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Drivetrain, Chassis and Tire-Soil Contact Influence on Power Shift Operations in Standard Tractors

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Today, electro-hydraulic gearshift actuation systems offer a wide range of opportunities to optimize power shift behavior and therefore handling comfort of tractors. However, the parameter application process is very time consuming, especially with regard to increasing comfort demands. To reduce the time consumption and to increase the repeatability of the parameter application process, it is promising to transform the process from field tests to test benches. One main requirement for the transformation is to adjust the dynamic drivetrain behavior on the bench until it is comparable with tractor field operations. Therefore, it is necessary to understand the influence of the tire-soil contact, the chassis design and the drivetrain design on power shift operations. Within this article, the influences are analyzed with the help of a complete tractor simulation model which is based on a CLAAS Axion 800 Hexashift. The resulting effects are evaluated by tractor driving speed, transmission output speed and transmission output torque.

Keywords
Tractor transmission, power shift transmission, shift optimization, tractor drivetrain, drivetrain dynamics

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 (Erwin and O'Harrow 1959, Buckingham 1962, Harris and Jensen 1964). 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. Till the end of the 1980s, both power shift transmission concepts were unpopular in Europe (Renius 2014). Today, there are various numbers of different partial power shift transmissions existing, as well as a few full power shift transmissions. Typically, the partial power shift transmissions offer two to eight power shifted ratios, combined with a synchronized range box, whereas the full power shift transmissions offer 16 to 23 power shifted ratios (Renius 2014). Moreover, they are mainly based on a counter shaft design instead of planetary drives, now.

To design transmissions with a high amount of ratios, but with reduced spur or planetary gears, the transmissions are designed as "grouped" transmissions with a serial arrangement of several sub-transmissions (Renius 1974). Each sub-transmission offers two or more ratios. The resulting number of ratios is the product of the ratios per sub-transmission. However, the demands on the gearshift actuation system increase because some gear changes require a simultaneously power shift within several sub-transmissions (Förster and Gaus 1971, Kraft 1972). For classification, the basic power shift operation within one sub-transmission, including one engaging and one releasing

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clutch, will be called a “one swap” gearshift. If two sub-transmissions are involved, it will be called a “two swap” gearshift, including two engaging and two releasing clutches. If three sub-transmissions are involved it will be called a “three swap” and so on. Full power shift transmissions require such “multi-swap” gearshifts including several sub-transmissions. The same principles apply to some partial power shift transmissions too.

As mentioned above, the demands on the gearshift actuation system raise, especially for gearshifts involving several sub-transmissions. These gearshifts are difficult to handle as the number of involved clutches increases and the energy management becomes more complex due to the higher moments of inertia during one gearshift (Eike and Stoever 1999). As a result the electro-hydraulic shift parameter set enlarges and the required time for the parameter application process increases. For example, such shift parameter sets need to cover the filling time and corresponding kiss-point pressure for all clutches, which are furthermore depending on oil temperature and rotational speed of the clutch. The timing and pressure modulation during the gearshift need to be defined and stored in the parameter set too. In addition, power shift transmissions are more and more in competition with continuously variable transmissions (CVT) and therefore handling comfort is becoming more important. Handling comfort can be optimized by more carefully chosen shift parameters which enlarge the shift parameter set too. Especially for agricultural tractors the shift parameters can vary between different gears and load conditions as well as with different implements and working conditions. However, the gearshift algorithm and parameter set need to be robust enough to provide a smooth gearshift in any condition (Tanelli et al. 2011). Consequently, a repeatable and fast parameter application process is required to reduce the development time and cost and to improve gearshift quality. A transformation of the application process from field tests to test benches seems to be a promising approach, at least in the automotive sector (Hagerodt 2003). However, as a main requirement, the dynamic drivetrain behavior on the bench should be comparable with its corresponding behavior on a tractor. Due to this, it is necessary to analyze how the tire-soil contact, the chassis design and the drivetrain design influence power shift operations. As a result of the analysis the differences can be evaluated and the test bench can be designed in an optimal way.

As a basis for the investigations, the basic power shift operation and a simplified sketch of a detailed tractor model will be presented. With the help of this theoretical approach the drivetrain, chassis and tire-soil related influencing effects on power shift operations are going to be explained. To which extent the effects influence a power shift operation will be evaluated by the results of the detailed tractor model. This model is parameterized to simulate the power shift behavior of a CLAAS Axion 800 Hexashift.

Basic Power Shift Operation ("One Swap")

All power shift operations, even the complex "multi-swap" gearshifts, refer to the main characteristics of basic power shift operations. A basic power shift operation, consisting of one engaging and one releasing clutch, can be explained with the help of a simplified drivetrain model (Figure 1). It consists of two parallel power flow paths, each including one clutch and offering one or more different ratios between the engine and the load. The engine and load are described by speed, torque and inertia. During the power shift operation the power flow shall switch from one path to the other without significant torque interruption. In case of slipping clutches during the power shift operation the drivetrain 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. Both subsystems adjust their speed individually and they are only connected by the transferred torque of the clutches.

