TRANSPORT
2008
23(1): 37–43

INVESTIGATION OF TRACTOR ENGINE POWER AND ECONOMICAL WORKING CONDITIONS UTILIZATION DURING TRANSPORT OPERATION

Antanas Juostas[1], Algirdas Janulevičius[2]

Lithuanian University of Agriculture, Studentų g. 15, LT-53361 Kaunas-Akademija, Lithuania
E-mails: [1]antanas.juostas@keskoagro.lt; [2]algirdas.janulevicius@lzuu.lt

Received 28 June 2007; accepted 20 November 2007

Abstract. Article analyzes tractor working and its engine conditions from economical point of view. Overview of tractor wheel slippage reliance on the traction force and weight utilization coefficient is given. Tractor maximum driving force according to road and field conditions, and driving speed are submitted. Literature and theoretical investigation analysis is done, where interaction between tractor wheels made-up driving force and grip is analysed. Driving speed and driving force dependence on rolling resistance and total aggregate weight using nominal power is described. In the present experimental research reduction in fuel consumption of tractor transport aggregate by reducing engine speed and by keeping the same work speed, was determined.

Keywords: tractor, slippage, wheel, fuel consumption, engine speed, power, stubble, gravelled road.

1. Introduction

One of the important factors is tractor working efficiency, that is efficient employment of its power for work to be done. Tractor engine's nominal working condition is not an efficient condition by SAME DEUTZ-FAHR … (2002), Ivanov *et al.* (2006) and Air pressure … (2005). Such features could be observed in most tractor characteristics. Tractor Deutz Fahr Agrotron TTV 1160 engine characteristic shows that maximum engine power is reached at 1800 rpm, whenever nominal engine speed is 2 350 rpm (Fig. 1) – SAME DEUTZ-FAHR … (2002). Engine has 30 % of torque rise. Constantly high torque range is reached at 1 400–1 800 rpm. Low fuel consumption of 209 g/kWh is in the optimum torque range at speed range of 1 450–1 750 rpm. Shifting can be delayed until the speed drops as low as 1 200 rpm. Constant engine performance at wide speed range gives relaxed work, power reserve, less gear shifting without traction interruption, high flexibility values with lower fuel consumption at lower speeds, but the same engine performance. Deutz engine characteristic curves sheet (Fig. 1) shows that economical engine working conditions (maximum torque and power and lowest fuel consumption) are achievable at lower than nominal engine speed.

During tractor maintaining in following conditions we could get the lowest fuel consumption, see Air pressure … (2005), Kraujalis (2002) and Neunaber (1997).

Work objective is to evaluate tractor working and its engine load from economical point of view, also to investigate economical condition during tractor transport aggregate use.

2. The analysis of literature and theoretical investigations

Important dimension in tractor dynamics analysis is a load, i.e. pulling force. All other indicators depend on this dimension. In turn, tractor pulling force is in straight dependence on effective engine torque.

For theoretical calculation, by Litvinov, Farobin (Литвинов, Фаробин 1989), Carrol (1992) and Wong (1989), of possible tractor pulling force (on even surface), which is the sum of rolling resistance and pulling force $F_v = F_f + F_t$, we have to know engine power P_e, driving speed v and transmission coefficient of efficiency η_{tr}:

$$F_v = \frac{\eta_{tr} P_e}{v}. \tag{1}$$

Driving force is not depending only on machine engine, transmission and chassis parameters but it as well depends on wheel interaction with soil. The maximum driving force, when driving torque is big, is limited by soil mechanical resistance or grip between wheel and rolling surface. If soil is not resistant enough to mechanical impact or grip between wheels and rolling surface, driving wheels will slip. Then tractor would not move

