EXAMINATION OF ENERGETICS REALTIONS OF UNDERLOAD GEAR SHIFTING

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ABSTRACT: Transmissions used in present-day tractors feature underload gear shifting even in lower performance categories in order to perform gear shifting during constant load. Shifting under load presents a critical part of operation. Tractors are subjected to high cinetic energy losses during shifting due to high drawbar force and low traction speed. Thus, shifting must be executed in a fairly short time. For this reason, underload gear shifting is used the major requirement of which is preventing torque transmission from ceasing entirely during the shifting within the shortest possible time. In our paper, we carried out a ploughland traction test with a tractor measuring traction characteristics during gear shifting. The short gear shifting, consuming 0.3-0.5 seconds, presented a significant challange to the measurement method and the measurement equipment used. At constant speed, the scattering of drawbar force and slip was examined during shifting. We aimed to determine moving speed, engine speed, drawbar force and slip during the shifting process as well as to examine their cause-effect relation and evaluate the results obtained.

KEY WORDS: Gear shifting, moving speed, engine.

1 INTRODUCTION

Efficient energy utilisation is an important element of the development and operation of agriculture prime movers. In addition to these criteria, there is an increasing pressure on developers to comply with economic and environmental requirements. This makes higher and higher demands not just towards permanent engine development but tractor drivetrain efficiency as well. The structural design of the transmission and the right choice of the momentary gear ratio—shifted automatically or manually—retroact on engine load, significantly influencing its pollutant emission as well, as demonstrated by studies in relevant literature [8] [11] [12]. Considering all this, it is clear that tractor drivetrains are to be regarded as one holistic system from engine to tyres both during development and operation. New results in terms of energetics and operation aspects cannot be achieved unless modifying and developing the system elements in a consistent way. After powershift transmissions having been in use for decades and after stepless drives having emerged recently—both aiming optimal engine performance utilisation and operator comfort—most recent developments pertain to maintaining underload shifting and the most simple drive technology solutions in order to minimalise performance losses. Discovering transmission losses, whether via previous calculations or real tests, calls for cautiousness and due meticulousness [9]. Investigations disclosing energy losses of tractor transmissions showed that there are significant losses due not just to friction losses of mechanical drive transmission emelents but—mainly at higher vehicle speeds—lubricant viscosity and the operation of the own hydraulic system of the transmission, as demonstrated by several studies in relevant literature [1] [4]. In the process of ratio change, there are further losses since gear shifting is only feasible at reduced torque transmission, with the relative slip of the hydraulically actuated wet multidisc clutch. The transmission discs are subjected to not inconsiderable friction and heat load due to a relative angular velocity difference and depending on the extent of the torque transmitted. From the perspective of transmission operation and its process control, it is important to know the processes taking place in the actuating system. Based on simulation [15] calculations, optimisation opportunities of pneumatic transmission control were investigated by Szimandl [10]. Control processes of hydraulically actuated powershift transmissions were in detail examined by D.C Kim [7] using EASY 5 simulation [16]. While looking into shifting processes it is important to examine the cinetic energy loss of the machine group since affecting the operational perfection through work speed reduction. During shifting time, because of the lack of drive the decelerated machine

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group has to be reaccelerated increasing the specific energy consumption. For having shifting comfort and an optimised transfer of power during shifting, more and newer control aspects and solutions are emerging [2] [13]. As far as examination methods are concerned, computer simulations are becoming more and more common. In my opinion, however, these are primarily suited for preliminary tests, for determining the expensive test equipment to be used in real conditions as well as the test parameters. In case of agruculture prime movers, besides the ”traditional” ploughland tests, public road tests including acceleration, emerged as well. Based on such inventive tests, Bietresato [6] [14] strove to find out how the traction performance varies using transmissions with different construtions. Knowing the engine characteristics diagrams, he also determined the efficiency of the drivetrain and the losses due to driving resistance.

Our investigations, based on ploughland traction tests, included the measurement of changes of traction and energetics characteristics during shifting as well as the disclosure of interconnections of parameters influencing these factors.

