Efficient analysis of complex continuously variable power split transmissions with multiple in- and outputs

Dipl.-Ing. **V. Warth**, ZF Friedrichshafen AG, Friedrichshafen[1]
Dipl.-Ing. **B. Reick**, ZF Friedrichshafen AG, Friedrichshafen[1]
Prof. Dr.-Ing. **B. Bertsche**, Universität Stuttgart, Stuttgart
Prof. Dr.-Ing. **M. Geimer**, Karlsruher Institut für Technologie, Karlsruhe

Zusammenfassung

Moderne Traktoren besitzen Fahrantriebe mit mehreren Leistungsausgängen für unterschiedliche Energieformen, z.B. die Zapfwelle, die Arbeitshydraulik, den Fahrantrieb oder elektrische Steckdosen. Eine schnelle und einfache Berechnung von Leistungen und Drehzahlen neuer Getriebekonzepte ist in der frühen Phase der Produktentwicklung besonders wichtig. Es wird daher, aufbauend auf dem Stand der Technik, eine Methodik zur effizienten Darstellung und Berechnung solcher Getriebestrukturen vorgestellt und anhand eines ausgewählten Beispielsystems erläutert. Abschließend werden der Leistungsfluss und der Getriebewirkungsgrad dreier Getriebestrukturen in unterschiedlichen Detaillierungsstufen verglichen. Die Ergebnisse unterstreichen die Effizienz und Flexibilität der erarbeiteten Methodik für unterschiedliche Anwendungsgebiete.

Abstract

Modern tractors have drives with several power outputs for different forms of energy, e.g., the PTO shaft, the working hydraulics, the vehicle drive or electrical power sockets. Quick and simple calculation of power and speed of new transmission concepts is particularly important in the early phase of product development. Thus, basing on the state-of-the-art a method for efficiently describing and calculating such transmission structures is presented and explained using a specific exemplary system. As a conclusion, the power flow and the transmission efficiency of three transmission structures are compared in different levels of detail. The results underline the efficiency and flexibility of the developed method for different areas of application.

---

[1] equally contributed

1. Introduction

Continuously Variable Transmissions (CVT) enable to automatically set the engine operating point independent on the loads applied to the vehicle. Using a CVT, the performance, comfort and efficiency of vehicle drive systems can be ideally matched to the driver's needs and the tasks to be fulfilled [8]. Figure 1.1 shows a systematic overview of different designs of continuously variable transmissions. For tractors, hydraulically power-split CVTs in the medium power class have already been established for some time. Furthermore, there is a trend to use this technology also in high and low power classes to an increasing extent [5].

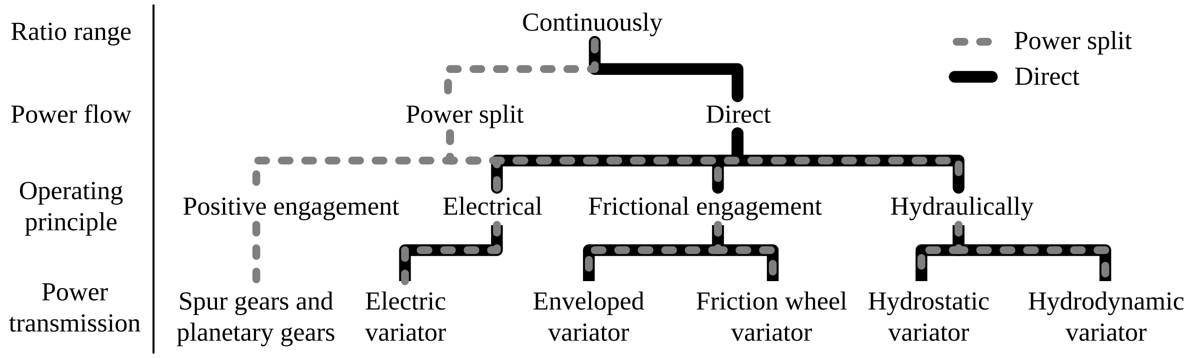


Figure 1.1 Overview of continuously variable transmissions

2. Basics

A description and calculation method is presented in the following enabling the efficient and automated analysis of such transmission systems.

2.1 State of the art

Planetary gears can be numerically described using a so-called transmission code. In a first step, the individual shafts must be indexed by numerical values z. The input shaft has the index one, the output shaft is indexed with two and housing-mounted shafts with zero. All other planetary gear shafts are numbered in an ascending order. Thus, a number N_{PS} of planetary gearsets PS_j can be mathematically described as a numerical order of the three shafts with the sequence sun $z_{s,j}$, carrier $z_{c,j}$ and ring gear $z_{r,j}$. Furthermore, an existing number N_{SE} of switch elements SE_m can be described as a combination of the numbers of the shafts A_m and B_m to be connected. If the code of all planetary gearsets is then combined with the one of the switch elements, the transmission code is created according to [6]. This mathematical description together with the calculation method from VDI 2157 enables an automated analysis of kinetics, kinematics as well as the efficiencies of conventional planetary transmissions [10]. Apart from planetary gears, dual clutch transmissions can also be calculated by assigning shaft numbers and adapting the VDI 2157 method. However, no definite description of the total transmission using transmission codes is known [7].