ISSN 1648-4142 print / ISSN 1648-3480 online
www.transport.vgtu.lt

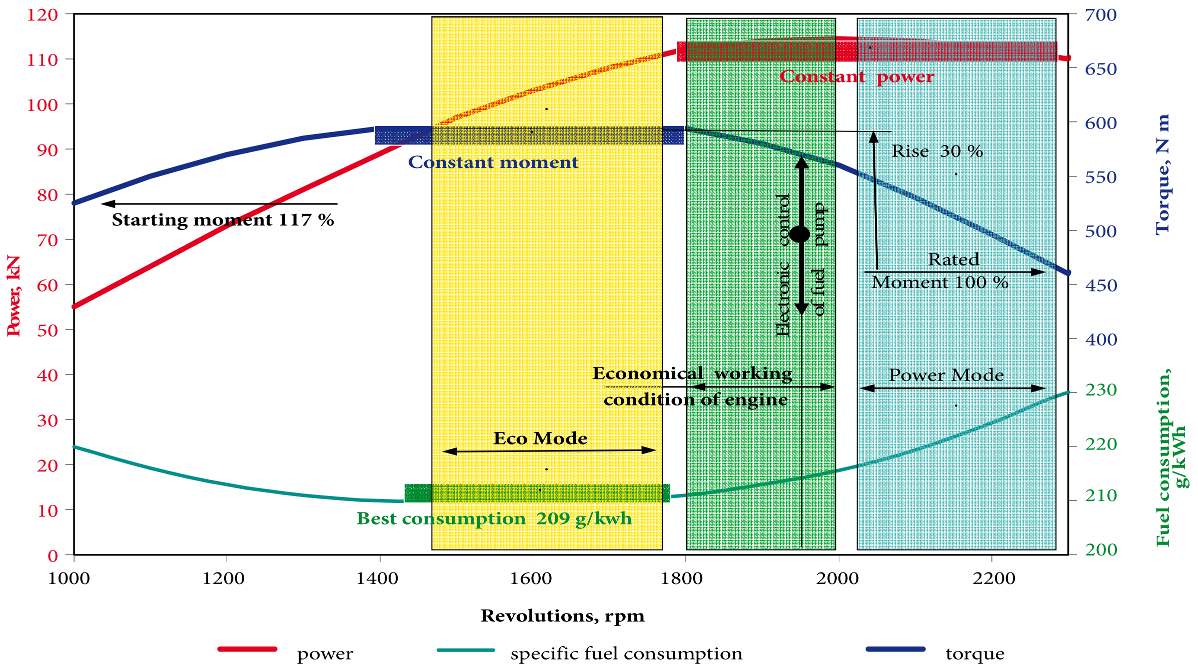

Fig. 1. Characteristic curves sheet for Deutz diesel engine of tractor Agrotron TTV 1160

from its place. Driving wheel capability to interact with soil or rolling surface is evaluated by grip coefficient φ. By knowing grip coefficient we could calculate grip force and maximum driving force by Carrol (1992) and Giedra, Janulevičius (2005):

$$F_{v\max} = F_{\varphi} = \varphi R_{yv}, \tag{2}$$

where: F_{φ} – grip force, kN; R_{yv} – soil vertical reaction to driving wheels, kN.

Theoretically to determine when tractor driving wheels would get a slippage of 100 %, we can calculate possible maximum traction force, which is the difference between grip and roll resistance $F_t = F_{\varphi} - F_f$. At given tractor traction force, driving wheels would slip absolutely ($\delta = 1$). Slippage dependence on traction force, in the same field conditions, would vary according to weight of tractor. Slippage would depend on vertical load (G) of driving wheels by researches of Giedra, Janulevičius (2005), Prentkovskis, Bogdevičius (2002) and Jun *et al.* (1998). This load exactly determines the slippage and driving wheel grip with soil. Therefore, comparing different tractor models we have to use comparative factors. One of these factors is weight utilization coefficient φ_g. This coefficient is expressed by proportion $\varphi_g = F_t / G$ of traction force F_t and vertical load G on driving wheels. The dependence $\delta = f(F_t/G)$, $\delta = f(\varphi_g)$ of the slippage δ on the weight utilization coefficient φ_g at the various working conditions was determined by Giedra, Janulevičius (2005) (Fig. 2).

The needed tractor pulling indicators could be reached and used effectively only when the main parameters such as weight of tractor, driving speed (transmission ration) and engine power will be chosen correctly. Those parameters are taken into account for tractor pulling force calculation.

For pulling force calculation we have to choose the weight of tractor. It is necessary to make a distinction

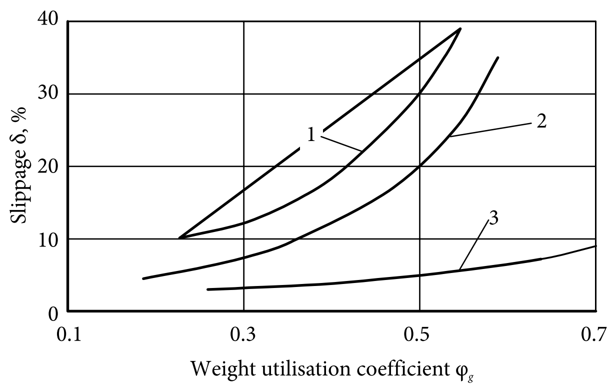

Fig. 2. Dependence of the wheel slippage on the weight utilization coefficient of tractors: 1 – on field prepared for sowing; 2 – on stubble; 3 – on asphalt road

between constructional (empty) weight m_0 and running (loaded) weight m_e. The constructional weight is comprehensible as the weight when tractor is out of: fuel, driver, tools, and optional equipment without front and rear weights. Tractor running (loaded) weight is tractor working weight. It is always bigger than constructional weight. Its minimum value $m_{e\,\min}$ is equal to the sum of constructional weight, fuel and driver weights. For most of tractors $m_{e\,\min} = (1.07\ldots1.1)\, m_0$, that is presented by Skotnikov *et al.* (Скотников *et al.* 1986).

