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Predicting vehicle waste heat recovery potential in road gradient driving cycle - A case study

V Sandu[1], V B Ungureanu[1], C Leahu[1] and A Mazilu[1,*]

[1]Mechanical Engineering Faculty, Transylvania University, Brasov, Romania

[*]E-mail: adrian.mazilu@unitbv.ro

Abstract. A commercial vehicle was instrumented for assessing the engine operation modes on a mountain route in Romania, in a 70.4 km round trip with mean road gradients +4.2% and -4.5%. The real world test cycle was processed to find the distribution of the most frequent engine operation modes (time intervals versus *n* x *n* speed and load rankings) in order to extract a steady-state gradient road cycle. The direct injection turbocharged diesel engine has been tested on the dynamometric brake in the selected operation modes, investigating the energy recovery potential of the exhaust system. An exergy analysis has been performed by processing the variables of the exhaust gas flow rates for each operation mode (gas composition, pressure and temperature) which have been weighted with the time share of each mode in the driving cycle. The exergy exhaust potential has been evaluated by means of several indicators such as mean weighted exergy rate, weighted exergy to energy ratio and weighted exergy to fuel exergy ratio. By comparing the results to local urban and extra-urban cycles from previous works, it has been revealed that the gradient road cycle has a higher energy potential than the urban cycle, but a lower one than the extra-urban cycle.

1. Introduction

Nowadays, the environmental concerns strongly affect the evolution of internal combustion engines (ICE) which are continuously forced to reduce both fuel consumption and emissions. The low engine efficiencies limited by Carnot principle could be compensated, in a certain extent, by energy recovery technologies. The exhaust gas system appears to have the highest energy harvesting potential, hosting also emission after-treatment devices and silencers, thus producing contradictory demands and competition among their functions; it involves variables such as temperature, pressure and velocity which are related to engine operation parameters.

Among the most promising and investigated waste heat recovery systems (WHRS) are thermoelectric generators (TEG), Organic Rankine Cycle (ORC) power systems, turbo-compounding and heat exchangers [1-4].

For vehicular applications, exhaust gas WHRS should consider the influence of vehicle size, weight and speed, aerodynamic parameters, road characteristics and traffic conditions.

Road gradient is a static property of the infrastructure influencing both vehicle fuel consumption and emissions. It is defined as the change in elevation divided by horizontal travelled distance, usually expressed in percentage.

When a vehicle is travelling on a positive gradient road (uphill), the engine should produce more work to compensate the gravity component which is opposing motion; when it is travelling on a negative gradient road (downhill), gravity accelerates the vehicle in a forward motion; as a consequence, driving uphill will increase the fuel consumption and CO_2 emissions, whereas driving downhill will reduce both.

Some studies have highlighted the sensitivity of vehicle fuel use to road gradient: research work [5] reported a (+/-8%) variation of fuel consumption on road grades of max. +14%/-12% downhill, for a gasoline light duty vehicle; the study [6] confirms for the same type of vehicle the increase of fuel consumption with 15 - 20% when driving on a +6% gradient than in the case of a flat road.

The fuel consumption can be reflected by measured CO_2 emissions; in case of a spark engine passenger car, the higher emissions during uphill operation are not compensated by the lower ones, in downhill operation [7].

For heavy duty vehicles, the road gradient influence is more important due to the higher mass. The fuel consumption and emission functions were studied in Swiss and German roads at gradients (-6%, -4%, 0%, 4%, 6%), in four classes of heavy goods vehicles, being reported in the German Emission Factor Program [8]. By applying regression analysis on a massive experimental data base, both fuel consumption (FC) and emissions (CO, VOC, NO_x, PM) could be predicted by a polynomial function with variable the mean vehicle speed; also, a correction factor providing the influence of road gradient was implemented, reported to the 0% gradient road.

In order to assess the influence of the dissimilar driving conditions upon fuel consumption and emission levels, representative driving cycles were implemented. A driving cycle is a sequential set of speed profiles versus time, issued by certification authorities to evaluate fuel consumption and emission levels, on the vehicle or on the engine, for each one in specific test facilities.

There is no specific driving cycle dedicated for the evaluation of exhaust energy harvesting. Most of the research works use either steady engine operation points or standardized emission cycles; for the latter situation, a detailed methodology has been presented in [9] for a light duty diesel vehicle, assessing the exergy potential equivalent in 8 - 19% fuel savings, when the vehicle has been tested under New European Driving Cycle (NEDC) on a chassis dynamometer.

For the particular case when energy recovery is done using a thermoelectric generator, the literature reports the significant influence of the driving conditions on TEG performance. By using both engine experiments and numerical simulations, the maximum potential of TEG has been evaluated in power output (37.6 W) in NEDC extra-urban sequence for a light duty diesel vehicle [10].

