CONTINUOUSLY VARIABLE TRACTOR TRANSMISSIONS

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Abstract. *The paper starts with a longer analysis of the colorful CVT history including typical early problems. Market demands for tractor transmission systems with CVTs address gross traction, speeds, efficiencies, typical control strategies, expected load spectra and others. Physical CVT principles are described in a survey, focusing on tractor drives but considering also the automobile sector. Hydrodynamic torque converters have no chances for tractors, while mechanical, hydrostatic and electric CVTs have. A single unit is however not sufficient, a system is needed. A combination with power shifted ranges can realize the required wide band of tractor speeds. Hydrostatic and electric CVTs (driven by Diesel engines) are in most applications however not able to meet the efficiency targets in a direct mode. The fuel cell could improve this situation for electric drives, but is not yet economically available. Mechanical chain variators offer outstanding efficiencies, but need at least two ranges with a power split for active zero speed control. The power split technology thus has the potential for adequate CVT transmission systems. Performances of basic systems are analyzed. Commercially available tractor CVTs with hydrostatic-mechanical power split are described as examples.*

Keywords. CVT, Efficiency, Electric CVT, Hydrostatic CVT, HST, IVT, Load Spectra, Mechanical CVT, Planetary, Power split, Power train, Toroid drive, Tractor, Tractor transmission, Transmission.

INTRODUCTION AND HISTORY

Agricultural mechanization is going to continue with its importance for the world`s nutrition changing to high sophisticated mechatronic systems in the high developed countries as addressed by Schueller and Stout (1995). The tractor remains still the most important machine and its transmission system is a key component representing about 25-30 % of the total tractor first cost. Stepped transmissions have been used since the beginning but infinitely variable drives with automatic controls have been introduced in Europe 1996 for standard tractors opening a new era of power train design principles.

The tractor transmission system

The complete tractor transmission (also called transaxle) is usually defined as demonstrated by **Fig. 1**: A combination of the vehicle speed change gearbox, the rear axle with brakes, the power take off (PTO) and - if required - arrangements for the front axle drive and for the drive of auxiliary units
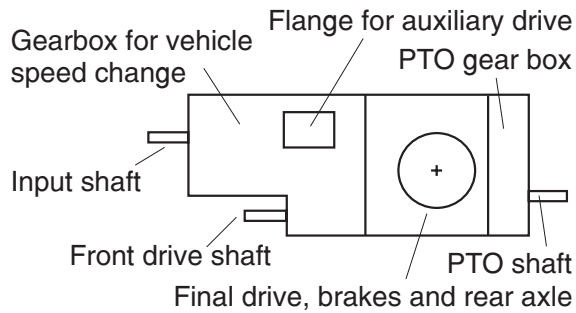
Fig. 1 Tractor transmission system, side view (Renius, 1999).

(mainly hydraulic pumps). The functions can also be addressed by a power flow diagram as used by E. Paul Browning in his well known and still very useful ASAE Lecture No. 4 (Browning 1978).

"CVT" and other used abbreviations

CVT is an abbreviation for "Continuously Variable Transmission". It has initially often been used to address continuously variable variators for car transmissions with steel belt friction drives.

IVT is an abbreviation for "Infinitely Variable Transmission". It is used like the proposed general meaning of "CVT", expresses however little better the infinite ratio of nominal speeds down to an "active" zero speed.

HST is an abbreviation for a (direct) Hydrostatic Transmission (without power split), often used in Japanese publications.

HMT stands for "Hydro-Mechanical Transmission". It addresses the combination of hydrostatic and mechanical power transfer, in particular for power split concepts.

History I: The early phase

CVTs are the dream of every progressive vehicle transmission engineer since a long time. About 100 years ago, battery-driven electrical drives allowed already an infinitely variable speed control and low noise levels. The main problem was the poor capacity for stored energy. This problem could be solved by the introduction of internal combustion engines, which how-

ever created the demand of a shiftable gear box.

Some other very early and at the same time high sophisticated CVTs should be mentioned. **Fig. 2** shows the famous idea of Renault (Renault 1907) to use the principle of internal hydrostatic-mechanical power split for a passenger car gearbox. The input shaft (left) from the IC engine is driving the cylinder block of the left axial piston pump, while the right cylinder block is rigidly fixed with the chassis. The power thus is transferred by both, the hydrostatic path and the housing. The higher the vehicle speed, the lower the hydrostatic portion and the higher the efficiency. The pump is controlled manually while the hydrostatic motor is automatically controlled by the system pressure. Realization of this outstanding concept failed because of problems with the required tolerances, materials, skills and production costs (Molly 1966). Early developments of hydrostatic drives for vehicles in the US have been addressed by Mortenson (1960).

At the same time agricultural engineers invented a friction drive CVT for a self propelled plough of the German company Stock, **Fig. 3**. The vehicle speed is controlled by the radii of the friction contacts. This vehicle had two main traction wheels carrying almost the whole vehicle weight and vertical ploughing force. The CVT was in this case also used to offer a power steering with a simple steering mechanism, which enabled different radii for both friction contacts. The complete machine is on display at the "Deutsches Museum" (German Museum of Technology), Munich. This invention has not been realized, probably because of the poor power density and missing durability of the friction drive.

Modern hydrostatic vehicle drives with and
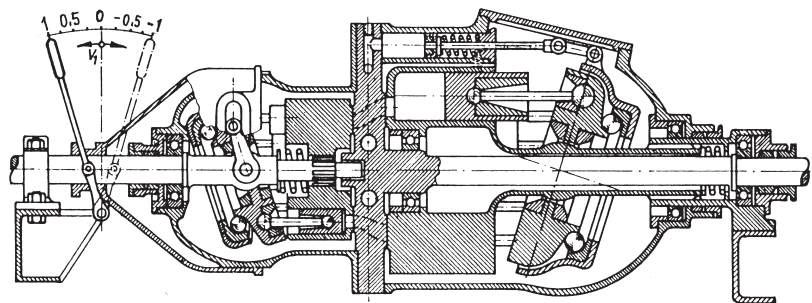
Fig. 2 Passenger car CVT with internal hydrostatic power split (Renault 1907).
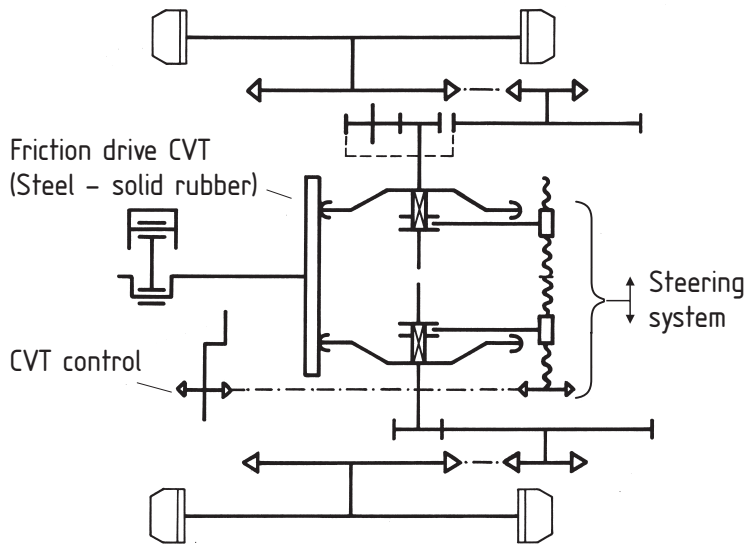
Fig. 3 Early design of a friction type CVT prototype of a Stock motorplow 1907 (Renius 1974a).

without power split mostly use so called "axial piston units" for both, pump and motor, **Fig. 4**. The

Fig. 4 Bent axis axial piston unit (left) and swash plate axial piston unit (right).

first concept, which gained importance, was the so called "Thoma" axial piston unit. Hans Thoma (1887-1973) is called to be the father of the today so called "bent axis axial piston unit". This promising development was first produced in series by Vickers, USA, for aircraft hydraulics according to a license contract 1935-1941 (Schunder 1995).
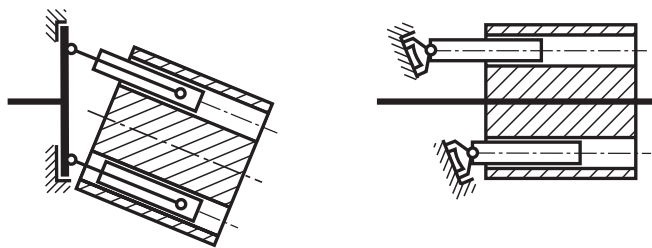
Pittler, Demag and other companies started in Germany with license productions from 1937. Around 1940, series production was mainly used for military applications such as aircrafts, tanks and warships with peak pressures up to 250 bar (about 3600 psi). After World War II the concept was improved and introduced in series production for civil use by several companies. Improvements came also from ideas of H. Molly at this time (Schunder 1995).

History II: The intermediate phase

The so called "swash plate axial piston unit" was first developed in the US since 1942 by Denison and could also gain commercial importance in the first stage mainly in the US.

The two mentioned axial piston units came in competition in the US from about 1950. In Europe this happened much later from about 1970, by Sundstrand, Mannesmann Rexroth, Linde and others (Schunder 1995). The friction between piston and cylinder allowed in the first stage tilting angles only around 18 degrees (bent axis unit much higher) and limited applications for motor use in case of low rpm. Therefore early research had been carried out investigating the friction mechanics and possible improvements (Renius 1974b).

Although the bent axis principle has little higher efficiencies and better start up behavior in motor applications, the strong competition between both was decided more and more in favor of the swash plate unit due to the lower specific production costs, the "drive through shaft", the compact design and other advantages.

The first broad research on hydrostatic CVTs for tractors had been carried out in the former National Institute of Agricultural Engineering (NIAE) Silsoe, UK, developing a hydrostatic driven prototype (Hamblin 1952), which was presented 1954, **Fig. 5**. The variable displacement axial piston pump has been driven by the IC engine working di-
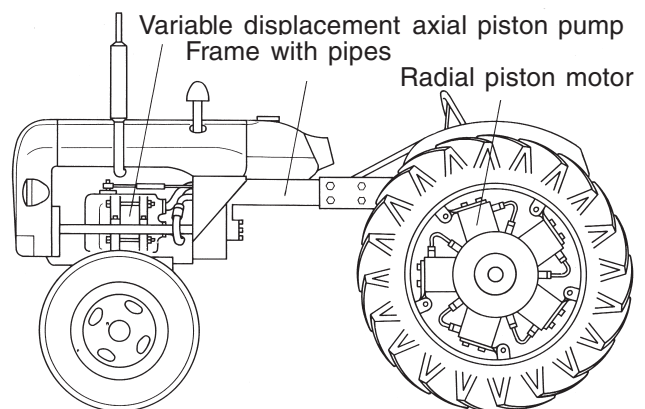
Fig. 5 Silsoe research tractor with hydrostatic CVT, presented 1954 by the National Institute of Agricultural Engineering (NIAE), Silsoe, UK.

rectly by tubes on large radial piston motors being integrated with the driving wheels (NN 1954). Several tractor companies of the US have also developed prototypes at this time or little later (Meile 1961). One of the first commercially available vehicles with a hydrostatic drive was the Linde "hydrocar" (Hannover fair 1955), which became the basis of the today well known Linde forklifters with hydrostatic Linde transmissions.

Fig. 6   First in series produced hydrostatic drive system for tractors, International Harvester , USA 1967 (Renius 1969).

A first commercial introduction of a hydrostatic CVT for tractors was tried by the German tractor company "Eicher", launching the new "HR" tractor with the hydrostatic Dowty "Taurodyne" unit in 1965. Two bent axis units with variable displacement had been linked together using the connect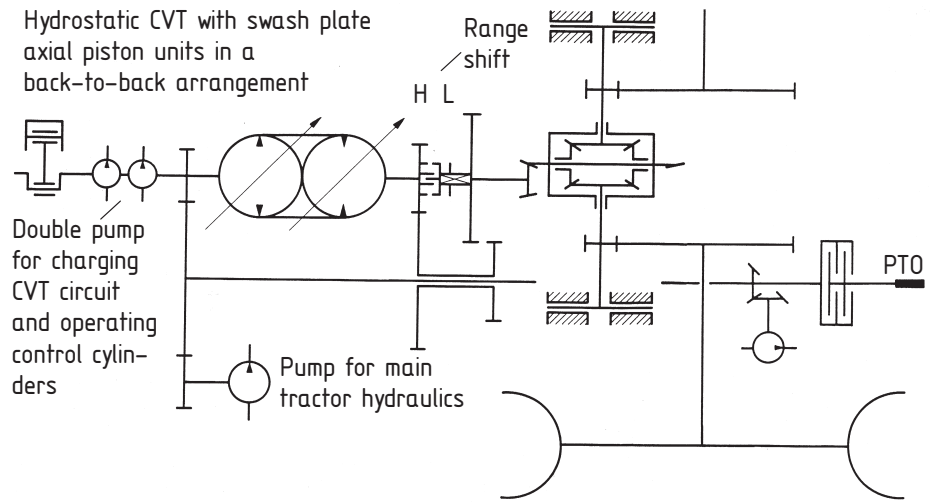ing oil pipes as four-bar-links, enabling automatically a suitable combination of tilting angles with low pressure losses (Worn and Walker 1966). The transmission system was working without ranges. Eicher could produce and sell only a small number because of several reasons. At the same time the German company Kopat tried to introduce a hydrostatic CVT with internal power split, system "Allgaier", for tractors and construction machinery – finally also without success.

