Influence of the Drive Train and Chassis on Power Shift Operations in Standard Tractors

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Abstract
The first partial power shift and full power shift transmissions for standard tractors were developed in the 1950s with mechanic and hydrostatic gearshift actuation systems [1], [2], [3]. These partial power shift transmissions usually had a two speed planetary drive in serial arrangement with a synchronized gearbox, whereas the full power shift transmissions consisted of a serial arrangement of several planetary drives with mechanic and hydrostatic gearshift actuation systems.

Today, electro-hydraulic gearshift actuation systems are state of the art and offer a wide range of opportunities to optimize power shift behavior and therefore handling comfort. On the other hand the number of setting parameters increases. To achieve an efficient transmission application process it is useful to have a general understanding of the influence of the dynamic tractor drive train and chassis behavior on power shift operations.

The influence is analyzed with the help of a complete tractor simulation model which is based on a given tractor. The resulting effects are evaluated by transmission output speed and torque.

1. Power Shift Transmissions for Standard Tractors
Till the end of the 1980s, partial and full power shift transmission concepts were unpopular in Europe [4]. Today, there are existing various numbers of different partial power shift transmissions as well as few full power shift transmissions. Partial power shift transmissions are offering two to eight power shifted ratios, combined with a synchronized range box, whereas full power shift transmissions offer 16 to 23 power shifted ratios [4]. Moreover, it became more likely to use a counter shaft design instead of planetary drives. To reduce the number of spur or planetary gears, the transmissions are designed as “grouped” transmissions with a serial arrangement of several sub-transmissions. Each sub-transmission offers two or more ratios. The resulting number of ratios is the product of the ratios per sub-transmission. This
leads to transmissions with a high amount of ratios, consisting of less spur or planetary gears [4]. But the demands on the gear shift actuation system increase with the number of sub-transmissions, especially for gear shifts involving several sub-transmissions. The basic power shift operation within one sub-transmission, including one engaging and one releasing clutch, is called a “one swop” gear shift. If two sub-transmissions are involved it will be called a “two swop” gear shift and so on. It is obvious that full power shift transmissions require such “multi-swop” gear shifts including several sub-transmissions. The same applies to some partial power shift transmissions, too.

2. Basic Power Shift Operation (“One Swop”)

Depending on the shift direction (up and down) and the torque flow direction (pulling and pushing) it is common to define four different shift conditions. Theoretically, an up-shift under pulling condition acts like a down-shift under pushing condition. The same applies for the other two conditions. Therefore, it is sufficient just to deal with up- and down-shifts under pulling condition. Regardless of the shifting direction, a basic power shift operation, consisting of one engaging and one releasing clutch, can be explained with the help of a simplified drive train model (Figure 1). It consists of two parallel power flow paths, each including one clutch and offering different ratios between the engine and the load. In case of slipping clutches during the power shift operation the drive train is divided in two dynamic subsystems. The first one includes inertia and stiffness of the engine and the primary transmission. The second one covers the secondary part of the transmission and the load.

Fig. 1: Simplified drive train model (“one swop”) based on [5]

Figure 2 shows the corresponding “one swop” power up shift process including several phases. It consists of a preparation phase, a torque transfer phase, a synchronization phase and a finalization phase. In the preparation phase the theoretical torque of the engaged
clutch is reduced by releasing the safety pressure against slippage. In addition, the hydraulic chamber of the other clutch is filled until the kiss-point with the friction plates is reached. During the torque transfer the power flow path changes from one gear ratio to the other, but no speed synchronization occurs. This is done by engaging the former disengaged clutch until it carries the applied torque. It is important that the former engaged clutch is fully released at the end of the phase and therefore it is continuously disengaged during the whole phase. The output torque decreases due to the changed gear ratio. In the synchronization phase the differential speed of the new engaged clutch is reduced by “overtorque” of the clutch and/or by engine torque control. If engine torque control is used the output torque remains nearly constant, whereas “overtorque” of the engaged clutch leads to an output torque increase. For comfortable down-shifts the order of torque transfer phase and synchronization phase must be interchanged. In the finalization phase the theoretical torque of the new engaged clutch increases by reapplying the safety pressure.

Fig. 2: Basic power up-shift process without engine torque control based on [5]

To perform comfortable power shifts, it is important to sense the transferred torque of the transmission at least during the preparation phase. This can be done by estimation, by torque sensors, by detecting differential speed while carefully releasing the engaged clutch until the slippage point is reached, or by a combination of these methods. Among these, only
the method of detecting differential speed gives the exact correlation between hydraulic pressure and required clutch torque. Therefore, this method is required for the final pressure release before starting the torque transfer or synchronization phase. It is used to control the differential speeds during the synchronization phase, too. Unfortunately, this method is influenced by the dynamic behavior of the drive train and the chassis.