The influence of four regulatory vehicle test cycles, namely, the Federal Test Protocol (FTP-75), the Highway Fuel Economy Test (HWFET), the ARTEMIS (Assessment and Reliability of Transport Emission Models and Inventory Systems) Rural Road (ART-R) and Worldwide harmonized Light vehicles Test Cycle (WLTC) was investigated on a TEG fitted on a diesel passenger car, finding energy gains of 1.54 - 1.68% [11].

For the Japanese 10-15 Mode Cycle, NEDC and Urban Driving Dynamometric Schedule (UDDS) simulations have estimated, for a TEG fitted on a gasoline light vehicle, an improvement of the vehicle performance of 0.52%, 0.78%, and 0.74%, respectively [12].

There is a debate on representativeness of standardized emission cycles versus real-world ones. In real life situations the driving conditions are differentiated from standardized emission cycles in terms of road characteristics, traffic flows and driver behavior.

For example, reference [13] reported TEG energy recovery in a light-duty diesel vehicle, in real-world driving conditions, at different altitudes, with three different fuels; the best ratio of exergy to exhaust gas energy, 35%, was reached in extra-urban sequence, at the highest altitude during testing (2300 m), running with diesel fuel; reference [14] has investigated medium duty diesel truck in two real world cycles, urban and extra-urban, reported exhaust gas exergy potential as fuel savings of 7.2% and 12.6%, respectively.

The investigation of the vehicle waste heat recovery potential during road grade operation is rather limited and dependent on the specific parameters of the WHRS. In reference [12], the behavior of a TEG mounted on a gasoline light vehicle has been predicted as function of road grade at 0 - 9% using a numerical model validated with experimental results; the model has indicated that the higher the road increment, the higher the exhaust temperature and flow rate, thus increasing the harvested power from 69.7 W at 0% to 127.7 W at 9%.

The objective of this study is to analyze the correlation between a real-world road grade driving cycle specific to Romanian routes and the potential of harvesting energy from the exhaust gas of a commercial vehicle diesel engine by means of exergy analysis and to compare it to the potential of urban and extra-urban driving cycles.

2. The characteristics of the real-world cycle

The road network in Romania is classified according to the public transportation law, in five categories (I - Highways, II - Express-ways, III - European and first class national roads, IV-V- Second class national roads, county and rural roads), which correlates the designed road speed with the maximum road grade, as described in table 1.

Table 1. Road technical categories versus designed speed and maximum road grades [15].

Technical category	Designed speed (km/h)	140	120	100	80	60	50-40	30	25
I Highways	Max. road grade (%)	3	4	5	6	-	-	-	-
II-V		-	5		6	6.5	7	7.5	8

Worldwide, the distribution of high gradient roads seems to fit the same tendency observed in table 1. The highways which are designed for high traffic speeds are built with small gradients, while county and rural roads at high altitude areas have higher gradients [16].

Romania`s landscape encloses a large area, 31% mountains and 33% hills, situated at relative high elevation.

The highest peaks are over 2500 m and the highest two roads, Transalpina (2145 m) and Transfagarasan (2042 m) represent strategic roads through the Carpathian Mountains, connecting historical provinces.

An elevation profile of Transfagarasan route indicates a 4.7% mean positive grade and -3.8% negative, as illustrated in figure 1 (the cumulative elevation gain is shown at the top of the image).

Figure 1. Elevation versus road length for Transfagarasan route.

The road on which the measurements have been performed, is situated in the central part of Romania, in Brasov and Prahova counties, between Izvoarele - Cheia - Bratocea Pass. The length of the route is 35.2 km and the elevation profile is envisaged in figure 2.

Figure 2. Elevation versus road length for driving cycle route.

The elevation profile was automatically plotted using the Google Earth application after creating a path through the selection of points across the road representation at approximately 100 m distance between two consecutive points, the average slopes being +4.2% and -4.5%. The traffic on this segment has more than 20% heavy duty vehicles, this artery being an alternative to Ploiesti - Brasov express-way which limits the access of 7.5 tonne commercial vehicles. The route has no speed limits under 35 km/h or roundabouts and there are very few pedestrian crossings, because Cheia village has a ring road. This route was considered to be representative for the road slopes on the hill and mountain areas of Romania as most of the roads belong to class III, as the segment under test.

The vehicle is a 7-tonne commercial truck which operated unloaded during the road test, being powered by a four-stroke, four-cylinder, turbocharged direct-injection diesel engine, model 392-L4-DT (Euro III), with its technical parameters included in table 2.

Table 2. Engine main characteristics.

Engine type	Diesel
Bore x Stroke [mm]	102 x120
Displacement [l]	3.92
Compression ratio	17.5:1
Rated power [kW]	82
Rated speed [rpm]	2700
Max. torque [N·m]	320
Max. torque speed [rpm]	1600-1800

While the vehicle has travelled two round-trips of 70.4 km, the time interval measurement was of 14630 seconds, resulting an average speed of 35 km/h; the weather was dry, during spring time, with a mean atmospheric temperature of 15°C. The vehicle was instrumented to measure engine speed and load with two sensors, mounted on the crankshaft and on the injection pump rack, respectively; a data logger has distributed the engine operation in *n x n* classes, counting the time intervals for each class.

