Design of a test bench for determining the general characteristics of an internal combustion engine using a hydraulic power take-off system

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1 Introduction

The general characteristics of the engine include information about the regions of the engine's 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. While the driver of a utility vehicle has a main impact on the fuel consumption during regular operation [1-5], the power take-off unit (PTO), such as the garbage truck's propulsion system, is determined by the characteristics of the driven machine. It has been shown in the paper [6] that an appropriate configuration of the drive system based on the general characteristics of the engine reduces the fuel consumption of the utility vehicle by up to 35%. Unfortunately, the information provided by the engine manufacturers contains only external characteristics and cannot be used to determine the PTO performance, where the maximum received torque is several times lower then nominal. This introduces the need to determine the general characteristics by measuring fuel consumption in many operating points and approximation of these values using proper mathematical function [7,8].

2 Determination of the general characteristics of the engine

Static characteristic of the engine is given as a vector function by equation [7-11]:

$$Y_S = f(M,n);(M,n) \subset L \tag{1}$$

Where L is a range of all operational point possible to obtain using PTO.

A method of characteristic determination described in [7] is the approximation of the 'Spline' function of the hourly fuel consumption B the torque M domain and the rotational speed n. The function consists of polynomials of N degree linked in nodes j = 1. K. Polynomials have the same degree (N), have the same values and derivatives up to $N-1$. Function parameters are calculated by the least squares method [12] by searching for a minimum of functions:

$$Min = \sum_{p=1}^{p_{\max}} [Z_p(x,y) - z_p(x,y)]^2 \tag{2}$$

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.

Although the distribution of measuring points can be arbitrary from the point of view of approximation, uniform distribution is suggested [9]. In currently produced trucks it is possible to accurately control the engine speed both by the driver (cruise control) and by the power take-off control unit. Moreover, a hydraulic system driven by PTO can generate particular value of resistance torque (in its operating range). These two facts lead to possibility of obtaining an arbitrary dense and uniform grid of measuring points in operating range. It should be pointed that to obtain characteristic of the engine in static state, it is required to operate the engine in particular point long enough to stabilize the operating conditions.

According to the Polish Standard [13] static states can be considered as torque and rotational speed changes within the permissible limits (Table 1). The range of permissible variations of engine performance such as particulate matter emissions, temperature and air pressure, exhaust gas, etc. are also specified.

Table 1. Acceptable deviations of the measured engine parameters acc. [13]

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

In addition, according to the Polish Standard [14], permissible fuel consumption deviations ΔG_e and engine power deviation ΔP should satisfy the following relation:

$$\sqrt{\Delta G_e{}^2 + \Delta P^2} \leq 3\% \quad (3)$$

An analysis of the possibility of determining the static characteristics using the operational data of a vehicle engine in terms of criteria mentioned above was discussed in work [15]. Unfortunately, the standard does not define the time, in which the specified deviations must be kept to conclude the engine is operating in the static state. Assuming the transient time (time which engine needs to adjust from one operating point to another) t_p = 1 min, measuring time in static state t_s = 15 s and density of operating points: $M \subset (0,25,\ldots550)$, $n \subset (600,650\ldots1300)$, the total test time should not exceed 7 hours.

3 Load application using PTO

A common practice to test engine operating parameters in applied load, is to use engine test benches. In case of testing a complete vehicle usually the only available options are chassis dynamometers. Load application is realized there by resistance of the rollers, while the vehicle is fixed to the restraining device. However, for utility vehicles' manufacturers it can be problematic and costly. In this paper an alternative solution was described, which is based on load application using PTO as an 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 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:

$$\eta \cdot M \cdot \omega = Q \cdot p \quad \text{[kW]} \quad (4)$$

where η – pump efficiency.

In general, the range of received power is determined by pump's displacement and parameters of proportional relief valve. However, some others constrains are discussed later in the paper.

