Experimental determination of general characteristic of internal combustion engine using mobile test bench connected via Power Take-Off unit

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Abstract. The general characteristics of the engine include information about the regions of its operating area that are most efficient, where specific fuel consumption reaches the smallest values. Economic operation based on those characteristics can contribute to a significant reduction of fuel consumption and consequently less pollutant emissions and lower costs. The paper presents an experimental method of determination of general characteristic of internal combustion engine mounted in a truck chassis. Experimental tests focus on low torque and engine speed range, which is observed during operation of the body Refuse Collection Vehicle. To conduct the measurements, the specially designed test bench was used. The collected data from two engines was processed applying machine learning methods, which were described and compared to obtain best regression model. Determined general characteristic was used to evaluate fuel consumption in a time period of variable engine load. Finally, simulated total fuel used for both engines in different engine speeds and load were compared. Results show significant differences in total fuel consumption depend on engine speed, thus it is an evidence that there is a large field to improvement in propulsion system configuration.

1. Introduction
The general characteristics of the engine include information about the regions of its operating area that are most efficient, where specific fuel consumption reaches the smallest values. Economic operation based on those characteristics can contribute to a significant reduction of fuel consumption and consequently lower costs and less pollutant emissions. Similarly, work related to the improvement of fuel supply systems [1, 2] and minimization of the tire rolling resistance of vehicle [3, 4] can bring a positive economic and ecological effect. Many studies shows that the driver of a vehicle has a main impact on the fuel consumption during regular operation [5,6,7,8,9]. However, in vehicles equipped with power take-off unit (PTO), such as the Refuse Collection Vehicle (RCV), fuel consumption is determined by the characteristics of the driven machine. It has been shown in paper [10] that an appropriate configuration of the drive system based on the general characteristics of the engine can reduce the fuel consumption of the utility vehicle by up to 35%. However, in this study information about the engine was used from external reference in which the engine was tested in the whole power range. In case of PTO performance, maximum torque is never obtained and produced power is several times lower than maximal. Moreover, the information provided by the engine manufacturers contains
mostly external characteristics, which have little value in drive system configuration process. Thus, there is a need to determine detailed general characteristics in low range of torque and speed, to allow us to find most efficient, environmental friendly and cost-effective drive system configuration. Due to the fact, that reconstruction of characteristic using real-world data is rather difficult [11], there is a need of method to determine it during controlled tests, but by low costs and on-site at the factory. Once the general characteristic is obtained, then it is possible to simulate different loads on PTO propulsion system and evaluate fuel consumption.

Refuse collection (collection, compaction, transportation) is a process, in which not much research on fuel economy and energy consumption was made. Consequently, Refuse Collection Vehicle (RCV) are at the early stage of optimization in comparison to other automotive branches. Thus, as a reference for future work and research direction it is beneficial to make use of e.g. construction equipment literature. In work [12] a mathematical model of hydraulic excavator was presented, which includes combustion engine characteristic reconstruction. In [13] a wheel loader and its working cycles were analysed, using i.a. engine general characteristic. Similar research needs to be done on RCV. A mathematical model of refuse collection process would allow to evaluate efficiency and improvement direction. This paper is one of the first steps in detailed examination of RCV energy consumption and fuel economy.

2. Determination of the general characteristics of the engine
The most popular way to determine general characteristics is to apply constant load at constant speed in many operating points and measure the fuel consumption. In this approach it is important to cover the whole desired engine operating range, otherwise the most efficient areas on the map may be omitted/miscalculated by approximation function. Moreover, measuring points grip should be uniform, because too much concentration in one area has high influence on cost function used in approximation algorithm.

Using a chassis dynamometer can be problematic and costly for utility vehicles’ manufacturers. In terms of costs, it requires chassis transportation to the nearest chassis dynamometer, or building own facility. In terms of parameters analysis, the torque measured on the rollers is different to the torque on PTO, due to the friction loses in the drivetrain and on the wheels. Finding accurate method to correlate these two torque values may be problematic and time-consuming.