Based on the data logger records, a real world driving cycle can be designed, which can be used for measuring fuel consumption, exhaust gas emissions, component reliability and energy harvesting. The procedure of cycle design which was adopted for the actual road gradient cycle has been described in the previous work of Negrea et al. [17]. For six classes of distribution, the operation modes become n^2 at which the idle mode is added, thus representing 37 modal points. Some time, cost, representativeness

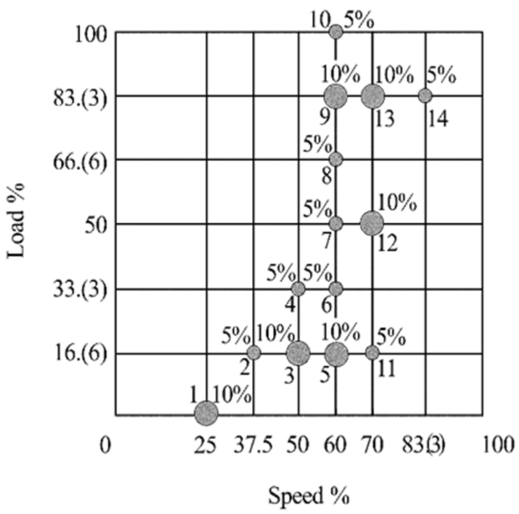

Figure 3. Road grade cycle configuration.

and reproducibility conditions have limited the modal point number. By filtering the modes under 2% of time operation and by redistributing their shares to the closest modes, the modal cycle was limited to 14 steady - state engine points, which are represented in figure 3.

3. Engine testing

During the vehicle operation on the route, it was not possible to measure the specific parameters of the exhaust gas required for the energy and exergy analysis such as fuel and air flow rates, exhaust temperature and pressure. It was decided to simulate the cycle by measuring the engine parameters on the 220 kW eddy-current dynamometric test bench at the Road Vehicle Institute Brasov-RO, according to the cycle composition represented in figure 3. The engine performance has been corrected according to the net power requirements of the standard ISO 1585, taking into account the influence of atmospheric pressure and temperature during tests (atmospheric pressure 97.19 kPa, temperature 290 K and relative humidity 40%). The engine set-up during testing is given in figure 4.

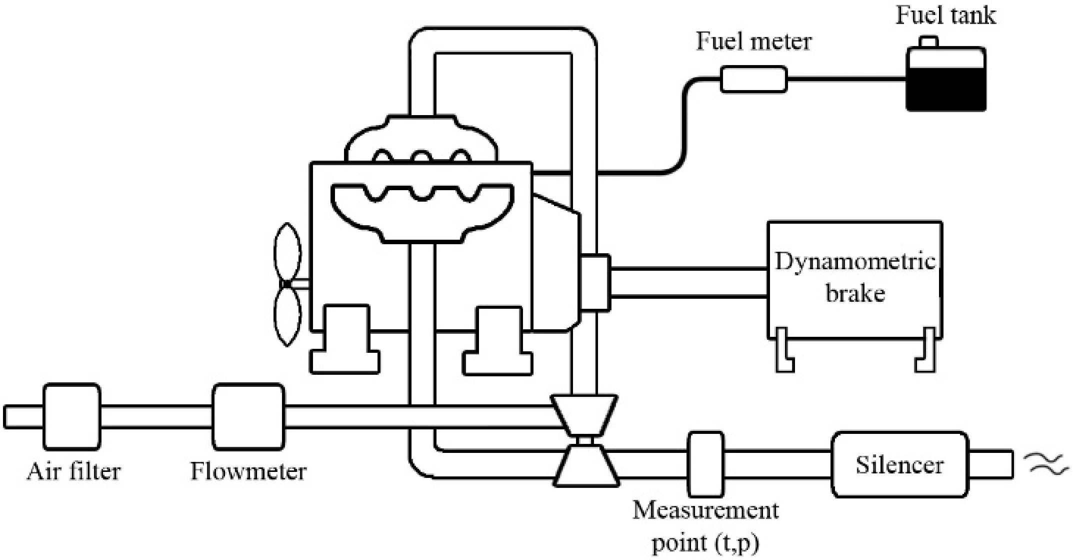

Figure 4. Engine test configuration.

The exhaust gas temperature was measured at 0.5 m downwards the turbocharger exit, with the engine exhaust duct connected to the test bench ventilated exhaust system. The engine was instrumented with K-type temperature sensors (cooling liquid, lubricant and exhaust gas), pressure sensors (lubricant and exhaust gas), laminar flow element flowmeter for air, AVL gravimetric fuel balance and an inductive type tachometer. The engine has been supplied with standard EN 590 diesel fuel and with SAE 15W40 viscosity class lubricant.

