parameters corresponding to specific traffic conditions that represent the resistance to motion, i.e. rolling resistance and air resistance. Technical and operating characteristics of a vehicle and its design features, such as maximum weight, transmission system ratios, dynamic wheel radius, drag coefficient, width and height, and efficiency of propulsion system, had a significant impact on their individual contribution. The efficiency of transmission was adopted from a simulation for different types of propulsion system. It was important in determination of the value of instantaneous fuel consumption for constant vehicle velocities used in the UDC test (Urban Driving Cycle – subtest of the EUDC cycle). The lowest fuel consumption for a given speed of a car occurred for the front-wheel drive transmission system, whereas the highest for the all-wheel drive system (4x4).

Keywords: *instantaneous fuel consumption, UDC cycle, passenger car, propulsion system, constant velocity*

1. Introduction

Fuel consumption is an important operational parameter defining the energy efficiency of a vehicle. The value of this indicator measured under real-world conditions differs from that furnished by a car manufacturer [9]. A carmaker producing a specific range of vehicles determines this parameter according to strictly defined methodology. Experiments involving fuel consumption are associated with granting the vehicle approval and are carried out under chassis dynamometer conditions. This is a preliminary prediction referring, among others, to emission of toxic compounds that is based on a specific type of dynamometer test, during which the level of exhaust gas and the percentage of particular toxic compounds coming out from vehicle's exhaust system are examined, and then fuel consumption is calculated against this background [4, 11]. For Europe, the test being used by many car manufacturers for determining this parameter is the NEDC (New European Driving Cycle) test. It is a part of the UNECE regulation and is composed of two subcycles: UDC (Urban Driving Cycle) and EUDC (Extra Urban Driving Cycle; 1990). In the United States of America, the EPA Federal Test (SFTP US06/SC03; 2008) is an equivalent of this test, while the JC08 test (2008) in Japan [9, 11].

A certain alternative for the above-mentioned NEDC test is the ADAC EcoTest, which incorporates three modules: NEDC cold (35% of the whole cycle), NEDC hot (35% of the whole cycle), and ADAC motorway (30% of the whole cycle), being distinctly different in the conditions of fuel consumption measurement [1]. The ADAC EcoTest NEDC cold allowed obtaining the CO_2 emission in 2010 being by 1% higher than that furnished by a car manufacturer, while by 20%

lower in relation to the data provided by car users [1, 9].

A different solution than testing the fuel consumption with a chassis dynamometer is the real-world driving cycle called CUEDC-P (Composite Urban Emission Drive Cycle for Petrol Vehicles). It lasted thirty minutes and was composed of four subcycles: Residential, Arterial, Freeway and Congested. The adopted model assumed determination of instantaneous fuel consumption on the basis of theoretical formulas, as well as on the basis of the above-mentioned driving cycle. A very high level of reliability for estimation of instantaneous fuel consumption was demonstrated, being slightly different from the values measured during the real-world CUEDC-P cycle [2]. The problems of fuel consumption under real-world driving conditions have been also taken up by other authors [5, 6].

Thus, literature provides the concept of instantaneous fuel consumption tests, both these experimental ones performed under laboratory conditions and those conducted under the real-world ones. Usually, these are "mileages" of a vehicle using a chassis dynamometer; examples of them are the NEDC test, ADAC EcoTest, or the CUEDC-P test for real-world conditions. Therefore, the authors decided to take up this problem on the example of standard subcycle Urban Driving Cycle and compare the simulation instantaneous fuel consumption being determined experimentally for a Fiat 1.3 JTD engine for three types of simulationally adopted propulsion systems:

a) classical drive system – engine is positioned lengthwise at the front of the vehicle; the driving moment is transmitted through the clutch, multi-speed manual gearbox and propeller shaft with joints to the driving axle, which constitutes an assembly of final drive together with differential and axle shafts that transmit power to wheels,
b) block front-wheel drive system – engine, clutch, multi-speed manual gearbox, final drive and differential are in a common body being directly connected with the engine; the driving moment is transmitted successively through the above-mentioned subassemblies to axle shafts, and then to wheels,
c) four-wheel drive (4x4) system – power is transmitted to all vehicle's wheels; an additional system is the transfer box, while drive transmission is accomplished by means of at least two propeller shafts (per one axle) [8].

2. Research objective and experimental methods

The aim of this study was to conduct a comparative analysis of vehicle fuel consumption for three types of simulationally adopted propulsion system, i.e. classical, block front-wheel drive and all-wheel drive (4x4). This analysis was performed for a FIAT Panda vehicle equipped with a MultiJet 1.3 JTD engine. Its load characteristics were determined according to the methodology and requirements referring to the measurements of respective parameters being provided in the standard [10]. Comparison of instantaneous fuel consumption was conducted for the established conditions of vehicle motion.

