The usage of CAN-Bus messages for engine power determination

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Abstract

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Determination of the actual power of tractor's engine in the operation can be done by calculation which requires the use of a range of parameters such as coefficient of rolling resistance, mechanical efficiency, the moments of inertia, etc. Their values are usually tabulated and therefore the engine power cannot be determine without simplifying. Another solution is to use the actual engine torque message from the CAN-Bus, which brings a specific value of the actual torque. The aim is to use the current torque to calculate the engine power in the deployment of tractor's set in transport operation. The results show that at a uniform movement of the flat section the engine power reached 73 kW. When driving uphill, the value of the actual power reached from 130 to 150 kW depending on the selected gear. Using the actual parameters of torque makes possible to identify the known full speed characteristics of the current engine without the need for assembly demanding measurement techniques.

Keywords: actual engine torque; engine; tractor; engine power; CAN-Bus

The tractor is more often used for working operations during the year. The manufactures try to reduce effort to driving or operating the tractor. The way how to affect a need for driving is the usage of components such as suspension of front axle, cab and operator's seat, the location of controls and indicators. The second possibility may be improvement of performance, e.g. increase in engine power, automatic shifting, etc. From a technical point of view it seems to be interesting to assess the performance of the engine in the transport of specific sets.

Theoretical analysis: All procedures for the determination of the actual power are based on the calculation of the effects of external forces applied to vehicle. The development of electronic control for tractors with an option to use current information about the activities of selected functional components of the CAN-Bus can be used as a base of input data of analysis (Suvinen, Saarilahti 2006). This network brings huge information about operation, for example the current torque, the torque or engine load. Whether the parameters are associated with specific torque measured on the dynamometer, it may provide engine power under various conditions in operation.

Figs 1–3 bring schematic representation of the three typical situations which may occur during transport. The simplest case would be run on the flat. Driving force brought to the driving axle must be in balance with the resistance expressed in Eq. 4.

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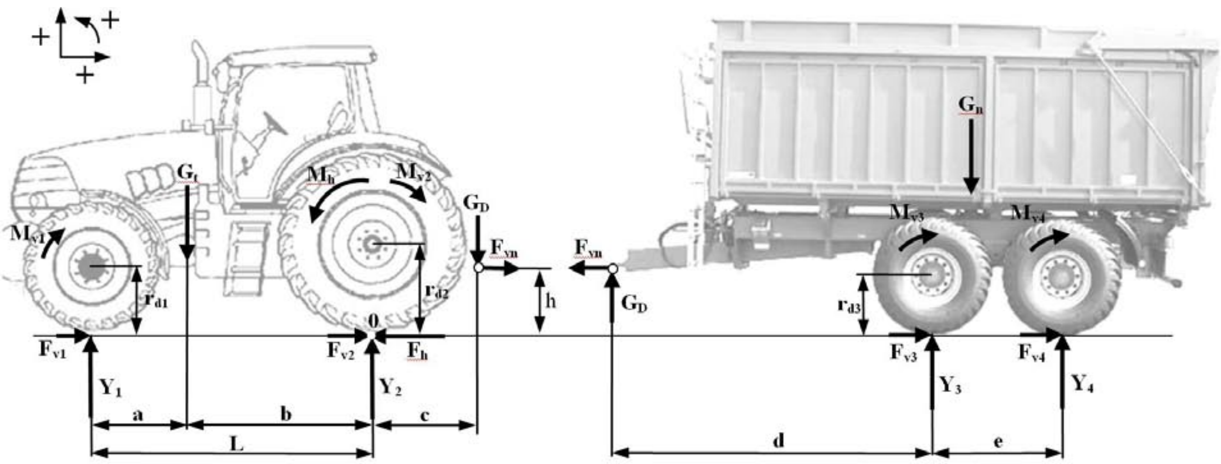

Fig. 1. Force and torque balance of the tractor when driving on the flat road in an unequal movement