To calculate the speeds in a compound planetary gear, the Willis equation is used for every planetary gearset, spur gear pairs, however, are calculated using the angular velocity ratio of the spur gear shaft A and the second spur gear shaft B [10]. A boundary condition and the constraints by the closed switch elements together with the equations mentioned above form a linear equation system in matrix notation consisting of a square coefficient matrix $\underline{A}_\omega$, an unknown vector of rotational speeds $\vec{\omega}$ as well as the solution vector $\vec{b}_\omega$, refer to equation (1). For simplification, $\underline{A}_\omega$ can be split up into three sub-matrixes consisting of the assigned boundary condition $\underline{A}_{\omega,RB}$, the equations of the planetary and spur gear stages $\underline{A}_{\omega,UE}$ as well as the equations of the closed switch elements in $\underline{A}_{\omega,SEG}$. The solution vector $\vec{b}_\omega$ contains the numerical value of the defined boundary condition r_ω.

$$\underline{A}_\omega \cdot \vec{\omega} = \vec{b}_\omega \qquad \underline{A}_\omega = \begin{pmatrix} \underline{A}_{\omega,RB} \\ \underline{A}_{\omega,\mathrm{UE}} \\ \underline{A}_{\omega,SEG} \end{pmatrix} \qquad \vec{b}_\omega = \begin{pmatrix} r_\omega \\ 0 \\ 0 \end{pmatrix} \tag{1}$$

In case of a compound planetary gear the torques are calculated by considering the torque equilibrium at the planetary gearset as well as the relation of the torque ratio of both bull gears and the fixed carrier gear train ratio which is independent of the component's form. Spur gear stages are included by the torque ratio of the spur gear on shaft B and the spur gear on shaft A. Furthermore, the torque equilibrium of the elements with identical shaft numbers is to be considered [10]. These equations and the definition of a boundary condition as well as the specification of a torque zero for opened switch elements are required to set up a linear equation system in matrix notation. Apart from the sub-matrixes $\underline{A}_{T,RB}$ and $\underline{A}_{T,UE}$, the coefficient matrix also contains the opened switch elements $\underline{A}_{T,SEO}$ and the torque equilibrium of the shafts $\underline{A}_{T,W}$.

$$\underline{A}_T \cdot \vec{T} = \vec{b}_T \qquad \underline{A}_T = \begin{pmatrix} \underline{A}_{T,RB} \\ \underline{A}_{\mathrm{T,UE}} \\ \underline{A}_{T,SEO} \\ \underline{A}_{T,W} \end{pmatrix} \qquad \vec{b}_T = \begin{pmatrix} r_T \\ 0 \\ 0 \end{pmatrix} \tag{2}$$

By integrating a stationary transmission efficiency η_0 as well as the corresponding pitch power factor w into the torque equations, the lossy torque can be calculated [10, 7]. The pitch power factor includes the pitch power's flow direction and thus is responsible for assigning the input or output torque to the corresponding bull gears.

By multiplying their inverted matrix $\underline{A}^{-1}$ by $\vec{b}$, the linear equation systems of kinematics and kinetics can be solved. Based on the determined values, the power output can be calculated and the transmission concept can be analyzed and evaluated. The load-dependent gear efficiency can be roughly calculated by relating the ideal $T_{ab,id}$ to the lossy output torque $T_{ab,v}$, [10].

As early as in 1958, *WOLF* [11] considered the illustration of transmission elements using simple symbols - e.g., the epicyclic gearing as circle with three lines radially pointing outwards - to be quite helpful when analyzing complex transmission structures. To analyze continuously variable transmission structures with several inputs and outputs, a similar symbol system was developed, refer to figure 2.3. Thus, the basic version of these kinds of transmission structures can be illustrated in a simple way. As shown in figure 2.1, there are different base structures: output coupled (OC), input coupled (IC) and compound coupled (CC). They have different physical characteristics, for instance, with regard to the power flow through the variator. By integrating the variator into the transmission, the base structures can be identified. If the variator is coupled to the input shaft, this is an IC system. Coupling to the output shaft is an OC system. If none of the two coupling types apply, it is a CC system.