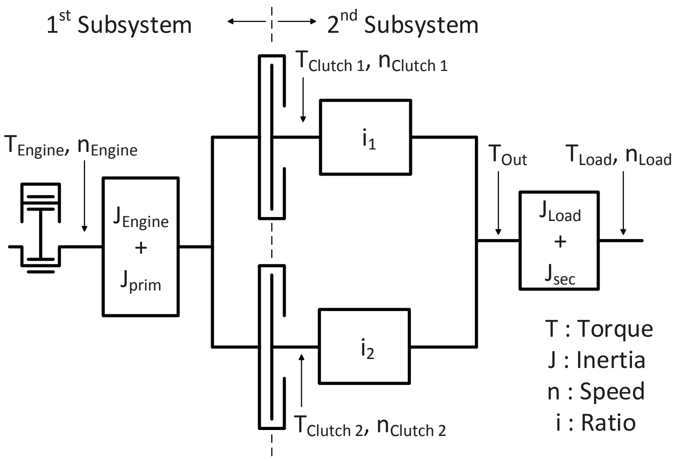


Figure 1: Simplified drivetrain model ("one swap") based on Fischer et al. (2012)

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 upshift under pulling condition acts like a downshift under pushing condition (Alvermann 2008). The same applies for the other two conditions. Therefore, it is sufficient just to deal with up- and downshifts under pulling condition. As a first approach, the following investigations are based on upshifts under pulling condition. Figure 2 shows the idealized corresponding basic "one swap" power upshift process. For simplification the engine torque is kept constant during the whole process and the load is slightly accelerating. The process 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 until the slippage point is reached. At this point the theoretical torque of the clutch equals the real applied torque. Simultaneously, the hydraulic chamber of the other clutch is filled with oil until the kiss-point with the friction plates is reached. The required time to reach the kiss-point should be smaller than the required time to reach the slippage point of the engaged clutch. At the kiss-point no torque will be transferred, but any further pressure increase will directly result in a torque transfer. This is the reason why the precise time of reaching the kiss-point cannot be seen in Figure 2.

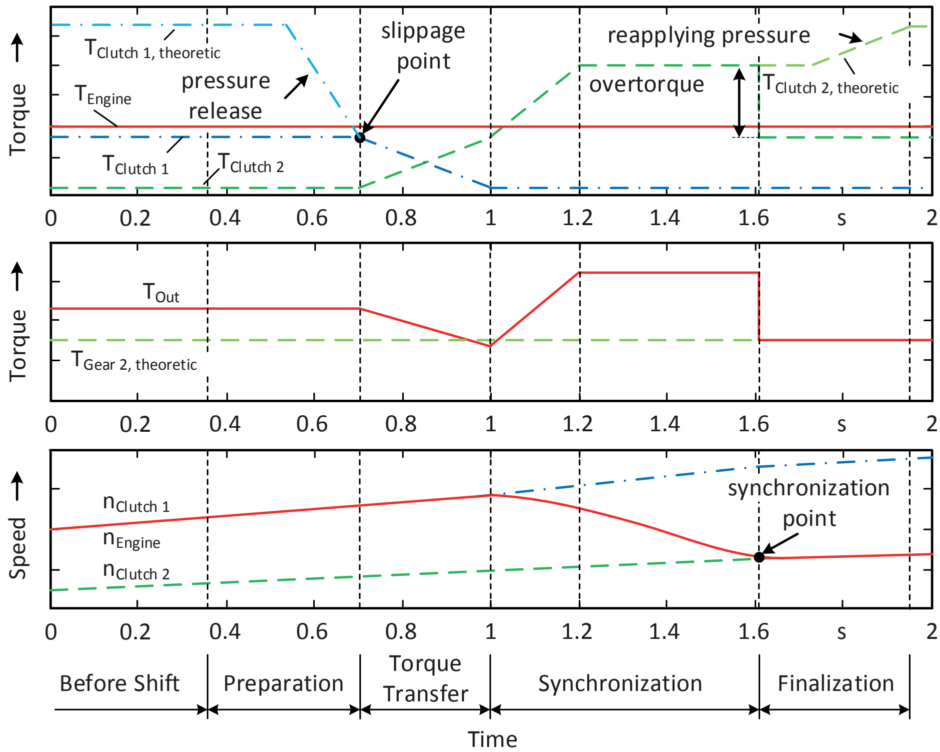


Figure 2: Basic power upshift process without engine torque control based on Fischer et al. (2012)

During the torque transfer phase 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. The former engaged clutch must be fully released when the new engaging clutch is able to carry the applied engine torque. This is the case at the end of the phase. If the clutch is not fully released, it will cause inner tension of the transmission which leads to high output torque reduction. Due to these reasons, the safety pressure of the disengaging clutch must be released during the previous preparation phase to detect the applied engine torque and it is continuously disengaged during the torque transfer phase. Nevertheless, the output torque decreases during the torque transfer because of the changed gear ratio. In the synchronization phase the differential speed of the new engaged clutch is reduced by “over-torque” of the clutch and/or by engine torque control. Figure 2 shows the method of “over-torque” that was the most commonly used method for tractors in the past. Using this method means to increase the hydraulic pressure of the new engaging clutch until the transferred torque of the clutch is higher than the applied engine torque during the synchronization phase. This forces the engine to decelerate till the synchronization point of the clutch is reached at the end of the synchronization phase. However, this method causes an output torque increase equivalent to the over-engagement of the clutch. When the synchronization point is reached and the differential speed of the clutch is zero, the over engagement of the clutch and the increased output torque suddenly decrease down to the level of the applied engine torque. Both, the increase and the sudden torque decrease, cause jerks which are uncomfortable for the driver and lead to wear of friction linings of the clutch. If the method of engine torque control is used, the new engaged clutch will not be over engaged and the transferred torque equals the output torque after the power shift.