4. Numerical road gradient influence on fuel consumption and emissions

To obtain fuel consumption and emission factors for fleet inventories, the European Union has introduced standard vehicle emission models based on vehicle mean velocity, which can be easily obtained from vehicle flow and density. Among them, it is worth mentioning COPERT- Computer Programme to calculate Emissions from Road Transport and MEET - Methodologies for Estimating air pollutant Emissions from Transport. For heavy duty vehicles, COPERT [18] has applied a correction factor or gradient factor, *as*, which could evaluate the contribution of the road gradient reported to flat road; it is defined by a sixth-order polynomial function of the vehicle mean speed, *v,* which can be applied both for fuel consumption, *FC*, and emission (CO, VOC, NO_x, PM) calculations. The subscripts represent *i* - polynomial ranking, *j* - vehicle category, *k* - type of road and $A_{i,j,k}$ - constants for fuel consumption, as function of vehicle weight and gradient class, $FC_{j,k}$ - fuel consumption in g/km for vehicle category *j* driven on the road of type *k*, $FC_{corr,j,k}$ - corrected fuel consumption in g/km for vehicle category *j* driven on the road of type *k*.

$$as_{j,k} = \sum_{i=0}^{6} A_{i,j,k} \cdot v^i \,, \tag{1}$$

$$FC_{corr,j,k} = as_{j,k} \cdot FC_{j,k} \,. \tag{2}$$

The constants $A_{i,j,k}$ are based on data processing of the research work already mentioned in [8] and are also considered identically in other standard vehicle emission models as MEET [19]. For the variable *j*, four classes of vehicles have been considered, according to their unloaded weight (3.5 - 7.5 t, 7.5 - 16 t, 16 - 32 t and 32 - 40 t).

Numerically, for the category of the diesel heavy duty vehicle, with a weight range of 3.5 - 7.5 tonnes and speed range of 0 - 60 km/h, with the correlation coefficient R^2 = 0.99, fuel consumption for 0% gradient is:

$$FC_{j,k} = 1425.2 \cdot v^{-0.7593} \,. \tag{3}$$

Taking into account that the route includes 2 round-trips, the mean value of the gradient road is 4.35%, higher than 4%, which indicates the class of road type *k* and selection of $A_{i,j,k}$ coefficients. The literature indicated four classes of coefficients corresponding to slopes, in percent (0...4), (-4...0), (4...6) and (-6...-4), limiting the mean speed of the vehicle to 13-50 km/h for a fair accuracy of the results. The $A_{i,j,k}$ coefficients became A_i , variable *j* having no longer sense, the gradient factor *as* being limited to (3.5-7.5t) and variable *k* of the road is limited to (4...6) uphill and (-6...-4) downhill, so the values of A_i are presented in table 3.

Table 3. Gradient factor coefficients for heavy duty vehicles [16].

A_6	A_5	A_4	A_3	A_2	A_1	A_0	Slope (%)
0	4.27E-07	-5.74E-05	2.97E-03	-7.42E-02	9.35E-01	-3.03E0	4...6
0	-7.74E-08	1.33E-05	-8.78E-04	2.72E-02	-3.93E-01	2.65E0	-6...-4

The gradient factor, *as,* calculated with equation (1) for the mean speed of the route of 35 km/h with coefficients from table 3, for uphill route is 2.30 and for downhill route 0.46, with a mean value of 1.38. The interpretation of the figures is that by combining the uphill and downhill routes, the fuel consumption increases with 38% reported to the consumption measured on a flat road, confirming the findings from reference [7].

5. Exhaust gas exergy

From the second law of thermodynamics which limits the conversion of heat, as disordered energy, in any ordered form of energy such as mechanic work or electricity, the expectations for energy harvesting from exhaust gas are limited due to the irreversibility of the processes.

The exergy analysis applied in this paper is based on the methodology described in the previous work of Sandu et al [14]. Briefly, the diesel fuel composition was considered equivalent to $C_{14.4}H_{24.9}$ hydrocarbon, with a molar mass of 198 kg/kmol and lower heating value (*LHV*) 43.1 MJ/kg. The stoichiometric air fuel ratio is 14.53 kg air/kg fuel and the combustion products were limited to CO_2, H_2O, N_2 and excess air (i = 1…4). The specific heats at constant pressure have been weighted with the mass fraction g_i and calculated according to temperature variation using a third order polynomial function (j = 0…3), as described in equations (4) and (5):

$$c_{p,g} = \sum_{i=1}^{4} g_i \cdot c_{p,i} \,, \tag{4}$$

$$c_{p,i} = \sum_{j=0}^{3} a_j \cdot T^j \, . \tag{5}$$

Neglecting the minor contribution of specific entropy and potential energy in gravitational field, the exergy rate of exhaust gas in each operation mode, $\dot{E}_g$, in W, was calculated based on equation (6):

$$\dot{E}_g = \dot{m}_g \left[c_{p,g}\left(T_g - T_0\right) - T_0\left(c_{p,g} \ln\frac{T_g}{T_0} - R_g \ln\frac{p_g}{p_0} \right) \right], \tag{6}$$

with $\dot{m}_g$ - mass flow rate of exhaust gas, in kg/s; T_g - absolute temperature of the gas, in K; R_g - gas mixture constant, in J/(kg K); p_g - absolute gas pressure, in kPa; subscript 0 - the reference state. Exhaust gas mass flow rate was calculated as the sum of measured air and fuel mass flow rates.