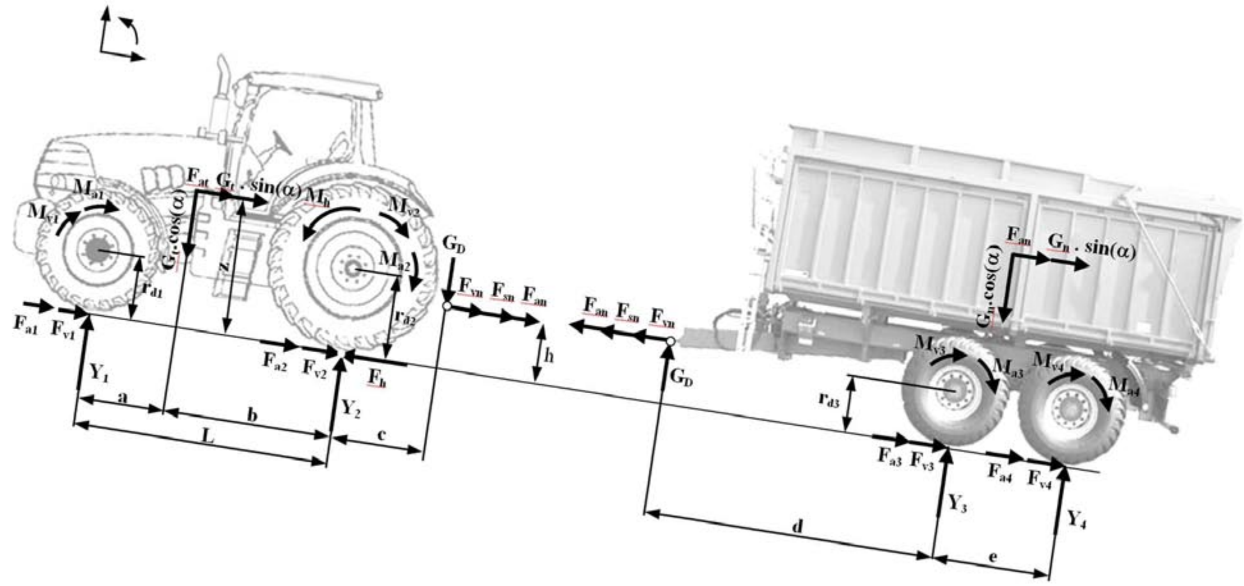

Fig. 2. Force and torque balance of the tractor when driving uphill in an unequal movement

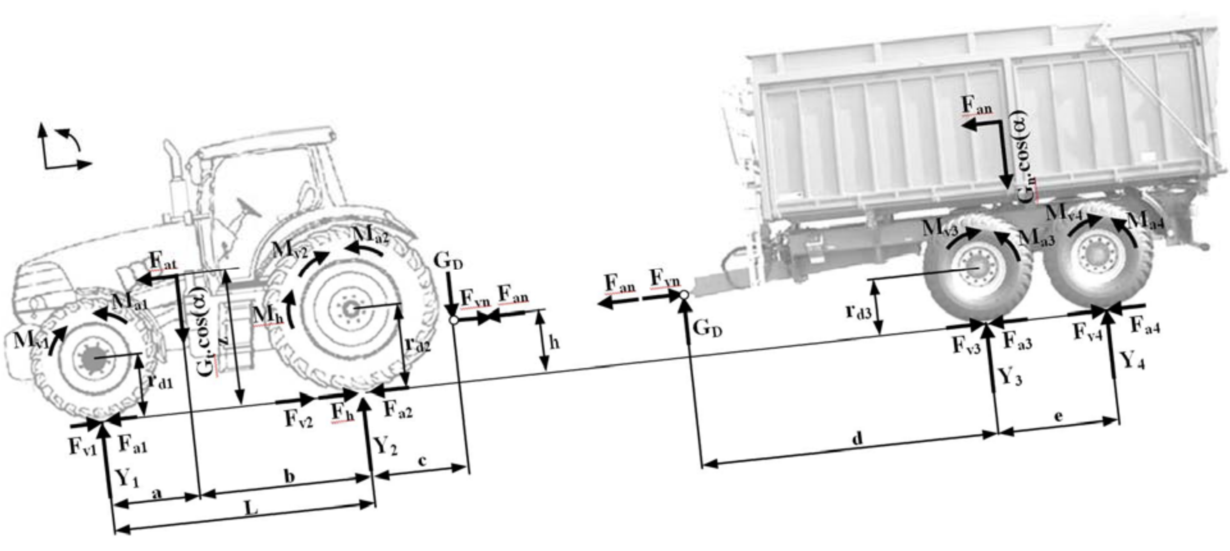

Fig. 3. Force and torque balance of the tractor when driving down the hill in an unequal movement

$$\sum_{i=1}^{\infty} F_{xi} = 0; F_{v1} + F_{v2} - F_h + F_{vn} = 0 \quad (1)$$

$$\sum_{i=1}^{\infty} F_{yi} = 0; Y_1 + Y_2 - G_t - G_d = 0 \quad (2)$$

$$\sum_{i=1}^{\infty} M_{oi} = 0; G_t \times b - M_{v1} - Y_1 \times L - M_{v2} + M_h - \\ - G_d \times c - F_{vn} \times h = 0 \quad (3)$$

where:
F_v – rolling resistance (N)
F_h – driving force (N)
Y – normal force (N)
G_t – tractor gravity (N)
G_d – gravity transferred from trailer to tractor (N)
M_v – torque of rolling resistance (N m)
M_h – driven torque (N m)
f – coefficient of rolling resistance (–)