Consequently, the output torque remains nearly constant and no jerk occurs. To reduce the differential speed of the clutch, the engine torque is reduced and its speed goes down. However, the engine torque must be reapplied early enough before the synchronization point is reached, otherwise the output torque decreases. In the finalization phase the theoretical torque of the new engaged clutch increases by reapplying the safety pressure. For downshifts, the order of torque transfer phase and synchronization phase must be interchanged to improve driving comfort.

As mentioned above, it is important to identify the transferred torque of the transmission at least during the preparation phase to perform comfortable power shifts. 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 drivetrain and the chassis as explained in the following chapter.

Design of the Tractor Simulation Model

In order to analyze the drive train, chassis and tire-soil influence on power shift operations a detailed simulation model is required. First of all, this model should include the main drive train components, which are engine, transmission, axles and tires as well as their parameters to describe their dynamic behavior. For the mechanical components these are the torsional spring rate, damping and inertia characteristics. For the engine and the transmission the dynamic behavior of their controllers need to be included. The axle and tire models need to cover two and four wheel drive (2WD, 4WD) as well as tire slippage. Furthermore, to analyze the influence of the tire-soil interaction correctly, a 2D tractor chassis model is required. This model should include the tractor body component masses, a suspended front axle, a suspended cabin including a seat, as well as dynamic axle load distribution and variable implement forces acting on the tractor.

Both, the drivetrain and the chassis model are created with the software „LMS Amesim“ and are used in co-simulation with a model (software „Matlab“) for the transmission controller. To get a better understanding of the detailed tractor model and to get a first impression of the drivetrain, the chassis and the tire-soil influence on power shift operations, the main model components like inertia and torsional spring rate are illustrated as a rough schematic drivetrain model in Figure 3. For simplicity reasons, only one involved power shift clutch is shown.

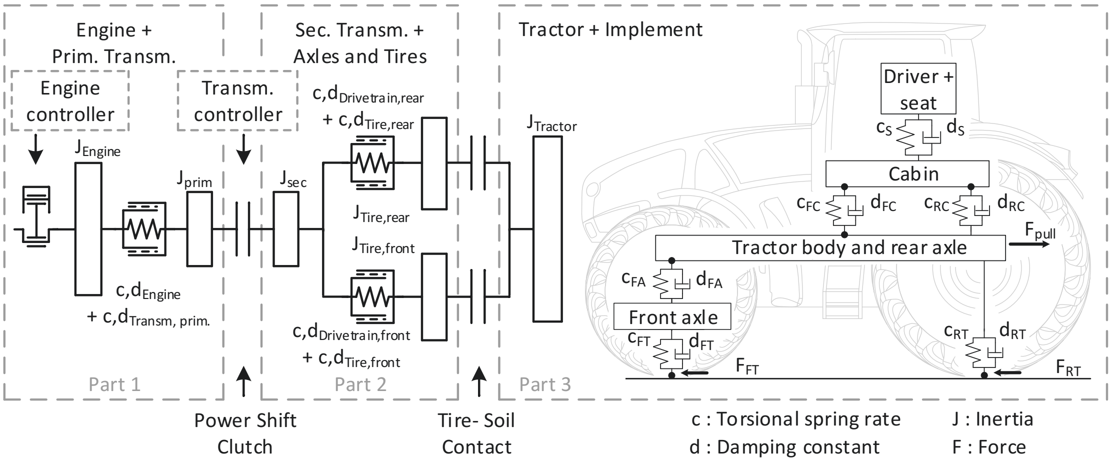


Figure 3: Schematic tractor drivetrain and chassis model

The model consists of three dynamic parts which are divided by the power shift clutch and the tire-soil contact of the front and rear axle (Figure 3). Part 1 represents the engine plus the primary part of the transmission. Part 2 includes the secondary part of the transmission as well as the front and rear axle drivetrain plus tires. Part 3 represents the longitudinal and vertical tractor behavior including pitching. Vertical and longitudinal implement forces act on the tractor to simulate implement forces. The Amesim drivetrain and chassis model is more detailed with regard to inertia and torsional spring rate distribution.

Preliminary Consideration of Tire-Soil Contact and Drivetrain Stiffness Influence

The power shift clutch and the tire-soil contact divide the schematic tractor drivetrain model into three dynamic parts (Figure 3). All three parts are arranged in a row and are only connected by the transferred torque of the power shift clutch and the tire-soil contact. Each of the dynamic parts adjusts its speed individually, depending on the acting torques and inertias. Due to this, it can be assumed that the slippage in the tire-soil contact will have an influence on the differential speed or slippage of the power shift clutch, at least during the synchronization phase.

In addition, Figure 3 gives an impression of the main distribution of the torsional spring rate and inertia for the tractor drive train model. Both can have an influence on the method of differential speed detection and therefore on the power shift operation itself. The responsible effect will be explained with the help of Figure 4 in more detail. In general, the method of differential speed detection starts with reducing the torque capacity of the power shift clutch by reducing the hydraulic pressure (Figure 4a). If the pressure reduction exceeds its theoretical target value, the clutch will start to slip (Figure 4a, t_S). Due to the shaft inertia, the primary and secondary clutch side accelerate and decelerate with a time delay and the differential speed will increase (Figure 4a). If the differential speed exceeds a threshold value, the slippage will be detected and the pressure reduction stops (Figure 4a, t_1 and t_2). At this point the final clutch torque is reached and remains constant, but on a lower level than the theoretical target value. Due to the described reasons the response time is mainly influenced by the acceleration of the primary and secondary clutch inertia. The acceleration itself depends on the torsional spring rate of the shaft and the inertia. If the torsional spring rate between secondary clutch inertia and tire inertia is high, the torsion angle between both inertias is low (Figure 4c). They can be seen as one huge inertia, decelerating or accelerating slowly. As a result the differential speed of the clutch increases slowly (Figure 4a, t2). If the torsional spring rate is low, the torsion angle between secondary clutch inertia and tire inertia is high (Figure 4d). In case of clutch slippage the tension releases and the torsion angle reduces. Therefore, the small secondary clutch inertia decelerates or accelerates faster than the bigger tire inertia. This effect results in quicker differential speed increase (Figure 4a, t_1). As a consequence the slippage point of the clutch is detected earlier and the torque remains on a higher level.