3. Course of experimental testing

Experimental testing consisted in making, based on measurements, the characteristics of specific fuel consumption in relation to engine torque for particular engine speeds. They were determined for vehicle velocities equal to 15, 32, 35 and 50 km/h used in the UDC cycle from the following relation [3, 11]:

$$v = \omega_K \cdot r_d = \frac{2\pi \cdot n_K}{60} \cdot r_d = \frac{2\pi \cdot n}{60 \cdot i_{UN}} \cdot r_d , \qquad (1)$$

where:

v – vehicle velocity [m/s],

ω_K – angular velocity of wheels [1/s],
r_d – dynamic wheel radius [m],
n_K – wheel speed [min^{-1}],
n – rotational speed of engine crankshaft [min^{-1}],
i_{UN} – drive system ratio.

After transformation, it assumed the following form:

$$n = \frac{60 \cdot v \cdot i_{UN}}{2\pi \cdot r_d}. \tag{2}$$

In order to determine the values of engine speeds, overall transmission ratios of respective drives were to be taken into consideration:

a) classical and block drive transmission system:

$$i_{UN}=i_{SP} i_{SB}\, i_{PG}, \tag{3}$$

b) 4x4 drive:

$$i_{UN}=i_{SP} i_{SB}\, i_{SR}\, i_{PG}, \tag{4}$$

where:
i_{SP} – clutch ratio (equal to 1),
i_{SB} – gearbox ratio (selectable),
i_{SR} – transfer box ratio (adopted constant ratio = 1),
i_{PG} – final drive ratio (constant).

On the basis of the aforesaid relationships, the values of engine speeds corresponding to particular vehicle velocities were determined (Tab. 1).

Tab. 1. Overall drive transmission ratio i_{UN}, vehicle velocity v, engine speed n, dynamic wheel radius r_d

Parameter	Gear	i_{UN}	v	v	n	r_d
Unit		[-]	[km/h]	[m/s]	[min^{-1}]	[m]
Value	I	13.439	15	4.17	1980	0.27
	II	7.419	32	8.89	2332	
	II		35	9.72	2590	
	III	4.624	35	9.72	1590	
	III		50	13.89	2271	

For the rotational speeds of engine crankshaft being determined in Tab. 1, the load characteristics for a FIAT 1.3 JTD engine was made, which is presented in Fig. 1.

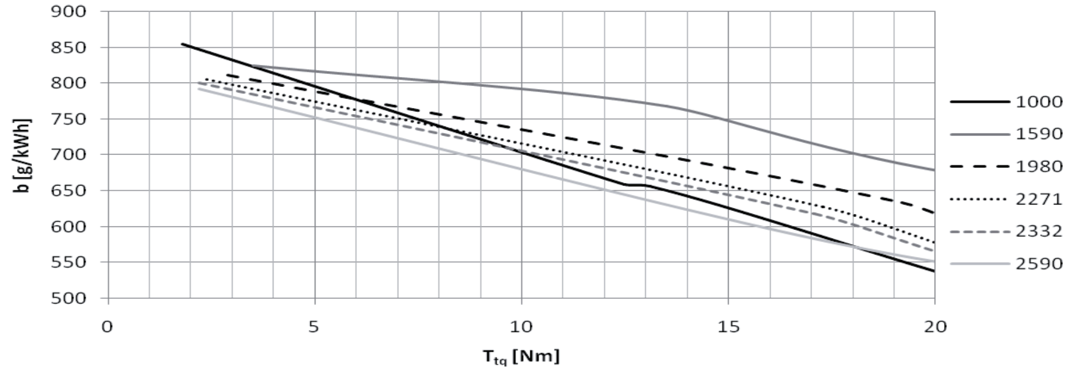

Fig. 1. The load characteristics of FIAT 1.3 JTD engine: b – specific fuel consumption, T_{tq} – engine torque; Legend – rotational speeds of engine crankshaft

In order to read the values of specific fuel consumption for engine load moment from the above curve and make a comparative analysis of instantaneous fuel consumption, vehicle data were needed and motion conditions were to be adopted by simulation.

4. Technical and operating characteristic of a vehicle and motion conditions

Specification of the resistance to motion and, as a consequence, the value of engine torque needed to overcome it, required to use vehicle technical and operating characteristics and to determine specific motion conditions (Tab. 2).