Maximum value of tractor running weight $m_{e\,\max}$ has to be chosen so, that tractor will not exceed permissible slippage limits during tractor work with determined traction force on hook. In this instance, slippage can not exceed $\delta_{leist.}$. Prescribed requirements for wheeled tractors under steady work conditions can be expressed by equation, which is presented by Litvinov *et al.* (Литвинов *et al.* 1989) and Skotnikov *et al.* (Скотников *et al.* 1986):

$$\varphi_g \lambda m_{e\max} g = F_n + f_1 m_{e\max} g, \tag{3}$$

where: F_n – nominal pulling force on hook, N; φ_g – weight utilization coefficient, which can be achieved under given respective soil conditions with allowed slippage of driving wheels; λ and f_1 – load of driving wheels and rolling resistance coefficient under given work conditions.

From the formula (3) calculating $m_{e\max}$:

$$m_{e\ \max} = \frac{F_n}{(\varphi_g \lambda - f_1) g}. \tag{4}$$

Tractor pulling force at transport work is equal to trailer resistance force. Trailer resistance force F_p on horizontal surface under steady work conditions is equal to:

$$F_p = f_1\ m_p\ g, \tag{5}$$

where: m_p – trailer weight, kg.

Equations 3 and 5 of tractor transport work can be written as follows:

$$\varphi \lambda m_{e\max} g = f_1 g (m_p + m_{e\max}), \tag{6}$$

$$m_{e\ \max} = \frac{f_1\ m_p}{\varphi\ \lambda - f_1}. \tag{7}$$

By taking into account high pulling force, weight distribution between front and rear wheels, for tractors with two–wheeled drive is $\lambda = 0.75\ldots0.8$, for tractors with four–wheeled drive – $\lambda = 1$, that is presented by Giedra, Janulevičius (2005) and Skotnikov *et al.* (Скотников *et al.* 1986).

USA scientist Frank Zoz suggests transferred weight count, from the front axle to the rear, according to the simplified formula, which is presented by Carrol (1992) and Upadhyaya (1997):

$$\Delta G = \zeta F_t, \tag{8}$$

where: ξ – coefficient, evaluating transferred weight; ΔG – transferred weight, N.

During work, at given nominal pulling force, the load on rear wheels has to create 60 % of all tractor weight by Wong (1989) and Skotnikov *et al.* (Скотников *et al.* 1986).

Usually, for increasing tractor working weight up to value $m_{e\ \max}$, additional weights can be added or a tyre filled full with liquid. Then tractor ballast weight is calculated as follows:

$$m_\sigma = \lambda (m_{e\max} - m_{e\min}), \tag{9}$$

where: m_σ – ballast weight, kg.

Because of big job variety tractor has to have three types of gears:

- auxiliary – for getting extremely slow gear speeds (when technological procedure is required);
- main – for main agricultural work performance;
- transport – for goods transporting and driving to and back from the field.

The minimum technological driving speed υ_{n1} has to ensure full engine load at nominal torque M_n with nominal pulling force on hook. In this instance tractor working weight is equal to $m_{e\ \max}$.

Maximum technological driving speed $\upsilon_{n(z)}$ is needed when minimal pulling force on hook F_n / δ_t is needed for work performance. In this instance it is enough that tractor working weight is minimal $m_{e\ \min}$, and engine is not fully loaded.

This requirement can be formulated by the following equations:

$$F_n + f_1 (m_{e\min} + m_\sigma) g\ r_r = M_n\ i_{tr1} \eta_{tr}, \tag{10}$$

$$(F_n / \delta_t + f_2 m_{e\min} g) r_r = \gamma_{\partial\min} M_n i_{\ tr(z)} \eta_{tr}. \tag{11}$$

Tractor transport work equations (10) and (11) can be formulated as follows:

$$f_1 m_p g + f_1 (m_{e\min} + m_\sigma) g r_r = Mn\ i_{tr1} \eta_{tr}, \tag{12}$$

$$(f_1 m_p g / \delta_t + f_2 m_{e\min} g) r_r = \gamma_{\partial\min} M_n i_{\ tr(z)} \eta_{tr}, \tag{13}$$

where: i_{tr1} and $i_{tr(z)}$ – tractor transmission ratio, working at lowest and highest main gears; f_1 and f_2 – tractor rolling resistance coefficient when tractor works at nominal and minimal torque on hook; η_{tr} – transmission ratio coefficient; $\gamma_{\partial\min}$ – permissible minimal engine load coefficient, usually $\gamma_{\partial\min} = 0.85$; δ_t – tractor torque range; r_r – wheel rolling radius, m.

