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Design Automation of Complex Hydromechanical Transmissions

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Abstract

This thesis proposes an automated methodology for the design of complex multiple-mode hydromechanical transmissions. High fuel prices and strict emission regulations are today drivers of the development of new fuel-efficient drive transmissions for construction machinery. Hydromechanical transmissions have high energy efficiency and a wide torque/speed conversion range. They are today strong candidates to replace the fuel-thirsty torque converters conventionally used in heavy construction machines. The trend towards more complex transmission architectures increases the need for more sophisticated product development methods. Complex multiple-mode transmissions are difficult to design and prototype and can be realised in a great number of different architectures. This increases the need for reliable concept evaluation in early design stages. The design of the transmission is also strongly coupled to its energy consumption and for a fair comparison between transmission concepts optimal designs are necessary.

Design automation and optimisation with detailed simulation models can support the industrial engineer in the design task and increase the available knowledge early in the design process. The proposed methodology uses simulation-based optimisation to design the transmission for a specific vehicle application. Various aspects of the transmission's characteristics may be targeted, although energy efficiency is in great focus in this work. To evaluate the energy efficiency, the transmission designs are simulated using backward-facing simulations with detailed power loss models. The methodology is applicable for designing the drive transmissions of construction machines and other mobile working vehicles such as agricultural machines, forest machines and mobile mining equipment.

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Linköping, October 2013
Karl Pettersson

Papers

The following papers make up the basis for the thesis and will be referred to by their roman numerals. In all three papers the first author is the main contributor and responsible for the work. The papers are printed in their original form with new formatting and minor changes in the nomenclature and spelling to maintain consistency throughout the thesis.

- [I] K. Pettersson and P. Krus. “Design Optimization of Complex Hydromechanical Transmissions”. In: *Journal of Mechanical Design* 135.9 (2013), p. 091005.
- [II] K. Pettersson and P. Krus. “Optimisation and Concept Sensitivity of Continuously Variable Hydromechanical Transmissions”. In: *The 8th International Conference on Fluid Power Transmission and Control, (ICFP13)*. Hangzhou, China, 2013.
- [III] K. Pettersson and P. Krus. “Modular Design of Hydromechanical Transmissions for Mobile Working Machines”. In: *The 13th Scandinavian International Conference on Fluid Power (SICFP2013)*. Linköping, Sweden, 2013.

The following papers are not included in the thesis but relate to the same topic and constitute an important part of the background.

- [IV] K. Pettersson, K.-E. Rydberg, and P. Krus. “Comparative Study of Multiple Mode Power Split Transmissions for Wheel Loaders”. In: *The 12th Scandinavian International Conference on Fluid Power (SICFP2011)*. Tampere, Finland, 2011.
- [V] K. Pettersson and P. Krus. “Optimering av komplexa hydraulmekaniska transmissioner för hjullastare”. In: *Hydraulikdagarna*. Linköping, Sweden, Apr. 2012.

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Abbreviations

CVT	Continuously Variable Transmission
HMT	Hydromechanical Transmission
HST	Hydrostatic Transmission
ICE	Internal Combustion Engine
ICPS	Input-Coupled Power-Split transmission
IVT	Infinitely Variable Transmission
KBE	Knowledge-Based Engineering
OCPS	Output-Coupled Power-Split transmission
TR	Theoretical Range

Nomenclature

C	Cost	$[-]$
C_m	Machine speed constant	$[mrev^{2/3}/s]$
D	Maximum displacement	$[m^3/rev]$
E	Energy	$[J]$
F	Force	$[N]$
N	Quantity	$[-]$
P	Power	$[W]$
Q	Hydraulic flow	$[m^3/s]$
R	Planetary gear ratio	$[-]$
T	Torque	$[Nm]$
X_{dp}	Design parameters	$[-]$
X_{sp}	System parameters	$[-]$
η_{gear}	Spur gear efficiency	$[-]$
η_{hm}	Hydromechanical efficiency	$[-]$
η_{pg}	Planetary gear efficiency	$[-]$
η_{vol}	Volumetric efficiency	$[-]$
λ	Weight factor	$[-]$
ω	Angular velocity	$[rad/s]$
ε	Relative displacement	$[-]$
i	Gear ratio	$[-]$
m	Number of modes	$[-]$
p	Pressure	$[Pa]$

r	Radius	$[m]$
t	Time	$[s]$
v	Linear velocity	$[m/s]$
w	Weight factor	$[-]$

1

Introduction

Hydromechanical Transmissions (HMTs) have great potential to increase drive performance and reduce fuel consumption of construction machinery. A Hydrostatic Transmission (HST) combined with a mechanical configuration can achieve a continuously variable speed ratio with high efficiency. By replacing the conventional hydrodynamic transmission with an HMT, the engine speed and the vehicle speed are decoupled, which allows for optimal engine management to reduce the fuel consumption further. By using multiple modes (gear shifts), a wider torque/speed conversion range with higher overall efficiency is reached with reasonably small displacement machines (hydraulic pump/motors). Both industry and academia have acknowledged this trend and the number of patents and publications on the subject have accelerated rapidly. In recent years, a few concepts have also been launched commercially. This thesis treats the design of complex multiple-mode HMTs to make the product development process more effective and increase knowledge in early design stages.

1.1 Background

Higher fuel prices and stricter emission regulations are drivers of development of new, fuel-efficient drive transmissions. In construction machinery, the hydrodynamic transmission (torque converter) is commonly used in heavier applications with high tractive force requirements. The main reasons for that are the robust design, high reliability, price and smooth behaviour for the operators. The torque converter is often found in series with a powershift gearbox to increase the torque/speed

range without loss of tractive force between gear shifts. The torque converter, however, has a limited range where the efficiency is acceptable for modern drive transmissions [1]. The focus is instead turned towards HMTs.

For multiple-mode HMTs there is a great variety of concepts due to the exponential increase in possible configurations for every additional mode. Complex HMTs contain large numbers of mechanical gears and components, which makes them difficult to design. Furthermore, the design space of each transmission concept also increases with the number of modes. The assessment of HMT concepts is consequently a complicated and difficult process which needs the support of computer aided tools. Some research has been done in modelling and simulating HMTs, mostly with the focus on energy efficiency. These techniques have, to some extent, been used to support evaluation of transmission concepts and to compare concepts. For a fair comparison, however, optimal designs are necessary for all considered concepts. Otherwise, a concept might be mistakenly underestimated and rejected. This risk becomes higher as the concepts become more complex, since an optimal design becomes more difficult to find. There is consequently a great need for methods for designing concepts in early product development stages. This can be achieved by using design automation and design optimisation.

Little work has been done on the design process of HMTs. In his dissertation, Erkkilä [2] proposed a design methodology where a preliminary design is iteratively redesigned by the engineer until the technical requirements are fulfilled. Volpe et al. [3] suggest an optimisation-based design process for power-split transmissions using kinematic models of the transmission. The objective was to avoid recirculative energy at the most frequently used operating points of the vehicle's working cycle. In this work, power loss models are not used; instead it is assumed that the operating range with the most recirculating power has the worst efficiency. Macor and Rosetti [4] also applied optimisation on the design of basic hydromechanical power-split concepts. The proposed design methodology uses more detailed simulation models of the components. The single objective was to minimise the consumed energy from a zero to a maximum speed acceleration and does not take into account the operating behaviour of the vehicle. It is highlighted that optimisation

can be used as an effective tool in the design process. In [5], the authors reformulate the problem into a multi-objective design optimisation by adding the total installed hydraulic displacement as another objective.

1.2 Design Automation

Design automation is a well-recognised field for more effective development of complex products. Although various definitions exist, it generally refers to the process of letting a system (computer) perform a design task in an automated way. The goal is to eliminate or at least minimise repetitive or non-creative design activities for the engineer [6]. This way, lead times and costs can be reduced in the product development process. Other possible advantages are fewer human errors in the designs and more customizable products. Figure 1.1 shows some of the problems with today's product design processes. The dilemma is that the knowledge that is gained throughout the process becomes more and more difficult and costly to apply to the product. The goal is to fully ex-

Figure 1.1 *Cost, available knowledge and design freedom during the design process, adapted from [6, 7].*

plore the possible designs in an early stage before making decisions with a high impact on cost. In this context, design optimisation with high fidelity models is a powerful tool. Knowledge-Based Engineering (KBE) is a collective term for methods and processes that automate or assist in the engineering task. KBE is today more common within geometry design automation by using parametric CAD models [6, 8]. For further details and definitions of design automation and KBE, see for instance [6, 7].

1.3 Contributions

This thesis proposes an automated design methodology for complex HMTs, with a special focus on the energy consumption of the transmission. Designing the transmission concept means, in this context, the process of sizing the components, such as gear ratios and displacement machines. The methodology is based on design optimisation of the transmission concept and it is particularly suitable for complex multiple-mode transmissions, which are difficult to design manually. A simulation environment is established based on previously known steady-state models and backward-facing simulations to evaluate the energy consumption of each transmission design. A methodology of designing modular transmissions for a range of applications is also presented and implemented. A transmission concept is, in this regard, designed with a fixed hydraulic variator and varying number of modes for each application.