More important became the decision of International Harvester, to start 1967 series production of agricultural tractors with the 656 (engine power about 60 kW) using two swash plate units "back-to-back". Both were variable in displacement and used pressure supported control (Morris 1967, Asmus and Borghoff 1968). The development had been carried out in a close co-operation

with Sundstrand, USA. The transmission system works with two ranges, **Fig. 6**. This leads to a better overall efficiency for small and high tractor speeds. The system had been applied for several tractor types of IHC but the production was finally terminated after "some 10.000 units produced" (personal estimation). The design was in the opinion of many experts well done, **Fig. 7**. But the used hydrostatic swash plate units created high losses and high first costs in comparison with the relatively simple conventional stepped transmissions of the 1960s. **Fig. 8** gives an impression of the full load efficiency of the complete tractor transmission system according to
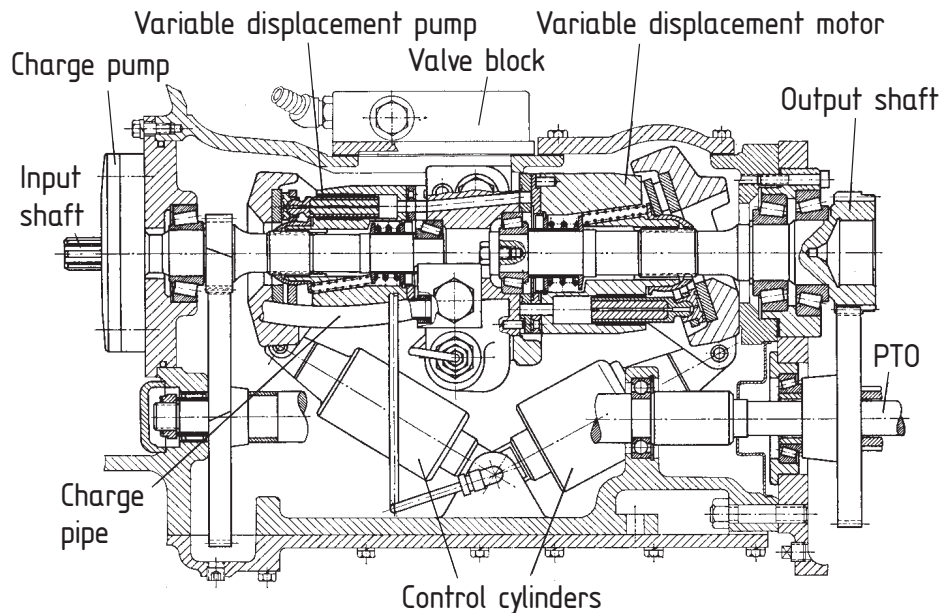
Fig. 7   Design of the hydrostatic section of the IHC transmission system plotted in Fig. 6. Swash plate units for pump and motor (Morris 1967).
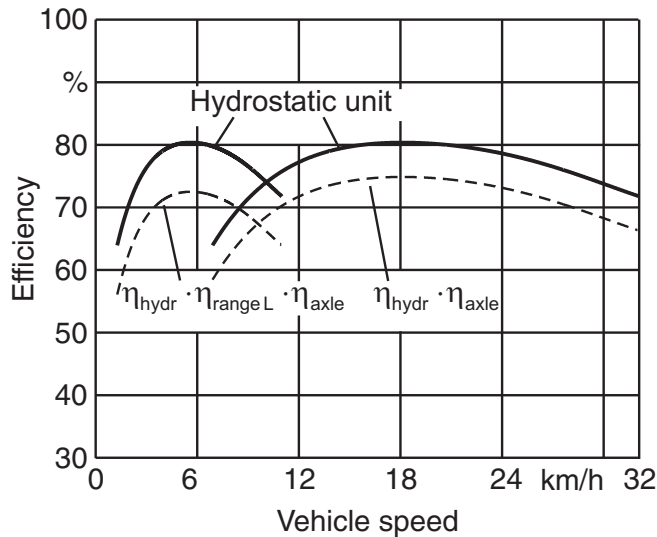
Fig. 8 Full load efficiencies estimated for the transmission system of Fig. 6 (Renius 2003).

data from Sundstrand (Martin 1969) and own calculations. The best efficiency of the hydrostatic unit is 80.5 %, but the best total system efficiency is only 73 and 75 % because of the additional gear wheels including the final drive. These figures are still typical for similar structures.

Best values for conventional transmissions are however above 90 % for the gearbox and about 85-87 % for the total transmission system (Reiter 1990). It was well known at that time, that the energy losses could be reduced by the power split principle (Jarchow 1964, Molly 1966, Kress 1968 and others). One important approach of this kind was the Cummins Sundstrand "Responder" (Ross 1972) with "dual mode": The combination of a first range with direct hydrostatic drive for low speeds with a second range working with power split. The project arrived however only at a pilot production (Wadman 1973). An other interesting power split system was proposed 1973 by the Orshansky Transmission Corp., USA (Orshansky 1972). This system used a four-shaft planetary and offered, for example, 4 power split ranges with shifts at

synchronous speeds. The promising concept was not commercially applied to tractors. At the same time Deutz tried the introduction of a hydrostatic drive for its new tractor concept "INTRAC" (Breuer 1974), but the project was also terminated after a low number of units produced. Other companies such as Versatile worked on intelligent power split systems for tractors, finally also without success. A flood of patents were crated regarding hydrostatic power split systems, but no system could achieve a series production for tractors, nor for other vehicles. One reason may be, that the competing conventional transmissions had been relatively cheap and efficient at that time. Secondly comfort and automation was not yet a strong demand in the 1970s and electronics had not been available.

Hydrostatic drives could however gain importance for very small tractors (Witt 1965, Moyer 1966), becoming popular mainly in Japan and USA and offering a high comfort level. The costs could be balanced by "simplified" components, improved production methods and high production volumes. Energy losses and fuel economy had been considered to be less important in this market.

History III: The modern phase

The 1988 presented "Munich Research Tractor" (Renius 1988, Kirste 1989) marked the beginning of a series of new developments. Its transmission contains a mechanical pull type chain CVT, **Fig. 9**,
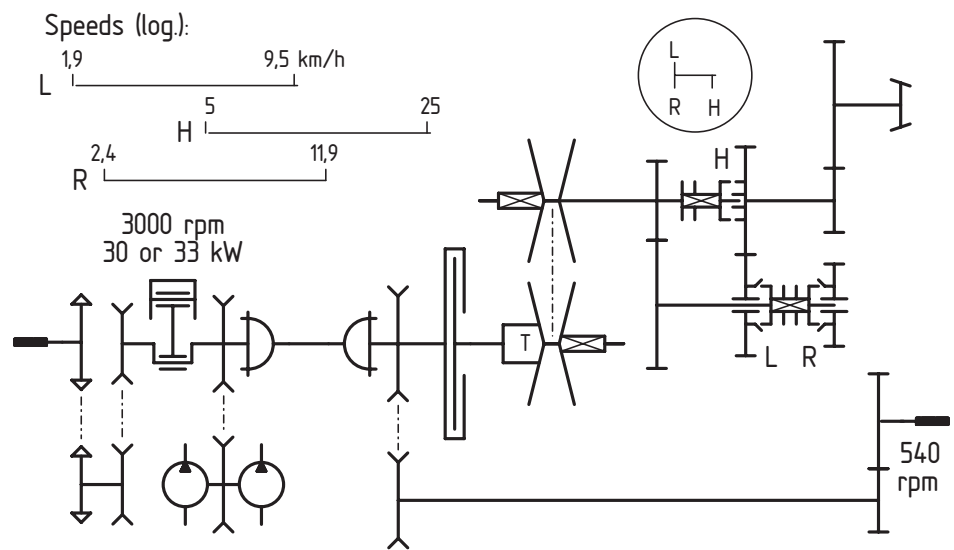
Fig. 9 Continuously variable transmission of the Munich Research Tractor 1988, CVT from P.I.V. (Kirste 1989). T torque sensor.

combined with two forward ranges and one reverse range. Hydraulically controlled clamping forces using the signal of a torque sensor enabled full load efficiencies of the variator unit up to about 90 % including hydrostatic losses. The tractor, which had been developed with broad support of the German industry, is now on display at the German Museum of Agriculture at University of Hohenheim. A prototype for a 60 kW tractor developed by Schlüter together with Hurth and P.I.V.-Reimers around 1990 was promising but did not reach series production due to the decline of Schlüter. Fendt was also very interested but the limited torque capacities of chain variators did not allow their application for large tractors. Fendt and CLAAS started therefore the development of hydrostatic power-split concepts in the late 1980s. The CLAAS transmission with a four-shaft planetary was based on Jarchow patents and could achieve outstanding high efficiencies, but was finally produced only in a limited number of units. At that time some power split transmissions have been developed also for other mobile machinery (Mitsuya et. al 1994, Schueller and Khan 1995). The Fendt "Vario" CVT was presented for tractors at Agritechnica 1995 and produced for a first large tractor model (926 Vario) from 1996 (Dziuba and Honzek 1997). The outstanding success motivated Fendt to apply the principle to more and more models in the following years. Meanwhile ZF and Steyr followed with their own concepts also based on hydrostatic units in power-split arrangements. John Deere used in a first step the ZF-concept, but developed later on its own hydrostatic power split transmission for larger tractors. The systems of Fendt, Steyr, ZF and John Deere will be presented in a later chapter.

Global demands: technology levels

Not every tractor on the world market needs a CVT. Only a small section of the total volume is and will be equipped in the future with such a "high tech" transmission. This shall be demonstrated by the principle of technology levels for tractors. While

Table 1 Basic transmission specifications of standard tractors by technology levels – word-wide view (Renius 2000, 2002).

Level	Nomin. speeds, km/h		No. of speeds	Shift	PTO
	forward	reverse	forw./rev.		speeds rpm
I	2–20(25)	3–8	6/2 to 8/2	SG, CS	540
II	2–30	3–10	8/4 to 12/4	CS, SS	540/(1000)
III	(0.5)2–30(40)	3–15	12/4 to 16/8	SS, HL	540/1000
IV	(0.3)2–40(50)	2–20	16/12 to 36/36 (or more)	SS, PPS, FPS	540/1000 (750/1250)
V	0–50(60)	0–25	∞	Automatic	

SG Sliding gear, CS Collar shift, SS Synchro shift, HL HiLo power shift, PPS Partial power shift (3 or more speeds), FPS Full power shift, ( ) options

the "Fordson" tractor could achieve in the 1920s a world-wide market share above 50 % with only one model, this situation has completely changed because of the wide-spread agricultural mechanization demands. The typical tractor in India, China or South America has, for example, not many common components with the typical tractor in Europe, USA or Japan. Today`s global demands thus require a complicated "multiple model strategy" regarding both, tractor functions and tractor power. Five technology levels have been defined for global product planning of tractors (Renius 2000, 2002) by typical component specifications. The transmission being the most complicated component is classified and specified by **Table 1**. Transmissions using a CVT unit can be regarded to represent the highest technology level V addressing markets with high productivity and comfort demands.

MARKET DEMANDS

Tractor families

Table 2 shows a typical structure of a world-wide standard tractor program (model). It contains at least 4 families according to 4 basic engine concepts. Every family consists of some models with a large number of common parts. This reduces the total number of parts and thus the production and logistics costs (Welschof 1974, Jenkins 1997, Renius 2002).

CVT transmission families should follow the tractor families. But family 1 is very sensitive in first costs and thus not yet equipped with CVTs.
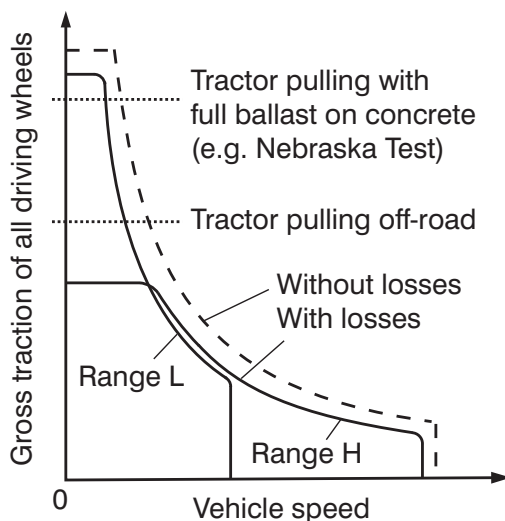
Table 2 Production program for standard tractors with (at least) 4 tractor families. Rated engine power for rated speed and ECE R24. Maximum engine power may be higher.

Tractor family	1	2	3	4
Rated eng. power, kW	30 – 55	60 – 90	90 – 125	135 – 210
Rated engine speed, 1/min	2100 ... 2500			
Diesel engine	3 cylinder Displ. ~ 3l	4 cylinder Displ. ~ 4l	6 cylinder Displ. ~ 6l	6 cylinder Displ. > 7l
	Turbo charger & intercooler very common			
Volume of functions	moderate	very large (options!)		large
Comfort level	moderate	high	very high	

The following market demands are therefore listed for typical standard tractors above about 50 kW rated engine power.

Basic drive transmission requirements, tractor speeds

Basic drive diagram: A typical gross traction versus speed diagram is shown by **Fig. 10** for a tractor with CVT and two mechanical ranges. Maximum gross traction on concrete is about equal to the weight of the ballasted tractor resulting in the maximum axle output torques. In case of a hydrostatic CVT, pressure is usually limited according to a

level little above. This very high load must be considered for life calculations with a limited time portion (see load spectra).

Speed change: Infinitely variable; if possible, under load for the full speed range. Short acceleration interruptions (range shift) accepted above 8-10 km/h. Ability for automatic control is a "must".

Speed range forward: From zero to a maximum of 40, 50 or 60 km/h (future versions in Europe).

Transmission efficiency forward: Total drive line efficiency (engine to wheels) for full load very near to those of stepped full power shift transmissions, see the target in **Fig. 11**.

Overdrive: Maximum vehicle speed should be possible at reduced engine speed to save fuel and reduce noise.

Speed range reverse: From zero to about 25 km/h.

Shift forward-reverse: Must be possible under load (power shift).

Active zero speed control: Must be possible under all practical load conditions.

Mechanical front wheel drive: An output shaft must be available, integrated design with ability for automatic control.

Fig. 10 Gross traction versus vehicle speed for a CVT with two ranges L and H.
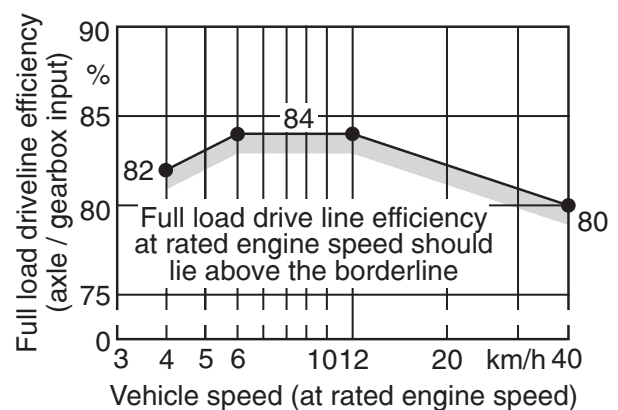
Fig. 11 Efficiency target for CVTs of larger tractors above 100 kW (Renius 1993, 1994, 1999).

Basic PTO requirements

Rear power take off for implements (PTO): Specification according to relevant standards, but up to four shaft speeds. If possible, CVT also for PTO.

Other power take off functions: The PTO "live shaft" should be able to drive hydraulic pump(s) and other auxiliary components.

PTO shift: Should be able for automatic control.

PTO efficiency (engine to shaft under full load, tractor not moving): About 95 % (auxiliary drives such as oil pumps and idling elements of the drive transmission excluded).

Closed loop control and drive line management

Closed loop controls are used for tractor-implement systems because of

– increased performance and thus improved productivity,
– better working accuracy and thus higher working quality
– reduced energy input
– improved driver comfort, health and safety
– contribution to environment protection.

These advantages are closely connected with the requirements of precision farming methods. Closed loop controls are working considerably better with infinitely variable actuators than with graduated actuators. Typical control strategies for tractor transmissions are listed by **Table 3**. Electronic engine control simplifies a complete drive line management (Scarlet 1993, Harms and Jaufmann 1996). It may be mentioned, that fuel saving strategies for tractors with CVTs have been already proposed in 1964 (Howson 1964).

Electronic control strategies and related design directives must consider the new standard ISO 11 783 (Goering et al 2003).

Comfort, safety, service

Comfort and safety

Applications of electronics for semi-automatic or fully automatic control strategies require skillful solutions for the man-machine-interface. Settings

Table 3 Manual and automatic control strategies for practical operations of tractors with CVT.

- Manual transmission control
- Closed loop speed control for PTO
- Combined open loop control of engine and transmission (transports)
- Closed loop power control for heavy operations (soil cultivation, combined implements, transports)
- Closed loop cruise control with slip compensation (distribution of chemicals and seeds)
- Fuel economy oriented combined open loop control of engine and transmission (all partial load operations)
- Closed loop control of the total working process of tractor and implements

for system configuration should widely be self explaining for the driver. This is a very important point as most drivers of CVT tractors quote the comfort to be the most important advantage. In case of a break down of the automated control system emerging systems must be available (Martinus and Freimann 2002). The typical stepwise development of safe mechatronic systems can be demonstrated by the so-called V-model, **Fig. 12**. Initially it had been a development standard for government IT systems (N. N. 2003) but is now going to become a general scheme for the development of mechatronic systems such as tractor and tractor-implement systems, including embedded software.
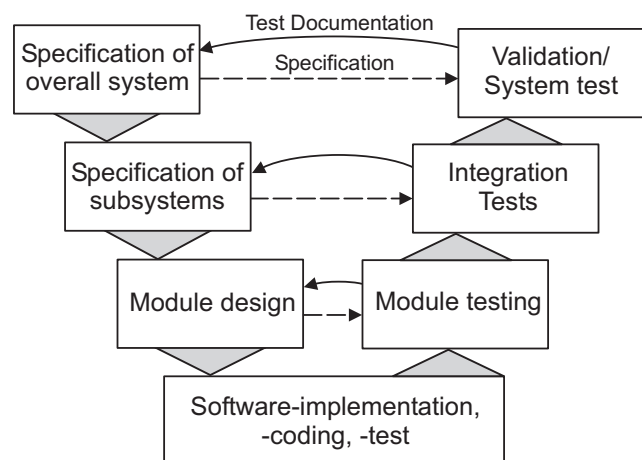
Fig. 12 V-model for the development process of mechatronic systems.