Thus, a special test bench was designed and manufactured. Papers [15, 16] contain its design principles. Test bench concept is based on load application using PTO as controllable engine load generator. Pomp attached to the PTO drives the hydraulic system, in which the load is completely adjustable. Engine’s power determined by multiplication of its torque and rotational speed is transmitted to the hydraulic system, whose power is defined by a multiplication of oil flow Q and pressure p:

$$\eta \cdot M \cdot \omega = Q \cdot p$$

where $\eta$ – pump efficiency. Using test bench, it is possible to obtain an arbitrary dense grid of measuring points in the range of PTO maximum power.

To perform general characteristics determination two chassis were chosen, Scania P320 and MAN TGS28.320. These are very popular chassis in RCV production and were consider as a good reference for tests. Chassis details are shown in Table 1. Test bench mounted on Scania is shown in Figure 1.

Table 1. Tested chassis parameters

| Chassis     | Scania P320 | MAN TGS 28.320 |
|-------------|-------------|----------------|
| Engine      | 9.3 liter disp. | 10.5 liter disp. |
| $P_{max}$   | 235 kW / 320 KM | 235 kW / 320 KM |
| $T_{max}$   | 1600 Nm     | 1600 Nm       |
| $T_{PTO, max}$ | 600         | 570            |
According to the Polish Standard [17], to conclude the engine is operating in the static state acceptable deviations of the measured engine parameters are defined in Table 2.

Table 2. Maximal deviation of engine parameters according to Polish Standard [12]

| Parameter          | Permissible deviation |
|--------------------|-----------------------|
| Torque             | ±2%                   |
| Engine speed       | ±2%                   |
| Fuel consumption   | ±3%                   |

However, the standard does not define the time, in which the deviations must be kept lower than specified. Thus, during the tests hydraulic system was set on particular constant load and speed for each point for 20 s. Based on performed experiments, it can be concluded that in most cases 15 s is enough for both engines to set in quasi-static state (at least complying Polish Standard), so for the analysis only points from the last 5 s of the quasi-static state were considered. Figure 2 shows histograms of uniform grid of measuring points in available range of speed and torque.

3. General characteristic approximation function

Static characteristic of the engine is given as a vector function by equation [14]:

$$Y_S = f(M, n); (M, n) \in L$$

(2)

where $L$ is a range of engine operational points. For the purpose of this study, $L$ is considered as all operational point possible to obtain using PTO. Range boundaries are described later in the text.

To determine engine characteristic a linear regression model was used as n-degree polynomial function. As engine characteristic is considered as the function of two variables, the result of approximation is a surface. An example of 2nd degree polynomial surface represents equation (3).

$$f(x, y) = w_0 + w_1 x + w_2 y + w_3 xy + w_4 x^2 + w_5 y^2$$

(3)
where \( w_0, w_1, \ldots, w_5 \) are polynomial coefficients calculated using the least squares method by searching for a minimum of cost functions (4).

\[
Min = \sum_{p=1}^{P_{max}} \left[ Z_p(x, y) - z_p(x, y) \right]^2
\]

(4)

where \( Z_p(x,y) \) – value measured in the point \( \{x,y\} \), \( z_p(x,y) \) – value approximated in the point \( \{x,y\} \), \( P_{max} \) - number of point in approximated range. In this analysis variables \( x \), \( y \) and \( z \) correspond to the hourly fuel consumption \( B \), the torque \( M \) and the rotational speed \( n \) respectively.

![Figure 2. Histogram of number of measuring points in static general characteristic tests for a) MAN b) Scania](image)

According to statistical learning methodology [18, 19] the experimental data was scaled to values 0-1 and then in search of optimal polynomial degree cross-validation was performed. It is widely used technique for assessing how the results of a statistical analysis will generalize to an independent data set. In other words, it shows if regression model overfits the data and how sensitive the prediction is to choosing different train set. As a scoring method two parameters were calculated: \( r^2 \) and mean squared error (MSE) [19].