For increasing performance of tractor aggregates need to approach the maximum tractor rate values, which is allowable by modern agro-engineering of agricultural machines. To set up whatever marginal values is not purposeful, because continuous agricultural process is creating new tractor working speed possibilities.

For different tractor works the driving force should be close to grip force, then maximum engine power and slippage do not exceed allowable limits. For tractors, under sustainable work conditions on horizontal surface, regulated requirements are expressed by equation $F_v \approx F_\varphi$:

$$P_e \eta_{tr} / v = m_t \varphi g. \tag{14}$$

Close to his value, Deutz Fahr Tractor TTV 1160 driving and grip force dependence on driving speed, under different soil conditions using maximum engine power is presented in Fig. 3.

In Fig. 3 it can be seen, that using nominal engine power of tractor, driving wheels develop driving force close to grip force, when driving speed on icy and tight snow roads is higher than 15 km/h, on cultivated and ploughed field – 9–12 km/h, gravelled, stubble and grassland field – 7–10 km/h, asphalt road – 7–8 km/h. Working at lower driving speed, driving wheel slippage will be very big.

In Fig. 3 it can be seen, that maximum driving force on stubble is from 31 to 40 kN, and driving speed is 10 km/h. Working at lower driving speed, maximum driving force will not be used, because tractor will exceed the slippage limit. To use all engine power at low driving speed so that driving wheels are not slipping, tractor has to be loaded with weights, to lower wheel air pressure

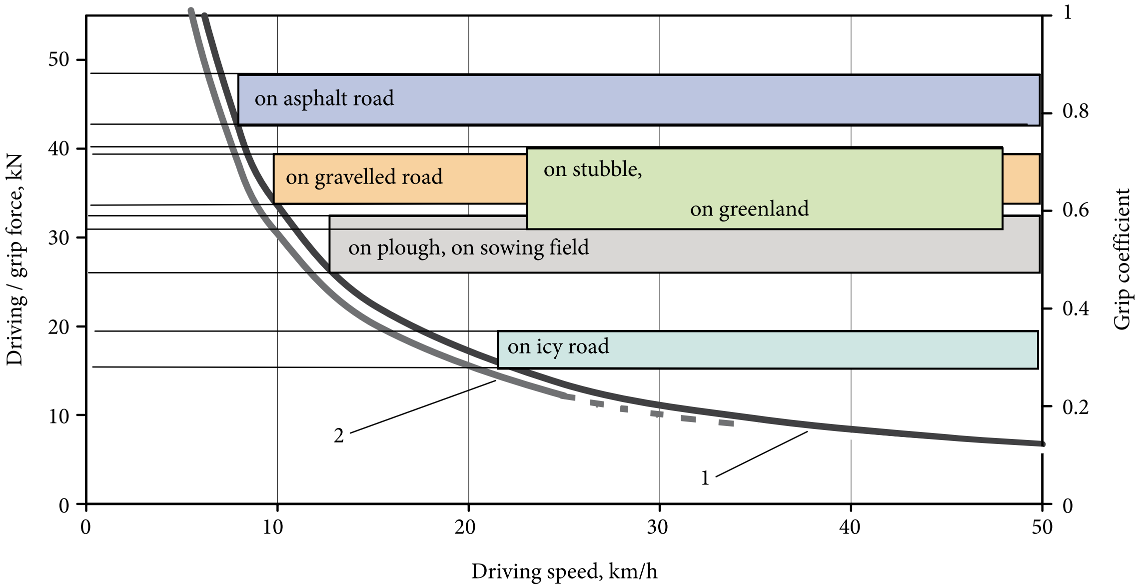

Fig. 3. Driving and grip force dependence on driving speed, under different soil conditions using maximum engine power: 1 – tractor maximum driving force, 2 – soil grip force

or use other devices. Otherwise, because of big slippage, tractor would not move from its starting position.

At transport work, tractor maximum engine power could be used by optimally choosing weight and driving speed of tractor transport aggregate. Weight and speed dependence could be expressed by equation:

$$P_e \eta_{tr} / v = (m_t + m_p) fg. \quad (15)$$

Tractor traction condition depends not only on engine dynamic factors and transmission parameters, but also on exploitation of driving wheel factors and soil conditions we are driving on, physical–mechanical factors as well on interaction of driving wheels with exploitation. The same factors have influence on trailer traction resistance. To calculate traction resistance of trailer we must know its weight m_p and coefficient of rolling resistance *f*. It depends on wheels and soil conditions, working conditions and driving speed. Rolling resistance coefficient is increasing as driving speed increases. At low speed, with nominal tyre load and air pressure, on smooth road, coefficient *f* varies insignificantly (Fig. 4), see Litvinov, Farobin (Литвинов, Фаробин 1989) and Nagaoka *et al.* (2001).