The process starts with the definition of the basic product specifications as outlined before, specifying on this basis the subsystems. Actual implementations of the functional structures take place in the lower part of the V-model while the right hand leg indicates all verification activities beginning with the modules and ending with the complete system. Tools like FMEA (Failure-Mode and -Effects Analysis) or model-based developments of electronic systems can be adopted from the automobile sector for tractor mechatronics.

Maintenance and repair

Requirements of service and repair should be as low as possible (e.g. long oil change intervals or replacement of filters only "on demand"). All transmission elements (including clutches and brakes) should be dimensioned for life. For diagnostic purposes an electronic interface of the transmission system must be on hand. Thereby networked architectures enable central access to system data using standardized protocols.

Transmission life

Tractor transmissions have to be designed and tested for the total tractor life. Nominal tractor life should always be defined together with a statistical probability. The authors prefer the B10 definition: 10 % can fail, 90 % last longer. Tractor B10 life should be about 8.000 to 12.000 hours (real time), increasing with the tractor power.

Load and speed spectra

Transmission speed collectives can be picked up from the local requirements. A generalized estimation for Europe can be found in the CIGR Handbook III, page 142 (Renius 1999).

The torque load spectra for the input shaft of CVTs are stronger than those for conventional stepped transmissions due to the popular control strategy "power control for heavy operations", which increases the use of the installed power considerably. This is demonstrated by **Fig. 13**.

The torque is counted in time portions and related to the rated engine torque at rated engine speed (not maximum torque!) in order to generalize the diagram. It shows as a basis the "Renius" load spectrum, which has been applied in Europe for many conventional transmissions as a first approach. This curve could be modified (Vahlensieck 1999) and was used for CVT developments.

In addition, load spectra for the transmission output are necessary. **Fig. 14** shows a standard load spectrum, which was published for 2WD (Renius 1976) and later confirmed for 4WD and typical European conditions (e.g. Rempfer 2003). These statistical fundamentals have been considered as a guideline for calculation and tests of several stepped transmissions and CVT developments in Europe. High torque reserves above about 25 % need little stronger load spectra for the transmission input. The experience shows, that the transmission input load spectra are mainly determined by the engine characteristics and the transmission control
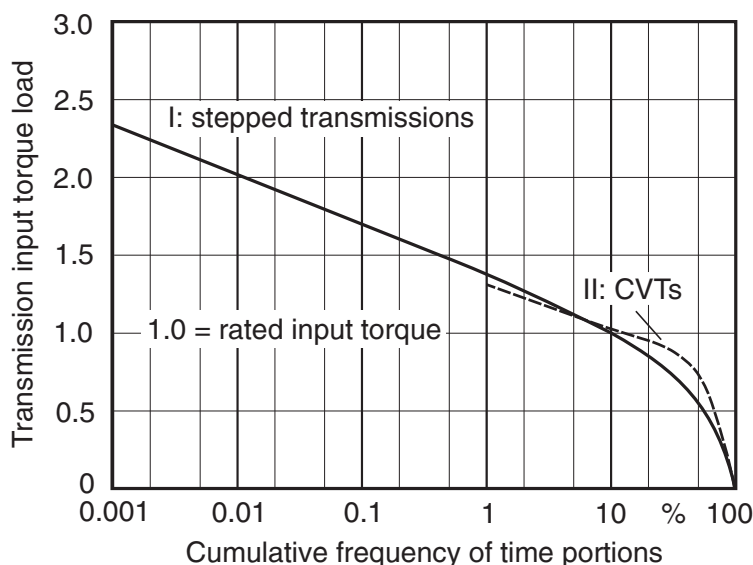
Fig. 13 Total torque load spectra for tractor transmission input shafts. Probability about 90 %.
Curve I Standard load spectrum "Renius" for the input shaft of stepped transmissions with dry master clutch, moderate torque reserve and no automatic power control (Renius 1976).
Curve II Typical deviation for CVT transmissions with closed loop power control and moderate torque reserve (torque back-up) (Vahlensieck 1999).
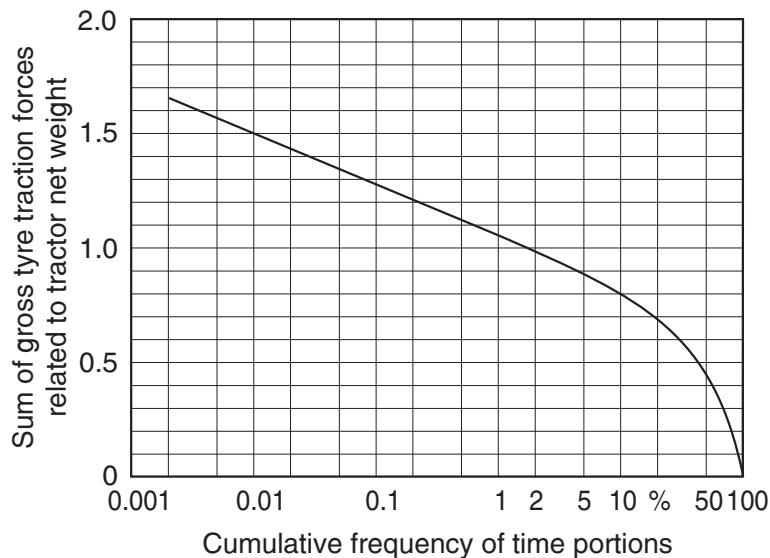
Fig. 14 Total torque load spectra for tractor drive line output. Torque calculation with effective tire radii for normal inflation pressures of about 1.2 bar; probability for European conditions and 4WD about 95 % (Renius 1976, Rempfer 2003).

strategies (but scarcely by local conditions) while the output load spectra are mainly determined by the local traction conditions and the tractor weight.

Relation of load spectra to speed ranges

A complete outline would exceed the purpose of this paper. But a simplified rule may be given. In principle, the transmission output load spectrum must be balanced with the input spectrum by considering the limited power of the engine including dynamic overloads.

This leads for typical weight-to-power ratios of standard tractors to a characteristic vehicle speed of about 5 to 7 km/h. Below this speed, the transmission loads must be derived from the traction dictated load spectrum of Fig. 14. Above this speed, transmission loads should be derived from the input load spectra of Fig. 13. A modification is recommended to increase the curve a little within the typical working range for heavy duty operations up to about 12 km/h and to decrease the curve a little for the resting speed range, which is mainly used for light field operations, transports and tractor movements farm-field-farm.

Regarding the transmission output load spectrum, the special conditions of Nebraska Test and OECD Test are not included. Extremely high pulling forces are generated for speeds much below the mentioned 5 to 7 km/h due to cleaned but well gripping concrete with optimized tire conditions and high ballast. Thus the loads during preparing and

running this part of the mentioned tests should be added to the curve of Fig. 14 creating a certain hump in the left strong part of the spectrum.

A typical dimensioning of a pair of gear wheels meeting a load spectrum with the lowest possible dimensions has been presented first in ASAE paper 76-1526 (Renius 1976).

PRINCIPLES OF CVTs AND THEIR INTEGRATION

Survey on physical CVT principles
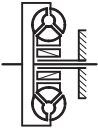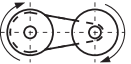
Four typical groups of continuously variable vehicle drives can be called to be important, **Table 4**. Their physical principles are completely different, also their ratio controls and typical fields of application.

The "hydrodynamic torque converter" is achieving by far the highest production volume for vehicle transmissions and offers in the same time the lowest production cost, but has two major weak points regarding the use for tractors:

1. The maximum efficiency is not poor but only available within a very limited band of ratios.

2. The ratio cannot be controlled in a closed loop, as the ratio results automatically from the load.

A solution to diminish weak point 1 is to add a free wheeling element or even a clutch blocking the unit. But this reduces the CVT effect and thus requires a high number of additional conventional

Table 4 Important physical principles of continuously variable transmissions for vehicle drives.

	Type of CVT		Principle of energy transmission	Ratio control	Application	Efficiency
1	Hydrodynamic		Mass forces at pump and turbine	Usually automatically by load	Important for pass. cars and construction machinery	poor
2	Mechanical		Traction forces within friction contacts	Radius of traction force	Important for passenger cars	excellent
3	Hydrostatic		Hydrostatic forces at pump(s) and motor(s)	Displacement of the units	Important for mobile machinery	moderate
4	Electrical		Electro-magnetic forces at generator(s) and motor(s)	Frequency of current or electric flux or load	Upcoming	moderate

ranges. Weak point 2 could be diminished only very little by adjustable vanes of the reaction wheel.

Thus all developments trying an introduction for tractors finally failed. As the authors do not see a principal change of this situation for the near future, the torque converter will not be included in the following considerations.

Those CVTs, which are most considered for tractor transmissions, are listed little more in detail by **Table 5**. As noted by Table 4, the CVT principles No. 3 and 4 are not excellent in efficiency. They are not able to meet the target of Fig. 11, if they are applied in a direct mode. A power split system can increase system efficiency over unit efficiency. Only mechanical CVT units can offer excellent efficiencies in a direct mode, which is their main advantage.

Mechanical CVTs and their integration

Survey

The ratio is controlled by variable effective radii, see, for example, the historical prototypes of Fig. 3 and Fig. 9.

The torque T is in principle a function of normal force F_N , friction coefficient μ and radius r:

$$T = \mu \cdot F_N \cdot r$$

Modern traction drive CVTs are lubricated by oil. The maximum usable friction coefficients (steel/steel) are about 0.06 to 0.12 and are mainly influenced by the type of fluid: high values with special "traction fluid", low values for example with rape seed oils. There is also some research on small dry friction drives resulting in higher friction coefficients.

Best efficiency of the traction contact results from a compromise between slip and rolling resistance losses – similar to traction mechanics of pneumatic tractor tires. The rated Hertz pressures can arrive at levels of 3000 N/mm^2 for toroidal CVTs (Machida et al 1995) and to about 800 N/mm^2 for chain type friction drives. The high normal forces are usually generated by hydrostatic pressures.

The following traction type CVTs are in production for passenger cars:

– Push type chain variators (Van Doorne/Bosch, used in millions of mostly smaller cars)
– Pull type chain variators (P.I.V./Luk/Audi, first introduced in 2000 for cars)
– Toroidal CVTs (NSK Ltd./Nissan, half toroidal version, first introduced for a larger car late 1999).

Table 5 Relevant CVT principles for tractor drives.

|	pros`s	con`s	comments
Mechanical by friction "direct"	High efficiency, low noise level	No start from zero, low shift speeds, power limits (chains)	Push type CVT: Japanese cars Pull type CVT: Audi-LuK chain CVT (2000) Toroid Nissan (1999)
Mechanical power split	Start from zero, low noise level	Efficiency can be lower than with "direct"	Prototypes known under the name "geared neutral"
Hydrostatic "direct"	Long experience, high shuttle comfort	Low efficiency high noise level	Popular for small tractors and crawlers
Hydrostatic power split	High efficiency	Noise, limited flexibility, shift can be complicated, high R&D costs	CLAAS "HM-8" (1996) Fendt "Vario" (1996) "Steyr S-Matic" (2000) ZF "Eccom 1.5" (2001) J. Deere/USA (2001)
Electrical "direct" with Diesel engine	Low noise level, high shuttle comfort	Expensive, low efficiency	Several research activities on units and systems.
Electrical "direct" with fuel cells	Low noise level, high shuttle comfort	Very expensive, extra motor for PTO, high R&D costs	Long term potential, commercial use 2022?
Electrical power split with Diesel engine	High efficiency low noise level	Expensive, high R&D costs	Several research activities on units and systems. Commercial use 2012?
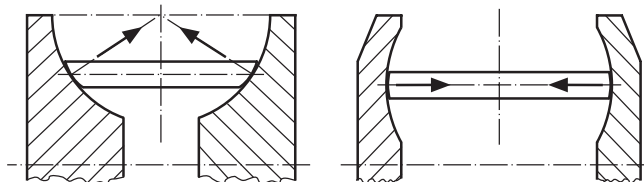
Fig. 15 Half toroidal (left) and full toroidal (right) traction drive CVT, shapes and reaction forces.
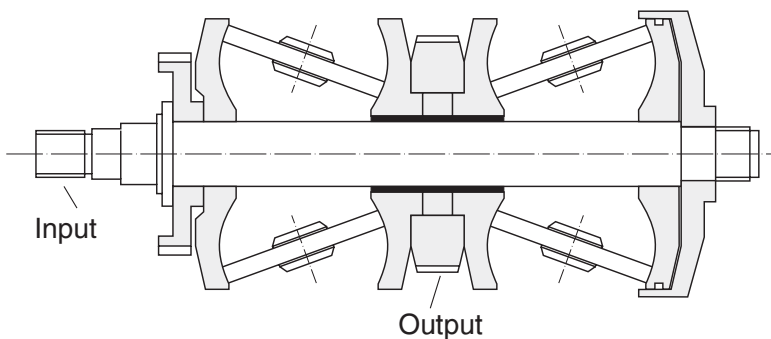
Fig. 16 Traction drive CVT, full toroidal type in double cavity arrangement. Position: High output speed (courtesy Torotrak Ltd.).

Toroidal drive CVTs

Toroidal drives are classified in "full toroidal" and "half toroidal" concepts, **Fig. 15**. A "full toroidal" CVT has been developed by the British company Torotrak Ltd., **Fig. 16** (Greenwood and Soar 1991) and a "half toroidal" CVT by NSK/Nissan (Machida et al. 1995). Double cavity arrangements are recommended for both concepts improving power and efficiency (Machida et al. 1995).

The "Torotrac" has been proposed for various vehicles, also for a tractor transmission with power split for LG Cable Ltd., South Korea (Kim et al. 2000).

A typical advantage of toroidal CVTs is the potential for high torque capacities and compact design due to parallel power flow. The slip is higher than for chain drive CVTs, same for the losses born by drilling friction. Half-toroidal concepts are little better in efficiency than full-toroidal drives. Efficiency drops at high loads and high speed reductions. This is not so important for car transmissions, but is a weak point for tractors.

Steel belt drive CVTs

Infinitely variable steel belt drive CVTs are classified in push type and pull type concepts. Both have been developed over several decades (Schönnenbeck and Wagner 2002).

Very first and today famous fundamentals for the pull type CVT have been presented by Dittrich (1953). The push type version was also subject of many papers; fundamentals see, for example, Fushimi/Fujii/Kanehara (1996).

