

Improving the energy efficiency of tractors in agricultural operations

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Abstract. The article proposes and substantiates a method for indirect determination of the energy parameters of the operation of an agricultural tractor engine, which makes it possible to calculate and establish the most energy-efficient mode for performing an agricultural operation. The proposed method for determining the energy parameters of the engine operation by measuring the turbocharging pressure, with high accuracy, is highly informative and has a number of advantages compared to traditional methods for instrumental control under the working conditions of agricultural tractors and machines.

1 Introduction

Agricultural tractors work in combination with implements mainly as tractors and energy sources through the power take-off shaft, thus forming a mobile power tool that should provide high performance and efficiency when performing the entire range of agricultural work, with the appropriate quality and within the established agrotechnical terms. At the same time, the output performance of the machine-tractor unit is significantly affected by such characteristics as: the size and complexity of the configuration of the site, the length of the rut, macro- and microrelief, clogging with stones, soil density and moisture, etc [1-5].

Due to ever-increasing engine power on the one hand, and legal or soil-physical restrictions on size or weight on the other hand, effective thrust limits are increasingly being reached. About 40% of the losses that occur between the crankshaft and the work tool are due to rolling resistance and slip during typical traction work in the field. Modern transmissions are at a high technical level and contribute only about a quarter of the total losses. Auxiliary and service units, depending on their purpose, have a similar order of magnitude [6-9]. The distribution of power losses of an energy-saturated tractor when working with a heavy cultivator in stubble cultivation is clearly shown by the Sankey diagram presented in Figure 1 [9]. Thus, only 20% of the energy stored in diesel fuel remains as traction for the implements.

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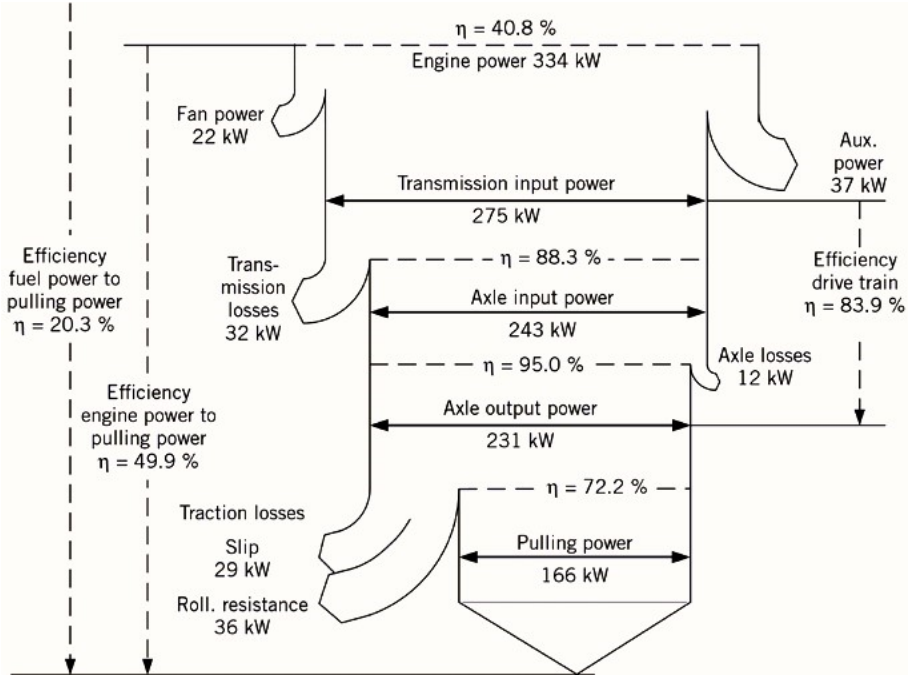


Fig. 1. Sankey-diagram for a large tractor operating at heavy pull.

One characteristic of diesel engines is the variation in energy efficiency at different load and speed modes (Figure 2). Consequently, when conducting less energy-intensive tasks at a load level different from the standard one, fuel efficiency decreases further.