2

Hydromechanical Transmissions

For vehicles with an Internal Combustion Engine (ICE) as the primary energy source, the engine power needs to be transformed into rotational mechanical power at the wheels of the vehicle. For construction machines, a steplessly variable speed ratio is often required with a wide torque/speed conversion range. For this reason, energy converters, such as hydraulic displacement machines (hydraulic pumps/motors), can be used that allow power transformation to be made steplessly in another physical domain. An HST with two hydraulic displacement machines is consequently a Continuously Variable Transmission (CVT) with an intermediate hydraulic power transformation. Modern hydraulic axial-piston machines often have the ability to be controlled to zero displacement, which makes the HST an Infinitely Variable Transmission (IVT) that has the ability to transform the input shaft speed into zero output shaft speed.

HMTs are transmissions that transfer power both hydraulically and mechanically, either in series, like an HST with mechanical gear steps, or in parallel. The parallel *power-split* architecture divides the power into a mechanical branch and a hydrostatic branch, which allows for a wide conversion range and high efficiency. Hydromechanical architectures can be divided into single-mode transmissions and multiple-mode transmissions. The single-mode transmissions are the basic hydromechanical configurations, either of power-split or hydrostatic type. Single-mode transmissions have no clutches, whereas multiple-mode

transmissions are always a combination of the basic configurations.

2.1 Performance Indicators and Requirements

One of the most important aspects of the transmission is the ability to transfer high power over a wide speed range. Based on this, two performance indicators can be identified; maximum power capacity and conversion range, see Fig. 2.1. The conversion range requirement can be

Figure 2.1 *Conversion range and maximum power capacity of a transmission.*

quantified by the Theoretical Range (TR) [1]. The TR-value is defined as the highest vehicle speed over the lowest vehicle speed where maximum power can be transferred. The TR-value has previously successfully been used to compare the capacities of different transmission concepts [9].

A key requirement of modern drive transmissions is energy efficiency, which has gained importance with stricter emission regulations and increased fuel prices. This requirement has in recent years been the main driver for the development of more advanced transmission concepts with higher efficiency. The main power losses in HMTs are from the hydraulic machines, which, in standard axial piston machines, have reduced efficiency in parts of the operating range.

An additional aspect of the transmission characteristics is the cost, both of development and manufacturing. The development costs can be hard to estimate and are partly related to the complexity of the concept. The manufacturing costs are also difficult to estimate in early design

stages, but by making reasonable assumptions at least a normalised comparison between concepts can be achieved [9]. This is possible by using statistical data for the production costs of the transmission elements, such as housing, bearings, gears and shafts [10]. The physical weight and volume of the gearbox are also aspects to consider, although weight is of less importance for ballasted applications. Additional performance indicators worth mentioning are the required control effort and the service life of the gearbox. The control effort can be difficult to quantify and will only be discussed briefly. The service life of the gearbox is difficult to estimate due to the strong dependency on the load spectra of the application. This aspect is not further discussed in this thesis, but could be included in a design methodology if suitable models are available.

2.2 Single-mode Transmissions

Single-mode HMTs can be classified into four main configurations, see Fig. 2.2. All configurations include a hydraulic variator (an HST) with

Figure 2.2 *The single-mode hydromechanical configurations.*

at least one variable displacement machine to achieve the IVT functionality. Kress [11] also identifies the differential transmission, first patented in 1907 by Renault [12], as a hydromechanical power-split configuration. This transmission configuration is not commonly used and represents the same functionality as an HST. The differential transmission is not further addressed in this work.

2.2.1 Hydrostatic Transmission

An HST, with or without a multiple-speed gearbox in series, can commonly be found in low power applications, where relatively small displacement machines are enough to meet the tractive force requirements. In recent years, however, hydrostatic drives have become more common also in larger applications. Figure 2.3 shows a simplified schematic of an HST with a variable pump and a variable motor in a closed circuit as traditionally seen in drive transmissions. During operation, one side acts as high pressure side and the other as low pressure side depending on the direction of the load. A charge pump is attached to the input shaft to supply oil to the low pressure side of the transmission. A flush valve ensures cooling and cleaning of the oil.

Figure 2.3 *Hydrostatic transmission with charge pump and flushing system.*

Design Example

This section presents a design example of the HST for a reference vehicle to show the functionality and characteristics of the transmission. Figure 2.4 shows a simplified schematic of the standard arrangement of an HST in a vehicle drive train. The pump shaft is directly connected to the engine and a final gear ratio, including a differential gear, is active between the motor shaft and the wheels. The transmission is traditionally sequentially controlled. From standstill, the pump displacement is increased while the motor displacement is at maximum. The vehicle speed is then ideally proportional to the pump displacement. When the pump reaches its maximum displacement, the motor displacement is decreased to increase the vehicle speed further, see eq. (2.1). Investigations have been made to optimise the control of the displacements to improve the energy efficiency of the transmission, see [10, 13].

$$v_{veh} = \frac{\varepsilon_1 D_1 \eta_{vol1} \eta_{vol2}}{\varepsilon_2 D_2} \omega_{ice} i_0 r_{tire} \quad (2.1)$$

Figure 2.4 The reference vehicle equipped with an HST and a final gear ratio, i_0 , including the differential to the wheels.

The reference vehicle is a small wheel loader which would today typically be equipped with a hydrostatic transmission and a power-shift gearbox. The vehicle parameters are given in Table 2.1. Table 2.2 shows the sys-

Table 2.1 The parameters of the reference vehicle.

Vehicle Parameter	Value
Maximum speed	30 km/h
Maximum tractive force	66 kN
Maximum tractive power	56 kW
Engine speed	1800 rpm

tem parameters of the transmission design, in which Unit 1 is an in-line machine and Unit 2 a bent-axis machine. The maximum system pressure difference is assumed to be 400 bar. Figure 2.5 shows a simulation

Table 2.2 Design example of an HST for a small wheel loader (not described in SI units).

Design Parameter	Value
Displacement D_1	91 cm ³ /rev
Displacement D_2	185 cm ³ /rev
Final Gear i_0	0.023

of an acceleration of the vehicle from zero to maximum speed during maximum tractive force. The transmission is designed to utilise the maximum pressure to overcome the maximum tractive force at standstill. The simulation is performed using the models described in chapter 3. The efficiency peaks when both displacements are at maximum and decreases at higher speeds due to the high speed and low relative dis-

Figure 2.5 Simulation of an HST design for a small wheel loader.

placement of Unit 2. Note that differently chosen design parameters would result in different simulation results.

Summary

The energy efficiency of the HST is favourable compared to a hydrodynamic transmission, especially for applications with high tractive forces and low speeds, such as wheel loaders [1]. As mentioned above, the HST normally has a larger torque/speed conversion range and can consequently be used, to a larger extent, without a multiple-speed gearbox for smaller applications. The displacement requirements of the machines become very high for high power classes since the motor must cover the corner power of the tractive force requirements. This is a limiting factor which necessitates the use of more advanced transmissions in larger applications.

2.2.2 Input-coupled Power-split Transmission

The term power-split refers to the parallel architecture of the transmission that allows power to be split up between a mechanical branch and a hydrostatic branch. The idea is to achieve a larger continuously variable conversion range by arranging the hydraulic variator together with a mechanical configuration. Since power is partially transferred mechanically, with low power transformation losses, the efficiency of a power-split transmission is normally also higher than a direct HST. The key component is the planetary gear that allows for the power-split

functionality.

The Input-Coupled Power-Split transmission (ICPS), also called *output-split* or *split-torque*, has the planetary gear connected to the output shaft of the transmission and the hydraulic variator coupled to the input shaft. Pohl [14] identifies six different configurations for the ICPS depending on how the planetary gear shafts are connected. All six configurations, however, do not have unique functionality. By switching the connection on the ring gear and the sun gear, the same functionality is achieved by inverting the planetary gear ratio according to $R' = \frac{1}{R}$, see [14, 15]. By eliminating the functionally equivalent configurations, three configurations remain, shown in Fig. 2.6 with the input shaft to the left and the output shaft to the right. They are here simply named configuration A-C and will be denoted ICPS A-C. Configuration A (ICPS A) is the

Figure 2.6 All possible input-coupled power-split configurations, with the input shaft to the left and output shaft to the right.

configuration traditionally referred to only as *input-coupled power-split*, see for instance [12, 16–18]. Configurations B and C are more often seen in multiple-mode transmissions.

The functionality of a power-split transmission is well understood by analysing the shaft speeds of the planetary gear by using lever analogy diagrams, introduced in [19]. Figure 2.7 shows lever diagrams for the planetary gear and the power flow of the ICPS A at different output speeds. Recirculating power is defined as positive when the power flows from left to right in the hydraulic variator and negative when the power flows from right to left. Recirculating power occurs because of the planetary gear configuration and affects the energy efficiency negatively, since power losses occur for all power transformations. The transmission has positive recirculating power for the reverse motion and negative recirculating power when the forward motion starts. Unit 1 then operates

Figure 2.7 Lever diagrams and illustration of the power flows for the ICPS A.

as motor and Unit 2 as pump. When the vehicle speed increases, the transmission reaches the full mechanical point which is the speed where all power flows through the mechanical branch. At this point the ring speed of the planetary gear, and consequently the speed of Unit 2, is zero. At higher vehicle speeds, the transmission operates with additive power flow.

Design Example

The considered concept is an ICPS A designed for the reference vehicle described in section 2.2.1. Figure 2.8 shows the drive line equipped with an ICPS A including two variable displacement machines, where Unit 1 is an in-line machine and Unit 2 a bent-axis machine. The vehicle speed

Figure 2.8 The reference vehicle equipped with an ICPS.

is given by eq. (2.2):

$$v_{veh} = \left(\frac{\varepsilon_1 D_1 \eta_{vol1}^k \eta_{vol2}^k}{\varepsilon_2 D_2} \frac{i_1 i_2}{1 - 1/R} + \frac{1}{1 - R} \right) \omega_{ice} i_0 r_{tire} \quad (2.2)$$

where $k = 1$ during positive variator power flow and $k = -1$ during negative variator power flow. Table 2.3 shows the system parameters for the transmission design and Fig. 2.9 shows the simulation results. A

Table 2.3 Design example of an ICPS for a small wheel loader.