For tractor transmissions the pull type steel belt variator has been preferred in the P.I.V. version with many prototypes (Fig. 9). **Fig. 17** represents a modern design. The variator works in principle like an adjustable V-belt drive. There are no

PIV chain

Torque sensor

Torque sensor

Fixed pulley

Movable pulley/clamping cylinder (20-50 bar)
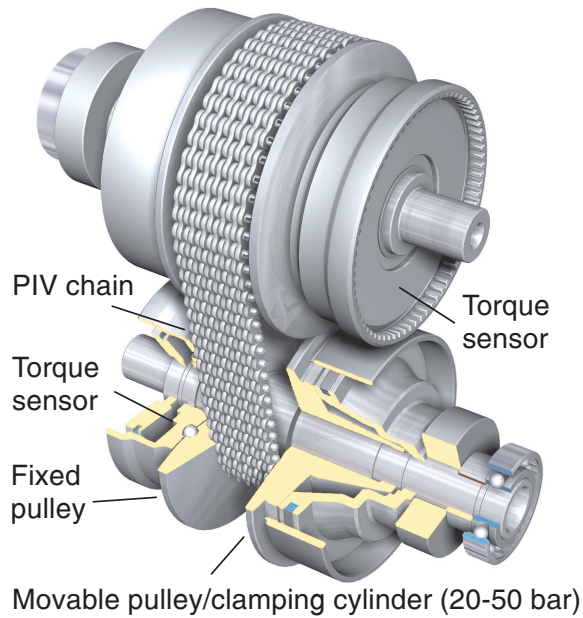
Fig. 17   Pull type chain CVT concept PIV (courtesy PIV Drives).

sliding losses within the steel chain due to the "rolling contact" joints (instead of sliding joints). The joint pins are in traction contact with the pullies. A stiff design is recommended in many publications to prevent losses by radial slip.

This CVT type could catch a lot of interest due to its introduction for cars with the "multitronic®" by Audi/Luk in 2000 (Nowatschin et al. 2000). It is called to be the worldwide "best in efficiency" automatic car transmission with some 100.000 units produced until 2004.

A survey on the fundamentals (Sauer 1996, Srnik 1999) and several other publications confirm the outstanding high efficiency potential – probably the highest of all presently known CVT concepts. If the clamping forces are properly adjusted to the actual torque load, mechanical full load efficiencies of about 95 % can be achieved within a wide band of speed ratios (Sue/Dittrich/Poll 2002). Practical values are lower due to the losses of the hydraulic control system. A simple system with a constant displacement pump doubles the losses, resulting in best efficiencies of about 90 % . An improved hydraulic control system with a variable displacement pump could achieve a measured best efficiency of 92.5 %, **Fig. 18** (Sauer 1995). This map has been picked up for 1000 rpm, higher input speeds such as 3000 rpm are not reducing efficiencies. High sophisticated hydraulic control systems have been developed (Faust/Homm/Reuschel 2002, Mölle 2003).
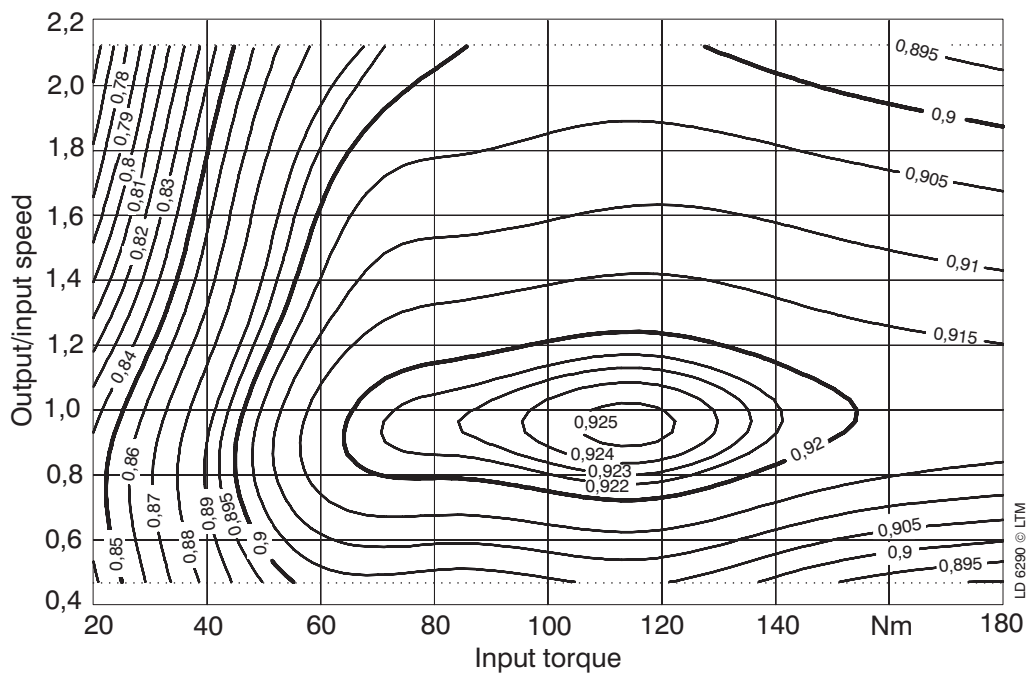
CVT from P.I.V., Input speed 1000 /min, double-sensor system (8,9 Nm/bar, variable displacement vane pump, oil flow 7 l/min

Fig. 18 Measured efficiency map of a chain CVT (similar to Fig. 17). Variable displacement hydraulic pump compensates changing input speeds (no flow surplus at high speeds) thus reducing the losses as compared with a constant displacement pump (Sauer 1995).

The mechanics are complicated. It can, for example, not be tolerated to assume equal clamping forces at both belt pulleys. The following correction factor ζ has been introduced by P.I.V. (Schönnenbeck and Wagner 1992):

$$\zeta = \frac{\text{clamping force driving pulley}}{\text{clamping force driven pulley}}$$

Practical values depend on ratio and torque load. If very low loads are excluded, ζ varies between about 1.0 and 1.5 influenced by stiffness, output speed and load (Sauer 1995). A typical ζ map has been measured by Westenthanner (2000). Actual information on hydraulic low energy control systems with adjusted flow rates can, for example, be picked up from Stöckl (2002). Interesting approaches with speed modulated electro-driven constant displacement pumps have been presented by Koberger (2000), Bradley/Frank (2002) and others.

The live of steel belt CVTs is mainly dictated by chain fatigue, only in the second order by surface wear and pittings. Internal chain dynamics have been measured by Sauer (1995) and confirmed by a precise model of Srnik (1999). Fatigue modeling with external load spectra and internal dynamics was first published by Vahlensieck (1999).

Friction drive CVTs can only be used for future tractor drive transmissions, if two basic problems are solved: The enlargement of the transmission output speed ratio forward and a reverse range. High tech markets will also require an active (powered) zero speed control.

A simple combination of the CVT unit with a conventional power shifted range package has the disadvantage, that the range shifts last long and that an active zero speed control is not possible.

An improved very early approach of solving these problems is plotted in **Fig. 19** based on a patent of J.H. Kress/John Deere (1962). The system works in mode L with power split thus realizing active zero speed control by "geared neutral". Modes R and H are "direct". There is high relative parasitic power circulating in mode L, but this is mitigated by the traction limit. Range shift between L and H is made in "synchronous points" which cuts adjustment times of the CVT unit. An application of this interesting idea for series production did not take place until now, but the principle of combining a power split range with one or more direct ranges is again under investigation now (Resch 2004), even for passenger car CVTs such as those of Audi-Luk (Englisch et al. 2002).

Another approach of increasing the output speed ratio of systems with chain variators is the so-called i^2-principle, proposed for passenger car transmissions. The variator is used in two ranges with alternative opposite power flow direction and synchronous shift between both (Höhn/Pinnekamp 1994). Four shift elements are necessary. Prototypes at TU Munich and ETH Zurich are working very well. The main disadvantage of the i^2-principle for tractors is a poor power density of the variator: High torques/low speeds in the low range and high speeds/low torques in the high range. Full performance of the variator would however need a combination of high speeds and high torques. A certain
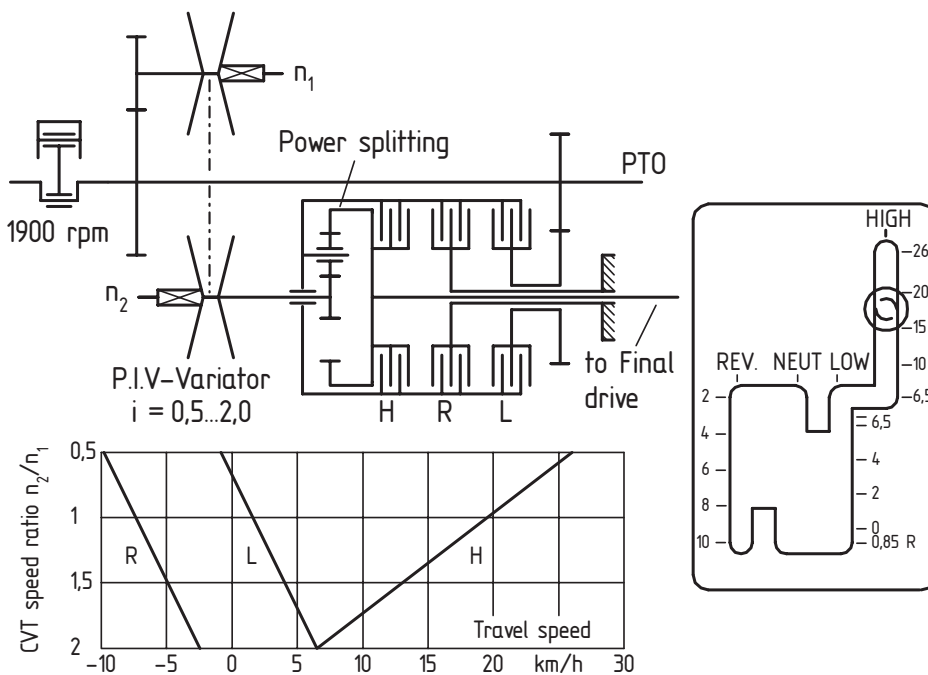

Fig. 19 System with steel belt CVT and 3 ranges (principle Kress 1962).

load limitation in the low range may be possible by the traction limit. A reverse ratio as well as an active zero speed control needs additional efforts.

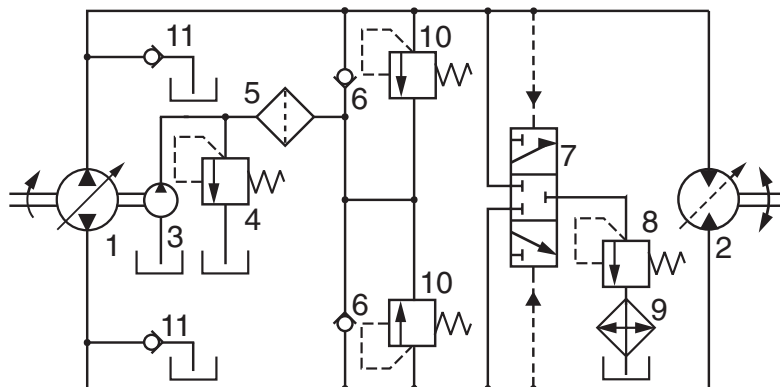## Hydrostatic CVTs and their integration

Survey and basic functions

A hydrostatic CVT is formed by the combination of at least one hydrostatic pump and at least one motor. At least one unit must have a continuously variable displacement.

The circuit of a simple typical hydrostatic CVT unit is shown by **Fig. 20.** It usually works with one variable pump 1 and one constant or variable motor 2 in a closed circuit. A variable motor can improve efficiency for high speeds. The charge pump 3 with safety valve 4 is always feeding the low pressure pipe passing through filter 5 and check valve 6. Surplus flow leaves the low pressure pipe automatically by the flushing valve 7 arriving at the tank through pressure relief valve 8 and cooler 9.

The charging system (with, for example 20 bar) is in addition often used for transmission control thus having in total the following duties:

– to replace leakage
– to control fluid temperature
– to control fluid contamination
– to serve as auxiliary power
– to enable high pump speeds

The pressure relief valves 10 are safety elements. If the charging system fails, an emergency re-filling of the circuit is possible by the suction check valves

11. They are however not used in general. If a blow out takes place for a longer period of time, fluid temperature will exceed the limits and the CVT can be destroyed. Modern systems prevent this danger by a pressure limitation without blow out: The pump reduces its displacement automatically, if a pressure signal exceeds a given limit. Such limits are typically between 300 and 550 bar (high values for high power in case of piston units).

Basic equations

The ideal "force balance" between mechanical and hydrostatic variables is described for pumps (P) and motors (M) by the the torque T, the displacement V and the pressure difference p as follows:

$$T_P = \frac{p \cdot V_P}{2 \cdot \pi}$$

$$T_M = \frac{p \cdot V_M}{2 \cdot \pi}$$

The ideal "speed balance" between rotational shaft speed n and fluid flow Q is

$$Q = n \cdot V = \frac{\omega \cdot V}{2 \cdot \pi}$$

with ω as angular velocity. The ideal CVT torque ratio thus is

$$\frac{T_P}{T_M} = \frac{V_P}{V_M}$$

and the ideal CVT speed ratio

$$\frac{n_P}{n_M} = \frac{V_M}{V_P}$$

These ideal formulas are influences in reality by two groups of losses:

– leakage (reducing output speed)
– all types of friction plus charging pump torque (reducing output torque)

A typical working point of a well designed hydrostatic CVT with swash plate units can have about 8 % speed loss and 10 % torque loss at a typical working point (pressure 200-300 bar).

Low pressures result in low speed losses but very high relative torque losses. CVT working points with pressures below about 100 bar should there-

Fig. 20 Typical circuit of a hydrostatic CVT unit. Symbols ISO 1219-1. Legend see text (Renius 2003).

fore be avoided. This is one reason, to use variable motors.

Rules for high efficiency

The following rules can be recommended:

– Pump and motor variable
– Units with large tilting angles
– Actual heavy duty pressure about 200-300 bar
– At least the motor in bent axis concept
– Motor displacement above pump
– Input speed not above Diesel engine speed
– Controlled circuit pressure losses
– Charging system not overdesigned
– No major paddling losses

Hydrostatic CVT integration to drive trains

Power split configurations will be presented in separate chapters.

The following information thus concentrates on CVTs with "direct power flow". They are only popular for small tractors. Their efficiency is relatively poor but this is not so important in most applications. Turf maintenance and garden tractors with rated engine power of about 10-20 kW (Betz 1991) require, for example, high comfort and proper speed control and they often use not all the engine power for driving due to powered implements.

There are several possibilities to combine a hydrostatic CVT unit with mechanical transmission components. **Fig. 21** shows two concepts. The upper configuration is typical for most CVTs of smaller tractors. The standard pattern of displace-
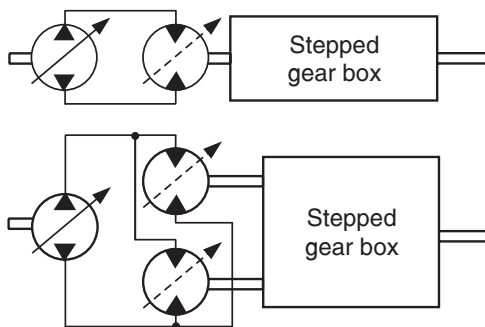
Fig. 21 Combination of hydrostatic CVTs with conventional gear boxes. Above: Standard concept for small tractors, below: new concept of a wheel loader.
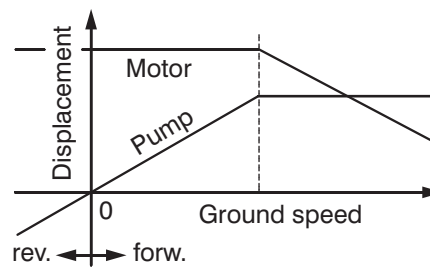
Fig. 22 Pump and motor displacement versus speed for the standard concept of Fig. 21 (Browning 1978).

ment control in case of variable motor is shown by **Fig. 22**.