Driving force can be calculated from Eq. 1 as a sum of out effects.

$$F_h = F_{v1} + F_{v2} + F_{vn} = Y_1 \times f_1 + Y_2 \times f_2 + Y_3 \times \\ \times f_3 + Y_4 \times f_4 \quad (4)$$

$$F_h = \left(G_t + G_d + G_n\right) \times f = G_c \times f \quad (5)$$

Another form of equation includes knowledge of engine torque, ratio gear, mechanical efficiency and dynamic diameter of a wheel.

$$F_h = \frac{M_t \times i_c \times \eta_m}{r_{d2}} \quad (6)$$

where:
M_t – engine toque (N m)
i_c – ration gear (–)
η_m – mechanical efficiency (–)
r_{d2} – dynamic diameter of driving wheel (m)

Engine torque can be rewritten according to Eqs 5 and 6 as follows:

$$M_t = \frac{G_c \times f \times r_{d2}}{i_c \times \eta_m} \quad (7)$$

The calculation of engine torque brings difficulties due to coefficient of rolling resistance and mechanical efficiency. Both parameters are already tabulated, but especially the value of the mechanical efficiency depends on used references or experimental results of different authors. Common mechanical efficiency at spur gears is according to the implementation of at least 0.98, at 0.97 bevel wheel shaft in the case of 0.995.

$$\eta_m = \eta_{čk}^{m} \times \eta_{kk}^{n} \times \eta_{uh}^{p} \quad (8)$$

where:
$\eta_{čk}^{m}$ – efficiency of spur gearing (–)
η_{kk}^{n} – efficiency of bevel gearing (–)
η_{uh}^{p} – efficiency of shaft support (–)

The overall mechanical efficiency according to Semetko lies between 90% and 94%, Remus, Culshaw present 88%, Gega from 85% to 95%, Reiter between 80% and 88%, Grečenko writes 85–90%, and Okamoto et al. 72% to 80% (Tinker 1993). It means that the accuracy of the calculation of load will be determined by the value of the mechanical efficiency and coefficient of rolling resistance. Therefore, a better offer is monitoring of the actual torque from the CAN-Bus. When driving uphill, the calculation of the driving forces is difficult, since it appears that resistance rises and the effects of rotary inertia and sliding mass. Force and torque balance are shown in Fig. 2.

$$\sum_{i=1}^{\infty} F_{xi} = 0; F_{a1} + F_{v1} + F_{at} + G_t \times \sin(\alpha) + F_{a2} + \\ + F_{v2} - F_h + F_{vn} + F_{sn} + F_{an} = 0 \quad (9)$$

$$\sum_{i=1}^{\infty} F_{yi} = 0; Y_1 + Y_2 - G_t \times \cos\left(\alpha\right) + G_n = 0 \quad (10)$$

$$\sum_{i=1}^{\infty} M_{oi} = 0;\ G_t \times \cos\left(\alpha\right) \times b - M_{v1} - M_{v2} - F_{at} \times z - \\ - G_t \times \sin(\alpha) \times z + M_h - M_{a2} - M_{a1} - G_d \times c - \\ - F_{vn} \times h - F_{an} \times h - F_{sn} \times h = 0 \quad (11)$$

The balance of power can be calculated by the value of the driving forces. The calculation is difficult to realize in order to find the effects of rotary inertia mass connected with the driving axle.

$$F_h = F_{a1} + F_{a2} + F_{at} + F_{an} + G_t \times \sin\left(\alpha\right) + \\ + F_{sn} + F_{v1} + F_{v2} + F_{vn} = F_{ac} + F_{vc} + F_{sc} \quad (12)$$

where:
F_{a1} – inertia force of front axle (N)
F_{a2} – inertia force of turning parts (N)
F_{at} – inertia force of tractor (N)
F_{an} – inertia force of trailer (N)
F_{sn} – drag force of trailer (N)
M_a – inertia torque (N m)
F_{ac} – total inertia force (N)
F_{vc} – total force of rolling resistance (N)
F_{sc} – total force of climbing drag (N)