2 MATERIAL AND METHOD

2.1. Test location

Traction tests took place on an area of the Cegléd-Cifrakert district of the South-Pest County Agriculture Zrt., covered by cereal stubble field, on 5th September, 2013. After harvest, the cereal stubble field was harrowed with medium weight harrow and a depth of 10-15 centimetres. The moisture content of the soil was 20-25%. The soil, medium bound quality and harrowed a month back, featured medium density. The test area was plain and the measurement lenght exceeded 500 metres which is ideal for measurement reasons.

2.2. The prime mover used in the traction test

For the test, we used a tractor make CLAAS ARION 420 featuring 16 gears and multiplication type transmission. Its gears within a group can be shifted under load (1-2-3-4). The engaging of the chosen group (A-B-C-D) can be executed in no-load-mode.

Major technical data of the tractor used in the test are given in Tabel 1.

Tabel 1: Technical data of tractor CLAAS Arion 420.

Parameters	Data
Model	CLAAS ARION 420 DPS
Chassis number	A2114DA/12103714 LZZ009
Engine number	CD4045L216148
Engine displacement	4525 cm^3
Make	John Deere
Performance	88 kW/2200 f/min
Weight of tractor	4900 kg
Size of back tyres	520/70 R38

Figure 1. Tractor CLAAS and braking vehicle MAZ during field test.

From the perspective of our research, it is important to overview the transmission construction of the prime mover. The structure of the multiplication type tractor transmission can be seen in figure 2. The engine torque goes first through the planetary gear group, marked 4, which gives four gears, shiftable under load.

Subsequently, drive goes through the reverser, marked 3. A smooth, joltless start both forward and backward is provided by its multidisc hydraulic clutches. For this, the machine operator only has to flip a switch in the right direction. By doing so, the start is always carried out with equal precision, irrespective of the operator's capability and momentary concentration.

At the end of the line, the group switch can be seen, marked 2, shiftable in an electro-hydraulic way too, but only during standing.This provides the groups A-B-C-D with their four gears. Thus, with these four gears per group, 16 gears forward and 16 gears backward are available.

The creeper, marked 1, is an optional supplement to the transmission. Due to the high ratio of the planetary gear located in it, the tractor can reach a moving speed of up to 500 metres per hour at nominal engine speed.

Figure 2. The structure of the Qadrishift transmission of the tractor CLAAS.

1.creeper gear, 2. group switch, 3. reverser, 4. planetary gear section, shiftable under load

Ratios given by the drivetrain of the tractor determine the drawbar force values available in the ploughland speed range. Based on these, the optimal operation range of the values speed, drawbar force, slip can be determined. The speed range of the measurement tractor is between 0 and 40 km/h which is split up to 16 gears. The traction characteristic diagram and the consumption characteristic diagram were plotted in every gear.

Figure 3 shows the bar charts of work speeds available in cetrain gears, calculated with factory sized tyres for an engine speed range of 1400-2100 1/min. For our measurements, we chose group B which provides a higher drawbar force and used for lower speed polughland tasks.

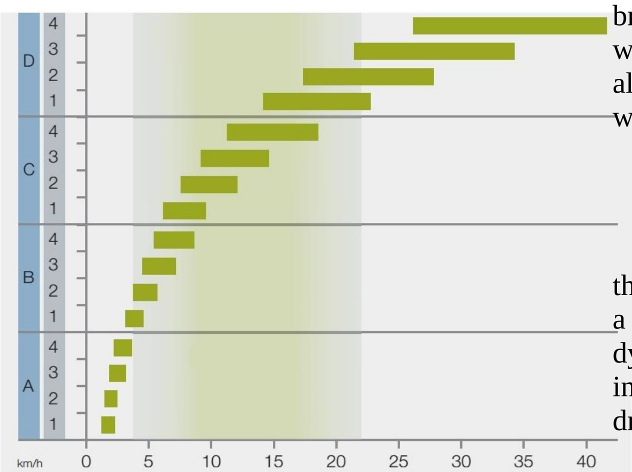

Figure 3. Speed range covered by certain gears at an engine speed range of 1400-2100 1/min.

2.3. Measurement vehicle used for plotting the traction characteristic diagram

For ascertaining the traction characteristics of the tractor, the load was provided by a specially designed braking vehicle. Both changing and constant load characteristics could be adjusted.

