Energy Condition of Gear Shifting under Load in Commercial Vehicles

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Abstract: In the fight against global warming, all increase in energy efficiency contributes to reaching the target numbers. This is especially true for agriculture, where a huge number of large vehicles work. Currently, used modern tractors are equipped with powershift transmissions, which allow for shifting underload. In the case of tractors, owing to the large traction force and low speed of tractors, kinetic energy loss during shifting is high, which reduces efficiency. The traction test presented here aimed to analyse the shifting process of a tractor from an energetic perspective. It was a challenging task to select the appropriate measuring devices and to create a measurement system for this purpose, due to the extremely short (0.3-0.5 s) shifting time. In order to be able to analyse the parameter changes and the cause-and-effect relationships between speed and engine revolution, and between traction force and slip in the shifting process. Two scenarios were examined: fixed speed and fixed traction force. Data characterizing shifting while high traction force is exerted was analysed by methods of mathematical statistics. The unit changes of traction force and slip and the required time were analysed. Traction force increased after shifting (+ 5 kN), which indicates that extra traction force is needed to make up for speed lost during shifting, and because of which slip is considerably increased, typically by more than 10%. After shifting, traction force vacillates at a frequency around 2 Hz, similarly to the vacillation of engine revolution. When changing down, the process starts with a decrease in slip. After almost 0.15 s, traction force plummets by 15 kN at the beginning of shifting. At the same time, slip also decreases, for a short time it can also exhibit negative values. The analysis proved that slip continuously causes considerable losses to various degrees during shifting. In order to minimize these losses, recommendations are given both for driving technique and for the parameter settings of the controlling software.

Keywords: transmission; gear shifting; agricultural vehicles

1 Introduction

Efficient energy consumption is a crucial factor in the development and operation of agricultural engines. According to studies, the correct specification of fuels can influence the fuel consumption and emission of commercial vehicles [1]. However, irrespective of the applied fuel mixture, the aim is to maximize the efficiency of energy conversion in the powertrain [2].

Furthermore, the need is growing to improve the utilization rate, the energy consumption [3] and the compliance to environmental expectations of tractors. Consequently, besides the constant development of engines and the addition of various biofuels [4], it is also a requirement to increase the energy-efficiency of tractor powertrains. Efficiency and emission are interconnected: other publications surveyed the air pollutant emission of agricultural engines [5]. They proved that the optimal load of the engine exercises a significant effect on decreasing the particle matter (PM) emission of diesel engines (smoke value) [6]. In the case of agricultural heavy-duty vehicles, the optimal load can be achieved by ensuring optimal transmission. Battery electricity is under research as well as energy provider for agriculture, but it has several barriers [7].

During the operation of a tractor, the choice of the setup of the power-shift gear and the actual transmission rate (both for automatic and manual transmission) influence engine load. As a result, during development and operation, the powertrain of tractors must be examined as one system from the engine to the soil. Results in the field of energetics and operation can only be reached if the modifications and developments of the various elements of the transmission system are coordinated. This development direction follows one of the major trends of the mobility: i.e. the increasing artificial and human cognitive level [8]. A design method for a new power-cycling hydro-mechanical continuously variable transmission (PCHMCVT) system is proposed to solve it [9].

After the creation of powershift and continuously variable transmissions (CVT), the aim was to create the simplest possible transmission technologies while keeping the possibility of shifting underload [10]. While earlier the main goals of development were optimizing the utilization of engine power and the enhancement of driver comfort, today the major aim is to minimize power loss. After examining the characteristics of tractor CVT systems through precise simulations, it was proved that tractor CVT systems can be used in a wider range of fields, and they ensure the lowest specific fuel consumption owing to the optimal load of the engine [11]. A deeper analysis [12] proved that for the power-split infinitely variable transmission (IVT) systems applied in tractors, the best power transmission efficiency (η = 0.90–0.94) is witnessed if the infinitely variable member's transmission is in the range of i = 0.85-2.4. The structural build-up of mass-produced power-shift CVT systems and the relevant transmission ranges were compared in mass-produced tractors [13]. The power dissipation was explored by preliminary calculations and real-life tests, which gave important information for the evaluation of power loss of different constructions.