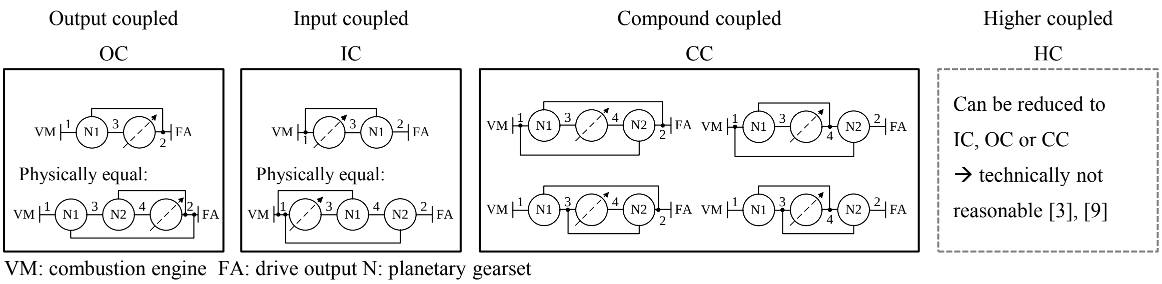
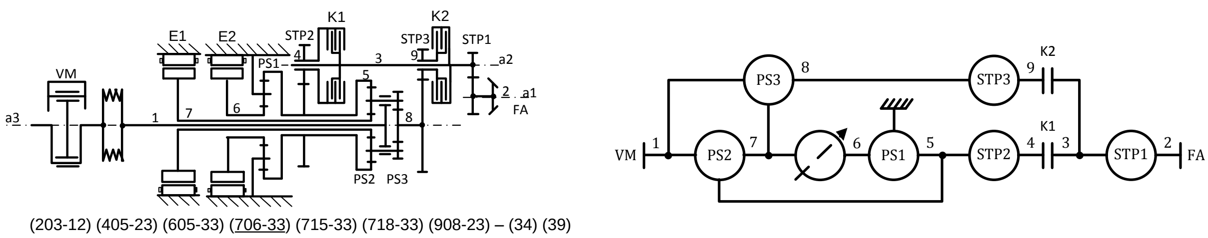


Figure 2.1 Basic structures of power-split transmissions

2.2 Calculating continuously variable transmissions with one input and one output [2]

Figure 2.2 shows an electrically power-split CVT with one input, one output and equalized power balance at the variator which will serve as example to illustrate the method used.



Code: (2 0 3 - 1 2) (4 0 5 - 2 3) (6 0 5 - 3 3) (7 0 6 - 3 3) (7 1 5 - 3 3) (7 1 8 - 3 3) (9 0 8 - 2 3) - (3 4) (3 9)

STP_1 STP_2 PS_1 Var_1 PS_2 PS_3 STP_3 SE_1 SE_2

Figure 2.2 Continuously variable transmission b with one input and one output as example

[2] Work result of Mr. Warth

Transmission structure description

Power-split continuously variable transmissions consist of a number N_{Var} of variators Var_l and often a number N_{STP} of spur gear pairs STP_k apart from the planetary gearsets. The equations from chapter 2.1 to calculate the speeds and torques of spur gear pairs can be derived from the planetary gearset equations in case of a stationary transmission. Thus, a spur gear pair can be described as a planetary gearset with housing-mounted carrier. To differentiate the planetary gearset and the spur gear pair, the rotational axes a also have to be considered. So the spur gear pair's number code consists of the first spur gear's A shaft number z_A, the second spur gear's B shaft number z_B as well as the rotational axis $a_{A/B}$ belonging to the corresponding spur gear.

$$STP_k \coloneqq \left(z_{A,k}\ 0\ z_{A,k} - a_{\mathrm{A,k}} a_{\mathrm{B,k}}\right) \text{ with k} \in [1;\ N_{STP}] \quad (3)$$

To make the description consistent, the planetary gearset description is also extended by the rotational axes. Due to the coaxial design of the planetary gearset, all elements are located on the sun's rotational axis.

$$PS_j \coloneqq \left(\mathrm{z}_{s,j}\ \mathrm{z}_{c,j}\ \mathrm{z}_{r,j} - a_{\mathrm{s,j}}\ a_{\mathrm{s,j}}\right) \text{ with j} \in [1;\ N_{PS}] \quad (4)$$

Planetary gearsets and spur gear stages have discrete ratios. Variators, on the other hand, enable a continuously variable speed and torque conversion corresponding to a continuously variable ratio. Depending upon the type of variator, the power is transmitted based on a mechanical, hydraulic or electric operating principle, refer to figure 1.1. All construction types have in common that they convert the speed and the torque between two shafts and according to an adjustable ratio. An exception are special mechanical variators in coaxial design which can also be used as differential with variable ratio und thus the power output is split up to three shafts [2]. The variator's shafts can be arranged coaxially or with axial offset. The variators can also be described with the number code which has already been introduced for spur gear pairs and planetary gearsets. It consists of numbering the three possible variator shafts $z_{A,l}$, $z_{C,l}$ and $z_{B,l}$ and the corresponding rotational axis $a_{A/B}$. To clearly differentiate the variator from the planetary gearset or spur gear pair number code, the variator's code is underlined.