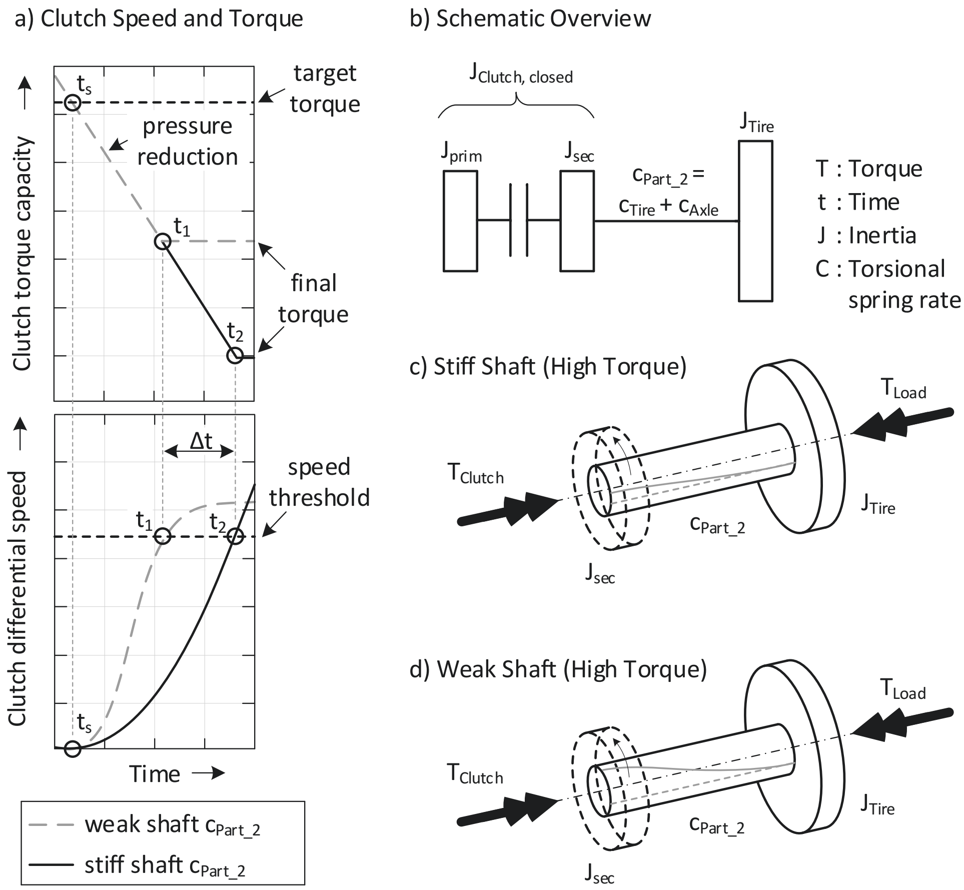
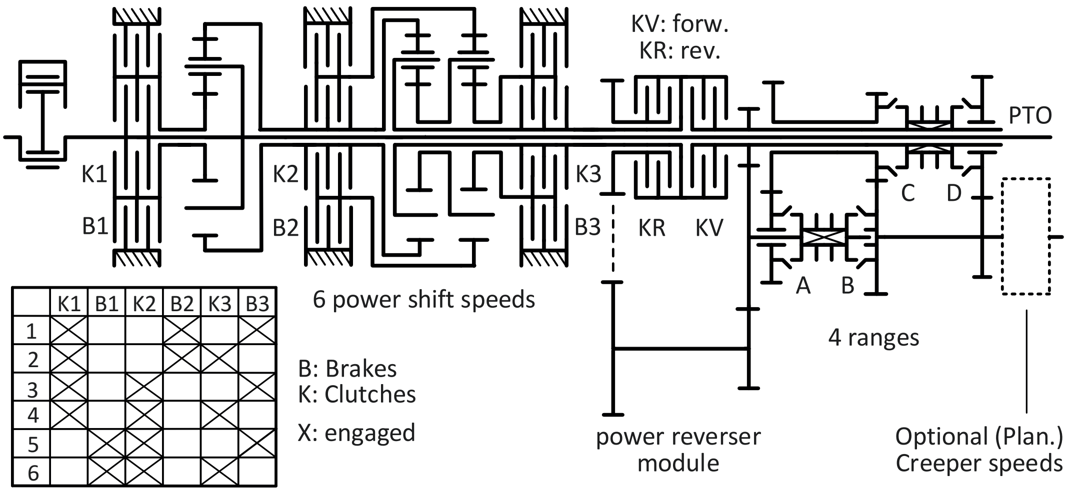


Figure 4: Influence of the torsional spring rate of the shaft on differential speed detection

In a real tractor the torsional spring rate between secondary clutch inertia and tire inertia is low because of the high rear axle ratio. In addition, the torsional spring rate is depending on the drivetrain configuration and the tires which can also vary in size and pressure. Moreover, if 4WD is engaged, the torsional spring rate will be defined by the front and rear axle drivetrain, superposed by the influence of the front axle lead. In an optimal case, the torsional spring rate is the sum of both but, as the load and slip ratio of the front and rear axle vary, the resulting torsional spring rate varies too. Depending on the axle load distribution the torsional spring rate is more similar to the front or to the rear axle drivetrain.