Although hydrostatic transmission offers the best regulation possibilities for vehicles moving in the fields, its efficiency is lower that that of mechanical transmission [14, 15]. Therefore, hydrostatic transmission is primarily used in slow harvesters or as the IVT member of power-split powershift systems in agricultural engineering. In a study on power dissipation of tractors [16], it was revealed that especially at higher speed, significant power loss is due to the viscosity of the lubricant and the operation of the transmission's own hydraulic system, besides the friction of the parts of the mechanical transmission system. It is also mentioned that the even the unloaded powershift transmission's own hydraulic system increases the frictional and inertial resistance, which are altogether responsible for 52% of losses in the powertrain. Furthermore, 40% of losses arise from oil splashes and the dynamic viscosity of the oil. According to the results of powershift transmission tests, a 10% decrease in the amount of oil in the system resulted in a 13% decrease in frictional resistance. Indirect tests showed that more than half of the losses in the hydraulic system arose in the final reduction and the resistance in the brake plates that are submerged in oil.

For the operation and control of clutches, it is important to know the processes present in the operating system. Model calculations based on simulations [17] proved that during the optimization of the control of pneumatic clutches, pneumatic control might result in a 100 ms delay. They examined the release and the closure of the pneumatic clutch in several scenarios (e.g. dynamic acceleration, normal operation), taking into consideration the operation of the control valve and the filling and emptying of the operating cylinders.

Research [18] presents results on agricultural vehicle tractive performance on an asphalt road. Under typical working circumstances, the tractor tractive efficiency depends on its rolling efficiency when the tractive force is smaller, and depends on its slip efficiency when the tractive force is larger. As the variation range of slip coefficient turns to be minor, and the variation trend of slip coefficient becomes understandable when the gearbox of the tractor is switched from low gear to high.

Considerable losses emerge at the soil–wheel interaction as well [19]. During their survey of rolling resistance, special emphasis was given to the determination of work necessary for the deformation of the soil [20].

When analysing the shifting processes, it is important to determine the kinetic energy losses of the machine group, as it influences the quality of work through the lowering of work speed. Earlier studies [21, 22] revealed that the machine group that had slowed down during gear shifting, had to be accelerated again, which increased unit energy consumption. In order to increase shifting comfort and to optimize performance transmission during gear shifting, a number of new criteria and solutions have arisen in the design of transmissions [23, 24]. It was proven [25] that the actual speed of the tractor must be taken into consideration for the programming of the shift duration to ensure the continuity of performance transmission during shifting.

Besides classic field tests for agricultural heavy-duty vehicles and models and simulations, road acceleration tests also emerged [26]. By applying different constructions of power-shift transmissions, the traction performance of tractors was determined [27] in public road traction tests. They also determined the efficiency of the powertrain and the losses due to running resistances with the help of engine mapping curves. One of the three tractors examined had a powershift transmission, while the other two were equipped with CVT powershift transmissions, and their performance was comparable. The best efficiency was 85%, achieved by one of the CVT transmissions, while the other two types produced 64 and 65%.

The changes in the torque characteristics of powershift transmissions of tractors were examined in detail [28], especially concerning the soil-dependent periodicity of the motor torque load, which influences the tractor wheel slip. With the slip growing, the shifting process is accompanied by the kinetic energy loss of the tractor+trailer combination, i.e. by a decrease in work speed. To eliminate this loss, if the engine's revolution gets out of the sensitivity range, the resulting torque change might result in shifting. It is only possible to fulfil this requirement if the total inertia torque is minimally decreased, i.e. during a partial torque transmission it is advisable to change the transmission rate, i.e. to shift gears. A detailed model and simulation of partial power shift transmission is presented as well [29]. During the research it was observed that the engine load and fuel consumption were directly proportional to the engine load levels. However, it was statistically proved that there was no significant difference between the simulation and measured engine torque and fuel consumption at each load level.

The present study describes a series of field measurements conducted on a tractor+trailer combination with respect to the changes of traction and energetic characteristics during shifting and explores the connections between the influencing factors. The above references proved that for slow energy-consuming processes with high traction power, the traditional choosing of optimal work speed can only rarely be achieved. When the machine group slows down during gear shifting, the arising energy loss must be compensated for, which increases the unit energy need of the work process. This study explores how much energy loss arises in the case of powershift transmissions which can be operated underload. The hydraulic control of the transmission of the examined current tractor works with constant durations, and constant pre-calibrated cross-section through a pulse-width modulation (PWM) directional control valve (DCV). The aim of the research is to explore how optimal these parameters are while changing the traction force load and the work speed. The present article discusses the field traction tests of tractors, including the measurement techniques, the test results and their analysis.

