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ADVANCED DESIGN METHODS FOR HYDROMECHANICAL TRANSMISSIONS IN ROAD AND AGRICULTURAL VEHICLES

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*When you give everything **you have**,
you've really given **everything**.*

SUMMARY

Nomenclature	V
1 Introduction	1
1.1 General	1
1.2 Objectives of the thesis	2
2 Literature review	5
2.1 State of the art	5
2.2 Theory and components of hydromechanical transmission	8
2.2.1 The planetary geartrain	8
2.2.2 Hydrostatic continuously variable transmission	11
2.3 The hydromechanical transmission	13
2.3.1 The Input-Coupled HMT	15
2.3.2 The Outut-Coupled HMT	21
3 Optimization of components and layouts of hydromechanical transmission	25
3.1 Introduction	25
3.2 Model	26
3.2.1 Hydrostatic group	27
3.2.2 Mechanical elements	28
3.3 Overall system of equations	30
3.4 Optimization	31
3.4.1 Degrees of freedom	32
3.4.2 Design parameters	33
3.4.3 Design constraints	33
3.4.4 Objective function	34
3.5 Optimization algorithm	34
3.5.1 Definition of the possible configurations	35
3.5.2 Particle Swarm optimizer	37
3.6 Optimization procedure application	38
3.7 Results	43

4	Simultaneous optimization of layout and components	49
4.1	Introduction	49
4.2	Hydromechanical transmission layouts	50
4.2.1	Layout I	52
4.2.2	Layout II	55
4.2.3	Layout III	57
4.2.4	Layout IV	58
4.2.5	Layout V	60
4.2.6	Layout VI	61
4.3	Simulation model	63
4.3.1	The hydrostatic transmission model	63
4.3.2	The real hydraulic machines	64
4.3.3	The engine model	66
4.3.4	The planetary gertran model	66
4.3.5	The vehicle model	68
4.3.6	Load control loop	69
4.3.7	Vehicle target speed and displacement control loop	70
4.3.8	High-speed mechanical losses	74
4.4	General optimization statement	74
4.5	The polar function	76
4.6	Manual search optimization	82
4.7	Procedure's check	83
4.7.1	Design variables	84
4.7.2	Reference vehicle	84
4.7.3	Solution of the design equations and simulation results	85
4.8	Results: the efficiency matrix	88
5	Prediction of sound pressure level for a Dual-Stage Hydromechanical transmission	91
5.1	Introduction	91
5.2	Dual-stage hydromechanical transmission	92
5.3	Simulation model of the vehicle	93
5.3.1	Engine	94
5.3.2	Hydromechanical transmission	95
5.4	Sound pressure level prediction models	98
5.4.1	Internal combustion engine noise emissions	98

5.4.2	Hydrostatic CVT noise emissions	101
5.5	Simulation and results	104
6	Experimental activity: the Toyota Prius Hydraulic Test Rig	113
6.1	Maha Fluid Power Research center	113
6.2	The Hydraulic Hybrid Toyota Prius	114
6.2.1	The output-coupled hydraulic hybrid driveline	115
6.3	The Toyota Prius hydraulic test rig	120
6.3.1	Engine and Load simulators	120
6.3.2	Hydraulic Unit 1 and Unit 2	121
6.3.3	The hydraulic power supply	122
6.4	Test rig schematic	123
6.4.1	The Engine and Load simulators	125
6.4.2	Unit 2	126
6.4.3	The isolating valve and the bleeding valve	126
6.4.4	Check valves	128
6.4.5	Pressure and temperature sensors	128
6.5	Engine and Load simulator test	128
6.5.1	Schematic and calculations	129
6.5.2	Modifications of the original circuit	131
6.5.3	The test procedure	131
6.5.4	The calibration of the linear variable differential transformer	132
6.5.5	The rotational direction of the shaft	133
	Conclusions	135
	References	VII
	Appendix	XI
	Aknowledgments	XIII

NOMENCLATURE

α	fractional displacement	[-]
a_{abs}	acoustic absorption coefficient	[-]
A, B, C	noise coefficient A	[dBA]
D	gear diameter	[m]
η	efficiency	[-]
F	force	[N]
fmp	full mechanical point expressed in % of the maximum speed	[-]
ϑ	design variable and coordinate in the polar function	[rad]
k	exponent for the power flow direction	[-]
K, J	reduction noise factor	[dBA]
l	length	[m]
L_w	sound power level	[dBA]
L_p	sound pressure level	[dBA]
M, m, T	torque	[Nm]
n	rotational speed	[rev/min]
n	number of iteration	[-]
ω	angular velocity	[rad/s]
p	pressure	[Pa]
P	Power	[W]
Q	directivity factor	[-]
r	distance from noise source	[m]
ρ	coordinate in the polar function	[m]
R	radius	[m]
R	room constant	[m ² Sabine]
s	accelerator partialisation	[-]
S	total plane surface	[m ²]
t	time	[s]
τ, i	gear ratio	[-]
$T, i0$	gear ratio of the planetary geartrain	[-]
v	velocity	[m/s]
V	displacement of the hydraulic unit	[cm ³]
χ_ω	speed effect correction factor of the hydraulic efficiency	[-]
$\chi_{\alpha,p}$	pressure and fractional displacement effect hydraulic efficiency	[-]
z	number of teeth	[-]

ACRONYMS

CNG	Combustion Natural Gas
CVT	Continuously Variable Transmission
FMP	Full Mechanical Point
HST	Hydrostatic transmission
HMT	Hydromechanical transmission
IC	Input Coupled
ICE	Internal Combustion Engine
OC	Output Coupled
SPL	Sound Pressure Level
PGT	Planetary geartrain

SUPERSCRIPTS AND SUBSCRIPTS

'	refers to the hydraulic machine shaft
0	referred to the PGT gear ratio
1,I	hydrostatic group inlet
2,II	hydrostatic group outlet
1,2,3	referring to the three-shaft spur gear
A	sun gear
A ₁	sun gear 1
A ₂	sun gear 2
B	carrier gear
C	ring gear
CE	internal combustion engine
C,S,R	carrier, sun, ring of the epicyclical gear
eng	engine
ex	engine exhaust system
h	engine housing
hydr	hydraulic portion
hym	hydromechanic
int	engine air intake system
loss	power losses
max	maximum
mech,fmp	full mechanical point
mech,2	second full mechanical point
mech,1	first full mechanical point
min	minimum
out	elements downstream of the transmission
o ₁	transmission output (first gear)
o ₂	transmission output (second gear)
O,p,PL	planet elements
rif	maximum efficiency condition
shift	gear shift
v	volumetric
wheel	wheel axis

1. INTRODUCTION

1.1. General

In the current scenery of worldwide shortage in energy and raw materials, and the resulting sharp increase in prices, the energy and resource efficiency of machines and systems has now become more important than ever.

Fluid power, as a high-performance technology for the creation of drive solutions, plays a very active role in this process of continuous improvement. From a financial point of view, considering the total cost of ownership over the entire life of a product, energy costs have progressively become the center of focus. It has become clear that, currently, the only way to reduce energy costs is through energy conservation.

In the high-power vehicles sector, which comprises agricultural tractors, heavy trucks, earthmoving machines, and passenger vehicles, there is the need to reduce fuel consumption owing to the higher cost of the petrol, and to reduce emissions to the environment, as well (Figure 1.1).

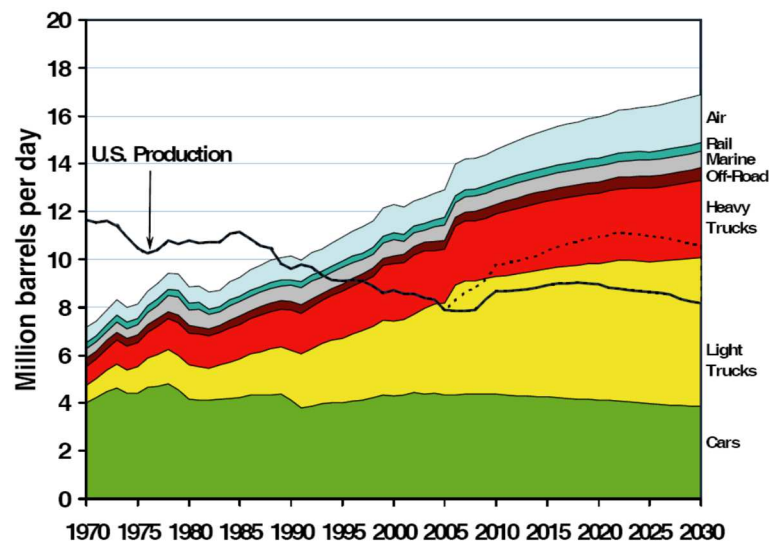


Figure 1.1 Petroleum production and consumption. United States (ORNL 2008)

To make matters even more challenging, a considerable decrease in the sales of agricultural tractors in European countries has occurred, as illustrated in Figure 1.2, caused by the decrease in price of cereal and agricultural products. Furthermore, the new components added to high-power vehicles to improve emissions and fuel saving

have further increased product costs. The purchase of vehicles has consequently become more reserved.

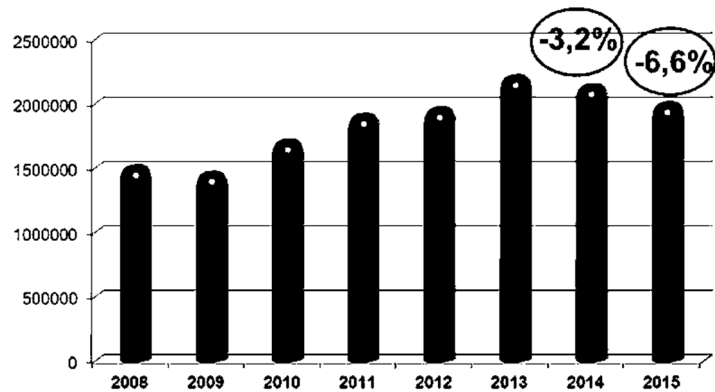


Figure 1.2. Worldwide sales of agricultural machines (source: Agrievolution-FederUnacoma).

The development of hydromechanical transmissions has grown, in an effort to overcome the aforementioned issues. In fact, these transmissions can manage the engine and its emissions, grant reliability and flexibility during operation, and offer a high level of comfort and drivability to the driver.

However, they suffer from a lower global efficiency than their mechanical counterpart does, and to overcome this disadvantage, an optimal transmission design needs to be realized.

On the other hand, due to the presence of hydraulic components, hydromechanical transmissions generate high noise levels. The recent regulations for machines require the conformance to high standards, such as generation of lower noise levels. A new challenge is therefore to study the transmission noise in order to determine the components that need improvement to making the system even more comfortable.

1.2. Objectives of the thesis

The hydromechanical transmission transfers the mechanical power from the engine to two different branches: one that is mechanical and is constituted of a planetary gear train, and one that is hydraulic, and is constituted of a hydrostatic transmission. The traditional design procedure of the hydromechanical transmission is based on certain assumption regarding the design parameters; consequently, the global

efficiency is a function thereof. However, the efficiency value would not be the best achievable by the system, if a different assumption on the parameters were to be made.

Therefore, the objective of this work is to create innovative design methods to help improve the hydromechanical transmission in terms of global efficiency and noise level.

The hydromechanical transmission, owing to the flexibility of the planetary gear train, can assume different geometrical configurations and layouts, leading to different design solutions. In order to find the design parameters that can achieve the best efficiency for each layout, two different optimization design procedures have been developed. In previous studies, optimization was applied to transmissions, the structures of which were a priori assigned. However, the various layouts do not have the same performance, because they have different internal power flows during the operations, as well as different internal losses.

The first study is dedicated to finding the optimum efficiency point of the simple input- and output-coupled configurations, for different layouts and planetary gear train ratios. The design is transformed into a mathematical programming problem, the goal of which is the optimization of both the configuration and the internal components of the transmission. The structure of the transmissions is described through graph theory, and the solution of the optimization problem was implemented by using a “direct search” algorithm, based on the particle swarm method. The procedure was applied to the transmission of a high-power agricultural tractor.

The second work is dedicated to finding the optimal efficiency point for the simple input-coupled configuration, and differs from the first work regarding the choice of the design parameters. Moreover, the efficiency changes by changing the speed of the full mechanical point, which is a strategic speed in which the optimal efficiency point during operation occurs, and this consideration will be taken into account in the work. In the first work the substantial difference is that the speed of the full mechanical point was assigned as a constant in the input of the optimization algorithm, and differs from the approach of the second work, where it was considered as a variable. The optimization method is a “manual search” procedure, based on an efficiency matrix function of the global parameters. This method is useful for obtaining a complete overview of the design problem. The design equations were solved in MATLAB, and the global efficiency was evaluated by using the transmission Amesim models. The procedure was applied for a 200 kW passenger bus.

In the third work, a predictive noise model for a hydromechanical transmission is developed. The scope of the study is first to build a tool with which to predict the sound pressure level during the design phase. The model permits to identify the physical parameters (pressure, speed, etc.) which mostly affect the global noise. The model consists in the calculation of the sound pressure level that is generated by the two main components of the transmission: the engine and the hydrostatic system. For the engine, a model from the literature was applied, whereas for the hydrostatic transmission, a numerical regression of the experimental results was developed. The two contributions were summed, and the global sound pressure level was evaluated for a four-shaft transmission (Dual-stage) installed in a city bus, equipped with a compressed natural gas (CNG) engine. The predictive model is also useful for pointing out the difference in terms of noise between the engine and the hydrostatic transmission.

The thesis is organized in three main sections: the first analyzes and explains two design optimization methods for the improvement of the global efficiency of the system; the second explains the predictive noise model applied to a complex hydromechanical transmission, and the third section describes the experimental research activity conducted in the American fluid power laboratory “Maha Fluid Power Research center” at Purdue University.

2 LITERATURE REVIEW

2.1 State of the art

Transmissions are the most essential component in a vehicle powertrain, which converts the high speed and low torque produced by the engine to a lower speed and higher torque as input to the wheels. The powertrain performance, as well as the vehicle performance and the fuel economy, are greatly affected by transmission operation. In general, a power transmission provides a variable connection between the vehicle primary power source (the engine) and the primary power consumer (wheels), as shown in Figure 2.1. Currently, several types of power transmission systems exist, and are based on a wide range of architectures. Conventional transmissions are generally classified as either manual or automatic transmissions; there are certain new designs, such as the automated manual transmission, which is a combination of the two. For passenger vehicles such as cars, the automatic transmissions are preferred in American countries (i.e., in the United States and, mostly, in North America) and emerging markets, whereas manual transmissions are preferred in Europe and in certain developing countries.

In addition to the conventional transmission types previously discussed, there are also the hydraulic transmissions, and, particularly, the Hydrostatic Transmissions (HSTs), mainly used for applications in machinery and off-road vehicles.

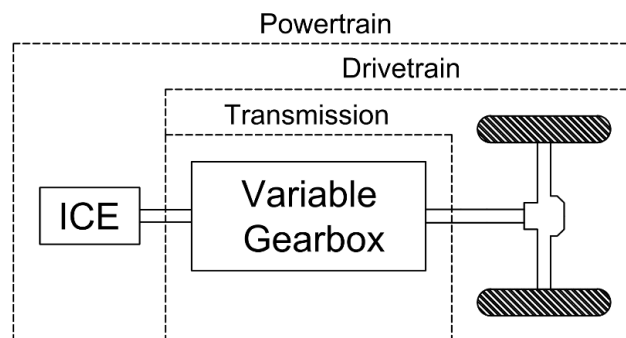


Figure 2.1: Powertrain overview.

Hydrostatic transmissions transmit power by moving pressurized fluid and by using positive displacement hydraulic units to convert the power between the rotational and the fluid domains. These volumetric machines come in two assortments: fixed displacement units, which move a fixed volume of fluid per revolution, and variable displacement units, which can continuously adjust the volume of fluid transferred. The assembly of two hydraulic units in series, at least one of which is variable displacement, constitutes a continuously variable hydrostatic transmission (CVT). A schematic of a simple hydrostatic transmission with the hydraulic units both variable displacements is shown in Figure 2.2.

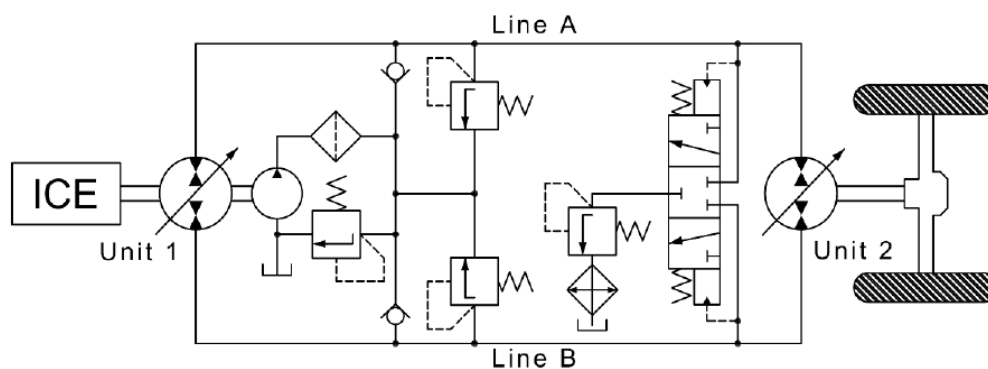


Figure 2.2: Hydrostatic transmission.

One of the first works on the HSTs for off-road applications began in the early 1950s with the development of a research tractor (Hamblin,1952). Probably in the same year, several tractor manufacturers investigated the HST as an alternative to the power transmission system; however, no products were introduced to the market (Meile,1961). In 1956, Linde introduced one of the first commercial vehicles with HST, and perhaps, owing to its first application in 1967, the American International Harvester, in cooperation with the Sunstrand Corporation, began the commercial production of HSTs for agricultural tractors (Morris,1967). While the design was successful, the HST presented inefficiencies and high production cost when compared to the conventional mechanical transmissions; a more commercial product emerged, with relatively low power, which was up to approximately 30 kW (Renius and

Resch, 2005). Currently, the HSTs are still widely available in applications, such as small tractors and wheel loaders.

The hydrostatic transmission has two main limits: low efficiency and a small range of gear ratios, thus eliminating the possibility of being installed in low-power vehicles.

Another class of hydraulic transmissions, which improves the inefficiencies of the HST, is called the hydromechanical transmission (HMT). This class of transmissions combines the benefits of the continuously variable output speed of the HSTs and the great efficiency of the mechanical transmissions. The core of the HMT is the planetary gear train (PGT), a device, which enables the splitting and combining of the power between three separate mechanical shafts (Figure 2.3).

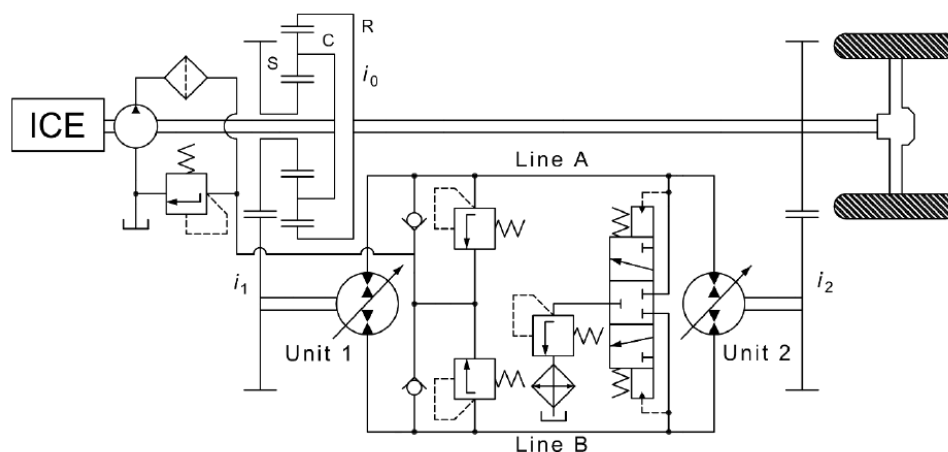


Figure 2.3: Hydromechanical transmission.

The HMT uses the PGT to split power between an efficient mechanical path and a flexible variable hydrostatic path. Several configurations and layouts exist for the HMT, based on either the position or the number of the PGTs or the connections between the shafts.

The first successful applications of HMTs emerged near the 1970s with Sundstrand Corporation, which developed notable hydrostatic and hydromechanical transmissions for the off-highway market. One example includes the “Responder”, an output-coupled hydrostatic drive (Ross,1972). In

1995, Fendt introduced the first commercially successful hydromechanical transmission with the “Vario,” for agricultural tractors (Renius and Resch, 2005). Consequently, multiple manufactures released other HMT configurations, for example the Styer compound input-coupled “S-Matic” in 2000, the ZF Dual-stage input-coupled “Eccom” in 2001, and the John Deere dual-stage input-coupled “AutoPower” in 2001.

2.2 Theory and components of hydromechanical transmissions

In this section, a more detailed explanation of the architecture of the HMT is provided. The simplest HMT configuration is composed of three main elements connected to each other: a power source (engine), the continuously variable path (hydrostatic transmission), and the planetary gear train.

2.2.1 The planetary geartrain

The planetary gear train (PGT) is a mechanical device that has the ability to sum and split the mechanically transferred power. Willis (1841) described a typical PGT consisting of three main elements: the sun gear, the ring gear, and the carrier (Figure 2.4).

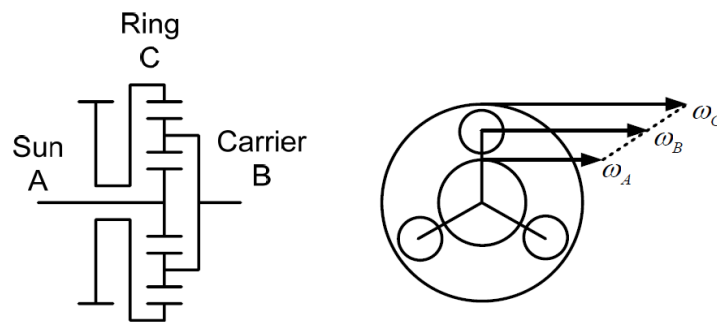


Figure 2.4: Planetary gear train and angular velocity vectors.

In the absence of constraints, the planetary gear train has three degrees of freedom. To permit the transfer of torque between two elements, the input and the output, one of the elements should be fixed on the chassis. In this case,

the PGT has two degrees of freedom; therefore, the rotational speed of two elements must be known in order to determine the speed of the third element. The kinematic motion of a PGT is represented by the following:

$$\frac{\omega_A - \omega_B}{\omega_C - \omega_B} = \text{constant} = \left. \frac{\omega_A}{\omega_C} \right|_{\omega_B=0}. \quad (2.1)$$

The equation states that the ratio of the relative motion between the sun gear and the carrier to the relative motion between the ring gear and the carrier is constant.

We can define the standing gear ratio as in the following:

$$i_0 = \left. \frac{\omega_A}{\omega_C} \right|_{\omega_B=0} = - \frac{Z_C}{Z_A}. \quad (2.2)$$

Where Z_C and Z_A denote the number of the teeth of the ring gear and the sun gear respectively. From Equations (2.1) and (2.2), the Willis equation can be derived:

$$\omega_A - i_0 \cdot \omega_C - (1 - i_0) \cdot \omega_B = 0. \quad (2.3)$$

There is a method, which permits to represent the velocity terms of the Willis equation as vectors. The magnitude and the direction of each vector represent the magnitude and the rotational direction of the corresponding term in the Willis equation, respectively. This vector representation is shown in Figure 2.4.

The force diagram of the PGT is shown in Figure 2.5 and can be used to derive the torque equations.

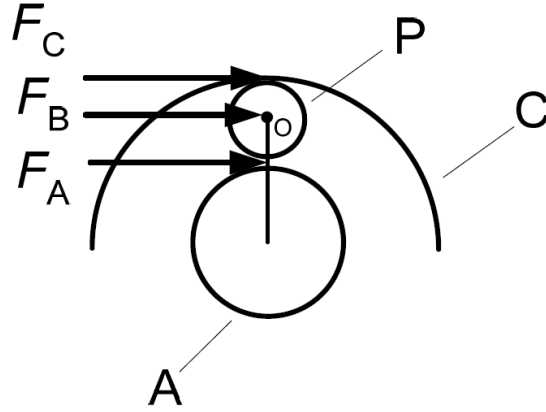


Figure 2.5: Force diagram on the PGT.

The sum of moments about the center of any of the planet elements is

$$\sum M_0 = 0, \quad (2.4)$$

and can be written in the following form:

$$\sum M_0 = F_C \cdot \frac{D_P}{2} - F_A \cdot \frac{D_P}{2}, \quad (2.5)$$

where D_P is the diameter of the planet gear.

Relating the force on the i^{th} planet element of the PGT, F_i to the torque on that element, T_i we have,

$$M_i = F_i \cdot \frac{D_i}{2}, \quad (2.6)$$

where D_i is the diameter of the i^{th} element of the PGT.

Considering the conservation of moment in the PGT,

$$M_A + M_B + M_C = 0, \quad (2.7)$$

Eqs. (2.5), (2.6), and (2.7) yield:

$$\begin{aligned} M_B &= (i_0 - 1) \cdot M_A, \\ M_C &= -i_0 \cdot M_A. \end{aligned} \quad (2.8)$$

The PGT is a statically determinant system, and the torque transmitted by any element of the PGT can be completely described by any other one torque, as shown in Equation (2.8). The magnitude and direction of power transmitted by

the i^{th} element of the PGT is the product of the angular velocity ω_i and torque M_i :

$$P_B = \left(i_0 \cdot \frac{\omega_C}{\omega_A} - 1 \right) \cdot P_A, \quad (2.9)$$

$$P_C = \left(-i_0 \cdot \frac{\omega_C}{\omega_A} \right) \cdot P_A.$$

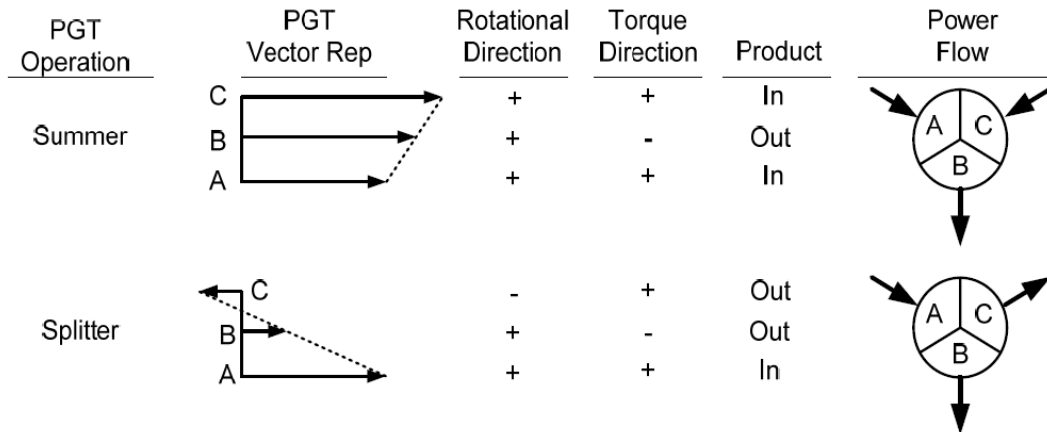


Figure 2.6: PGT Power flow analysis.

Equation (2.9) states that the PGT power splitting is dependent on the relative rotational direction of each PGT element, as depicted in Figure 2.6, below.

The power summing and splitting capability of the PGT is the fundamental principal behind hydromechanical transmissions.

2.2.2 Hydrostatic continuously variable transmissions

Hydrostatic transmissions operate in a manner similar to that of a flow control. In a flow-controlled system whatever flow leaves one unit must immediately pass through the second unit. During the operation, a given combination of engine speed and pump displacement will produce a certain flow rate. This flow will then pass through the motor, which is coupled with the axle/wheels. The generated wheel speed will thus be a function of the developed flow rate, the motor displacement, and the associated volumetric losses. The required wheel torque is a function of speed and acceleration, and it is back-

propagated to the HST where, along with motor displacement and hydromechanical losses, it determines the system pressure.

The HSTs are typically controlled using sequential, as opposed to simultaneous, control. In the sequential control, only one unit is adjusted at a time, with the second unit remaining at full displacement (Figure 2.7).

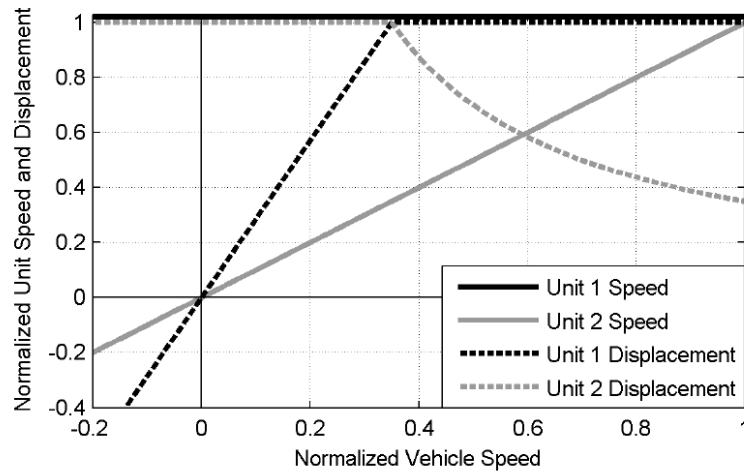


Figure 2.7: Sequential displacement control.

HSTs operate in a closed control circuit configuration; that is, a configuration where the flow recirculates predominantly between the two units. A characteristic of closed circuit configurations is that both primary transmission lines can operate at high pressure. During the vehicle motion, one HST line (for example, line A) operates under high pressure, while the other line (line B) operates at low pressure. During braking, Unit 2 changes from motoring to pumping mode and pressurizes line B, while line A reverts to low pressure.

Another characteristic of closed-circuit configurations is that the primary transmission lines are always pressurized to a certain nominal low pressure (e.g., 30 bar). Maintaining this minimum pressure is essential for preventing cavitation within the system, as well as for increasing the overall transmission stiffness and response.

2.3 The hydromechanical transmission

A complete theory on hydromechanical drives was developed by Kress (1968). According to him, these types of transmissions can be divided into two fundamental groups, namely the input-coupled (IC) and the output-coupled (OC) group. The input-coupled HMT is characterized by the mechanical coupling of the HST with the input shaft. The output-coupled HMT is characterized by the mechanical coupling of the HST with the output shaft.

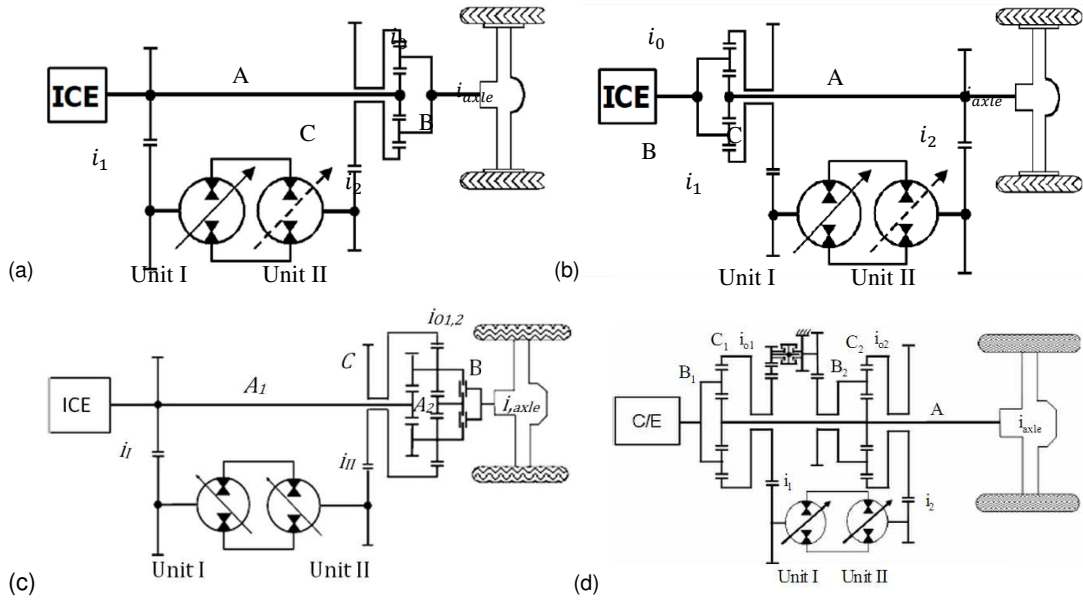


Figure 2.8: HMT architectures. (a) Input-coupled HMT, (b) Output-coupled HMT, (c) Dual Stage HMT, and (d) Compound HMT.

However, starting from the simple IC and OC architectures, more complex configurations of hydromechanical transmissions can be developed. Essentially, Kress defined complex HMT configurations as the transmissions, which realize an intermediate discrete gearshift.