$$Var_l \coloneqq \left(\underline{z_{A,l}\ \ z_{C,l}\ \ z_{B,l} - \ a_{A,l} a_{B,l}}\right) \text{ with l} \in [1;\ N_{var}] \quad (5)$$

The random sequence of all elements of a continuously variable power-split transmission according to the equation (3), (4) and (5) followed by the switch elements results in the extended transmission code when compared to [6], refer to equation (6). According to the described procedure, figure 2.2 shows the extended transmission code as example.

$$PS_j - STP_k - Var_l - SE_m \quad (6)$$

Calculation

The calculation of power-split continuously variable transmissions with equalized power balance is based on the determination of the individual transmission elements. Chapter 2.1 describes how to calculate planetary gearsets and spur gear stages. Depending on its design, a variator has different control variables to set the desired operating point. In general, adapting the control variable results in changing the speed and the torque between shaft A and shaft B of the variator Var_l. It is described by a variable ratio $i_{Var,l}$.

$$\omega_{A,l} - i_{Var,l} \cdot \omega_{B,l} = 0 \qquad\qquad T_{B,l} + i_{Var,l} \cdot T_{A,l} = 0 \tag{7}$$

For instance, pressure losses in the hydrostatic variators, result in torque losses that are calculated using the efficiency $\eta_{T,l}$.

$$T_{B,l} + i_{Var,l} \cdot \eta_{T,l}{}^{w_{Var,l}} \cdot T_{A,l} = 0 \quad \text{with} \quad \mathrm{w}_{Var,l} = sgn[-\omega_{A,l} \cdot T_{A,l}] \tag{8}$$

Opposed to fixed ratio transmissions in case of which power is transmitted with positive engagement and thus without slip, variators are subject to so-called slip losses in practice. These are considered in form of the efficiency $\eta_{\omega,l}$.

$$\omega_{A,l} - \frac{i_{Var,l}}{\eta_{\omega,l}{}^{w_{Var,l}}} \cdot \omega_{B,l} = 0 \tag{9}$$

Thus, multi-dimensional characteristic maps are required to calculate power-split continuously variable transmissions; these characteristic maps in turn depend on the characteristics of the variator used. In the early concept development phase in particular, the required efficiency maps are unknown; consequently, an efficient analysis cannot be conducted using equations (7) to (9). This is only possible if a real variator ratio $i_{var,l}{}^*$ can be assumed instead of a theoretical variator ratio $i_{var,l}$ and the efficiency $\eta_{\omega,l}$. Thus, equation (9) can be simplified and as a consequence corresponds to the calculation of a spur gear pair [1].

$$i_{Var,l}{}^* = \frac{i_{Var,l}}{\eta_{\omega,l}{}^{w_{Var,l}}} \qquad\qquad \omega_{A,l} - i_{Var,l}{}^* \cdot \omega_{B,l} = 0 \tag{10}$$

If the real variator ratio is inserted into equation (8) and if it is taken into account that the multiplication of $\eta_{\omega,l}$ and $\eta_{T,l}$ results in the total efficiency $\eta_{Var,l}$, the lossy torque calculation can also be attributed to the calculation of the torques at a spur gear pair.

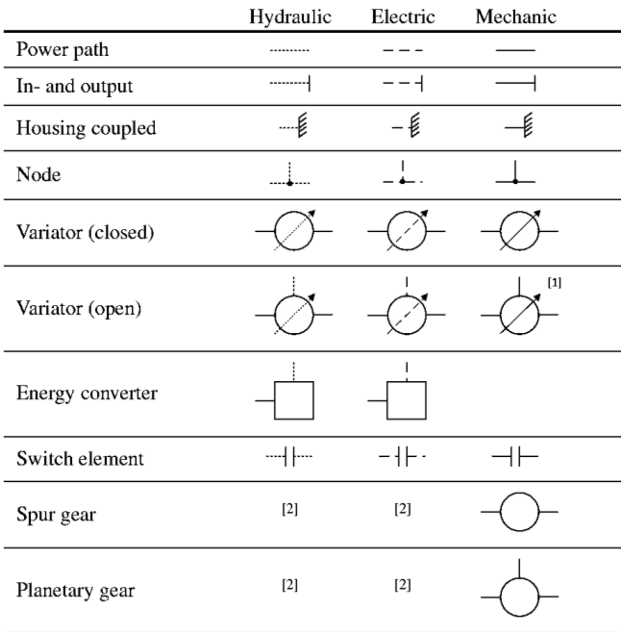
$$\eta_{Var,l} = \eta_{T,l} \cdot \eta_{\omega,l} \qquad\qquad T_{B,l} + i_{Var,l}{}^* \cdot \eta_{Var,l}{}^{w_{Var,l}} \cdot T_{A,l} = 0 \tag{11}$$