The braking vehicle was made out of a missile and tank traction vehicle make MAZ 537. From the original vehicle, only the running gear left. A diesel engine make IVECO V8 Cursor, performance 400 kW, was installed into the measurement vehicle. The original drivetrain was replaced by a closed system hydrostatic drive with continuously alterable fluid flow which is suitable for driving the vehicle in the targeted way. With the alterable fluid flow closed system hydrostatic drive, the adjustable load torque and traction speed of the braking vehicle, the load of the tractor can be controlled. The data, representing the simultaneous operation of the braking vehicle and the tractor, are determined by means of force and speed measurement, the way it can be seen in figure 4. For operating the measurement system in figure 4 as well as for data collection, a computer system was installed. The traction slip of the tractor can be calculated as the quotient of the actual distance covered and the theoretical distance calculated from the actual wheel speed.

Major characteristics of the braking vehicle:

The braking vehicle features a normal drive mode suitable for traffic. In this case, the wheels of the vehicle are driven by the hydrostatic system.

In braking mode, when hydraulic motors, driven by the wheels and functioning as pumps, the braking force can be steplessly controlled by counterpressure, adjusted by throttling, in a hydraulic way.

The adhesive force, necessary for ploughland braking, can be altered by putting additional wheight on the braking vehicle. The controlling algorhythm provides two different opportunities which can be chosen by the operator:

- Measurements at constant speed and
- Measurements at constant braking force.

(Speed or drawbar force controlled mode).

The concact between the examined tractor and the braking vehicle is maintained by a steel rod with a length of 5 metres. On its end near to the tractor, a dynamometer equipped with strain gauge was installed which was responsible for turning the drawbar force into electronic signals.

The major characteristics of the braking vehicle are given in Tabel 2.

Tabel 2. The major characteristics of MAZ 537.

Parameters	Data
Engine performance	400 kW
Braking/traction performance	250 kW
Maximal braking/drawbar force	150 kN
Speed range	0-35 km/h

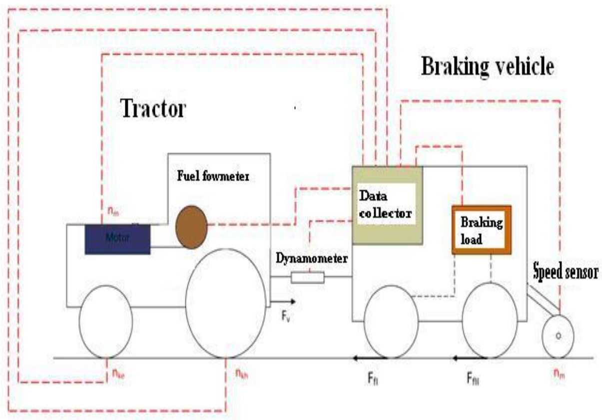

Figure 4. Concept of the measurement.

According to the scheme in figure 4, following traction parameters were recorded:

Drawbar force (Fd) [kN]; it is measured by a load cell located in the dragbar of the measurement car which can be seen in figure 4. The values measured by it, are stored by the data collector of the load cell located on the measurement vehicle.
The peripheral velocity (vp) [m/s] of the driven wheel of the tractor can be determined based on the speed (nw), measured by the speed transmitter installed on the wheel, and the rolling radius (rr) as follows:

$$v_p = 2 \cdot \pi \cdot r_r \cdot \frac{n_w}{60} \quad (1)$$

In case of all vehicles moving on terrain, there is a slip occurring in the contact between the wheel and the terrain the extent of which is indicated by slip (s), thus, it was necessary to measure the actual moving speed of the machine group. The actual moving speed is measured by a pushed wheel measurement system connected to the braking vehicle where the moving speed is determinable from the speed of the pushed wheel and the rolling radius, with the calculation method of the speed of the driven wheel. The extent of the slip is indicated in percent by the difference of the two speeds measured this way as follows:

$$s = \frac{v_p - v_m}{v_p} \cdot 100 \quad (2)$$

where vp is the peripheral speed and vm is the moving speed.

The engine speed could be measured by means of an impulse transmitter indirectly connected to the PTO shaft of the tractor.