Design Parameter	Value
Displacement D_1	$41 \text{ cm}^3/\text{rev}$
Displacement D_2	$118 \text{ cm}^3/\text{rev}$
Planetary Gear R	-1.5
Spur Gear i_1	-2.6
Spur Gear i_2	-0.67
Final Gear i_0	0.039

Figure 2.9 Simulation of an ICPS design for a small wheel loader.

high amount of recirculating power reduces the efficiency at low speeds even though power is partially transferred mechanically. The efficiency peaks at the full mechanical point where all power is transferred mechanically. At this point, the displacement of Unit 1, and consequently the speed of Unit 2, is zero. At higher speeds the transmission has additive power flow, where Unit 1 operates as pump and Unit 2 as motor. When Unit 1 reaches maximum displacement, the displacement of Unit 2 can

be decreased to increase the vehicle speed further. At higher speeds, more power is transferred hydraulically which causes the efficiency to decrease slightly. Another factor in the decrease in efficiency is the final gear, which has poor efficiency at high speeds.

Summary

The ICPS requires smaller displacement machines than an HST to accomplish the same conversion range and can consequently be realised in a relatively compact design [17]. The efficiency is also higher throughout the speed range even though there is a high amount of recirculating power at low speeds. The configuration is, however, more complex and contains more components than an HST. The control complexity is also an issue, since the transmission starts with recirculating power and an inverse relationship between the speed of the hydraulic variator and the vehicle, see [20]. The ICPS can also be realised with Unit 2 as a fixed displacement machine, which might be less costly than a variable machine but result in a lower conversion range.

2.2.3 Output-coupled Power-split Transmission

The Output-Coupled Power-Split transmission (OCPS), also called *input-split* or *split-speed*, has the planetary gear connected to the input shaft of the transmission. Figure 2.10 shows the functionally different configurations, which are simply achieved by turning the input-coupled power-split configurations around and setting the output to input and vice versa. Configuration A is the one traditionally denoted *output-*

Figure 2.10 All possible output-coupled power-split configurations.

coupled power-split, see for instance [12, 16–18]. Figure 2.11 shows lever diagrams of the planetary gear and power flows for the OCPS A. The forward motion starts with additive power flow, where a high amount of

Figure 2.11 Lever diagrams and illustration of the power flow for the OCPS A.

power is transferred through the hydraulic variator. During this phase, Unit 1 operates as pump and Unit 2 as motor and the total speed ratio is proportional to the speed ratio of the variator. At the full mechanical point, the speed ratio of the variator approaches infinity. This point is often designed to be the maximum speed of the vehicle to obtain a high efficiency throughout the speed range [17]. To further increase the vehicle speed, Unit 2 has to be controllable over-centre to achieve a negative speed ratio for the variator. During this phase, the transmission operates with negative recirculating power.

Design Example

Figure 2.12 shows the OCPS A in a driveline arrangement with a final gear ratio. The vehicle speed is given by eq. (2.3):

Figure 2.12 The reference vehicle equipped with an OCPS.

$$v_{veh} = \left(\frac{1 - R}{1 - \frac{\varepsilon_2 D_2 R}{\varepsilon_1 D_1 \eta_{vol1}^k \eta_{vol2}^k i_1 i_2}} \right) \omega_{ice} i_0 r_{tire} \quad (2.3)$$

Table 2.4 shows the system parameters of a transmission design for the reference vehicle in section 2.2.1. The displacement machines are sized

Table 2.4 Design example of an OCPS for a small wheel loader.

Design Parameter	Value
Displacement D_1	151 cm^3/rev
Displacement D_2	151 cm^3/rev
Planetary Gear R	-0.25
Spur Gear i_1	-0.64
Spur Gear i_2	-0.39
Final Gear i_0	0.0505

to meet the tractive force requirements during start where all power flows through the hydraulic variator. This makes the required machine sizes larger than for the ICPS. Both machines are of bent-axis design and the full mechanical point is at the maximum vehicle speed. Figure 2.13 shows simulation results of the transmission design. At low speeds,

Figure 2.13 Simulation of an OCPS design for a small wheel loader.

a high amount of power is transferred hydraulically and the total efficiency is consequently almost the same as the hydraulic efficiency. At the full mechanical point the efficiency peaks. At this point the variator speed ratio is zero since the speed of Unit 2 is zero. Both machines use their full operating speed range to reach the maximum vehicle speed.

The conversion range does not necessarily increase if Unit 2 can be controllable over-centre, since the maximum allowed speed decreases, see Fig. 3.3.

Summary

The OCPS is advantageous with respect to the control effort compared to the ICPS since the vehicle speed is proportional to the variator speed ratio at low speeds [17]. It is often stated that the efficiency of an OCPS is generally higher than for an ICPS and an HST over the entire speed range [16, 17]. This is, however, dependent on the considered transmission designs [5] and the required size of the charge pump. The displacement machines are generally larger than for an ICPS since all power flows through the hydraulic variator during start. For high-power applications, very large displacement machines are required, which results in high costs. This limits the range of suitable applications for the OCPS.

2.2.4 Compound Power-split Transmission

The compound power-split configuration is also called *variable bridge* and has two planetary gears. Both displacement machines are connected to a planetary gear shaft and the configuration is accordingly neither input- or output-coupled. It has been shown that a compound configuration can be achieved by replacing the hydraulic variator of one power-split configuration with another power-split configuration, see [21, 22]. All compound power-split configurations can consequently be built by combining the input-coupled power-split configurations in Fig. 2.6 and the output-coupled power-split transmissions in Fig. 2.10. Figure 2.14 shows an example of how a compound power-split configuration is formed by arranging an OCPS C with an ICPS A as variator. By combining all input-coupled and output-coupled power-split configurations, all possible compound configurations can be achieved. Compound power-split configurations are rarely seen in single-mode power-split transmissions due to the high amount of recirculating power. They are, however, more often seen in multiple-mode transmission, where two full mechanical points can be achieved in each mode [23]. Due to this, no design example is given for a compound power-split transmission.

Figure 2.14 *Functional equivalency of a compound power split configuration.*

2.3 Multiple-mode Transmissions

A multiple-mode HMT combines several basic HMTs by using clutches. Each mode consists of one of the single-mode configuration described above. By switching between the modes, the total speed ratio is increased and a wide conversion range with high efficiency is achieved. Often only parts of the configuration ranges are used in a multiple-mode concept. This way, the best features of each configuration can be utilised. The number of possible multiple-mode configurations is infinite and rapidly increases with the number of modes. The multiple-mode transmission allows for a wide conversion range without unreasonably large displacement machines. By using multiple-mode transmissions, the use of HMTs is consequently also possible for larger applications.

2.3.1 State-of-the-Art

A great number of concepts have been patented over the years and some also realised commercially. The two-motor transmission, presented in the 90s [24], was an attempt to increase the use of hydrostatics in higher power classes. The principle consists of two HST modes where the second motor is decoupled for the second mode. During shifting, the shafts are synchronised to avoid loss of tractive force. The concept is further studied in [9] and can today be found in some applications of relatively high power [25, 26]. Starting in the late 90s, several manufacturers released multiple-mode power-split concepts for agricultural tractors [27–29]. The new transmission concepts were well accepted by the market and multiple-mode HMTs are today state-of-the-art. See [12] for an overview of some of the concepts developed for agricultural applications. During recent years, a few commercial multiple-mode power-split

concepts have been developed for construction machinery, see [30–32].

2.3.2 Functionality

Figure 2.15 shows a common design principle of a general multiple-mode HMT with two output shafts from the gear configuration. During mode shifts, the two shaft speeds are synchronised to avoid loss of tractive force. One mode increases the vehicle speed by increasing the HST speed ratio while the next increases the vehicle speed by decreasing the variator speed ratio. This functionality is possible with power-split configurations and allows for an arbitrary number of modes. This makes it possible to adapt the number of modes to match the tractive force requirements of different power classes, see [III]. Other design principles

Figure 2.15 A general representation of a multiple-mode HMT with an arbitrary number of modes.

exist, with additional output shafts [27] and clutches within the gear configuration [33]. Common to all multiple-mode HMTs, however, is the functionality achieved by combining the single-mode configurations described in section 2.2. Each multiple-mode transmission architecture can consequently be described by listing the individual mode configurations. The detailed functionality, however, is not fixed since the complete range of a configuration is not necessarily used. The actual gear

configuration may also look different since several modes can share mechanical gears to minimise the number of components. The magnitude of the design space for multiple-mode concepts quickly becomes very large with an increasing number of modes, although all combinations are not functionally meaningful.