2 Materials and Methods

Tractor speed choice depends on agronomic operations. Work speed determines the size of the possible traction force, whose transmission towards the soil ensures the optimal implementation of agricultural operations at constant speed. In order to optimize the efficiency of the engine's performance, and to ensure the optimal work speed corresponding to high traction force, modern tractors are equipped with powershift transmissions, which are shifted when loaded. Field tests that take the energetic behaviour of the tractor into consideration may be carried out to determine the optimal speed and traction force in the soil–wheel interaction with different agrotechnical attachments.

Tractor tests should be carried out in agrotechnical circumstances where resistance arising from slopes, acceleration and aerodynamic drag need not be taken into account. Therefore, these tests are run in horizontal fields, where soil structure, soil humidity and soil cohesion are homogenous. Aerodynamic drag does not play a significant role in the work speed range of tractors, i.e. this factor can be neglected.

Figure 1

The tractor–brake cart combination used in the field tests

Deceleration arising in the first phase of gear changing is only influenced by horizontal forces (Figure 1). Therefore, based on the balance of horizontal forces, the deceleration value of the first phase can be calculated.

In the second phase of gear shifting, the field machine group's state of motion can change in three ways, hypothesizing that the revolution of the engine stays constant:

(i) it continues deceleration (if changing down, at low rolling resistance);

(ii) speed stays constant (if changing down, at high rolling resistance);

(iii) the machine group accelerates (if changing up).

Our field experiments of tractor–brake cart group focused on traction tests with a maximal engine load. Gear shifting was done by a powerhsift transmission system. Force tests determined the values of traction forces during gear shifting, an analysis of which helped reveal the energetic characteristics of shifting.

The traction tests were carried out in a wheat stubble field in the Cegléd-Cifrakert area of the Dél-Pest Megyei Mezőgazdasági Zrt., Hungary. After harvesting, the land was cultivated by a medium-heavy disc, in 10-15 cm depth. Soil humidity was

20-25%, while soil cohesion was medium. One month after cultivation, the soil compaction was also medium. The examined area is utilized as a plough land, it is plain, and the length of the measurement was more than 500 meters, which is ideal from the testing perspective.

2.1 Tractor Characteristics

A CLAAS ARION 420 tractor was used in the experiment. It had a powershift transmission system with 16 gears, with 4 power shift gears (1–2–3–4) in 4 power shift ranges (A–B–C–D). Shifting between ranges (A–B–C–D) is only possible when unloaded, while gear shifting (1–2–3–4) is possible when loaded. The traction force of the tested tractor running in the field ranged from 10 to 25 kN in the 5 to 18 km/h speed range.

The major technical parameters of the tested tractor are given in Table 1.

Table 1
Technical specifications of CLAAS Arion 420

Name	Data
Type	CLAAS ARION 420 DPS
Vehicle identification number (VIN)	A2114DA/12103714 LZZ009
Year of production	2015
Engine identification number (EIN)	CD4045L216148
Displacement	4525 cm3
Producer	John Deere
Performance	88 kW/2200 f/min
Weight	4900 kg
Back tires	520/70 R38

2.2 Trailer Characteristics

In order to determine the traction characteristics of the tractor, a special brake cart developed by NAIK-MGI was used, in which different load parameters could be set. A similar attachment was used by the Italian CREA-ING laboratory in 2015 [32]. The data characterizing the tractor–brake cart group were determined by force measurements (see Figure 2). The measurement system depicted in Figure 2 was operated, and the data was collected by the data collector computer installed into the brake cart. The major characteristics of the brake cart are given in Table 2.

Table 2

Technical specifications of the A–MAZ brake cart

Technical parameter	Value
Engine performance	400 kW
Braking/traction performance	250 kW
Maximal braking/traction force	150 kN
Operational speed range	0–35 km/h

During field tests, the change in the adhesive force is modelled by extra loading of the brake cart. The regulating algorithm offers two possibilities (freely chosen by the operator):

(i) measurements at constant speed (regulated for speed);

(ii) measurements at constant traction force (regulated for traction force).

The connection of the brake cart and the build-up of the measurement system are shown in Figure 2. Traction force was measured by a strain gauge inside the towing bar, which sent the data to the data collector system.