The lower version of Fig. 21 has been introduced 2002 for Liebherr wheel loaders. The special gear box enables synchronous power shifts between three ranges (Renius 2003). Efficiencies cannot compete with power split CVTs as introduced for tractors but are on a high level for the "direct power flow" philosophy (Krauss and Ivantisynowa, 2004) and clearly above those of transmissions with hydrodynamic torque converters.

Coming back to tractors: A high number of hydrostatic CVTs with direct power flow is applied up to about 30 kW. A typical structure is presented by **Fig. 23**, based on material which has been kindly provided by Yanmar, Japan. The diesel engine drives the variable displacement swash plate pump working on a constant displacement swash plate motor. Motor output shaft is combined with a conventional three range gear box in order to meet the tractor speed range demands.

Electrical CVTs and their integration

Electrical CVTs have the following advantages:

– Low noise level,
– low maintenance/repair costs,
– low energy transfer losses
– high ability for control systems
– low required energy for ratio control
– environmental friendly

Critical points are safety aspects for the high required voltage, costs and efficiency.

An example may demonstrate the voltage demand. If we assume 100 kW power and 200 A current, a voltage of 500 V is needed.

Electrical drives could gain importance for mobile machinery drives because of the availability of the following new technologies:
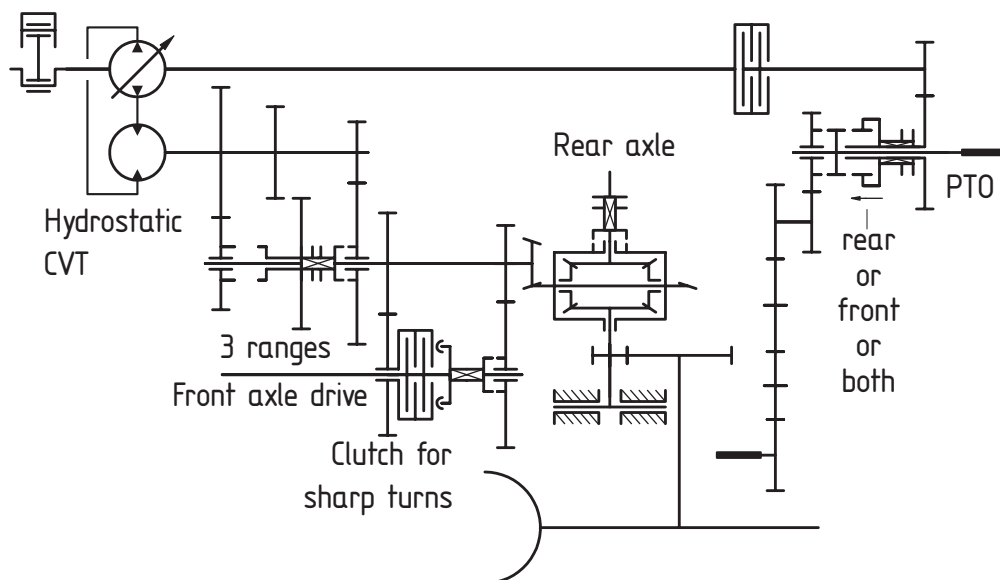
Fig. 23 Drive train of Yanmar tractors with hydrostatic CVT (swash plate units) for about 20-25 kW rated engine power.

– new power electronics for
  • AC/DC resp. DC/AC transformation
  • continuously variable voltage
  • continuously variable frequency
  • fast induction current control
– new magnetic materials and cooling principles
– digital overall control systems

As outlined earlier by Table 5, electrical CVTs have a potential for tractor drive transmissions in power split systems, later on perhaps also in a direct mode with fuel cells. All possible solutions may require additional conventional transmission elements, such as ranges and gear wheels for speed reduction.

Efficiency trends of electrical generators and motors under typical loads have been collected by

Fig. 24 Typical efficiencies for three concepts of electrical machines (Tarasinski 2004).

Tarasinski (2004), **Fig. 24**. The efficiency band has about the same level as for hydrostatic units, but the level of rotational speed is much higher. The asynchronous machine can achieve efficiency levels above 85 % in a wide spee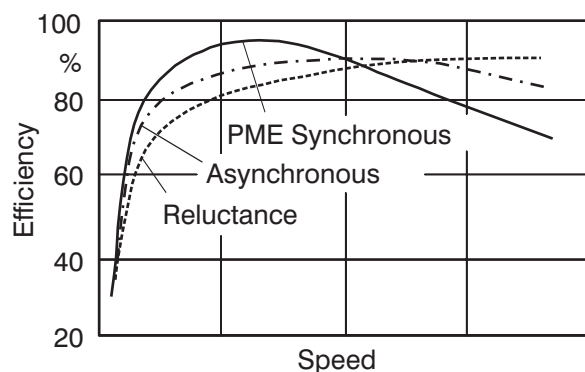d band, while the PME concept with permanent magnets has better peak values but they are limited to low speeds. The reluctance machine can offer high efficiencies at high speeds. Speed levels should be as high as possible as the force density is rather poor. According to Tarasinski (2004), a circumferential force density of only 20 kN/m^2 between rotor and stator is typically for a fluid cooled asynchronous machine and 40 kN/m^2 for an intensively fluid cooled synchronous machine. The prices for power electronics are still very high, but a decrease is expected by their introduction for passenger car hybrid transmissions.

Such a transmission has been introduced commercially first by Toyota for its "Prius" in 1997 (Killmann et al. 1999). It is since 2003 in the second generation on the market without changing its basic structure **Fig. 25.** The engine drives the carrier of a planetary, where the power is split into a CVT path via sun gear/generator/controller/motor and a mechanical path to the ring gear, where merging of both takes place. The system can be classified to be output coupled (see following chapter). It has several additional functions such as starting the gasoline engine, electrical braking (charging the battery), charging the battery in case of power surplus, electrical boosting, reversing and driving slow speeds electrically for example for stop and go traffic.

The Prius reinforced broad research on hybrid CVTs in the car industry. But it may also encourage research for electrical tractor drives, which is carried out since some years for example at the Univer-
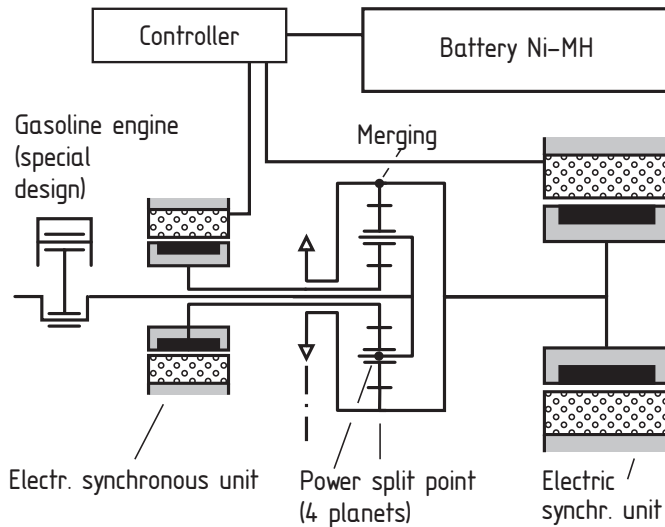
Fig. 25 First commercial hybrid car CVT, working with an electro-mechanical power split system: Toyota "Prius" 1997.

sity of Dresden (Barucki 2000). Also tractor companies, mainly Agco-Fendt and John Deere, indicated activities.
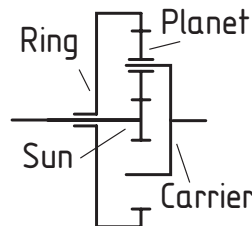
PRINCIPLES OF POWER SPLIT CVTs

Basic functions of power split CVTs

The previous chapters indicate, that the efficiency demands for tractor CVTs above about 50 kW cannot be met by "direct" hydrostatic or "direct" electrical transmissions

A special transmission system is needed, called "power split" system in order to increase efficiency. It often needs a planetary for splitting or merging the power.

Fig. 26 shows the three-shaft standard planetary set.

Three planetary gear wheels are typical as they offer outstanding self centering of the sun without a bearing and optimal power distribution. Speed rela-

Fig. 26 Standard planetary gear set.

tions can be briefly analyzed by plotting the straight lines of circumferential speeds for all elements. Further fundamentals see Browning (1978).

The basic idea of the power split principle is demonstrated by **Fig. 27**.

The input power is split into

– a constant ratio mechanical path and
– a variable ratio path using a CVT unit

On the output side of the transmission both power flows are merged again. Six basic power split configurations with planetaries like Fig. 26 have been analyzed by Jarchow (1964). Total efficiency is typically above that of a CVT "direct" due to the high efficiency of the mechanical path. This shall be demonstrated by a typical example with realistic values for full power:

We assume for the structure of Fig. 27 a split ratio of "70 % mechanical" and "30 % with CVT unit" with a related mechanical efficiency of 95 % and a CVT efficiency of 80 % (hydrostatic, swash plate units). The resulting total efficiency can be calculated as

$$0.7 \cdot 0.95 + 0.3 \cdot 0.80 = 0.905 = 90.5 \text{ %}$$

The efficiency benefit increases with decreasing CVT power portion. Best total efficiency can therefore usually be expected with zero power in the CVT path, which is possible with hydrostatic units. Total efficiency would be for such a lockup point in this example above 90.5 %, but below 95 % due to the idling losses of the hydrostatic unit and its leakage (it is still loaded, but delivering no power). Another not so favorable phenomena with power split CVTs is the possibility of circulating power. Regarding again the system of Fig. 27, we can assume, that the variator output speed to the sun gear can be reversed in case of a hydrostatic or electrical CVT unit. This means, that its power flow direction is re-
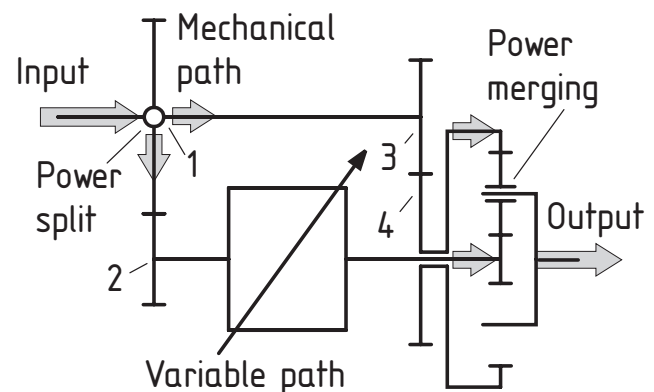
Fig. 27 Principle of a CVT with external power split and two power transmitting pathes.

versed - in Fig. 27 now from right to left. This power is superimposed to the system input power, it must be transferred by the mechanical path again to the right and it is thus called "circulating". If its value is low compared to the input power, total system efficiency can still remain above that of a direct CVT. This strategy is therefore often applied in hydrostatic power split tractor transmissions in order to increase output speed range.

Efficiency versus speed ratio range

An increased system efficiency results in a reduced speed ratio range and an enlarged system speed ratio range creates efficiency losses (both cases compared with CVT mode "direct"). In order to demonstrate a typical speed and speed ratio modeling, the system of Fig. 27 shall be analyzed as an example in the following section.

The rotational speed of the sun and the ring gear in Fig. 27 can be determined by

$$\omega_{sun} = \frac{\omega_{in}}{i_{12} \cdot i_{Var}}$$

where

ω_{in} rotational speed of input shaft

$i_{12} = \dfrac{\text{speed shaft 1}}{\text{speed shaft 2}}$

$i_{Var} = \dfrac{\text{speed shaft 2}}{\text{speed sun wheel}}$

The speed of the ring gear can be expressed as follows:

$$\omega_{ring} = \frac{\omega_{in}}{i_{34}}$$

where

$i_{34} = \dfrac{\text{speed shaft 3}}{\text{speed shaft 4}}$

The shaft speeds at a planetary gear set, Fig. 26, can be expressed by the equation of Willis:

$$\omega_{sun} - \omega_{ring} \cdot i_0 = \omega_{carrier} \cdot (1 - i_0)$$

where

ω_{ring} rotational speed of ring gear

$\omega_{carrier}$ rotational speed of carrier

$$i_0 = -\frac{\text{number of the teeth at ring gear}}{\text{number of the teeth at sun gear}}, \quad \text{here}$$

$$i_0 = -3$$

With these equations, the ratio of system input to output speed can be found as

$$i_{total} = \frac{\omega_{in}}{\omega_{carrier}} = \frac{\text{speed input shaft}}{\text{speed output shaft}} =$$

$$= \frac{(i_0 - 1) \cdot i_{12} \cdot i_{34} \cdot i_{var}}{i_0 \cdot i_{12} \cdot i_{var} - i_{34}}$$

The variator speed ratio range is defined as

$$S_{Var} = \frac{i_{var,max}}{i_{var,min}}$$

where

$i_{var,min}$ minimum variator speed ratio

$i_{var,max}$ maximum variator speed ratio

In this example S_{Var} is assumed to be 4, symmetrically to $i_{var} = 1$ resulting in the values $i_{var,min} = 0.5$ and $i_{var,max} = 2.0$.

If we assume $i_{12} = i_{34} = -1$, the speed ratio range of the total transmission system S_{total} gets to

$$S_{total} = \frac{\max(|i_{total}|)}{\min(|i_{total}|)} = \frac{1.14}{0.8} = 1.429$$

This means that the total transmission ratio range is 1.429 and thus only about 35 % of the variator speed ratio range.

Internal and external power split configuration

Power split CVT concepts can be divided in the two groups

– CVTs with internal power split

– CVTs with external power split

The transmission of Renault described already in Fig.2 is an example for the first group. **Fig. 28** shows a similar concept.

The swash plate axial piston units are placed in a compact back-to-back design. The cylinder block 1 at the primary unit is driven by the input speed n_1, the cylinder block of the secondary unit is fixed in the surrounding transmission housing. The unit capsule and both swash plates revolve with the output speed n_2. Thus the fluid displacement at the
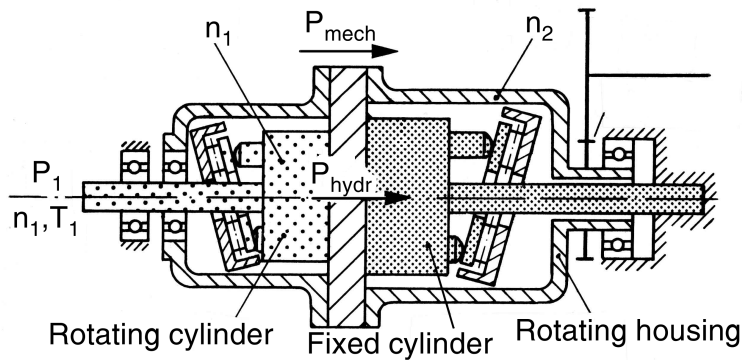
Fig. 28 Continuously variable transmission with internal power split (n rotational speed, T torque, P power, 1 input, 2 output).

primary unit is determined by the speed difference $n_1 - n_2$ and at the secondary unit by the output speed n_2. The input torque of the pump is working on both, the fluid and via the swash plate on the revolving capsule. Therefore the power split ratio is controlled by the speeds:

$$P_{hydr} = 2 \cdot \pi \cdot T_1 \cdot (n_1 - n_2)$$
$$P_{mech} = 2 \cdot \pi \cdot T_1 \cdot n_2$$

where

$\omega = 2 \cdot \pi \cdot n$

T_1 torque on input shaft

n_1 primary rotational speed

n_2 secondary rotational speed

P_{mech} power input to mechanical path

P_{hydr} power input to hydraulic path

Power merging takes place at the swash plate of unit 2.