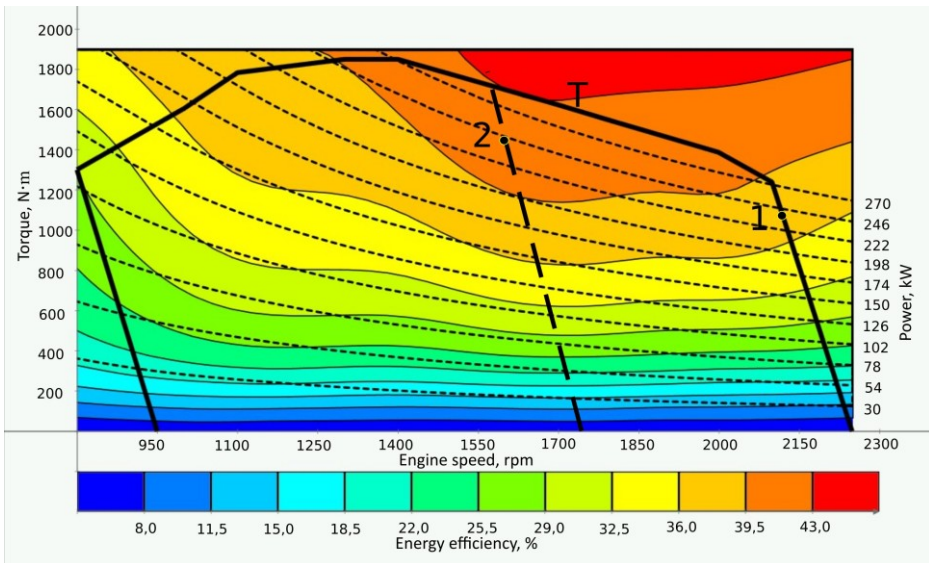


Fig. 2. Energy efficiency of diesel engine.

In this case, for tractors with stepped transmissions, it is recommended to use the SUTB (shifted-up and throttled-back) technology, which allows you to maintain the same speed in different gears and implies a decrease in the crankshaft speed when switching to a higher transmission stage and vice versa [10]. To do this, the operator must independently assess

the energy demand necessary to perform the technological operation and compare it with the power, fuel and economic characteristics of the engine and a possible range of transmission ratios. Then, taking into account the restrictions imposed by the requirements for the quality of the agricultural operation, make a decision on the possibility of increasing the speed of the unit, or on the possibility of reducing the crankshaft speed and switching to an increased transmission stage to maintain the same speed. As input data for such an analysis, the operator can use information obtained from the instrument panel of the tractor (engine crankshaft speed, set speed of the tractor), as well as information obtained by the organoleptic method (tonality and character of the engine sound, color and smoke of exhaust gases), motion uniformity, vibration change, etc.), while the effectiveness of analysis and subsequent decision-making is subjective and largely depends on the qualifications and experience of the operator. It is possible to increase the efficiency of decisions made by him only with the help of available means and methods of instrumental control.

2 Materials and methods

Direct measurements of engine torque require the installation of torque devices on the drive shafts and, in most cases, are not available to the average user. Therefore, in order to solve the problem, the author carried out studies of parameters, the measurement of which, on the one hand, does not present significant difficulties, on the other hand, it allows to give a fairly accurate assessment of the degree of use of engine power and its current energy efficiency.

3 Results

At present, for energy-saturated agricultural tractors, diesel engines with gas turbine supercharging are widely used, which have intermediate cooling of the air entering the cylinders. When implementing such a boost scheme, the energy of the exhaust gases is used to drive the turbine wheel mounted on the same shaft as the compressor (supercharger) wheel. Thus, the turbocharger rotor rotates at a high frequency, providing compression and air supply to the engine cylinder. Since the turbocharger rotor in such a scheme is not kinematically connected to the crankshaft, the amount of air supplied by the compressor also changes with a change in engine load. Thus, the boost pressure p_k , is characterized by the presence of a stable, for a given engine, connection with one of the basic performance indicators of an internal combustion engine - the mean effective pressure p_e , MPa. The mean effective pressure is a virtual indicator and characterizes the pressure value in the engine cylinder, which must be maintained constant at any position of the piston during various parts of the working cycle and at which the work carried out in it in one stroke will be equal to the effective work for the entire cycle:

$$p_e = \frac{W_e}{V_h} \tag{1}$$

W_e – effective cycle work, J; V_h – cylinder volume, cm³.