More specifically, the complex configuration of the simple IC is called Dual Stage HMT, and the complex configuration of the simple OC is called Compound. Figure 2.8 shows the previously mentioned four HMT architectures.

In this section, the simple IC, OC, and Dual Stage HMT will be described in detail.

The HMT has three modes of operation: the power additive mode, the full mechanical mode, and the power recirculation mode. The mode in which the HMT operates depends on the rotation of the PGT elements, as shown in Figure 2.9.

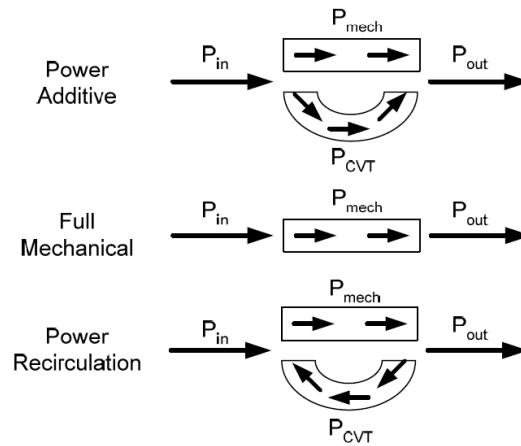


Figure 2.9: Power flows in the HMT.

When the PGT integrates power, the input-coupled HMT operates in the power-additive mode, and the output-coupled HMT operates in the recirculation mode. When the PGT acts as a power splitter, the input-coupled HMT operates in recirculation mode, and the output-coupled HMT operates in additive mode.

During the power-additive mode, the power from the input shaft, P_{in} , is split to the mechanical power, P_{mech} , and the CVT power, P_{CVT} . The two powers are added up before being transmitted to the output shaft, P_{out} .

In the fully mechanical mode, all power is transmitted mechanically. The fully mechanical mode occurs when every C element (the ring gear) of the PGT is locked, apart from the CVT; thus, P_{CVT} is zero.

During the power recirculation mode, the power is split before being transmitted to the output shaft, and is summed back at the input shaft. The mechanical path receives power from the CVT.

2.3.1 The Input-Coupled HMT

The following section describes the fundamental characteristics of the IC system. One configuration of the IC drive is characterized by the coupling of the HST with the input shaft, as shown in Figure 2.10. Alternative configurations may exist, and they are explained in the section 4.2.

Renius and Resch (2005) have described the order of operational modes as follows.

0. Recirculation (reverse)
1. Recirculation (forward)
2. Full mechanical
3. Additive

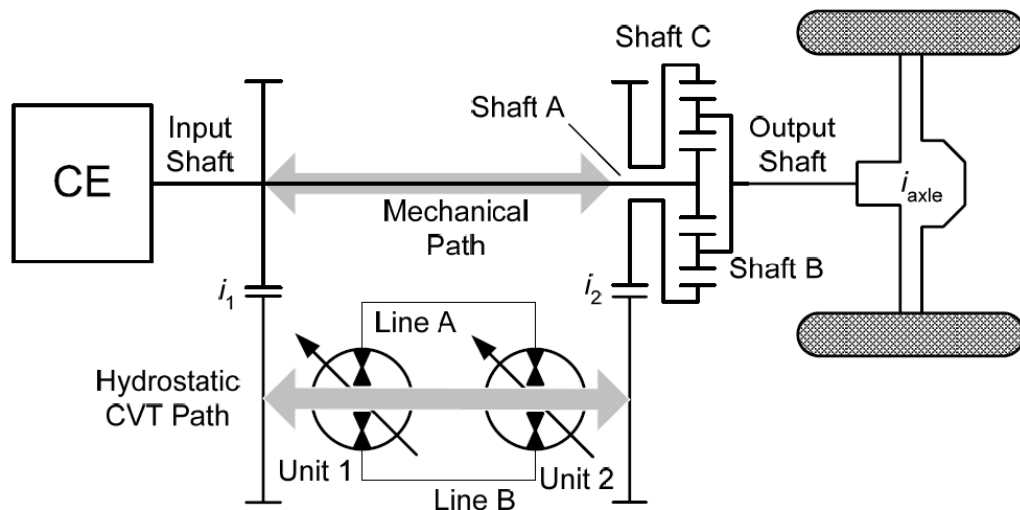


Figure 2.10 : IC hydromechanical transmission.

From standstill velocity up to a developed velocity, the input-coupled HMT starts its operation in a full power-recirculation mode, where all the power from the input shaft is recirculated between the mechanical path and the hydrostatic path. The displacement of both hydraulic units is maximum. Unit 1 is in motoring mode, and its displacement, V_1 , decreases to increase the vehicle speed.

As the vehicle speed increases, less power is recirculated hydraulically through the HST, and power is transferred from the engine to the wheels.

When the displacement of Unit 1 becomes zero, the point of full mechanical operation has been reached. At the speed corresponding to full mechanical point, v_{mech} , all the power is transferred mechanically through shaft A to the PGT. Unit 1 is now in pumping mode, and its displacement increases again to its maximum value.

As the vehicle speed increases above the value of v_{mech} , the amount of power through the hydrostatic path increases, and the IC drive is in power-additive mode. Unit 2 is in the motoring mode, and its displacement, V_2 can be decreased while is operating in power additive mode to the further increase vehicle speed.

To achieve reverse (negative) velocities, the displacement of Unit 2 decreases, thus requiring Unit 2, and therefore, shaft C to increase the angular velocity. Consequently, shaft B rotates in the reverse direction, according to Figure 2.12.

Figure 2.11 describes the sequential control scheme of the hydraulic displacements according to Renius and Resch (2005), showing the relative displacement of each unit as a function of the vehicle speed (the reverse speed is not displayed). The engine speed ω_{CE} is assumed constant.

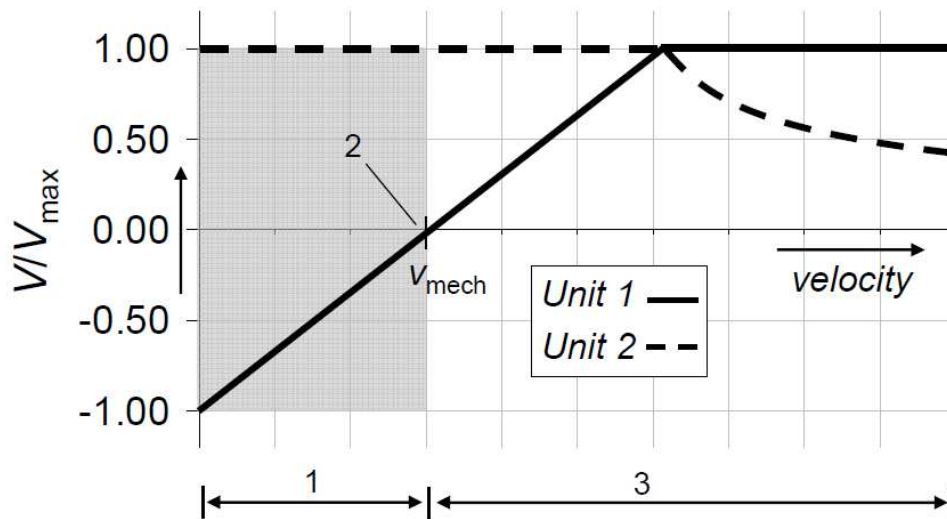


Figure 2.11: Hydraulic displacement sequential control for the IC HMT.

The shaded velocity range, marked as “1”, corresponds to the power recirculation mode, related to diagram (1) in Figure 2.12. The speed annotated with number “2” is the fully mechanical point v_{mech} , corresponding to diagram (2) in Figure 2.12. The third velocity range corresponds to the power-additive mode, corresponding to diagram (3) in Figure 2.12.

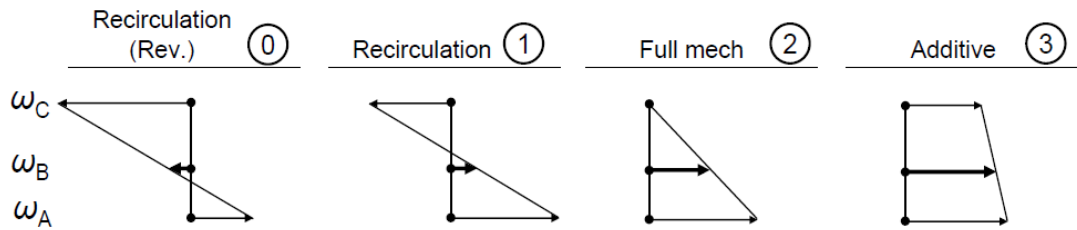


Figure 2.12: PGT operation of the IC hydromechanical transmission.

The full mechanical point operation is reached at point 2 in Figure 2.12, which is the point corresponding to the highest operating efficiency of the IC drives. The IC transmission overall efficiency is represented as the ratio of the output power delivered to the wheels to the input power from the engine. The overall efficiency of the input-coupled transmission, η_{HMT} , is given as:

$$\eta_{HMT} = \frac{P_{OUT}}{P_{IN}}. \quad (2.10)$$

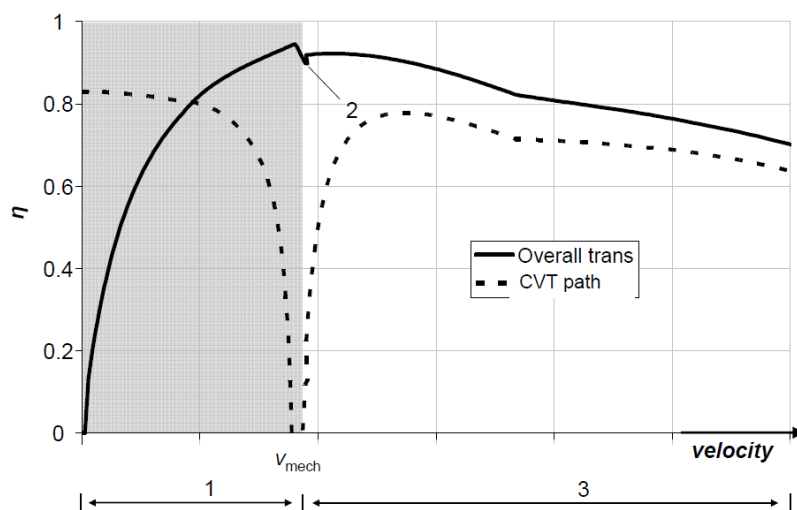


Figure 2.13: Input-coupled HST efficiency. (Kress, 1968)

The overall transmission efficiency can be approximately described as a function of the vehicle speed and the operational mode, as shown in Figure 2.13.

According to Kress (1968), the efficiency of the entire mechanical path is assumed to be 98%, and that of the hydrostatic path is assumed to be 80%. The transmission efficiency η_{HMT} increases as the transmission operating point approaches the full mechanical point (point 2), as a decreasing amount of power is transferred via the HST path. As the transmission operating point moves away from the full mechanical point, the global efficiency falls off as an increasing amount of power is transferred through the hydrostatic path.

According to Blake et al. (2005) in designing the input coupled transmission, the first decision to be made is the full mechanical point speed, and it depends on the choice of the axle ratio i_{axle} , as described in Equation (2.10).

$$v_{mech} = \frac{n_{CE}}{(1 - i_0)i_{axle}} R_{tire} \cdot 3.6 \cdot \frac{2\pi}{60} \quad (2.10)$$

The gear ratio between Unit 2 and the ring gear depends on the maximum speed that hydraulic unit 2 can achieve, and is given by

$$i_2 = n_{2,max} \cdot i_0 \cdot \left[\frac{v_{max} \cdot i_{axle}}{R_{tire} \cdot 3.6} (i_0 - 1) \cdot \frac{60}{2\pi} + n_{CE} \right]^{-1} \quad (2.11)$$

The gear ratio between the engine and Unit 1 depends on the maximum speed that Unit 1 can achieve.

$$i_1 = \frac{n_{CE}}{n_{1,max}} \quad (2.12)$$

The displacement of Unit 2 depends on the maximum pressure and the maximum wheel torque, according to the following equation.

$$V_2 = M_{wheel} \left(1 - \frac{1}{i_0} \right)^{-1} (i_{axle} \cdot i_2 \cdot \Delta p_{max})^{-1} \cdot 20 \pi \quad (2.13)$$

Meeting the flow requirement of Unit 2, Unit 1 is sized as in the following:

$$V_1 = V_2 \cdot \frac{i_1 \cdot i_2}{|i_0|}. \quad (2.14)$$

A more advanced input coupled HMT has been introduced by John Deere (Mistry and Sparks, 2002). The John Deere input-coupled solution involved in a complex planetary gear train, which has four input/output shafts, as shown in Figure 2.14.

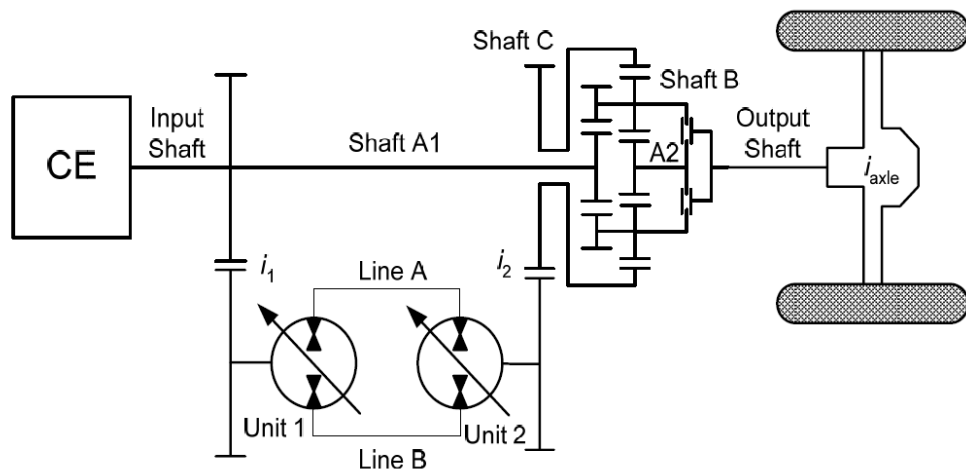


Figure 2.14: dual-stage input coupled HMT.

This power split drive has two-stage output shafts, and will henceforth be referred as dual-stage input-coupled HMT. At low speeds, shaft B is connected to the output shaft, and at high speeds, shaft A2 is connected to the output shaft. This switch between shaft B and A2 occurs at a synchronous speed, when both shaft B and A2 are rotating with equal velocities at the point of switching.

The order of the operating point for a dual-stage transmission is listed below:

1. Recirculating I
2. Full mechanical I
3. Additive I
4. Switch point
5. Recirculating II
6. Full mechanical II
7. Additive II

After the shaft switch has occurred, the transmission is operating in power recirculating mode again. Thus, the sequence of power recirculation mode, full mechanical and power-additive, occurs twice. Similar to the simple IC transmission, the dual stage can operate under a displacement sequential control, as shown in Figure 2.15.

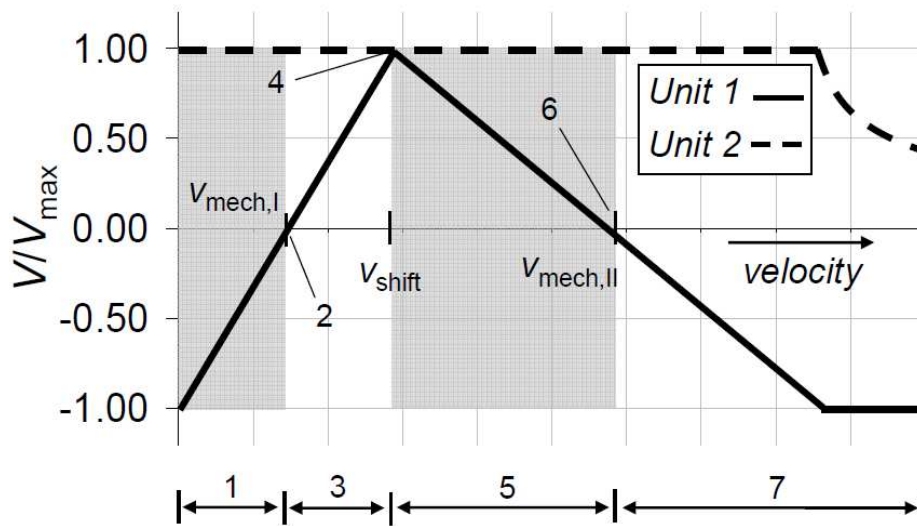


Figure 2.15: Hydraulic displacement sequential control for the Dual Stage HMT.

The shaded regions in the figure represent the speed ranges where the power recirculation occurs. The non-shaded regions represent speed ranges where the power-additive mode occurs. Region 1 through 7 of Figure 2.15 corresponds to diagrams 1 through 7 shown in Figure 2.16, wherein the kinematic behavior of the complex PGT during the dual-stage HMT operation is shown.

1. Power recirculation (reverse)
2. Power additive
3. Full mechanical point
4. Power recirculation (forward)

To reverse the direction, Unit 1 moves over the center and starts its pumping operation, while Unit 2 is in motor operation mode, at full displacement. Pumping over the center causes the oil to flow from Line A to Line B in a counterclockwise direction, resulting in Unit 2 (and consequently the vehicle) operating in reverse. A certain amount of power leaving Unit 2 is transferred to the wheels, however, as the ring gear is now rotating backwards, kinematics dictate that a certain amount of power also flows back to the PGT through the mechanical path, hence the name “recirculation”. Once back to the PGT, power from the mechanical path is combined with power from the engine before flowing back out to Unit 1 and HST path. In the power-additive mode, Unit 1 pumps from Line B to Line A in clockwise direction, whereas Unit 2 is in motor operation mode at full displacement, resulting in forward driving.

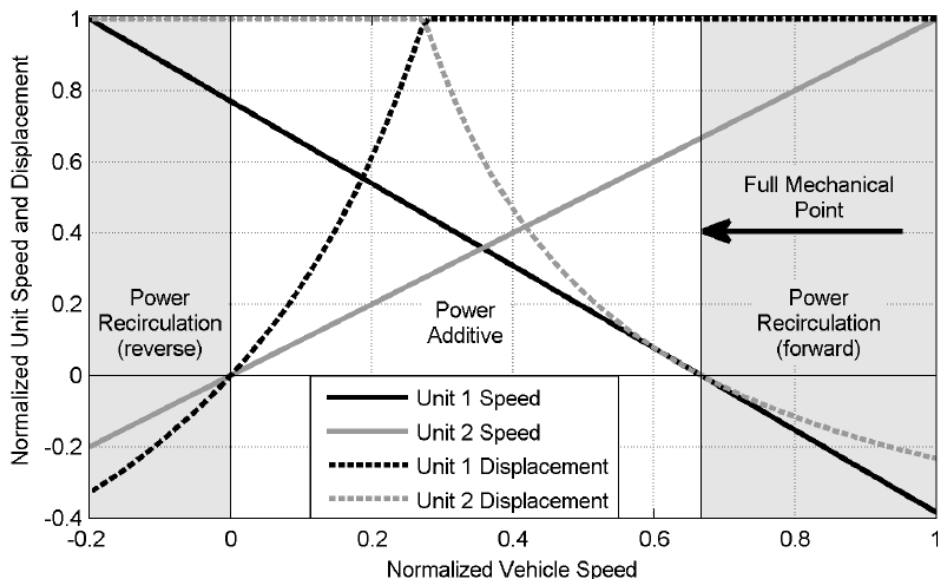


Figure 2.18: Hydraulic displacement sequential control for the Output coupled HMT.

This causes power from the engine to be split at the PGT and flow to the wheels through both the mechanical and HST paths. To increase vehicle speed Unit 1 continues to increase the displacement until both Unit 1 and 2 are at the full displacement, a point of saturation. To further increase the vehicle speed, Unit 1 remains at full displacement while Unit 2 displacement decreases. At near zero vehicle speed almost all the power flowing through the transmission transits to the HST path. However, as vehicle speed increases, kinematics cause the sun gear to slow down resulting in a great percentage of power passing through the more efficient mechanical path. As vehicle speed further increases through the power additive mode, Unit 2 will reach a condition of zero displacement. At this point Unit 1 is locked at zero velocity and Unit 2 is freewheels. Known as the full mechanical point, this is HMT's most efficient mode of operation, as the entire power flows through the mechanical path.

To increase vehicle speed over the full mechanical point, Unit 2 moves over center and begins pumping from Line B to Line A in a counterclockwise direction. Unit 1, remaining at full displacement, now begins motoring in reverse. With the sun gear rotating in reverse, PGT kinematics dictates that power from the HST path now flows into the PGT.

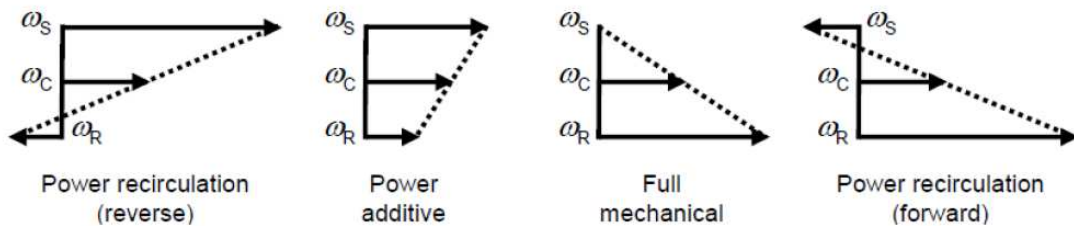


Figure 2.19: PGT operation for an output coupled HMT.

This results in some of the power flowing from the engine through the mechanical path to the wheels to be diverted through the HST path where it flows back to the PGT. As vehicle speed further increases, a greater percentage of power recirculates through the less efficient HST path, lowering the overall transmission efficiency.

To design the output-coupled transmission, the equation for the speed of the full mechanical point is given by

$$v_{\text{mech}} = \frac{n_{\text{CE}} \cdot (1 - i_0)}{i_{\text{axle}}} R_{\text{tire}} \cdot 3.6 \cdot \frac{2\pi}{60} . \quad (2.15)$$

The gear ratio between Unit 2 and the ring gear depends on the maximum speed that hydraulic Unit 2 can achieve, and is given by

$$i_2 = \frac{n_{2,\text{max}}}{v_{\text{max}} \cdot i_{\text{axle}}} \cdot R_{\text{tire}} 3.6 \cdot \frac{2\pi}{60} . \quad (2.11)$$

The gear ratio i_1 depends on: the maximum speed of Unit 1, the engine speed, and the sun speed (n_A) following relation 2.12.

$$i_1 = \frac{n_{1,\text{max}} \cdot i_0}{n_A - (1 - i_0) \cdot n_{\text{CE}}} \quad (2.12)$$

The maximum displacement of Unit 2 is given by

$$V_2 = M_{\text{wheel}} (i_{\text{axle}} \cdot i_2 \cdot \Delta_{\text{pmax}})^{-1} \cdot 20 \pi . \quad (2.13)$$

Finally, the displacement of Unit 1 is determined by the maximum engine torque and pressure:

$$V_1 = \frac{M_{\text{CE}}}{\frac{1}{i_0} - 1} \frac{1}{i_1} \cdot \frac{20\pi}{\Delta_{\text{pmax}}} . \quad (2.14)$$

The basic input- and output-coupled HMT in addition to the dual-stage HMT are the basic architectures for the current work.

3 OPTIMIZATION OF COMPONENTS AND LAYOUTS OF HYDROMECHANICAL TRANSMISSIONS

3.1 Introduction

The recent evolution of transmission systems for agricultural and work machinery led to the improvement of comfort and drivability without loss in efficiency and, where possible, increases in cost. These objectives involve the use of continuous transmissions that allow the elimination of gear shifts, which, occasionally, can be challenging, and the management of the engine at minimum specific consumption conditions. Hydromechanical transmissions meet these requirements well; therefore, they have been increasingly installed in medium-to-large machines. However, the double conversion of energy in the hydrostatic units lowers the average efficiency of the entire driveline to values below the typical efficiency of conventional mechanical transmissions. As with all continuously variable transmissions, hydromechanical transmissions separate the wheel speed from the engine speed, thus enabling optimal engine management. It has been shown that the optimal management of the engine produces beneficial effects greater than the negative effects induced by the transmission (Macor and Rossetti, 2013).

Owing to the abovementioned advantages, hydromechanical transmissions can be comparable to conventional transmissions and be successful only if the transmission is optimally designed and if the engine is optimally operated. In this study, only the optimization of the design will be investigated.

In the literature, the design has been examined in various ways, which are all based on the dimensioning of the configuration to meet power and speed requirements (Sung et al., 2005; Linares et al., 2010), occasionally followed by a simulation (Kirejczyk, 1984; Mikeska and Ivantysynova, 2002; Krauss and Ivantysynova, 2004; Blake et al., 2006; Casoli et al., 2007).

A more progressive approach was carried out by Macor and Rossetti (2011), who considered the sizing as an optimization problem, the objective function of which is to maximize the average efficiency of the transmission along with the rated speed range of the vehicle. In contrast, in the work of Rossetti and

Macor (2013), the design was a multiobjective problem, the objective functions of which are the maximization of efficiency and the minimization of bulk transmission.

Undeniably, numerous other aspects must be considered in the design procedure, such as noise. An attempt in this direction has been made by Macor et al. (2016), and is also later discussed in the respective section of the present work.

In previous studies, optimization has been applied to transmissions whose structure had been a priori assigned. However, the various layouts did not have the same performance because of different internal power flows during operation and different internal losses. Therefore, the optimization of transmission should also include the layout, and not merely the components.

For this reason, we present an optimization approach for the design of three-shaft hydromechanical transmissions; the goal of the present work is the simultaneous optimization of the structure and the components.

The structure was represented by means of graph theory, whereas the components were described by their functional models.

As the problem is non-linear, a direct-search algorithm based on the swarm theory (Particle Swarm Optimizer, PSO; Shu-Kai et al., 2014) was used to solve the optimization problem. Finally, the optimization procedure was applied to the transmission of a high-power agricultural tractor.

3.2 Model

To calculate the performance of a power-split transmission, two different approaches can be employed: the stationary and the time-variant approach. For a model to be coupled with an optimizer, the former approach seems to be more suitable because it is faster, provided the efficiency of the hydraulic machines is not considered a constant. Here, the adopted model bears these features.

The schematic of a generic transmission is illustrated in Figure 3.1, the main groups of which will be discussed below.

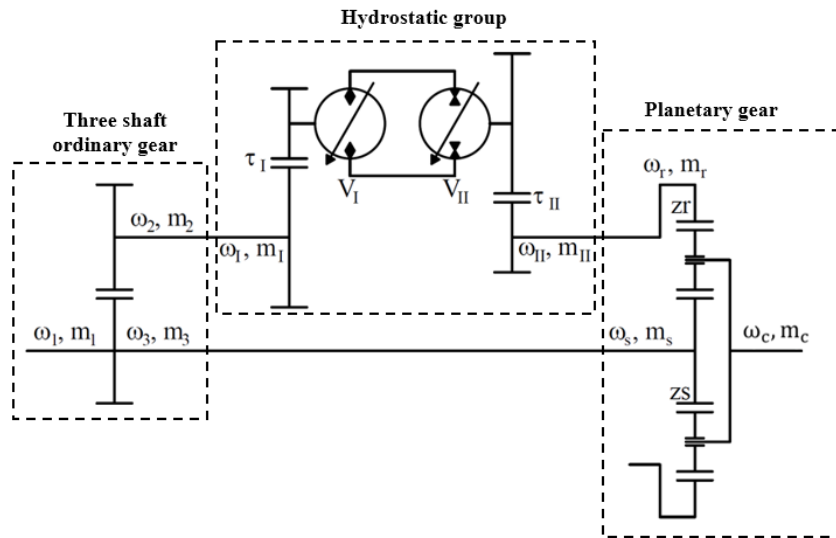


Figure 3.1: Schematic of the power-split transmission.

3.2.1 Hydrostatic group

The hydrostatic group is the continuously variable element of power-split transmissions; it consists of two variable-displacement reversible hydraulic machines (Figure 3.2).

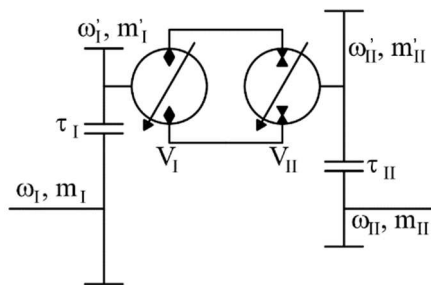


Figure 3.2: Schematic of the hydrostatic group.

By varying the two displacements, it is possible to vary the transmission ratio continuously. Two gears, τ_I and τ_{II} , are added to fit the input and output speeds to the operating range of the hydrostatic units.

The fundamental equations regulating the power transmission inside the unit, illustrated in Figure 3.2, are the continuity equation and the uniqueness of the

pressure drop of the two machines, which is derived from the equation of the torques.

The ratios τ_I and τ_{II} are defined as follows:

$$\omega'_I = \tau_I \omega_I, \quad (3.1)$$

$$\omega'_{II} = \tau_{II} \omega_{II},$$

$$[\tau_I \alpha_I V_I \eta_{vI}^k - \tau_{II} \alpha_{II} V_{II} \eta_{vII}^{-k}] \left\{ \frac{\omega_I}{\omega_{II}} \right\} = 0, \quad (3.2)$$

$$\left[\frac{(\eta_{hym I} \eta_{gear}^{nI})^k}{\tau_I \alpha_I V_I} - \frac{(\eta_{hym II} \eta_{gear}^{nII})^{-k}}{\tau_{II} \alpha_{II} V_{II}} \right] \left\{ \frac{m_I}{m_{II}} \right\} = 0, \quad (3.3)$$

where the exponent k is a function of the power flow direction. In particular, $k = 1$ if the power is transferred from unit I to unit II; $k = -1$ in the opposite case; the efficiencies η_v and η_{hym} are the volumetric and hydromechanical efficiency of units I and II, respectively, whereas η_{gear} takes into account the mechanical losses in the ordinary gears. Superscripts nI and nII represent the number of gear pairs needed to achieve gear ratios τ_I and τ_{II} ; they are estimated based on $\tau_{\max}$, which is the maximum allowable value for each gear pair:

$$n_I = \text{int}(\tau_I / \tau_{\max}) \quad (3.4)$$

$$n_{II} = \text{int}(\tau_{II} / \tau_{\max})$$

3.2.2 Mechanical Elements

The main mechanical elements inside the power-split transmission are the three-shaft ordinary gear (Figure 3.3(a)) and the planetary gear; the latter is the summation point of the speed of the three shafts (Figure 3.3(b)).

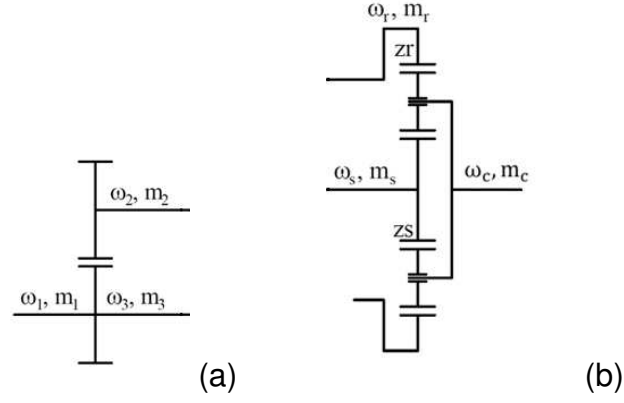


Figure 3.3: Mechanical elements: (a) schematic of the three-shaft ordinary gear, and (b) the planetary gear.

The kinematic and dynamic equations of the three-shaft ordinary gear are:

$$\begin{bmatrix} 1 & 0 & -1 \\ 1 & 1 & 0 \end{bmatrix} \begin{Bmatrix} \omega_1 \\ \omega_2 \\ \omega_3 \end{Bmatrix} = 0, \quad (3.5)$$

$$\begin{bmatrix} 1 & -1 & 1 \end{bmatrix} \begin{Bmatrix} m_1 \\ m_2 \\ m_3 \end{Bmatrix} = 0, \quad (3.6)$$

wherein the transmission ratio was assumed to be equal to one, because speed reductions or multiplication can be incorporated in the τ_I and τ_{II} gears of the CVT element. The velocity and torque equations of the planetary gear are:

$$\begin{bmatrix} (1 - T) & -1 & +T \end{bmatrix} \begin{Bmatrix} \omega_C \\ \omega_R \\ \omega_S \end{Bmatrix} = 0, \quad (3.7)$$

$$\begin{bmatrix} 1 & (1 - T\eta_0^{-t}) & 0 \\ 0 & T\eta_0^{-t} & 1 \end{bmatrix} \begin{Bmatrix} m_C \\ m_R \\ m_S \end{Bmatrix} = 0, \quad (3.8)$$

where T is the standing transmission ratio and η_0 is the efficiency of the planetary gearing. The exponent t can take values of ± 1 , and depends on the

power flow inside the gear. The values of t can be synthetically expressed as a function of the incoming sun gear power, measured with reference to the carrier, as $t = \text{sign}([m_S(\omega_S - \omega_C)])$.