Tab. 2. Basic data characterising a FIAT Panda vehicle and motion conditions

Parameter	Value	Unit	where:
m_c	1455	[kg]	maximum gross weight
r_d	0.27	[m]	dynamic wheel radius
i_{SBI}	3.909	-	first gear ratio
i_{SBII}	2.158	-	second gear ratio
i_{SBIII}	1.345	-	third gear ratio
i_{PG}	3.438	-	final gear ratio
i_{SR}	1	-	transfer box ratio
f_t	0.012	-	rolling resistance coefficient
c_x	0.33	-	drag coefficient
A	2.19	[m^2]	vehicle frontal area
ρ_P	0.8201	[kg/dm^3]	fuel density

Basic assumptions referring to the selection of values for vehicle data [3, 11, 12] were as follows:

- vehicle was loaded to its maximum gross weight (m_c); in order to make a more reliable comparison, weight differences resulting from the construction of vehicles with different types of drive system were not taken into consideration,
- dynamic wheel radius resulted from the size of tyre (inflated to the pressure specified by its manufacturer) and rim, taking into consideration dynamic loads during motion,
- the same values of gearbox and final gear ratios for different types of drive system were adopted in order to ensure there were no differences in fuel consumption resulting from application of different ratio values,
- the value of transfer box ratio (4x4 drive system) was selected so that the above-mentioned objective was preserved,
- vehicles moved on a smooth asphalt surface (value of rolling resistance coefficient f_t),
- drag coefficient was the same for all vehicle types,
- vehicle width and height defined its frontal area and it was the same for vehicles with different drive systems,
- fuel density (Diesel oil) was adopted for normal conditions.

The value of propulsion system efficiency was adopted from literature [3, 11] as a ratio of mean efficiencies of lower gears of all drive system subassemblies:

a) classical drive system:

Tab. 3. Efficiency of classical drive system

Gear	η_{SP}	η_{SB}	η_{WN}	η_{PG}	η_{UN}
I, II, III	0.998	0.96	0.992	0.96	0.91

b) block drive system:

Tab. 4. Efficiency of block drive system

Gear	η_{SP}	η_{SB}	η_{PG}	η_{UN}
I, II, III	0.998	0.96	0.96	0.92

c) 4x4 drive system:

Tab. 5. Efficiency of 4x4 drive system

Gear	η_{SP}	η_{SB}	η_{SR}	η_{WN1}	η_{WN2}	η_{PG}	η_{UN} *
I, II, III	0.998	0.96	0.93	0.992	0.992	0.96	0.84

** – there are three propeller shafts in this drive system but power transmission onto a given vehicle axle occurs when using two propeller shafts (thus only two propeller shafts were taken into consideration)*

where:
η_{SP} – efficiency of friction clutch,
η_{SB} – efficiency of gearbox,
η_{SR} – efficiency of transfer box,
η_{WN} – efficiency of propeller shaft,
η_{PG} – efficiency of final drive,
η_{UN} – efficiency of propulsion system.

In order to extrapolate instantaneous fuel consumption, basic vehicle resistance to motion needed to be determined, i.e. rolling resistance and air resistance. The former was determined in the following manner [3, 11]:

$$F_T = f \cdot m_C \cdot g \,, \tag{5}$$

where:
F_T – rolling resistance [N],
f – rolling resistance coefficient,
m_C – maximum gross vehicle weight [kg],
g – gravitational acceleration = 9.81 m/s^2.

Another type of forces acting in the direction of travel but having an opposite sense was air resistance (drag). After bringing the air density to normal conditions (1.16 kg/m^3), aerodynamic drag was represented by the following equation [3, 11]:

$$F_P = c_x \cdot A \cdot q = c_x \cdot A \cdot \frac{\rho \cdot v^2}{2} = 0.58 \cdot c_x \cdot A \cdot v^2 \,, \tag{6}$$

where:
F_P – air resistance (drag) [N],
c_x – drag coefficient,
A – vehicle frontal area [m^2],
q – dynamic pressure [N/m^2],
ρ – air density [kg/m^3],
v – relative vehicle and air (wind) speed [m/s].

The above-mentioned resistance to motion was basic elements when determining the established driving conditions being represented by constant vehicle velocities. This corresponded to particular instantaneous fuel consumption being determined in the next section.

5. Simulation instantaneous fuel consumption at constant velocity

The velocity profile and the load value defined the energy intensity of vehicle motion, which needed to be associated with particular fuel consumption being represented by the relationship presented below:

$$\beta = \frac{b \cdot P^d}{3600 \cdot \rho_P}, \tag{7}$$

where:

β – instantaneous fuel consumption [mdm^3/s],

b – specific fuel consumption [g/kWh],

P^d – engine effective power [kW],

ρ_P – fuel density [kg/dm^3].