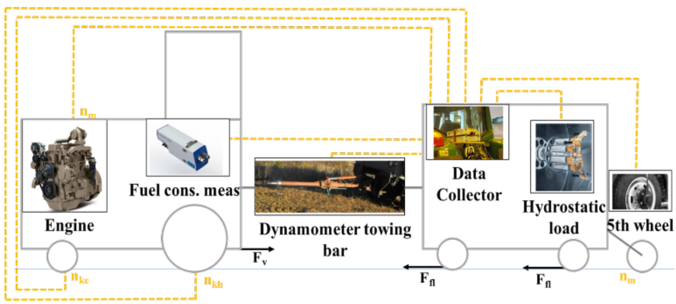

Figure 2

Sketch of the measurement system

The following traction characteristics were recorded by the measurement system shown in Figure 2. The peripheral speed v_k [m/s] of the tractor's driven wheel can be calculated from the revolution n_k measured by the revolution meter in the wheel, and the rolling radius r_g (1).

$$v_k = 2\cdot\pi\cdot r_g\cdot\frac{n_k}{6} \quad v_k = 2\cdot\pi\cdot r_g\cdot\frac{n_k}{60} \tag{1}$$

There is always some slip between the wheel and the soil for vehicles moving in the field, which is reflected by the quantity slip (s). Therefore, the actual speed of the machine group had to be measured. The actual speed was measured by a fifth wheel measuring system attached to the trailer. This calculates the actual speed v_H from the rolling radius and the revolution of the fifth wheel, in a similar manner to (1).

Slip percentage is calculated from the difference between these two speeds with equation (2).

$$s = \frac{v_k - v_H}{v_k} \cdot 100 \qquad (2)$$

where s is slip, v_k is the peripheral speed of the driven wheel, and v_H is the actual speed.

The revolution of the tractor was measured indirectly by an impulse signal transmitter mounted on the PTO of the tractor. The revolution could be calculated from the fixed transmission rates.

Fuel consumption was recorded by an AVL PLU flow meter with a limit of measurement of 600 ℓ/h, which was mounted on the braking attachment, but was inserted into the fuel system of the tractor. In order to determine fuel consumption precisely, measurement cycle times must be set. The difference between the measured flow in the forward and backward going branches of the system gives the quantity of consumed fuel in a unit of time. The accuracy of the fuel consumption meter, which is equipped with a temperature compensator, is ±1%. The measured data was forwarded to the central data collector on the brake cart.

2.3 Data Processing

The signals transmitted by the different signal transmitters were received by a 16-channel measurer and data collector system (SPIDER Mobil). Sampling density was 10 Hz. The collected data were recorded by the CATMAN software, without preliminary filtering. The data were depicted in a chart that conformed to the sampling density, where data lines followed each other at 0.1 second resolution.

From the measurement data, correlation calculations were done by methods of mathematical statistics; while the mean value of dispersed data (xi) was determined automatically by the applied software.

Based on the mean values and the values of standard deviation the dataset was corrected: outliers were eliminated. With this method, the standard deviation of the mean value was reduced. In a second round of calculations, the mean value and the standard deviation was determined again, for the dataset without the outliers. The new values were used for further calculations. Thus, the filtered database represented the measured data properly, with a uniform standard deviation.

During the experiment, the number of test runs for each type of measurement was determined in the following way. If the variance of the population is unknown, the required sample size can be calculated by Chebyshev’s inequality (3).

$$P\left(\bar{x} - k \cdot \frac{\sigma}{\sqrt{n}} \langle \mu \langle \bar{x} + k \cdot \frac{\sigma}{\sqrt{n}}\right) \geq 1 - \alpha \tag{3}$$

In the case of simple random choice, the formula in (3) is simplified to (4).

$$P(\bar{x} - \Delta \langle M(x) \langle \bar{x} + \Delta) = 1 - \alpha$$ $$P\left(\bar{x} - \Delta \langle M(x) \langle \bar{x} + \Delta\right) = 1 - \alpha \tag{4}$$

If equation (4) is rearranged, the required sample size arises (5):

$$n = \frac{t^2 \cdot s_k^2}{\Delta^2}$$ $$n = \frac{t^2 \cdot s_k^2}{\Delta^2} \tag{5}$$

where

n – required sample size;

t – probability constant;

s_k – corrected empirical standard deviation;

Δ – accuracy range.