2.3.3 The Jarchow Concept

The Jarchow concept is a multiple-mode HMT adapted from the patent by Friedrich Jarchow of Bochum University [34]. The concept forms the base of several known HMTs and has a compact design with relatively few mechanical gears. The concept is shown in Fig. 2.16 and consists of the patented transmission architecture plus a hydrostatic starting mode. The concept is shown with an arbitrary number of modes and follows the principle in Fig. 2.15. The figure only shows the principle of the concept,

Figure 2.16 The Jarchow concept with an arbitrary number of modes, m . In the figure, m is an odd number. The concept can also be realised with a variable Unit 2.

the physical gearbox may, however, have a different gear arrangement. The first forward mode is purely hydrostatic and is followed by an arbitrary number of modes.

trary number of input-coupled power-split modes for the forward motion. The first power-split mode is input-coupled configuration A (ICPS A) whereas the second power-split mode is configuration C (ICPS C). The backward motion follows the same pattern and starts with the hydrostatic mode with Unit 1 at negative relative displacement. The clutch configuration is shown in Table 2.5. The number of modes, m , is defined as the number of forward modes plus the hydrostatic mode. Three modes, $m = 3$, means one hydrostatic mode and two power-split modes for the forward motion. There are, however, additional $m - 1$ power-split modes for the reverse motion. Figure 2.17 shows lever diagrams for the

Table 2.5 *Clutch arrangement*

Mode	S_{fwd}	S_{rev}	S_0	S_1	S_2	S_{m-1}
R $m-1$		•				•
R2		•			•	
R1		•		•		
H			•	•		
F1	•			•		
F2	•				•	
F $m-1$	•					•

power-split modes. The figure shows both planetary gears with different gear ratios for the odd (ICPS A) and even (ICPS C) power-split modes. Shaft I and shaft II are synchronised during each mode shift to avoid loss of tractive force.

Design Example

Table 2.6 lists the system parameters of a design example of a three-mode transmission ($m = 3$) for the reference vehicle described in section 2.2.1. Unit 1 is an in-line machine controllable over-centre and Unit 2 is a fixed bent-axis machine. All gear ratios without notation are considered to be -1. Figure 2.18 shows simulation results for the transmission design. The required conversion range of the reference vehicle is achieved with very small displacement machines since several modes are used. The efficiency is also high throughout the speed range since a low fraction of power is transferred hydraulically. The efficiency peaks at the full mechanical points of the power-split modes, but remains high throughout the speed range.

Figure 2.17 Lever diagrams for the power-split modes of the Jarchow concept. The first row shows modes $F1$, $F3$, $F5$, etc and the second row shows modes $F2$, $F4$, $F6$, etc.

Table 2.6 System parameters of the simulated transmission design.

System Parameter	Value
Displacement D_1	$37\text{ cm}^3/\text{rev}$
Displacement D_2	$29\text{ cm}^3/\text{rev}$
Planetary Gear R_1	-2.6
Planetary Gear R_2	-2.1
Spur Gear i_1	-2.7
Spur Gear i_2	-0.31
Spur Gear $i_{s,1}$	-0.53
Spur Gear $i_{s,2}$	-0.53
Final Gear i_0	0.067

Summary

A multiple-mode HMT allows for a wide conversion range with small displacement machines and high overall efficiency. The efficiency increases with the number of modes up to the point where the mechanical gear losses become dominant. The disadvantages are the complexity of the gearbox, which relates to its robustness. A complex transmission con-

Figure 2.18 Simulation of a transmission design of the Jarchow concept for a small wheel loader.

cept also contains a higher number of components which makes the gearbox bulkier, heavier and more expensive. These disadvantages increase with the number of modes, since additional components are required. A high control effort is also needed, in particular for the synchronisation of shaft speeds during mode shifts [9].

3

Modelling and Simulation

This chapter describes the simulation environment used in the proposed design methodology to simulate the energy efficiency of a transmission design. The component models are based on known steady-state relations and power loss models from previous research. The research on modelling and simulation of HMTs has, to a large extent, focused on energy efficiency with the aim of supporting the evaluation of different transmission concepts. Mikeska et al. [35] developed such a simulation environment for power-split drives as a toolbox in MATLAB Simulink. The software is used in several studies for evaluation of transmission concepts or development of control algorithms [16, 17, 36]. A similar simulation approach is later made by Casoli et. al. [37]. Erkkilä [2] also developed a software for design and analysis of HMTs by using relatively simple power loss models. Kohmäscher [38] did extensive work on developing simulation models for HMTs. In contrast to Erkkilä's work, the necessity to use very detailed loss models is underlined, both for displacement machines, gears, clutches and other losses in the transmission.

The models described in the sections below are used in the simulation of the transmission's energy consumption, which is the primary focus in the methodology. It is important to reflect on the necessary accuracy and detail level of the loss models. The goal of this work is to present a design methodology and not to further develop simulation models of HMTs. Nevertheless, the simulation models need to be sufficiently accurate to give validity to the design choice. The aim of the

methodology is to achieve an optimal design for a transmission concept. It is not crucial to predict the exact energy consumption of a transmission concept. It is, however, important to model the losses that would differ from one transmission design to another. Some power losses may also differ between different designs, but are very small and for this reason would not be important to model. The necessary detail level of the loss models can be identified by sensitivity analysis, as shown in [II].

3.1 Backward-facing Simulation

The transmission simulations are performed by using backward-facing simulation of the driveline. A backward-facing simulation assumes that a prescribed vehicle motion cycle is carried out and sequentially tells each component how much it must perform to fulfil that cycle. The calculations are consequently made backwards, starting at the vehicle wheels, through the complete drivetrain. Each component's power losses are calculated using loss models and the total power losses are then calculated as the sum of all component losses. The backward-facing simulation is convenient for evaluating transmission performance and energy consumption, since the vehicle cycle is exactly the same for all transmission designs [39]. Another advantage is that no driver model is needed and the simulation execution time is very short [40], making it ideal for optimisation-based design, see also [41]. As an example, Fig. 3.1 shows a block diagram of the simulation model of the ICPS A from Fig. 2.8. The simulations are implemented with a fixed step solver in MATLAB Simulink or simply by m-scripts in MATLAB.

Figure 3.1 Principle of the backward-facing simulation of an ICPS A (shown in Fig. 2.8). The simulation does not need to manually adjust the control of the displacement machines for different transmission designs.

For components which have power losses dependent on the operating conditions, an algebraic loop is formed. One example is a displacement machine where the system pressure, p , is dependent on the torque losses, which in turn is dependent on the system pressure, see section 3.2. Since the equation system is hard to derive analytically, it is solved iteratively, see also [2].

3.2 Hydraulic Displacement Machines

The steady-state behaviour of the displacement machines follows eq. (3.1a) and (3.1b):

$$T = \frac{\varepsilon D \Delta p}{2\pi} \eta_{hm}^i \quad (3.1a)$$

$$Q = \frac{\varepsilon D \omega}{2\pi \eta_{vol}^i} \quad (3.1b)$$

where

$$i = \begin{cases} 1 & \text{for motoring mode} \\ -1 & \text{for pumping mode} \end{cases}$$

3.2.1 Efficiency

The power losses of the displacement machines often constitute the major part of the losses in hydromechanical transmissions and change depending on operating conditions. They must therefore be modelled very accurately. The efficiencies are here expressed as polynomials of the relative displacement ε , the hydraulic pressure Δp and the speed n according to eq. (3.2a) and (3.2b).

$$\eta_{hm} = \sum_{u=0}^p \sum_{y=0}^q \sum_{z=0}^r C_{hm,uyz} \varepsilon^u \Delta p^y \omega^z \quad (3.2a)$$

$$\eta_{vol} = \sum_{u=0}^p \sum_{y=0}^q \sum_{z=0}^r C_{vol,uyz} \varepsilon^u \Delta p^y \omega^z \quad (3.2b)$$

The polynomials are matched to fit a series of measurement points, using linear regression. The efficiency maps are then considered to be constant

for all machine sizes. This method is known as POLYMOD [42] and is today frequently used for modelling power losses in hydraulic displacement machines. The reason for adapting polynomials to match measurement data, instead of interpolating/extrapolating the values, is to capture a physically more realistic behaviour at the extremes. See [38] for more information on using mathematical loss models and an overview of previously developed models. Figure 3.2 shows examples of efficiency maps adapted to a series of measurement data supplied by a manufacturer.

Figure 3.2 Efficiency maps for a displacement machine of in-line design at full displacement.

3.2.2 Operating Range

The operating range of the displacement machines is limited in terms of pressure and rotational speed. These limits highly influence the required size of the displacement machines in a transmission concept. The maximum operating range is here investigated by examining data sheets of standard hydraulic machines.

Maximum Pressure

Although varying slightly between manufacturers, designs and series, the maximum pressure of standard displacement machines seems to be constant for all sizes of the machines. The maximum pressure of all displacement machines is here set to $p_{max} = 425 \text{ bar}$.

Maximum Speed

The maximum speed of a displacement machine is coupled to the size of the machine. Sannelius [9] showed that the maximum speed of a hydraulic motor is proportional to $D^{-1/3}$. This relationship is here expanded to include the overspeed factor β , according to eq. (3.3).

$$\omega_{max} = 2\pi C_m \beta D^{-1/3} \quad (3.3)$$

where C_m is a parameter considered constant for a geometrically uniform machine and β is the overspeed factor. Table 3.1 shows the values for C_m and β for different machine designs. These are average values from

Table 3.1 Machine Speed Constants and overspeed factors for different machine designs.

Machine Design	C_m	β
Fixed bent-axis	3.0	1.0
Variable bent-axis	2.8	$f(\varepsilon)$ (see Fig. 3.3)
Variable in-line	2.7	1.0

data sheets of standard displacement machines and are confirmed to some extent in [9]. The overspeed factor, β , varies according to Fig. 3.3 for the variable bent-axis design [4, 43].