The data was split in 50 folds (49 of them considered as trainset and 1 as test set). Then, for obtained model \( r^2 \) and MSE were calculated. This calculation was performed 50 times, each time using different fold as test set. An example of one train/test split was shown in the Figure 4. Results of model cross-validation are shown in Figure 3 as boxplots. They confirm the assumption, that there is no benefit in increasing polynomial degree to more than 4. Above this value overfitting is apparent, as MSE rises and \( r^2 \) remains constant or decreases. What is more, higher polynomial degree creates artificial extremities clearly visible just beyond the range of measuring points (Figure 5). Therefore, in approximation 4\textsuperscript{th} degree polynomial regression model was used.
4. Specific fuel consumption map evaluation

Results of approximation for both engines are shown in Figure 6 and Figure 7. As scaling the data was needed only to evaluate the regression function, in further analysis data was rescaled to real values, and then evaluation of specific fuel consumption was done, according to the formula:

$$g_e = \frac{G_e}{M \cdot n}$$

where \(g_e\) – specific fuel consumption, \(G_e\) - fuel rate.

Figure 3. Cross-validation results: a) Mean Squared Error b) \(r^2\)

Figure 4. Scania dataset split: train set and test set (red)

Figure 5. Artificial extremity beyond measuring points for 5th degree polynomial
Results of this calculation are shown in Figure 8 and Figure 9, as well as constant power lines (dashed). Maximum power, which is received by hydraulic system via PTO is usually no more than 45 kW. The lower the $n$ is, the higher torque should be, so it must be pointed out, that there are construction limitation in setting minimal $n$. Maximal torque for PTO and maximal torque for hydraulic pomp usually is no higher than 600 Nm. Thus, considering 45 kW power, the lowest possible $n$ value is about 700 rpm.

5. **Total fuel consumption prediction**

General engine characteristic can be used to evaluate fuel consumption in particular operational point. Next step is to consider set of points in time which corresponds to real load, changing in time. Using regression model it is possible to calculate total fuel consumption for arbitrary load in time, according to the formula:

$$T_{fuel\ used} = \sum_{i=1}^{k} (G_i \cdot \Delta t)$$  \hspace{1cm} (6)
where \( G_{ei} \) – fuel rate in \( i\)-th operational point, \( \Delta t \)-time step.

Hydraulic systems in refuse collection vehicles in most cases are equipped with fixed displacement pump, which is attached to PTO. To obtain constant oil flow in the system, engine rotational speed is set on constant value, while the PTO torque changes due to variable pressure in hydraulic cylinders. Explanation of RCV operation principles [8, 20] is not the aim this study, nevertheless it must be mentioned that its loading cycles may be very different. Based on measurement made on real RCV during regular operation, 4 different loads were specified, shown in Figure 10 and described in Table 3. They have different duration time, the factor which has high impact on total fuel consumption. They have also different average power, which means that the engine works in different operational point most of the time. Thus, specified loads are good reference to evaluate trends in fuel consumption of any load applied to the engine by PTO.

![Figure 10. Different engine load measured on Refuse Collection Vehicle during operation](image)

**Table 3.** Four specified loads based on real operation measurements

| Load  | Duration | Mean power | Description                  |
|-------|----------|------------|------------------------------|
| Load 1| 54 s     | 7.0 kW     | Waiting, bins lifting, compaction |
| Load 2| 80 s     | 10.9 kW    | Waiting, bins lifting, compaction |
| Load 3| 28 s     | 4.34 kW    | Waiting, bins lifting        |
| Load 4| 17 s     | 22.3 kW    | Bins lifting, compaction     |

For each speed \( n \) in range (700-1200 rpm, with step 50 rpm) torque values were calculated. Since the idling speed is 600 and requirement mentioned in paragraph 4, 700 rpm was assumed as minimum value to power the hydraulic system. Upper limit is set to 1200 rpm due to high noise emission, which is undesirable especially in urban areas. Using regression models of Scania and MAN engines, total fuel consumption was calculated for each load and each speed. Results are shown in Figure 11 and Table 4.

Considering MAN engine the lower speed is set, the lower total fuel consumption is obtained. It is valid for all considered loads. The maximum differences in fuel consumption vary from 23% to even 80% in extreme scenario (Figure 12). However, in case of rapid pressure increase in hydraulic system, it is possible for low \( n \) values to stop the engine due to high load peak. This fact should be taken into account, while choosing optimal \( n \) value.
In the contrary, for Scania engine optimal engine speed is 900 rpm (for load 1-3) and 750 rpm (for load 4). The maximum differences vary from 21% to even 60% (Figure 12). For load 1-3 minimum fuel consumption is obtained for $n = 900$ rpm and 950 rpm, whereas for load 4 $n = 750$ rpm. Anyway, it is clear that setting $n$ in range between 750-950 rpm is the most economical.