3.3 Overall system of equations

Equations (3.5)–(3.8) define a system of six equations and twelve unknowns. The missing equations are the relationships that link the elements of the transmission; thus, they define the transmission structure. These equations require the equality of speed and the equilibrium of the moments between the internal components of the transmission. In the case of the transmission illustrated in Figure 3.2, these are:

$$\begin{cases} \omega_3 - \omega_S = 0 \\ \omega_2 - \omega_I = 0 \\ \omega_R - \omega_{II} = 0 \end{cases} \quad (3.9)$$

$$\begin{cases} m_3 + m_S = 0 \\ m_2 + m_I = 0 \\ m_R + m_{II} = 0 \end{cases} \quad (3.10)$$

The final constraints are the speed and the torque of the engine at the input shaft. For example, for the case presented in Figure 3.2, the constraints $\omega_1 = \omega_{\text{engine}}$ and $m_1 = m_{\text{engine}}$ allow a determined system of fourteen equations and fourteen unknowns.

The model can be summarized in a matrix form, as follows:

$$\begin{aligned} \Omega \vec{\omega} &= b_{\omega}, \\ M \vec{m} &= b_m. \end{aligned} \quad (3.11)$$

Alternatively, the equations can be grouped into three classes: the elements, the links between elements, and the links with the outside components. Therefore, the system can be expressed in the modular form:

$$\begin{bmatrix} \Omega_{\text{elements}} \\ \Omega_{\text{links}} \\ \Omega_{\text{engine}} \end{bmatrix} \vec{\omega} = \begin{bmatrix} 0 \\ \vdots \\ \omega_{\text{engine}} \end{bmatrix}, \quad (3.12)$$

$$\begin{bmatrix} M_{\text{elements}} \\ M_{\text{links}} \\ M_{\text{engine}} \end{bmatrix} \vec{m} = \begin{bmatrix} 0 \\ \vdots \\ m_{\text{engine}} \end{bmatrix}.$$

The strong non-linearity of the efficiency of the hydraulic units prevents the direct solution for the system. Consequently, an iterative solution was used, in which the efficiencies at the first iteration were set to be equal to one. The convergence criterion was assumed equal to 0.1% between two successive iterations. A flowchart of the iterative scheme is presented in Figure 3.4.

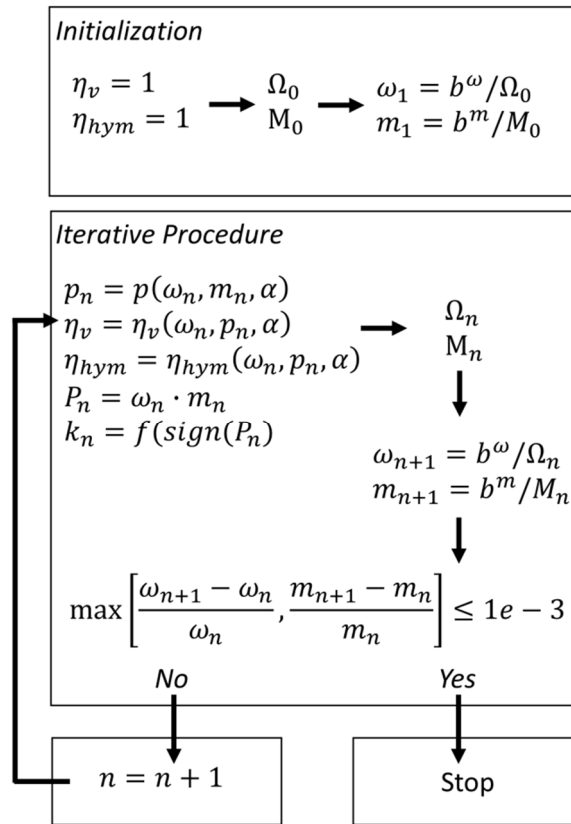


Figure 3.4: Flowchart of the iterative procedure for the solution of the system.

3.4 Optimization

The simulator was inserted into an optimization procedure that allows simultaneous optimization of both the internal components and the structure

of the transmission. The optimization procedure can be formulated in the following general algorithm:

Find $x = [x_1 \ x_2, \dots, x_n]^T$ minimizing $f(x)$, subject to constraints $g_j(x) \leq 0 \ j = 1, \dots, m$ and $l_j(x) = 0 \ j = 1, \dots, p$.

where:

- $x_1 \ x_2, \dots, x_n$ are the free optimization variables (degrees of freedom);
- the equality constraints $l_j(x) = 0 \ j = 1, \dots, p$ are the design parameters;
- the inequality constraints $g_j(x) \leq 0 \ j = 1, \dots, m$ are the design constraints; and
- $f(x)$ is the objective function.

3.4.1 Degrees of freedom

The variables of the optimization can be divided into two groups: the variables linked to the transmission structure and the component variables. The latter are:

- the displacements V_I and V_{II} ,
- the transmission ratios τ_I and τ_{II} ,
- the standing transmission ratio T of the planetary gear, and
- the transmission ratio τ_{out} of the gears (both ordinary and planetary) between the output shaft of the power-split transmission and the wheel axis.

Since the transmission structure can be represented in matrix form, as seen in Equations (3.11) and (3.12), the variables associated with the layout of the transmission are:

- the internal relationships of the transmission Ω_{links} , and
- the connection between the transmission and the engine Ω_{engine} .

These two variables were assembled in a unique matrix, which completely defined the transmission layout:

$$\Omega_{layout} = \begin{bmatrix} \Omega_{link} \\ \Omega_{engine} \end{bmatrix}.$$

3.4.2 Design parameters

The design parameters supplied as input to the optimization procedure are:

- the internal combustion engine (ICE) design power (which is, typically, close to the optimum efficiency power),
- the design speed of ICE,
- the maximum speed of the vehicle,
- the maximum wheel pulling force, and
- the wheel radius.

3.4.3 Design constraints

The optimization procedure must be forced to satisfy certain design and structural constraints, namely

- the maximum pressure of hydraulic machines: $\Delta p \leq \Delta p_{\max}$;
- the minimum and maximum transmission ratios of the planetary gear:
 $T_{\min} \leq T \leq T_{\max}$;
- the minimum and the maximum volume allowed for the units displacement: $V_{\min} \leq V_I \leq V_{\max}$;
- the maximum speed of the hydraulic units as a function of displacement:
 $\omega'_I \text{ and } \omega'_I \leq \beta(\alpha)\omega_{\max}(V)$.

The maximum speed of the hydraulic units was considered a function of the nominal displacement, $\omega_{\max}(V)$. Because the maximum speed of the units refers to the full displacement condition, an overspeed factor $\beta(\alpha)$ was added to model the increase in the maximum speed under the partial displacement conditions ($\alpha < 1$).

The constraints were implemented according to penalty functions, which add a quantity proportional to the exceeded distance from the allowed domain.

3.4.4 Objective function

The objective function is the integral average loss calculated between the zero speed and the maximum design speed of the vehicle:

$$\text{fitness} = \overline{\eta_{\text{driveline}}} = \frac{1}{v_{\text{max}}} \int_0^{v_{\text{max}}} \eta_{\text{driveline}} dv, \quad (3.13)$$

where $\eta_{\text{driveline}}$ is the transmission efficiency, defined as the ratio of the power delivered at the wheels to the engine power is given by:

$$\eta_{\text{driveline}} = \frac{P_{\text{wheel}}}{P_{\text{engine}}}. \quad (3.14)$$

3.5 Optimization algorithm

The optimization of the system was structured according to a hierarchical logic, by dividing the structural variables (i.e., Ω_{links} and Ω_{engine}) from the internal variables. This need comes from the lack in continuity and order of the set of the possible structures of a transmission. In fact, while the optimization of internal parameters is possible via classical algorithms or a direct-type search, the lack of an order relation between the possible values of Ω_{links} and Ω_{engine} does not allow the definition of the sum or of other algebraic operations based on which are all the advanced optimization algorithms. Given that the set of configurations is finite, it is still possible to proceed to the optimization via the enumeration of the possible configurations. Therefore, the optimization was performed according to the following scheme.

All the possible configurations of a three-shaft transmission were initially defined; for each of these configurations, a process of optimization of the internal variables was carried out to obtain the value of the local optimum of that configuration. Finally, the global optimum was obtained by considering the best value among the local optima identified.

3.5.1 Definition of the possible configurations

In order to enable an effective enumeration and storage of the possible configurations, an abstract representation based on the graph theory was used.

A graph is a mathematical structure consisting of nodes connected by arcs. In the case in question, the nodes represent the transmission elements, such as a pump, a motor planetary gear; the arcs represent the shafts. The number of arcs that connect to a node is called the degree of the node. Five nodes were identified:

- 1 node–degree 2, corresponding to the CVT variator, represented by the symbol Δ ;
- 2 nodes–degree 3, corresponding to the ordinary gear (symbol Ξ) and the planetary gear (symbol Θ); and
- 2 nodes–degree 1, corresponding to the transmission input (I) and the output (Ω).

Following this nomenclature, IC and OC transmissions can be represented as in Figure 3.5.

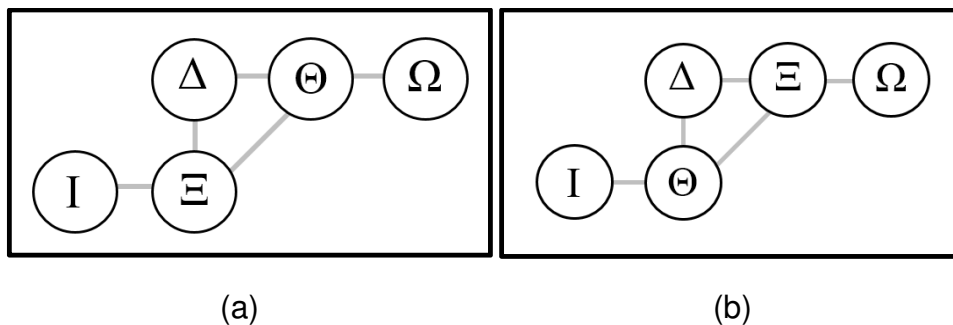


Figure 3.5: Base graph for the IC configuration (a) and the OC configuration (b).

The definition of the possible graphs was obtained through the following steps:

1. Identification of possible graphs;
2. Removal of meaningless graphs (Figure 3.6), such as:
 - physically undetermined graphs (Figure 3.6(a));

- mechanically wrong (when the CVT is connected to an input or an output, i.e., Figures 3.6(b) and (c))

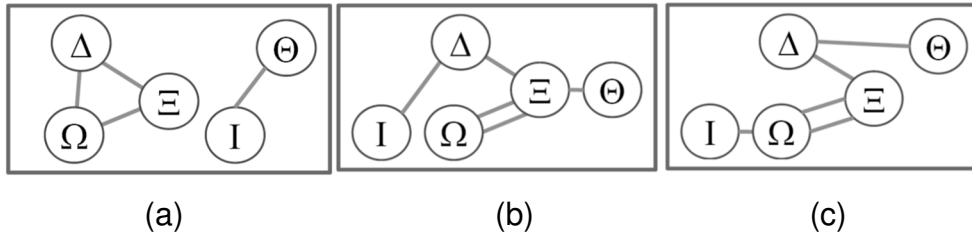
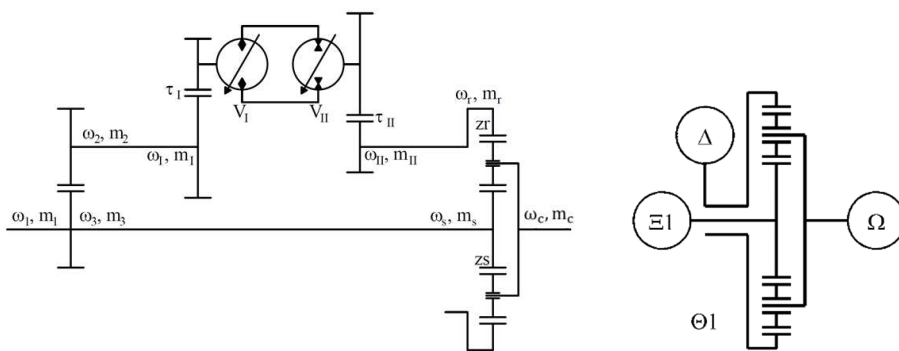


Figure 3.6: Graphs physically not acceptable or not significant

This procedure obtained a known, nevertheless nontrivial, result: the only meaningful graphs for a three-shaft layout are the IC and OC of Figure 3.5.

In searching for possible graph configurations, the asymmetry of the shafts of the epicyclic gear must be taken into account. Therefore, together with the graphs of Figure 3.5, several other combinations must be considered. For example, Figure 3.7 shows two possible configurations of the base graph of IC, which differ in the connections of the planetary gear.

Owing to the fact that we consider three-shaft planetary gears, the number of possible permutations of the connection with the remaining elements is six. Thus, any graph generates $6^{n_{\text{epi}}}$ configurations, where n_{epi} is the number of planetary gears in the transmission. Therefore, graphs of the possible configurations are six for the IC-graph-based and six for the OC-graph-based.



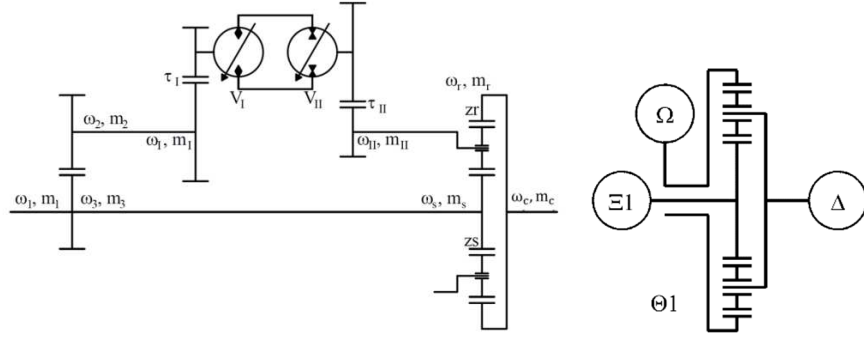


Figure 3.7: Different layouts for the IC configuration and their representations.

3.5.2 Particle Swarm optimizer

Because the objective function of the optimization is not directly expressed through an analytical formulation, it was decided to resort to optimization algorithms of the direct-search type. Furthermore, the particular form of the objective function, which probably presents several local minima, suggests the use of evolutionary algorithms, which are able to reduce the importance of the initial research point, as well as the trapping in local minima that are far from the absolute minimum.

Search algorithms based on swarms (Particle Swarm Optimizer, PSO) are stochastic, based on the social behavior of bird flocks and insects swarms, in which information acquired by an individual and the sharing of this information with other individuals leads to numerous evolutionary advantages, including improved efficiency in the search for food.

The most important feature of the particle swarm algorithm is the process that moves the particle of the swarm (initialized in random position in the first iteration), between two subsequent iterations. The position of the i^{th} particle in the space of the allowable solutions at iteration $n+1$ is obtained by summing a displacement d to the position x of the same particle at the n^{th} iteration, as in the following.

$$x_{n+1}^i = x_n^i + d_{n+1}^i \quad (3.15)$$

The displacement is computed according to the following expression:

$$d_{n+1}^i = w \cdot d_n^i + c_1 \cdot rand(y_n^i + x_n^i) + c_2 \cdot rand(y_n + x_n^i). \quad (3.16)$$

The displacement includes three terms, which model the prime basis on the social behaviors.

The first term models the resistance to changes; individuals tend to repeat their behavior and maintain their opinions, even if new environmental changes suggest modification thereof. This term is ruled by the inertial coefficient w , which is multiplied with the displacement at the previous iteration.

The second term represents the tendency of any individual to move toward the position of best fitness, according to his/her own experience. The position y_n^i is, in fact, defined as the best position discovered by the i^{th} particle from the beginning of the optimization to the current iteration. This component is weighted by the so-called “self-confidence” coefficient c_1 and the uniformly distributed random variable $rand$, which can receive any value between 0 and 1.

The third term models the interaction between individuals, such as knowledge sharing and emulations. The goal of this part of the displacement is to move toward position y_n , which is the position of best fitness to be discovered by the swarm. This component is weighted by the so-called “swarm confidence” coefficient c_2 and the random variable $rand$.

The optimizer used in the present study, originally presented by Shu-Kai et al. (2014), combines the classical formulation of PSO with the Nelder–Mead algorithm (Nelder and Mead, 1965). This formulation improves the efficiency of local convergence, keeping the research capabilities of the algorithm unchanged. Such an algorithm was successfully applied to hydromechanical transmission optimization problems by Macor and Rossetti (2011).

3.6 Optimization procedure application

The optimization procedure previously discussed was implemented in MATLAB, and was applied to the transmission of a high-power agricultural tractor. The overall dimensions and the main data of the vehicle are presented

in Figure 3.8 and Table 3.1, respectively. The optimization constraints are summarized in Table 3.2 and Figures 3.9 and 3.10.

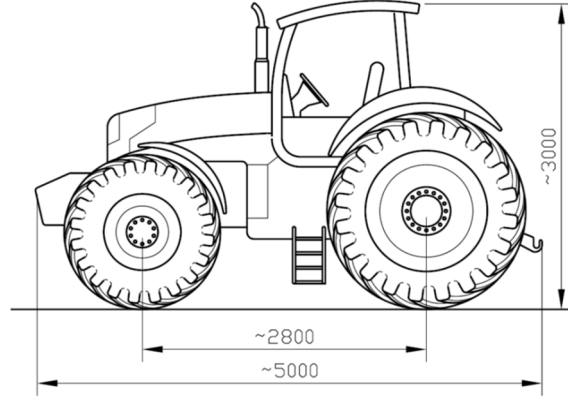


Figure 3.8: Overall dimensions of the tractor.

ICE Power	180 kW
ICE rated speed	2,200 rpm
Wheel radius	0.965 m
Maximum traction force	120 kN
Maximum speed	40 km/h

Table 3.1: Main data of the agricultural tractor

In order to contain the size of the volumetric machines, given the high value of the traction force at zero velocity, a two-speed gearbox downstream of the CVT transmission was adopted. Its gear ratio was assumed to be 4:1 at lower speed and 1:1 at higher speed. The gearshift operation was carried out when the transmission efficiency in the second gear was higher than that in the first gear.

$$\eta_{\text{Driveline}}^{\text{1st gear}}(v_{\text{change}}) = \eta_{\text{Driveline}}^{\text{2nd gear}}(v_{\text{change}}) \quad (3.17)$$

Consequently, the objective function could be rewritten in the following form:

$$e(x) = \frac{\int_0^{v_{\text{change}}} (1 - \eta_{\text{Driveline}}^{\text{1st gear}}) dv + \int_{v_{\text{change}}}^{v_{\text{max}}} (1 - \eta_{\text{Driveline}}^{\text{2nd gear}}) dv}{v_{\text{max}}}. \quad (3.18)$$

		Min	Max
Hydrostatic group			
Displacement	V	28 cc	250 cc
Pressure	$\Delta p_{\max}$	-	400 bar
Maximum speed	$v_{\max}$	0	Figs. 3.9 and 3.10
Epicyclical gear			
Standing gear ratio	T	-1/2	-1/7

Table 3.2: Optimization constraints.

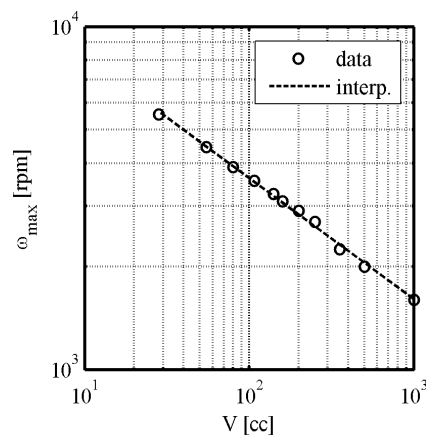


Figure 3.9: Maximum speed at maximum displacement as a function of the displacement.

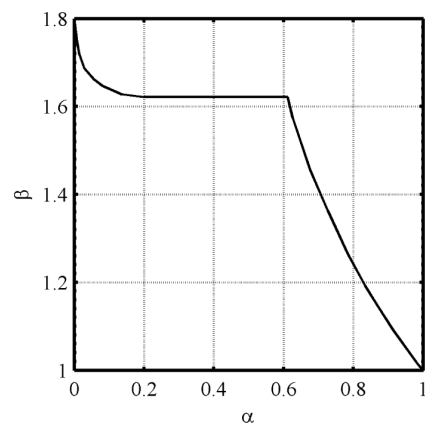


Figure 3.10: Increase factor of the allowable speed as a function of the partialization.

Regarding the efficiency laws for the hydraulic machines and the efficiency of all the gear pairs established by the model (section 3.2), the following assumptions were made.

The efficiencies η_v and η_{hym} of the hydraulic machines were expressed as a function of the operating variables ($\omega, \alpha, \Delta p$) via the following expressions:

$$\eta_v = \eta_{ref} \cdot \chi_{\omega}^v \left(\frac{\omega}{\omega_{max}} \right) \cdot \chi_{\alpha,p}^v \left(\alpha, \frac{\Delta p}{\Delta p_{max}} \right), \quad (3.19)$$

$$\eta_{hy} = \eta_{ref} \cdot \chi_{\omega}^{hy} \left(\frac{\omega}{\omega_{max}} \right) \cdot \chi_{\alpha,p}^{hy} \left(\alpha, \frac{\Delta p}{\Delta p_{max}} \right),$$

where η_{ref} is the reference efficiency value, assumed to be 0.96 for both efficiencies, reduced by coefficients χ_{ω} and $\chi_{\alpha,p}$. The former coefficient depends on the speed, and the latter depends on the pressure and the displacement, as shown in Figures. 3.11–3.14. These curves were calculated based on the experimental curves of commercial hydraulic machines.

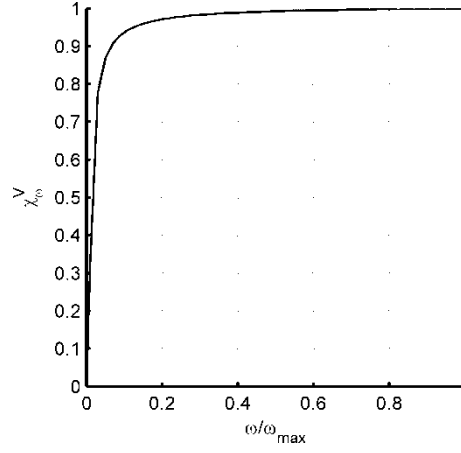


Figure 3.11. Effect of the speed on the volumetric efficiency.

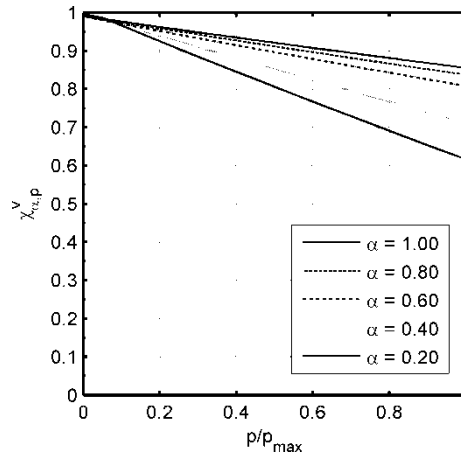


Figure 3.12. Effect of the pressure and displacement partialization on the volumetric efficiency.

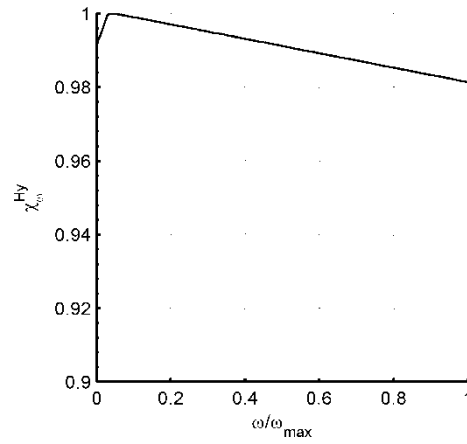


Figure 3.13. Effect of the speed on the hydromechanical efficiency.

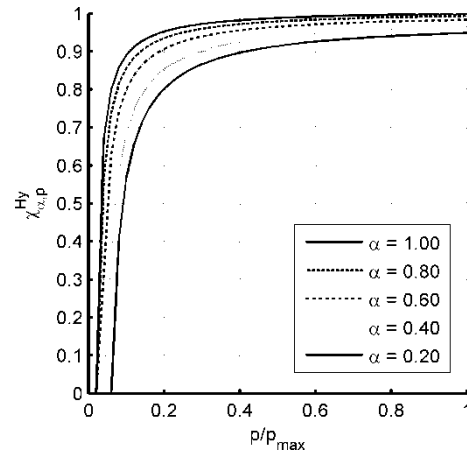


Figure 3.14. Effect of the pressure and displacement partialization on the hydromechanical efficiency.

The efficiencies of each gear pair η_{gear} and of the planetary gear η_0 were assumed equal to 0.985 and 0.98, respectively. The maximum gear ratio for each gear pair τ_{max} (Equation (3.4)) was assumed equal to 2.

The rear axle gear was assumed to be formed by a differential gear ($\eta = 0.975$) and by a planetary gear reducer ($\eta = 0.985$).

The full description of viscous and friction losses on the bearings and of ventilation and splash oil losses of the overall transmission would lead to a very complex model. Furthermore, a detailed description of the losses requires a full description of the shafts, gears, and bearings involved, which

could not have been known when preliminary design considerations were drawn. For this reason, the friction losses were neglected, as they do not substantially interfere with the optimum definition, despite changing (slightly) the overall efficiency. Nevertheless, preliminary tests showed that viscous effects have a strong influence on the optimum design choice. In the absence of viscous effects, the optimum configuration will move toward a high-speed output shaft, to reduce the momentum losses on the axle gears. On the contrary, the presence of viscous effects forces the shaft speed to reduce to a reasonable value, thus defining a “physical” constraint to the shaft speed.

Viscous losses were then included, using a very simple model. Because the input shaft speed is fixed, and the mechanical losses of the hydraulic unit were modeled inside the unit sub-system, the viscous losses on the mechanical path were considered only on the output shaft. Because the number and the disposition of the shafts of the axle gearbox are not known, for simplicity, the losses were considered to be caused by the shaft entering the axle gear reducer. The value of $0.10 \text{ [Nm}\cdot\text{rad}^{-1}\cdot\text{s}]$ was shown to lead to consistent results for the considered test case.

3.7 Results

The code enabled the evaluation of all the different layouts of the IC and OC configurations. The results for the IC configuration are concisely summarized in Table 3.3, whereas the values of the optimization variables for the solutions are listed in Table 3.4; Figure 3.15 shows the efficiency as a function of the vehicle speed at the rated power of the engine.

The same results are presented in Tables 3.5 and 3.6 and in Figure 3.16 for the OC configuration.

First, not all layouts meet the traction requirements of the vehicle, such as the IC-1, IC-3, and IC-5 layouts for the IC configuration (Table 3.3), and the OC-2, OC-4, and OC-5 layouts for the OC configuration (Table 3.5).

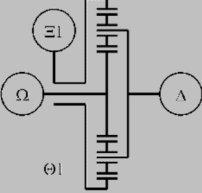
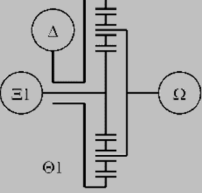
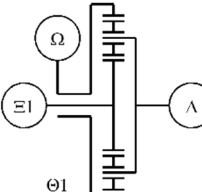
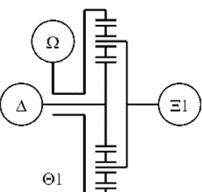
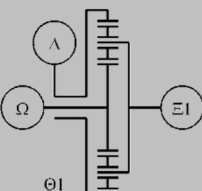
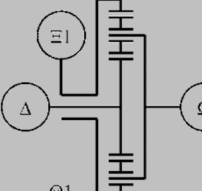
Input Coupled	$\max(\eta_{\text{mean}})$	Input Coupled	$\max(\eta_{\text{mean}})$
	$[-]$		0.847
	0.825		$[-]$
	$[-]$		0.704

Table 3.3: Optimal efficiencies for the IC configurations.

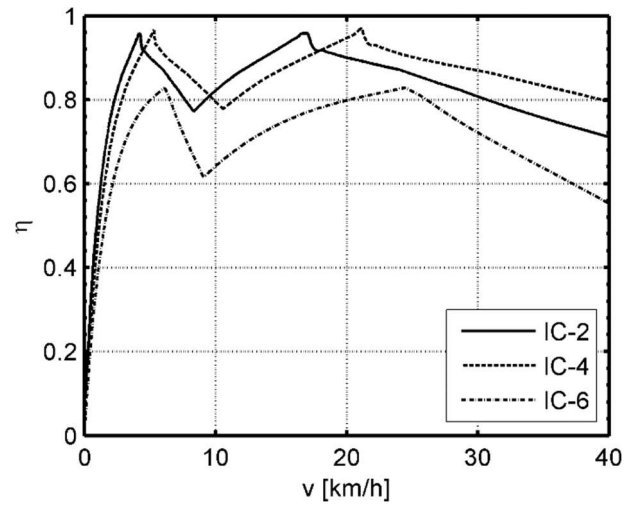


Figure 3.15: Total transmission efficiency of the IC layouts.

	IC-2	IC-4	IC-6
T	-0.15	-0.2	-0.5
V _I	28 cc	54 cc	244 cc
V _{II}	250 cc	250 cc	198 cc
τ_I	2.19	1.77	1.15
τ_{II}	4.0	3.90	0.72
τ_{out}	28.7	24.5	84.5
η_{mean}	0.825	0.847	0.704

Table 3.4: Optimal values of the design variables for the IC layouts.

The IC configuration reaches its maximum value of efficiency with the IC-4 layout, i.e., with the CVT section connected to the ring, the solar connected to the ordinary gear, and the carrier connected to the output..

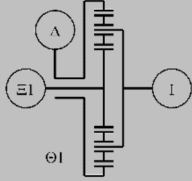
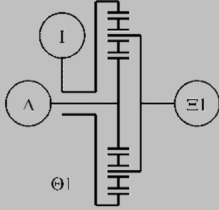
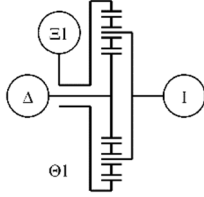
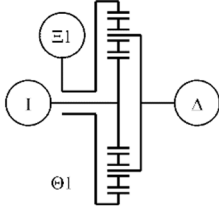
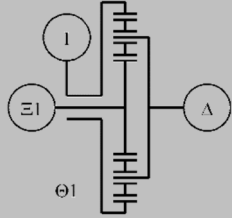
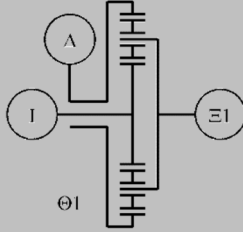
Output Coupled	$\max(\eta_{mean})$	Output Coupled	$\max(\eta_{mean})$
	.708		[-]
	[-]		[-]
	.682		.770

Table 3.5: Optimal efficiencies for the OC configurations

The optimal solution for the OC configuration is OC-5, i.e., with the CVT section connected to the ring, the solar connected to the engine, and the carrier connected to the ordinary gear.

The efficiency of the two optimal solutions is compared to that of the hydrostatic group in Figure 3.17.

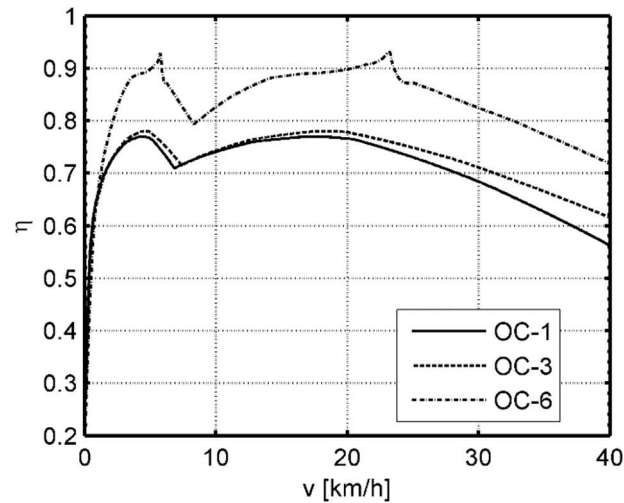


Figure 3.16: Total transmission efficiency of the OC layouts

The IC superiority is not only owing to the intrinsic qualities of the transmissions; in fact, as shown in Figure 3.17, the hydrostatic group of the IC configuration, which is the main source of losses, has a lower efficiency than that of OC.