By defining a real variator ratio and a variator efficiency which only depends on the variator ratio, power-split continuously variable transmissions of any design can be calculated using matrixes in different detailing levels and by using equations (10) and (11) and the method described in chapter 2.1. The matrix size for kinematics calculations is based on the degrees of freedom of the individual elements in the transmission to be calculated. A simplification of the equations results in the extension of the consideration according to [6]. The number of shafts is consequently derived from the number of planetary gearsets, spur gear pairs, variators as well as closed switch elements.

$$N_w = 1 + N_{PS} + N_{STP} + N_{Var} + N_{SEG} \tag{12}$$

2.3 Calculating continuously variable transmissions with several inputs and outputs [3]

Before the calculation of transmission systems with several inputs and outputs is explained, figure 2.3 gives an overview of the symbols developed. All required symbols for mechanical-hydraulic or mechanical-electric transmission systems are shown, the corresponding calculations are generally valid. The symbols from the column "mechanic" are already known from [2], [3] and [9]. Calculation of mechanical elements as well as closed variators with equally power balance are described in chapters 2.1 and 2.2. In the following, open variators, the paths and their nodes, the electric or hydraulic inputs and outputs as well as the energy converters are described and calculated. In general, electric energy converters are electrical machines. The closed variator in this context is a special form since it consists of two energy converters and one path. The open variator, in addition, has a node connecting the paths. For this reason, variators are calculated as if they were a combination of the mentioned individual elements. In order not to change the existing symbols from the state of the art and to reduce the transmission scheme symbols to a minimum of components, the variator's illustration is maintained nevertheless. The two energy converters regulating the vehicle drive ratio are illustrated as variators in general. Initially, the mathematical description of the calculation of electric and hydraulic switch elements as well as housing connections is omitted since they are not used in the transmission examples shown. Due to the symbolic illustration of the transmission systems, the variator's operating



[1] Special three-shaft variators

[2] Technically feasible, but illustration is limited to mechanics

Figure 2.3 Developed transmission symbols

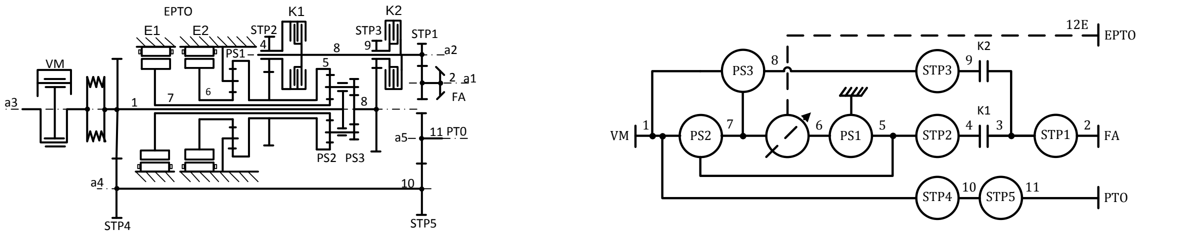
---

[3] Work result of Mr. Reick

principle and the CVT system's coupling type can be seen immediately. Furthermore, deducting the transmission code, drawing a transmission scheme and thus creating the transmission variants is decisively easier.

Transmission structure description

Unlike the well-known system from figure 2.3, the extended transmission system b, refer to figure 2.4, features a PTO (power take-off shaft) and an EPTO (electric interface) as well as an open variator. Thus, the transmission code must be extended for an automated calculation. Open variators have a third shaft or an additional power path. As a consequence, the power flow in the variator must be calculated.



Code:	(2 0 3 - 1 2)	(4 0 5 - 2 3)	(6 0 5 - 3 3)	(7 12E 6 - 3 3)	(7 1 5 - 3 3)	(7 1 8 - 3 3)	(9 0 8 - 2 3)	(1 0 10 - 3 4)	(10 0 11 - 4 5)	-	(3 4)	(3 9)	(1 VM)	(2 FA)	(11 PTO)	(12E EPTO)
	STP_1	STP_2	PS_1	Var_1	PS_2	PS_3	STP_3	STP_4	STP_5		SE_1	SE_2	S_1	S_2	S_3	S_4

Figure 2.4 Continuously variable transmission b with several inputs and outputs as example

The paths' starting number has the index $N_w + 1$, then all paths are numbered in an ascending order just like the mechanical shafts. In the transmission example b, the open variator has the number $z_{C,l} = 12E$ of the electric path in the center position. According to equation (5), the code is then (7 12E 6 - 33). The additional inputs and outputs (also interfaces) contain the number z and the interface's designation. Taking the PTO as an example, the code is: $(11\ PTO)$.