The last case is the ride down the hill, which once again engages the inertia effect but operates in the direction of movement and reduces the value of the forces. The measurements showed that the engine while driving down the hill (where the trailer pushed the tractor) reduces torque at zero. In fact, the torque can be regarded as negative, leading to a change in the direction of action of the driving forces. Balance of power exists between the rolling resistance, driving force, inertia effects and the moving force behind a set of F_s. Force and torque balance are presented in Fig. 3.

$$\sum_{i=1}^{\infty} F_{xi} = 0;\ F_{v1} + F_{a1} + F_{at} + F_{v2} + F_{a2} + F_{vn} + F_{an} + F_h - F_{st} - F_{sn} = 0 \quad (13)$$

$$\sum_{i=1}^{\infty} F_{yi} = 0;\ Y_1 + Y_2 - G_t \times \cos(\alpha) - G_n = 0 \quad (14)$$

$$\sum_{i=1}^{\infty} M_{oi} = 0; -M_{a1} - M_{v1} - Y_1 \times L + G_t \times \sin(\alpha) \times b - F_{at} \times z - M_{v2} - M_{a2} - G_n \times c - F_{an} \times h - F_{vn} \times h - M_h + F_{sn} \times h + F_{st} \times z = 0 \quad (15)$$

$$F_{v1} + F_{v2} + F_{vn} + F_h + F_{a1} + F_{a2} + F_{at} + F_{an} = F_{st} + F_{sn} \quad (16)$$

MATERIALS AND METHODS

Load measurement of combustion engine was realized with a support of tractor set of CASE IH Puma 195 tractor and Annaburger HTS 22.79 semi-trailer. Field measurement was done after laboratory measuring of full speed characteristic in Vehicle's laboratory of Department of Engineering and Automobile Transport (Mendel University in Brno, Czech Republic). Torque was measured on test bench equipped with Eddy current dynamometer VD 500 during partial and full supply of fuel. CAN-Bus messages were stored in the same time sampling; the file consisted of fuel consumption, engine speed, fuel temperature and air temperature before and after turbocharger, torque, etc. The measured torque values together with other needed parameters were recorded at a frequency of 18 Hz and stored in computer memory. The distance of testing track was 21.7 km divided into sub-sections, in particular the character of the road gradient. Total weight is given in Table 1. In the transport operation following parameters were measured or calculated: altitude, time travel, actual velocity, fuel consumption, engine speed, actual torque, fuel temperature, altitude, and location sets. Sampling frequency signals from the sensors was 5 Hz. Location tractor was determined by GPS receiver.

Table 1. Weight of set (in kg)

CASE IH Puma 195 + trailer Annaburger HTS 22.79 (load)	
Front axle of tractor	2,380
Total weight of tractor	12,260
Tractor + trailer	31,760
Trailer	19,460

System of terrain measurement includes three basic parts:

- data acquisition system of HBM Company,
- CAN-Bus monitoring through the PCMCIA CAN card of National Instruments,
- tractor location determined by GPS receiver.

Altitude measurement

Altitude measurement was realized using pressure sensors – Motorola MPX4115 – measured in the range of 15 to 115 kPa. The output signal is linear over the range of 0.2 to 4.8 V. Pressure accuracy is 1.5%. Calibration of the sensor is realized on the triangulation point with defined altitude. Uncertainty of the measurement is better than 0.5 m. The sensor was powered via USB voltage stabilizer and the value of supply voltage led to the input of card for measurement correction. Signal was processed by the USB DAQ card from National Instruments products.

The CAN-Bus monitoring

Software allows monitoring of CAN-Bus of a tractor. CAN-bus of tested tractor was fully compatible according to standard SAE J1939 in extended ArbID (29-bit). Communication speed was 250 kbps and for the analysis of the report relevant channels were selected.

RESULTS

Measured values were processed and evaluated by spreadsheet editor Microsoft Excel. Using re-

Table 2. Statistics of selected indicators during the plane ride

Index	Velocity v (km/h)	Altitude h (m)	Engine speed n (1/min)	Actual engine torque (%)
Averaged	41.40	210	1,745.68	49.30
Standard deviation	0.11	0.72	4.79	1.42
Minimum	41.11	209.70	1,734.00	46.00
Maximum	41.48	210.30	1,757.00	52.00
Number of values	133.00	133.00	133.00	133.00
Variation coefficient (%)	0.27	0.12	0.27	2.89

Table 3. Values used to calculation

i_c (for 19th gear)	(–)	14.798
r_{d2}	(m)	0.925
η_m	(–)	0.85–0.9

gression analysis they were processed according to $M_{takt} = f(M_t)$ and $M_t = f(n_{eng})$ to establish the full features, Figs 4, 5. Determinacy index values ranged from 0.97 to 0.99. Measured values of transport are graphically summarized in graphs; Figs 7–9.

