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Testing of Operating Characteristics with Powershift Multi-disc Clutches

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Abstract: *The diverse requirements pose complex demands on today's agricultural tractors. Effective work requires a modern diesel engine to have an efficient and versatile drivetrain, since the engine's performance must be transmitted to the ground and the wheel with minimal loss. At the same time, the activated gear ratio ensures optimum engine load in the otherwise wide range of speeds to reach low fuel consumption. For this reason, powershift transmissions are most often used under modern loads. The critical point of their operation is the gear shifting under load - the gear shifting process which is achieved by the controlled sliding of multidisc wet clutches. The research work focuses primarily on these clutches, starting from the electric signal controlling the shift, due to the displacement of the hydraulic actuation elements, taking into account the change in the drawbar force exerted.*

Keywords*: Tractor drivetrain, powershift transmission, multidisc clutch*

1. Introduction

Climate change and energy efficiency that are linked to each other are among the challenges of the present and the future.
Nowadays, transportation is the biggest branch causing greenhouse gases, that is why energy saving and low emission vehicles and prime movers are needed. In the case of vehicles, energy efficiency and emission are not simply a question of motorization, but it is also a machine development problem that includes drivetrain solutions providing efficient transmission.
The most frequent purpose of researches on the development of gear boxes is the reduction of power transmission loss. While earlier developments mostly aimed at the optimal use of engine power and meeting the expectations regarding the ergonomic demands of operation, nowadays, the minimization of performance loss is the most typical goal.
The results of researches on the energy losses of tractor transmissions can be found in various literature. *Mc Carthy and Kolozsi* [5] already dealt with the research method based on preliminary calculations of transmission energy loss and and actual tests in late 20th century. Their calculation principles are valid to this day. Based on the simulated examinations of the characteristics of the continuously variable tractor transmissions *Joóri et al.* [2] found that the continuously variable tractor transmissions can be more widely used than the stick shift transmissions and due to the more optimal transmission of engine performance, they provide lower specific fuel consumption. In subsequent researches, *Kerényi and Farkas* [3] found that the continuously variable, power branching engines provide the most optimal, η=0,90-0,94 transmission efficiency in the i=0,85-2,4 variation range.
Based on their research aiming at detecting kinetic energy losses of tractors, *Molari and Sedoni* [1] pointed out that beyond the friction losses of the mechanical power transmission parts - especially in case of higher travelling speed -, losses due to lubricant viscosity as well as due to the operation of the gearshift's own hydraulic system itself are significant.
In their publications, *Molari and Sedoni* [1] and *Bietresato et al.* [6] refer to the fact that the hydraulic system of the powershift transmission consumes 4% of the input power even when unloaded. Their researches also pointed out that half of that results from the losses due to oil filling viscosity.
In the process of gear shifting, the transmitted power is reduced by the friction losses depending on the time passing during the variation of transmission. The passage of time is influenced by the hydraulic accumulators built in the system to stabilize the supply pressure *D. Prodan and A. Bucuresteanu* [7]. During gear shifting, the energy transmission between the driving shaft and the driven shaft ceases and with the increase of the gear shifting time, the kinetic energy loss of the prime mover increases due to the decrease of the travelling speed.
Considerable losses occur in the connection of the wheel to the ground as well. *Kiss and Laib* [4]

examined the components of the rolling resistance and paid special attention to extent of work necessary for the deformation of the soil.
In the light of all the foregoing, it can be assumed that the energy losses on prime movers and vehicles due to gear shifting can be decreased by the optimization of the shifting time of transmissions.
The research objectives can be determined on the basis of my assumption. Objective of this research: exploring the possibilities of decreasing loss on prime mover transmissions based on the theoretical analyses and practical examinations. As far as the execution methods of the examinations are concerned, they are in accordance with the work of *Hristea Al. et al.* [8].
In order to explore the connections, the passage of time during the shifting process, that can be described with the changes of regulating signals, has to be examined. The conclusions drawn from results based on analyses and measurements support the proposals for decreasing loss in prime mover transmissions.