The relationship of the mean effective pressure p_e with the engine torque T_e is determined by the formula:

$$M_e = \frac{p_e \cdot i \cdot V_h}{0.314 \cdot \tau_d} \tag{2}$$

τ_d – engine cycle, (2 or 4); i – number of cylinders.

The relationship of the mean effective pressure and torque with the engine power is determined by the formula:

$$N_e = \frac{p_e \cdot i \cdot V_h \cdot n_e}{0.3 \cdot \tau_d} = \frac{T_e \cdot n_e}{9550} \quad (3)$$

n_e – engine speed, rpm.

Since, from a mathematical point of view, for a given engine, the average effective pressure can be represented as an empirical function of boost pressure p_k (from a physical point of view, the direction of the relationship is opposite), taking into account the above formulas, it becomes possible to write:

$$p_e = f(p_k) \quad (4)$$

$$T_e = f(p_e) = f(p_k) \quad (5)$$

$$N_e = f(T_e, n_e) = f(p_e, n_e) = f(p_k, n_e) \quad (6)$$

As a result, knowing the characteristics of the engine, it becomes possible, by measuring the boost pressure p_k , during operation, to determine the current values of the related indicators (Figure 3).

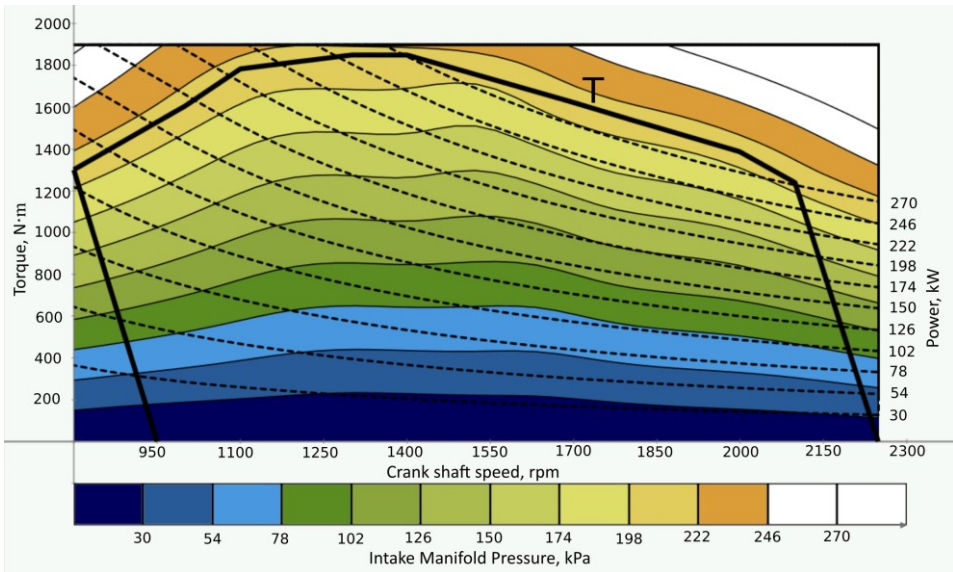


Fig. 3. Determining the current performance of the engine from a known characteristic.

Formulas (1), (2) and (3) show that the average effective pressure characterizes only the degree of forcing and its values for different engines of the same type, regardless of their power, will be in the same range. Mean effective pressure values for piston engines of various types are presented in Table 1.

Table 1. Mean effective pressure values for different types of reciprocating engines.

Engine's type	Mean effective pressure p_e , MPa
Four-stroke carbureted engines	0.6 – 1.1
Forced four-stroke carburetor and engines with electronic injection	up to 1.3
Four-stroke naturally aspirated diesel engines	0.65 – 0.85
Four-stroke supercharged diesel engines	up to 2.0
Two-stroke high-speed diesel engines	0.4 – 0.75
Gas engines	0.5 – 0.75

The values of turbocharging pressure p_k also for industrial diesel engines do not exceed 300 kPa (the most common engines, the maximum boost pressure of which is in the range of 200 ... 250 kPa).