In neutral position of swash plate the primary unit produces no oil flow and the output speed is zero. Driving through the forward speed range, the primary swash plate is first swivelled to the maximum tilt angle, while the secondary swash plate keeps its maximum position. Then the displacement of the secondary unit is reduced to zero. At start up the hydraulic power is 100 % in the very first moment. If both speeds are equal, the

power is transmitted fully mechanically, but the hydrostatic unit is of course still under pressure (leakage).

To the knowledge of the authors this concept has been realized only in prototypes (Molly 1966) and not introduced commercially for agricultural machinery. As explained in the following section, this system is similar in basic mechanics to output-coupled external power split concepts.

Power split CVTs with input- and output-coupled planetaries

The following considerations concentrate on external power split systems, as they seem to be the more important type for tractors and other mobile machinery. A huge number of solutions has been proposed (also by many patents): sometimes difficult to analyze. An excellent early classification and mathematical analysis has been presented by Kress (1968). For CVTs with external power split and three shaft standard planetaries (Fig. 26), Kress formed two typical groups, **Fig. 29**:

A Systems with input-coupled planetaries

B Systems with output-coupled planetaries

"Coupling" means to cover the surplus grade of freedom of the planetary. This classification is very useful and was confirmed by many other authors (Browning 1978, Ziebart et al. 1983, Renius 1999 and others). There are further modifications possible within the groups A and B by other planetary arrangements (Jarchow 1964).
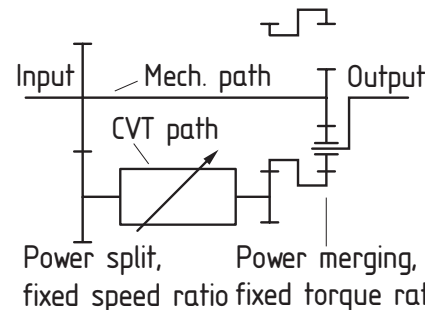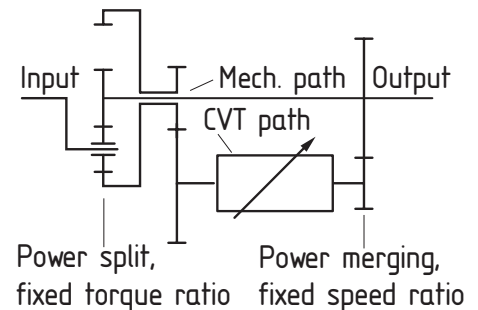
Fig. 29 Two basic concepts of continuously variable transmissions using the external power split principle (classification of Kress 1968).

The structures A and B are mirrored at each other, but their operational behavior is completely different, as shown in the following.

Analysis of power split concepts should address the expected efficiency characteristic, the portion of the power through the variable path and, the circulating power and the rotational speeds of transmission elements.

The following analysis is done with the software Matlab, based on constant full power efficiencies, which is a simplification but can aid as a first useful approach (Resch 2004). The following sign convention was used:

– Positive speeds have the direction of the input speed.
– Output power is considered to be negative, whereas input power is positive.
– The sign of torque is determined according to the signs of speed and power.

The diagrams in **Fig. 30** and **31** use 97 % efficiency for each pair of gear wheels and 85 % for the variator. The variator value is based on hydrostatic high angle bent axis units. System efficiency diagrams for other variator efficiencies follow later on.

Fig. 30 analyzes concept A (input coupled) by estimated efficiency plots and related power in the variable path indicating in the same time the circulating power. This is all plotted versus transmission output speed (as a measure for vehicle speed). Typical for this concept is the hyperbolic increase of the power in the variable path towards zero output speed. Near the start up point there is a huge amount of circulating power with the consequence of poor efficiencies although mitigated by the traction limits (Fig. 10). The circulating power for the forward direction is called "negative" for the variator, as its direction points from right to left – opposite to the normal direction. The circulating power for the backwards regime is called "positive", as the direction is normal. In this case, the CVT unit has however to handle a higher power level – always above input power – thus creating a very poor total efficiency level. The best efficiencies are achieved in the forward mode with low power portions in the variable path with an optimum for the lockup point. Practical peak values are some % lower due to the idling friction losses and the leakage of the variator being under load.

The target of Fig. 11 would require for the system of Fig. 30 efficiencies of about 90 %, as Fig. 11 includes the final drive. This limits the usable bandwidth of output speeds with the consequence of additional ranges, as realized by Claas, Steyr, ZF and John Deere.

If backward operations with small speeds are important, a mechanical reverser should be considered to supply acceptable efficiencies.

Fig. 31 shows for concept B (output coupled) completely different characteristics. Starting up from zero output speed the power in the variable path is first equal to the input power resulting in a poor efficiency. The absolute losses are again mitigated by the traction limit (Fig. 10). The power transmitted by the mechanical path is increasing proportionally with output speed. The total efficiency rises linearly as constant efficiencies are assumed for both power split paths. If the primary shaft of the variator is locked up, the power is transmitted fully mechanically arriving to the best possible total efficiency, as described earlier. Also in this case, practical peaks are some % lower as explained before. Once the primary shaft starts to turn in the opposite direction, negative circulating power arises and the efficiency decreases in a steeper decline. Also this power split system requires additional ranges as realized by Fendt. For backward operation the rotational speed of the sun wheel changes its direction and turns to be an input shaft to the planetary gear set. Positive circulating power occurs creating poor efficiencies, however in this case mainly for higher reverse speeds. If they are important, a mechanical reverser should be considered.

Fig. 32 and **33** demonstrate the influence of the variator efficiency on the system efficiency. Sensitivity is very high in working points with high variator power - in particular for reversed output speeds.
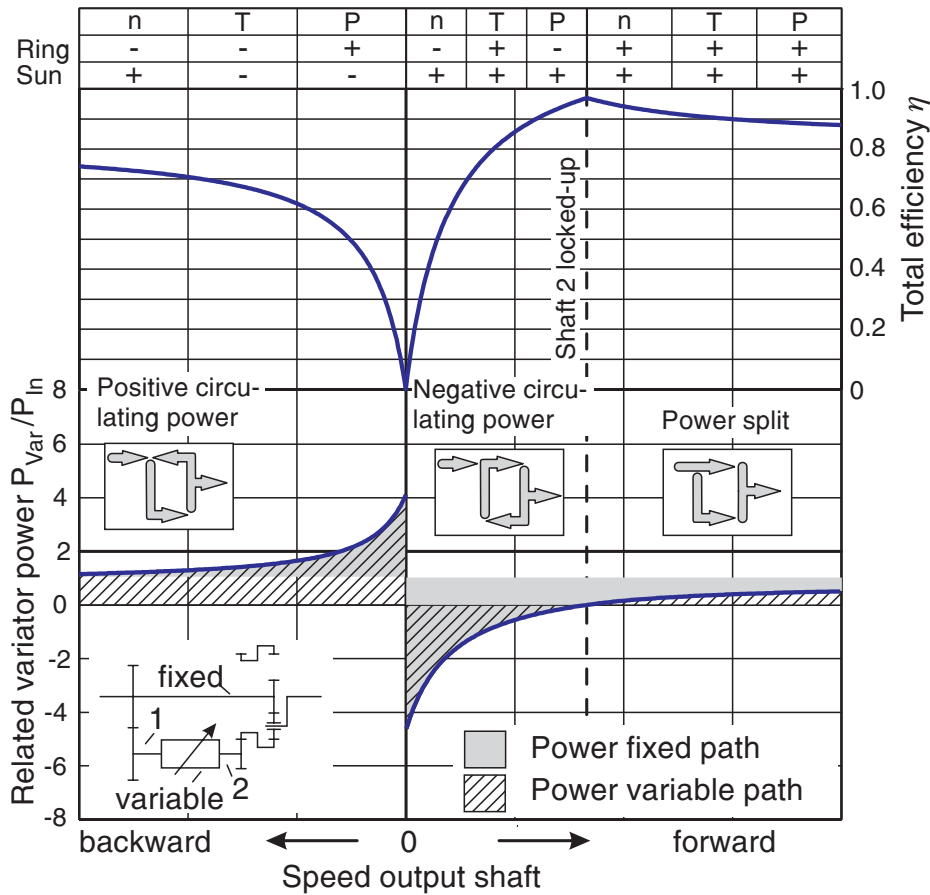
Fig. 30 Full power characteristics and efficiencies for an input coupled CVT configuration (upper part: direction of torque T, speed n and power P at the planetary gear set). Assumed efficiencies: spur gears 97 %, Variator 85 %.
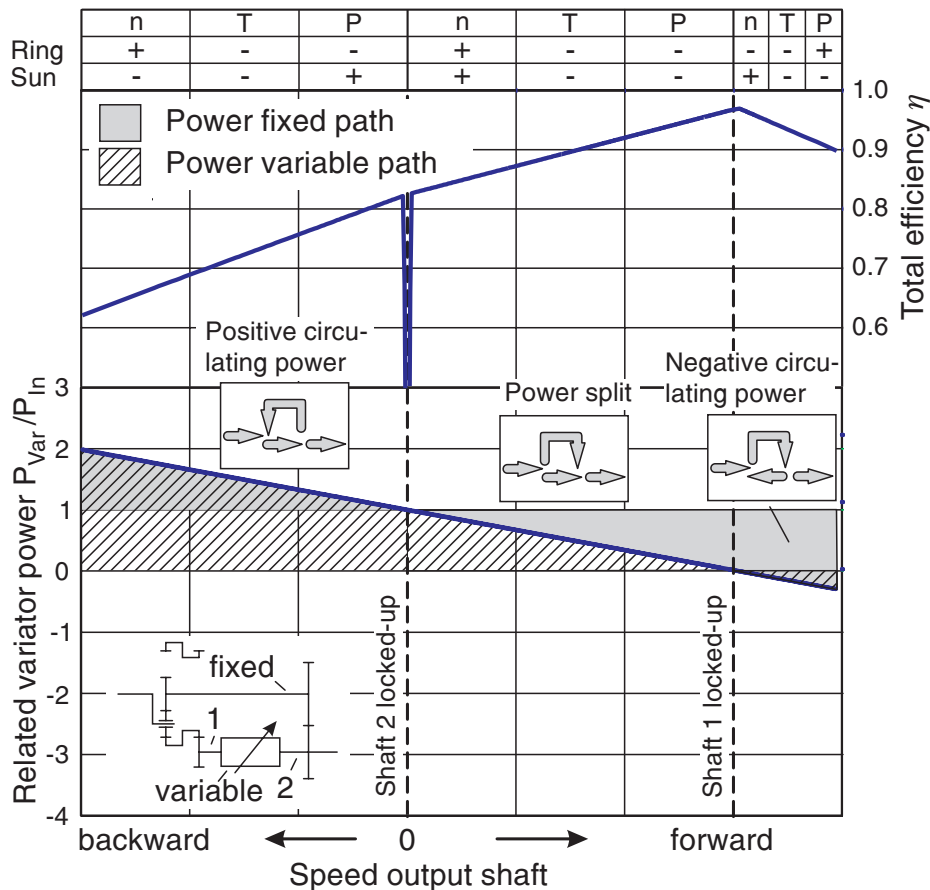
**Fig. 31 Power characteristics and efficiencies for an output coupled CVT configuration (upper part: direction of torque T, speed n and power P at the planetary gear set). Assumed efficiencies: spur gears 97 %, Variator 85 %; maximum torque ratio input to output 50.**

Conclusions for practical applications

1. Power split CVT systems are able to increase system efficiency above that of a "CVT direct" mode.

2. Both principles – power split systems with input and output coupled planetaries – are of interest and are commercially used.

3. Power split CVT systems cannot cover the full required tractor speed range, they need additional mechanical ranges. The smaller the relative variator power, the better the system efficiency, but the higher the required number of additional ranges.

4. Power split system engineering should consider efficiency benefits of the lockup point. It is favorable to apply this point for often used heavy duty working speeds (for example 6-12 km/h) and high transport speeds.

5. Computer aided analysis is recommended with tools like Matlab/Simulink, AMESim and others.

6. Losses of the spur gear wheels have a higher influence on the system efficiency than often expected, in particular for working points with circulating power or high rotational speeds. A proper active lubricating system (oil pump) and oil level management is recommended to prevent plunging losses.

7. Input coupled systems are able to produce zero output speed also with mechanical variators, such as chain variators or toroidal units ("geared neutral"). The disadvantage of very high circulating power portions are mitigated for tractors by traction limits (Fig. 10).
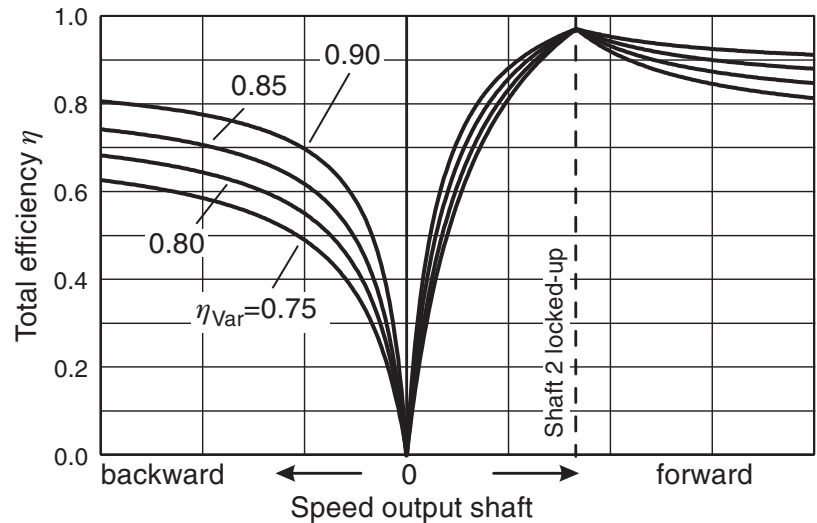
Fig. 32 Influence of the variator efficiency on the overall efficiencies for the structure from Fig. 30.
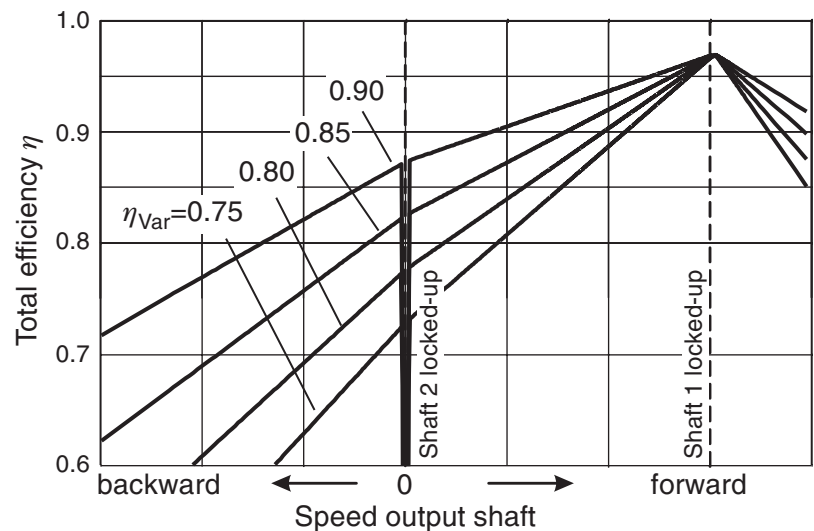
Fig. 33 Influence of the variator efficiency on the overall efficiencies for the structure from Fig. 31.