Tractor Model Configuration and Scope of Investigation

The following investigations are based on a „CLAAS Axion 800“ equipped with a “Hexashift” transmission (Figure 5). The Hexashift is a partial power shift transmission consisting of a planetary gear set for the power shift speeds at the input, a power reverser module in the middle and a synchronized range box at the output of the transmission. It offers six power shift speeds (1–6) and four synchronized ranges (A-D) resulting in 24 speeds (A1-D6) in total. The power reverser module consist of two clutches, one for forward and one for reverse operation which can be switched while driving.



	K1	B1	K2	B2	K3	B3
1	X			X		X
2	X			X	X	
3	X		X			X
4	X		X		X	
5		X	X			X
6		X	X		X	

Figure 5: Stick diagram of GIMA Hexashift transmission for CLAAS Axion 800 series based on Renius and Geimer (2008)

The corresponding parameters for the complete tractor model are based on CAD data, component data sheets and on unpublished measurements regarding the longitudinal and vertical tire parameters. These parameters are measured according to the considerations of Brinkmann et al. (2005) and Ferhadbegovic (2005). The tire parameters are assumed as constants for the simulation model. The parameters to describe the tire-soil contact are extracted from Söhne (1963). The used power shift algorithm is based on the serial software but kept very simple to point out the main characteristics. Unlike the described basic power shift operation, a power shift operation within the Hexashift transmission is more complex because the power reverser module is involved. During the preparation phase of the overall power shift operation the pressure of the engaged clutch of the power reverser module is reduced by a pressure control valve until the slippage point is reached. Afterwards, an internal power shift within the planetary gear set is performed. This internal power shift behaves like a basic power shift operation too, but as the brakes are controlled by on/off valves and the clutches are mechanically connected to the movement of the brakes, no pressure modulation is possible. Referring to the explanations of the basic power shift operation this results in torque peaks. The internal power shift can be seen as the torque transfer phase of the overall power shift operation. The slipping power reverser clutch filters the occurring torque peaks and smooths out the power shift. During the synchronization phase the power reverser clutch is carefully reengaged to synchronize the clutch by the method of “over-torque” without any specific engine torque control command. However, the engine speed governor increases the engine torque during the synchronization to adjust the commanded engine speed. When the synchronization point is reached the safety pressure against slippage is reapplied.

Even if the overall power shift operation of the Hexashift transmission is different to the basic power shift operation, it offers the main characteristic behavior and can be used for the investigations. The investigations base on exemplary power shifts of a non-ballasted tractor with constant engine speed command. In total, three different upshifts will be simulated to investigate the influence of the transmission ratio (Table 1). The first variant equals a power shift at 10 km/h driving speed with constant engine speed command and a vehicle load that causes middle engine load (Table 1, variant I). This should represent a frequently occurring, general load condition. Based on the explanations regarding to Figure 4, a remarkable influence of the torsional spring rate between the secondary clutch inertia and the tire inertia is expected. This torsional spring rate varies due to the different ratios of the four transmission ranges. Therefore, the other two simulation variants deal with a power shift in the lower and the upper transmission range. The power shift gear remains the same for all simulation variants. (Table 1, variants II and III). In the lower range the output load of the transmission stays constant and in the upper range the engine load stays constant in comparison to the power shift variant I. This is necessary to prevent excessive tire slippage or engine stalling.

Table 1: Overview of the different power shift variants

Power shift variant	**Variant I**	**Variant II**	**Variant III**
Engine speed command	1600 1/min	1600 1/min	1600 1/min
Gear upshift	C4 – C5	B4 – B5	D4 – D5
Driving speed in front of the gearshift	10 km/h	5.5 km/h	30 km/h
Vehicle load	middle engine load	output load like variant I	engine load like variant I

The corresponding power shift operations are simulated on concrete and clayey/sandy soil. This allows for the analysis of different tire-soil configurations. A variation between 2WD and 4WD is added to extend the tire-soil investigations by different chassis configurations. In addition, the drive train stiffness varies between 2WD and 4WD so that the influence on the differential speed detection can be analyzed too. The stiffness variation is extended by a synthetic simulation with a nearly rigid tire. This simulation configuration covers the conditions on a transmission test bench and gives an impression of the associated dynamic power shift behavior. Finally, five different simulation configurations result for each of the three different power shift variants.

Results of the simulation model

In Figure 6 the results for the five different simulation configurations of the tractor model for power shift variant I are shown. The general shapes of the transmission output speed, transmission output torque, power reverser differential speed and tractor drive speed are linked to the provided, simplified power shift algorithm.