Intensive change, depending on tyre type and pressure, starts when driving speed reaches 50 km/h and more. For calculating rolling resistance by estimating driving speed, there are various empirical formulas. A suitable formula for practical use is:

$$f = f_0 + k_f v^2, \quad (16)$$

where: f_0 – rolling resistance coefficient at low speed; v – driving speed. In case, when coefficient of k_f value is not known, Litvinov, Farobin (Литвинов, Фаробин 1989) recommended to accept $k_f = 7 \cdot 10^{-6}$.

Dependence of rolling resistance coefficient *f* on the tyre air pressure p_o is different according to variety of roads. Rolling resistance coefficient *f* on hard surface roads

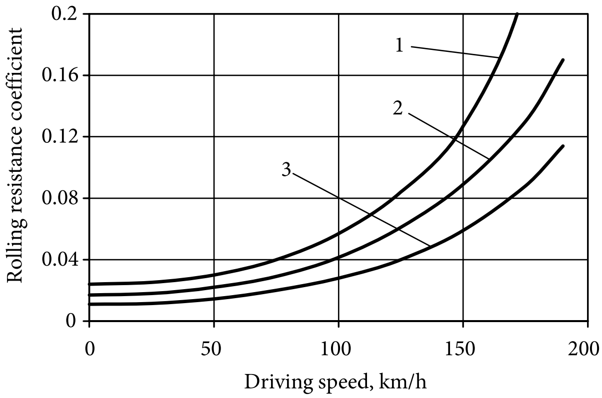

Fig. 4. Dependence of rolling resistance coefficient on driving speed, when air pressure in the tyre: 1 – 0.15; 2 – 0.25; 3 – 0.3 MPa

is bigger, when air pressure in the tyre is lower. However, too high tyre pressure, because of uneven wheel and road bigger dynamic load interaction, as often as not increases coefficient of rolling resistance, see Litvinov, Farobin (Литвинов, Фаробин 1989), Wong (1989), Upadhyaya *et al.* (1997) and Jun *et al.* (1998). Lower air pressure in the tyre while driving on a field increases rolling resistance coefficient *f* because of bigger tyre deformation. And, on the other hand rolling resistance coefficient *f* is decreased because of lower field. Always we can choose such air pressure p_o (optimal air pressure) when coefficient *f* will be of minimum value (Fig. 5).

Rolling resistance coefficient *f* varies, when wheel vertical load and developed torque M_v changes. On hard surface roads, when vertical wheel load varies in the range of 80–110 % of nominal load limit, rolling resistance coefficient *f* varies insignificantly. Overloading motoring wheels by 20 %, rolling resistance coefficient *f* increases by around 5 %. Vertical wheel load has sig-

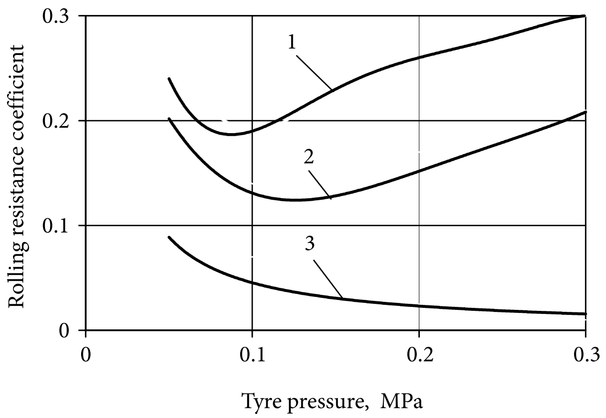

Fig. 5. Dependence of rolling resistance coefficient on tyre air pressure: 1 – on sand; 2 – on plough; 3 – on asphalt road

nificant influence on rolling resistance coefficient f for low pressure tyre (tractors) and for roads distorted by wheels. For calculating of rolling resistance coefficient f, estimating driving torque value, we could use Skotnikov's *et al.* (Скотников *et al.* 1986) formula:

$$f = a_p / r_d + M_v(r_{rv} - r_r) / R_z r_d r_{rv}, \quad (17)$$

where: a_p – distance between wheel vertical axis and attached point of vertical reaction force; r_d – radius of dynamic wheel; r_r – wheel rolling radius; r_{rv} – rolling radius of driven wheel; R_z – wheel vertical reaction force; M_v – wheel driving torque.

One of the first exploitation factors of tractor transport aggregate is its driving speed.