The share of each steady state mode was introduced in euation (7) by means of w_k coefficient - weight of the steady operation mode k in the steady-state test cycle to find the weighted mean exergy flow rate:

$$\dot{E}_{wm} = \sum_{k=1}^{n} w_k \cdot \dot{E}_k \, . \tag{7}$$

6. Interpretation of results

The exhaust gas exergy rate strongly depends on temperature and mass flow rate. There is a quality - quantity relation among these measures.

While mass flow rate varies approximately proportional with engine speed, the temperature level is influenced by speed and load [20]. At constant load, exhaust gas temperature increases with engine speed. At a small constant load, equivalent to 16.6% of rated load, the gas temperature increase is illustrated in figure 5. At a higher load, equivalent to 83.3% of rated load, the temperature level rises significantly, keeping the tendency of growth with increasing engine speed.

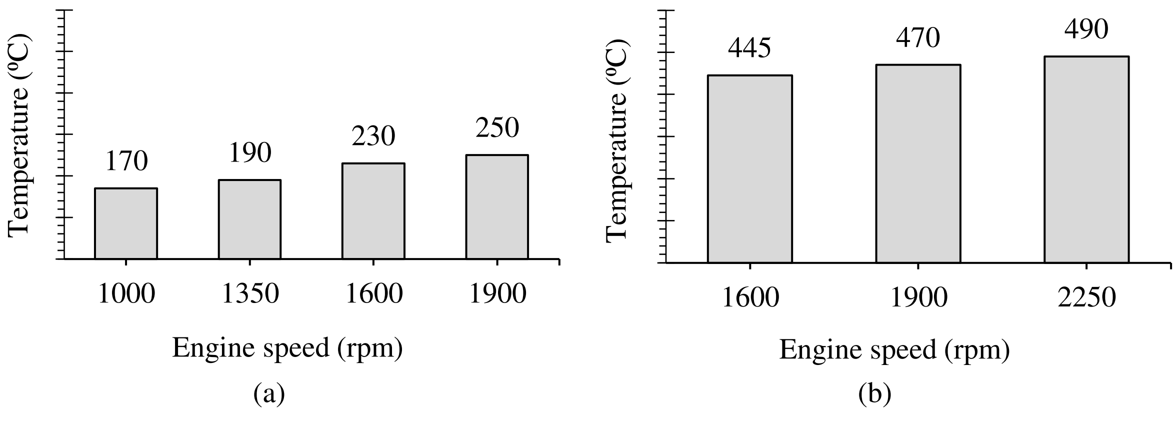

Figure 5. Exhaust gas temperature versus engine speed at constant load: a - low load, b - high load.

At constant speed, exhaust gas temperature increases with engine load; for the maximum torque speed, 1600 rpm, the temperature reaches the maximum value, 550°C, at full load, as shown in figure 6. In terms of relative temperature increase, it may be considered that the influence of load is greater than that of engine speed.

The exergy and exhaust gas energy rate are represented in figure 7 for 14 modal points. The greatest loss of energy through the exhaust gas is met for point 14, at the highest speed of the cycle (5/6 of rated speed, 5/6 of rated load), followed by point 10, at 1600 rpm and 100% load; in this situation, the highest temperature of point 10 does not correspond to the maximum exhaust gas heat, showing that the influence of the higher mass flow rate is decisive.

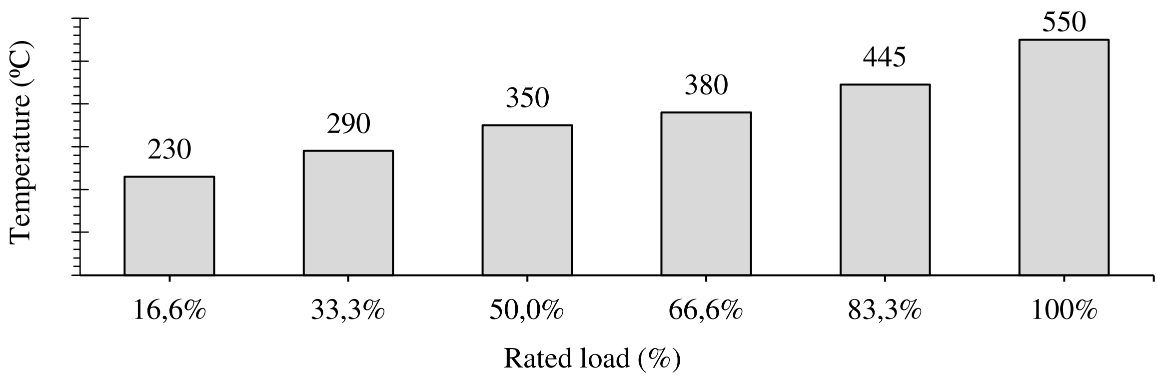

Figure 6. Exhaust gas temperature versus engine load at constant speed.