	OC-1	OC-3	OC-6
T	-0.5	-0.5	-0.5
V_I	250	250	205 cc
V_{II}	117	248	250 cc
τ_I	1	1	1.04
τ_{II}	1	1.03	1.05
τ_{out}	81.3	73.8	45.7
η_{mean}	.708	.682	.770

Table 3.6: Optimal values of the design variables for the OC layouts

This implies that there are other losses, not pertaining the transmission. These are the viscous losses in the output shaft. In fact, the transmission ratio τ_{out} of the OC solution is higher than that of the IC solution; this means that the speed of the output shaft is higher for the OC. Consequently, the losses will also be higher, which vary with ω^2 .

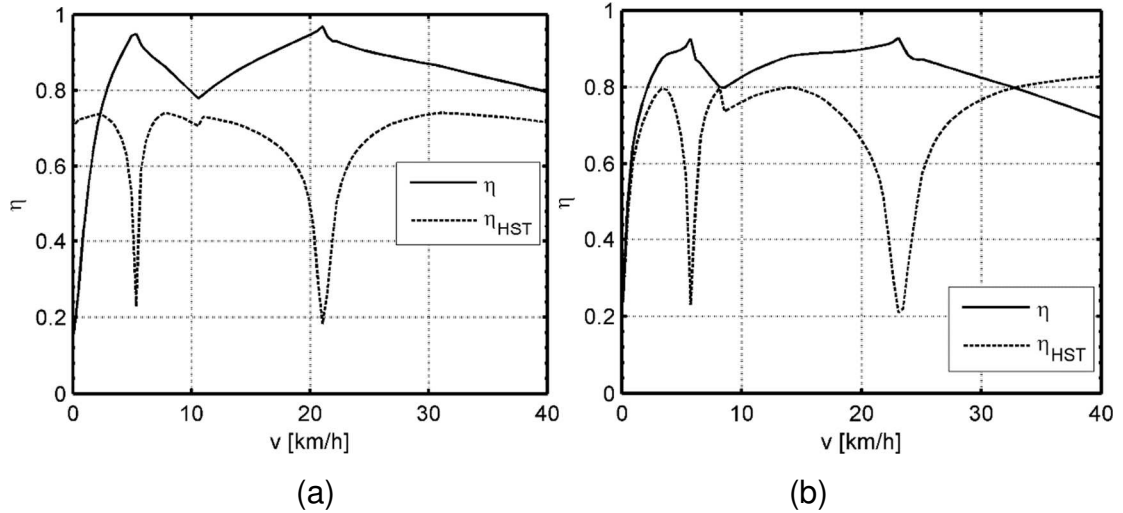


Figure 3.17: Total transmission efficiency and efficiency of the hydrostatic group;
a) optimal IC layout; b) optimal OC layout.

Incidentally, the optimal displacements in Tables 3.4 and 3.6 are greater than those currently in use on tractors of similar power.

The initial choice of ratios of 4:1 and 1:1 for the two-speed gearbox was found to be consistent with the typical use of the vehicle: in fact, the transmission presents the best efficiency (greater than 85%) within the range of low speeds (4–8 km/h), typical of plowing operations on the field, and for high speeds (20–30 km/h), typical of road transport. The trend of the efficiency as a function of speed is also significantly higher than the standards proposed by Renius and Resch (2005) for agricultural CVT transmissions.

4 SIMULTANEOUS OPTIMIZATION OF LAYOUT AND COMPONENTS

4.1 Introduction

The hydromechanical transmissions are generally adopted for vehicles, such as tractors, earthmoving machines, and for high-power vehicles in general, owing to their capability to achieve a continuously variable speed, high flexibility, and high comfort level. However, one of the disadvantages of these transmissions is the low efficiency, which could be addressed with an optimized design procedure. In fact, the traditional design procedure commonly used to develop these systems does not consider the difference between the HMT layouts in terms of global efficiency. Therefore, the global efficiency changes by changing the full mechanical point speed and the planetary gear train ratio, which are variables that are typically assumed in a traditional design procedure. In this study, an optimization design method for the input-coupled HMT is explained, which considers all layouts of the transmission as feasible solutions. Knowing the general data of a high-power vehicle, the objective of the design optimization is to find the best global HMT efficiency, considering the variability of the layout, the full mechanical speed, and the planetary gear train ratio. In the first section, a “manual search” optimization (sensitivity analysis) was developed, in order to assess how the efficiency changes within the full range of the design variables.

The model of the transmission layouts has been implemented using the simulation software Amesim, and the optimization procedure was performed using a MATLAB code. The optimization procedure has been performed for an urban passenger bus as the reference vehicle, and the results indicated that the optimal global efficiency was reached at low values of the full mechanical point speed and for the sixth layout, showing at the end of the procedure the design variables and parameters which permit that specific solution. The result is also in agreement with the literature (Blake et al., 2006).

4.2 Hydromechanical transmission layouts

The term “layout” is used to indicate the geometry of the transmission in terms of connections between the components. The most widely known IC layout for the hydromechanical transmission is shown in Figure 4.1, where the engine is connected to the sun gear of the planetary gear train, the HST is connected to the ring gear, and the wheels are connected to the carrier gear of the planetary gear train.

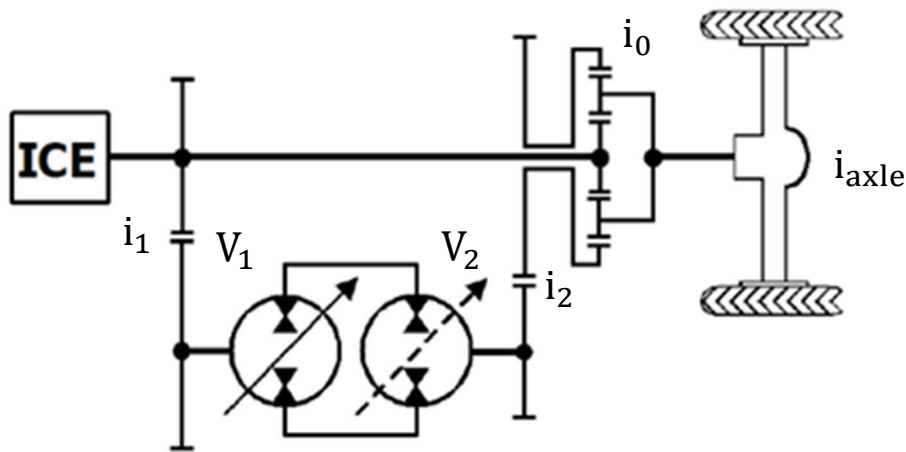


Figure 4.1: Input-coupled layout.

However, this is not the only layout that may be realized in the transmission. In fact, by changing the connection between the components of the planetary gear train, different layouts can be formed.

The gear ratios that can be obtained by the planetary gear train are three, which are the possibilities to fix one shaft to the chassis, times two, which are the possibilities to consider each element as input and output one by one).

$$N_{\text{layout,max}} = N_{\text{free,shafts}}! = 3! = 3 \cdot 2 = 6 \quad (4.1)$$

In this study, we considered all the layouts as feasible choices for the design, and no assumptions were made in advance to favor one of them over the others. However, we were aware in advance that certain layouts do not constitute good solutions for hydromechanical transmissions because they

produce higher planetary gear train output speeds, which are not feasible in real systems.

The objective is to develop a scientific approach for the design optimization, which is not yet present in the literature.

The six layouts are schematically illustrated in Figure 4.2; it should be noted that the transmission layout presented in Figure 4.1 is named “Layout VI” in Figure 4.2.

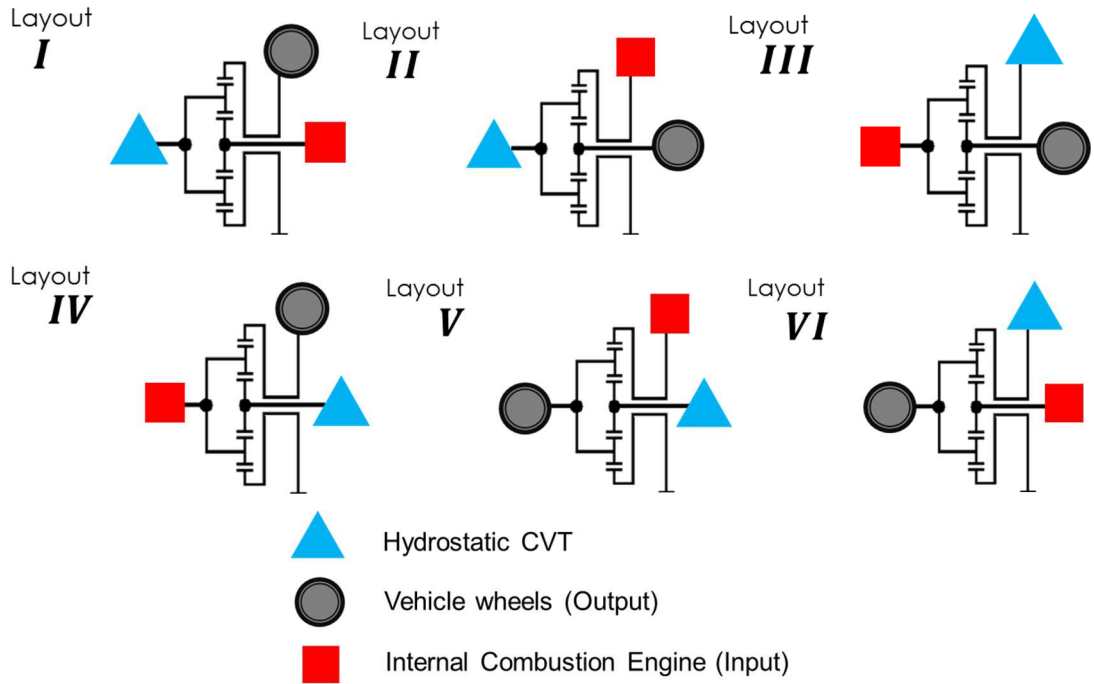


Figure 4.2: Schematic of input-coupled HMT layouts.

For each layout, the planetary gear train realizes different gear ratios, as functions of the input and the output shaft. Considering $i_0 = -Z_A/Z_C$ as a convention for the ordinary gear ratio ($-1 < i_0 < 0$), we could realize six gear ratios τ , as listed in Table 4.1.

Input shaft	Output shaft	Stator	Gear ratio
A	C	B	$\tau = \omega_A/\omega_C = 1/i_0$
C	A	B	$\tau = \omega_C/\omega_A = i_0$
A	B	C	$\tau = \omega_A/\omega_B = (i_0 - 1)/i_0$

B	A	C	$\tau = \omega_B/\omega_A = i_0/(i_0 - 1)$
B	C	A	$\tau = \omega_B/\omega_C = 1/(1 - i_0)$
C	B	A	$\tau = \omega_C/\omega_B = (1 - i_0)$

Table 4.1: Planetary gear train ratios in function of the input and output shafts.

In the following section, a detailed discussion and the design procedure is explained for each layout, because it is a fundamental part in the optimization sequence.

4.2.1 Layout I

In this layout, the engine is connected to the PGT sun gear, the HST is connected to the carrier, and the output is connected to the ring gear.

The typical design equations are listed below, which are useful for finding the design solution in terms of the displacements of the hydraulic units V_1 and V_2 , the gear ratio of the ordinary gears i_1 , i_2 , and the rear axle ratio i_{axle} .

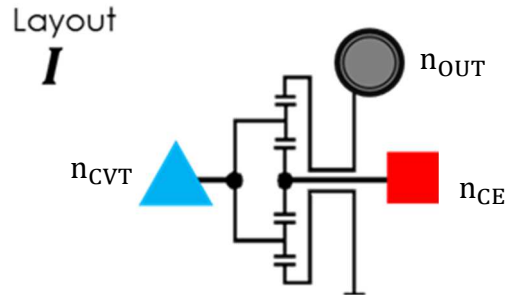


Figure 4.3: Schematic of Layout I.

To solve the design equations, it was assumed that all vehicle and engine data, as well as the planetary gear train ratio i_0 are known.

For the input-coupled transmission, the first assumption should be the vehicle speed at which the full mechanical point occurs. Therefore, the rear axle ratio is given as:

$$i_{\text{axle}} = \frac{v_{\text{max}} \cdot f_{\text{mp}}}{3,6 \cdot R_{\text{tire}} \cdot \frac{2\pi}{60}} \cdot \frac{1}{n_{\text{CE}} \cdot i_0}, \quad (4.2)$$

where $v_{\max}$ is the maximum speed of the vehicle in [km/h], f_{mp} is the fraction of the full mechanical point speed $f_{mp} = v_{fmp}/v_{\max}$, R_{tire} is the radius of the vehicle tires in [m], and n_{CE} is the rotational speed of the engine expressed in [rev/min].

Assuming a certain fixed value for the maximum speed of the hydraulic units $n_{1,\max}$ and $n_{2,\max}$, it is possible to calculate the ratio of ordinary gear 1:

$$i_1 = \frac{n_{1,\max}}{n_{CE}} \quad (4.3)$$

To calculate the ratio of ordinary gear 2, according to the Equation (4.4), we need to evaluate $n_{CVT,\max}$, which is the maximum rotational speed of the carrier during the operational conditions from zero speed to maximum speed.

$$i_2 = \frac{n_{2,\max}}{n_{CVT,\max}} \quad (4.4)$$

The aforementioned speed reaches zero value, when the full mechanical point occurs. Thus, n_{CVT} is required to be evaluated in two opposite conditions, namely for vehicle speed $v = 0$ km/h and $v = v_{\max}$.

For $v = 0$ km/h, the output speed is zero, and the Willis equation for the planetary gear train yields

$$n_{CVT}|_{v=0} = n_{CE} \left(\frac{i_0}{i_0 - 1} \right). \quad (4.5)$$

For $v = v_{\max}$, the output speed is given by Equation (4.6), which can be used to solve Equation (4.7).

$$n_{OUT} = \frac{V_{\max}}{3.6 R_{tire}} \cdot \frac{1}{\frac{2\pi}{60} i_{axle}} \quad (4.6)$$

$$n_{CVT}|_{v=v_{\max}} = n_{OUT} \left(\frac{1}{1 - i_0} \right) - n_{CE} \left(\frac{i_0}{i_0 - 1} \right) \quad (4.7)$$

Hence, $n_{CVT,\max}$ presented in Equation (4.4) is given by the following:

$$n_{CVT,max} = \max\{n_{CVT}|_{v=0}; n_{CVT}|_{v=v,max}\}. \quad (4.8)$$

The displacement V_2 can be determined by merging the two relations. The first relation determines the torque of unit 2 that occurs at the carrier gear, and it is denoted as M_{CVT} . The second relation determines M_{CVT} in a different manner, considering the planetary gear train Willis equations. In particular, for the current layout, the general equation, Equation (2.3), becomes

$$n_{CVT}(1 - i_0) = n_{OUT} - n_{CE} \cdot i_0. \quad (4.9)$$

The relation between M_{CVT} and M_{OUT} , which are the torques of the carrier and ring, respectively, is given in Equation (4.10).

$$M_{CVT} = M_{OUT} \cdot (1 - i_0) \quad (4.10)$$

The torque on the ring gear can be related to the wheels, as expressed in Equation (4.11), and the first relation of the torque on the carrier is written as Equation (4.12).

$$M_{OUT} = M_{wheel} \cdot i_{axle} \quad (4.11)$$

$$M_{CVT} = M_{wheel} \cdot (1 - i_0) \cdot i_{axle} \quad (4.12)$$

The second relation for M_{CVT} is given by the torque equation for unit 2, M_2 :

$$M_2 = \frac{\Delta p_{max} \cdot V_2}{20\pi}, \quad (4.13)$$

$$M_{CVT} = M_2 \cdot i_2 = \frac{\Delta p_{max} \cdot V_2}{20\pi} \cdot i_2. \quad (4.14)$$

Merging Equation (4.12) with Equation (4.14), the displacement of unit 2 is given by the following:

$$V_2 = \frac{M_{wheel}(1 - i_0) i_{axle}}{\frac{\Delta p_{max}}{20\pi} \cdot i_2}. \quad (4.15)$$

Through Equation (4.16), which was derived from the continuity and conservation of the flow rate in the hydrostatic transmission, it can be found that the displacement of Hydraulic Unit 1 can be obtained from the following.

$$V_1 = V_2 \cdot \frac{n_2}{n_1} \quad (4.16)$$

However, the operating point where $n_{CVT,max}$ is realized has to be considered. In fact, if $n_{CVT,max}$ is determined for $v = 0$ km/h, the two displacements are equal. Thus, the displacement of Unit 1 is given by the expression below:

$$V_1|_{\substack{n_{CVT,max} \\ v=0}} = V_2. \quad (4.17)$$

If $n_{CVT,max}$ is calculated for $v = v_{max}$, Equation (4.16) becomes

$$V_1|_{\substack{n_{CVT,max} \\ v=v_{max}}} = V_2 \cdot \frac{i_0}{i_0 - 1} \cdot \frac{i_2}{i_1}. \quad (4.18)$$

4.2.2 Layout II

In this layout, the engine is connected to the ring gear, the hydrostatic transmission is connected to the carrier gear, and the output is connected to the sun gear.

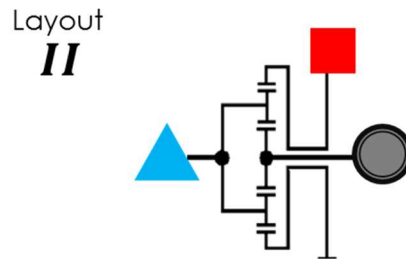


Figure 4.4: Schematic of layout II.

The traditional design equations for this layout are presented below, starting from the equation for the rear axle ratio, Equation (4.19).

$$i_{axle} = \frac{v_{max} \cdot f_{mp}}{3,6 \cdot R_{tire} \cdot \frac{2\pi}{60}} \cdot \frac{1}{n_{CE} \cdot \frac{1}{i_0}}. \quad (4.19)$$

Assuming the maximum speed of the hydraulic units as $n_{1,max}$ and $n_{2,max}$, it is possible to calculate the ratio of ordinary gear 1:

$$i_1 = \frac{n_{1,max}}{n_{CE}} \quad (4.20)$$

To calculate the ratio of ordinary gear 2, according to Equation (4.21), we need to evaluate $n_{CVT,max}$.

$$i_2 = \frac{n_{2,max}}{n_{CVT,max}} \quad (4.21)$$

For $v = 0$ km/h, the output speed is zero, and the Willis equation for the planetary gear train can be expressed as

$$n_{CVT}|_{v=0} = n_{CE} \left(\frac{1}{1 - i_0} \right). \quad (4.22)$$

For $v = v_{max}$, the output speed is given by Equation (3.6), which is used for the solution of Equation (4.23).

$$n_{CVT}|_{v=v,max} = n_{CE} \left(\frac{1}{1 - i_0} \right) - n_{OUT} \left(\frac{i_0}{1 - i_0} \right) \quad (4.23)$$

Hence, $n_{CVT,max}$ in Equation (4.21) is given by the following:

$$n_{CVT,max} = \max\{n_{CVT}|_{v=0}; n_{CVT}|_{v=v,max}\}. \quad (4.24)$$

Following the same procedure previously explained for Layout I, considering the new connections between the planetary gear train, the displacement of Unit 2 can be expressed as

$$V_2 = \frac{M_{wheel} \cdot i_{axle}}{\frac{\Delta p_{max}}{20\pi} \cdot i_2} \cdot \frac{i_0 - 1}{i_0}. \quad (4.25)$$

If $n_{CVT,max}$ was determined for $v = 0$ km/h, the displacement of Unit 1 is given by

$$V_1|_{n_{CVT,max} \atop v=0} = V_2. \quad (4.26)$$

If $n_{CVT,max}$ was calculated for $v = v_{max}$, the relation becomes

$$V_1|_{n_{CVT,max} \atop v=v_{max}} = V_2 \cdot \frac{1}{1 - i_0} \cdot \frac{i_2}{i_1}. \quad (4.27)$$

4.2.3 Layout III

In this layout, the engine is connected to the carrier gear, the hydrostatic transmission is connected to the ring gear, and the output is connected to the sun gear.

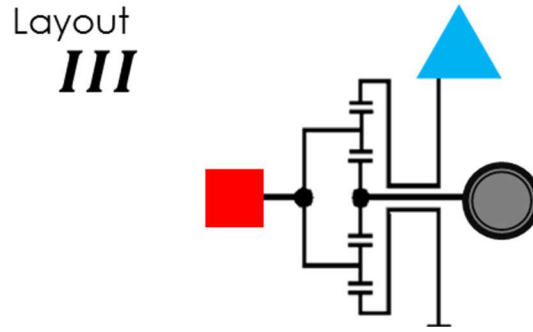


Figure 4.5: Schematic of the layout III.

Starting the design procedure, Equation (4.28) is the rear axle ratio:

$$i_{axle} = \frac{v_{max} \cdot f_{mp}}{3,6 \cdot R_{tire} \cdot \frac{2\pi}{60}} \cdot \frac{1}{n_{CE} \cdot \frac{i_0 - 1}{i_0}} \quad (4.28)$$

Assuming the maximum speed of the hydraulic units $n_{1,max}$ and $n_{2,max}$, it is possible to calculate the ratio of ordinary gear 1:

$$i_1 = \frac{n_{1,max}}{n_{CE}}. \quad (4.29)$$

To calculate the ratio of ordinary gear 2, according to Equation (4.30), we need to evaluate $n_{CVT,max}$.

$$i_2 = \frac{n_{2,max}}{n_{CVT,max}} \quad (4.30)$$

For $v = 0$ km/h, the output speed is zero, and the Willis equation for the planetary geartrain gives

$$n_{CVT}|_{v=0} = n_{CE}(1 - i_0). \quad (4.31)$$

For $v = v_{max}$, the output speed is given by Equation (4.6), which is used to solve Equation (4.23).

$$n_{CVT}|_{v=v,max} = n_{CE}(1 - i_0) + n_{OUT}(i_0) \quad (4.32)$$

Hence, $n_{CVT,max}$ in Equation (4.30) is given by the following:

$$n_{CVT,max} = \max\{n_{CVT}|_{v=0}; n_{CVT}|_{v=v,max}\}. \quad (4.33)$$

Following the procedure, as explained for Layout I, the displacement of Unit 2 is given by the expression:

$$V_2 = \frac{M_{wheel} \cdot i_{axle}}{\frac{\Delta p_{max}}{20\pi} \cdot i_2} \cdot \frac{1}{i_0}. \quad (4.34)$$

If $n_{CVT,max}$ is determined for $v = 0$ km/h, the displacement of Unit 1 is given by

$$V_1|_{n_{CVT,max}} = V_2. \quad (4.35)$$

If $n_{CVT,max}$ was calculated for $v = v_{max}$, the relation becomes

$$V_1|_{n_{CVT,max}} = V_2 \cdot (1 - i_0) \cdot \frac{i_2}{i_1}. \quad (4.36)$$

4.2.4 Layout IV

In this layout, the engine is connected to the carrier, the hydrostatic transmission is connected to the sun gear, and the output is connected to the ring gear.

Equation (4.37) is the rear axle ratio:

$$i_{axle} = \frac{v_{max} \cdot f_{mp}}{3,6 \cdot R_{tire} \cdot \frac{2\pi}{60}} \cdot \frac{1}{n_{CE} \cdot (1 - i_0)}. \quad (4.37)$$

Assuming the maximum speed of the hydraulic units $n_{1,max}$ and $n_{2,max}$, it is possible to calculate the ratio of ordinary gear 1:

$$i_1 = \frac{n_{1,max}}{n_{CE}}. \quad (4.38)$$

Layout
IV

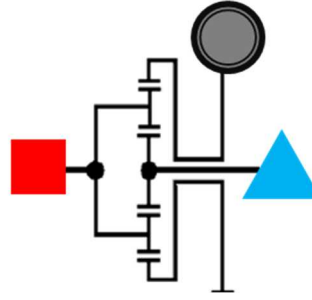


Figure 4.6: Schematic of layout IV.

To calculate the ratio of ordinary gear 2, according to Equation (4.39), we need to evaluate $n_{CVT,max}$.

$$i_2 = \frac{n_{2,max}}{n_{CVT,max}}. \quad (4.39)$$

For $v = 0$ km/h, the output speed is zero, and the Willis equation for the planetary geartrain gives

$$n_{CVT}|_{v=0} = n_{CE} \left(\frac{i_0 - 1}{i_0} \right). \quad (4.40)$$

For $v = v_{max}$, the output speed is given by Equation (4.6), and is used to solve Equation (4.41).

$$n_{CVT}|_{v=v,max} = n_{OUT} \left(\frac{1}{i_0} \right) - n_{CE} \left(\frac{1 - i_0}{i_0} \right) \quad (4.41)$$

Hence, $n_{CVT,max}$ in Equation (4.39) is given by the following:

$$n_{CVT,max} = \max\{n_{CVT}|_{v=0}; n_{CVT}|_{v=v,max}\}. \quad (4.42)$$

Following the procedure explained for Layout I, the displacement of Unit 2 is given by:

$$V_2 = \frac{M_{wheel} \cdot i_{axle}}{\frac{\Delta p_{max}}{20\pi} \cdot i_2} \cdot i_0 \quad (4.43)$$

If $n_{CVT,max}$ was determined for $v = 0$ the displacement of the Unit 1 is given by:

$$V_1|_{n_{CVT,max} \atop v=0} = V_2 \quad (4.44)$$

If $n_{CVT,max}$ was calculated for $v = v_{max}$, the relation becomes:

$$V_1|_{n_{CVT,max} \atop v=v_{max}} = V_2 \cdot \left(\frac{i_0 - 1}{i_0} \right) \cdot \frac{i_2}{i_1} \quad (4.45)$$

4.2.5 Layout V

In this layout, the engine is connected to the ring gear, the hydrostatic transmission is connected to the sun gear, and the output is connected to the carrier gear.

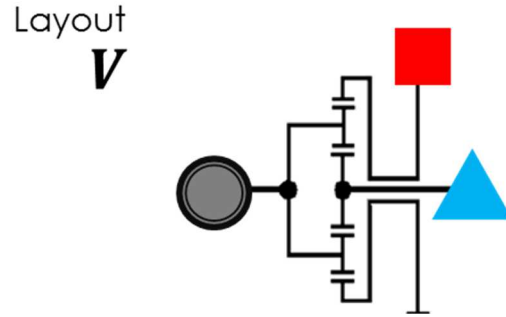


Figure 4.7: Schematic of layout V.

Equation (4.46) is the rear axle ratio:

$$i_{axle} = \frac{v_{max} \cdot f_{mp}}{3,6 \cdot R_{tire} \cdot \frac{2\pi}{60}} \cdot \frac{1}{n_{CE} \cdot \frac{1}{(1-i_0)} s} \quad (4.46)$$

Assuming the maximum speed of the hydraulic units $n_{1,max}$ and $n_{2,max}$, it is possible to calculate the ratio of ordinary gear 1:

$$i_1 = \frac{n_{1,max}}{n_{CE}}. \quad (4.47)$$

To calculate the ratio of ordinary gear 2, according Equation (4.48), we would need to evaluate $n_{CVT,max}$.

$$i_2 = \frac{n_{2,max}}{n_{CVT,max}}. \quad (4.48)$$

For $v = 0$ km/h, the output speed is zero, and the Willis equation for the planetary gear train is

$$n_{CVT}|_{v=0} = n_{CE} \left(\frac{1}{i_0} \right). \quad (4.49)$$

For $v = v_{\max}$, the output speed is given by Equation (4.6), and can be used for the solution of Equation (4.50).

$$n_{CVT}|_{v=v,\max} = n_{CE} \left(\frac{1}{i_0} \right) - n_{OUT} \left(\frac{1 - i_0}{i_0} \right) \quad (4.50)$$

Hence, $n_{CVT,\max}$ of Equation (4.48) can be expressed as in the following:

$$n_{CVT,\max} = \max\{n_{CVT}|_{v=0}; n_{CVT}|_{v=v,\max}\}. \quad (4.51)$$

Following the procedure explained for Layout I, the displacement of Unit 2 is given as

$$V_2 = \frac{M_{\text{wheel}} \cdot i_{\text{axle}}}{\frac{\Delta p_{\max}}{20\pi} \cdot i_2} \cdot \frac{i_0}{i_0 - 1}. \quad (4.52)$$

If $n_{CVT,\max}$ is determined for $v = 0$ km/h, the displacement of Unit 1 is given by the relation below:

$$V_1|_{n_{CVT,\max} \atop v=0} = V_2. \quad (4.53)$$

If $n_{CVT,\max}$ is calculated for $v = v_{\max}$, the relation becomes

$$V_1|_{n_{CVT,\max} \atop v=v_{\max}} = V_2 \cdot \left(\frac{1}{i_0} \right) \cdot \frac{i_2}{i_1}. \quad (4.54)$$

4.2.6 Layout VI

The sixth layout is characterized by the engine being connected to the sun gear, the hydrostatic transmission being connected to the ring gear, and the output connected being connected to the carrier gear.

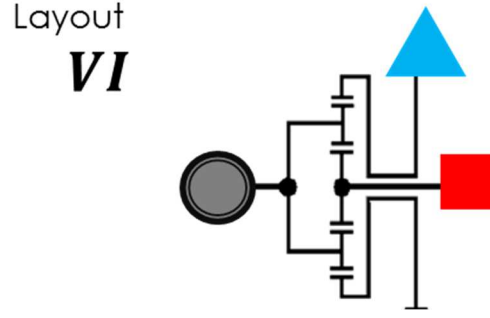


Figure 4.8: Schematic of layout VI.

The rear axle ratio is

$$i_{\text{axle}} = \frac{v_{\text{max}} \cdot f_{\text{mp}}}{3,6 \cdot R_{\text{tire}} \cdot \frac{2\pi}{60}} \cdot \frac{1}{n_{\text{CE}} \cdot \frac{i_0}{(i_0 - 1)}}. \quad (4.55)$$

Assuming the maximum speed of the hydraulic units $n_{1,\text{max}}$ and $n_{2,\text{max}}$, it is possible to calculate the ratio of ordinary gear 1:

$$i_1 = \frac{n_{1,\text{max}}}{n_{\text{CE}}}. \quad (4.56)$$

To calculate the ratio of ordinary gear 2, according to Equation (4.57), we need to evaluate $n_{\text{CVT},\text{max}}$.

$$i_2 = \frac{n_{2,\text{max}}}{n_{\text{CVT},\text{max}}}. \quad (4.57)$$

For $v = 0$ km/h, the output speed is zero, and the Willis equation for the planetary gear train becomes

$$n_{\text{CVT}}|_{v=0} = n_{\text{CE}} \cdot i_0. \quad (4.58)$$

For $v = v_{\text{max}}$, the output speed is given by Equation (3.6), and it is used to solve Equation (4.59).

$$n_{\text{CVT}}|_{v=v,\text{max}} = n_{\text{OUT}}(1 - i_0) + n_{\text{CE}}(i_0) \quad (4.59)$$

Hence, we can obtain $n_{\text{CVT},\text{max}}$ seen in Equation (4.57) via the following equation:

$$n_{\text{CVT},\text{max}} = \max\{n_{\text{CVT}}|_{v=0}; n_{\text{CVT}}|_{v=v,\text{max}}\}. \quad (4.60)$$

Following the procedure explained for Layout I, the displacement of Unit 2 can be expressed as

$$V_2 = \frac{M_{\text{wheel}} \cdot i_{\text{axle}}}{\frac{\Delta p_{\text{max}}}{20\pi} \cdot i_2} \cdot \frac{1}{1 - i_0}. \quad (4.61)$$

If $n_{\text{CVT,max}}$ is determined for $v = 0$ km/h, the displacement of Unit 1 can be obtained through the relation, below:

$$V_1|_{n_{\text{CVT,max}} \atop v=0} = V_2. \quad (4.62)$$

If $n_{\text{CVT,max}}$ is calculated for $v = v_{\text{max}}$, the relation becomes

$$V_1|_{n_{\text{CVT,max}} \atop v=v_{\text{max}}} = V_2 \cdot (i_0) \cdot \frac{i_2}{i_1}. \quad (4.63)$$

4.3 Simulation model

The optimization procedure is based on simulation models of the input-coupled transmissions, and requires the implementation of six simulation models, which correspond to the six layouts proposed in section 4.2. The models were implemented in the Amesim environment, and they differed only in terms of the connections between the shafts because it is fundamental to compare them to each other assuming the same parameters and sub-models. Thus, in this section, only one layout is considered as reference (Layout VI), and is explained. The complete schematic of the simulation model is presented in the Appendix.

4.3.1 The hydrostatic transmission model

In Figure 4.9, the hydrostatic transmission model is shown. It is characterized by two variable displacement units connected as a classical layout of closed-circuit applications. The two units are connected by two lines, whose maximum pressure is controlled via relief valves (opening pressure: Δp_{max}). They link the high pressure line to the low pressure line to avoid damages

caused by overpressure in the circuit, and are required in reversible systems that are connected back-to-back across loop lines.