It depends on variable quantity: crankshaft angular velocity ω, slippage of driving wheels δ, driving wheels rolling radius r_v, transmission ratio i_{tr}.

$$v = \omega r_v (1 - \delta) / i_{tr}. \quad (18)$$

The biggest influence of driving speed and driving force dependence on road/field conditions, using tractor engine nominal power, is presented in Fig. 6.

Transport aggregate, made of Deutz Fahr Tractor TTV1160, maximum power use dependence on driving speed and total aggregate weight is shown in Fig. 6. Driving on stubble with 15 t aggregate, its driving speed would be from 11 to 17 km/h, while with the same aggregate weight and using economical engine working conditions, on asphalt road its maximum driving speed would be from 30 to 50 km/h.

Use of stepless transmissions allows to get different speeds in different conditions, as well to increase minimum engine load. So, tractor could work in optimal conditions, close to nominal, that means with higher tractor output and economy. Practical use of stepless transmission advantages for tractors can be done by implementing operating systems, which allows automatically change transmission ration according to resistance of tractor transport aggregate.

To evaluate changes of engine load, during its work, required tractor engine power necessary to reserve for overcoming systematically uneven driving resistance. Part of engine reserve could be used for tractor aggregate run-up, without gear shifting.

Tractor aggregate should be combined so that average torque value would be a little lower than engine torque at the same engine speed. Ratio of indicated torque is called tractor engine running load coefficient χ_e, which varies from 0.8 to 0.85, by Skotnikov *et al.* (Скотников *et al.* 1986), depending on engine dynamic conditions and variation of tractor driving resistance. A reserve of needed engine power is determined by entering value of indicated coefficient to formula.

So, needed tractor engine power in kW, is counted:

$$P_e = \left[F_n + f_1(m_{e\min} + m_\sigma)g\right]\upsilon_{n1} / (10^3 \eta_{tr} \chi_e). \quad (19)$$

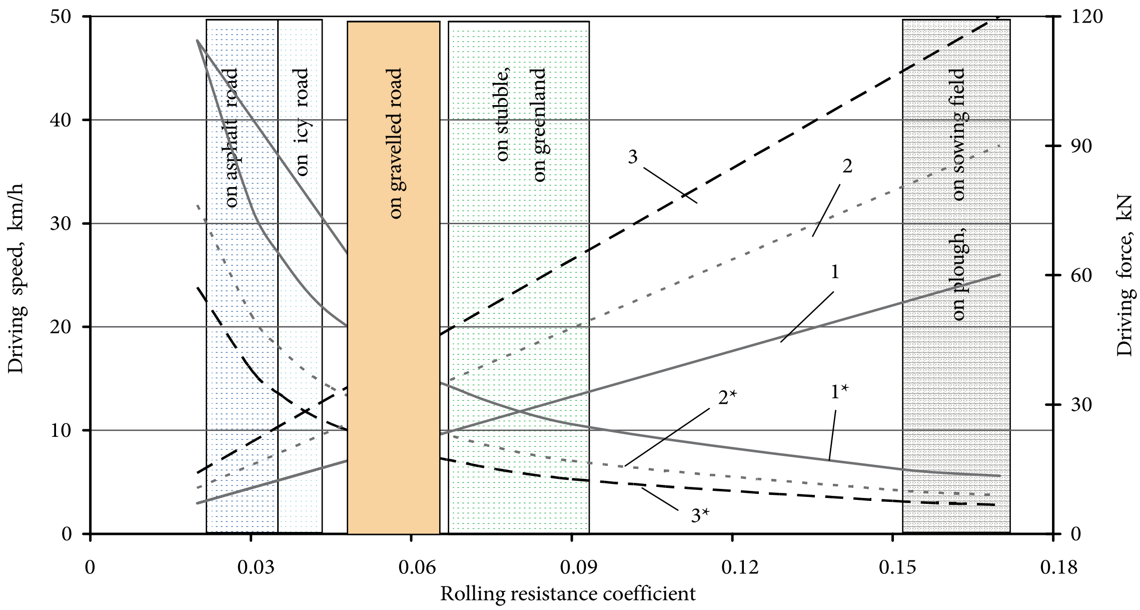

Fig. 6. Deutz Fahr Tractor TTV1160 driving speed and driving force dependence on rolling resistance and total aggregate weight, using nominal power: 1 and 1* – total aggregate weight 15 t; 2 and 2* – 20 t; 3 and 3* – 25 t

For tractor transport aggregates:

$$P_e = \left[f_1 g (m_p + m_{e\min} + m_\sigma) \right] \upsilon_{n1} / (10^3 \eta_{tr} \chi_e). \quad (20)$$

Thanks to this formula the value of calculated required engine power is rounding up, for reason when tractor works in variable conditions engine does not give full power. Ratio between engine power and tractor weight is called comparative tractor power. Comparative power is calculated according to formula, which is presented by Skotnikov *et al.* (Скотников *et al.* 1986):

$$P_{sal} = P_e / m_{e\min}. \quad (21)$$

Comparative engine power P_{sal} is a very important factor for tractor power estimation. Increasing of comparative tractor engine power allows to increase working speeds.