The main requirements of the design of test bench are listed below:

- Testing the engine in demanded states and with demanded operating parameters is achieved by controlling the engine speed and parameters of proportional relief valve
- Received power is equal to the maximum power of the hydraulic system of refuse collection vehicle (RCV)
- Test bench consists of dismountable components, ready to use both in static and mobile (in the future) tests
- Test bench can be used to test any chassis with PTO and GVM ≤ 26 t

Operating parameters of hydraulic system of RCV are determined by the vehicle's functions, which are: collection, compaction and transport of waste. The working elements usually consist of hydraulic cylinders. To achieve proper velocity of cylinders the required oil flow Q equals 120 l/min [16]. For safety reasons, the maximum pressure in the system is set to 180 bar. However, the components of the system allow to operate with the pressure of 230 bar, so the maximum received power can reach 46 kW and for such value the test bench is designed.

The n range is limited from below by the minimum idle speed (600 rev/min) and from above by noise emission (1300 rev/min), which is undesirable in urban areas. At this stage it is important to remember, that engine speed usually is different to pump speed due to PTO ratio (i_{PTO}= 0,86÷1,3). For low n values, high P value entails high torque values. Therefore, overall torque limitation both PTO and pump drive shaft should be taken into account. Whereas different pump can be chosen, PTO is delivered and assigned to the chassis and its maximum torque usually is not higher than 600 N·m. Another constrain is maximum pump displacement. For instance, for 150 cm^3/rev and n_p=600 rev/min oil flow Q reaches 90 dm^3/min, which with the 230 bar of pressure corresponds to the power P=34,5 kW and torque only of 549 N·m. So both a large pump and high torque PTO is required to receive high power in the lower range of n.

A constrain, which is design-independent is a minimum engine speed for demanded torque. It is probable, that despite proper test bench design high torque in low n will be unachievable. The magnitude of this constrain is associated with particular tested engine.

Constrains mentioned above are shown on the Fig.1., where the estimated achievable range of engine characteristic is hatched.

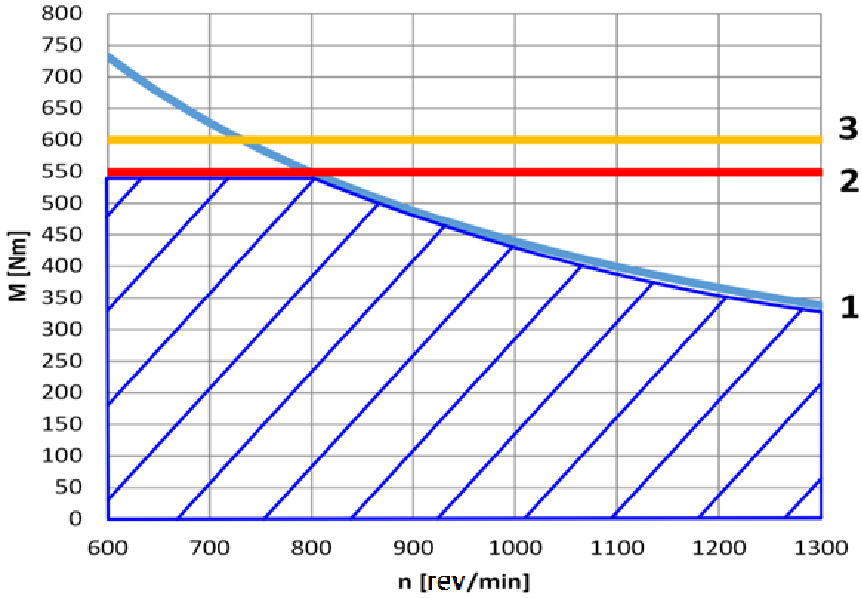

Figure 1: Achievable range of engine characteristic (hatched). (1) – maximum power P=46 kW, (2) – M_{max} on pump, (3) – M_{max} on PTO

4 Hydraulic system

To drive the hydraulic system of the test bench Load Sensing (LS) pump Leduc TXV150 [17] was chosen, with maximum displacement *q*=150 cc/rev. It allows to receive high value of torque from PTO even for low *n* and to maintain constant flow *Q*=120 dm^3/min for higher *n* (even when i_{PTO}=1,3). It should be pointed out, that pump must be able to rotate either direction due to the fact, that chassis may be equipped with PTO, which rotates clockwise or counterclockwise.