For measuring the fuel consumption, a PLU flowmeter make AVL was used, located in the measurement vehicle but installed in the fuel supply system of the examined tractor. This measurement instrument is consisted of a 600 ℓ/h volume flow flowmeter, installed into the pressure line of the fuel system, and a return line flowmeter. For a precise determination of consumption, measurement cycle times were to be set. The amount of fluid consumed in a certain time unit is given by the difference between the two measured volume flows.

The consumption meter features temperature compensation, its measurement accuracy is ±1% and it is connected to the data collector of the measurement vehicle processing and storing the signals and indicating hourly and specific consumption.

2.4. Method of the data processing

The signals provided by different transmitters were received by a 16 channel measurement and data collector system make SPIDER Mobil. During the measurements, the sampling density was 10 Hz. The data collected by the measurement system were recorded and stored by the software CATMAN without any previous selection. By means of the software, the data measured were read out and displayed in a chart, suitable for the sampling. The subsequent rows of the chart included the values measured every 0.1 second.

2.5. Measurement procedure

The objective of the ploughland test was to determine the traction characteristics during real operation conditions in regard to underload shifting. The traction test was carried out in both modes—speed controlled and drawbar force controlled braking vehicle operation—as follows.

- The traction test started with 30 kN controlled drawbar force, adjusted on the braking vehicle, at maximal load position throttle, in gear B1. Then, after the measured values having stabilised, the operator engaged gears B2, B3 and finally B4. Stage 2 of this test included down-shift measurements, in order B4-B3-B2 and B1. The stabilisation time between gear shifts was approx. 50 seconds.

- At the next measurement, in speed controlled operation mode of the braking vehicle, a moving speed of 1,4 m/s (5 km/h) was adjusted and the measurement just as well started at maximal load position throttle, in gear B1. Keeping the stabilisation times, traction characteristics were recorded, first shifting up to B2, B3 and B4, subsequently, shifting down B4-B3-B2-B1.
- Carrying out these high sampling density tests, it became possible to expose the transfer of the drawbar force during shifting. Thus, dynamic processes during underload shifting could be analysed.

3. MEASUREMENT RESULTS

3.1. Measured data in speed controlled operation mode

Data recorded by the braking vehicle at continuous speed control showed that traction parameters could be stabilised in gear B2 within the shortest time

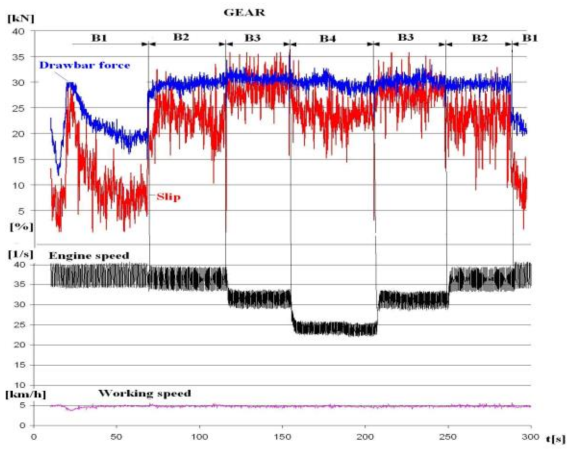

Figure 5. Traction characteristics during underload shifting at speed controlled traction.

(since in gear B1, the maximal speed of the tractor did not reach the adjusted speed of 5 km/h and the PLC control system of the braking vehicle reacted to this deviation by dramatically reducing drawbar force load). The values drawbar force, moving speed, slip and engine speed recorded during the measurement can be seen in figure 5.

Except for in gear B1, the drawbar force could be held between 25-30 kN. The scattering of drwabar force values in cetrain gears after stabilisation, reduced significantly, more than that, the scattering ceased. The measured values of slip changes represent the drawbar force changes, however, its scattering is rather high. The moving speed was held by the PLC of the braking vehicle at a value of 5 km/h. Logically, the engine speed decreased while shifting up, and while shifting down it increased, corresponding to particular ratios as well as the speed values set by us.