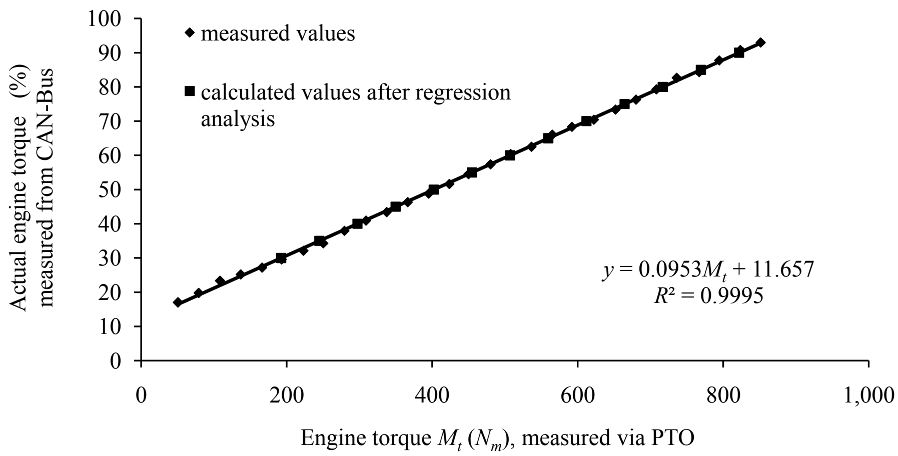

Fig. 4. Dependence of actual engine torque versus engine torque measured on dynamometer

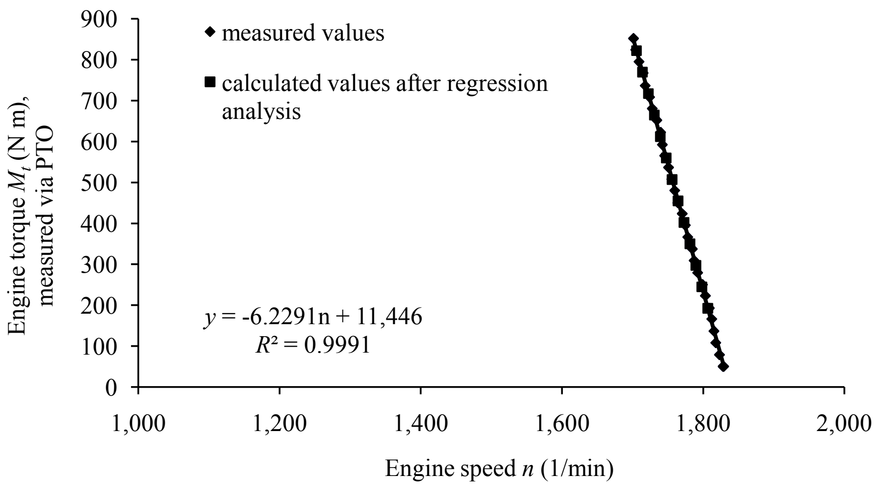

Fig. 5. Dependence of actual engine torque versus engine torque measured on dynamometer, regulation part from engine performance, the CASE IH 195 Puma tractor; power take off (PTO)