A fixed-displacement hydraulic unit (charge pump, where $V_{\text{charge}} = 20 \text{ cc}$ and $n_{\text{charge}} = 1000 \text{ rpm}$) is placed within the system, whose function is to release a certain quantity of fluid from the reservoir, and then, to reintegrate it into the circuit.

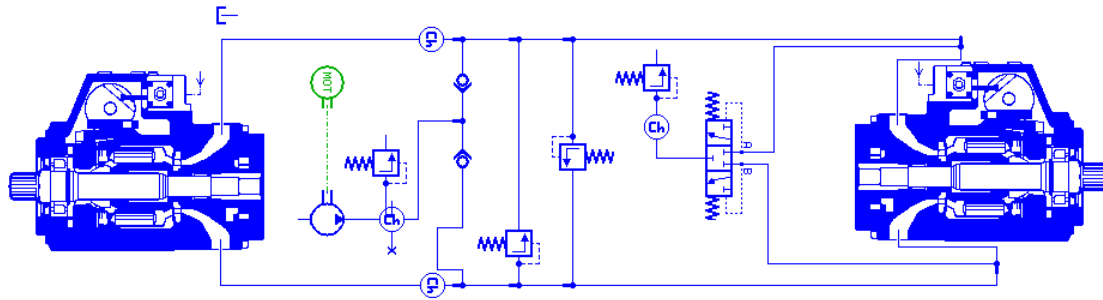


Figure 4.9: Amesim simulation model of the hydrostatic transmission.

According to the pressures in the high- and low-pressure lines, the fluid returns to the reservoir passing through a pressure relief valve (opening pressure: $\Delta p_{\text{max}}/2$) or through a flushing valve.

The flushing valve serves to discharge the excess loop oil, which is replaced by fresh oil from the charge pump, and the minimum pressure level of the closed circuit is fixed by the counter pressure valve (opening pressure: 10 bar).

4.3.2 The real hydraulic machines

The hydraulic units were modeled as real variable displacement machines, and the loss models were implemented according to the equations proposed (3.17). In particular, it was developed an accurate Amesim model to characterize the behavior of the hydraulic reversible machines (Figure 4.10).

Both the volumetric and the hydromechanical efficiency values depend on the pressure difference between the high and low pressure lines, the rotational speed of the unit, and the fractional displacement.

The volumetric efficiency was modeled considering the variable orifice in Figure 4.10, where the orifice diameter changes with the maximum displacement of the unit. The function that the orifice diameter has to follow is linear, and the coefficients were calibrated taking the reference value of the Rexroth datasheet of the A4VSO variable axial piston pump, for different displacement (40cc, 250cc, 500cc).

The hydromechanical efficiency was modeled considering an overall torque loss due to two main contributes: the first is the torque loss due to viscous friction on the shaft, and the second is the torque loss due to the differential pressure on the unit.

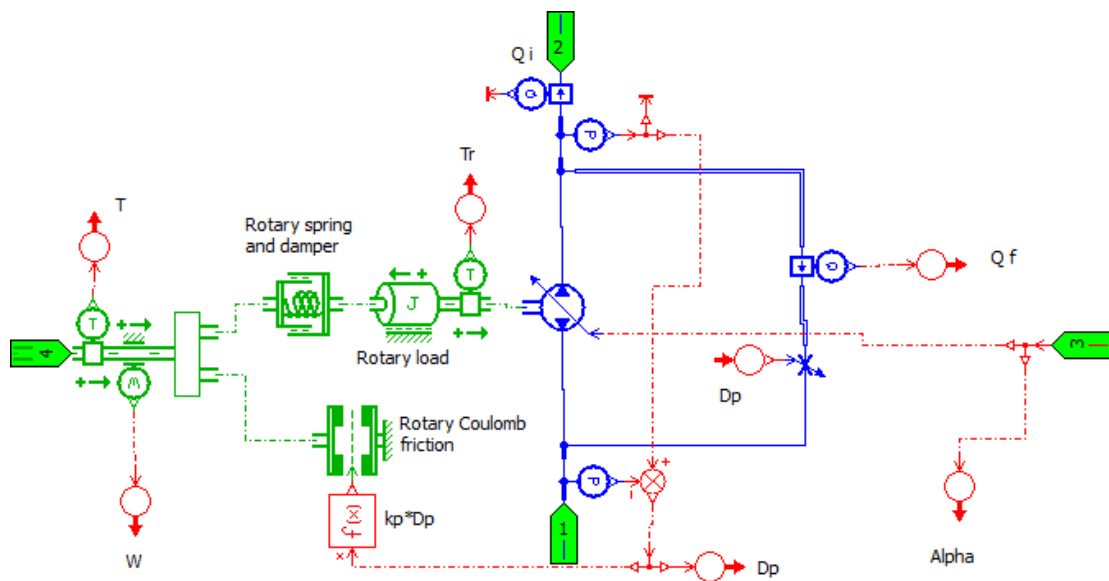


Figure 4.10: Real hydraulic unit integrated with a loss model.

The Amesim item, called Rotary load in the Figure 4.10, is a standard element in the mechanical library and it is useful to model the viscous friction loss through the Stribeck curve. The torque loss is function of three main coefficients: the stiction torque, the Coulomb friction torque and the coefficient of viscous friction and they change with the displacement through a linear law. The coefficients were calibrated taking the reference value of the Rexroth datasheet of the A4VSO variable axial piston pump, for different displacement (40cc, 250cc, 500cc), in order to reproduce the efficiency values declared.

A rotary spring and damper was added to the model in order to simulate the torsional elasticity of the shaft.

4.3.3 The engine model

The model of the engine in the transmission is a rotational speed generator (Figure 4.11). The target speed is imposed externally via the red control signal, and it was maintained constant during the entire simulation time.

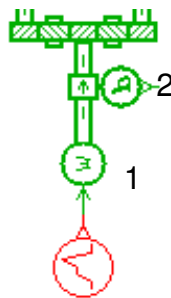


Figure 4.11: Engine simulation model.

The engine (item 1 in Figure 4.11) is directly connected to ordinary gear 1, and the instantaneous engine power was checked via the power rotary sensor that is placed over the speed generator (item 2 in Figure 4.11).

4.3.4 The planetary gear train model

As can be seen in Figure 4.12, the planetary gear train consists of the sun gear (item 1), the ring gear (item 2), the carrier gear (item 3), and the planets (item 4). The geometrical dimensions of the gears in the Amesim model can be related to each other considering the relations explained in this section. More specifically, by setting a fixed pitch radius for the sun gear ($R_A = 100 \text{ mm}$), consequently, the pitch radius of the planets (R_{PL}) and the pitch radius of the ring gears (R_C) can be calculated.

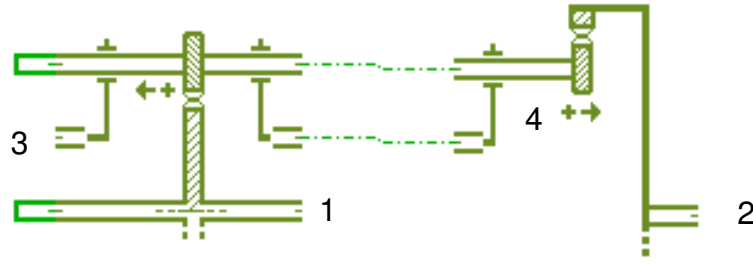


Figure 4.12: Amesim model of the planetary gear train.

The geometrical relations between the pitch radii of the gears in the PGT are given by

$$R_B = R_A + R_{PL} \quad (4.64)$$

$$R_C = R_A + 2 R_{PL} \quad (4.65)$$

The ordinary gear ratio i_0 is defined as in the following equation, and the pitch radius of the ring gear can be expressed as in Equation (4.67).

$$i_0 = -\frac{Z_A}{Z_C} = \frac{\omega_C}{\omega_A} = -\frac{R_A}{R_C} \quad (4.66)$$

$$R_C = -\frac{R_A}{i_0} \quad (4.67)$$

Substituting Equation (4.65) into Equation (4.67), we can find the relation for the pitch radius of the planet gears. From Equation (4.69), the pitch radius of the ring gear can be calculated.

$$R_{PL} = \frac{1}{2} \cdot R_A \left(-\frac{1}{i_0} - 1 \right) \quad (4.68)$$

$$R_C = R_A \left(-\frac{1}{i_0} \right) \quad (4.69)$$

Equations (4.68) and (4.69) were inserted to the planetary gear train model, whereas the ordinary ratio i_0 was considered as a global parameter. The efficiency of the planetary gear train was considered to be equal to a constant value of 0.98.

4.3.5 The vehicle model

The vehicle model, the rear axle, and the mission load are shown in Figure 4.13. The rotational motion from the planetary gear train is transmitted through the rear axle (item 1), and is then converted from rotary to linear motion in item 2, which represents the wheel tires. The mass (item 3) represents the vehicle, and the load is considered to be acting upon the mass through item 4, which is a clutch with a force input signal.

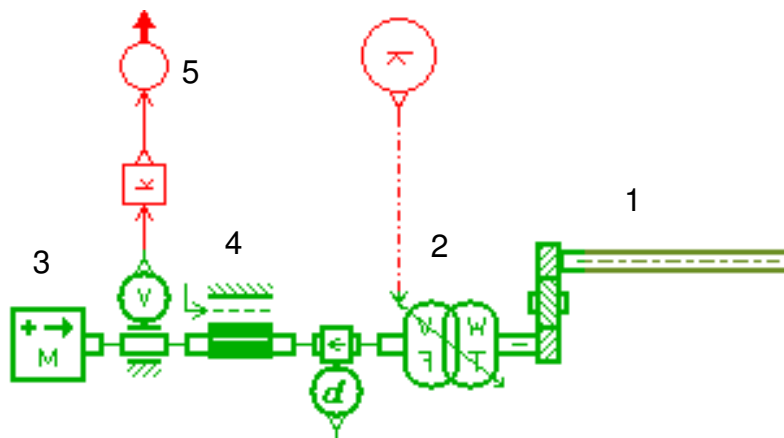


Figure 4.13: Model of the vehicle and the rear axle.

The mass of the vehicle M , the radius of the wheels R_{tire} , and the gear axle ratio i_{axle} are considered as global parameters. The variable load acting on item 4 will be explained in the following section. Finally, item 5 converts the linear motion expressed in [m/s] to a linear motion expressed in [km/h].

In this model, the rear axle was considered as a single component that reduces the speed from the planetary gear train. However, if a general design procedure is considered, it is possible to accomplish a global reduction that exceeds the maximum reduction achievable by a single element. In this case, the vehicle would need more than one gear reduction connected serially, for example an additional planetary gear or a simple ordinary gear.

This modification becomes important when the calculation of the mechanical efficiency occurs. In order to take into account of the mechanical losses of the elements in series, the following Matlab model is proposed.

```

if abs(Iaxle) > Idiff %case of the single axle gear
    Iaxle_Eff = Idiff_eff;
else
    n = log(abs(Iaxle)/Idiff)/log(Iepi);
    n = ceil(n);
    %case of the axle gear and multiple planetary geartrain
    %in series
    Iaxle_Eff = Idiff_eff*(Iepi_eff)^n;
end

```

If the value of the global reduction i_{axle} obtained from the design procedure is higher than the minimum value of the reduction of the differential gear $i_{diff} = 0.25$, a single reduction is enough, and the mechanical efficiency (i_{axle_eff}) is reduced to the single differential gear efficiency ($i_{diff_eff} = 0.97$).

If the value of the global reduction is lower than the minimum value $i_{diff} = 0.25$, the vehicle would need n planetary gear trains in series, where n would be calculated considering the minimum reduction that the PGT can achieve ($i_{epi} = 0.16$). In this case, the global efficiency is the result of the product between the single differential gear efficiency and the n -time PGT efficiency.

4.3.6 Load control loop

In this work, the load is applied on the vehicle following the curve of the maximum traction (constant power test), the control loop of which is shown in Figure 4.14. Item 1 calculates the instantaneous force obtained through the relation (4.70), where ICE_Power is the maximum engine power (item 2) and Actual_v (item 3) is the instantaneous speed of the vehicle.

$$F_{tr} = \frac{ICE_power}{Actual_v} \quad (4.70)$$

Item 4 is a loop that receives the difference between the ICE_Power and the ICE_Actual Power, and then feeds a PID that corrects the traction force calculated in item 1. The ICE_Actual Power signal is obtained from item 2 in Figure 4.11, and represents the instantaneous engine power.

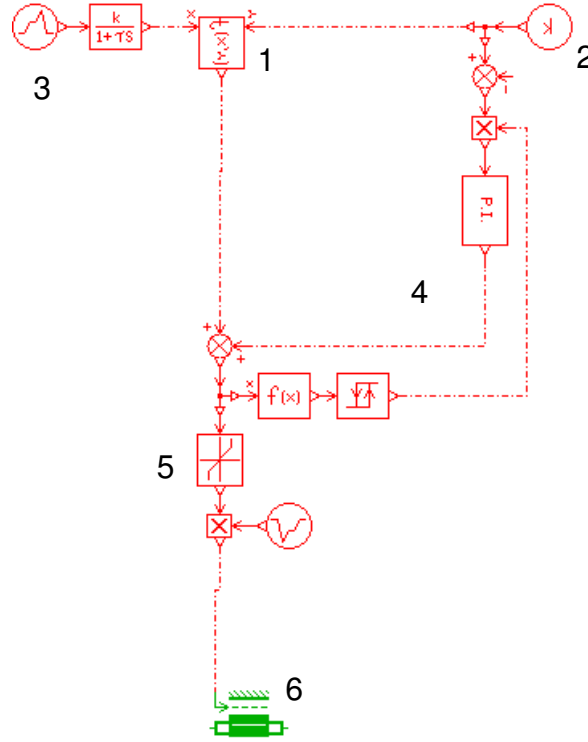


Figure 4.14: Vehicle load control loop.

The traction force is generally limited with its maximum value: $F_{tr,max}$, which is given via Equation (4.71), where M_{wheel} is a design parameter that indicates the maximum torque admissible at the wheels.

$$F_{tr,max} = \frac{M_{wheel}}{R_{tire}} \quad (4.71)$$

Item 5 limits the traction force at the maximum value expressed in the equation above, and the output signal is then sent to the virtual clutch, which represents the applied load (item 6).

4.3.7 Vehicle target speed and displacement control loop

In this section, the sequential control of the hydraulic displacement is explained (Figure 4.15). The control was divided into two branches: one where the real parameters, namely the ratio of fractional displacements α_1/α_2 , are calculated (*feed-forward* control), and another where a PI. controller corrects the first branch. Both branches need the vehicle target speed as an input variable.

The target speed is the speed that the vehicle has to follow, imposed by item 1. In the first branch, the target speed is compared to the real vehicle speed (item 2 in Figure 4.15, equivalent to item 5 in Figure 4.13), and their difference is imposed as input to a PI (item 3). This “speed error” will be summed (item 4) with the signal derived from the second control branch.

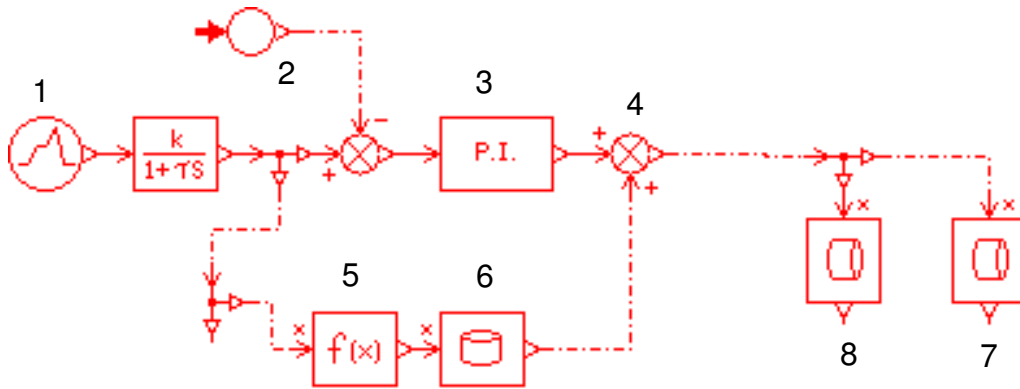


Figure 4.15: Displacement control loop.

The purpose of the second control branch is to regulate the displacement of the units by using the numerical relations expressed via the equation inside item 5. These relations are functions of the vehicle speed, and their output is the ratio of the two fractional displacements α_1/α_2 . The relations can be defined for each transmission layout; the equations for layout VI (Figure 4.8) are reported in detail below, Equations (4.72)–(4.75), whereas the expressions for the other layouts are described through Equations (4.76)–(4.80).

According to the following equations, the ratio α_1/α_2 is calculated with the feed-forward branch (item 5), and the PI element corrects the value, if differences between the real and the target speed occur.

The vehicle speed can be calculated by Equation (4.72), where n_{OUT} is the rotational speed of the carrier. Moreover, n_{OUT} can be defined through the Willis equation (4.73), and n_{CVT} can be related to the hydrostatic transmission parameters through Equation (4.74).

$$v = R_{tire} \left(3,6 \cdot \frac{2\pi}{60} \right) \cdot i_{axle} \cdot n_{OUT} \quad (4.72)$$

$$n_{OUT} = \frac{n_{CE} \cdot i_0 - n_{CVT}}{i_0 - 1} \quad (4.73)$$

$$n_{CVT} = n_{CE} \cdot \frac{V_1}{V_2} \cdot \frac{\alpha_1}{\alpha_2} \cdot \frac{i_1}{i_2} \quad (4.74)$$

By merging the three equations and extrapolating the fractional displacement ratio α_1/α_2 , we obtained the following relation, which is a function of the transmission parameters, and is included in item 5:

$$x|_{VI} = \frac{\alpha_1}{\alpha_2} = \left(\frac{-v}{R_{tire} \left(3,6 \cdot \frac{2\pi}{60} \right) i_{axle} \cdot n_{CE} \cdot \left(\frac{i_0}{i_0 - 1} \right)} + 1 \right) \cdot i_0 \left(\frac{V_2}{V_1} \cdot \frac{i_2}{i_1} \right). \quad (4.75)$$

Following the same procedure, the fractional displacement ratio can be calculated for each layout.

$$x|_I = \frac{\alpha_1}{\alpha_2} = \left(\frac{v}{R_{tire} \left(3,6 \cdot \frac{2\pi}{60} \right) i_{axle} \cdot n_{CE} \cdot i_0} + 1 \right) \cdot \frac{i_0}{1 - i_0} \left(\frac{V_2}{V_1} \cdot \frac{i_2}{i_1} \right) \quad (4.76)$$

$$x|_{II} = \frac{\alpha_1}{\alpha_2} = \left(\frac{v}{R_{tire} \left(3,6 \cdot \frac{2\pi}{60} \right) i_{axle} \cdot n_{CE} \cdot \left(\frac{1}{i_0} \right)} + 1 \right) \cdot \frac{1}{i_0 - 1} \left(\frac{V_2}{V_1} \cdot \frac{i_2}{i_1} \right) \quad (4.77)$$

$$x|_{III} = \frac{\alpha_1}{\alpha_2} = \left(\frac{-v}{R_{tire} \left(3,6 \cdot \frac{2\pi}{60} \right) i_{axle} \cdot n_{CE} \cdot \left(\frac{i_0 - 1}{i_0} \right)} + 1 \right) \cdot (i_0 - 1) \left(\frac{V_2}{V_1} \cdot \frac{i_2}{i_1} \right) \quad (4.78)$$

$$x|_{IV} = \frac{\alpha_1}{\alpha_2} = \left(\frac{-v}{R_{tire} \left(3,6 \cdot \frac{2\pi}{60} \right) i_{axle} \cdot n_{CE} \cdot (1 - i_0)} + 1 \right) \cdot \left(\frac{1 - i_0}{i_0} \right) \left(\frac{V_2}{V_1} \cdot \frac{i_2}{i_1} \right) \quad (4.79)$$

$$x|_V = \frac{\alpha_1}{\alpha_2} = \left(\frac{v}{R_{tire} \left(3,6 \cdot \frac{2\pi}{60} \right) i_{axle} \cdot n_{CE} \cdot \left(\frac{1}{i_0 - 1} \right)} + 1 \right) \cdot \frac{1}{i_0} \left(\frac{V_2}{V_1} \cdot \frac{i_2}{i_1} \right) \quad (4.80)$$

The ratio $\frac{\alpha_1}{\alpha_2} = x$ can receive values from -1 to 1, depending on the operating conditions of the transmission. The signal x is transferred into item 6, where it is amplified, as shown in Figure 4.16, and it is then corrected by the P.I. signal (item 4). After the summing junction, two displacement controllers (item 7 for Unit 1 and item 8 for Unit 2) are placed, and they permit the sequential control of displacements V_1 and V_2 , following the graph in Figure 4.17.

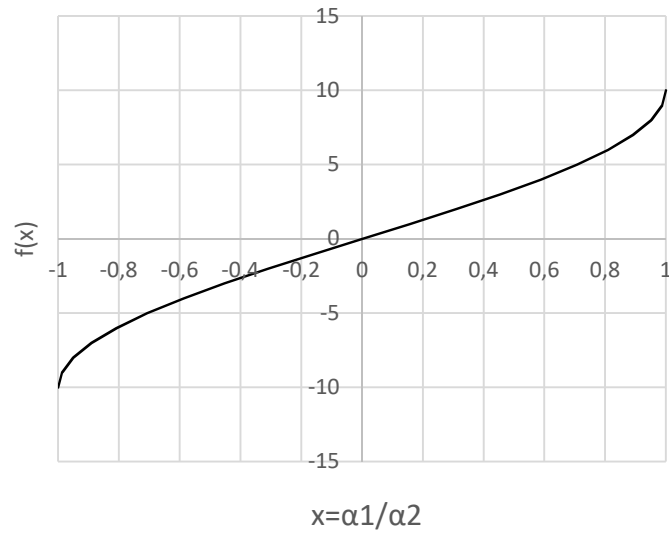


Figure 4.16: Fractional displacement ratio amplification.

More specifically, the trend of the fractional displacement of Unit 1, α_1 , is defined by the graph in item 7, and displayed in red in Figure 4.17. On the other hand, the fractional displacement of Unit 2 is defined by the graph in item 2, and is displayed in green in Figure 4.17.

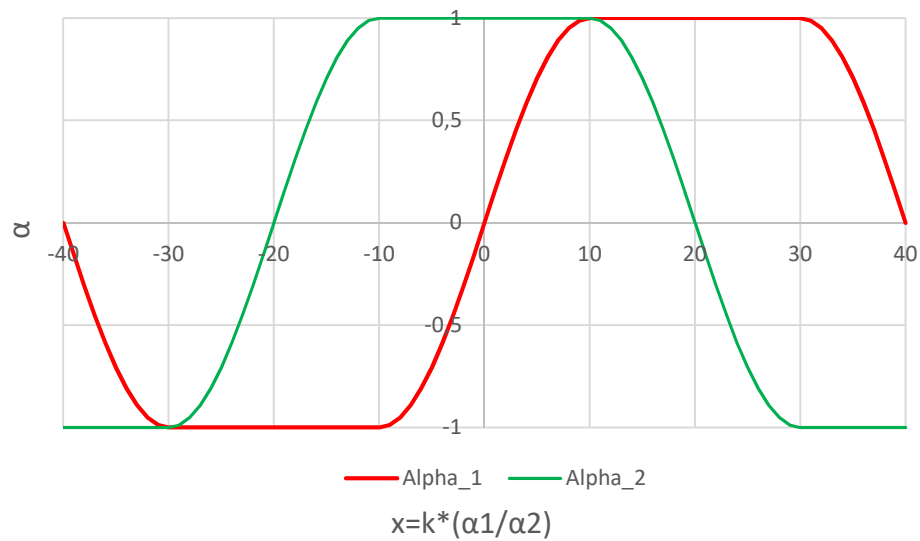


Figure 4.17: Sequential displacement control.

4.3.8 High-speed mechanical losses

The full description of the viscous and friction losses on the bearings, ventilation and splash oil losses of the overall transmission would lead to a very complex model, as just described in the chapter 3.6. Furthermore, a detailed description of the losses needs the full description of the shafts, gears and bearings, which can not be known when preliminary design consideration were drawn, such as in this work. For this reason, the friction losses were neglected because they do not reasonably interfere with the optimum definition, despite changing (slightly) the overall efficiency.

However, preliminary tests showed that viscous effects have strong influence on the optimum design choice. In the absence of viscous effects, the optimum configuration will move toward a high speed output shaft, to reduce the momentum losses on the axle gears. On the contrary, the presence of viscous effect forces the reduction of the shaft speed to a reasonable value.

In some of the HMT layout, when the planetary geartrain works as speed's multiplier, the rotational velocity of the output shaft (before the rear axle) is very high. For this reason, viscous losses were instead modeled, using a simple model. Because the input shaft is fixed, and the hydraulic unit's mechanical losses were modeled inside the unit subsystem, the viscous losses of the rotational shafts were thus concentrated in the output shaft. They are empirically estimated (Rossetti & Macor, 2012) considering a 0,1Nm/(rad/s) of loss torque. If this loss contribution was not considered, the layouts with higher output speed would have been favored. The global HMT efficiency, η_{HMT} is thus given in the (4.81).

$$\eta_{HMT}^* = \eta_{HMT} - 0.1 \cdot \frac{|n_{OUT}|}{ICE_{Power}} \quad (4.81)$$

4.4 General optimization statement

In the previous sections the traditional design procedure and the simulation models for each HMT layout were proposed; they are useful for the evaluation of the global efficiency of the transmission. However, the value obtained would not be the best achievable.

In this work, two optimization approaches could be adopted: the first is the “manual search” optimization, and it is useful for testing the stability of the entire procedure, as well as for the observation of how the efficiency changes by changing the optimization parameters. The second is a direct search algorithm, called “Particle Swarm Optimizer,” which is a fast optimizer used to verify the results obtained by the first, complete, approach.

In this section, solely the first approach will be explained: The Particle Swarm Optimizer has just been explained in the Chapter 3 and it could not more improve the originality of the work.

The optimal design procedure allows the simultaneous optimization of the internal components, as well as the geometry of the transmission. It can be formulated in the following algorithmic form, in a manner similar to the optimization procedure described in the section 3.4.

Find the vector of the free optimization variables $x = [x_1 \ x_2, \dots, x_n]^T$ that would minimize the objective function $f(x)$, subject to the design parameters $l_j(x) = 0$ for $j = 1, \dots, p$ and the design constraints $g_j(x) = 0$ for $j = 1, \dots, m$.

- The optimization variables are the following:
 - the hydraulic displacement V_1 and V_2 ,
 - the transmission ratios i_1 and i_2 ,
 - the standing gear ratio of the planetary geartrain i_0 , and
 - the rear axle ratio i_{axle} .
- The design parameters supplied as input of the optimization procedure are:
 - the speed and the maximum power of the engine,
 - the maximum speed of the vehicle,
 - the maximum wheel pulling force,
 - the wheel radius, and
 - the maximum speed of the hydraulic units.
- The optimization procedure has to satisfy certain structural and design constraints:

- the maximum pressure of the hydraulic machines: $\Delta p \leq \Delta p_{\max}$;
 - minimum and maximum displacement of the units: $V_{\min} \leq V_1$ and $V_2 \leq V_{\max}$;
 - minimum and maximum transmission ratios of the planetary geartrain: $i_{\min} \leq i_0 \leq i_{\max}$;
 - minimum and maximum full mechanical point ratio: $fmp_{\min} \leq fmp \leq fmp_{\max}$.
- The objective function is the integral average loss calculated between the zero speed and the maximum design speed of the vehicle.

$$\overline{\eta_{HMT}} = \frac{1}{v_{\max}} \int_0^{v_{\max}} \eta_{HMT} dv \quad (4.82)$$

Here, η_{HMT} is the transmission efficiency, defined as follows:

$$\eta_{HMT} = \frac{P_{\text{wheel}}}{P_{\text{engine}}}. \quad (4.83)$$

The difference between the design method described above and the method described in chapter 3 is the number of design variables. Here, the procedure depends on one additional variable than the previous method: the speed ratio of the full mechanical point fmp . The other two design variables are the layout and the planetary gear train ratio i_0 .

However, in the procedure explained in the following section, only two design variables are considered. In fact, a new variable θ that correlates the layouts and the planetary gear ratio i_0 is considered.

4.5 The polar function

Considering the planetary gear train in the HMT, it is possible to correlate each geometrical combination (layout) with the standing gear ratio i_0 , through a mathematical procedure explained below.

Starting from the Willis equation, Equation (4.84), for $(-1/2) \leq i_0 \leq (-1/7)$,

$$\frac{\omega_C - \omega_B}{\omega_A - \omega_B} = i_0, \quad (4.84)$$

where one by one, the respective rotational speed of the three shafts holds a fixed value that is equal to one, and thus Equation (4.84) is solved. Whereas, in the equation above there are still two variables, one at a time has to be considered equal to zero. The results are presented in Table 4.2, where six equations are definite. Here, the rotational speeds of the shafts are functions of i_0 , and are evaluated assuming that one of the shafts is fixed to the chassis (zero speed), and the remaining one holds a fixed positive speed ($\omega = 1$).

ω_B	ω_A	a) $\omega_C = 1 - i_0$
	ω_C	b) $\omega_A = \frac{i_0 - 1}{i_0}$
ω_A	ω_B	c) $\omega_C = i_0$
	ω_C	d) $\omega_B = \frac{i_0}{i_0 - 1}$
ω_C	ω_A	e) $\omega_B = \frac{1}{1 - i_0}$
	ω_B	f) $\omega_A = \frac{1}{i_0}$

Table 4.2: Equations suitable to create the polar function.

Considering for example the case for $\omega_B = 1$ as a shaft with a constant speed, we solve the equations a) and b) in the table above, for n values of the standing gear ratio i_0 (here we consider six values between $i_{0,\min} = -1/7$ and $i_{0,\max} = -1/2 : -1/2; -1/3; -1/4; -1/5; -1/6; -1/7$).

The two rotational speeds ω_C and ω_A could be plotted in a graph (orange and red points respectively), considering one as input and the other as output shaft in the PGT and consequently switching the roles. The procedure has to be applied to the remaining two cases ($\omega_A = 1, \omega_C = 1$). The result is shown in Figure 4.18, where the x-axis represents the speed of the input shaft and the y-axis represents the speed of the output shaft.

Following points are worth mentioning here. Each couple of values (e.g., (ω_A, ω_C) , (ω_C, ω_A) , etc.) calculated between $i_{0,\min}$ and $i_{0,\max}$, can be linked together and a curve appears as plotted by a dotted line in the Figure 4.18.

Each point of this curve definite the speed of the input and output shaft and the i_0 , and it is associated to the corresponding hydromechanical transmission layout, as shown in Figure 4.18.

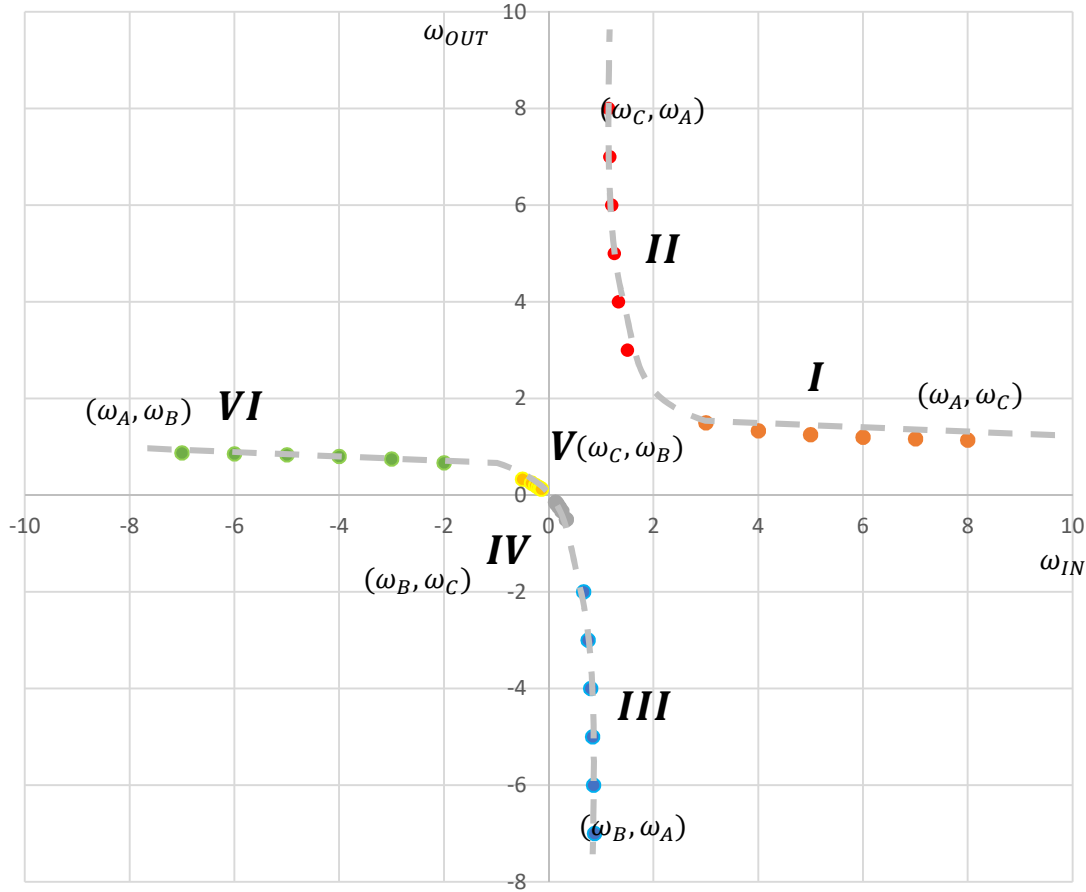


Figure 4.18: Homographic function that associates the layouts with the PGT standing gear ratio.