For the exergy rate, the highest potential belongs also to point 14, even in relative units, whereas in point 10, 43% of exhaust heat can be recovered, which is more than 41%, corresponding to point 14.

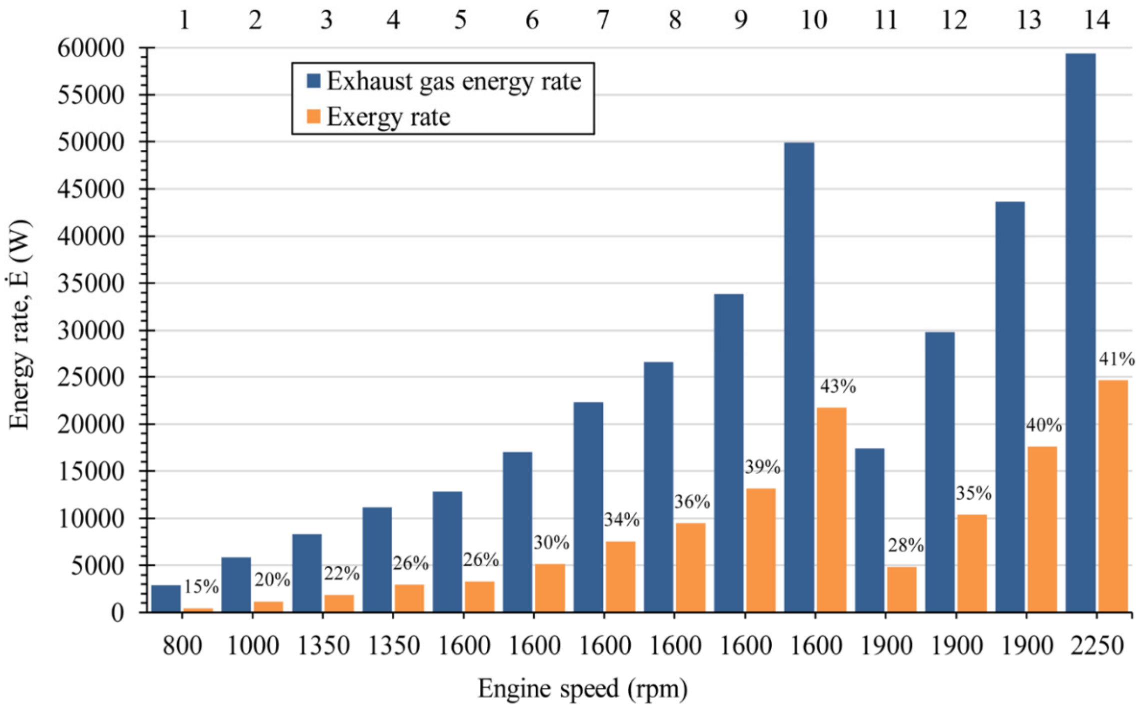

Figure 7. Exergy rates in road grade cycle.

The exergy rate is improved significantly at increasing load and speed, with the same tendency of a higher influence for load, as reported in [21].

At the constant speed of 1600 rpm, the increase of load significantly improves the exergy rates from 26% (16.6% load) to 43% (100% load).

A comparison of the energy recovery of the cycle could be made with two other simulated cycles which have been reported in previous work of Sandu et al. [14], for urban and extra-urban conditions, on the same tested engine.

The analysis of the exhaust gas temperatures is illustrated in figure 8, in terms of mean and maximum values. The mean temperatures were calculated with a weighted formula, similar to equation (7).

For each cycle, the mean weighted exergy rate was calculated, by means of equation (7). The results presented in figure 9 ordinate the maximum potential of energy recovery as function of driving cycles, the extreme cycles being the urban and extra urban ones. This hierarchy is also kept for mean weighted temperature, as illustrated in figure 8.