With the help of equation (5), the number of tests can be analyzed from the perspective of the proper range of standard deviation (i.e., accuracy range). The acceptable accuracy range for a given parameter at a given point at a given setup was determined based on the literature. In order to ensure uniform conditions and to minimize the number of influencing factors, the following measures were taken.

A test site with relatively homogenous surface and structure was selected within the larger test area (as far as this is possible in a ploughland). Test runs were repeated so that a new run would start on unaffected, untrodden 'virgin' soil.

Tests were carried out with the tractor's accelerator being fixed.

At the first round of the tests, the brake cart was set to fixed speed mode (in order to keep the pre-set speed, traction force was modified). In the second round, traction force was fixed. We have no knowledge of other device that could ensure uniform traction force more accurately than this device.

More than 10 seconds were left between shifting gears so that the regulatory cycle of the braking cart could stabilize traction. Therefore, the following shifting took place from this balanced state.

A test cycle lasted for 350 s, i.e. owing to the 10 Hz sampling density, for each parameter 3500 pieces of data were produced. The following five parameters were tested: (i) TLT revolution; (ii) wheel speed; (iii) actual speed; (iv) fuel consumption per hour; (v) actual traction force. Two other operational values were calculated from the measured data: (vi) slip from the wheel speed (ii) and the actual speed (iii); and (vii) peripheral speed of the tractor's driven wheel calculated from the revolution (i) and the rolling radius. The five measured parameters (i)–(v) and the two calculated ones (vi)–(vii) amount to 24,500 pieces of data per test run.

As an illustration, the result data for the 3rd gear are given in Table 3.

Table 3
Planned and actual number of measurements

		sexp.	scalc.	t	Δ	n [pc.]
Traction force	kN	0.5	0.45	1.96	0.05	312
Slip	%	0.2	0.14	1.96	0.05	30
Shifting duration	s	0.04	0.035	1.96	0.05	2
Fuel consumption	g/kWh	2	1.5	1.96	0.05	3500

s_{exp} – experienced corrected empirical standard deviation;

s_{calc} – calculated corrected empirical standard deviation;

t – probability constant;

Δ – accuracy range;

n – required sample size;

2.4 Measurement Process

The traction test was carried out in two scenarios: the brake cart was either in the fixed speed or in the fixed traction force mode of operation.

1) Fixed traction force Traction force was set to 30 kN on the brake cart. The accelerator was also fixed at the maximum position. The traction test started in gear B1. When the traction force was stabilized and traction became steady, the driver shifted to B2, then to B3, and finally to B4. In the second phase of this test, the driver changed down: B4–B3–B2–B1. The time between gear changes during which the traction force stabilized was 50 s. Data necessary for the distribution of the traction force during shifting was measured while changing up or down. Data were recorded by the measurer-collector system.

2) Fixed speed Speed was set to 1.4 m/s (5 km/h), with the accelerator being at the maximum position. Again, the traction test started in gear B1. After stabilization, the driver shifted to B2, then to B3, and finally to B4. In the second phase of this test, the driver changed down: B4–B3–B2–B1. Data were recorded at each shifting.

In one test cycle (changing up four gears and changing down four gears), approximately 50,000 pieces of data were recorded. Owing to the test circumstances (in a ploughland) and the high sensitivity of the measuring devices, the standard deviation of traction force and slip values were considerably high, with several outliers.

During processing, data series were cut up into sections. For the sections corresponding to steady traction phases, in the preprocessing phase methods of mathematical statistics were applied. Data that differed from the mean of the given section by more than ±5% were replaced by the mean of the neighbouring 5-5 pieces of data (i.e., the average of 10 pieces of data). In order to analyze the shifting phases, the raw data series was analyzed to be able to follow the changes accurately.

An exact methodology had to be created to determine the limits of each section from the data series. In all cases, the need for traction force grew after shifting. Then, after the machine group accelerated back and the brake cart automatically regulated the speed, traction force started to approach the previous value. The starting point of a stable traction section was defined as the second when the actual traction force first gets under the pre-set value. The endpoint of the stable section was defined as the starting point of shifting.

3 Test Results

The frequent sampling during the tests allowed for the analysis of the transmission of the traction force during shifting. The changes in slip and traction force could be depicted. The high standard deviation of the data was due to the measurement circumstances, but trends could be detected and analysed.

Measured data were analysed in two phases. At first, characteristics of the stable traction phases were described. Secondly, the dynamic changes occurring during shifting were compared.