3. Experimental investigation

All test measurements were carried out using a Deutz Fahr Agrotron TTV 1160 (114 kW/155 hp) and trailer – OZTP-9554. The tractor scaled 6 410 kg without extra weights, with 41 % (2 630 kg) resting on the front axle. Down below, the tractor was shod with 540/65R28 and 650/65R38 sized Continental Contract AC 65 front and rear tyre. Trailer's OZTP-9554 weight is 4 800 kg, number of axles – 3. For the experiment, trailer has been loaded with gravel.

All test measurements were conducted on field stubble and on gravelled road. Field of testing plot was smooth and soil structure almost not changing: moisture at 15 cm depth – 18 %, rigidity – 1.05 MPa. and moisture at 20 cm depth – 18.5 %, rigidity – 1.09 MPa. For testing straight and horizontal, in good condition, not bumpy swatch of gravelled road was chosen. Test was done on the same swatch and driving was performed forward and in reverse. Test quantity – 6. Measurements included engine speed, fuel use, forward speed, wheel slip and draft required. Draft was calculated by relating pulling power to travel speed.

4. Experimental investigation results of tractor's economic and dynamic indicators

Tests were done on stubble and gravelled road. Engine revolutions, fuel consumption, driving speed, wheel slippage and needed pull were measured. Pull force were calculated according to pulling force and driving speed.

For first test, trailer has been loaded with gravel, making a combined total of 18 t of unit weight. Tests were conducted on gravel road by increasing driving speed from 2 km/h up to 37 km/h, and by keeping steady engine speed at 1800 rpm. During the test measurements of fuel consumption, slippage and pulling force at different driving speed records were taken. Test results are shown in Fig. 7.

The figure shows, that gradually increasing driving speed of tractor transport unit from 2 km/h up to 37 km/h, hourly fuel consumption increases from 6 l/h up to 25 l/h. Pulling force during driving speed increase almost did

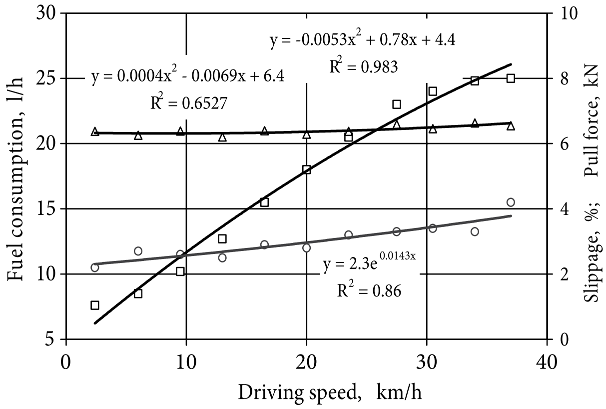

Fig. 7. Fuel consumption, pull force and slippage dependence on driving speed

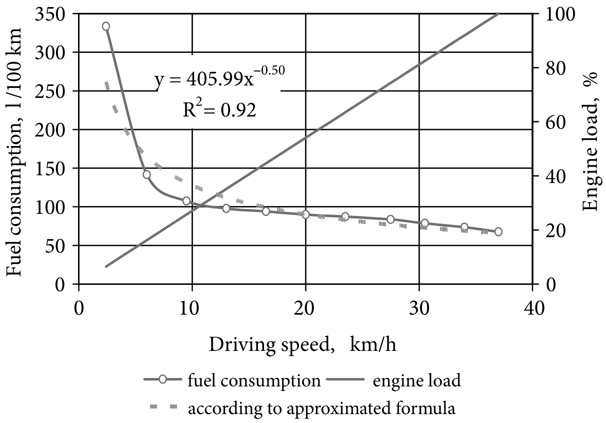

Fig. 8. Dependence of engine load and fuel consumption on driving speed at transport work

not change. At tested driving speed range, percentage of slippage increased slightly.

Hourly fuel consumption of transport unit is recalculated to fuel consumption in l/100 km. From diagram (Fig. 8) of engine load and fuel consumption at transport work it can be seen, that fuel consumption of 330 l/100 km at driving speed of 2 km/h was determined. Then engine has worked at very low torque. At chosen steady engine speed (1800 rpm) most economical working condition was reached. At driving speed of 34–37 km/h, fuel consumption was 70 l/100 km, and engine load has reached its maximum value – 100 % of engine load.