$$S_o := (z_o\ Bez) \text{ with } o \in [1;\ N_s] \quad \text{and} \quad Bez := \{VM, FA, PTO, EPTO\} \qquad (13)$$

The transmission system might feature further energy converters. This is the case, e.g., for the *ZF Terra*+ transmission system. The transmission consists of a hydraulic variator for the vehicle drive and a crankshaft generator to provide electric power for an EPTO [4]. The transmission code for further energy converters is, according to equation (14):

$$EW_n := (z_{A,n}\ z_{C,n}) \quad \text{with} \quad n \in [1;\ N_{EW}] \qquad (14)$$

Equation (15) describes the total code for a CVT system with energy converters and several inputs and outputs; see also the transmission code in figure 2.4.

$$(PS_j - STP_k - Var_l - SE_m - EW_n - S_o) \qquad (15)$$

Calculation

Continuously variable transmission structures with the requested complexity might have rotational degrees of freedom ≥ 1. The number of boundary conditions N_{RB}, which must be set for calculation purposes, are calculated by means of the number of shafts N_w, the number of ratio elements N_{UE} and the number of closed switch elements N_{SEG}.

$$N_{RB} = \mathrm{N}_w - N_{UE} - N_{SEG} \qquad N_{UE} = \mathrm{N}_{PS} - N_{STP} - N_{Var} \tag{16}$$

The speeds are mostly calculated analogous to equation (1). The number of boundary conditions corresponds to the number of rotational degrees of freedom in the system. Thus, the boundary conditions are provided in form of a vector $\vec{r}_\omega$. Since several energy converters, in- and outputs might be integrated in a transmission system, it first must be determined how many power and torque boundaries have to be set for the calculation. It is recommended to specify a torque for mechanical shafts and a power for hydraulic or electric paths. The number of boundaries N_{RB} is calculated using the amount of all calculated torques and powers N_{tp} and the number of constraints N_{ztp}. Apart from the elements already known, the number of open switch elements N_{SEO}, the energy converters N_{EW} and the paths N_L are integrated in the equations.

$$N_{RB} = \mathrm{N}_{tp} - N_{ztp} \tag{17}$$

$$N_{tp} = \mathrm{N}_S + N_{SE} + 3 \cdot N_{PS} + 2 \cdot (N_{ST} + N_{EW}) \tag{18}$$

$$N_{ztp} = \mathrm{N}_{ST} + 2 \cdot N_{PS} + N_{SEO} + N_{EW} + N_L + N_w \tag{19}$$

The torques and power in the paths are also calculated using a linear equation system in the form already presented with a coefficient matrix $\underline{A}_{TP}$ and a solution vector $\vec{b}_{TP}$. It also contains the boundary conditions $\vec{r}_{TP}$ to calculate the torques and power in $\overrightarrow{tp}$. The solution vector might also contain further transmission losses for a more detailed consideration. $\underline{A}_{TP}$ contains the known sub-matrixes $\underline{A}_{T,RB}$, $\underline{A}_{T,UE}$, $\underline{A}_{T,SEO}$ and $\underline{A}_{T,w}$. Furthermore, the equations of the energy converter $\underline{A}_{TP,EW}$ and the paths $\underline{A}_{TP,L}$ are mapped.

$$\underline{A}_{TP} \cdot \overrightarrow{tp} = \vec{b}_T \tag{20}$$

Variators, however, are no longer calculated in $\underline{A}_{T,UE}$, see also chapter 2.2, but calculated by using sub-matrixes $\underline{A}_{TP,EW}$ and $\underline{A}_{TP,L}$. The sub-matrix $\underline{A}_{TP,EW}$ contains the generally valid power equations for the energy converters. The mechanical power is converted into electric or hydraulic power of the path attached to the energy converter.

Ideally: $$\omega_{A,l} \cdot T_{A,l} - P_{C,l} = 0 \tag{21}$$

With loss: $$\omega_{A,l} \cdot \eta_{Var,l}{}^{w_{Var,l}} \cdot T_{A,l} - P_{C,l} = 0 \quad \text{with} \quad w_{Var,l} = \mathrm{sgn}\left(-\omega_{A,l} \cdot T_{A,l}\right) \tag{22}$$

The total power in the paths must be balanced and is determined in $\underline{A}_L$ analogous to the torques of the shaft equations in $\underline{A}_W$.