3.1 Fixed Speed Scenario

Data recorded by the braking cart in the fixed speed scenario prove that traction parameters could be stabilized the fastest in the B3 gear. The traction force and slip values recorded during the test are shown in Figure 3.

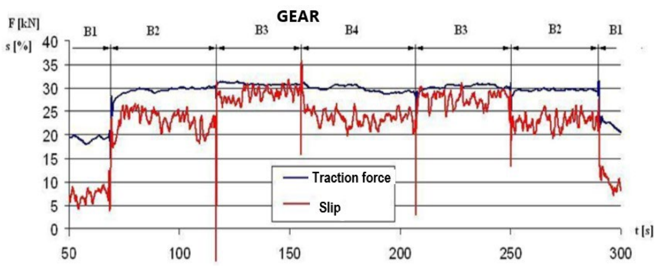

Figure 3
Changes of traction force and slip during shifting when loaded, fixed speed scenario

During the traction test, the value of the traction force could be kept in the range 25-30 kN, with the exception of the B1 gear. In the different gears, when traction force was stabilized, the standard deviation considerably diminished. Slip changes reflect the changes of the traction force, but with a different standard deviation. The speed was kept at the pre-set 5 km/h by the PLC in the brake cart. The standard deviation (SD) test proved that SD was the highest in the case of slip, out of all the parameters.

Table 3
Mean values of traction parameters and the corresponding standard deviation in the fixed speed scenario

Shifting	Δs [%/s]	ΔF [kN/s]	t_k [s]	$t_ö$ [s]
B1–2	38	26.8	0.15	0.6
B2–3	185	15.2	0.15	0.5
B3–4	80	29.3	0.15	0.4
B4–3	115	22.5	0.15	0.5
B3–2	23	7.5	0.15	0.6
B2–1	1.5	3.5	0.15	0.8

Table 4
Traction parameters during shifting

Gear	Stable traction parameters			
	F_{mean} [kN]	SD_F [kN]	s_{mean} [%]	SDs [%]
B1	19.32	0.29	8.31	1.36
B2	29.46	0.69	23.08	2.1
B3	30.82	0.25	28.67	1.3
B4	29.71	0.66	23.71	1.52
B3	30.29	0.36	27.81	1.41
B2	29.55	0.18	23	1.5

F_{mean} –mean traction force
SD_F – standard deviation of traction force
s_{mean} – mean slip
SDs – standard deviation of slip

Δs – slip change per second
ΔF – traction force change per second
t_d – clutch disengagement delay
$t_ö$ – total shifting time

With respect to the data series, the starting point of shifting can be defined as the first point in time after stabilization, when the traction force gets lower than that of the stable section at least by 10%; and the peripheral speed of the wheel also drops. The endpoint of shifting is when the rise in the traction force after shifting reaches its maximum. This process is illustrated by Figure 4, where the time scale is stretched for better visibility.

In order to be able to analyse the dynamic changes of the measured parameters accurately, those parameters must be selected that allow for comparing all shift changes in a uniform manner. As the above figures show, the process always starts with the change of slip, which is followed with a certain delay by the corresponding change in the traction force.

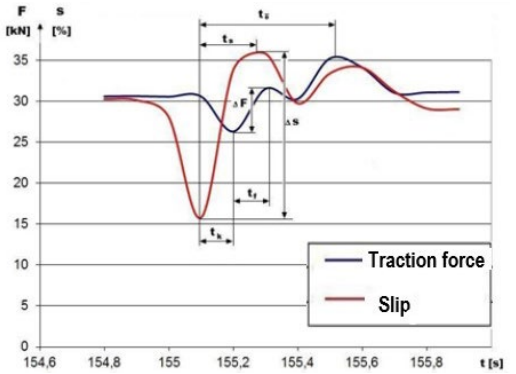

Figure 4
Changing of traction force and slip relative to time, at shifting from B3 to B4

The delay is marked as t_k in Figure 4 and Table 4. The delay can be explained by the characteristics of the forces arising at the soil-wheel interaction. As the driven wheel rolls over the surface, first the soil is deformed where the surfaces touch. This deformation requires time. Then, the tread of the wheel can adhere to this layer and exert traction force.

The size of the changes in traction parameters can be characterized by the change of traction force and the slip in a unit of time (1 s), i.e. the size of the change should be divided by the required time.

The total length of time needed for shifting (tö) is the time elapsed between the a starting point and the endpoint of shifting in seconds. The unit values of slip and traction force changes and the corresponding time required for shifting are given in Table 4 above.