EXAMPLES OF POWER SPLIT CVTs FOR TRACTORS

Introduction

Five concepts of power split CVTs have been presented commercially for larger tractors until now as mentioned in Table 5. Four of them could enter and stabilize a series production stage introducing meanwhile most of the control strategies as listed by Table 3.

The Fendt "Vario" was the first (1996), followed by Steyr S-Matic (2000), ZF "Eccom"

(2001) and the John Deere "AutoPowr" for the 7010 series (2001). All dates indicate use for tractors produced in series. The Steyr transmission business belongs since May 1, 2000 to ZF. The status of 1998 (Claas, Fendt, Steyr and ZF) has been reviewed by Lang et al (1998) collecting some basic specifications (1998). All transmissions have been commented also in the German Yearbook Agricultural Engineering 1997-2003, always within the chapter "engines and transmissions" (see, for example, Renius and Koberger 2001). A very detailed mathematical analysis of all mentioned transmissions has been presented recently (in Spanish) by P. Linares (2003) referring to the fundamentals of Kress (1968).

The transmission systems are represented by uniform own graphics using ISO 1219 symbols for hydrostatics and own symbols for mechanical elements (Renius 1968 and 1976a).

The Claas-Jarchow "HM-8" and "HM-II"

A lot of pioneering work was carried out since the late 1980s by Claas resulting in the concepts Claas "HM-8" (about 140 kW) and the later larger Claas "HM-II", also called "Traxion" (about 220 kW). Both have been developed for the Claas carrier tractor "Xerion" and both are based on a patent of Jarchow, a leading German specialist in power split systems (Jarchow 1981). The HM-8 had 7 power split ranges in order to keep the hydrostatic power portion low and efficiency high. One creeper range works "direct". The concept became public 1989 and was produced since 1996, while the HM-II (Traxion) with 5 power split ranges, one "direct" and a conventional power shift reverser by clutches was presented at Agritechnica 1999 (Fredriksen 1999). Both transmissions could only arrive to a limited production volume.

The Claas-Jarchow approach is characterized by an input coupled four-shaft compound planetary which consists of two standard planetaries (Jarchow 1981, Renius and Koberger 2001, Linares 2003). The shifts between the ranges are done at synchronous speeds with dog clutches and the power is handed over without any power interruption in a very short time, in which the ratio is however kept constant (no acceleration/ deceleration of the vehicle). This principle is very difficult to realize but doesn't need friction clutches (costs, idling losses). Safe, comfortable shifts require a high sophisticated electronic control system.

Measured Claas "HM-8" efficiencies have been found to be outstanding (diagrams see Renius and Vahlensieck 1996, Renius 2000), covering the target of Fig. 11. Both Claas CVTs will not be discussed more in detail because of limited space in this paper.

The Fendt "Vario"

Fendt announced its new, infinitely variable "Vario" transmission at Agritechnica 1995 for the largest new Fendt tractor "926 Vario" (191 kW), **Fig. 34**. It became the worldwide first in series produced power split transmission for standard tractors (1996). Its principle was soon expanded to more and more other Fendt models forming meanwhile 3 transmission families. At present (end of 2004) 4 tractor families 400, 700, 800 and 900 are in pro-
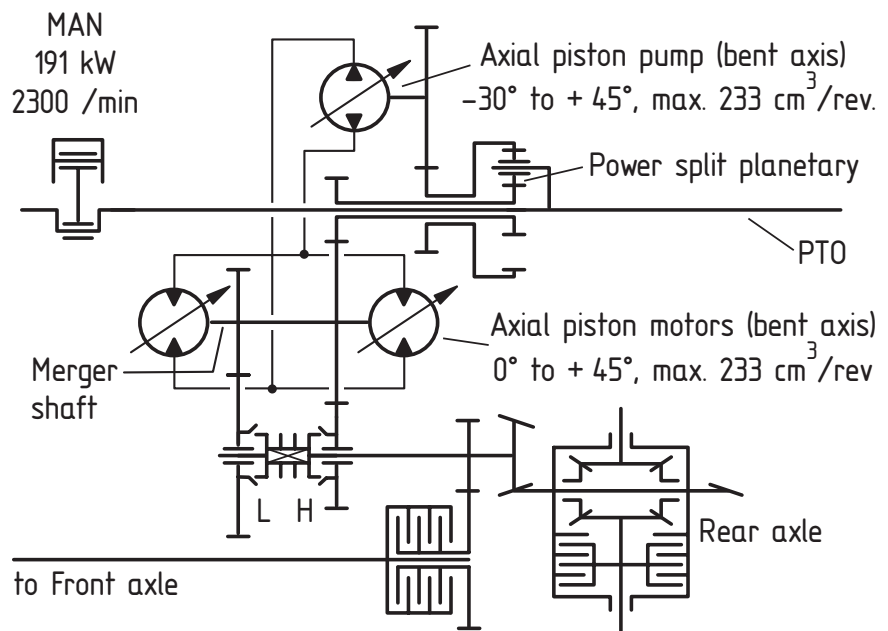
Fig. 34 First successful infinitely variable hydrostatic power split transmission for standard tractors, Fendt "Vario" for the tractor "Favorit 926" (1996).

duction with the "Vario" CVT as standard equipment. It became recently also adopted to MF tractor series 7400 (2003) and 8400 (2004), called "Dyna-VT". About 40.000 "Vario" transmissions have been produced end of 2004. It could catch an outstanding number of innovation awards and became the main driving force for the introduction of power split transmissions for tractors.

The Fendt "Vario" is a power split CVT with output coupled standard planetary (compare with Fig. 29). The structure is based on a principle of Molly (1966). Basic development was done by the Fendt engineer H. Marschall (1936-1989), who got several patents on it (Marschall 1973). In the forward direction, power is split by the planetary transmission and merged by the shaft of the two hydrostatic units (Dziuba and Honzek 1997). The basic power characteristics follow Fig. 31. Hydrostatic power portion is 100 % in the starting point, but zero at the top speeds of the ranges.

The power split unit is combined with two conventional ranges. Range L covers speeds up to 32 km/h, range H up to 50 km/h. These top speeds are near the lock-up points. The speed potential beyond the lock-up point is not used in all "Vario" transmissions in order to prevent any circulating power in the forward mode. Hydrostatic power flow is thus not reversed. Range shift is not needed in the field. Regarding on-road operations, an automatic shift on-the-go was introduced 2002/03. It allows finger tip range selection within a speed band up to 20 km/h. The shift is done by active synchronization support of the CVT. This is, for example, useful for heavy transports on hilly roads. The small power interruption is mostly not realized by the driver. For light duty transports, range H is sufficient for all speeds from zero to 50 km/h.

The Fendt concept requires very high maximum hydrostatic power and thus large units. All commercially available variable axial piston pumps and motors had been quoted to be too poor in efficiency to meet the target of Fig. 11. "Wide angle" units with spherical pistons and piston rings had been early proved at TH Braunschweig to offer outstanding efficiencies (Hoffmann 1970), but had been available only in fixed displacement versions.
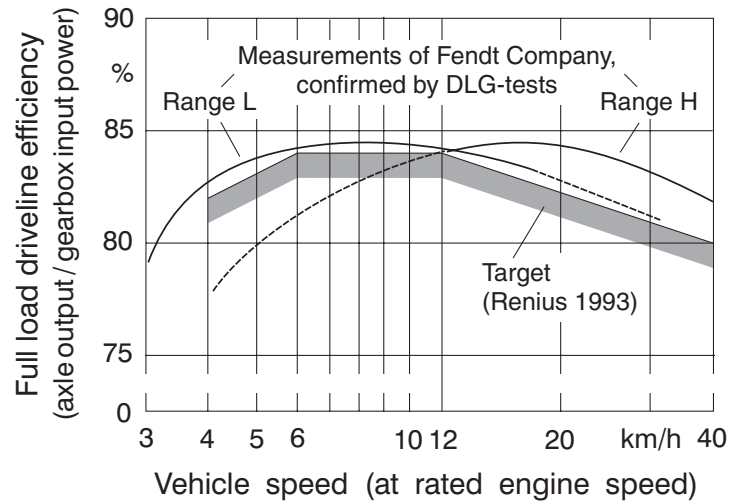
Fig. 35 Full load drive line efficiencies (axle power by transmission input power) of the Fendt 926 "Vario" and its comparison with the target of Fig. 11.

Fendt started therefore an own development of 45 degree variable bent axis units, in the final stage with support of Sauer-Sundstrand (today Sauer-Danfoss). These units offer in their best points efficiencies of 95-96 % (swash plate units rather 89-92 %). The combination with a very low number of mechanical transmission elements resulted in an excellent full load efficiency curve, **Fig. 35**, which could well meet the target of Fig. 11 in spite of the high hydrostatic power portions. Due to the high efficiency level it is even possible to reverse in power split mode in spite of regenerative power.

A serious problem of the Fendt philosophy was the high noise level of the large units. A break through in noise reduction could be achieved by isolating the CVT body noise by elastic suspensions (as usual for IC engines).

The Steyr "S-Matic" for Steyr, Case, New Holland and others

The Steyr "S-Matic" has been developed within the 1990s by the Austrian company "Steyr Antriebstechnik (SAT)" which was taken over by Zahnradfabrik Passau GmbH (part of ZF) in 2000. A first concept became already public in 1994 (Ulbrich 1994), series production was announced with a modified structure (Aitzetmüller 1999) and realized early 2000 for the standard tractors Steyr "CVT" and the Case "CVX" (both 88-125 kW).

Speed		K1	K2	K3	K4
1	0–8 km/h	×			
2	8–14 km/h		×		
3	14–30 km/h			×	
4	30–50 km/h				×

× engaged
forward: KV
backward: KR
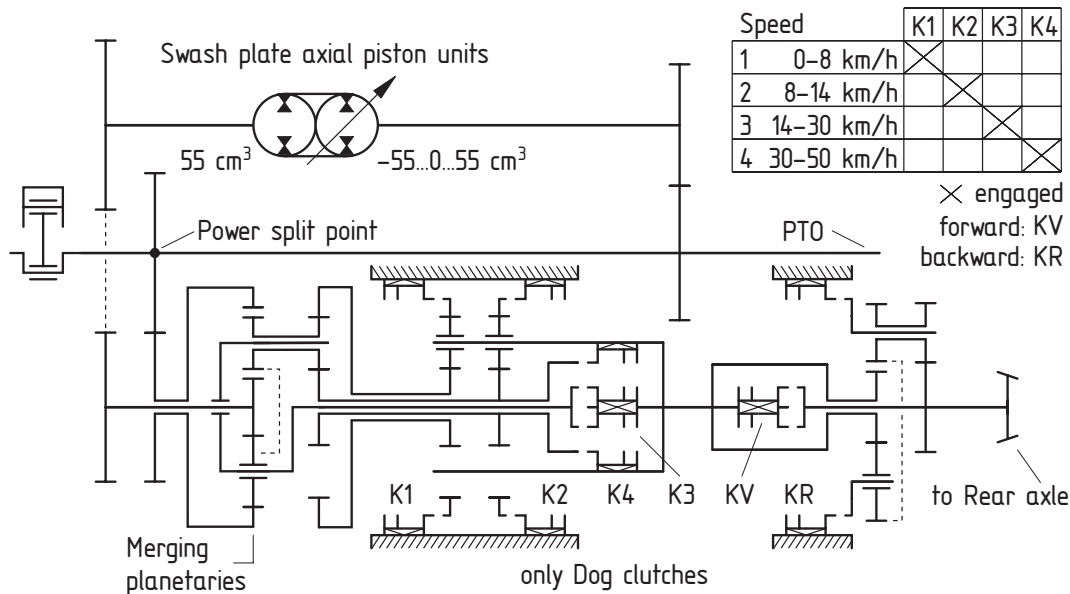
Fig. 36 Hydrostatic power split transmission Steyr "S-Matic" (2000).

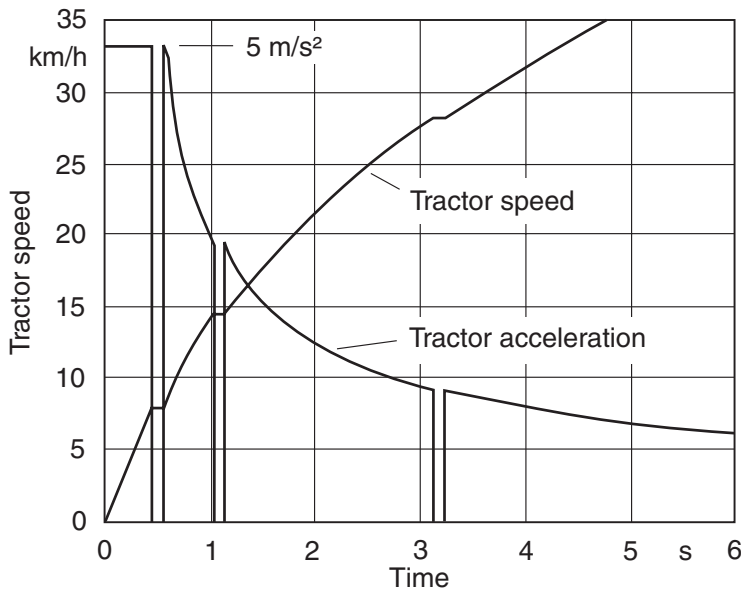**Fig. 37 Acceleration test of the Steyr "S-Matic" demonstrating range shift. 120 kW tractor on the road without trailer. Data courtesy Steyr.**

Fig. 38 Full load transmission efficiency of the Steyr S-Matic.

The transmission has been well accepted by the market and is meanwhile also used by the New Holland TVT tractor series (2004). McCormick has announced at EIMA 2004 to introduce it to a new tractor series VTX in 2005/2006.

The power split CVT works with an input coupled five shaft compound planetary, followed by two standard planetaries and a planetary for reverse, **Fig. 36**. Four ranges are used to keep the hydrostatic power level low. It varies between zero and 50 % (Lang et al. 1998). The hydrostatic unit is formed by swash plate units in back-to-back configuration (compare with Fig. 7) and can be dismounted/mounted easily at the transmission casing. The ranges are shifted at synchronous speeds using the dog clutches K1, K2, K3 and K4. The power is handed over without power interruption, similar to the Claas philosophy. The sophisticated function was optimised by simulation. The interruptions of acceleration could be made very short as demonstrated by **Fig. 37**.

Efficiencies of the S-Matic have been published by Leitner et al. (2000) excluding axles, **Fig. 38.**

If axle efficiencies are included (for example 95 % for 6-12 km/h), the result is little below the target of Fig. 11. According to Leitner et al (2000), efficiencies increase 2-3 % for reduced input speeds.