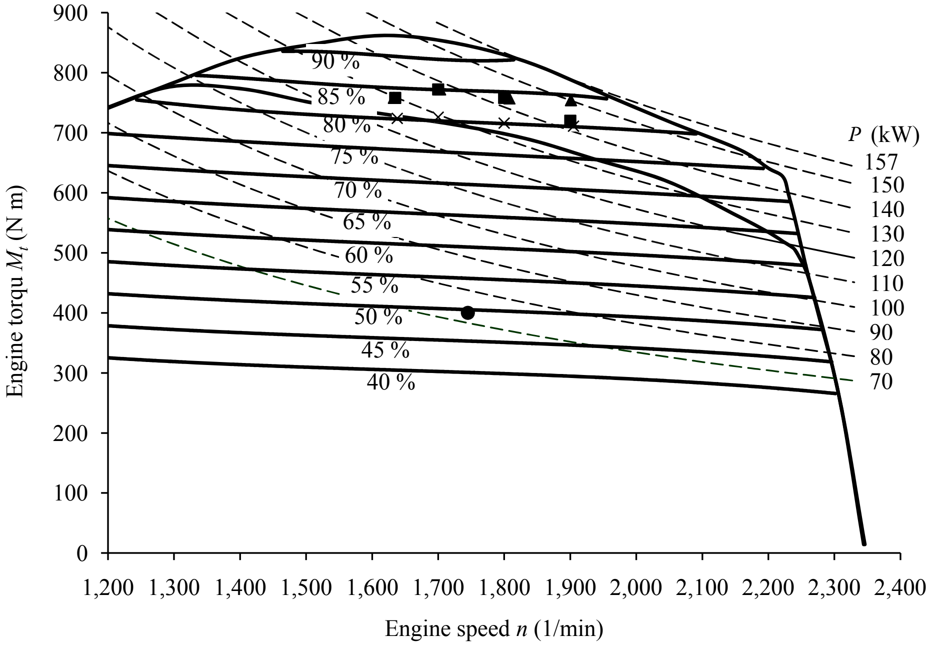

Fig. 6. Full speed map of the CASE IH 195 Puma tractor with constant curve of actual engine torque and power

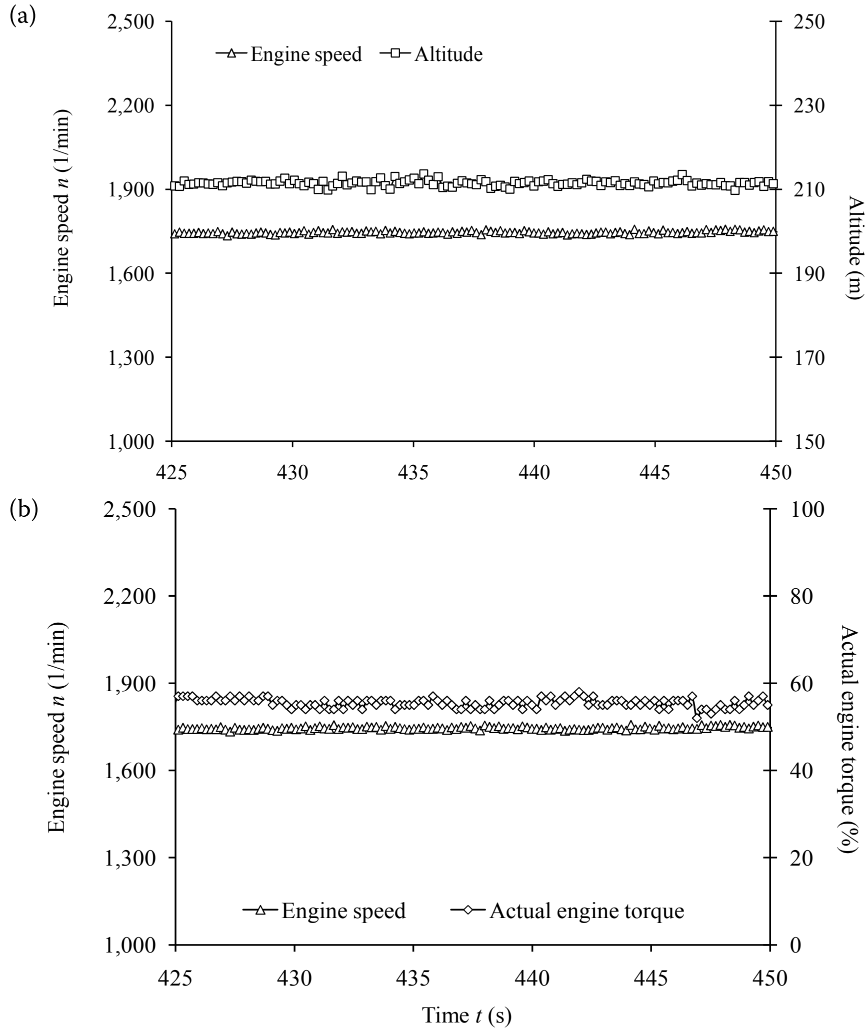

Fig. 7. Record of measured parameters in driving on the flat road; altitude (a), actual engine torque (b)