3.2 Fixed Traction Force Scenario

For the fixed traction force scenario, the changes of the traction parameters relative to time are illustrated in Figure 5. In the stable periods after shifting, traction force stays around the pre-set 20 kN value. The slip changes compared to the mean value correspond to the changes in the track force. The standard deviation of the measured slip values are similar to that of traction values, but the standard deviation of slip is higher. After shifting, the traction force is increased so that the tractor could be accelerated back to the speed before shifting, which results in increased slip values.

To be able to compare the two scenarios objectively, the standard deviation of measured values had to be determined. The data presented in Table 5 below show that the standard deviation values of traction parameters are similar in the two scenarios: speed values have the lowest SD, while slip values have the highest SD.

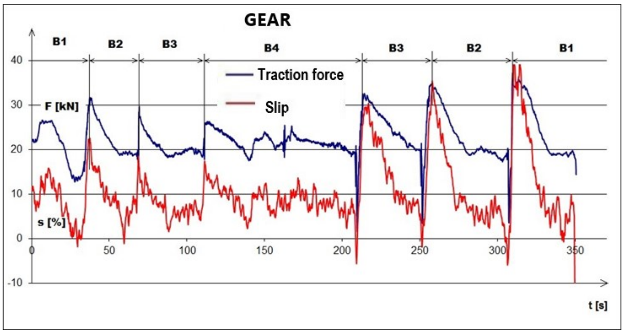

Figure 5

Changes of traction force and slip during shifting when loaded, fixed traction force scenario

Data in *Table 5* prove that the parameter changes are different when changing up from those when changing down. SD values show that the most stable operational parameters were recorded in gear B3. Conforming to the expectations of operators and owing to the electro-hydraulic operation system, shifting underload happens in less than half a second. Despite the short time, traction parameters considerably change in this interval. In order to be able to analyze data thoroughly, it is worth enlarging the graph depicting the data for the shifting period.

Table 5

Standard deviation of traction parameter values in the fixed traction force scenario

Characteristics of stable traction				
Gear	F_{mean} [kN]	SDF [kN]	s_{mean} [%]	SDs [%]
B2	18.99	0.43	6.51	4.02
B3	19.24	0.6	5.9	1.93
B4	21.52	1.3	9.23	1.98
B3	20.39	0.48	7.2	1.24
B2	19.08	0.95	4.76	2.72
B1	18.98	0.26	4.98	2.58

F_{mean} –mean traction force

SD_F – standard deviation of traction force

s_{mean} – mean slip

SDs – standard deviation of slip

Table 6

Traction parameters during shifting

Shifting	Δs [%/s]	ΔF [kN/s]	t_k [s]	$t_ö$ [s]
B1–2	–	–	–	–
B2–3	6.73	6.25	1	3.2
B3–4	6.5	6.09	0.5	1.5
B4–3	8.2	14.22	0.7	1.8
B3–2	6.72	8.35	0.3	2.1
B2–1	12	15.81	0.9	2.1

Δs – slip change per second

ΔF – traction force change per second

t_k – clutch disengagement delay

$t_ö$ – total shifting time

Figure 6 shows the data recorded during shifting from B4 to B3. In the shifting process, the traction force suddenly increases by 5 kN, and the corresponding increase in slip is 10%. In some seconds after shifting, traction force starts to vacillate with a frequency around 2 Hz, in a similar manner to the vacillation of the engine's revolution.

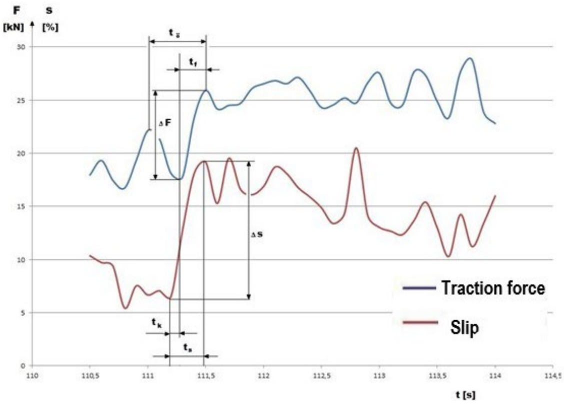

Figure 6

Changes of traction parameters relative to time when shifting from B4 to B3

The graph showing the parameter changes during the process of shifting proves that slip changes precede changes in the traction force. However, at the moment of shifting, for some milliseconds slip decreases, owing to the decreased torque and traction force transmission.