The homographic function, which links all the points is given by

$$\omega_{OUT} = \frac{\omega_{IN}}{\omega_{IN} - 1}. \quad (4.85)$$

However, this representation is still not practical, and is in need of certain modifications prior to being implemented in the future optimization procedure.

- Solution of the equations in Table 4.2, with absolute values.
- Change of coordinates system from Cartesian to polar.
- Centering of the new coordinate system in [1,1].
- Plotting in logarithmic scale.

Layout I between the extremes (a) and (b):

ρ_a	$\sqrt{\frac{1}{i_{0,min}^2} + i_{0,min}^2}$	ρ_b	$\sqrt{\frac{1}{i_{0,max}^2} + i_{0,max}^2}$
ϑ_a	$\arccos\left(-\frac{1}{i_{0,min} \cdot \rho_a}\right)$	ϑ_b	$\arccos\left(-\frac{1}{i_{0,max} \cdot \rho_b}\right)$

Layout II between the extremes (c) and (d):

ρ_c	$\sqrt{\frac{1}{i_{0,max}^2} + i_{0,max}^2}$	ρ_d	$\sqrt{\frac{1}{i_{0,min}^2} + i_{0,min}^2}$
ϑ_c	$\arccos\left(-\frac{i_{0,max}}{\rho_c}\right)$	ϑ_d	$\arccos\left(-\frac{i_{0,min}}{\rho_d}\right)$

Layout III between the extremes (e) and (f):

ρ_e	$\sqrt{\left(\frac{i_{0,min}}{1-i_{0,min}}\right)^2 + \left(\left \frac{1}{i_{0,min}}\right - 1\right)^2}$	ρ_f	$\sqrt{\left(\frac{i_{0,max}}{1-i_{0,max}}\right)^2 + \left(\left \frac{1}{i_{0,max}}\right - 1\right)^2}$
ϑ_e	$\arccos\left(\frac{i_{0,min}}{(1-i_{0,min})\rho_e}\right)$	ϑ_f	$\arccos\left(\frac{i_{0,max}}{(1-i_{0,max})\rho_f}\right)$

Layout IV between the extremes (g) and (h):

ρ_g	$\sqrt{\left(\frac{1}{i_{0,max}-1}\right)^2 + (i_{0,max} - 1)^2}$	ρ_h	$\sqrt{\left(\frac{1}{i_{0,min}-1}\right)^2 + (i_{0,min} - 1)^2}$
ϑ_g	$\arccos\left(\frac{1}{(i_{0,max}-1)\rho_g}\right)$	ϑ_h	$\arccos\left(\frac{1}{(i_{0,min}-1)\rho_h}\right)$

Layout V between the extremes (i) and (l):

ρ_i	$\sqrt{(i_{0,min} - 1)^2 + \left(\frac{1}{i_{0,min}-1}\right)^2}$	ρ_l	$\sqrt{(i_{0,max} - 1)^2 + \left(\frac{1}{i_{0,max}-1}\right)^2}$
ϑ_i	$\arccos\left(\frac{ i_{0,min} - 1}{\rho_i}\right)$	ϑ_l	$\arccos\left(\frac{ i_{0,max} - 1}{\rho_l}\right)$

Layout VI between the extremes (m) and (n):

ρ_m	$\sqrt{\left(\frac{1}{ i_{0,max} } - 1\right)^2 + \left(\frac{i_{0,max}}{1 - i_{0,max}}\right)^2}$	ρ_n	$\sqrt{\left(\frac{1}{ i_{0,min} } - 1\right)^2 + \left(\frac{i_{0,min}}{1 - i_{0,min}}\right)^2}$
ϑ_m	$\arccos\left(\left(\frac{1}{ i_{0,max} } - 1\right) \cdot \frac{1}{\rho_m}\right)$	ϑ_n	$\arccos\left(\left(\frac{1}{ i_{0,min} } - 1\right) \cdot \frac{1}{\rho_n}\right)$

Generalizing the equations of the extremes for each layout, the points along the six colored curves can be expressed by the following.

$$\text{Layout I: } \cos\vartheta + \frac{1}{i_0 \cdot \rho} = 0$$

$$\text{Layout II: } \cos\vartheta + \frac{i_0}{\rho} = 0$$

$$\text{Layout III: } \cos\vartheta - \frac{i_0}{(1-i_0) \cdot \rho} = 0$$

$$\text{Layout IV: } \cos\vartheta - \frac{1}{(i_0-1) \cdot \rho} = 0$$

$$\text{Layout V: } \cos\vartheta - \frac{|i_0|-1}{\rho} = 0$$

$$\text{Layout VI: } \cos\vartheta - \left(\left|\frac{1}{i_0}\right| - 1\right) \frac{1}{\rho} = 0 \quad (4.86)$$

The scope of the polar function is to know the corresponding layout and i_0 , for a fixed coordinate ϑ . The layout can be easily defined between the intervals of the previously defined ϑ coordinate. However, to extrapolate the i_0 in Equation (4.86), there is still an unknown variable ρ .

The key relations used for the correlation of the ϑ variables with ρ depending on the three curves, A, B, and C, in Figure 4.19, the general implicit equations of which are expressed as in the following.

$$fA = \rho \cdot \sin\vartheta - \frac{1}{\rho \cdot \cos\vartheta} = 0$$

$$fB = \rho \cdot \sin\vartheta + \frac{2\rho\cos\vartheta + 1}{\rho\cos\vartheta} = 0 \quad (4.87)$$

$$fC = \rho \cdot \sin\vartheta + \frac{1}{\rho \cdot \cos\vartheta + 2} = 0$$

Summarizing the procedure, to extrapolate the transmission layout and the i_0 , only one ϑ coordinate is required to be fixed in input. The first step is to understand which is the corresponding “sector” where ϑ should be placed. In other words, ϑ could be placed inside or outside the extremes of the six layout curves (the “sectors”). If ϑ is placed inside the designated area (or corresponding to a specific extreme of the sector) the layout is immediately known. In fact, the limits of the sector are defined above. If ϑ is placed outside, the procedure adopts the layout corresponding to the first extremity nearest to ϑ because the remaining area is in a non-feasible sector.

When the layout is assigned, the corresponding curve A, B, or C is known; solving equation (4.87), the coordinate ρ could be calculated. Curve A is valid for both layouts I and II; curve B is valid for both layouts III and IV, and curve C is valid for layouts V and VI.

Finally, with ϑ , ρ , and the layout, equation (4.86) could be solved for i_0 .

The procedure for the polar function was implemented in the MATLAB environment, and according to what was anticipated, ϑ became the new design variable, in addition to the speed of the full mechanical point, replacing in a singular variable both the layout and i_0 .

4.6 Manual search optimization

The first optimization approach consists in searching the best efficiency of the transmission (4.82) using the simplest optimization method. The procedure is shown in Figure 4.20. It is based on the design equations’ solution for each combination of the design variable. The resolution of the system using the data for a specific case study permits to set up the global parameters of the relative simulation model of the transmission, which give in output the value of the average global efficiency. If the procedure is performed for all the possible combinations of the initial variables, the results will be an efficiency matrix that shows the variability of the average efficiency in function of the two design variables.

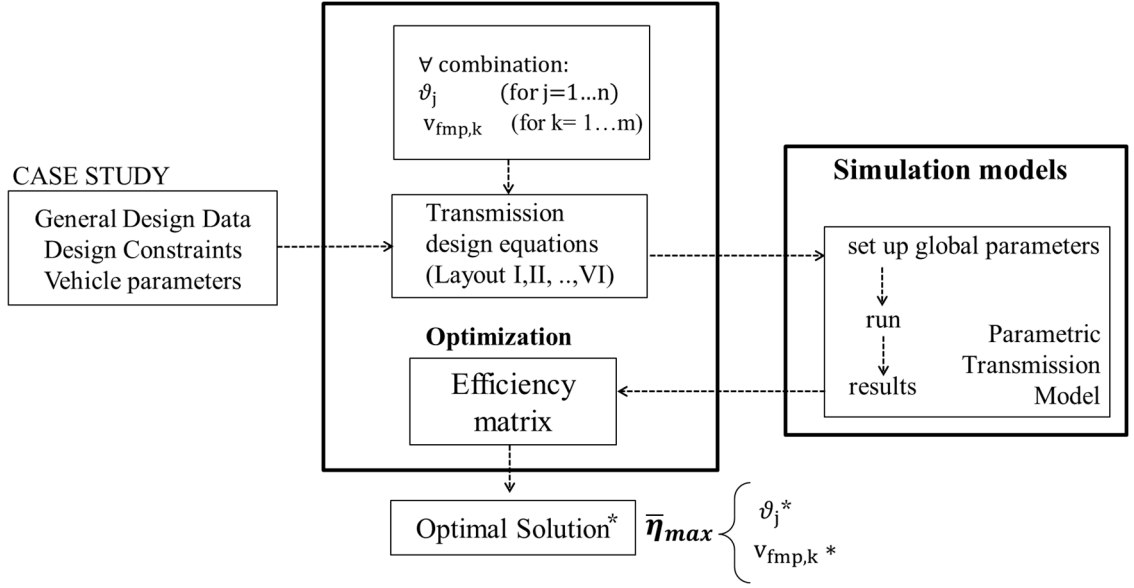


Figure 4.20: Procedure of the manual-search optimization.

The design procedure can be implemented for one or more case study. The general design data and design constraints of the case study have to be known as input of the transmission design. Some vehicle parameters have to be assumed: the mass, the tires dimension, the maximum speed, the maximum torque on the wheel.

In this work, the optimization procedure was applied for an urban passenger bus as reference vehicle, and the discussion of the results will be done in the following sections.

4.7 Procedural check

In this section, the procedure is show for a couple design variables. Thus the result, in terms of global efficiency of the transmission, will not be the best achievable, but only one of the values, which can be obtained.

In a second time (section 4.8) the optimization procedure is applied, and the result will be the efficiency matrix (Figure 4.20) where it can be seen the best efficiency point.

4.7.1 Design variables

The couple of design variable arbitrary chosen for the procedural check are the following:

$$\vartheta = 6.11 \text{ rad}$$

$$fmp = 0.3$$

Considering the variables above the polar function can be solved following the procedure in the previous section, and the corresponding layout and the standing gear ratio i_0 are given by:

Layout= VI

$$i_0 = -0.5$$

Therefore, the hydromechanical transmission has the PGT connected as indicated in Figure 4.8.

4.7.2 Reference vehicle

The reference vehicle is a city bus whose main data are shown in the Table 4.3 and come from Blake et al. (2006). This choice is useful to make a comparison of the results in literature at the end of the optimization procedure. The target speed of the vehicle was chosen linear from zero to the maximum vehicle speed. The maximum wheel torque instead, is function of the friction force between the wheels and the ground; the value was chosen by the experience, in order to avoid the wheels slip.

ICE Power	261 kW
ICE rated speed.	1800 rpm
Wheel radius	0.5281 m
Maximum wheel torque	25000 kN
Maximum speed	120 km/h

Table 4.3: Main data of the urban bus.

4.7.3 Solution of the design equations and simulation results

The design equations for the VI layout expressed in the section 4.2.6 can be solved now, and the results are shown in the Table 4.4. A MATLAB code permits to solve the equations of the layout VI and recall the corresponding simulation model in Amesim environment. Thus, the solution variables become the global parameters into the Amesim model.

Displacement Unit 1 V_1	220 cc
Displacement Unit 2 V_2	490 cc
Gear ratio 1 i_1	1.94
Gear ratio 2 i_2	1.38
Rear axle ratio i_{axle}	0.2686

Table 4.4: Design solution (non-optimal).

The reference vehicle has to reach 120 km/h in 120 s, and in the Figure 4.21 the difference between the target speed and the real speed of the bus is shown. There is a good correlation between the two lines, with no oscillations during the ride.

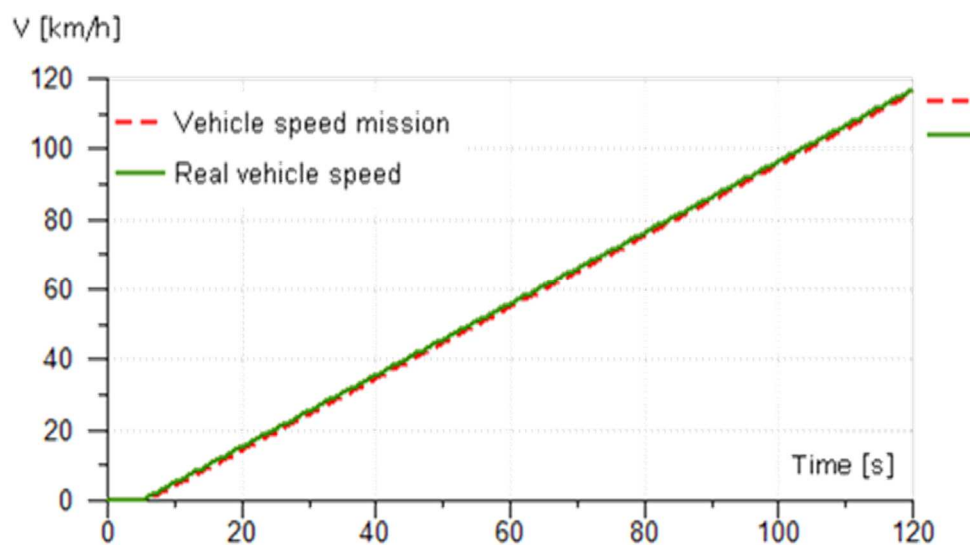


Figure 4.21: Target velocity and real velocity of the city bus.

The displacement control loop compares the real speed and the target speed, and the instantaneous difference is shown in the plot in Figure 4.22 with the output from the PI control.

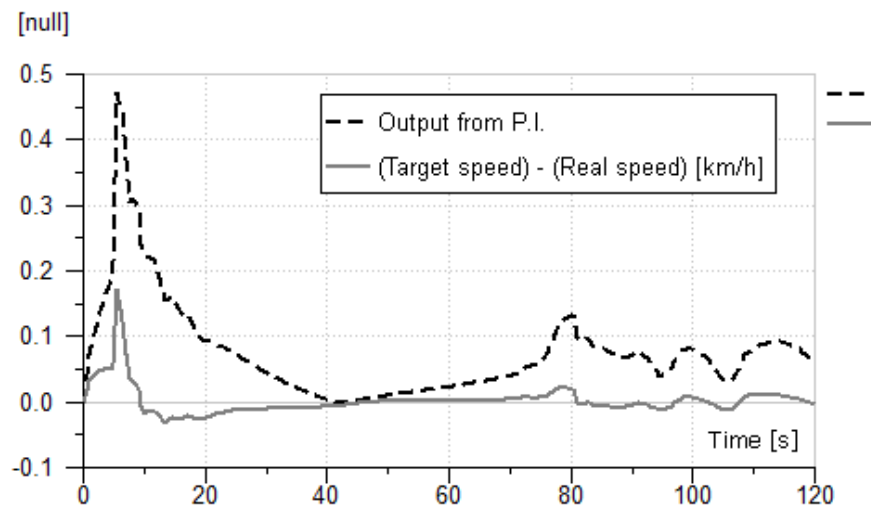


Figure 4.22: Difference between the target speed and the real speed of the vehicle.

During the drive, the two hydraulic units change the displacement according to the Figure 2.11. The trend of both the fractional displacement is shown in Figure 4.23, and the rotational speed of the two units is represented in Figure 4.24.

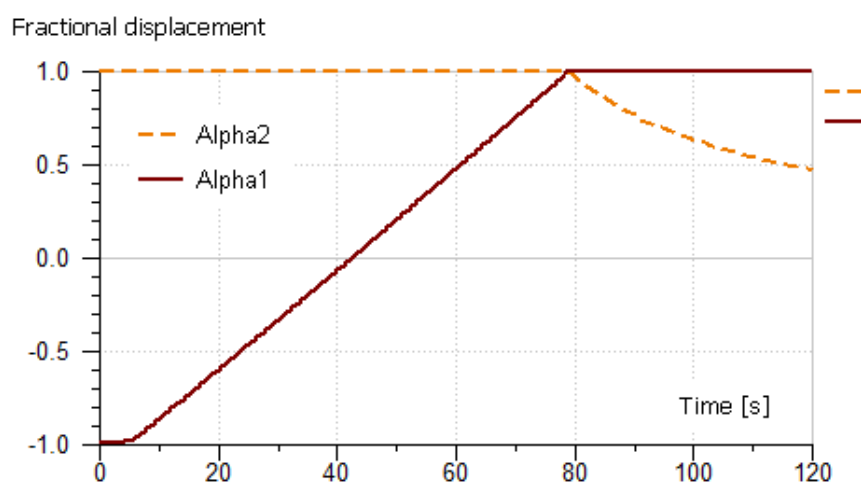


Figure 4.23: Fractional displacement of Unit 1 (Alpha 1) and Unit 2 (Alpha 2).

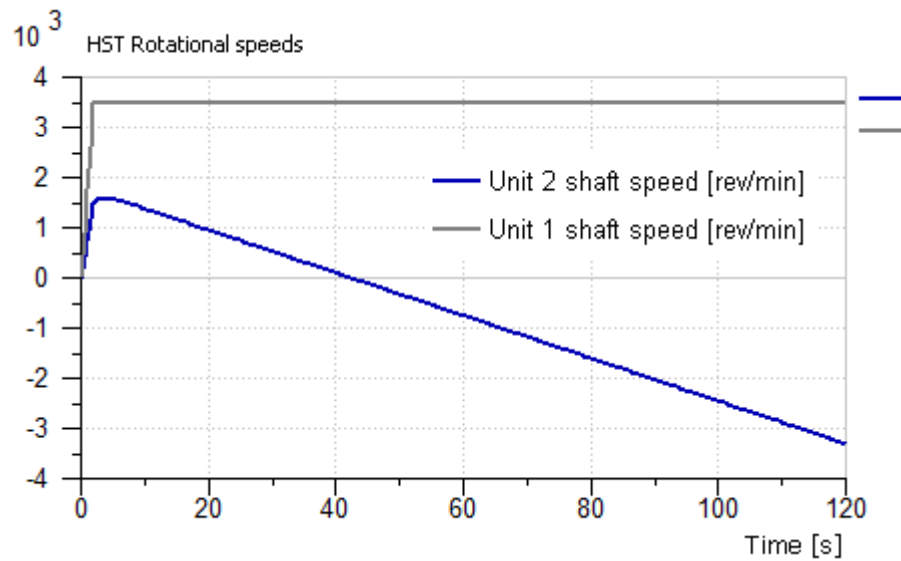


Figure 4.24: Rotational speed of Unit 1 and Unit 2.

The speed of Unit 1 is constant because it is connected to the engine, which rotates with a constant speed. Unit 2 decreases its speed from 1552 rev/min to -3286 rev/min; at 42.5 s the speed becomes zero and the full mechanical point occurs. The corresponding vehicle speed in that point is around 36 km/h, in agreement with the $f_{mp} = 0.3 * v_{max} = 36$ km/h.

The rotational speeds of the planetary geartrain and the rear axle speed are shown in the Figure 4.25.

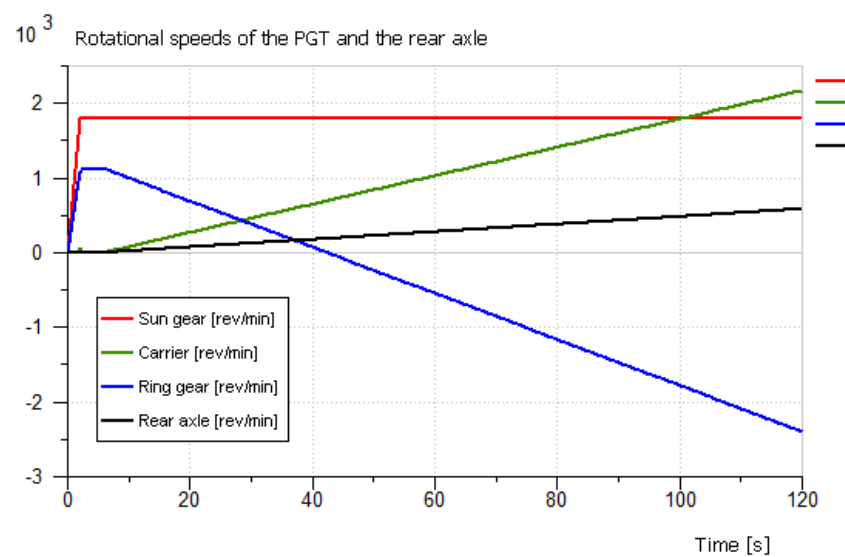


Figure 4.25: Rotational speed of the PGT and the rear axle ratio.

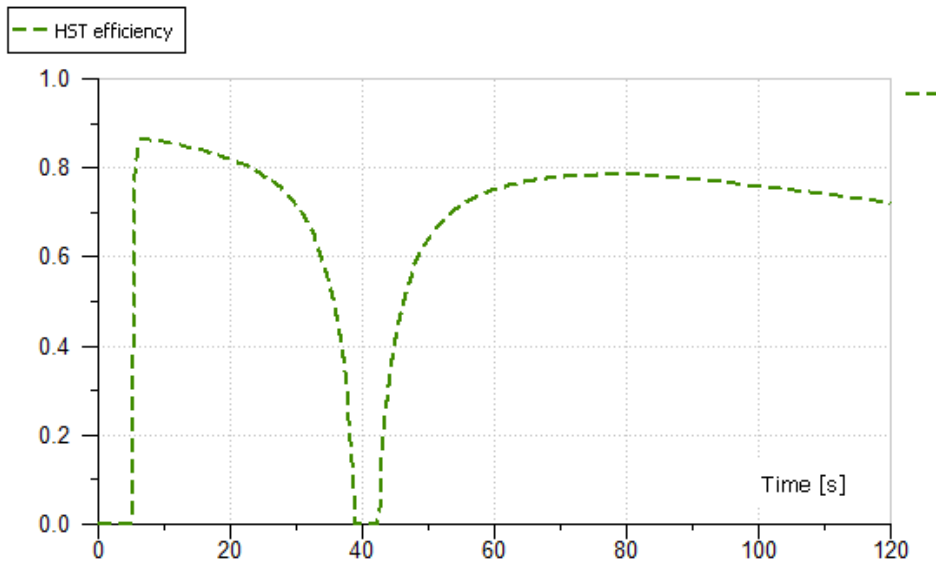


Figure 4.26: Efficiency of the hydrostatic transmission.

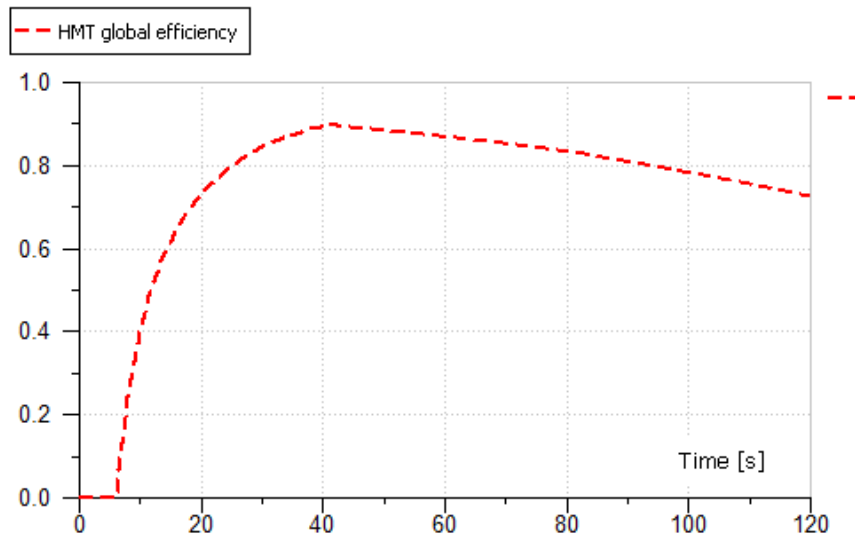


Figure 4.27: Global efficiency of the hydromechanical transmission.

In Figure 4.26 the efficiency of the hydrostatic transmission is shown. The optimal value is reached when the hydraulic units work far from the full mechanical speed. At last, the efficiency of the transmission is shown in Figure 4.27. The highest value, around 0.88 is reached in correspondence of the full mechanical point speed. Before that, the transmission works in recirculating mode, while after that point the additive mode occurs, according on what explained in the section 2.3.1. The integral average value of the

global efficiency is 0.818, calculated between zero and the maximum speed (equation 4.82).

4.8 Results: the efficiency matrix

The optimization procedure is implemented in MATLAB environment. Here, the efficiency of the transmission is calculated for several values of the design variables ϑ and the fmp and the best efficiency point is visible in the efficiency matrix.

To solve the procedure, we consider 24 values of the polar coordinate ϑ , and in particular four values within each interval $[a,b]$ $[c,d]$ $[e,f]$ $[g,h]$ $[i,l]$ $[m,n]$. No values are considered outside those intervals. For the fmp , we consider a minimum and a maximum values, 0.3 and 1 respectively, which means that the lowest value of the fmp is at 0.3 of the maximum speed, and the highest value corresponds with the maximum vehicle speed. In general, that limitation could be rearranged in base of the use. According to those design variables, the resulting efficiency matrix is shown in Figure 4.28. The efficiency is function of the two design variables and represented in normalized coordinates.

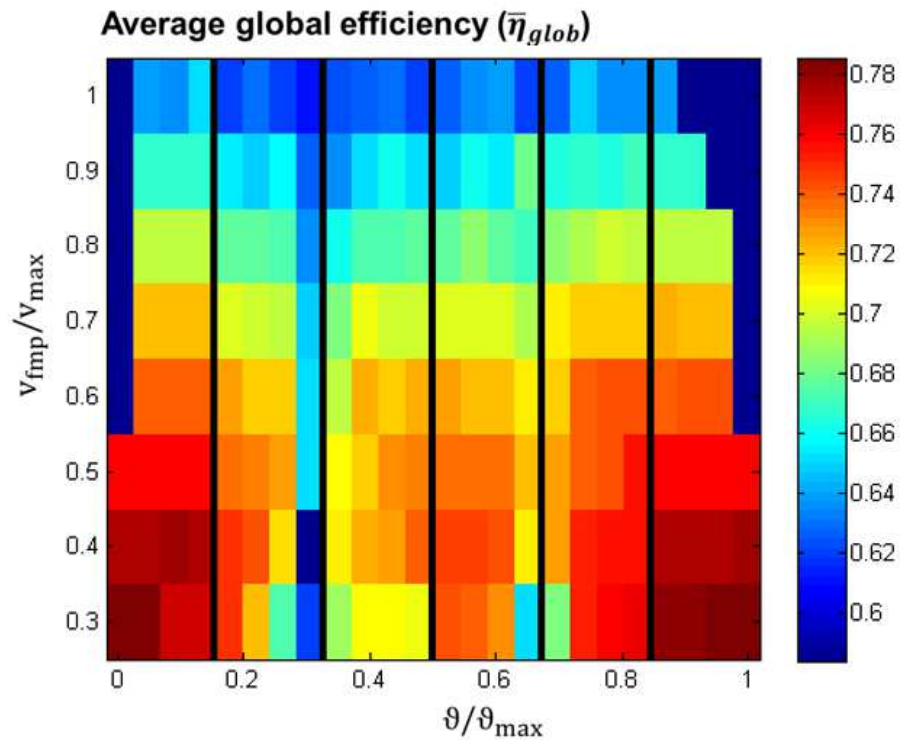


Figure 4.28: Optimization result: efficiency matrix for the city bus.

The plot is divided in six sections, one for each layout, and for each section, four different values of efficiency have been calculated.

The color scale indicates the average global efficiency, which goes from low values (0.55) represented with a blue squares to high values (0.78) represented with the red squares.

The maximum average efficiency in the case of the city bus $\overline{\eta}_{HMT} = 0.79$, reached for $\vartheta^* = 6,162$ and $fmp=0,3$. This solution corresponds to $i_0 = -0.423$ and the layout VI. Another good solution ($\overline{\eta}_{HMT} = 0.76$) is therefore obtained for the Layout I.

Some consideration on the results could be made. In the Figure 4.28 the difference between the layouts is very clear to see, because it depends on the polar function equations. In particular, there is a continuity in the plot between the layouts I and II, III and IV, V and VI, pointed out by the colors. In fact, for each of the three sections a local maximum could be reached. However, the global maximum is widely reached for the VI layout and for the layout I. This is primarily due to the lower speeds realized in output on the planetary geartrain, whose increment the efficiency than the other solutions.

The efficiency changes also by changing the fmp , however the variation of this last influences in the same way the efficiency for different layouts. In fact, the efficiency decreases going from low to high values of fmp . Considering lower fmp speed, the power recirculating in the hydrostatic transmission is lower; consequently, an improvement of the global efficiency occurs.

The results are also in agreement whit the literature (Blake et al. 2005) where the layout VI is taken into account to design a passenger bus with the same characteristics of the current one, but with a slight difference in the choice of the planetary geartrain ratio.

The procedure could be applied for different vehicles, changing the design constraints, the vehicle parameters and the design data.

In this study an input coupled transmission was considered, and the same methodology could be used to optimize an output coupled transmission and the relative layouts, adapting the design equations.

A possible future step of the method could be the application of a direct search algorithm, and in particular the Particle Swarm optimizator, to speed up the evaluation of the best efficiency point of the transmission.

5 PREDICTION OF THE SOUND PRESSURE LEVEL FOR A DUAL-STAGE HYDROMECHANICAL TRANSMISSION

5.1 Introduction

The hydraulic units inside the CVT create structural vibrations because of the internal pressure variation and the flow pulsations that are distributed along the hydraulic line (Klop *et al.* 2010). The manufacturers of hydrostatic transmissions are aware of the necessity to improve the acoustic characteristics at the early stages of the design. This necessity arises from two technological requirements: the compliance to regulations in terms of sound emissions, which impose a severe limitation on every type of apparatus; the improvement of the acoustic performance imposed by market requirements, because high noise levels in new products is increasingly less tolerated. Consequently, technical research has been focused on the investigation of noise sources (Klop and Ivantysynova 2011) and on the minimization of noise problems in hydraulic machines (Kumar *et al.* 2011, Kumar and Ivantysynova 2011). On the other hand, the engine manufacturers have made progress in minimizing the noise generated by their products.

Dual-stage transmissions, and more generally power-split transmissions (Renius 2005), have been mainly designed for agricultural use; therefore, they have never been subjected to the issue of noise. In fact, their widespread use has never created significant inconvenience.

Recent studies (Macor and Rossetti 2013) have shown that a dual-stage transmission could be suitable in applications where frequent stops and starts occur, such as urban buses. Accordingly, the noise problem can be decisive. For this reason, more literature has been focused on the investigation of noise sources (Klop and Ivantysynova 2011) and the minimization of the noise problem in hydraulic machines (Kumar *et al.* 2011).

Therefore, the subject of the present study is the noise emissions of a dual-stage transmission installed on an urban bus. It aims to understand whether a

dual-stage transmission generates a significant noise contribution compared to that generated by the engine.

To answer this question without resorting to expensive bench tests, a model of an urban bus was developed by using the Amesim simulation environment. The models of the engine and the transmission were integrated with a model for predicting sound emission. It's important to highlight that the current study was principally proposed as a tool for designing the system, considering noise emission the most important variable in the system.

5.2 Dual-stage hydromechanical transmission

The typical configuration of a dual-stage transmission is shown in Figure 5.1 (Blake *et al.* 2006). The dual-stage planetary gearbox, a feature from which the entire system takes its name, allows a gear change. This is performed by the clutch when the rotational speed of the planet carrier B and the sun gear A2 are synchronous.

Unlike the traditional IC transmissions, for dual-stage systems, the FMP occurs twice: once preceding and once following the gear shift. Consequently, the high efficiency operating zone occurs twice.