DISCUSSION

Rated speed characteristic was measured in order to determine the load of the combustion engine during its operation. Full speed map of engine was built of ten partial load curves (Fig. 6). The curves were plotted on constant actual torque in %, which is given by approximation using the specific values of engine torque and engine speed (Figs 4 and 5). Parameters of CAN-Bus were monitored to bring the full features and to determine the value of the actual torque and engine power. Testing track was divided into section with different profile where the load of engine varied from zero to full torque. Nineteenth gear was shifted on a flat road. Evaluation in selected section was targeted that the movement of tractor approached balance straightforward movement. This is also confirmed by statistical evaluation of selected parameters measured during the trip, Table 2 and Fig. 7. The location of actual engine torque in full speed map can be described on the basis of two major variables – engine speed and engine load (Fig. 6). Value of engine power on flat road has reached 73 kW, which corresponds to 46% of maximum power. The Eq. 7 can be calculated as the value of the coefficient of rolling resistance for the set. Parameters for calculation are listed in Table 3. The results show that the coefficient of rolling resistance for the set ranged from 0.0175 to 0.0185 for the mechanical efficiency of 0.85 and 0.9. Resulting values correspond to the range of reported references Marcín, Zítek (1985), Semetko et al. (1986), Grečenko (1994), Macmillan (2002). The same procedure can be used for determination of the engine power when driving uphill. The full speed map consists of three gears (16, 17, and 18), which were shifted in the selected parts of the track, Fig. 8. Four characteristic points were selected for

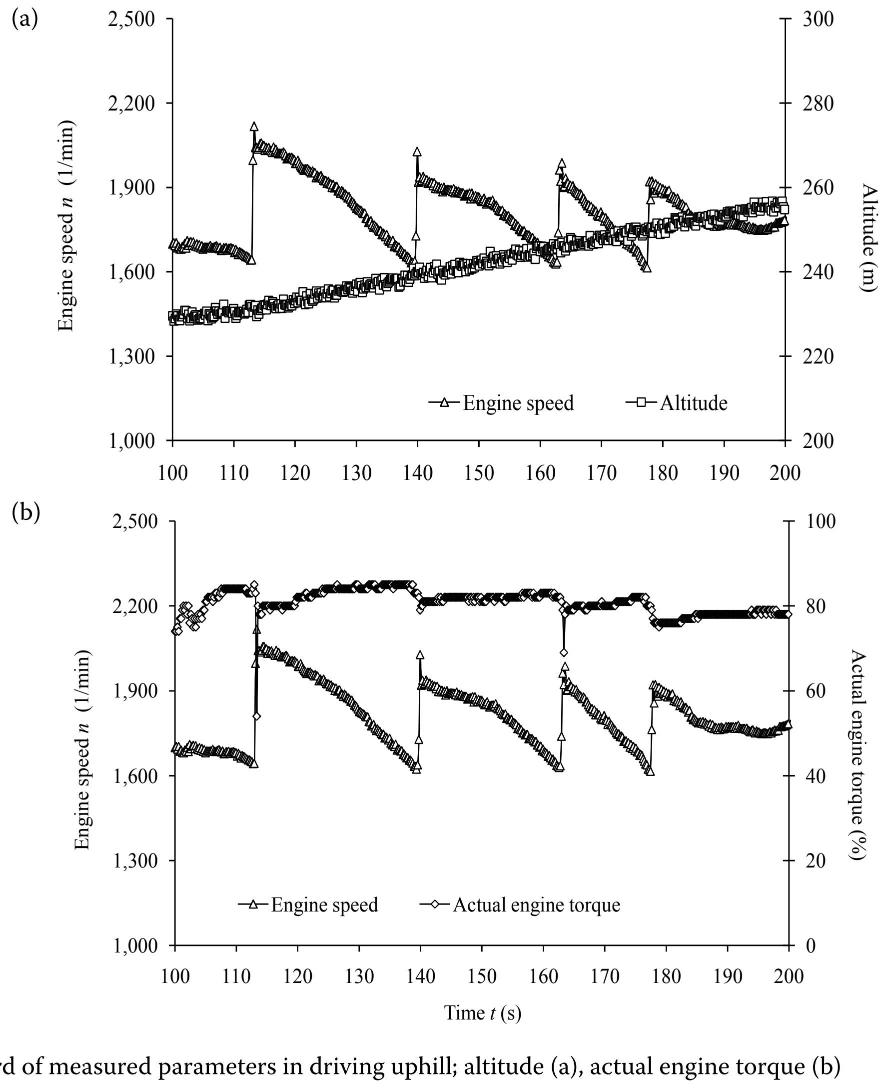

Fig. 8. Record of measured parameters in driving uphill; altitude (a), actual engine torque (b)