The energy potential of the engine available for use at the current time at any position of the engine speed regulator control can be determined using the engine power utilization factor K_N , determined by the formula:

$$K_N = \frac{N_{e(curr)}}{N_{e(max)}(n_{e(curr)}, \Delta_p)} \tag{7}$$

$N_{e(curr)}$ – current consumed effective power, kW; $N_{e(max)}(n_{e(curr)}, \Delta_p)$ – maximum effective power developed by the engine according to the regulatory branch of the regulatory characteristic at the current position of the control element of the engine speed controller, kW; $n_{e(curr)}$ – current engine speed, rpm; Δ_p – the degree of unevenness of the speed controller, determined by the formula:

$$\Delta_p = \frac{2 \cdot (n_{xmax} - n_p)}{n_{xmax} + n_p} \tag{1}$$

n_{xmax} – maximum engine speed without load, rpm; n_p – crankshaft speed at which the engine develops 90% of the maximum power when operating in the control section of the control characteristic, rpm.

Since the empirical dependences (4), (5) and (6) for a given engine are unchanged, then at any time, for any position of the engine regulator control, knowing the current values of the turbocharging pressure $p_{k(corr)}$ and engine speed $n_{e(corr)}$, it becomes possible to determine the engine power utilization factor:

$$K_N = \frac{p_{k(corr)} - p_{k(min)}(n_{e(corr)}, \Delta_p)}{p_{k(max)}(n_{e(corr)}, \Delta_p) - p_{k(min)}(n_{e(corr)}, \Delta_p)} \tag{9}$$

$p_{k(max)}(n_{e(corr)}, \Delta_p)$ – turbocharging pressure at maximum power developed by the engine according to the regulatory section of the regulatory characteristic at the current position of the control element of the engine speed controller, kPa; $p_{k(min)}(n_{e(corr)}, \Delta_p)$ – turbo boost pressure during engine idle operation ($N_e = 0$) in the regulatory section of the regulatory characteristic at the current position of the control element of the engine speed controller, kPa.

Compared to the direct torque measurement method, this method has a number of advantages:

- The method also allows you to determine the instantaneous values of the target parameters.
- To measure the boost pressure, it is not required to make changes to the engine design, which may affect its output parameters and functionality.
- To measure the boost pressure of most engines, you can use one type of pressure gauge with one measuring range.

To assess the accuracy of determining the degree of engine load according to the developed method and substantiate the metrological characteristics of the measuring tool, it is necessary to calculate the maximum allowable measurement error limit.

When determined in accordance with formula (9), the engine power utilization factor K_N is an indirect value determined by three independently measured quantities ($p_{k(corr)}$, $p_{k(max)}$ and $p_{k(min)}$), herefore, the value of the resulting absolute error ΔK_N will be determined by the sum of the products of partial derivatives of the first order with respect to each argument and the absolute errors of the measured values. Since each of these measured quantities will be determined by the same means of measurement, we will accept it as the same. In this case, the resulting error in determining the engine load factor will be determined by the formula:

$$\Delta K_N = \sqrt{\left(\frac{1}{p_{k(max)} - p_{k(min)}} \cdot \Delta p\right)^2 + \left(\frac{p_{k(corr)}}{(p_{k(max)} - p_{k(min)})^2} \cdot \Delta p\right)^2 + \left(\left(\frac{p_{k(corr)} - p_{k(min)}}{(p_{k(max)} - p_{k(min)})^2} - \frac{1}{p_{k(max)} - p_{k(min)}}\right) \cdot \Delta p\right)^2} \quad (10)$$

Δp – pressure gauge accuracy, kPa.

Knowing the required accuracy of determining the engine power utilization factor and the possible measurement ranges, it becomes possible, by solving equation (10) with respect to Δp , to determine the required permissible error of the measuring instrument. To simplify the solution of this problem, a contour diagram was built (Figure 4), thanks to which it is possible to determine both the compliance of the metrological characteristics of the pressure gauge with the requirements for the accuracy of determining K_N , and the resulting accuracy of determining K_N when using measuring instruments with known characteristics.