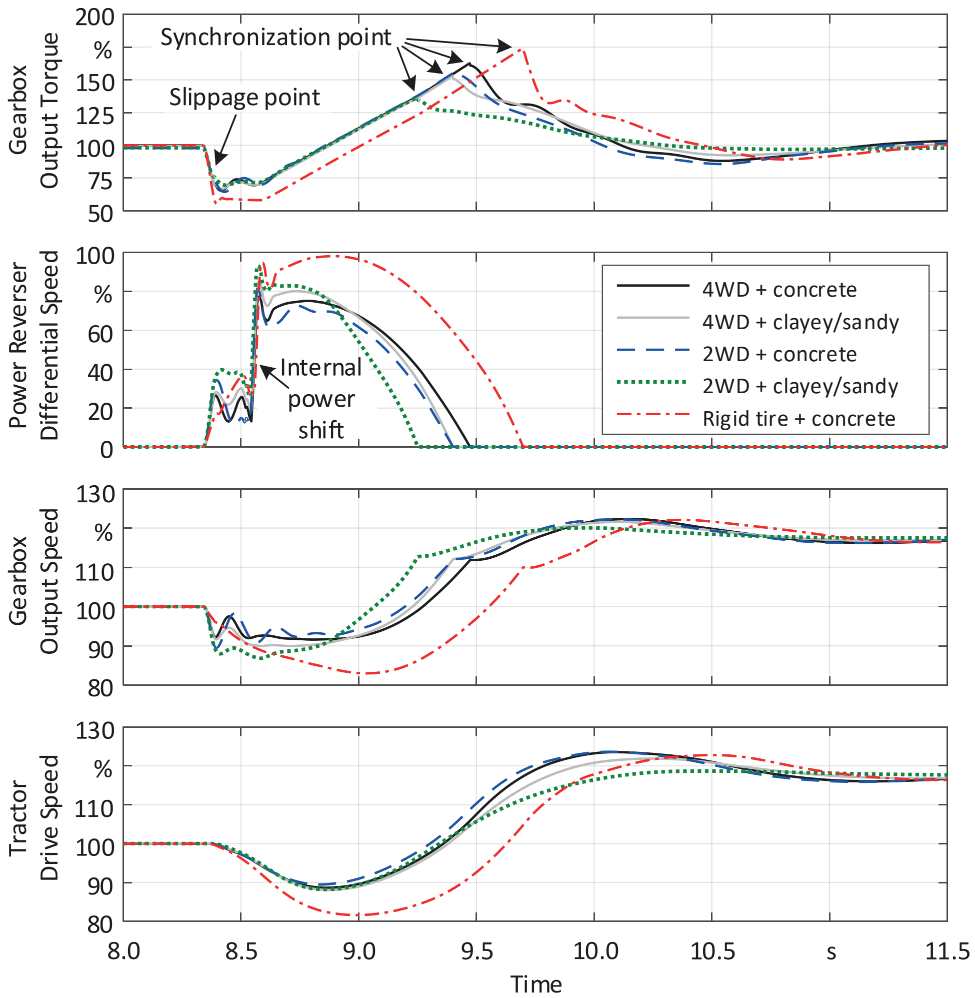


Figure 6: Simulated transmission output speed, transmission output torque, power reverser differential speed and tractor drive speed during power shift variant I

At the beginning of the shift, the pressure of the power reverser clutch is reduced until the slippage point occurs. At this point the transmission output torque reduces by 25% due to the time delay for the differential speed detection. Within the simplified simulation no subsequent pressure increase is commanded to compensate this occurring torque reduction. As a result the output torque stays constant till the internal power shift is finished. Due to this, the driving speed reduces by 10%. The internal power shift itself increases the differential speed of the power reverser clutch. Afterwards, during the synchronization phase, the output torque increases by a linear ramp which is proportional to the linear pressure increase of the power reverser clutch. When the synchronization point is reached, the output torque suddenly decreases and oscillates with the natural frequency of the drivetrain.

With regard to the provided shift algorithm a power shift on concrete with 4WD results in the highest output torque at the synchronization point and the 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 because the transmission torque is transferred by four wheels instead of two wheels. The simulation results for clayey/sandy soil show a comparable effect between 2WD and

4WD but the synchronization points are still reached earlier and the torque peaks are still lower. This means that the complete synchronization is partly covered by clutch slippage and by additional wheel slippage. The weaker the traction coefficient of the soil, the more tire slippage and the less clutch slippage occurs. As a result the tractor drive speed increases much smoother with 2WD than with 4WD and on clayey/sandy soil than on concrete soil.

It was expected that the 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. However, the simulation results show that the effect can be neglected based on the little torsional spring rate variation. In case of the rigid tire simulation the torsional spring rate variation is much higher. This leads to a considerably slower transmission output speed decrease at the beginning of the gearshift, resulting in a longer response time of the differential speed detection. Therefore, the output torque decrease and the tractor drive speed decrease are higher, the peak torques are higher and the synchronization time last longer. As a consequence, the overall dynamic power shift behavior of this synthetic configuration is clearly different compared to the normal power shift behavior of a tractor.

In Figure 7 the results of the five different simulation configurations for power shift variant II are shown. The general shape of the power shift bases on the provided, simplified power shift algorithm, like it is the case for power shift variant I. In contrast to the results of power shift variant I, the previous remarkable difference between the simulation of the rigid tire and the other configurations is little. One reason for the small difference is the higher ratio of range B in comparison with range C, which minmizes the reduced torsional spring rate of the drivetrain on the power reverser shaft. As a result, the time delay of the differential speed detection is minimized for all simulation configurations. Therefore, the output torque reduction as well as the differences between them gets smaller. However, the overall percentage of the output torque reduction is 40 %, which is higher compared to power shift variant I. This is caused by the lower torque level within the power reverser clutch, resulting from the higher ratio of range B in conjunction with the constant output torque level compared with variant I. Further simulations show less percentage of output torque reduction if the same torque level, like for variant I, is used within the power reverser clutch. The percentage of transmission output speed and tractor driving speed reduction is 40 %, which is higher than the result for the rigid tire simulation of variant I. The main reason for this is the lower speed level of the tractor. During the synchronization phase the differences regarding the synchronization point and the torque peaks between 2WD and 4WD are comparable with the results of simulation variant I. The same applies for the differences between clayey/sandy soil and concrete.