Using proportional relief valve as load generator means that the whole power delivered by the engine is converted into the heat of hydraulic oil. Oil tank with capacity of 400 dm^3 and dimensions 1000x1000x500 mm is able to dissipate approximately 1,2 kW. Thus, it is necessary to equip the test bench with air oil coolers, with cooling capacity 1,12 kW/^{o}C. In presented design two air oil coolers – Parker LDC033 [18] were used.

In the Fig.2 simplified hydraulic diagram is presented. Load sensing pump (1) supplies the system maintaining constant pressure drop on proportional directional valve (2), so that for n_p>800 rev/min pump displacement is controlled by (2) setting and the flow does not exceed 120 dm^3/min. In case of a pause in engine examination the valve (2) closes and the pump sets itself to minimum displacement minimizing power take-off. Proportional relief valve (3) is responsible for applying the load to the pump, causing the oil to heat. Directional valve (4) opens a circuit of air oil cooler when oil temperature in the tank reaches 70^{o}C. Pressure relief valve (6) is set to 230 bar.

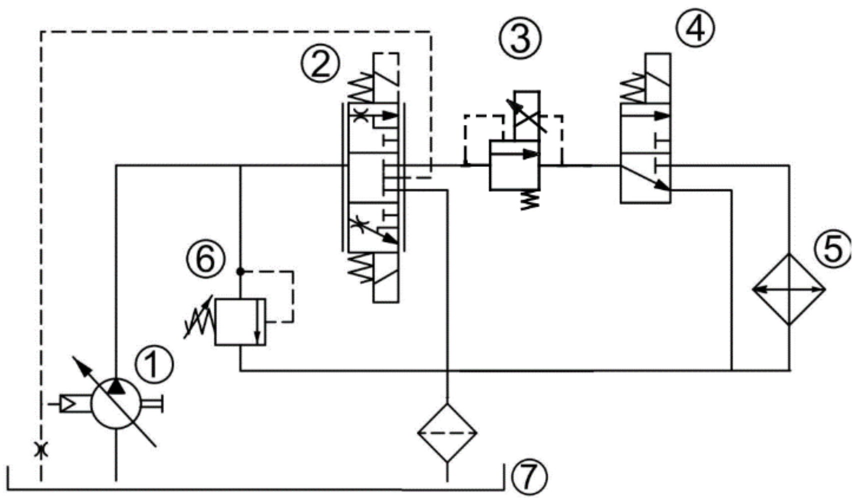

Figure 2: Simplified diagram of the hydraulic system of the test bench (description in the text)

Parameters needed to determine general characteristic such as *n*, *M* and hourly fuel consumption *B* can be obtain directly from the chassis. Utility vehicles produced after year 2002 transmit these information via CAN bus according to FMS standard [19]. It is possible to connect data recorder using interface in driver's cabin and it does not require any interference with vehicle electronic system. While the n and B values are determined directly by engine control unit (ECU), the torque is calculated using other parameters. This calculated value is described as percent of maximum engine torque and corresponds to mean effective pressure in the cylinder. In practice, it means that the M reaches even 250 N·m while idling. Thus, verification of this value is necessary. To measure M accurately, torque dynamometer was mounted between PTO and the pump.

5 Test bench control unit

The main function of the control unit (CU) is to set the engine into desired operating point and data acquisition. The diagram of CU is shown in Fig.3. Engine speed is set by sending to chassis proper message via CA, which contains information about n (n_s signal). Engine load is realized by adjusting proportional relief valve (3) by signal Y_1, but to set exact torque value, feedback from torque dynamometer is used (M_m). Engine operating parameters M_s, n and B are read from CAN and decoded by Programmable Logic Controller (PLC) and stored in the memory of data logger. Signal Y_1, which operates proportional directional valve (2) determines start, pause or end of the test. Oil cooler circuit is opened by signal Y_3. When oil temperature t reaches 70ºC, signal Y_4 is sent and turns on the oil coolers.