2. The losses of tractor transmissions

Based on the results in literature, the losses of transmissions can be divided into two main groups on the basis of their source:

First, the mechanical losses of transmission that are caused by the friction loss at the gear connections, bearings and shaft sealings
- In addition to these, in case of power branching transmissions, there are the losses of the power transmission parts (variator, toroidal friction disc).
- In case of underload gear shifting, the friction losses of the multidisc clutch (at start and underload transmission)

Secondly, the hydraulic losses which, in most tractor transmissions occur due to the viscosity of oil filling, also used as hydraulic fluid. By moving and mixing it, they cause the heating of the transmission. As the transmission losses are accounted for more than half of the transmission losses according to literature sources, this is what will be examine in more detail.

2.1 Hydraulic losses in the gear shift

- Losses of the transmission's own hydraulic circuit
    - for the flow and vaporization of the lubricant
    - for the operation of clutches (TLT, POWERSHIFT)
    - for operation of steering and braking
- The fluid friction between the released brake and the clutch lamellae

Nowadays, hydraulically (and hydro-pneumatically) actuated multidisc clutches (Figure 1) are used in high performance prime movers and vehicles.

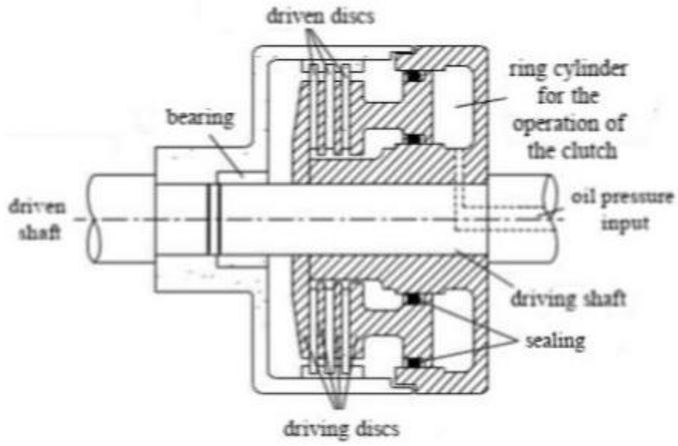

Fig. 1. The structure of a hydraulically actuated multidisc clutch [9]

Torque is transmitted from the drive shaft to the driven shaft by the friction occurring on the discs fixed to them. When the clutch is engaged, transmission is provided by the hydrostatic pressure that presses the discs against each other. The pressure transfer medium is hydraulic oil (fluid). However, even with this it takes time for the pressure necessary for the torque transmission of the drive shaft to be built up.

3. The method and toolset of measurement

Measurements were carried out to explore the operation characteristics of powershift clutches, paying special attention to under load gear shifting.
The measurements to be carried out aim at supporting the possibilities for decreasing losses that occur on the clutches of prime movers.
The measuring method to be applied is based on the fact that at gear shifting, torque transmission between the drive shaft and the driven shaft of the clutch stops for a measurable time and this fluctuation results in a detectable drawbar force fluctuation as well.
Experimental toolset:

Prime mover:
Type: New Holland T7040
Gear Shift: Full Powershift, multiplication system 3x2x3=18 and one direct gear
Tow: HW 80.11 trailer
Total weight during the examination: 9800 kg
Equipment: hydraulic cylinder built in towing device and pressure transmitters (Figure 2)

Fig. 2. Measurement of indirect drawbar force with hydraulic cylinder

Table 1: Features measured and their measurement tools

Nr.	Feature measured	Unit of measurement	Frequency	Measurement tool
1.	PWM valve operating current	mA	200 Hz	Scan Tool software
2.	Operating pressure of clutch B	bar	100 Hz	HydacHMD3444
3.	Operating pressure of clutch D	bar	100 Hz	HydacHMD3444
4.	Operating pressure of clutch E	bar	100 Hz	HydacHMD3444
5.	Chamber pressure of cylinder built in the drawbar eye	bar	100 Hz	HydacHMD3444

Number of repetitions: 2
Location of measurements: Periphery of the town Szerencs

3.1. The relevant characteristics of the experimental toolset

The detailed examination was carried out on a New Holland T7040 type tractor with a Full Powershift clutch.
The clutch makes 19 forward and 6 reverse gears shiftable under load. This clutch provides speeds of 3-9 km/h and 5-13 km/h during field, and a speed of about 40 km/h during transportation.
Gear shifting is done by 9 hydraulically actuated multidisc wet clutches. A characteristic of the gear shift is the fact that during transportation in 19th gear, optimal efficiency direct drive can be transmitted from the flywheel through the TLT drive shaft, thus decreasing heat development and friction losses [1].