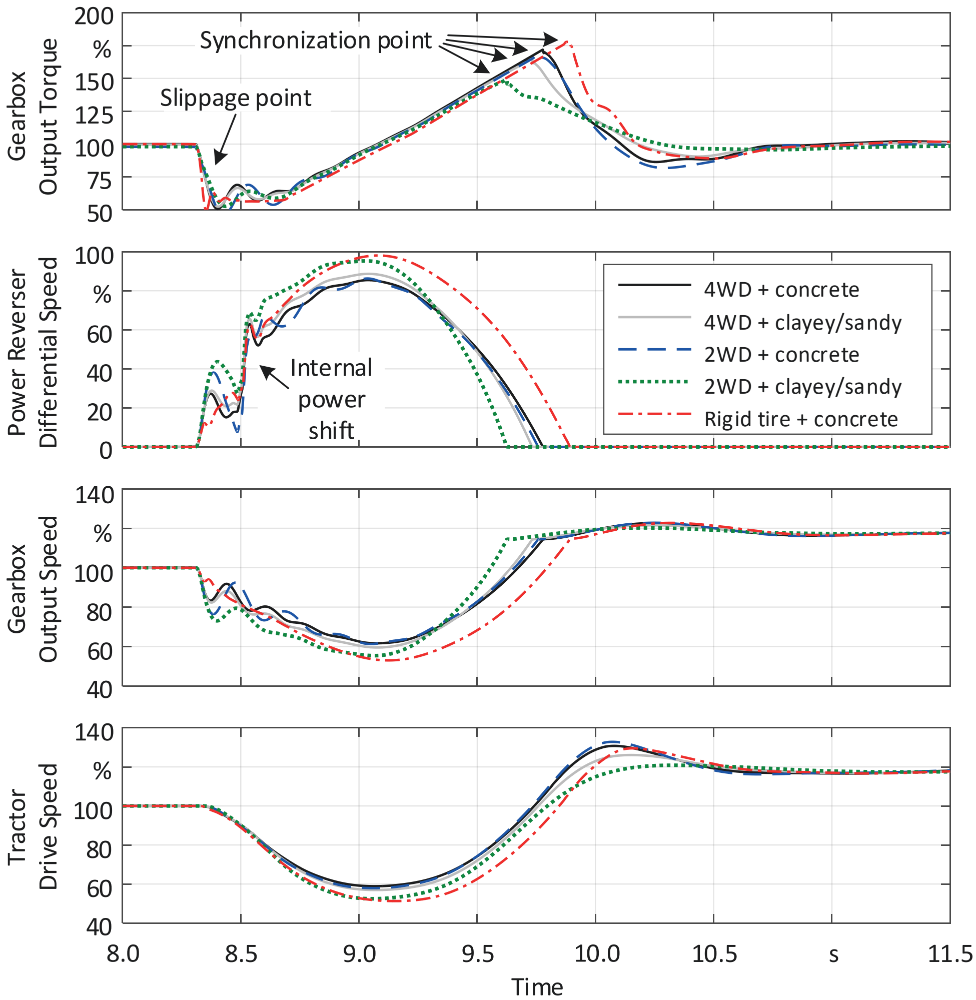


Figure 7: Simulated transmission output speed, transmission output torque, power reverser differential speed and tractor drive speed during power shift variant II

In Figure 8 the results of the simulation configurations for power shift variant III are shown. Within this variant the ratio of the transmission range is lower than for power shift variant I. Consequently, the reduced torsional spring rate of the drivetrain on the power reverser clutch increases for all simulation configurations as well as the reduced tire inertia. With respect to the explanations for Figure 4, both effects lead to longer time delays for the differential speed detection and to higher output torque decrease at the slippage point. The output torque decrease is about 60 to 80%. This is a higher absolute decrease in comparison to variant I, because of the same torque level within the power reverser clutch. The higher level of the reduced torsional spring rate of drivetrain for power shift variant III leads to higher absolute differences between the reduced torsional spring rate of the drivetrain for the five simulation configurations. As a result there are remarkable differences between the output torque reductions of the 2WD, the 4WD and the rigid tire configuration. The percentage of transmission output speed reduction and tractor driving speed reduction is 3 to 5%, which is less compared to the results of power shift variant I. Comparing the percentage of speed reduction to the higher driving speed level of power shift variant III, the absolute driving speed reduction remains on a comparable level with power shift variant I, even if the output torque reduction is higher.