The design of the transmission has been performed by following the guidelines suggested by Blake *et al.* (2006). The main technical data of the vehicle and the engine are reported in Table 5.1, and the results of the transmission design are summarized in Table 5.2.

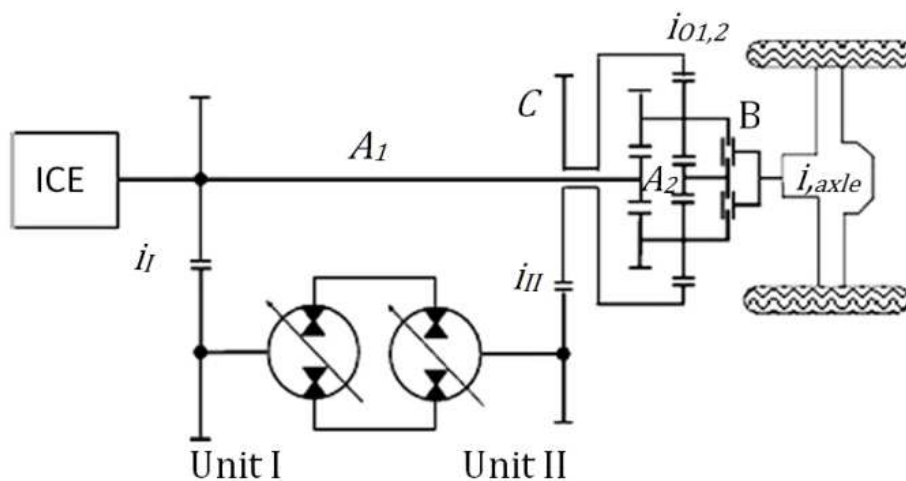


Figure 5.1: Schematic of the IC dual-stage transmission.

General bus features	
Model	Citelis-12
Constructor	Iveco Irisbus
Number of seats	86
Total weight	12000 [kg]
Total height	3.3 [m]
Total width	2.5 [m]
Total length	12 [m]
Total admissible weight	19845 [kg]
Maximum speed	68 [km/h]
Type of tires	275/70 R22.5

General engine features	
Model	Iveco Cursor 8
Fuel	CNG
Motor type	6 cylinders in line
Maximum power	190 [kW]
Max rotational speed	2050 [rpm]
Maximum torque	1076 (at 1100 rpm) [N·m]

Table 5.1: Main engine and vehicle features for the IC dual-stage transmission.

Design transmission variables	
n_{ICE}	1500 [rpm]
R_{tire}	0.448 [m]
p_{max}	450 [bar]
T_{wheel}	20600 [N·m]
$\tau_{mech,1}$	0.5
$\tau_{mech,2}$	2
τ_{shift}	1
i_{axle}	10.143
$i_{0,1}$	-1
$i_{0,2}$	-3
i_I	0.58
i_{II}	0.581
V_I, V_{II}	125 [cc]
$n_{I,max}, n_{II,max}$	2600 [rpm]

Table 5.2: Main design variables of the transmission.

5.3 Simulation model of the vehicle

The block scheme of the vehicle model is shown in Figure 5.2. As the transmission is kinematically separate from the engine, two different control systems provide for the engine management and for the transmission management. In the block simulating the driver, the vehicle speed is compared to the mission speed, to produce as output, a command s , either for the accelerator or the brake. The accelerator command s enters the

control strategy block to manage the engine according to a predetermined criterion. In this way, the engine supplies the required power to the transmission.

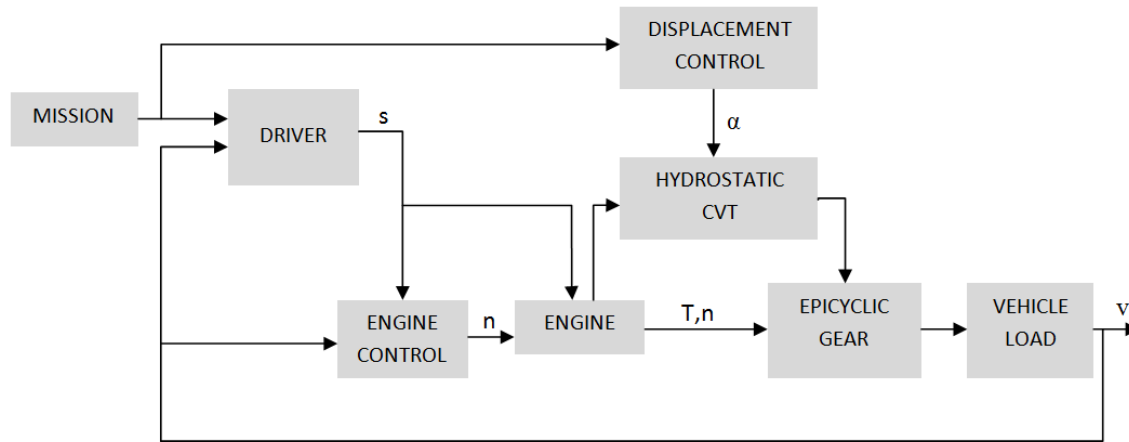


Figure 5.2. Block diagram of the vehicle test.

The transmission is equipped with a control system that receives the vehicle speed signal, and acts on the displacement partialization of the two hydraulic machines in order to impose the desired speed.

From the perspective of efficiency and noise emission, the most important parts of the model are the engine and the hydraulic CVT. They were shaped as described in the following paragraphs.

5.3.1 Engine

The engine model is an ideal torque generator, whose driving torque was a function of the rotational speed and the accelerator position. The fuel consumption was calculated based on the actual load and rotational speed. The speed–torque characteristic of the engine and the fuel consumption data were derived from experimental measurements reported in the literature by De Simio *et al.* (2010). The normalized engine map and the experimental points are reported in Figure 5.3. In order to take into account the engine dynamics, the engine inertia and a first-order delay on the accelerator signal have been added to the torque generator model, as shown in Figure 5.4.

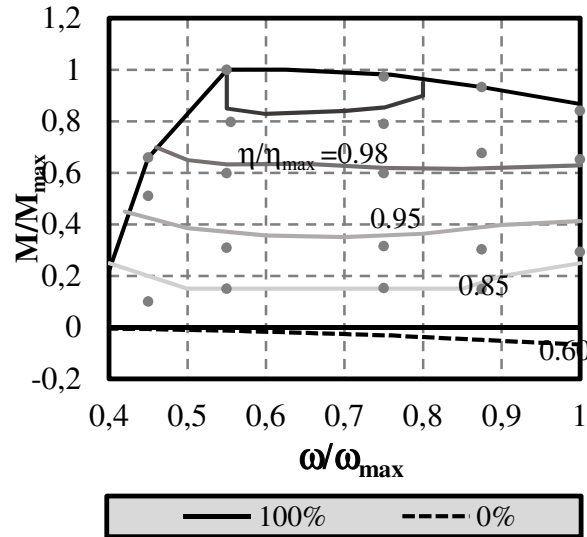


Figure 5.3: Normalized efficiency map of the engine.

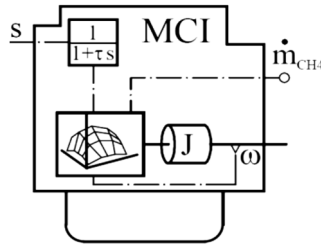


Figure 5.4: Dynamic model of the engine.

5.3.2 Hydromechanical transmission

The model of the hydraulic CVT (Figure 5.5) is based on a standard scheme for a hydrostatic transmission, comprising relief valves, a boost pump, and counter pressure and flushing valves. The hydraulic units were modelled by applying to an ideal unit the main loss sources (Figure 5.6): friction by means of braking (C), and leakage from the orifices d .

In order to consider the influence of the operating conditions on the unit losses, both the friction torque of the brake C and the diameter of the orifices (d_{flow} , d_{lk}) were expressed as polynomial functions of the rotational speed, the pressure difference, and the actual to maximum displacement ratio. The coefficients of the loss models were fitted to the experimental data that were supplied from a

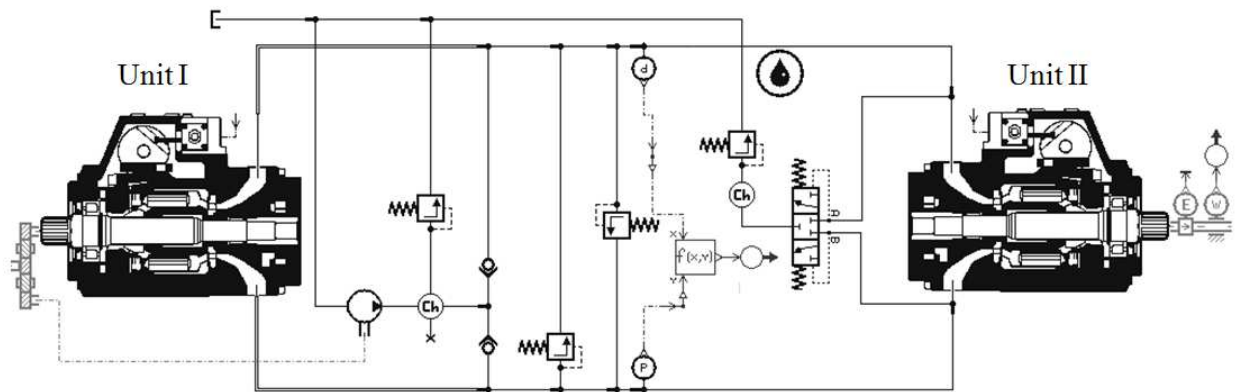


Figure 5.5: Scheme of hydrostatic CVT transmission.

manufacturer. The overall performance of the hydraulic CVT is reported in Figure 5.7 for different operating conditions.

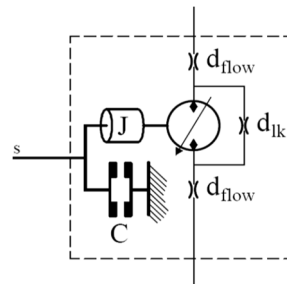
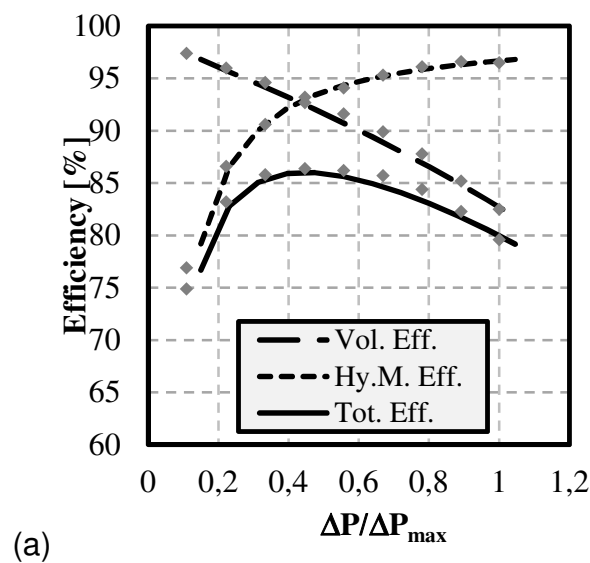
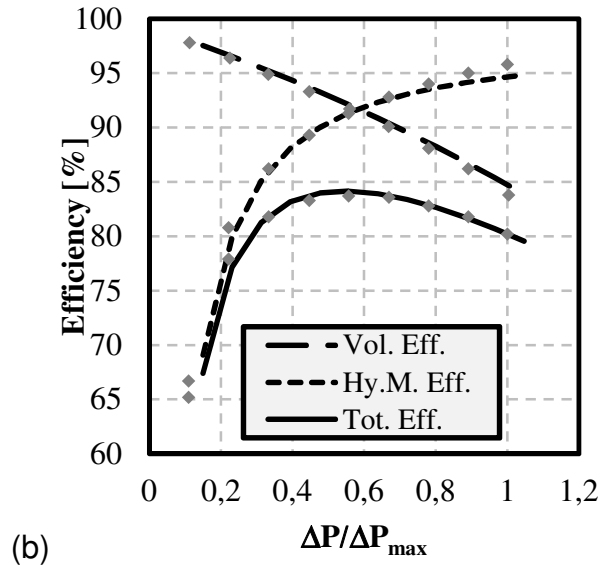


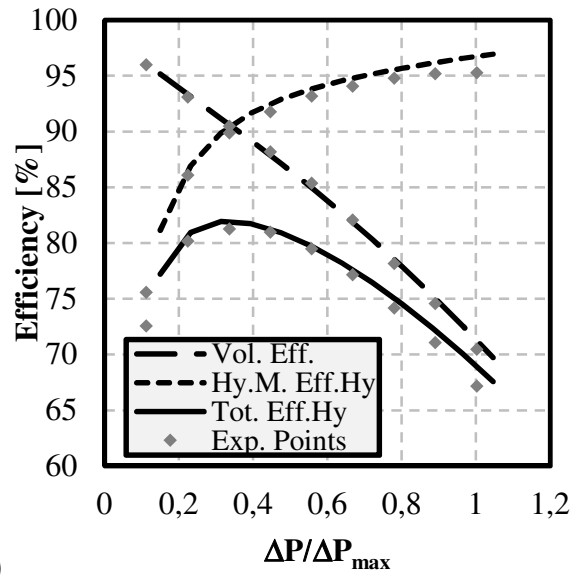
Figure 5.6: Real pump functional scheme.

Each gear pair of the model, both ordinary pairs and pairs belonging to the epicyclical gear, was assumed to have a constant efficiency $\eta_{\text{gear}} = 0.980$.





(b)



(c)

Figure 5.7: Efficiency of the hydraulic CVT for different operating conditions. Comparison between the model (lines) and the experimental data (dots):

(a) $n_I = 2000$ rpm, $\alpha_I = 1$, $\alpha_{II} = 1$;

(b) $n_I = 3000$ rpm, $\alpha_I = 1$, $\alpha_{II} = 1$;

(c) $n_I = 2000$ rpm, $\alpha_I = 0.5$, $\alpha_{II} = 1$.

The vehicle loads, i.e., the aerodynamic friction and the rolling friction, were modelled according to: $F_a = 0.5 c_d v^2 A_f$ and $F_r = k_r Mg$, whereas for the vehicle mass, the drag coefficient, the frontal area, and the rolling resistance

coefficient the following values were respectively assumed: $M = 14,900 \text{ kg}$, $c_d = 1.18$, $A_f = 7\text{m}^2$, $k_r = 0.008$.

The engine control strategy adopted in this model is the simplest one, known as *Speed Envelope* (Pfiffner and Guzzella 2001). It operates based on vehicle speed and accelerator position, without any optimization procedure.

5.4 Sound pressure level prediction models

The sound pressure contributions of the engine and the hydrostatic transmission are considered as predominant. The sound emission of other components, such as the planetary gearbox, the mechanical gears, and the hydraulic elements, proved to be very low, and therefore, negligible.

5.4.1 Internal combustion engine noise emissions

The noise generated by an engine is a function of several design variables. It would be a hard task to take all of them into account. A simplified predictive model for high power engines (Blair and Spechko 1972, Bies and Hansen 1988), therefore, was applied, which considers the three most important contributions to noise emissions: the engine exhaust system noise, the engine housing noise, and the air intake system noise. This is a good reference model for noise forecasts; however, it may overestimate the noise values because it is dated. The new-generation engines are quieter and more isolated than the old-generation ones.

The sound power level generated by the engine exhaust system with a muffler can be estimated using expression (5.1):

$$L_{W,ex} = 108 + 10 \log_{10} P + K - (l_{ex}/1.2), \quad (5.1)$$

where the K parameter indicates the reduction factor owing to the presence of the turbocharger. The l_{EX} parameter indicates the length of the exhaust pipe, and P indicates the instantaneous gas engine power.

The noise emission related to engine housing is not univocally determined from only one relation, because it is a function of both the engine speed and the structure of the system. The sound power level is obtained by using expression (5.2):

$$L_{W,h} = 93 + 10 \log_{10} P + J_1 + J_2 + J_3 + J_4, \quad (5.2)$$

where P indicates the instantaneous engine power, and parameters J_1, J_2, J_3 are the correction terms as functions of the engine rotational speed (5.3), the fuel type (5.4), and the cylinder arrangement (5.5), respectively:

$$\begin{cases} J_1 = -5 & n \leq 600 \text{ rpm} \\ J_1 = -2 & 600 \text{ rpm} < n \leq 1,500 \text{ rpm} \\ J_1 = 0 & n > 1,500 \text{ rpm} \end{cases} \quad (5.3)$$

$$\begin{cases} J_2 = 0 & \text{diesel engine or dual fuel engine: diesel/CNG} \\ J_2 = -3 & \text{CNG engine} \end{cases} \quad (5.4)$$

$$\begin{cases} J_3 = 0 & \text{cylinder arrangement: in – line} \\ J_3 = -1 & \text{cylinder arrangement: V – type and radial type} \end{cases} \quad (5.5)$$

In order to obtain the sound power level values filtered through the A-weighting curve, a correction (5.6) of their total value must be made.

$$\begin{cases} J_4 = -4 & n \leq 600 \text{ rpm} \\ J_4 = -3 & 600 \text{ rpm} < n \leq 1,500 \text{ rpm} \\ J_4 = -1 & n > 1,500 \text{ rpm} \end{cases} \quad (5.6)$$

To predict the sound power level for the engine intake air system with the turbocharger, the following relation (5.7) may be used:

$$L_{W,int} = 92 + 5 \log_{10} P - l_{int}/1.8, \quad (5.7)$$

where l_{int} is the length of the input duct.

To compare the results obtained from the expression 5.7 and the output result from the hydrostatic noise model, the sound power level must be converted

into a sound pressure level (SPL). Toward this end, it is possible to calculate the sound pressure level at a certain distance r if the sound power level of a point source and its acoustic directional factor Q are known, using expression (5.8) for diffused fields:

$$L_p = L_W + 10 \cdot \log \left(\frac{Q}{4\pi r^2} + \frac{4}{R} \right). \quad (5.8)$$

The acoustic room constant R was obtained in (5.9), knowing the total surface S and the acoustic absorption coefficient a_{abs} by using Sabine's law (Moncada and Santoboni 1995).

$$R = S \cdot a_{abs} / (1 - a_{abs}) \quad (5.9)$$

Assuming the noise point source in the free field is omnidirectional, the simplified expression (5.10) can be applied for a power source placed on a reflective semispherical plane ($Q = 2$), and assuming that the surface is not characterized by absorption. For a distance equal to 1 m from the noise source, the expression becomes

$$L_p = L_W - 20 \cdot \log r - 8. \quad (5.10)$$

When more than one source emits a known sound pressure, the overall sound pressure level can be calculated by means of relation (5.11):

$$L_{p,TOT} = 10 \cdot \log_{10} \sum_{k=1}^N 10^{\frac{L_{p,k}}{10}}, \quad (5.11)$$

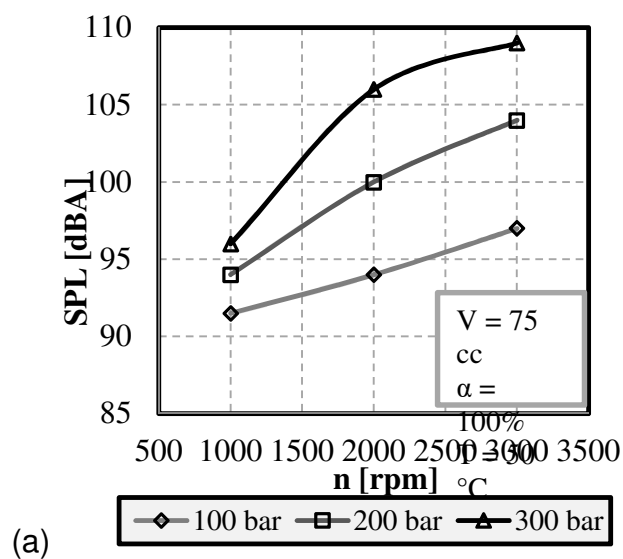
which expresses the sum of the individual sound pressures in decibels. Expression (5.11) will also be used for summing the sound contributions of the gas engine and of the hydrostatic CVT.

5.4.2 Hydrostatic CVT noise emissions

The literature indicates there are three main noise sources in a hydraulic system (Bies and Hansen 1988): the *air-borne noise*, that is, the noise transmitted through the air, which is that most easily heard by humans; the *fluid-borne noise*, generated by an irregular fluid flow and by certain pressure oscillations in the hydraulic pipeline; the *structure-borne noise*, which is the noise generated by the mechanical and structural elements of the machine.

The author developed a polynomial formulation of the noise produced by the Hydrostatic CVT, based on experimental noise data collected by the manufacturer under various operating conditions. Phonometric measurements were performed in an anechoic chamber on two hydraulic axial piston pumps, at a fluid temperature of 50 °C. They relate to the A-weighting curve.

The curves of the sound pressure level are presented as a function of hydraulic pressure and rotational speed. In Figures 5.8(a) and (b) the curves refer to a displacement of 75 cc and a partialization equal to $\alpha = 100\%$ and $\alpha = 25\%$, respectively. In Figures 5.8(c) and (d) the curves refer to a displacement of 90 cc and a partialization equal to $\alpha = 100\%$ and $\alpha = 25\%$, respectively. As can be seen in the graphs, the SPL increases with both the pressure and the rotational speed; it decreases with the displacement of the hydraulic units.



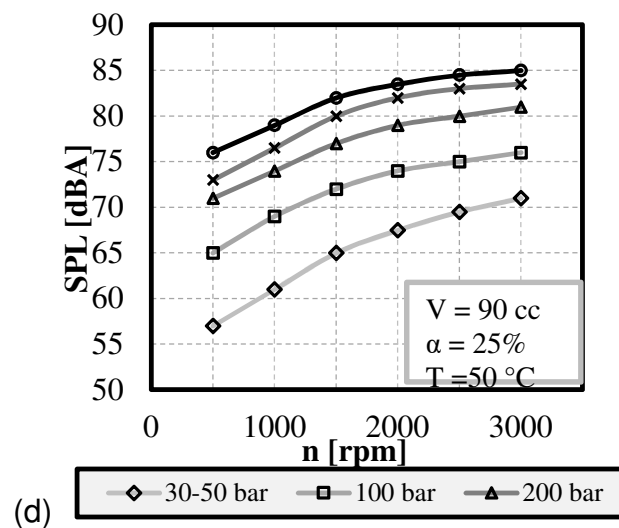
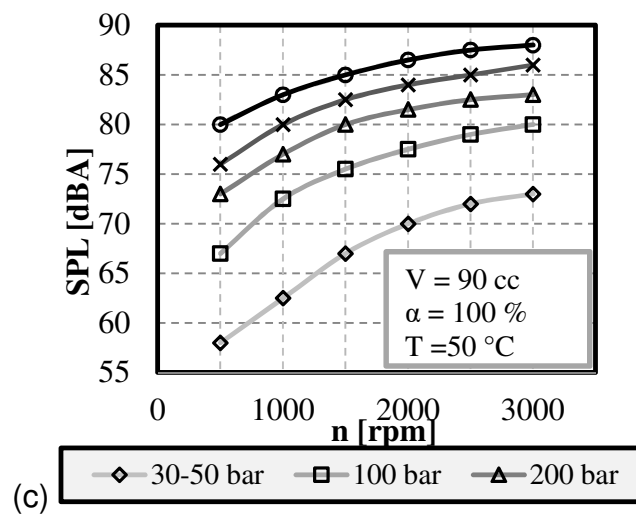
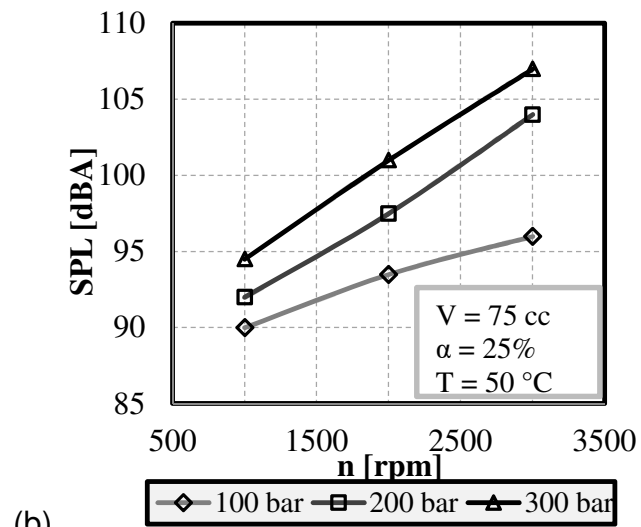


Figure 5.8: SPL as a function of pressure and rotational speed of the hydrostatic unit: (a) $\alpha = 100\%$ and $V = 75$ cc; (b) $\alpha = 25\%$ and $V = 75$ cc; (c) $\alpha = 100\%$ and $V = 90$ cc (d) $\alpha = 25\%$ and $V = 90$ cc.

The experimental curves have been interpolated by means of a polynomial equation expressed as a function of pressure, rotational speed, displacement, and partialization of displacement:

$$SPL[dBA] = f(p, n, V, \alpha). \quad (5.12)$$

The general form of the equation is the following (5.13):

$$SPL(dBA) = An^2 + Bn + C, \quad (5.13)$$

where coefficients A, B, and C have been obtained through regression of the experimental data and are of the form shown in (5.14):

$$A = a_A \cdot (\alpha V) + b_A \quad B = a_B \cdot (\alpha V) + b_B \quad C = a_C \cdot (\alpha V) + b_C \quad (5.14)$$

$$\begin{aligned} a_A &= k_A \cdot (p) + \gamma_A & a_B &= k_B \cdot (p) + \gamma_B & a_C &= k_C \cdot (p) + \gamma_C \\ b_A &= k'_A \cdot (p) + \gamma'_A & b_B &= k'_B \cdot (p) + \gamma'_B & b_C &= k'_C \cdot (p) + \gamma'_C \end{aligned}$$

In Equation (5.14), coefficients A, B, and C are expressed as a function of the product of the partialization and displacement, and of parameters a_A , b_A , a_B , b_B , a_C , and b_C . These parameters are considered dependent on the hydraulic operative pressure only.

In Figure 5.9, the curves resulting from the application of the model (dots) are compared with the experimental data (lines) for a displacement of 90 cc and a fractional displacement equal to 100%.

The low-grade polynomial nature of the experimental model permits a satisfactory forecast of the noise values, even for pressures and rotational speeds outside the ranges shown in Figure 5.9. The average root mean square error between the experimental data and the prediction of the polynomial model was approximately 0.469 dBA, with a maximum value of 1.14 at 200 bar and 1,500 rpm.

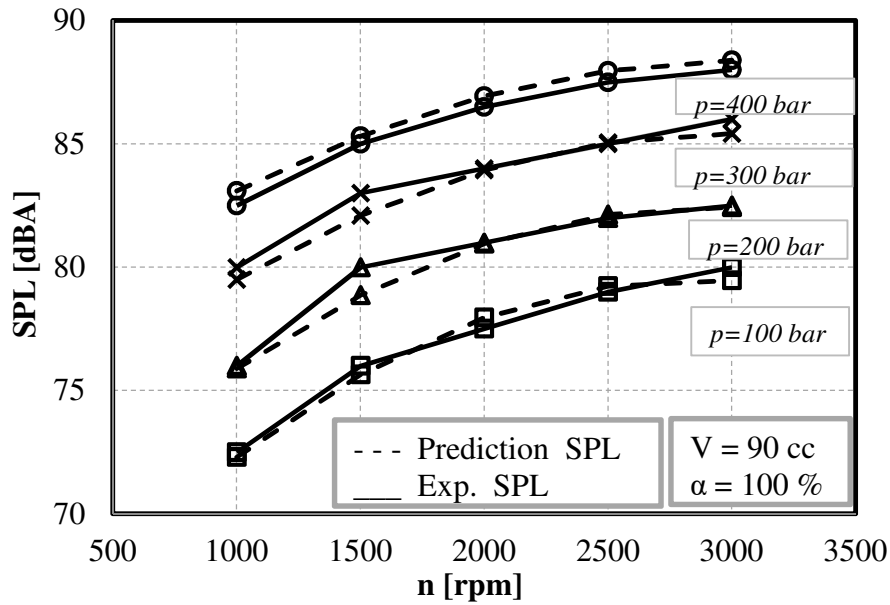


Figure 5.9: Experimental (lines) and prediction (dots) curves for a displacement of 90 cc and a fractional displacement $\alpha = 100\%$.

5.5 Simulation and results

Two speed missions were used in the simulations: the trapezoidal mission (Figure 5.10) and the Manhattan mission (Figure 5.11) (DieselNet 2014). They are presented along with their characteristic parameters in Table 5.3 and Table 5.4, respectively. The latter simulates the real route for an urban bus with sudden accelerations, deceleration, and frequent starts and stops, which is typical for this type of vehicle; the former is used to highlight the relative importance of the noise sources in a simple acceleration–deceleration test.

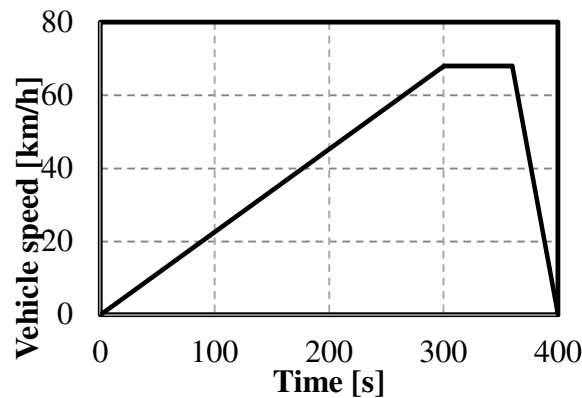


Figure 5.10: Trapezoidal speed mission.

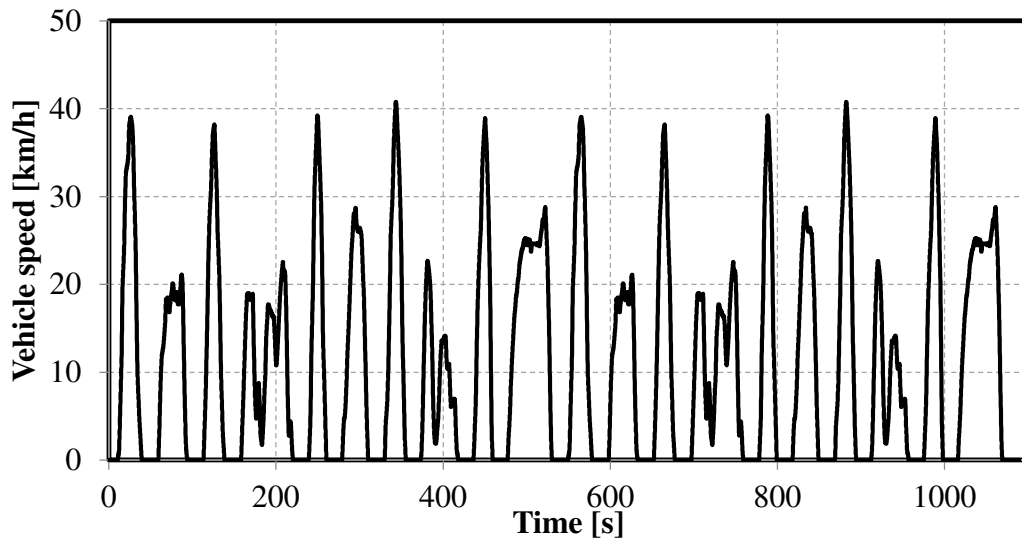


Figure 5.11: Manhattan speed mission.

Duration	400	[s]
Stops time/Duration	100	[%]
Total driving distance	4.3	[km]
Maximum speed	68	[km/h]
Time at constant speed	60	[s]
Maximum acceleration	0.063	[m/s ²]
Number of stops per km	0	[-]

Table 5.3: Characteristic parameters of speed mission in Figure 5.10.

The sound pressure level of the two hydrostatic units is shown in Figure 5.12; it has been calculated according to equation (5.14). In the section where the vehicle has reached its maximum speed, the SPL is maintained at a constant level of approximately 80 dBA for both hydraulic units. In fact, in this section, no quantity affecting the noise varies. In the acceleration section, the SPL is affected by the variation in the hydraulic pressure (Figure 5.13(c)). During the gearshift operation, the pressure of Unit I decreases, whereas that of Unit II increases; this defines the discontinuity in the graph of sound emissions.

Duration	1089	[s]
Stops time/Duration	36	[%]
Total driving distance	3.33	[km]
Maximum speed	40.9	[km/h]
Average speed	11	[km/h]
Average speed without stops	17.17	[km/h]
Maximum acceleration	2.24	[m/s ²]
Number of stops per km	6	[-]

Table 5.4: Characteristic parameters of the Manhattan speed mission.