After shifting, slip values increase, but after some seconds, a decrease is witnessed. This shows the excess need for traction force after shifting in order to gain lost speed. The increased slip is the result of increased traction force. Similarly, to the other scenario, changes during shifting were compared. The relevant data are given in Table 6 above.

Processes during changing down are depicted using a stretched time scale. Data recorded during shifting from B4 to B3 are illustrated in Figure 7. In the case of changing down, the process starts with the decreasing of the slip, then after 0.15 s, traction force plummets by more than 15 kN at the beginning of shifting. At the same time, slip decreases. Due to the momentum of the brake cart, the minimum traction force is almost zero. As a result of, the tractor's inertia, the slip value becomes negative for the back wheels for a very short time during the slowing of the tractor.

As the data prove, the revolution of the engine cannot grow when the load is reduced. The load arising at the end of the shifting process causes a reduction of revolution by Δn = 3-5 1/s (compared to the mean of the vacillating revolution). This fall in revolution does not exceed the revolution sensitivity of the engine, thus no reaction follows. The increase in traction force after shifting is due to the excess traction force exerted for accelerating back after "stopping short" during shifting.

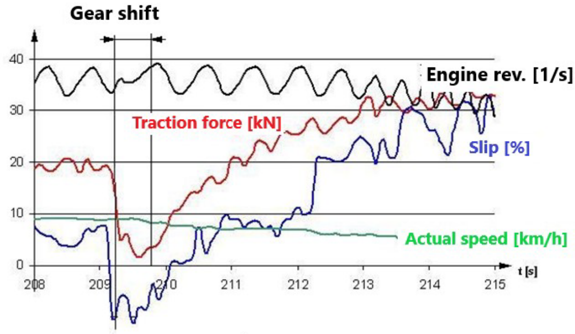

Figure 7
Traction parameters relative to time during shifting from B4 to B3

As the mapping curve for the traction force shows, the traction force gets back to the pre-set 30 kN within 3-5 seconds.

Conclusions, suggestions

The measurement technique described in this article proved to be appropriate for testing the traction parameters during shifting process underload in tractors. The methods and devices applied in the field traction test make it possible to follow the changes of traction force, actual speed and slip in the course of time. The huge amount of data obtained owing to the high sampling frequency allows for the detailed analysis of the shifting process.

The measured data proves that information about shifting time and the changes in traction parameters is crucial for the operation of tractors and other vehicles with constant load. The analysis presented here shows that the slip, occasionally with changing signs, constantly causes considerable traction loss, although its amount is varied.

In order to decrease traction losses that arise during shifting under load, which are primarily caused by the increasing slip, the following recommendations can be made based on our traction test.

In the field of operation of machines with powershift transmission it is suggested to turn off the fixed engine revolution (hand accelerator) before shifting, so that the operator can make the following corrections with the accelerator pedal.

When changing up, in the moment of shifting the accelerator pedal should be retracted by 30-50% (depending on the actual load). After shifting, acceleration should be continuous in order to speed up the machine group with minimal slip.

When changing down, it is worth slowing down gradually so that the exerted traction force could slow down the machine group while maintaining useful work. Otherwise, the driven wheels slow down the machine group with vacillating, dynamic load, occasionally accompanied by negative slip values.

These actions require attention and experience on behalf of the operator. At the same time, they enhance the efficiency of operation and lengthen the lifespan of the parts of the clutch.

In the software development for powershift transmissions area compared to the simple electro-hydraulic control, more refined regulatory processes are required during shifting. The following should be taken into consideration.

- When changing down, the clutch disengagement delay should be prolonged in order to avoid negative slip. Based on the revolution of the engine and the chosen gear, the system calculates the corresponding speed. It only shifts into the chosen lower gear if the machine group has already slowed down appropriately. In ploughland circumstances, this delay only lasts for some deciseconds.
- When changing up, irrespective of the position of the accelerator pedal, the performance of the engine should be restricted if the slip were to exceed the pre-set limit (e.g. 25%), in order to avoid the sudden growth of slip.

The above suggestions are important not only from an energetic, but also from agrotechnical and environmental perspectives. If slip is high during agrotechnical operations, it leads to higher compacting of the soil, the formation of deep tracks, and thus the microstructure of the soil is damaged.

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