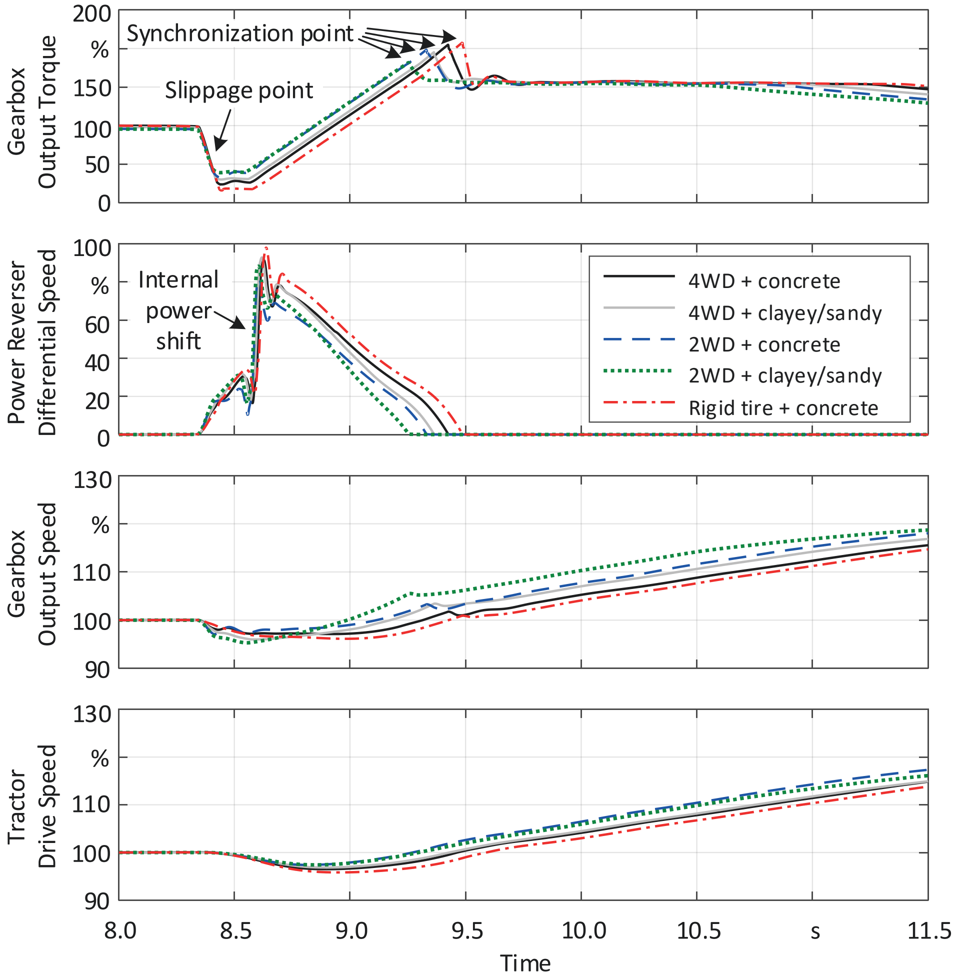


Figure 8: Simulated transmission output speed, transmission output torque, power reverser differential speed and tractor drive speed during power shift variant III

During the synchronization phase, one can recognize the same qualitative differences between clayey/sandy soil and concrete as in the previous configuration, with respect to the synchronization point. After finishing the power shift, subsequent to the synchronization point, there is no constant driving speed level within the shown time interval. This is caused by the engine, which needs more time to accelerate the vehicle when driving in a higher gear.

Conclusions

Power shift transmissions can be a cost effective alternative to CVTs if they offer several gears and small gear steps. Grouped transmission design is a favorable solution for this. Moreover, driving comfort is of increasing importance because power shift transmissions are in competition with CVTs. Both factors increase the demand for enlarged and optimized power shift parameter sets. Enlarged power shift parameter sets require time consuming parameter application processes. A transformation of the parameter application process from field tests to test benches is a promising method to improve the gearshift quality and to reduce the application time. As a basis for this transformation the influences of the drivetrain and chassis design as well as the influences of the tire-soil contact on a power shift operation need to be investigated. In order to do this, the basic power shift operation is explained in theory and it is pointed out that differential speed detection is a very crucial method to perform comfortable power shifts. Furthermore, a detailed simulation model of a tractor is created. Based on a schematic view of this model, it is explained how the torsional spring rate of the drivetrain and the tire-soil contact can influence the power shift operation. The results of the detailed simulation show a remarkable influence of the tire-soil contact which acts like an overload protection and smooths jerky power shift operations by increasing slippage. Weaker traction coefficient curves lead to smoother power shifts. The same applies for 2WD in comparison with 4WD. The influence of different torsional spring rates between 2WD and 4WD drivetrains depends on the ratio of the transmission range between the power shift clutches and the tires. The differences between 2WD and 4WD can be neglected in a lower transmission range with a higher gear ratio. Even a higher torsional spring rate variation, like in the case of the rigid tire configuration, lead to imperceptible differences. The same applies to the differences between 2WD and 4WD in the medium transmission range. However, a higher torsional spring rate variation, caused by a rigid tire configuration, leads to a remarkable different dynamic behavior in the medium transmission range. The main reason is a longer time delay for the differential speed detection, leading to a lower torque level at the beginning of the power shift. In the higher transmission range, a variation between 2WD, 4WD and the rigid tire configuration lead to different time delays for the differential speed detection. Due to this the resulting torque levels at the slippage point differ too. However, the percentage of transmission output speed reduction and driving speed reduction is lower than in the medium transmission range.

As a conclusion, a well-designed test bench drivetrain for power shift application shall at least offer a torsional spring rate which is comparable to a real tractor drivetrain. This can be challenging, mainly for pure transmission test benches. Otherwise, especially in the medium transmission range the occurring differences of the dynamic behavior can be obvious. For an even more realistic power shift behavior the tire-soil contact should be taken into account. Finally, to improve validity of the results and to optimize the simulation model, field measurements with a real tractor are planned for the future.

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