$$\sum T_{PS,z} + \sum T_{STP,z} + \sum T_{EW,z} + \sum T_{S,z} = 0 \tag{23}$$

$$\sum P_{EW,z} + \sum P_{S,z} = 0 \tag{24}$$

In case of a transmission system with only one closed variator, the following applies:

$$P_{C,1} = P_{C,2}. \tag{25}$$

3. Application of Calculation Method

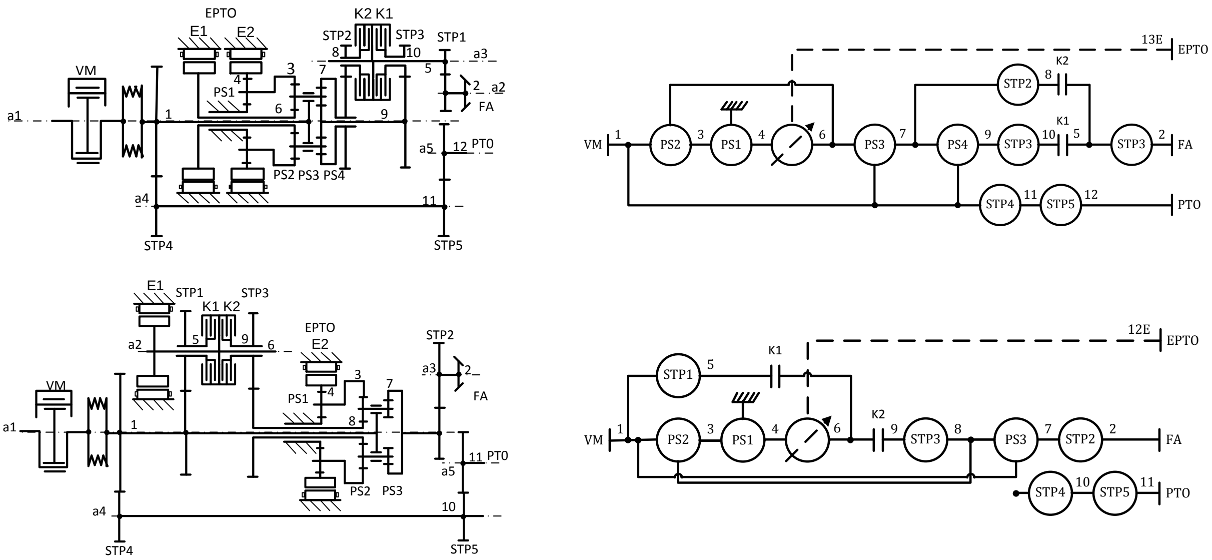


Figure 2.5 Continuously variable transmissions ‘Example a’ and ‘Example c’

Speeds and torques of all transmission elements can be calculated using the presented method. Furthermore, the system efficiency can be analyzed in different degrees of detail. Depending on the available data basis, the variator efficiency might be considered either one-dimensionally as constant, two-dimensionally depending on the variator ratio or as multi-dimensional characteristic map which depends on both the speed and the torque.

Based on the sample systems from figures 2.4 and 2.5, an exemplary analysis is made and the method is verified by comparing it to simulation results. The system has 2 speed ranges, the first of which is synchronized by closing clutch K1 and the second by closing clutch K2. Reversing is possible either directly using an electric variator or by an in- or output sided reversing unit. Considering the Wolf symbols from the sample transmissions, it is clear that all systems have a CC structure in the 2nd speed range whereas they differ in the 1st speed range. Transmission ‘Example a’ has also a CC structure whereas transmission ‘Example b’ has an OC structure and transmission ‘Example c’ an IC structure when closing K1.

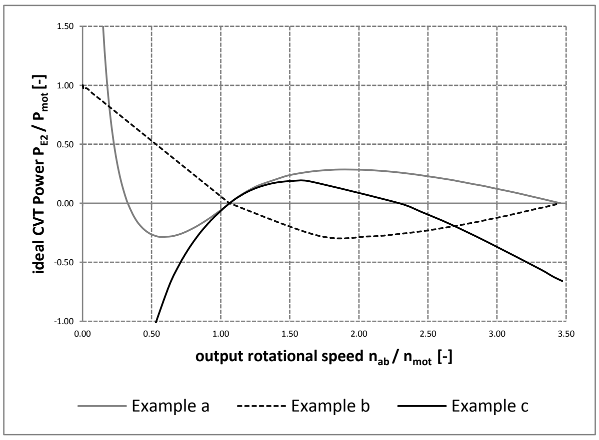


Figure 2.6 Related ideal variator power of the sample systems

Figure 2.6 shows the variator power of electric motor E2 without losses relating to the drive power of the combustion engine. It is clearly visible that the power quotient curve depends on the speed range's coupling type. Furthermore, the coupling type also influences the power flow in the second speed range which is set differently due to the transmission kinematics despite the identical shift point. When comparing the efficiency of the three systems with a constant variator efficiency, the curves shown in figure 2.7 a) are formed.