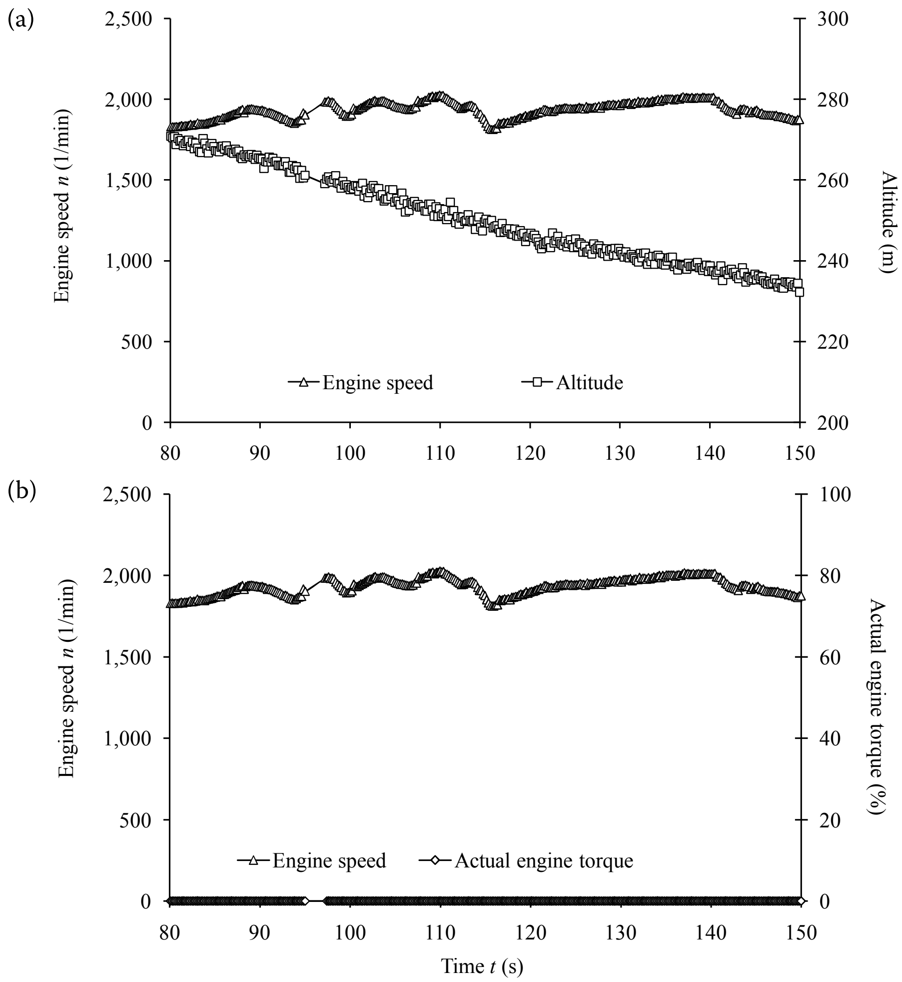

Fig. 9. Record of measured parameters in driving downhill; altitude (a), actual engine torque (b)

Table 4. Selected parameters brought up into total engine performance calculation (*average values)

	Measured value		Computed value from total engine performance	
	engine speed (1/min)	actual engine torque (%)	engine power (kW)	engine torque (N m)
Riding on the flat	1,745*	49	73.05	400
Ride uphill 16th ration gear	1,905	80	141.76	711
	1,800	80	134.89	716
	1,700	81	129.17	726
	1,638	80	124.12	724
Ride uphill 17th ration gear	1,900	82	143.18	720
	1,800	84	142.80	758
	1,700	85	137.36	772
	1,635	83	129.71	758
Ride uphill 18th ration gear	1,901	84	150.02	754
	1,808	84	143.25	757
	1,703	85	137.60	772
	1,633	83	129.55	758
Ride downhill	1,905*	0	0	0

each gear, Figs 4, 5 and these points were plotted into the full speed map Fig. 6. The results show that the tractor operated with an increase in the engine power between 130 to 150 kW at 17th and 18th gear. Maximum engine power was consumed at 83% to 96%. Downshifting to 16th gear brought the reduction in boost due to the higher gear ratio and overload in transmission. Driving down the hill, actual torque message cannot be used to calculate power, as the set is pushed and the engine sets the zero fuel supply and the actual torque generated by engine is zero, Fig. 9.

CONCLUSION

The actual load can provide the value of actual engine power whether the full speed map had been measured. This procedure brings advantages in needs for installed instrumentation what is positive, especially in field measurement.

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