Figure 3.3 Overspeed factor for variable bent-axis machines

3.3 Charge Pump

Standard hydrostatic transmissions include a charge pump for cooling, anti-cavitation and to supply pressure to the control circuit. This results

in a constant power loss proportional to the size of the charge pump. The charge pump is often mounted in the same housing as the transmission pump and rotates at the same speed. For power-split transmission, the charge pump is considered to rotate with the maximum speed of the pump. The size of the charge pump needs to be adjusted with respect to the size of the displacement machines to cover the leakage flow. From studying data sheets of standard transmission pumps, it is concluded that the size of the charge pump is directly proportional to the transmission pump size, according to eq. (3.4).

$$D_{cp} = C_{cp}D_p \quad (3.4)$$

Figure 3.4 shows the size of the charge pump for the Bosch Rexroth transmission pumps A4VG/32 and A4VG/40 [44] and a trendline for $C_{cp} = 0.22$ that is used in this work. The charge pump pressure is

Figure 3.4 Size of the charge pumps for Rexroth transmission pumps and a trendline for 22% of the pump displacement. Data taken from [44].

considered to be constant, $p_{cp} = 25 \text{ bar}$. All power required to drive the charge pump is lost and will therefore decrease the total transmission efficiency. This is an important power loss to consider since it will punish transmission designs with large displacement machines.

3.4 Spur Gears

The steady-state equations of a simple mechanical spur gear (Fig. 3.5) are:

$$\omega_{out} = i\omega_{in} \quad (3.5a)$$

$$T_{out} = -\frac{T_{in}\eta_{gear}}{i} \quad (3.5b)$$

where $i < 0$ is the gear ratio and η_{gear} is the mechanical efficiency responsible for the torque losses of the spur gear.

3.4.1 Efficiency

The power losses from the mechanical gears in a hydromechanical transmission often constitute a small part of the total losses, which motivates the use of a simple friction model. Conveniently, the losses are often divided between load-dependent and load-independent losses. The load-dependent losses include sliding and rolling friction between the gear teeth and friction losses in the bearings. The load-independent losses are often speed-dependent and include churning losses caused by the gear wheels when rotating in an oil bath and losses in sliding bearings. Previous studies have used different levels of model complexity for simulation of hydromechanical transmissions. Kohmäscher [38] used an advanced loss model for the spur gear efficiency, which took into account oil viscosity, the size of the gear wheels, spur gear assembly and other effects. Other studies assume constant efficiency or even ideal spur gears to conclude behaviours and characteristics of different transmission concepts, e.g. [2, 4].

For this work, a semi-empirical model is used for the power losses in a spur gear. The model has been validated in test rig experiments [45] and is partially supported by previously known analytical models [46]. The load-dependent losses also include losses from the bearings and are assumed to be 0.5% of the input torque. The speed-dependent torque losses are modelled as a linear speed dependency and the total output power is hence:

$$P_{out} = P_{in} - P_{loss} = 0.995P_{in} - (C_1n_{in} + C_2n_{in}^2) \quad (3.6)$$

The resulting efficiency map is shown in Fig. 3.6. Large gear ratios,

Figure 3.5 *Spur gear*

Figure 3.6 *Efficiency of the spur gear*

however, are hard to accomplish with only one gear pair. It is therefore assumed that the maximum gear ratio for one spur gear is limited and two or more spur gear stages are needed. The power loss for the total gear ratio is hence:

$$P_{out,tot} = P_{in} - \kappa P_{loss} \text{ with } \kappa = \left\lceil \frac{i_{tot}}{i_{max}} \right\rceil \quad (3.7)$$

where i_{max} is the maximum allowed gear ratio for one spur gear pair. It is reasonable to take the effect into account since different designs will result in widely different gear ratios. A similar assumption is also made in [4, 5].

3.4.2 Operating Range

The operating range of the spur gears is not limited by size in the same way as the displacement machines. The spur gears' operating range is therefore considered to be infinite.

3.5 Planetary Gears

The kinematics of the planetary gear (Fig. 3.7) follow eq. (3.8)

$$R = \frac{\omega_{sun} - \omega_{carr}}{\omega_{ring} - \omega_{carr}} \quad (3.8)$$

where $R < 0$ is the planetary gear ratio. The torque equilibrium follows eq. (3.9):

$$T_{sun} + T_{carr} + T_{ring} = 0 \quad (3.9a)$$

$$T_{ring} = -T_{sun} R \eta_{pg}^i \quad (3.9b)$$

where

$$i = \begin{cases} 1 & \text{if } T_{sun}(\omega_{sun} - \omega_{carr}) > 0 \\ -1 & \text{if } T_{sun}(\omega_{sun} - \omega_{carr}) < 0 \end{cases}$$

Figure 3.7 Planetary gear

3.5.1 Efficiency

The efficiency of the planetary gear is here considered to be constant, $\eta_{pg} = 0.99$. The same assumption is also made in [4, 35]. For an overview of algebraic efficiency models of planetary gears, see for instance [47].

3.5.2 Operating Range

The operating range of the planetary gear is assumed to be infinite.

3.6 Clutches and Bearings

Clutches and bearings are also a possible source of power losses in the transmission. Losses from bearings are included in the power loss models for the spur gears. Power losses from clutches depend on the type of clutch that is used. Speed-dependent losses from disengaged wet clutches can be a significant power loss, whereas dog clutches are loss-free. Theoretically, dog-clutches can be used since the shafts are synchronized during each mode shift. Dog clutches can, however, be hard to use from a control perspective, see for instance [9]. For this study, no clutch losses are included in the simulations.

4

Design Methodology

This chapter describes the proposed design process on a high level, whereas chapter 5 shows an implementation of the methodology on the previously described Jarchow-concept.

4.1 Concept Design

The proposed concept design process is applicable to any HMT concept and can be used to optimise the design of a transmission for a specific vehicle application. To compare different architectures, this process needs to be performed for all considered concepts. A transmission concept may, however, have a non-fixed number of modes, as described in section 2.3. Figure 4.1 shows the flow chart for the design methodology.

4.1.1 Vehicle Requirements and Typical Operating Behaviour

Tractive Force Requirements and Maximum Speed

The inputs to the transmission design process are the requirements of the transmission for the considered vehicle. These normally include the desired maximum speed of the vehicle and how the required tractive force varies with the vehicle speed. It is consequently necessary to identify the desired performance of the vehicle in an early design stage. By doing so, it is easier to compare the characteristics of the different designs,

Figure 4.1 *The proposed design methodology.*

since they all have the same performance. It is also an advantage when comparing transmissions of different architectures.

Typical Operating Behaviour

To evaluate the energy consumption of a transmission design, pre-recorded data from typical operating cycles are used, i.e. the operating points which are used most frequently. By simulating the operating

cycles, the energy consumption of a transmission design can be evaluated. This is a common way of evaluating system concepts and allows for an optimised system design specifically for the typical operating behaviour of the vehicle [41]. Note the difference between the tractive force requirements and the typical operating behaviour of the vehicle. For instance, a wheel loader is rarely operated at its maximum speed, although the drivetrain must be able to reach it. The typical operating behaviour is usually defined with data series from representative working cycles measured in real-world experiments. For some applications, standardised cycles are used in both industry and academia to evaluate the characteristics of drivetrains.

4.1.2 Explicit Design Relations

The explicit design relations translate the design parameters into system parameters, i.e. gear ratios and size of displacement machines. The design relations are identified with physical relationships between the components, the vehicle parameters and the required tractive force data. These equations are used to size the components in order to ensure the functionality and performance of the transmission. For multiple-mode HMTs, which require uninterrupted traction force during mode shifts, it is necessary to synchronise the speeds of the active output shafts. Technical limitations on the components are also important considerations. Such specifications are often provided by the manufacturers and define the allowed operating range of the product. The most critical components are usually the displacement machines with a fixed maximum speed and load torque for a certain size. Once all design relations have been set up, an equation system can be formed where the design parameters are included.

4.1.3 Equation System and Degrees of Freedom

By analysing the formed equation system, it is possible to identify the degrees of freedom of the system (by subtracting the number of equations from the number of system parameters). The degrees of freedom of the equation system will also be the number of design parameters for the optimisation. To arrive at an automated design, the equations need to be rearranged in order to achieve expressions of the system parameters as functions of the vehicle parameters and the design parameters. At this point it is possible to form an automated design of the transmission by

manually choosing the design parameters. To achieve an optimal design, the design equations are instead used to set the model parameters in an optimisation algorithm.

4.1.4 Simulation-based Optimisation

This step follows the process described in [48] and shown in Fig. 4.2. The objective function of the design optimisation should reflect the de-

Figure 4.2 *Simulation-Based Design Optimisation according to Krus [48]*

sired characteristics of the final gearbox. The minimisation of consumed energy is central to this methodology, but the objective function may also include other aspects, such as the ones discussed in section 2.1. Predicting such properties may be hard at an early stage, but can still be of help in the design process [10]. Macor et al. [4] showed that a design optimisation of single-mode power-split HMTs with energy efficiency as the only objective can lead to significant oversizing of the displacement machines. A multi-objective optimisation may instead be more suitable for the industrial engineer by modelling the negative aspects of using many components and large displacement machines, such as cost, size, weight, etc.

The optimisation algorithm is simply used as a tool to explore the design space of the transmission and cleverly choose values of the design parameters. The algorithm may be of any non-gradient based method suitable for simulation-based optimisation. After choosing values of the design parameters, they are translated into system parameters. Simulation of the system model results in some gearbox characteristics which are then fed back and evaluated with respect to the objectives.