Presentation of the measurements carried out

The measurements were carried out during road towing, with the loaded prime mover being accelerated continuously. In the acceleration cycle, the indirect drawbar force measurement was carried out by recording the pressure values built up in the chamber of the hydraulic cylinder placed in the towing structure. During the measurement process, the pressure values of the operating current and the hydraulics operating the gear shifting clutch were recorded against time. The measurements were carried out at fixed throttle position. The rotational speed of the tractor engine had been set to 1500 1/min beforehand.
The measurement was carried out in two phases.

As a first step, the electric signal of the actuator operating he hydraulics was read from the control circuit of the actuator through the CAN network of the tractor, with New Holland's original software (New Holland Electronic Service Tool). The amount of current that passed through the solenoid of the hydraulically shifted multidisc wet clutch was saved with this software.

In the second measurement phase, the changes were examined in hydraulic pressure. A Hydac analysis tool /meter+data collector/ was applied to measure the pressure operating the clutch during gear shifting. The ***Hydac HDA 3444-A-600-000*** pressure transmitters were connected to the pressure measuring service ports of clutch B C and D on the control blocks of the transmitter. These can be found in the hydraulic control circuit right after the PWM valves, so the amount of pressure operating each clutch can be directly examined. Data were retrieved on the time and course differences between the controlling current and the pressure it passed through.
The measured pressure values were recorded by a Hydac HMG 2020 data recorder from which data was imported to a PC and could be processed.

Fig. 3. Connection of the pressure transmitters and the data collector to the control block, valve C, D and E.

3. Analysis of the measurement results

The control signals, the number of which reached 8000 during the examination, were recorded in the case of every clutch. The current changes during the gear shifting process based on the measured data are presented in Figure 4.

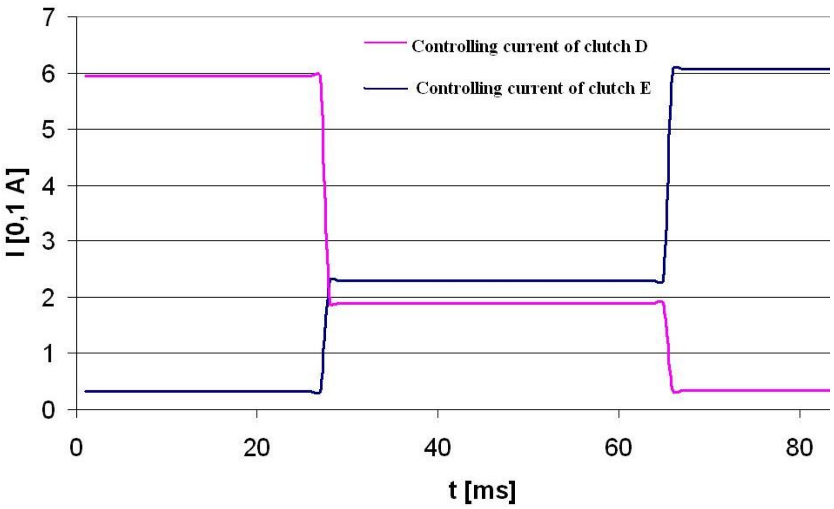

Fig. 4. Time course of the electrical control signals of the subsequent gears during gear shifting

The amount of control current of the hydraulic system decreased from 0.6 A to 0.2 A during gear shifting when the clutch D was engaged, while the same amount of current starts to switch the control valve of the hydraulic clutch in order to reach the next gear. After the time (in this case 47 ms) that can be set in the software which operates the interconnected system, clutch D completely stops, meanwhile its controlling current decreases to 0.05 A. On the contrary, clutch E becomes completely engaged, thus its controlling current increases to 0,6 A. It was found important to note that as I was preparing for this series of measurements, preliminary control measurements were carried out in non-loaded mode, without a tow and at 1200 1/min and 1800 1/min engine speeds as well. The current of the electronic signals and the controlled time of the gear shift were the same as the values measured in underload mode.
For the analysis of the relations of the operation process, the electronic signal at the particular clutch and the hydraulic control pressure was presented on an identical time scale.