Simulation results allows to compare two engines, which can be used to power the same RCV. They may be characterized by similar parameters (equal max power and torque), but fuel used to operate RCV hydraulic system may significantly vary. For load 2, even if $n$ is set likewise, the difference can reach up to 44%. This fact exposes the need of further research on RCV energy efficiency and RCV-chassis configuration method. Apart from customer preferences in choosing chassis manufacturer, fuel economy and thus environment impact should be taken into account.

Table 4. Total fuel consumption for 4 specified loads based on real operation measurements

| $n$ [rpm] | Load 1  | Load 2  | Load 3  | Load 4  | Load 1  | Load 2  | Load 3  | Load 4  |
|-----------|---------|---------|---------|---------|---------|---------|---------|---------|
| 700       | 54.8    | 98.6    | 23.8    | 31.6    | 54.6    | 101.0   | 23.4    | 33.7    |
| 750       | 50.4    | 93.6    | 21.3    | 31.4    | 57.4    | 104.9   | 25.0    | 34.2    |
| 800       | 47.5    | 90.6    | 19.6    | 31.6    | 60.5    | 109.2   | 26.7    | 34.8    |
| 850       | 45.7    | 89.2    | 18.5    | 32.0    | 63.9    | 113.8   | 28.6    | 35.6    |
| 900       | 44.8    | 88.9    | 17.9    | 32.5    | 67.6    | 118.9   | 30.6    | 36.4    |
| 950       | 44.8    | 89.6    | 17.8    | 33.0    | 71.6    | 124.4   | 32.7    | 37.3    |
| 1000      | 45.8    | 91.4    | 18.3    | 33.6    | 75.7    | 130.2   | 34.9    | 38.2    |
| 1050      | 47.9    | 94.6    | 19.4    | 34.3    | 79.8    | 136.0   | 37.2    | 39.2    |
| 1100      | 51.4    | 99.7    | 21.3    | 35.2    | 83.8    | 141.5   | 39.3    | 40.2    |
| 1150      | 56.9    | 107.2   | 24.2    | 36.4    | 87.4    | 146.6   | 41.2    | 41.0    |
| 1200      | 64.8    | 118.1   | 28.5    | 38.2    | 90.4    | 150.6   | 42.8    | 41.7    |

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Figure 11. Total fuel used in simulation for 4 loads dependent on engine speed

Figure 12. Difference in fuel used dependent on engine speed
Due to many exploitation and manufacturing reasons, it is not always possible to set optimal engine speed for RCV operation. However, as a useful tool to comparing fuel consumption for engine speeds in different loads, a percentage difference map was drawn (Figure 13). On the X axis currently set speed is shown, and on the Y axis is an alternative speed. Plot shows how many percent total fuel consumption will increase or decrease, if speed from axis X is changed to speed from axis Y. For example, for load 1 and Scania engine, while $n$ is set on 1150 rpm by default (x axis), if $n$ is changed to 800 rpm (y axis), then the fuel consumption will change by -20%.

![Figure 13. Map of % difference between fuel used in given load depend on set $n$ (x axis) and alternative $n$ (y axis)](image)

6. Summary
Presented method of engine general characteristic determination appears as a useful tool for fuel consumption predicting for vehicles equipped with PTO propulsion system. Based on experimental measurements two characteristics were determined using numerical methods. Polynomial regression models were calculated and evaluated by Mean Squared Error and $r^2$ in cross-validation, and it was shown that appropriate polynomial degree is 4th. As a result, specific fuel consumption maps were obtained. They can be used to unveil the most efficient operation areas of the engine. Regression models also allow to perform simulations with arbitrary loads to calculated total fuel consumption. In this paper 4 different load were applied to 2 engines for different engine speeds. Results show, that there is high potential in optimizing propulsion system of RCV, due to significant differences in total fuel used. For Scania and MAN engines setting optimal rotational speed can lead up to 8% and 50% respectively, if compared to default RCV manufacturer setting (1000 rpm). Additionally, useful graphs were presented as a tool for assessing potential benefits in changing propulsion system configuration.

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