4.2 Modular Design

The concept of modularity and platform design within product development is a well-recognised method to decrease the variety of components in a product family. The main advantages include lower manufacturing costs and lower development costs due to the increased communality of the products. The trade-off for the company lies instead in the decreased individual performance of the products [6, 49]. When dealing with multiple-mode hydromechanical transmissions, much can be gained from using similar concepts for many applications, including different power classes of the same type of vehicle. The development costs are significantly reduced if the gearbox can be scaled up and down or at least easily adapted to each application. Additionally, manufacturing costs can be reduced by producing or buying larger quantities of the same component.

The tractive force requirements and the demands of the gearbox characteristics naturally vary between applications. The demand for low cost and high energy efficiency may vary greatly also within different sizes of the same type of vehicle. One approach to achieve a modular gearbox is to use the same number of modes in each transmission and to vary the sizes of the displacement machines to match the tractive force requirements of each application, as shown in [30]. Another approach is to use the same hydraulic variator, i.e. the same sizes of displacement machines but with different numbers of modes depending on tractive power demands. The advantages include lower manufacturing costs for the gearboxes since the variator can be bought or manufactured in larger series. The development costs may also be reduced and robustness increased if similar control software can be used for all transmissions. This is the design approach further examined in this paper. The trade-off for this scenario is that transmissions with a small hydraulic variator, leading to lower power losses, require a greater number of modes, leading to higher costs.

A design methodology is here presented for the design of a modular gearbox suitable for a range of applications. The modularity is confined to the use of the same transmission concept and a collective hydraulic variator, but with a varying number of modes. The choice of hydraulic variator and number of modes for each application should consequently

also be the subject of a design optimisation. Figure 5 shows the proposed methodology for the design process. The local optimisation loop optimally designs each application one by one, with a given number of modes and a given variator size. This design process follows the methodology described in section 4.1 and may be a single- or multi-objective optimisation. The global optimisation loop sets the optimal size of the variator and the number of modes for each application. Here, the derived characteristics of each transmission need to be weighted in the objective function according to the importance of each application. These weight factors relate to the number of sold vehicles, cost margins, fuel price, customer demands, acceptable payback period, etc.

Figure 4.3 The design process for the modular gearbox. The inner loop optimises the transmission designs for all applications individually given a number of modes and a variator size. The outer loop sets the number of modes and variator size optimally for the considered range of applications.

5

Application Example: The Jarchow Concept

This chapter applies the proposed design methodology on the Jarchow concept. The number of modes is not fixed since this design choice is also supported by the methodology. A concept design is made for a 31-tonne wheel loader, which would typically be equipped with a torque converter and a powershift gearbox. A wheel loader of this size is a suitable application for a more complex driveline, since the high tractive force requirements make simple HMT concepts insufficient. The wheel loader is a targeted application for HMTs, much because hydrodynamic transmissions have such poor efficiency in the low speed range.

The concept is also used in the design of a modular gearbox for three wheel loader applications, to demonstrate the methodology described in section 4.2.

5.1 Concept Design

This section follows the design methodology described in section 4.1 for the considered design task. The transmission concept is shown in Fig. 5.1 with Unit 2 as a fixed displacement machine. All gears without notation are considered to be -1 since they add no additional functionality to the transmission.

Figure 5.1 The Jarchow concept with a fixed Unit 2.

5.1.1 Vehicle Requirements and Typical Operating Behaviour

Tractive Force Requirements

Figure 5.2 shows the tractive force requirements and Table 5.1 lists the parameters of the considered wheel loader. The engine speed is assumed

Figure 5.2 The tractive force requirements of the wheel loader with the dashed line as the tractive power.

to be constant and the transmission itself must hence be capable of covering the complete vehicle speed range from 0 to v_{max} .

Table 5.1 The parameters of the reference vehicle (not described in SI units).

Quantity	Description	Value
v_{max}	Maximum speed	40 km/h
F_{max}	Maximum drag force	240 kN
P_{max}	Maximum load power	205 kW
r_{tire}	Tire radius	0.884 m
ω_{ICE}	Engine speed	1800 rpm

Typical Operating Behaviour

Wheel loaders are versatile machines and have many different application areas. Many are sold as multi-purpose machines although operating with a bucket on a production site is the most common task [50]. Two operating cycles are often used to characterise the operating behaviour of the wheel loader: the short loading cycle (also called the Y-cycle or the V-cycle) and the load-carry cycle (also called the long loading cycle). Briefly explained, in the short loading cycle, the wheel loader repetitively approaches and fills the bucket from a gravel pile and then reverses and unloads the gravel into a load receiver. The load-carry cycle is a longer cycle where the unloading takes place farther away from the gravel pile. Figure 5.3 shows the principle of the two operating cycles. For further details see [51]. Figure 5.4 shows corresponding bubble plots of the two cycles recorded in real world operation. A large bubble represents a point where the wheel loader operates frequently. The maximum speed of the short loading cycle is around 8-10 km/h and the bucket fill operation mostly takes place at under 5 km/h [30]. In the load-carry cycle, the

Figure 5.3 Typical operating cycles of the wheel loader, from [51].

Figure 5.4 Bubble plots of the typical operating cycles.

wheel loader reaches higher vehicle speeds during the transportation phase. To achieve an operating behaviour representative of the wheel loader's life-cycle, each loading cycle is weighted according to how often it is used. The complete operating behaviour of the wheel loader is therefore achieved by weighting each cycle according to its significance. For the considered wheel loader, the weighting factors in Table 5.2 are used.

Table 5.2 Estimated time distribution of the operating cycles of the wheel loader.

Short loading cycle	35%
Load-carry cycle	25%
Idling	40%

5.1.2 Explicit Design Relations

This section derives the kinematic and torque relations of the transmission concept. The output speed and torque acting on the shafts during each mode shift are defined as:

$$\omega_{shift,k} = \frac{v_{shift,k}}{r_{tire}i_0} \text{ for } k = 1, 2, 3, \dots, m-1 \quad (5.1a)$$

$$T_{shift,k} = F_{shift,k} r_{tire} i_0 \text{ for } k = 1, 2, 3, \dots, m-1 \quad (5.1b)$$

The maximum output speed must be reached by the last mode, which makes it possible to treat the maximum speed as the last mode shift:

$$\omega_{shift,m} = \frac{v_{max}}{r_{tire}i_0} \quad (5.2)$$

For each mode shift, the speed of shafts I and II must be synchronised. The output speeds of the even mode shifts (shifting to Shaft I in Fig. 2.16) are:

Shaft I:

$$\omega_{shift,k} = \omega_{ICE} i_{s,k} \text{ for } k = 2, 4, 6, \dots \quad (5.3a)$$

Shaft II:

$$\omega_{shift,k} = \omega_{ICE} i_{s,k-1} \text{ for } k = 2, 4, 6, \dots \quad (5.3b)$$

The output speeds of the odd mode shifts (shifting to Shaft II in Fig. 2.16) are:

Shaft I:

$$\omega_{shift,k} = \left(1 - \frac{2}{R_1}\right) \omega_{ICE} i_{s,k-1} \text{ for } k = 3, 5, 7, \dots \quad (5.4a)$$

Shaft II:

$$\omega_{shift,k} = \left(1 - \frac{2}{1 - R_2}\right) \omega_{ICE} i_{s,k} \text{ for } k = 3, 5, 7, \dots \quad (5.4b)$$

It is assumed that each displacement machine uses its entire speed and torque operating range to minimise the necessary size of the displacement machines. In other words, it is assumed that the optimal solution will be found on these design constraints. This assumption is made to simplify the design equations and to reduce the degrees of freedom. The synchronisation speed of the first mode shift is therefore:

$$\omega_{shift,1} = \frac{\omega_{2,max}}{i_2} i_{s,0} \quad (5.5)$$

Unit 1 is directly connected to the engine shaft and runs with a constant speed at all vehicle speeds:

$$i_1 = \frac{-\omega_{1,max}}{\omega_{ICE}} \quad (5.6)$$

Unit 2 reaches maximum speed in all mode shifts:

$$i_2 = \frac{-\omega_{ICE}}{\omega_{2,max}} = \frac{\omega_{ICE}}{-\omega_{2,max}} \quad (5.7)$$

Unit 2 is dimensioned to handle the load requirements in each mode shift. The load constraint on Unit 2 for the startup mode is:

$$\frac{\Delta p_{max} D_2}{2\pi} \geq T_{max} i_{s,0} i_2 \quad (5.8)$$

The loss-free load constraints on Unit 2 for the odd mode shifts (shifting to Shaft II in Fig. 2.16) are:

$$\frac{\Delta p_{max} D_2}{2\pi} \geq \frac{T_{shift,k} i_{s,k} i_2}{1 - R_2} \text{ for } k = 1, 3, 5, \dots \quad (5.9)$$

The loss-free load constraints on Unit 2 for the even mode shifts (shifting to Shaft I in Fig. 2.16) are:

$$\frac{\Delta p_{max} D_2}{2\pi} \geq \frac{T_{shift,k} i_{s,k} i_2}{-R_1} \text{ for } k = 2, 4, 6, \dots \quad (5.10)$$

Unit 1 is dimensioned to meet the loss-free flow demand of Unit 2:

$$\omega_{1,max} D_1 = \omega_{2,max} D_2 \quad (5.11)$$

The maximum speed of Unit 1 and Unit 2 is calculated according to section 3.2 with $\beta_1 = \beta_2 = 1.0$.