3. Tractor Drive Train and Chassis Model
In order to analyze the influence of the drive train and chassis on the differential speed detection and therefore on the power shift operation itself, a complete tractor model optional with two and four wheel drive (2WD, 4WD) is created. The drive train model includes transmission and axle ratio, inertia and stiffness plus tire slippage, inertia and stiffness. The chassis model includes the component masses, a suspended front axle, a suspended cabin plus seat as well as dynamic axle load distribution. The drive train and chassis models are created with the help of AMESim and used in co-simulation with a Matlab model for the transmission controller.

To get a better understanding of the resulting effects, the main influencing components like inertia and stiffness are illustrated as a rough schematic drive train model (Figure 3). For simplicity reasons only one involved power shift clutch is shown.

Fig. 3: Schematic tractor drive train and chassis model
sion as well as the front and rear axle drive train plus tires. Part 3 represents the longitudinal and vertical tractor and implement behavior. The AMESim drive train model is more detailed with regard to inertia and stiffness distribution. In addition, part 3 of the model, representing the tractor and implement behavior, consists of a detailed 2D chassis model (Figure 3). Figure 4 helps to explain how the method of differential speed detection is influenced by the primary and secondary inertia connected to the clutch. While reducing the torque capacity of the clutch by reducing hydraulic pressure, the clutch will start to slip after a certain time. If the differential speed between primary and secondary clutch side exceeds a threshold value, the slippage is detected. At this point the pressure reduction will stop and the clutch torque capacity remains constant. If the stiffness between secondary clutch inertia and tire inertia is high, they can be seen as one huge inertia, decelerating or accelerating slowly. But in a tractor the stiffness of the tires and the axle drive train are low. Therefore, the torsion angle between secondary clutch inertia and tire inertia is high. In case of clutch slippage the tension releases and the small secondary clutch inertia decelerates or accelerates faster than the bigger tire inertia. This effect can result in quicker differential speed detection (Figure 4). As a consequence the slippage torque of the clutch remains on a higher level.

![Diagram of clutch and differential speed detection](https://doi.org/10.51202/97831810230006-439)

**Fig. 4:** Influence of shaft stiffness on differential speed detection

In a real tractor this stiffness is depending on the drive train configuration and the tires which can also vary in size and pressure. Moreover, if 4WD is engaged, the stiffness will be defined by the front and rear axle drive train, superposed by the influence of the front axle lead. In an optimal case the stiffness is the sum of both, but as the load and slip ratio of the front and rear axle vary, the resulting stiffness varies, too. Depending on the axle load distribution the stiffness is more similar to the front or the rear axle drive train.
4. Results of the Simulation Model

In Figure 5 the results of the tractor simulation model are shown as transmission output speed and torque during a power shift operation. The comparison includes simulations with 2WD and 4WD, as well as with concrete and clayey/sandy soil. With regard to the provided shift algorithm a power shift on concrete with 4WD results in the highest output torques and longest clutch synchronization time. With 2WD the synchronization time is shorter and therefore the torque peak is lower. The main reason is that the tire slippage of a 2WD tractor increases much earlier during the reengagement of the clutch than in a 4WD tractor. As a result the synchronization point is reached earlier and the peak torque reduces.

![Graph showing transmission output speed and torque during a power shift operation.](https://doi.org/10.51202/9783181023006-439)

Fig. 5: Simulated transmission output speed and torque during a power shift operation

The effect of a weaker drive train of a 2WD tractor compared to a 4WD tractor should result in earlier differential speed detection and therefore in a higher torque level at the slippage point, but it can be seen that the influence on the torque level is very small. Additional investigations with different tire stiffness support this conclusion.
The influence of different traction coefficient curves can be stronger than the influence of an engaged four-wheel drive, whereas the effect is very similar. A smooth traction coefficient curve like for clayey/sandy soil leads to an earlier synchronization based on higher tire slip-page and therefore to less jerky shifts.

5. Summary and Conclusion

In order to investigate power shift operations a detailed simulation model of a tractor is created. The model includes the complete tractor drive train plus chassis behavior. The simulation results show a remarkable influence of the chassis and drive train configuration on the power shift behavior, whereas the main reason is based on different tire-soil interaction. As a next step the simulations need to be validated with real tractor measurements.

References