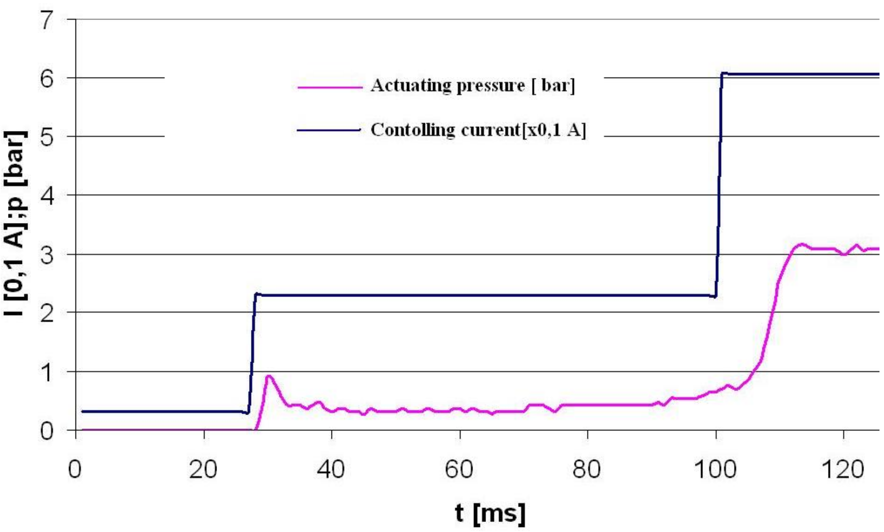

Fig. 5. Time course of the pressure of the electronic signal at clutch D and the hydraulic actuator during gear shifting

On Figure 5 it is clear that pressure of the hydraulic system builds up with a 10-15 ms delay, following the changes of the control signal. The explanation to this is that certain moving occurs during the compression of the lamellae, which takes time to happen. In addition, the accumulator built in the system also has to charge for a smooth gear shift.

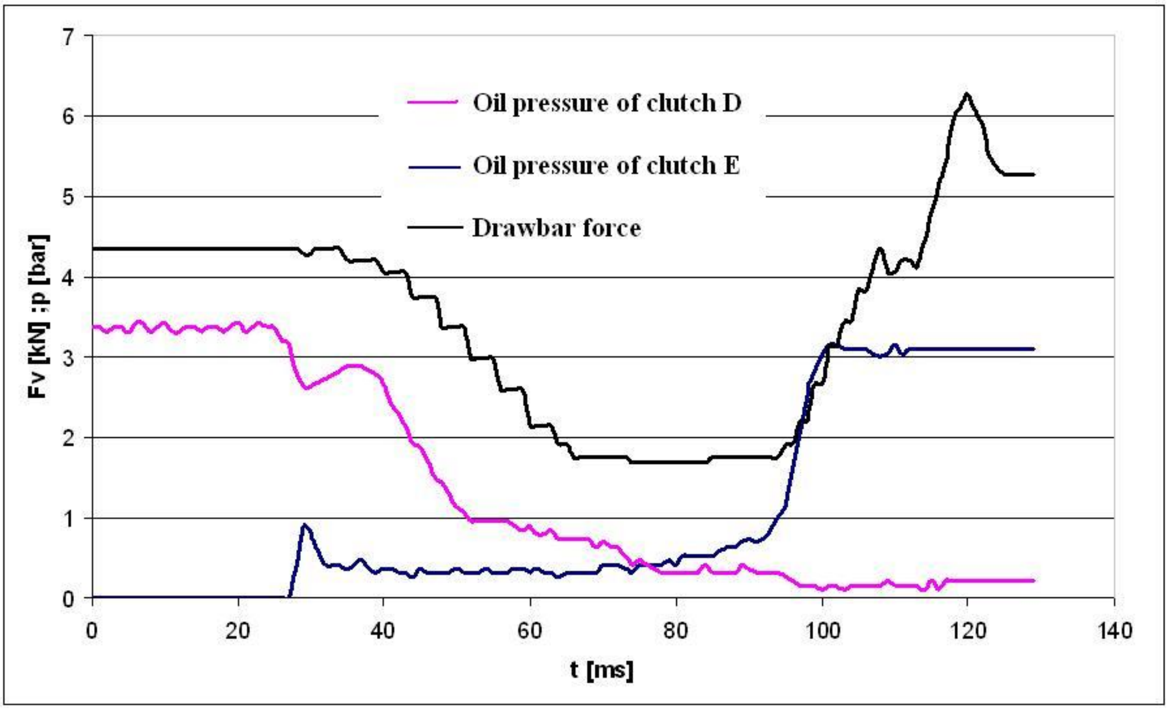

Fig. 6. Time course of the hydraulic control pressure and the drawbar force during gear shifting

Figure 6 shows the change of the measured drawbar force. From this, it can be ascertained that the control pressure of the clutch is closely followed by the drawbar force exerted by the prime mover.