Because of the scattering of the measured values of the major traction characteristics, the characteristic value of the drawbar force had to be determined. For this, we used the method of the average calculation with the following formula:

$\bar{x}$ - Average value of the parameter in question;

$$\bar{x} = \frac{\sum_{i=1}^{n} x_i}{n} \tag{3}$$

where:

x_i – momentarily value of the parameter in question;
n – number of the data recorded.

$$\sigma = \sqrt{\frac{\sum (x_i - \bar{x})^2}{n-1}}, \tag{4}$$

Table 3. Scattering of values in speed controlled operation mode.

Traction parameter	Scattering
Drawbar force	4,147624
Slip	7,549112
Moving speed	0,211738
Engine speed	5,077212

The scattering examination proved numerically that the slip fluctuation is the highest of out all examined parameters. In addition, the chart data show that the moving speed could be held almost perfectly with the help of the braking vehicle.

3.2. Measured data in drawbar force controlled operation mode

Traction parameters recorded during underload shifting in drawbar force controlled operation mode can be seen in chart 6. After stabilisation, drawbar force becomes even, coming close to the value of 20 kN previously set. The change of slip compared to average value equals the drawbar force changes and the scattering of the measured values is nearly equal, however, it is higher than the scattering of drawbar force values. The higher drawbar force values after shifting suggest that there is an increased demand of drawbar force in order to reaccelerate the machine group, resulting in increased slip.

From the recorded data, it is manifest that the change of moving speed is proportional to the change of engine speed.

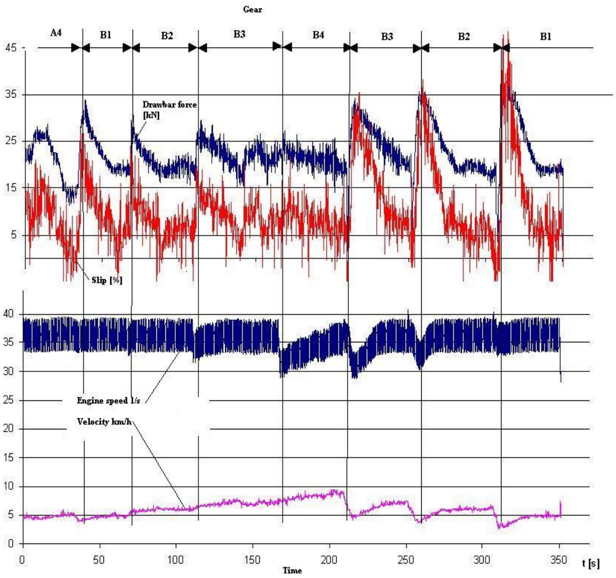

Figure 6. Traction characteristics during underload shifting at drawbar force controlled traction.

For comparing the two measurement operation modes in an objective way, it was necessary to execute a statistical scattering examination of the measured data, here too. From the data in tabel 4, it can be seen that the scattering values of traction characteristics are similar to the values recorded in speed controlled operation mode: the scattering of speed values is the lowest, whereas slip values featuring the highest scattering.

Tabel 4. Scattering of values in drawbar force controlled operation mode.

Traction parameter	Scattering values in percent in the gears examined			
	B1	B2	B3	B4
Drawbar force	3,93	0,87	1,35	2,00
Slip	4,99	3,90	2,65	3,33
Moving speed	0,27	0,21	0,16	0,48
Engine speed	2,02	2,01	2,00	2,14

The data show that the traction characteristics are different at shifting up and shifting down. Based on scattering values, it can be ascertained that the most stabil operation characteristics could be measured in gear B3. According to the user's expectations and thanks to the electrohydraulic actuating system, the underload shifting usually takes less than half a second. Despite this short time, there are significant changes taking place in traction characteristics. For examining these changes effectively, it is advisable to look into the data recorded during shifting in an enlarged way.

In order to present the changes in a more detailed way, the data redorded during shifting B2-B3 are given in figure 7.