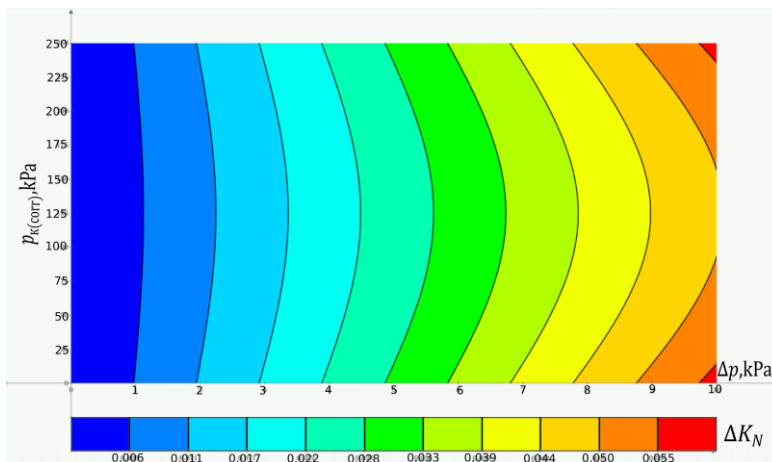


Fig. 4. Relationship diagram of $p_{k(corr)}$, Δp , and ΔK_N values.

4 Discussion

The calculation of alternative combinations of engine speed and gear ratios (stages) of the transmission, depending on the strategic goals defined by the MTU operator, can pursue one of the following goals:

- Achieve the highest possible performance.
- Achieving the maximum possible fuel and energy efficiency while maintaining the current speed.

The solution to this problem is to find such a point "1" on the combined characteristics of the engine (Figure 2), which provides the necessary power reserve, but increases energy efficiency. Since this changes the speed of the engine crankshaft, in order to provide the necessary speed, it is necessary to correspondingly change the gear ratio of the transmission (switch to another stage). The ranges of possible theoretical travel speed (excluding slippage) in various gears are determined by the tractor design and are given in the operation manual in the form of diagrams or tables. Using the data of these tables, the value k' , obtained by the ratio of the actual speed v'_{corr} to the engine speed n'_e when the tractor is moving without load (with zero slip) (formula 11), the value of normal tire deformation f (формула 12) and slipping δ and solving equation (13) with respect to v_{corr} , it becomes possible to determine the expected speed of the unit at any speed of rotation of the engine crankshaft for any transmission stage. It is possible to determine the required speed of rotation of the engine crankshaft to achieve the set speed of the unit by solving equation (13) with respect to n_e .

$$k' = \frac{v'_{corr}}{n'_e} \tag{11}$$

$$f = \frac{C_2 \cdot G_W}{2 \cdot (p_W + p_0)} + \sqrt{\left[\frac{C_2 \cdot G_W}{2 \cdot (p_W + p_0)} \right]^2 + C_1 \cdot G_W} \tag{12}$$

$$\delta = \left(1 - \frac{v_{corr}}{k' \cdot n_e} \right) \cdot 10^2 \tag{13}$$

G_W – wheel load, kN; p_W – internal tire pressure, kPa; C_1, C_2, p_0 – constant coefficients for a given tire, kPa.

The task of determining point "2" on the combined characteristic of the engine is easily solved by a graphical method. For the possibility of an analytical solution, the relationship between the parameters of the engine characteristic should be presented in the form of functional dependencies. Since the indicated relationship of engine parameters is empirical and is not described by simple mathematical functions, it is also convenient to use the LOESS method to represent it in an analytical form [11-12].

5 Conclusion

The proposed method for determining the energy parameters of the engine by measuring the turbocharging pressure, with high accuracy, has high information content and a number of advantages compared to traditional methods for instrumental control under the working conditions of agricultural tractors and machines. Unlike traditional methods and tools, boost measurement does not require serious intervention in the engine design, complex and expensive equipment, and highly qualified service personnel. Practical experiments on the

application of the proposed method made it possible to reduce energy costs in the performance of various agricultural operations within 15–30 MJ/ha and increase the energy efficiency of agricultural operations by 12–14%.

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