$$\omega_{1,max} = C_{m,1} D_1^{-1/3} \quad (5.12a)$$

$$\omega_{2,max} = C_{m,2} D_2^{-1/3} \quad (5.12b)$$

5.1.3 Equation System and Degrees of Freedom

By arranging eq. (5.1) - (5.12) an equation system is formed where two degrees of freedom are achieved. The system parameters and design parameters are:

$$X_{sp} = [D_1 \ D_2 \ R_1 \ R_2 \ i_1 \ i_2 \ i_{s,0} \ i_{s,1} \ i_{s,2} \ \dots \ i_{s,m-1}]^T \quad (5.13a)$$

$$X_{dp} = [x_1 \ x_2]^T \quad (5.13b)$$

The final gear ratio i_0 does not enable any design freedom to the system and is considered to be constant for all designs. The design parameters, which represent the degrees of freedom for the system, can be expressed as two fractions describing how the mode shifts can be positioned:

$$x_1 = \frac{v_{shift,k+1}}{v_{shift,k}} \text{ for } k = 1, 3, 5, \dots \quad (5.14a)$$

$$x_2 = \frac{v_{shift,k+1}}{v_{shift,k}} \text{ for } k = 2, 4, 6, \dots \quad (5.14b)$$

The design parameters are directly related to the gear ratios of the planetary gears according to:

$$R_1 = \frac{2}{1 - x_2} \quad (5.15a)$$

$$R_2 = \frac{1 + x_1}{1 - x_1} \quad (5.15b)$$

5.1.4 Simulation-based Optimisation

The design optimisation problem is formulated as:

Find $X_{dp} = [x_1 \ x_2]^T$ minimising $F(X_{dp})$ subject to the constraints $g_j(X_{dp}) \leq 0$ for $j=1..n$

where:

X_{dp} are the design parameters;

$F(X_{dp})$ is the objective function;

$g_j(X_{dp}) \leq 0$ $j=1..n$ are the inequality constraints.

Objective Function

The objective function for this implementation focuses on two aspects of the gearbox characteristics - consumed energy and total installed hydraulic displacement. The total installed displacement can be thought of as the cost of the gearbox, since larger hydraulic machines are normally more expensive. To handle the multiple objectives, the weighted-sum approach is used:

$$F(x_1, x_2) = \lambda_1 f_1 + \lambda_2 f_2 \quad (5.16)$$

where f_1 is the total energy consumption from the simulation and f_2 is the total installed displacement:

$$f_1 = \sum_k w_k \int_0^{t_{end,k}} P_{tot,k}(t) dt \quad (5.17a)$$

$$f_2 = D_1 + D_2 \quad (5.17b)$$

where k is the operating cycle and w_k is the corresponding weight of the cycle according to Table 5.2. Note that f_2 can be directly deduced

from the design equations while f_1 is a result of the simulation of the specific transmission design. The weight factors λ_1 and λ_2 are changed to sample points to form a Pareto optimal front.

Optimisation Algorithm

Since the objective function of the optimisation problem is based on simulation, a non-gradient based method is required. For this implementation, the Complex RF method is used. The Complex RF is an updated version of the original complex method first developed by Box in 1965 [52]. Briefly explained, the algorithm generates a certain number of random points in the design space and reflects the worst point towards the centroid of the other points until it is no longer the worst point. This process is iterated until the minimum function value is found or the maximum number of evaluations is reached. It is simple, easy to use and has previously been proved suitable for direct-search design optimisation of complex systems [48]. Note that the choice of optimisation algorithm is not critical to the methodology, since the execution time of a simulation is normally very short when using backward-facing simulation.

Constraints

From the above explicit design relation, each generated transmission design has the performance to meet the tractive force requirements. To avoid transmission designs that are unsuitable for other reasons, design constraints are added to the optimisation. In the complex RF algorithm, the design constraints are included in the objective function as penalties.

Two inequality constraints are here used in the optimisation to restrict the freedom of positioning the mode shifts in the speed range, see eq. (5.18).

$$g_1(X_{dp}) = 5 - 3.6v_{shift,1} \leq 0 \quad (5.18a)$$

$$g_2(X_{dp}) = 2 - 3.6(v_{shift,k+1} + v_{shift,k}) \leq 0 \text{ for } k = 1, 2, \dots, m-1 \quad (5.18b)$$

The first constraint relates to the need to avoid mode shifts during the bucket fill operation, which normally takes place at under 5 km/h. The second constraint prevents the mode shifts to be positioned too closely together. Between two mode shifts, the relative displacement of Unit 1

is controlled from -1 to 1 or vice versa. During quick accelerations and decelerations, there are consequently high demands on the control speed of the displacement machines. For this design process, a minimum of 2 km/h is allowed between two mode shifts.

5.1.5 Results and Discussions

Consecutive optimisations are performed to find the global optimum for $m = 2, 3, 4$ and 5. The backward-facing simulations as well as the optimisation algorithm is implemented in MATLAB m-scripts on a 64-bit Intel® Core™ 2 Duo (3.07 GHz). The execution times for a three-mode transmission are:

- 2.67 s for one simulation.
- 2 min 44 s for one optimisation (65 evaluations, 37 simulations).

The execution times vary slightly depending on tolerances, optimisation parameters, etc.

For the considered design task, the design parameter values that minimises the energy consumptions also minimises the total installed displacement, i.e. a clear Pareto front is not formed. Figure 5.5 shows the characteristics of the optimised designs for $m = 2$ and $m = 3$. Figure

Figure 5.5 Optimised designs for $m = 2$ and 3.

5.5a shows the functionality of the designs and how the maximum vehicle speed is reached. The mode shifts occur when the hydrostatic speed

ratio is at minimum and maximum. Figure 5.5b shows a simulation of the transmission efficiency at maximum load conditions (see Fig. 5.2). The distribution of the typical operating behaviour is also shown in the graph, where a high value represents a speed interval where the wheel loader operates frequently. Some peaks in the distribution can be identified: the bucket fill operation at 0 to 5 km/h, the reverse motion from the gravel pile at around -8 to -10 km/h and the transportation phase at around 25 km/h. In the figure, it is obvious that the optimisation algorithm positions the mode shifts to match the full mechanical points with the most frequent operating points. This minimises the total energy consumption of the simulated cycles. Figure 5.6 shows the same graphs for $m = 4$ and $m = 5$. It can be seen that the first mode

Figure 5.6 Optimised designs for $m = 4$ and 5.

shift is positioned at exactly the same speed for $m = 3, 4$ and 5. This relates to the constraint that restricts mode shifts under 5 km/h, see eq. (5.18a). The other mode shifts are cleverly positioned to match the most frequent operating points. Figure 5.7 shows the relative displacement and system pressure for the simulations of the transmission. The transmissions are automatically designed to utilise the maximum pressure and maximum speeds of the displacement machines. For all transmissions, the maximum pressure is reached in the first mode that sets the required sizes of the displacement machines. This is confirmed in Table 5.3, which shows the total installed displacement and equivalent energy consumption for the optimised designs. Since the first mode shift

(a) Pressure and relative displacement of Unit 1 for $m = 2$

(b) Pressure and relative displacement of Unit 1 for $m = 3$

(c) Pressure and relative displacement of Unit 1 for $m = 4$

(d) Pressure and relative displacement of Unit 1 for $m = 5$

Figure 5.7 Simulation of an acceleration from 0 to v_{max} at maximum load conditions for $m = 2, 3, 4$ and 5.

Table 5.3 Comparisons of the optimised transmission designs.

m	$D_1 + D_2$	Energy Consumption
2	$650 \text{ cm}^3/\text{rev}$	100%
3	$370 \text{ cm}^3/\text{rev}$	86.8%
4	$370 \text{ cm}^3/\text{rev}$	86.4%
5	$370 \text{ cm}^3/\text{rev}$	86.1%

is the same for $m = 3, 4$ and 5, the displacement requirements are also the same. The energy consumptions of the weighed operating cycles are also very similar between $m = 2, 3$ and 4. This confirms the tight connection between transmission efficiency and total installed displacement. Transmissions with a large hydraulic variator transfers a larger amount of the power hydraulically, which causes higher power losses. As previously noted, larger displacement machines also require a larger

charge pump, which increases the parasitic losses of the transmission. Besides this, the three-mode transmission is well capable of matching the main peaks in the speed distribution of the typical operating behaviour, as seen in Fig. 5.5b. By adding more modes, the efficiency is essentially only raised at points which are not frequently operated, as seen in Fig. 5.6b. The total energy consumption is then only slightly decreased. This is of course strongly dependent on application and transmission concept. Nevertheless, this is an unexpected result that would be hard to find without using design automation.

Figure 5.8 shows the optimised values of the design parameters. For

Figure 5.8 The optimal values of the design parameters for all transmissions.

$m = 3, 4$ and 5 , the optimal values seem difficult to find manually, which shows the proficiency of the methodology. For $m = 2$, only one planetary gear is used and only one design parameter, x_1 , is consequently available. The optimal value of x_1 also minimises the total installed displacements, see Fig. 5.9. For $x_1 \leq 5.5$, the second mode becomes the mode that sets the requirements for the displacement machines.

For all transmissions, the objective functions, f_1 and f_2 , share the same optimum. The minimisation of the total installed displacement

Figure 5.9 Functional space of the two-mode transmission.

does not, however, always guarantee an optimal energy consumption, which is dependent on the positioning of the mode shifts. The necessary power losses of the charge pump can be different depending on application, but they are inarguably important to consider when designing the transmission. In [I, II], multi-objective design optimisations are performed where several designs can be found along a Pareto front.