In order to solve noise problems, critical actual frequencies have been analyzed. If it was found, that they are amplified by resonance, a shifting up of the Eigen frequencies of the relevant
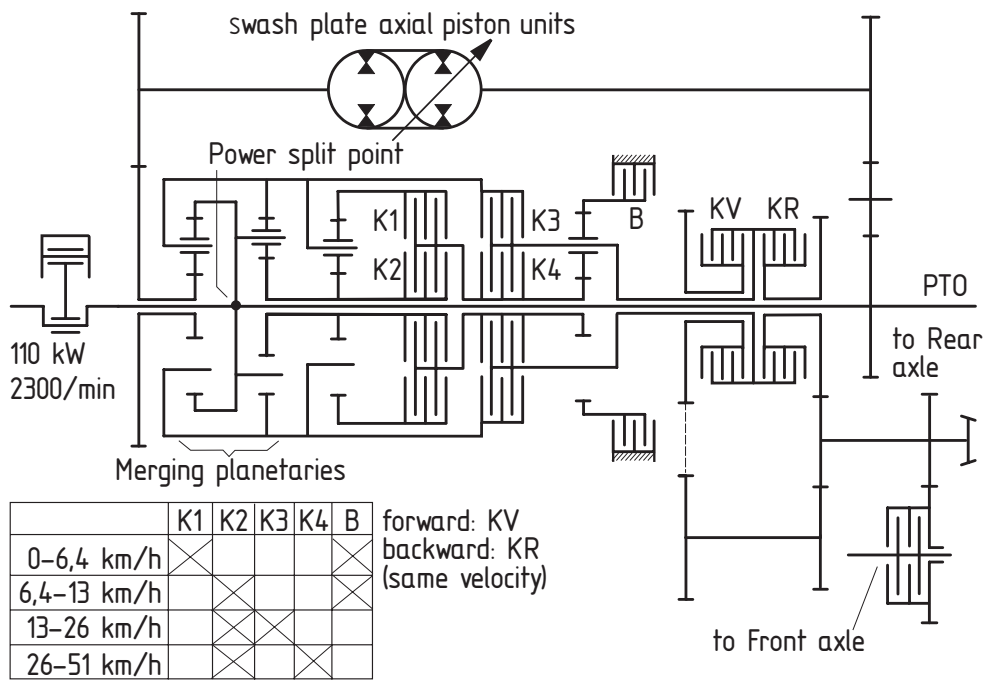
Fig. 39 Hydrostatic power split transmission ZF "Eccom" (2001).

housing sections was done by increased their stiffness. This method is very successful and can be generally recommended (Kirste 1989).

Load spectra on the basis of Fig. 13 and 14 and speed spectra on the basis of the CIGR Handbook III (Renius 1999) have been applied for dimensioning the S-Matic (Leitner et al. 2000).

The ZF "Eccom" for Deutz-Fahr, John Deere, Claas and others

Development of the ZF Eccom has been presented first by ZF/ZP at Agritechnica '97. It is at present realized with the common basic structure of **Fig. 39** by 3 transmission families (Pohlenz and Gruhle 2002). Eccom 1.5 (max. 110 kW) and Eccom 1.8 (max. 130 kW) are in production. A third version Eccom 3.0 (max about 220 kW) shall be available soon.

Deutz-Fahr was the first tractor company using the Eccom (version 1.5) in its new series "Agrotron TTV" (92/103/110 kW) starting a pilot series in 2000 and series production in 2001. John Deere applies the Eccom since 2001 for its 6000 tractor line advertising it as "AutoPowr". Claas has announced to use the Eccom 3.0 for its new Xerion carrier tractor family.

Also other tractor companies are considering to use the Eccom. About 12.000 units are in the market.

The four ranges concept allows it to keep the hydrostatic power level low. It varies between about plus 40, zero and minus 35 % of total power, **Fig. 40** (Pohlenz and Gruhle 2002) and uses – similar to Steyr - a compact hydrostatic package of swash plate units in back-to-back arrangement (compare with Fig. 7). The power split system works with an input coupled planetary. The variable path is starting at the split point via the PTO shaft arriving at the variator by the right three gear wheels. Fig. 40 shows well the linear output speed variation of the hydrostatic unit between minus 2300 and plus 2300 rpm crossing at zero the lockup
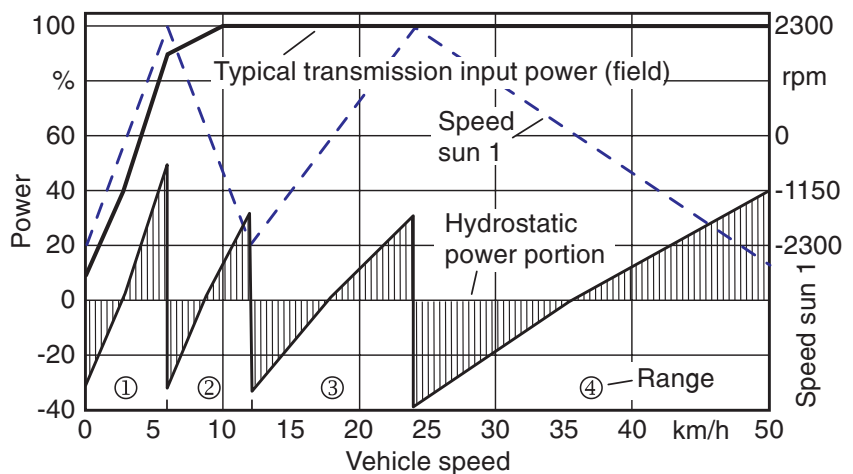
Fig. 40 ZF "Eccom": Hydrostatic power portions and speed of first left sun gear within the 4 ranges (Pohlenz and Gruhle 2002).

point. Three combined standard planetaries (two of them merging) form a five shaft compound planetary. This could produce in principle 3 ranges using friction clutches K1, K2 and K3. They are shifted at synchronous speeds. A forth standard planetary (K4, B) enables enlargement of the system to four ranges. Forward-reverse shift is realized by two conventional reverser clutches (KV, KR). Clutch dimensions are large due to the high possible slipping speeds but they offer a very powerful reversing.

Efficiencies of this transmission are published excluding the axles (Pohlenz and Gruhle 2002). If axles are included, they can cover the target of Fig. 11 near the lockup points. Noise reduction has been achieved by measurements and analytical methods.
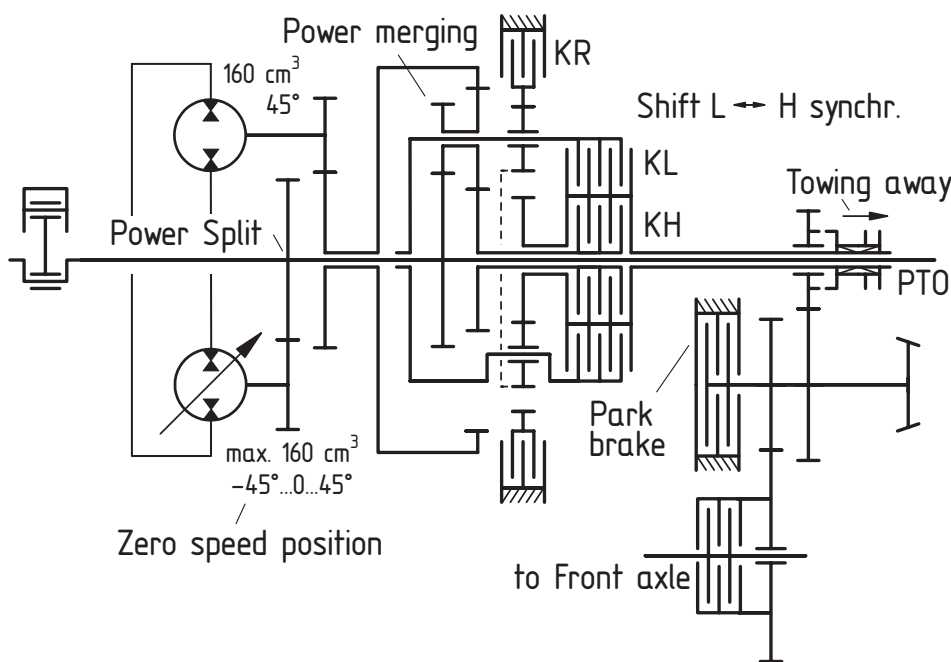
John Deere "Autopowr" (IVT-USA)

JD presented a newly designed transmission for its 7710 and 7810 tractors in 2001, **Fig. 41**. Rated engine power was 118 and 129 kW but the engines offered some extra power. The system works with two 45° bent-axis axial-piston units from Sauer-Danfoss (160 ccm/rev), similar in design to those which have been earlier installed in the Fendt "Vario" transmissions. However the transmission structure is completely different, as the concept is input-coupled and uses a 4-shaft compound planetary for the power split system (Mistry and Sparks 2002, Linares 2003). The engine drives both, the variable axial piston unit and the left sun gear of the merging compound planetary. Its ring gear speed is controlled by the fixed displacement hydrostatic unit (both directions). Its carrier output is used for range L (KL, max. 15 km/h) and its output of the right sun gear used for range H (KH, max. 50 km/h). Shift between both is done at synchronous speeds. The transmission is prepared for top speeds up to 62 km/h. An extra planetary for reverse (max. 17 km/h) uses the carrier output of the merging planetary (as for L), but reverses output direction by brake KR and generates output speed at its sun gear. The concept of the power shifted reverser is well known from the John Deere "PowrQuad" transmission (CIGR Handbook III, Renius 1999).

At start-up the ring gear turns in opposite direction of the engine speed. Here regeneration occurs as the variable unit acts as motor and the fixed unit as a pump. At the lock-up of the ring gear, in about the middle of the range, efficiency reaches its maximum value. When the ring gear speed has the same direction as the engine, the variable unit works as a pump and the fixed unit acts as motor. The transmission control is described in the paper of Mistry and Sparks (2002) explaining hydraulic circuit in detail and electrical circuit in concept.

Measured efficiencies are not known by the authors. But the 45° degrees units and a special oil level management provide favorable conditions for good values. The hydrostatic system pressure relief valves are set at 550 bar (compare with Fig. 20), but typical pressures under heavy load (field 6-8 km/h) are about 250 bar. Charge pump delivers 40 l/min at rated engine speed with flushing output of 24 l/min at 18 bar (Mistry and Sparks 2002).

Fig. 41 Hydrostatic power split transmission John Deere "AutoPowr" for John Deere 7010 (and later 7020) tractor series.
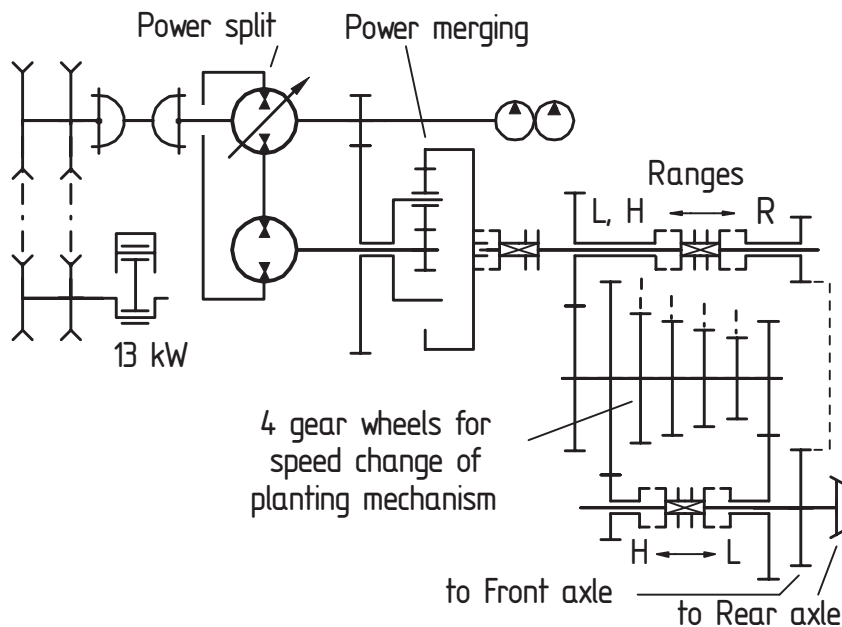
Fig. 42 Vehicle drive with power split CVT of Yanmar rice transplanter (2002).

EXAMPLE OF A POWER SPLIT CVT FOR A RICE TRANSPLANTER

Yanmar Agricultural Equipment Co., Ltd., Japan, produces since Febr. 2002 a self propelled rice transplanter in series, which uses a power split CVT for the vehicle drive. This is – as far as the authors know – the first series application of a power split CVT for the ground drive of a self propelled agricultural working machine and could become a trend also for other self propelled agricultural machinery. The Yanmar CVT was therefore included briefly in this paper. The information for developing the graphics of **Fig. 42** has been kindly provided directly by Yanmar, Japan.

The 13 kW engine drives both, the variable hydrostatic axial piston unit and the carrier of the standard planetary. Power split takes place within the variable unit, power merging in the planetary with the sun connected to the second hydrostatic unit. The system thus works with an input coupled planetary and delivers characteristics similar to that of Fig. 30. Power is directed from the planetary ring gear to a conventional gear box, which contains forward-reverse range, two forward ranges L, H, a speed shift for the planting mechanism and output shafts for rear and front axle. Measured efficiency curves are not available, but the system has been well accepted by the market according to information from Yanmar.

CONCLUSIONS AND OUTLOOK

Power split CVTs with hydrostatic variator units will continue to gain importance for tractors for countries with high tech farm mechanization, and in addition also for other mobile machinery such as self propelled agricultural working machines or construction machinery.

This trend is supported by new electronically controlled Diesel engines and electrohydraulic CVT control strategies and leads to improvements in productivity, precision of work, energy efficiency, environment protection and driver comfort. All these benefits are important, driver comfort is sometimes most underlined by customers.

The progress by CVTs influenced meanwhile also the development of stepped transmissions, mainly those with partly or full power shift. If the engine offers a high torque back up, the drive train gross traction hyperbola (Fig. 10) can be approached relatively well, if a certain engine speed variation is accepted. This is becoming a common strategy for pulling operations while there are some limitations in engine speed variation for PTO operations.

These improvements balance however a little the market trend to CVTs.

Hydrostatic units are in the foreground building power split CVTs for tractor drives. Two axial pis-

ton concepts are common, swash plate and bent axis units, as demonstrated by the examples. Swash plate pumps and motors are available in a large variety by many producers worldwide while variable wide angle bent axis units are by far not so common.

For tractors below about 100 kW, mechanical CVTs like the pull type PIV variator remain interesting because of their outstanding efficiencies. An active zero speed control is possible by a power split configuration for low speeds ("geared neutral") which could be combined with a "direct" mode and a reverser similar to the famous proposals of J.H. Kress (1962). Combinations of this kind are also in discussion for car transmissions with friction drive CVTs. Range change can be made very comfortable today due to model based control lay outs.

More attention than in the past should be paid in the future to infinitely variable drives also for the PTO in order to make its speed independent from the engine speed. The required band width of output speeds is much smaller than for the ground drive but efficiency must be extremely high, as tractors are sold by PTO power in several countries.

A broad discussion and some research has started on electrical units. They need a considerably higher speed level for adequate power density and first costs, but are quoted to have potential for tractor drives in power split systems.

The use of fuel cells is still far away in the future but could make one day an electrical drive without generator possible with high overall efficiencies. A close combination of an electrical motor with each individual tractor tire would offer a maximum flexibility but seems to be not realistic because of the wide band of speeds and torques for tractors (Fig. 10). The authors rather expect a combination of an electric motor with a range shift and final reduction drives. It could however be attractive to drive and control at least each axle independent to prevent circulating power and achieve better traction in turns (see, for example, the pull-turn-clutch of Fig. 23 and the research results of Grad and Brenninger at TU Munich).

The standard tractor will probably remain a key machine worldwide for further decades. One reason is, that about 90 % of all countries are on a lower mechanization level than the highly developed industrialized countries. The low developed areas may not need high tech solutions because of low labor costs and other reasons, but they may need reliable, simple tractors, adopted to local conditions.

Automation will however continue to penetrate the upper technology levels (Table 1). After the presentation of the first commercially available autonomous guidance system by GEO TEC (Bittner 2000, Freimann 2000) at Agritechnica 1999 (Gold Medal), research and development has been reinforced considerably on this subject. Master-slave systems as well as small autonomous vehicles (platforms) are under research for further steps (Schueller 2002). They all will need mechatronic drive systems, so that this business will remain important anyway. The authors see a certain shortage in skilled engineers in this field and would like to encourage young professionals to develop their skills in this fascinating field.

Acknowledgments

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