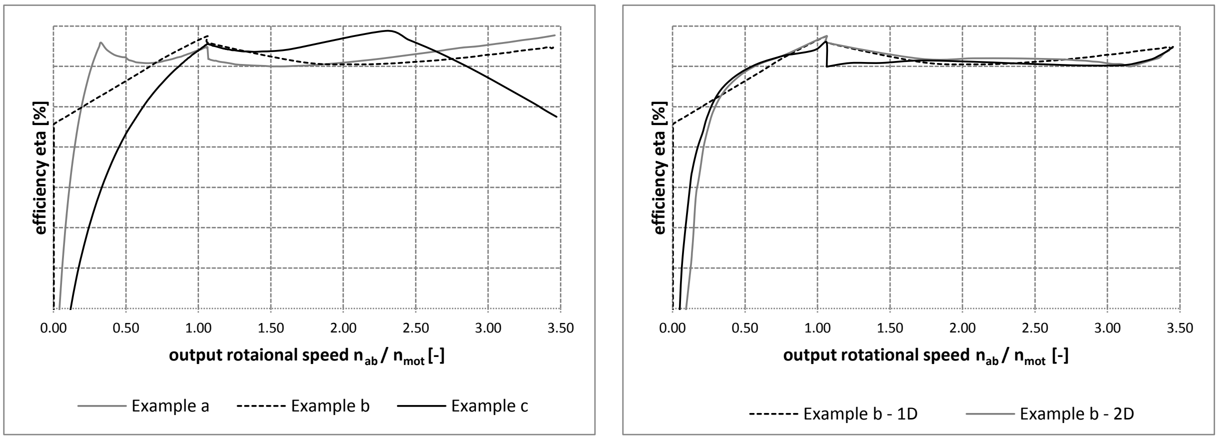


Figure 2.7 a) Efficiency comparison of sample systems
b) Levels of detail of the variator efficiency

A correlation to the variator power share can also be seen here. The transmission efficiency decreases in general when a high proportion of power flows through the variator. Three levels of details for the calculation are shown in figure 2.7 b) for the transmission 'Example b'. In case of the simplest level of detail, a constant variator efficiency is assumed. This simple calculation is compared to an averaged 2D characteristic map and a calculation with a specific speed- and torque-dependent 3D characteristic curve. The 2D calculation combines a good result quality with an acceptable calculation time. The 3D calculation shows variator-specific efficiency curves and thus offers the option for further investigation, such as an optimization of the electric motor's operating points.

4. Literature

[1] Brems, W.: Beitrag zur Berechnung leistungsverzweigter Getriebe unter Berücksichtigung mechanischer, hydraulischer und elektrischer Stellgetriebe. Masterarbeit. Karlsruher Institut für Technologie, Lehrstuhl für Mobile Arbeitsmaschinen, 2014.

[2] Förster, H. J.: Stufenlose Fahrzeuggetriebe in mechanischer, hydrosta-tischer, hydrodynamischer, elektrischer Bauart und in Leistungsverzweigung. Verlag TÜV Rheinland, Köln, 1986.

[3] Gitt, C.: Analyse und Synthese leistungsverzweigter Stufenlosgetriebe. Dissertation. Universität Stuttgart, 2004.

[4] Götz, M., Grad, K. und Weinmann, O.: Elektrifizierung von Landmaschinen. ATZ Off-Highway, 2, 2012, 10–20.

[5] Grad, K., Bailly, G. und Haas, M.: Stufenloses Getriebekonzept für Klein- und Schmalspurtraktoren. ATZ Off-Highway, 3, 2013, 28–37.

[6] Gumpoltsberger, G.: Systematische Synthese und Bewertung von mehrgängigen Planetengetrieben. Dissertation. Technische Universität Chemnitz, 2007.

[7] Leesch, M.: Beitrag zur systematischen Synthese und Bewertung von Doppelkupplungsgetrieben. Dissertation. Technische Universität Chemnitz, 2012.

[8] Reick, B., Volpert, B. und Ikonen, T.: Regelstrategien stufenlos leistungsverzweigter Getriebe. 5. Fachtagung Hybride und energieeffiziente Antriebe für mobile Arbeitsmaschinen, KIT Scientific Publishing, 30, Karlsruhe, 2015, 19-33.

[9] Schröder, H.: Konzeptionierung leistungsverzweigter Getriebestrukturen für Hybridfahrzeuge. Dissertation. TU Braunschweig, Logos-Verlag, 2009.

[10] VDI Richtlinie 2157: Planetengetriebe – Begriffe, Symbole, Berechnungsgrundlagen. Beuth-Verlag Berlin, 2010.

[11] Wolf, A.: Die Grundgesetze der Umlaufgetriebe. Vieweg Verlag, 1958.