The few ms delay is primarily due to the inertia of the prime mover as the drawbar force in the wheel-soil interaction was not examined, but between the prime mover and the tow. The drawbar force peak at the end of the shift clearly presents the extra force necessary for the prime mover to accelerate again.

4. Conclusions, proposals

Prime movers have to exert a variable amount of drawbar force in a wide speed range. Based on the measured data, the relations between the electronic control signal and the course of the shifting process can be detected. It can be found that the examined prime mover works with the same duration and current of control signals independently from the speed and the exerted drawbar force. From an energetic point of view, it would be advisable to differentiate the hydraulic pressure that controls the course of the under-load gear shifting depending on the drawbar force and the travelling speed. With the modified control, in addition to the expected (otherwise slight) fuel saving, the lifespan of the clutch parts may considerably increase, the operating temperature of the lubricant may decrease and depending on the operating conditions, the shifting comfort improves.

With the analysis of the construction and the operating conditions of the multidisc clutch, the following relations can be determined the connection between the control pressure and the transmitted torque.

From the geometric and friction characteristics of hydraulically (and pneumatically) operated clutches, a coefficient can be derived (specific torque transmission). Multiplied by the control pressure, it provides the amount of the momentary gear ratio.

The specific transmission coefficient (C_{MP}):

$$C_{MP} = \left(D_h^2 - d_h^2\right) \cdot \pi \cdot z \cdot \mu \cdot D_k \tag{1}$$

Where:

D_h is the outside diameter of the ring plunger;
d_h is the inside diameter of the ring plunger;
D_k is the average diameter of the friction discs;
z is the number of friction discs.

Due to the dynamic viscosity of the lubricant, friction occurs between the friction discs even when multidisc wet clutches are released.

Friction between the discs when multidisc wet clutches are released (F):

$$F = \eta \cdot \frac{dv}{dx} \cdot A \tag{2}$$

Where:

η is the dynamic viscosity of the lubricant;
Dv is the average speed difference of the friction discs;
Dx is the average gap between the discs of the released clutch;
A is the the frontal area of the friction disc.

The average speed difference of the discs (dv):

$$dv = q_i \cdot n \cdot \left(D - d\right) \cdot \pi \tag{3}$$

Where:

q_i is the quotient of the gear of examined clutch and the transmission of current gear;
N is the input speed;
D is the outside diameter of the friction disc;
d is the inside diameter of the friction disc.

The average gap between the friction discs (dx):

$$dx = \frac{x_z}{z} \tag{4}$$

Where:

X_z is the operating travel of the ring cylinder;
z is the number of friction discs.

The total surface of the friction discs (A):

$$A = \frac{(D^2 - d^2) \cdot \pi \cdot z}{4} \tag{5}$$

By substituting relations (3), (4), (5) described in detail above in the equation (2), multiplying by the medium radius of the clutch after simplification, with the help of the following relation and based on the construction characteristics of the clutch and the viscosity characteristics of the lubricant, the amount of fluid friction transmission occurring on the released clutch can be calculated.

$$M_S = \frac{n \cdot q_i \cdot (D^2 - d^2) \cdot (D^2 - d^2) \cdot \pi^2 \cdot z \cdot \eta \cdot \varphi}{8 \cdot x_z} \tag{6}$$

Where the fill factor φ, the value of which can be between 0 (dry clutch) and 1 (completely submerged in oil), was introduced as a supplement to describe the oil filling between the friction surfaces.

From the relations described, the following solutions can be suitable for reducing hydraulic losses:

- application of dry clutches;
- lower viscosity lubricant;
- lower oil filling (+ active lubrication);
- bigger lamellae gap;
- application of fewer clutches.

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