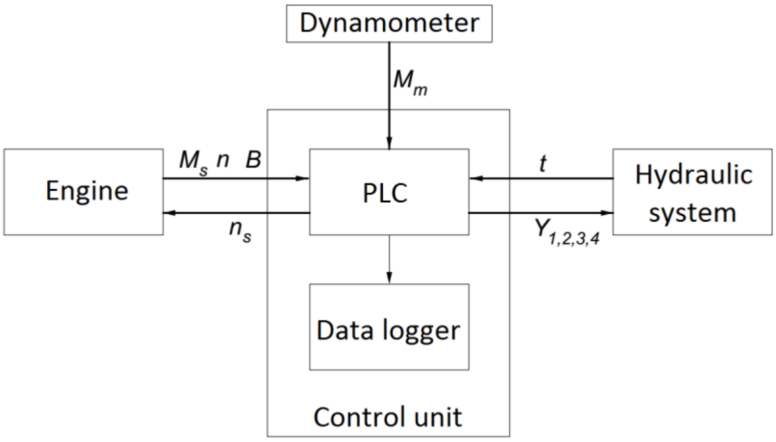

Figure 3: Test bench control unit diagram

6 Mobile test bench

The design of the test bench is shown in the Fig.4. It is mounted to the chassis frame (8) by screws using standard holes in the frame. The construction consist of steel frame (dimensions: 3250x1400x1370 mm) (7), with the intended space for oil tank (4), oil coolers (9) and additional components planned in future investigations. Hydraulic valves (5) and test bench control unit (6) are located in the interface at the side of the frame. Hydraulic pump and dynamometer are a separate unit (3) connected to the engine (1) via homokinetic shaft. The pump is connected to the tank and other hydraulic components by hoses, which can have various lengths. Such separation makes mounting operation much easier. Presented in this paper test bench construction allows us to mount it on any GVM ≤ 26 t chassis. In the Fig.5. and Fig.6. the test bench mounted on the chassis frame is shown.