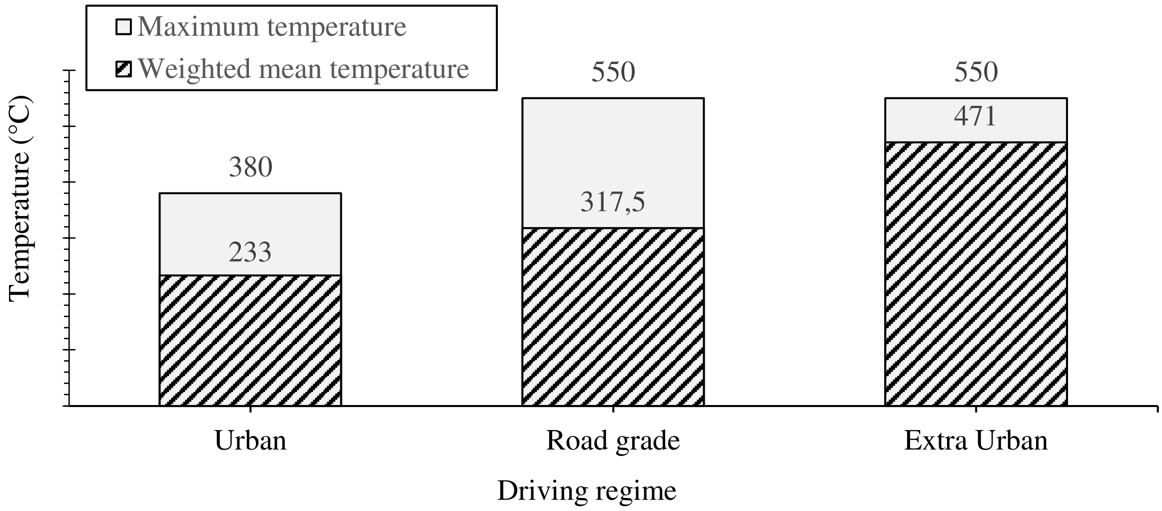

Figure 8. Maximum and mean exhaust gas temperatures versus driving cycle.

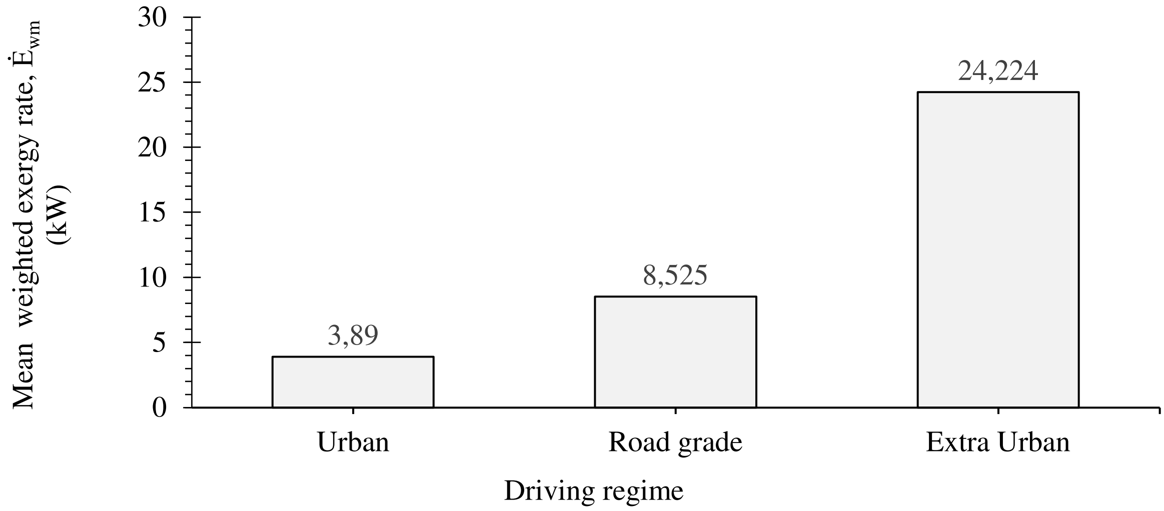

Figure 9. Mean weighted exergy rates.

Two approaches have been selected for the present study to express the exergetic efficiency. The first approach is applied locally to the exhaust manifold, being the maximum extractable work available from exhaust gases, as defined by Alkidas in [22]; according to this definition, it represents the ratio of exergy to exhaust gas thermal energy [23], which has been expressed by the parameter δ, identified in equation (8):

$$\delta = \frac{\dot{E}_g}{\dot{m}_g c_{p,g} \left(T_g - T_0\right)} . \tag{8}$$

This ratio has also been adopted in other works, for example reference [9], such as the exergy to energy ratio further defined as the fraction of exhaust gas energy which can be transformed into an ordered form of energy such as mechanical work or electricity in an ideal energy recovery device.

The second approach is applied to the whole engine and to the energy within the fuel, revealing how effectively the available fuel energy could be converted into work [24]; the definition expresses how much energy is harvested from the fuel chemical exergy as a result of energy destruction and loss. This form of efficiency is expressed as the ratio of exhaust gas exergy to fuel (chemical) exergy, ε, defined by equation (9):

$$\varepsilon = \frac{\dot{E}_g}{\dot{m}_f a_f}. \tag{9}$$

In equation (9) $\dot{m}_f$ represents fuel mass flow rate and a_f the fuel chemical exergy. For diesel fuel, the chemical exergy can be approximated from the hydrocarbon based formula [25]:

$$a_f = LHV\left(1.04224 + 0.011925\frac{y}{z} - \frac{0.042}{z}\right), \tag{10}$$

with z, y - number of carbon, respectively, hydrogen atoms. For y =14.4 and z = 24.9, it yields for diesel fuel $a_f = LHV \cdot 1.0599$.