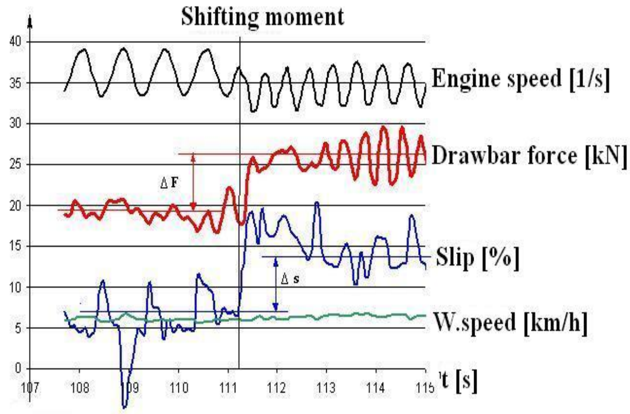

Figure 7. Traction characteristics plotted against time at shifting from B2 to B3.

It can be well discerned that there is a sudden increase in drawbar force of 5 kN during shifting and parallel to that, slips goes up by 10%. Several seconds after the moment of shifting, the drawbar force starting fluctuating with a frequency of about 2 Hz can be observed, similar to engine speed fluctuation. After shifting up, the average value of engine speed shows a decrease of Δn=5-6 1/s resulting from increased engine load.

From the processes describing drawbar force changes in figure 7, it can be determined that slip change is equal to drawbar force change. It can also be found that, at the moment of shifting, the slip lessens for a period of several miliseconds which can be explained by reduced torque and drawbar force transfer at the moment of shifting. Slip values increasing after shifting, then lessening after several seconds, show the increased darwbar force demand for reaccelerating the ploughland machine group after shifting as well as the increase in slip caused by an increase in drawbar force. In the same way, the processes taking place during shifting down can be well examined on an expanded time scale. Data recorded during shifting down from B4 to B3 can be seen in figure 8.

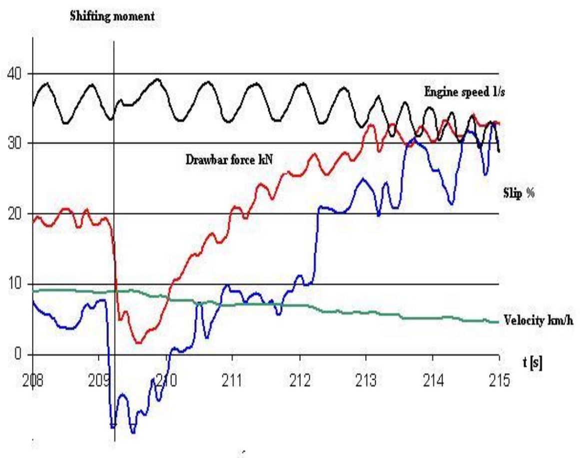

Figure 8. Traction characteristics plotted against time at shifting from B4 to B3.

At shifting down, drawbar force falls by over 15 kN at the beginning of shifting entailing a reducing slip. Due to the momentum of the braking vehicle, the minimum of the drawbar force is almost zero and due to the inertia of the tractor, the slip has a negative value for a short period of time at the rear wheel.

Looking into the engine speed it is visible that the engine works at its nominal speed, thus, it cannot become higher during load reduction.

At the end of the shifting process, the reappearing load results in an engine speed reduction of $\Delta n=3\text{-}5$ 1/s (referred to the average value of the fluctuating engine speed). This engine speed reduction does not exceed the speed sensibility of the engine, consequently, it does not react to these. To the increased drawbar force after shifting, the excess drawbar force gives an explanation which is necessary for reaccelerating after the slight momentary speed loss caused by the shifting. Following through on the development of the drawbar force on the drawbar force diagram, figure 4, we can see it recovering within several seconds to the set value of 20 kN.

4. CONCLUSIONS

Based on the measurements it can be ascertained that the method used by us is suitable for examining traction characteristics during underload shifting. With the appliances used, changes of drawbar force, moving speed and slip can be effectively monitored during the entire test period. The significant amount of the recorded data makes a detailed analysis of the shifting process possible.

At the measurements, the extent of changes in traction characteristics could be determined both at shifting up and shifting down.

From the data published above, we can inference that knowing the changes in shifting time and traction characteristics are important information regarding the operation of tractors and other constant load mobile machines. Examining the data it can be determined that

- the slip causes continuous, significant and fluctuating traction losses, even during shifting.
- after shifting, the similar frequency of the fluctuation of engine speed and drawbar force is caused by a tight interconnection.

The data received are the basis of our future work at which we will aim at determining the traction losses of tractors during and between shifting.

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