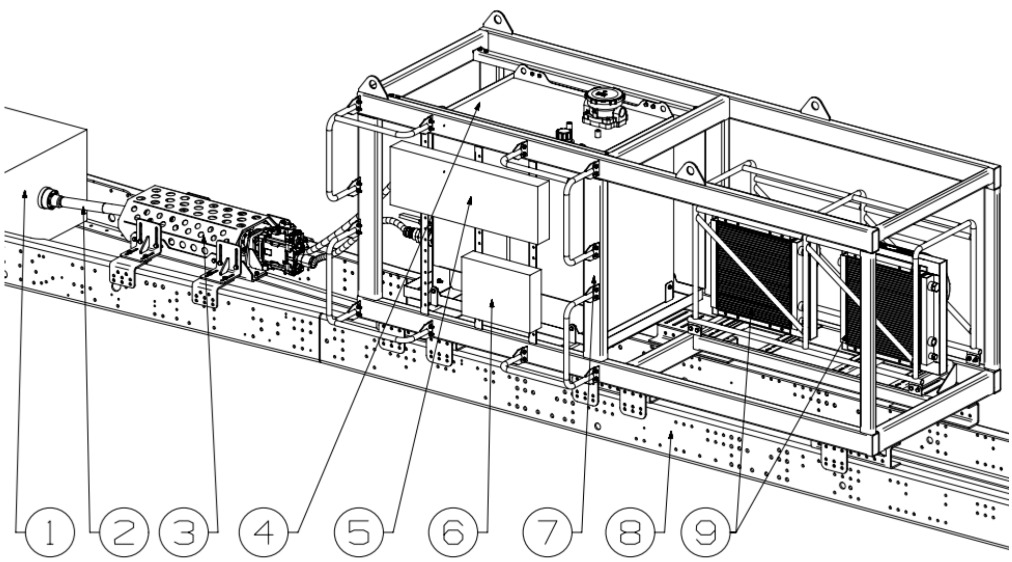

Figure 4: Test bench 3D model

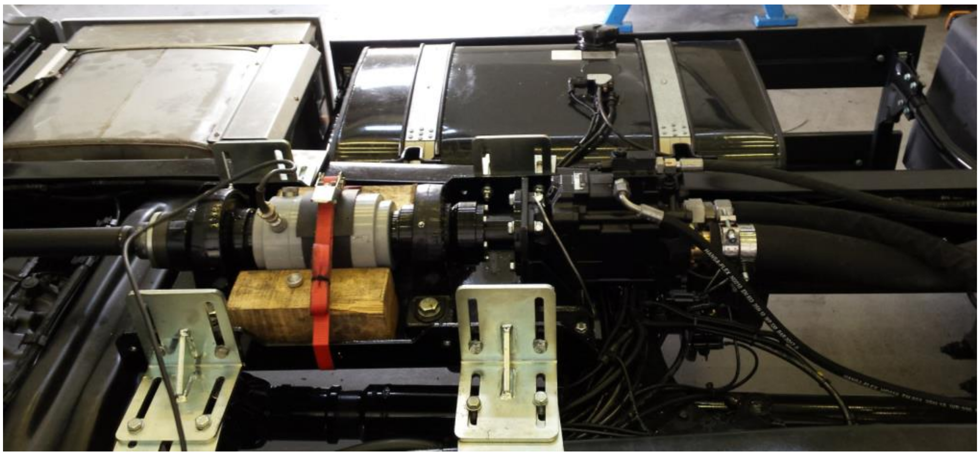

Figure 5: Dynamometer and pump unit, connected to the engine by the shaft

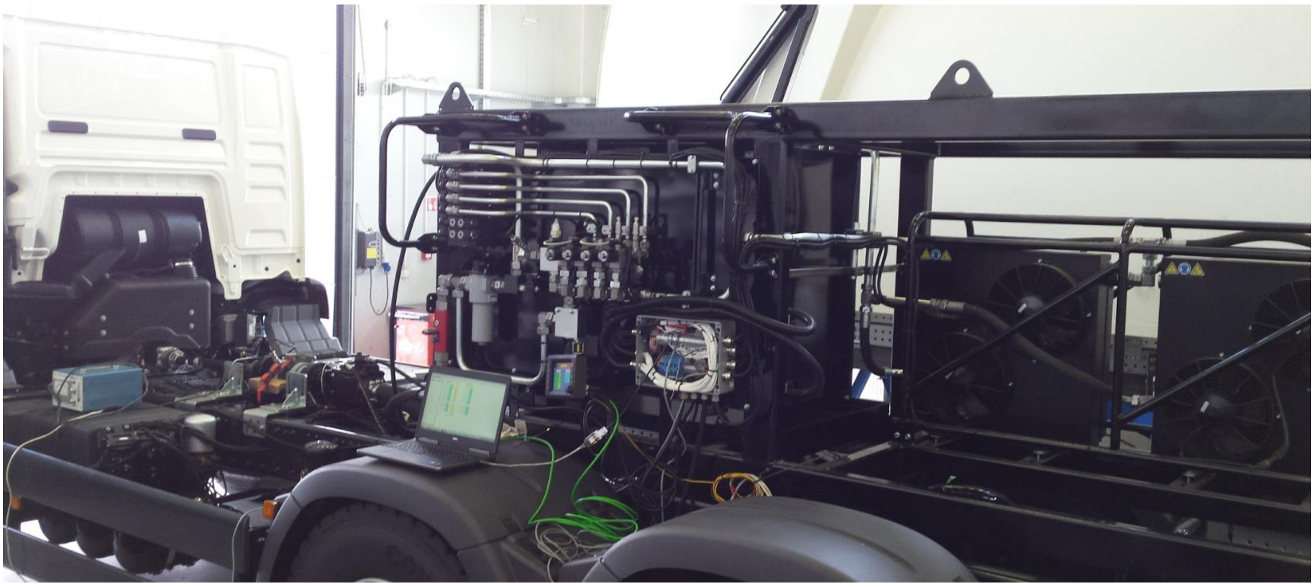

Figure 6: Test bench mounted on the chassis frame (MAN 26.320)

7 Conclusions

Test bench presented in this paper allows to determine the general engine characteristic in the power range of 46 kW for any chassis of utility vehicle at the body production phase, using PTO as load applying device. It creates possibility to obtain optimal configuration of drive system of the RCV, which indicates lower fuel consumption and emission of toxic exhaust gases, particularly important in vehicle operating areas - urban areas. Proposed hydraulic system receives the power effectively and allows the control unit to generate demanded resistance torque at particular engine speed, while the data logger registers all the information and operating parameters. It means, that the test bench can be used in the future investigations on operating parameters of the engine in transient states and to simulating the engine load by operating conditions of real RCV. As a consequence, an assessment of energy consumption for each RCV would be possible.

8 Acknowledgement

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