For the whole cycle, the parameters δ and ε were averaged with the factor w representing the weight of the operation mode k in the cycle, resulting δ_{wm} and ε_{wm} with a weighting formula similar to equation (7). According to figure 10, the road gradient cycle has an intermediate capacity for energy harvesting compared to urban and extra-urban cycle.

The composition of the road gradient cycle is closer to urban operation due to the lower speeds, but it is operated at higher loads. The weighted mean value of exhaust efficiency δ_{wm} is 30.6% and the ideal energy saving expressed in fuel availability is 8.5%.

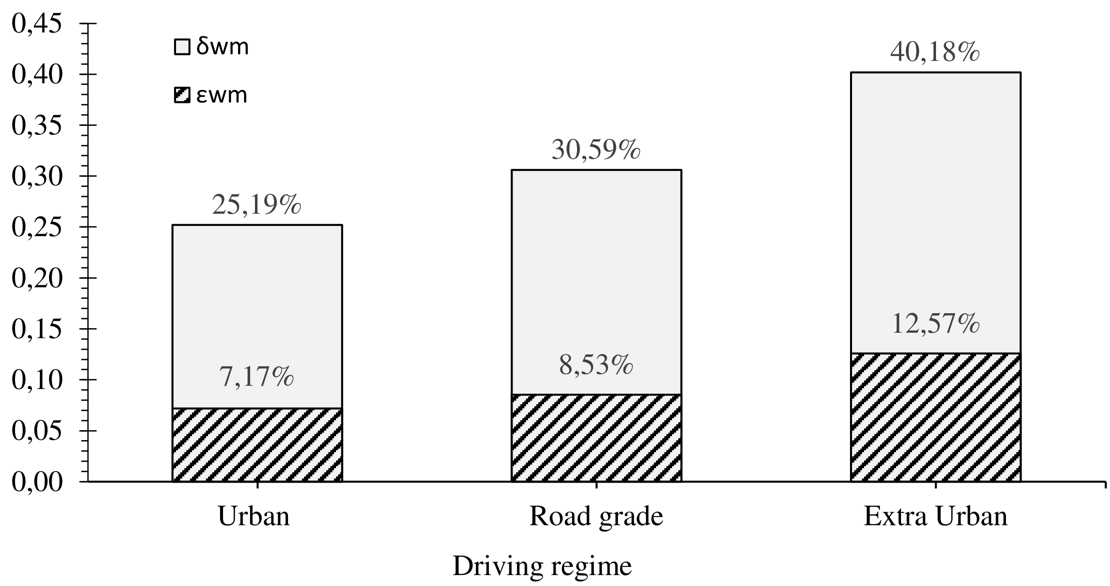

Figure 10. Weighted exergy to energy ratio, δ, and exergy to fuel exergy, ε.

Both exergy to energy ratio, δ, and exergy to fuel exergy ratio, ε, depend on the position along the exhaust duct; their values, indicated in figure 10, appear to be very high compared to the energy harvesting efficiency of real devices. The selection of the temperature measurement point on the exhaust gas duct, closer to the turbocharger exit, has not considered the anti-pollution devices which usually require a high level of exhaust gas temperatures to work efficiently. If the measurement point on the exhaust duct had been chosen downwards of the antipollution device, the exergy rate would have decreased drastically.

Among the strengths of the case study are the investigation of the vehicle operation in real-world traffic conditions and the evaluation of the relationship between road gradient and commercial truck engine operation modes in Romania, which, to the knowledge of the authors, have not been published in literature. Also, the comparison of the urban, road gradient and extra-urban cycles were based on the real world measurements of the same vehicle and engine type. Some limitations of the study are related to the reduced test time, limited mean road gradient and a single vehicle type. Even though the engine speed and load have been measured continuously during the road test, the engine has been operated both in steady-state and transient modes; a source of errors consists of the exhaust temperature and pressure measurements being collected on the engine test bench instead of on-board measurements.

7. Conclusions

A 7-tonne diesel truck driven on a mountain route, with a relatively high average slope (+4.2%, -4.5%), has been equipped for identifying the most frequent engine operation modes and further analyses were made to evaluate the potential of exhaust gas energy harvesting. The fuel consumption on the gradient road has been assessed to have a 38% increase compared to a flat road.

The processed road gradient cycle has a relatively symmetrical distribution around the engine speed at the maximum torque, with dominant shares at medium and high loads.

The exhaust energy harvesting potential in road gradient cycle has been expressed in absolute exergy rates and in two exergetic efficiencies - exergy to thermal energy and exergy to fuel exergy; in absolute units, 8.5 kW can be converted into useful work, representing more than 30% of exhaust gas thermal energy and more than 8% of fuel exergy.

For particular conditions of the presented case study, it may be predicted that the exhaust energy harvesting potential in road gradient cycle is greater than in urban operation, but smaller than in extra-urban operation.

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