5.2 Modular Design

In this section, a design optimisation is made of a modular Jarchow transmission for three wheel loaders of different power classes. All wheel loaders follow the tractive force requirements shown in Fig. 5.2. Table 5.4 shows the main parameters of the considered wheel loaders, denoted A-C, where A is the largest and C is the smallest. The typical operating

Table 5.4 Wheel loader parameters.

Wheel loader	A	B	C	Unit
P_{max}	219	161	96	kW
F_{max}	245	181	130	kN
v_{max}	40	40	40	km/h

behaviours are described by recorded data of the operating cycles shown in Fig. 5.3 of all three wheel loaders. The complete operating behaviours of the wheel loaders are then achieved by weighting each cycle according to its significance. The weight factors differ for each application. For

instance, larger wheel loaders normally operate more frequently in short loading cycles.

5.2.1 Local Optimisation

The local optimisation loop chooses the local design parameters, $X_{dp,l}$, to minimise the energy consumption of each application. The Jarchow concept is here realised with Unit 2 as a variable displacement machine. For this implementation, the minimum displacement of Unit 2, $\varepsilon_{2,min}$, is left as a design parameter. The system parameters and the design parameters for the local optimisation are consequently:

$$X_{sp,l} = [D_1 \ D_2 \ R_1 \ R_2 \ i_1 \ i_2 \ i_{s,0} \ i_{s,1} \ i_{s,2} \ \dots \ i_{s,m-1}]^T \quad (5.19a)$$

$$X_{dp,l} = [x_1 \ x_2 \ \varepsilon_{2,min}]^T \quad (5.19b)$$

The objective function for the local optimisation is simply the simulated equivalent energy consumption of the transmission for the specified operating cycles, shown in eq. (5.20):

$$f_l = \sum_k w_{l,k} \int_0^{t_{end,k}} P_{tot,k}(t) dt \quad (5.20)$$

where w_k is the weight factor for the operating cycle, as described above. The constraints in eq. (5.18) are used to limit the positioning of mode shifts.

5.2.2 Global Optimisation

The global optimisation loop sets the optimal size of the variator and the number of modes for each wheel loader. The design parameters for the global optimisation are consequently:

$$X_{dp,g} = [m \ D_{var}]^T \quad (5.21)$$

where

$$m = [m_A \ m_B \ m_C]^T \quad (5.22a)$$

$$D_{var} = D_1 + D_2 \quad (5.22b)$$

In section 5.1, the sizes of the displacement machines are calculated and can take any continuous value. A more realistic approach for an industrial designer is to choose from a discrete series of sizes. To represent

this scenario, the listed sizes of the Bosch Rexroth standard displacement machines A4VG/40 (Unit 1) and A6VM/71 (Unit 2) [44] are used as possible variator sizes. One variator size means, in this case, only one combination of machine sizes, according to eq. (5.23):

$$\begin{bmatrix} D_1 \\ D_2 \end{bmatrix} \in \left(\begin{bmatrix} 145 \\ 170 \end{bmatrix} ; \begin{bmatrix} 175 \\ 215 \end{bmatrix} ; \begin{bmatrix} 210 \\ 280 \end{bmatrix} \right) \quad (5.23)$$

The target of the global optimisation problem is a multi-objective one. The objective formulates the trade-off between energy efficiency and the manufacturing cost of the gearbox according to eq. (5.24):

$$f(X_{dp,g}) = \lambda_1 f_{g1} + \lambda_2 f_{g2} \quad (5.24)$$

The first objective, f_{g1} , is the weighted sum of equivalent energy consumptions from the local optimisation results according to eq. (5.25):

$$f_{g1} = \sum_k w_{g1,k} E_{k,equiv} \text{ for } k = A, B \text{ and } C \quad (5.25)$$

The weight factors $w_{g1,k}$ specify the importance of energy efficiency for one application versus another. The values of the weight factors are difficult to determine, since they are specific to the company business case. In this application example, the equivalent energy consumption is evenly weighted for the considered wheel loaders. The second objective, f_{g2} , is the weighted manufacturing costs of the gearboxes, according to eq. (5.26):

$$f_{g2} = \sum_k w_{g2,k} C_k \text{ for } k = A, B \text{ and } C \quad (5.26)$$

The weight factors $w_{g2,k}$ also relate to the above mentioned market-related factors and are here assumed to be equal. The manufacturing costs are simply modelled as the sum of all component costs plus a fixed cost for the housing, additional hydraulic components, etc [II].

5.2.3 Results and Discussions

Figure 5.10 shows optimisation results for consecutive runs with linearly changed weight factors λ_1 and λ_2 to form the Pareto optimal front. The solution will generally travel to the bottom right corner of the graph with a greater number of modes for the transmissions. A suitable weight between the two objectives is probably found somewhere in the middle,

Figure 5.10 The Pareto front for the multi-objective design optimisation. Each marker represents a set of numbers of modes as indicated, according to $[m_A, m_B, m_C]$. The top left corner represents low energy efficiency and low costs, whereas the bottom right corner represents high costs and high energy efficiency.

where a greater number of modes for a larger application is a reasonable choice. The awkward selections of m on the Pareto front show that an optimised manual design would be difficult to achieve. The absence of the 4-mode transmission is explained by the high capabilities to match the typical operating behaviour of the 3-mode transmission. The 3-mode transmission is accordingly almost as efficient as the 4-mode transmission, but with fewer components. To identify the decreased individual energy efficiency of the applications, the optimised designs are simulated

Table 5.5 Percentage increase in energy consumption when using the largest variator instead of the smallest one.

Wheel loader	A (Largest)	B	C (Smallest)
$m = 2$	2.6%	1.9%	2.5%
$m = 3$	-1.4%	2.1%	3.3%
$m = 4$	-1.1%	1.1%	2.9%
$m = 5$	-1.0%	1.5%	3.0%
$m = 6$	-2.3%	0.5%	2.6%

with two different variator sizes. Table 5.5 compares the equivalent energy consumptions of the wheel loaders using a larger variator than is actually needed to achieve the specified performance. Normally, a transmission with a smaller variator suffers from lower power losses, as the parasitic losses from the HST are reduced. This seems to be the case for wheel loaders B and C. A small variator used in a large application, however, is forced to operate with smaller relative displacements, which lowers the efficiency of the transmission. This is the reason why wheel loader A has a lower energy consumption when a larger variator is chosen.

6

Summary and Conclusions

This thesis deals with the design of complex hydromechanical transmissions and proposes a design methodology using simulation-based optimisation of the transmission concept. The methodology is application-focused and adapts the transmission design to the typical operating behaviour of the vehicle. By using detailed models of the power losses in the transmission components and backward-facing simulations, the energy efficiency of a transmission design is evaluated effectively and used for design optimisation. Design automation of the transmission concept has been proven to be effective in the design process. The results confirm the difficulties to find an optimal transmission design manually, especially for transmissions with many modes. The energy efficiency of a transmission is to some extent coupled to the size of the hydraulic variator, which is responsible for most of the power losses in the transmission. Nevertheless, it is important to consider the losses of the mechanical part and the parasitic losses of the transmission in the search for the optimal design. The operating range of hydraulic displacement machines is also an important consideration since it influences the required size of the machines. The energy consumption of a transmission concept is strongly coupled to the design of the transmission, since the energy efficiency varies a great deal throughout the speed range. It is consequently necessary to use design optimisation to compare the capabilities of different transmission concepts. A modular transmission for a range of applications makes the design task even more complex and multi-dimensional, which increases the need for design automation.

As the development of new high-efficient components progresses, the choice of transmission concept and design may change. This might also be the case for different manufacturers due to market-related factors, fuel prices, customer demands, etc. With an automated design methodology, new models and parameters are easily imported and accounted for in the design process. The trend towards more intelligent control and hybrid transmissions also increases the need for a more effective design process. Design optimisation with backward-facing simulations is a promising tool in this respect since the drive line control can be automatically adjusted for each design. The design methodology could also be further expanded to optimise the transmission architecture by formulating rules for how a multiple-mode transmission can be assembled. The outcome of the design optimisation would then be an optimal transmission concept for the vehicle application.

7

Review of Papers

Paper I

Design Optimisation of Complex Hydromechanical Transmissions

This paper introduces the proposed design methodology for hydro-mechanical transmissions. The design process is established on a high level and implemented on the Jarchow concept for a heavy wheel loader application. The functional space of the transmission is investigated together with how different designs affect the energy consumption and required installed hydraulic displacement of the transmission. The proficiency of the methodology is thereby established and the difficulty of finding the optimal design manually is shown.

Paper II

Optimisation and Concept Sensitivity of Continuously Variable Hydromechanical Transmissions

The design methodology is here used on the same concept to minimise the energy consumption of the transmission and the manufacturing costs of the gearbox. The trade-off between energy efficiency and cost in the design task is described. This type of analysis supports the choice of number of modes. The concept's sensitivity is also investigated by letting the power loss models vary and examining the change in the results.

Paper III

Modular Design of Hydromechanical Transmissions for Mobile Working Machines

This paper extends the design methodology to consider a range of applications. The goal is to design a modular transmission concept with a common hydraulic variator. For high-power vehicles, additional modes are used to match the increased requirements on the transmission. For this design task, the choice of number of modes for each application and the size of the hydraulic variator are the global design parameters, whereas the design of each individual transmission is made in a local optimisation loop.

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