When entering into equation (7) the efficiency of propulsion system in accordance with the formula below:

$$\eta_{UN} = \frac{P_K}{P^d}, \tag{8}$$

where:

η_{UN} – efficiency of propulsion system,

P_K – power on wheels [kW],

the following relationship was obtained:

$$\beta = \frac{b \cdot P_K}{3600 \cdot \rho_P \cdot \eta_{UN}}. \tag{9}$$

In the case of steady states, the power on vehicle wheels was represented by the following equation:

$$P_K = P_T + P_P = \frac{F_T \cdot v + F_P \cdot v}{1000}, \tag{10}$$

where:

P_T – power needed to overcome rolling resistance [kW],

P_P – power needed to overcome drag [kW],

F_T – rolling resistance [N],

F_P – drag [N],

v – vehicle velocity [m/s].

The motion conditions being defined by vehicle velocity and basic resistance to motion determined the values of engine speed and engine load moment. Equalising engine torque was determined from the relations presented below:

$$T_{tqK} = (F_T + F_P) \cdot r_d, \tag{11}$$

$$T_{tq} \cdot \eta_{UN} \cdot i_{UN} = (F_T + F_P) \cdot r_d \rightarrow T_{tq} = \frac{(F_T + F_P) \cdot r_d}{\eta_{UN} \cdot i_{UN}}, \tag{12}$$

where T_{tqK} – torque on wheels.

When reading the values of specific fuel consumption from the load characteristics for the determined engine torque and using relation (13), instantaneous fuel consumption at constant velocities 15, 32, 35 and 50 km/h was determined for a FIAT Panda vehicle with different types of drive system. Finally, instantaneous fuel consumption for constant velocities was presented by the following formula:

$$\beta = \frac{b \cdot v \cdot (F_T + F_P)}{3600 \cdot 10^3 \cdot \rho_P \cdot \eta_{UN}} \cdot \tag{13}$$

The collected values of instantaneous fuel consumption for the velocities being determined above and three types of drive systems are presented below in Tab. 6, 7 and 8.

Tab. 6. Instantaneous fuel consumption at constant velocity – classical drive (η_{UN} = 0.91)

Gear	v	v	n	T_{tq}	b	F_T+F_P	β
	[km/h]	[m/s]	[min^{-1}]	[Nm]	[g/kWh]	[N]	[mdm^3/s]
I	15	4.2	1980	3.94	788.6	178.57	0.218
II	32	8.9	2332	8.17	713.1	204.41	0.482
II	35	9.7	2590	8.43	696.0	210.88	0.531
III	35	9.7	1590	13.53	735.7	210.88	0.561
III	50	13.9	2271	16.18	634.0	252.15	0.826

Tab. 7. Instantaneous fuel consumption at constant velocity – block front-wheel drive (η_{UN} = 0.92)

Gear	v	v	n	T_{tq}	b	F_T+F_P	β
	[km/h]	[m/s]	[min^{-1}]	[Nm]	[g/kWh]	[N]	[mdm^3/s]
I	15	4.2	1980	3.90	789.1	178.57	0.216
II	32	8.9	2332	8.09	714.1	204.41	0.478
II	35	9.7	2590	8.34	697.0	210.88	0.526
III	35	9.7	1590	13.38	737.5	210.88	0.557
III	50	13.9	2271	16.00	635.8	252.15	0.820

Tab. 8. Instantaneous fuel consumption at constant velocity – 4x4 drive (η_{UN} = 0.84)

Gear	v	v	n	T_{tq}	b	F_T+F_P	β
	[km/h]	[m/s]	[min^{-1}]	[Nm]	[g/kWh]	[N]	[mdm^3/s]
I	15	4.2	1980	4.27	784.8	178.57	0.235
II	32	8.9	2332	8.86	705.5	204.41	0.517
II	35	9.7	2590	9.14	688.6	210.88	0.569
III	35	9.7	1590	14.66	722.7	210.88	0.597
III	50	13.9	2271	17.53	620.0	252.15	0.876

where: v – vehicle velocity, n – rotational speed of engine crankshaft, T_{tq} – engine torque, F_T+F_P – basic resistance to motion, β – instantaneous fuel consumption, η_{UN} – efficiency of propulsion system

6. Conclusions

Instantaneous fuel consumption for different drive transmission systems depends on the overall efficiency of propulsion system. It defines the rate of engine load, which is important for its overall efficiency being expressed in the form of specific fuel consumption. This explains why for selected vehicle velocities the block drive system was characterised by the lowest fuel consumption. Due to additional drive assemblies, the 4x4 drive transmission system was characterised by the highest fuel consumption when compared to classical design and block drive system.

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