Test of transport unit has been done when maximum engine power was not needed. Fuel consumption varies considerably depending on engine speed. Many tractors can reach optimal fuel consumption at reduced engine speed. To determine how much fuel could be saved when tractor works at engine speed of 2 300 rpm and after when engine is working at 1 800 rpm, test measurements on stubble (driving speed – 8 km/h) and gravelled road (driving speed – 18 km/h) were taken. At different engine speed the same driving speed was reached and 4.5–5 % of fuel saving was determined (Fig. 9).

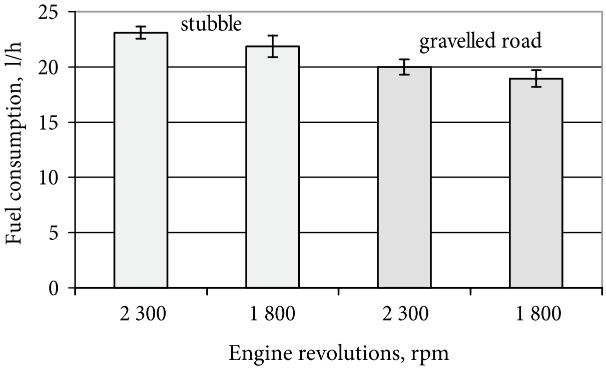

Fig. 9. Fuel consumption at different engine speed, when driving speed on stubble – 8 km/h, on gravelled road – 18 km/h

5. Conclusions

1. The tractor optimum fuel consumption is at lower engine revolutions than at rated revolutions.
2. By using tractor Deutz Fahr Agrotron TTV 1160 engine nominal power, driving wheels develop driving force close to grip force, when driving speed on the icy road is higher than 15 km/h; on field prepared for sowing, on plough field, on stubble and on gravelled road – 8–10 km/h. At lower tractor driving speed the driving wheel slippage would be bigger.
3. During tractor transport works under different road and field conditions, economical engine working conditions or nominal power could be fully used by choosing suitable unit proportion and driving speed.
4. Working at reduced engine speed can cut the fuel consumption by 5 %.

References

Air pressure, weight and fuel consumption: Diesel savings of 10 %. 2005, *Tractor profit and farm machinery* 9: 20–23.

Carrol, E. 1992. *Gearing engine and tractor power*: ASAE Text Book No. 7. Michigan. 539 p.

Giedra, K.; Janulevičius, A. 2005. Tractor ballasting in field transport work, *Transport* 20(4): 152–158.

Ivanov, R.; Rusev, R.; Ilchev, P. 2006. Laboratory investigation of tyre sliding grip coefficient, *Transport* 21(3): 172–181.

Jun, H.; Kishimoto, T.; Way, T. R.; Tauiguhi, T. 1998. Three-directional contact stress distributions for a pneumatic tractor tire on soft soil, *Transactions of the ASAE* 41(5): 1237–1242.

Kraujalis, A. 2002. Pavarinių traktorių agregatų degalų sąnaudų analizė [Analysis of fuel input of driving tractor units, *Agricultural engineering*] 34(3): 35–42.

Nagaoka, A. K.; Lancas, K.; Castro Neto, P.; Benez, S. H. 2001. Evaluation of single wheel testing device with mechanical transmission. *ASABE technical library*. USA. California. 13 p.

Neunaber, M. 1997. Correct ballast boost draft by 20 % or more, *Profi* 10: 46–49.

Prentkovskis, O.; Bogdevičius, M. 2002. Dynamics of a motor vehicle taking into consideration the interaction between wheels and road pavement surface, *Transport* 16(6): 244–253.

SAME DEUTZ-FAHR Agrarsystem GmbH. 1130 / 1145 / 1160. Agrotron, Lauingen, Germany. 2002. 15p.

Upadhyaya, S.; Sime, M.; Radhuashi, N.; Adler, B. 1997. Semi-empirical traction prediction equations based an relevant soil parameters, *Journal Terramechanics* 34(3): 141–154.

Вонг, Д. Ж. 1989. Теория наземных транспортных средств. [Wong, J. Y. *Theory of ground vehicles*]. Москва: Машиностроение. 237 с.

Литвинов, А. С.; Фаробин, Я. Е. 1989. Автомобиль. Теория эксплуатационных свойств [Litvinov, A. C.; Farobin, J. E. *Automobile. Theory of performance*]. Москва: Машиностроение. 237 с.

Скотников, В. А.; Мащенский, А. А.; Солонский, А. С. 1986. *Основы теории и расчета трактора и автомобиля* [Skotnikov, V. A.; Mashensky, A. A.; Solonsky, A. S. *Theory and count of tractor and automobile*]. Москва: Агропромиздат. 384 с.