The sharp drop in noise emission of the two hydraulic units at the FMPs (at approximately 48 s and 190 s) is mainly due to the speed variation for Unit II, and the displacement variation for Unit I, as shown in Figures 5.13(a) and (b). Initially, the circuit works in a *negative circulating mode*: the first hydraulic unit works as a motor and the second unit works as a pump. After the FMP, the flow is inverted into the CVT, and the shift to *power-split mode* occurs.

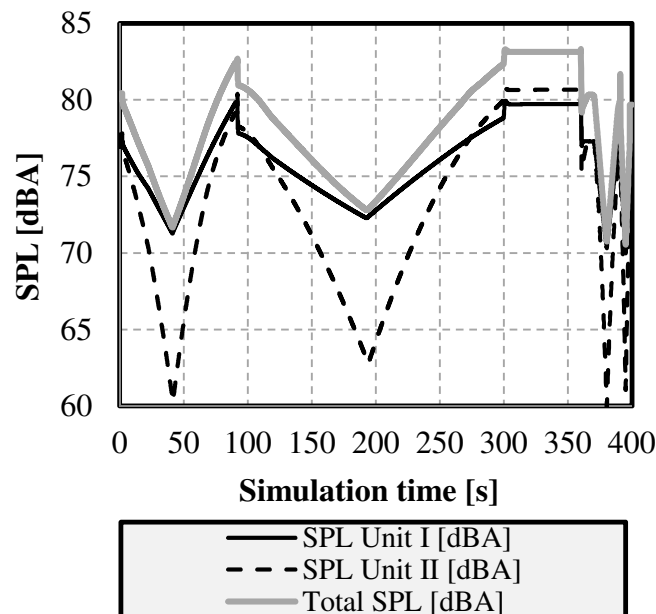
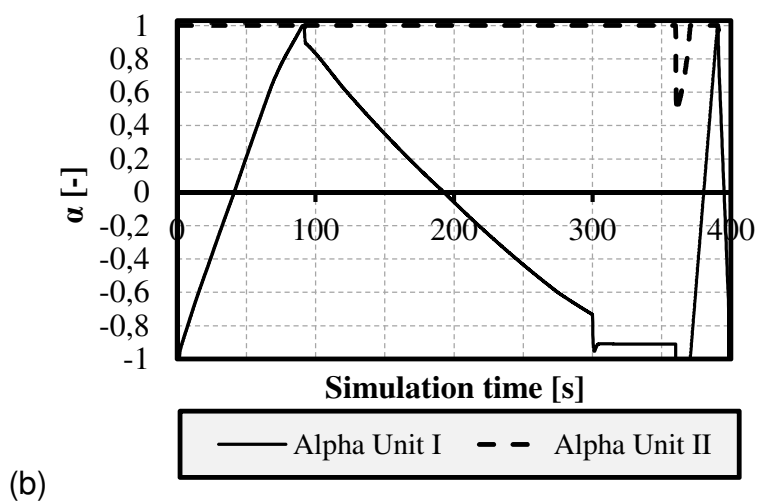
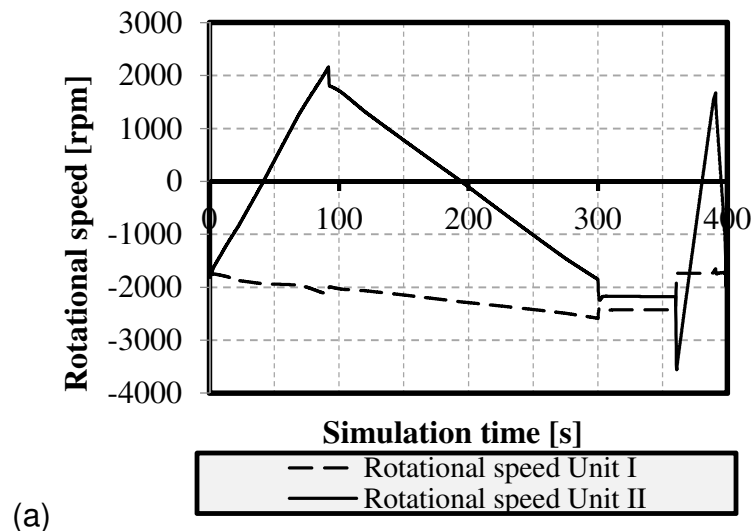


Figure 5.12. Sound pressure level of the hydraulic units.

In Figure 5.13, the speed, the displacement, and the pressure variation of the hydraulic units are shown. The speed of Unit II is imposed solely by the incoming flow rate, i.e., by the combined action of the speed (Figure 5.13(a)) and the displacement of Unit I (Figure 5.13(b)).



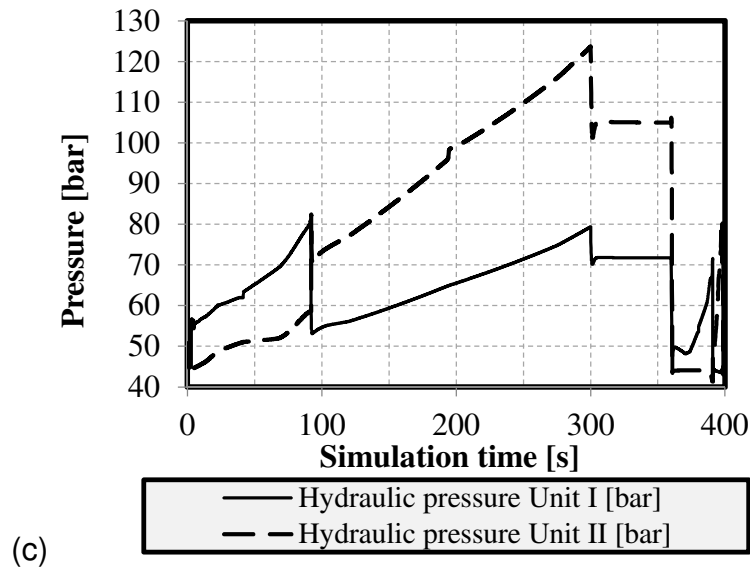


Figure 5.13: (a) Variation of rotational speed,(b) displacement partialization, and (c) hydraulic pressure of Unit I and Unit II.

The former slightly increases, according to the control strategy of the engine. In the first acceleration phase, and hence with the first gear inserted, the rotational speed of the wheels is imposed by the planet carrier B . When its rotational speed is synchronous with the speed of the sun gear A_2 , the clutches are activated, and the gear shift occurs. Then, the sun gear A_2 imposes the motion.

In Figure 5.14, the trend of the SPL of the gas engine along with all its contributions is reported. The trends are similar because the SPL depends mainly on the engine power, as shown by Equations (5.1), (5.2), and (5.7).

The predominant contribution to the sound emission in Figure 5.14 is the engine housing, which is followed by the exhaust system. This result agrees with what was observed in practice: the noise is created by the fuel combustion inside the cylinders, coupled with the effects of the structural vibration of the engine. During the deceleration, the contribution of the air intake system is the most significant. The other sound source in this phase, the braking system, is not considered in the present analysis.

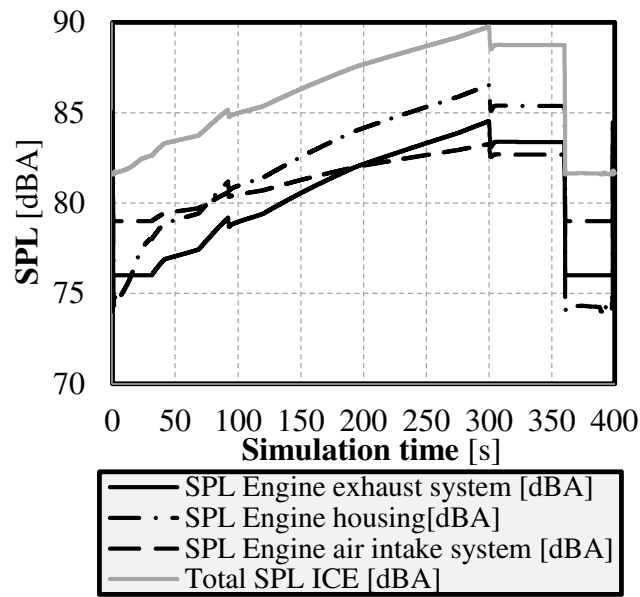


Figure 5.14. SPL of a gas engine with noise contributions.

The SPLs of the CVT and the engine are shown in Figure 5.15. They have been calculated via Equation (5.11) as the sum of all the contributions shown in Figure 5.13. As can be seen, the SPL of the engine is always greater than that of the transmission; the difference is particularly evident at the FMP, where it reaches 15 dBA.

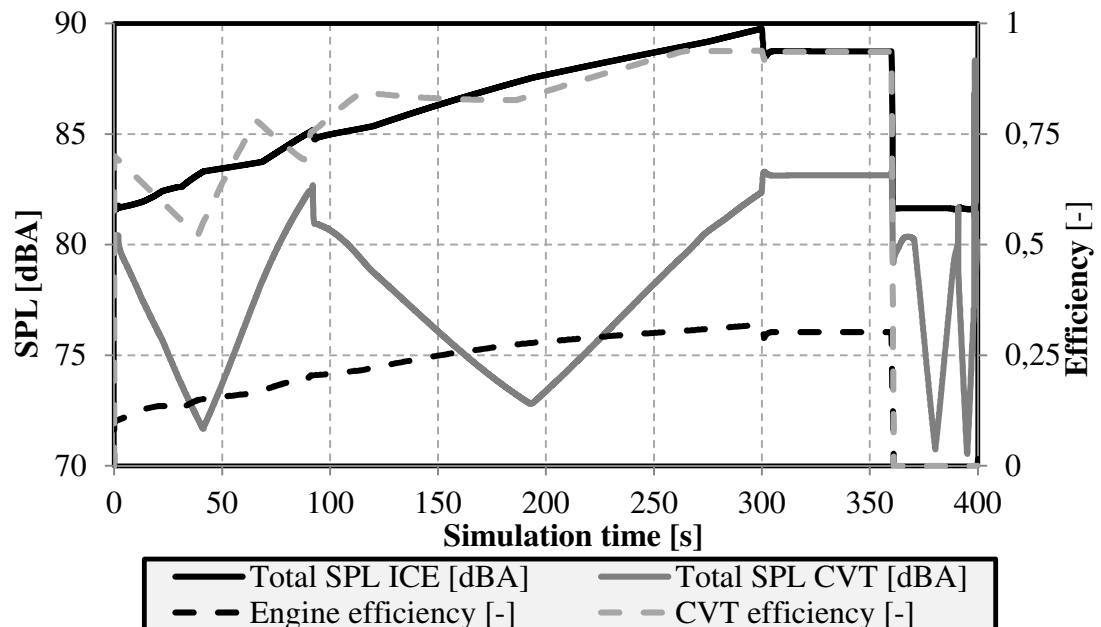


Figure 15. Comparison between the total SPL of a gas engine and the total SPL of a CVT.

The efficiencies of the engine and the CVT are also shown in Figure 5.15. The efficiencies and the SPL have similar trends. This can be explained by the fact that the efficiency and the noise depend on the same parameters. For the engine, they mainly depend on the power; for the CVT, they depend on pressure, speed, and partialization.

We can observe that the SPL value of the gas engine is generally higher than the SPL of the CVT, for values that are between 5 and 15 dBA. In the worst case, the SPL of the engine reaches approximately 90 dBA, versus the 83 dBA of the transmission. Despite this high SPL value, the CVT contributes slightly to the SPL of the engine-transmission group. In fact, considering the engine as the main noise source, the CVT contribution becomes equal to 0.8 dBA (Equation (5.11)).

The overall SPLs for the Manhattan mission are presented in Figure 5.16. Instead, in Figure 5.17, the first 100 s are zoomed in.

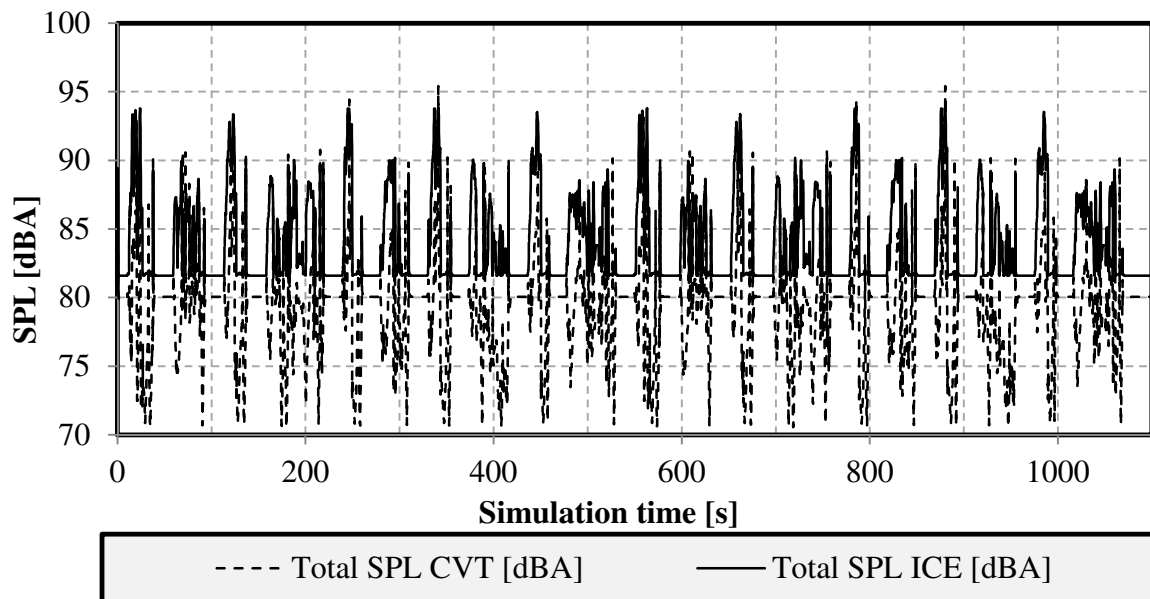


Figure 16. Total SPL for the *Manhattan* speed mission.

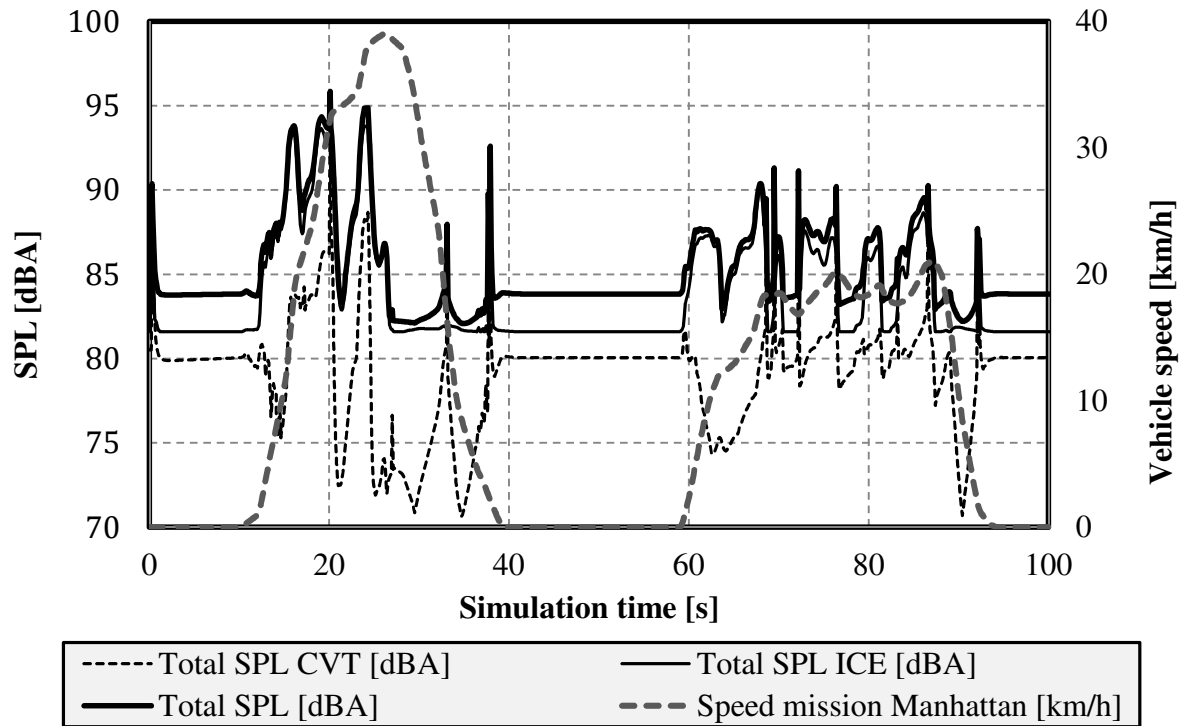


Figure 17. Total SPL for Manhattan speed mission in the first 100 s of simulation.

Even in this mission, the SPL of the engine is always greater than that of the CVT, and the transmission contribution to the noise generation is minor, as shown in Figure 5.17.

However, the CVT contribution also depends on the engine type. For CNG engines, as the present one, presenting SPL peaks of 85–90 dBA, the transmission contribution is equal to 0.8–2 dBA, because the transmission peak is 83 dBA. If diesel engines are considered, the SPL peak values of which range between 95 and 100 dBA, the transmission contribution drops to 0.1–0.3 dBA.

6. CHAPTER: EXPERIMENTAL ACTIVITY: THE TOYOTA PRIUS HYDRAULIC TEST RIG

6.1. Maha Fluid Power Research Center

The experimental activity performed within the scope of the present PhD work was conducted in the Maha Fluid Power Research Center at Purdue University (West Lafayette, IN, USA). The research center was formed in 2004 from an endowment from the fluid power pioneer, Otto Maha. Professor Monika Ivantysynova moved from TUHH (Hamburg, Germany) to Purdue University, to lead the new center.

The research in the laboratory is mainly focused on fluid-power innovative projects, and the working area is equipped with certain vehicles for testing, such as earthmoving machines, agricultural tractors, forklifts, and wheel loaders. A special area is dedicated to several bench tests for power transmission systems; each system is controlled and managed by means of remote supervision, and the supervisor is placed in a specific office. A south view of the laboratory is shown in Figure 6.1.



Figure 6.1: South view of the Maha Fluid Power Research Lab.

The Ph.D. research at the Maha Laboratory was carried out under the supervision of Professor Monika Ivantysynova, and the main topic was focused on the study of the

hydraulic version of the Toyota Prius power transmission. In the past, a test rig had been developed to test the output-coupled hydraulic hybrid powertrain. Although the measurements were in good agreement with the target variables, small oscillations in the results occurred. In Figure 6.2, an example of the original measurements is shown, illustrating the comparison between the real vehicle speed obtained from the bench test and the design target speed of the vehicle.

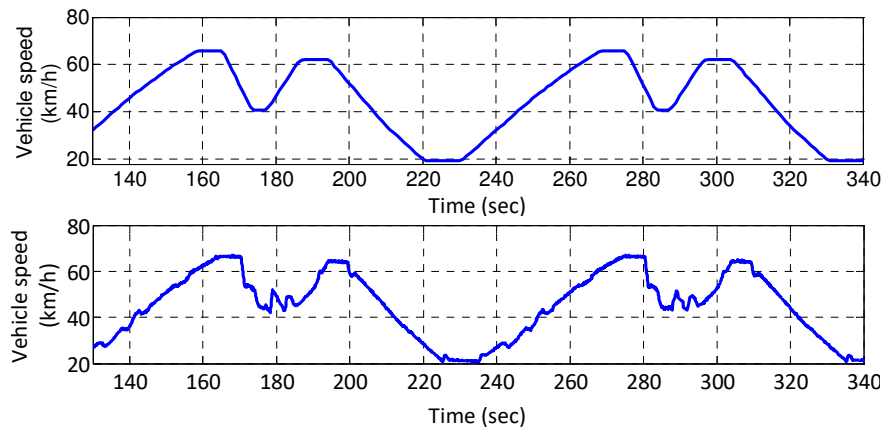


Figure 6.2: Design target speed versus the real vehicle speed.

The objective of the activity was to analyze and verify the test rig and all its components, thus developing a full technical documentation of the entire system, a complete and detailed schematic of the driveline, in order to acquire full knowledge of the transmission operation. A parallel objective was to propose changes in the hydraulic part with respect to the original test in order to overcome the issues that occurred in the first measurements.

The current study was conducted under the supervision of the author, whom exclusively had the responsibility to develop and carry out the project. It is important to note that the work explained here is only a part of the final project, to be carried out until the end of 2017, when the final transmission will be fully functional.

6.2. The Hydraulic Hybrid Toyota Prius

Over the past years, the increase in oil prices and the growing environmental concerns have forced the automotive manufacturers to show interest in advanced technologies. Lately, hybrid vehicles have been drawing a lot of attention, particularly after the surge in oil prices, and considering the US market, the Toyota Prius became

one of the best selling cars. It was originally equipped with a combined electric hybrid, a double connection between the engine and the drive axle: mechanical and electrical, allowing the decoupling of the power supplied by the engine from the power demanded by the driver. In a traditional vehicle, a larger engine is typically used to provide acceleration from a standstill to a certain steady speed. This is owing to the minimal torque of the engine at lower rotational speeds. The electric motor, instead, allows the maximum torque at stall and can be used to overcome the engine deficiency at low speeds. According to this, a smaller and highly efficient engine can be adopted. At lower speeds, this system operates as a series hybrid electric vehicle (HEV), whereas at high speeds, where the series powertrain is less efficient, the engine takes over.

The hydraulic hybrid vehicle (HHV) is composed of an internal combustion engine as primary source of power, and a hydraulic accumulator as the secondary source of energy. In addition, it includes a hydraulic pump and a motor that charges and discharges the accumulator. HHV is an alternative to the HEV and offers some advantages compared to its electric counterpart. For instance, the hydraulic units (pump and motor) have a substantially higher specific power rating compared to the electric units (generator and motor) in the HEV, which render them more compact. Moreover, the hydraulic accumulators have a higher specific power rating than the typical battery in the HEV (Kumar, 2010).

The higher power density of the HHV components makes them particularly attractive for vehicle applications, when many stop-and-go are required.

Considering the Toyota Prius 2004 as a reference for electric vehicles, a study conducted by Kumar (2010) proved the hypothesis that there is hydraulic hybrid architecture of hydraulic hybrid that, when combined with a control strategy, it can achieve higher fuel economy than its electric counterpart.

After investigating different architectures of hydraulic hybrid power-split transmissions, the output-coupled was selected as the reference because it is the most promising regarding the reduction in fuel consumption.

6.2.1. The output-coupled hydraulic hybrid driveline

A hydraulic output-coupled power split transmission has been sized to meet the performance specifications of the electric hybrid Prius. The goal was to come up with all sizes of hydraulic units and gear ratios to realize an efficient transmission that meets the performance of the electrical transmission.

The schematic of the transmission is shown in Figure 6.3.

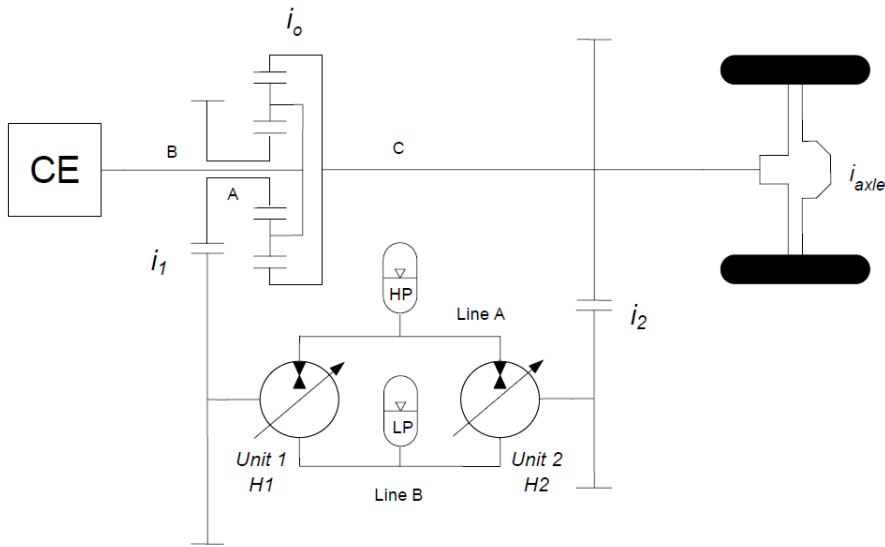


Figure 6.3: Toyota Prius transmission, hydraulic version.

To compare the two transmissions, the same engine was retained for the sizing of the hydraulic transmission. The specifications of the performance and the variables of the system are listed in Table 6.1; the other vehicle parameters are assumed equal to those of the electric hybrid transmission.

Vehicle mass	M	1,311	kg
Radius of the tires	R_{tire}	0.305	m
Engine Power	P_{CE}	57	kW
Engine Rotational speed	n_{CE}	4500	rpm
Wheel torque	M_{wheel}	1,729.8	N·m
Maximum vehicle speed	V_{max}	180	km/h
Maximum engine torque	M_{CE}	121	N·m
Maximum differential pressure	Δp_{max}	400	bar

Table 6.1: Transmission parameters.

The design procedure of the output-coupled transmission is shown below.

The relation of the maximum speed of the axle is given by equation (6.1):

$$n_{axle} = \frac{v_{max}}{3.6 R_{tire}} \frac{60}{2\pi} = 1,565 \text{ rpm.} \quad (6.1)$$

Assuming that the maximum speed of the vehicle is equal to the speed of the full mechanical point, the speed of the ring gear at the full mechanical point, $n_{C, fmp}$ is given by

$$n_{C, fmp} = n_{axle} \cdot i_{axle} \quad (6.2)$$

The standing gear ratio of the transmission is $i_0 = -2.6$. The speed $n_{C, fmp}$ could be calculated using the relation for the planetary gear train (6.3).

$$n_{C, fmp} = \frac{-(1 - i_0) \cdot n_B}{i_0} = \frac{-(1 - i_0) \cdot n_{CE}}{i_0} = 6,231 \text{ rpm} \quad (6.3)$$

Merging the two relations (6.2) and (6.3), the resulting axle gear ratio is $i_{axle} = 3.98$. The maximum torque is known to be the maximum engine torque M_{CE} . The maximum engine torque of the sun and the ring are therefore expressed by relations (6.4) and (6.5).

$$M_A = \frac{M_B}{(i_0 - 1)} = -33.6 \text{ N} \cdot \text{m} \quad (6.4)$$

$$M_C = -i_0 \cdot M_A = -87.4 \text{ N} \cdot \text{m} \quad (6.5)$$

The maximum axle torque is equal to the maximum wheel torque: $M_{axle} = M_{wheel} = 1,729.8 \text{ N} \cdot \text{m}$. Furthermore, it may be given as the sum of two contributions—the torque of the ring of the PGT ($M_{axle, mech}$) and the variable hydraulic branch ($M_{axle, U2}$).

$$|M_{axle, max}| = |M_{axle, mech}| + |M_{axle, U2}| \quad (6.6)$$

If no mechanical losses are considered, the $M_{axle, mech}$ is given by the following:

$$M_{axle, mech} = |M_C| i_{axle} = 349.6 \text{ N} \cdot \text{m}. \quad (6.7)$$

The torque generated by the hydraulic path is given by:

$$M_{axle, U2} = |M_{axle, max}| - |M_{axle, mech}| = M_2 \cdot i_2 \cdot i_{axle} = 1,380.2 \text{ N} \cdot \text{m} \quad (6.8)$$

The torque provided by the variable displacement 2 can be calculated from its maximum volume V_2 , the maximum differential pressure Δp_{max} , and the hydromechanical efficiency η_{hm2} :

$$M_2 = \frac{V_2 \cdot \Delta p_{max} \cdot \eta_{hm2}}{20\pi}. \quad (6.9)$$

A 55 cc/rev bent-axis hydraulic unit (Sauer–Danfoss Series 90) was considered, owing to its ability to deliver the maximum torque of 349.8 N·m and to reach speeds of over 8,000 rpm at zero displacement. The gear ratio i_2 is calculated via the following relation (6.10).

$$i_2 = \frac{M_{axle,U2}}{M_2 \cdot i_{axle}} = 1 \quad (6.10)$$

The required maximum speed of Unit 2 is calculated via (6.11), and it is in agreement with the accepted speed range.

$$n_{2,max} = n_c \cdot i_2 = 6,231 \text{ rpm} < 8,000 \text{ rpm} \quad (6.11)$$

The maximum speed of the sun gear is reached when the vehicle is at a standstill. However, in most situations when the accumulator is being filled by the engine at standstill, the engine speed has to be limited. According to this, the maximum engine speed is lowered until $n_{CE,ss} = 2,500 \text{ rpm}$; therefore, the maximum speed of the sun gear is calculated as in the following:

$$n_{A,max} = (1 - i_0) \cdot n_{CE,ss} = 9,000 \text{ rpm}. \quad (6.12)$$

The gear ratio i_1 is selected so that unit 1 does not overspeed when the sun gear is running at $n_{A,max}$. A 28 cc/rev bent-axis unit (Rexroth AA6VM) was chosen because it is capable of delivering the maximum torque of 179 N·m under the maximum differential pressure of 400 bar, and has a maximum speed of 5,550 rpm at the maximum displacement.

The gear ratio of Unit 1 is:

$$i_1 = \frac{n_{A,max}}{n_{2,max}} = 1.62. \quad (6.13)$$

Unit 1 must be capable of handling the maximum torque of the sun gear M_A . Assuming a hydromechanical efficiency of $\eta_{hm1} = 0.92$ and a nominal differential pressure of $\Delta p_{nom} = 200 \text{ bar}$, the minimum required displacement is:

$$V_{1,min} = \frac{M_A \cdot i_1 \cdot \eta_{hm1} \cdot 20\pi}{\Delta p_{nom}} = 16 \frac{\text{cc}}{\text{rev}}. \quad (6.14)$$

Therefore, the selected unit is in agreement with the torque requirement. Additionally, hydraulic unit 1 must supply flow to unit 2 for all operating conditions. In the worst case, the hybrid transmission has to work as simply hydrostatic, and the maximum flow requirement occurs at the maximum speed, when the displacement of the Unit 2 is small. For the Unit 2 previously chosen, the maximum flow rate is 91.7 l/min. The maximum speed of the Unit 1 at its full displacement is 5500 rpm. To determine whether U1 is capable of meeting the maximum flow requirement imposed by the Unit 2, its displacement must be at least equal to $V_{1,min}$, calculated from the relation (6.15).

$$V_{1,min} = \frac{Q_{max}}{n_{1,max} \cdot \eta_{v1} \cdot \eta_{v2}} = 21 \text{ cc/rev} \quad (6.15)$$

Here, η_{v1} and η_{v2} are 0.95 and 0.83, respectively. The displacement of Unit 1 is also in agreement with the relation (6.15).

The design of the hydraulic hybrid transmission does not include the sizing of the accumulators because the research conducted in the laboratory was confined to only the hydrostatic version of the transmission.

Table 6.2 summarizes the size of the components in the transmission.

Displacement of the Unit 1	V_1	28 cc/rev
Displacement of the Unit 2	V_2	55 cc/rev
Axle gear ratio	i_{axle}	4
Gear ratio related to the Unit 1	i_1	1.62
Gear ratio related to the Unit 2	i_2	1
Planetary gear train ratio	i_0	-2.6
Maximum system pressure	Δp_{max}	400 bar
Minimum system pressure	Δp_{min}	200 bar

Table 6.2: Size of the transmission components.

6.3. The Toyota Prius hydraulic test rig

For the testing of the previously designed output-coupled transmission, a hardware-in-the-loop test rig was developed. The hydraulic hybrid transmission was connected to an Engine Simulator and a Load Simulator. The Engine Simulator represents the behavior of the internal combustion engine, whereas the Load Simulator should replicate the behavior of a real vehicle, incorporating all the vehicle properties, such as the air drag, the mass of the vehicle, and the rolling resistance. The Load Simulator represents the speed response of the vehicle when a certain acceleration and deceleration torque is transmitted. The configuration of the transmission test rig is shown in Figure 6.4.



Figure 6.4: Hydraulic hybrid Toyota Prius Test Rig.

6.3.1. Engine and Load simulators

Both the Engine and Load Simulators (Figure 6.5) are 180 cc secondary-controlled axial-piston hydraulic units (Rexroth A4VSO-180), able to operate in either speed or torque control mode. They can simulate the engine and vehicle characteristics, such as a strong acceleration and deceleration, owing to their fast dynamic response. In such operations, an overspeed of the units could be easily generated; to avoid such event, both units were operated in speed control mode.

The engine simulator controller receives an analog signal correlated to the target speed of the engine. Instead, the Load Simulator receives a speed command, based

on the mission that the load has to follow. According to this, the Engine Simulator always operates as a motor, whereas the load simulator operates as a pump during the propulsion. The direction of applied torque changes, when the vehicle brakes are applied; Unit 2 may operate over center, thereby filling the accumulator. The speed and the torque of the two hydraulic units were measured in real time using the torque meters placed along the same shaft.

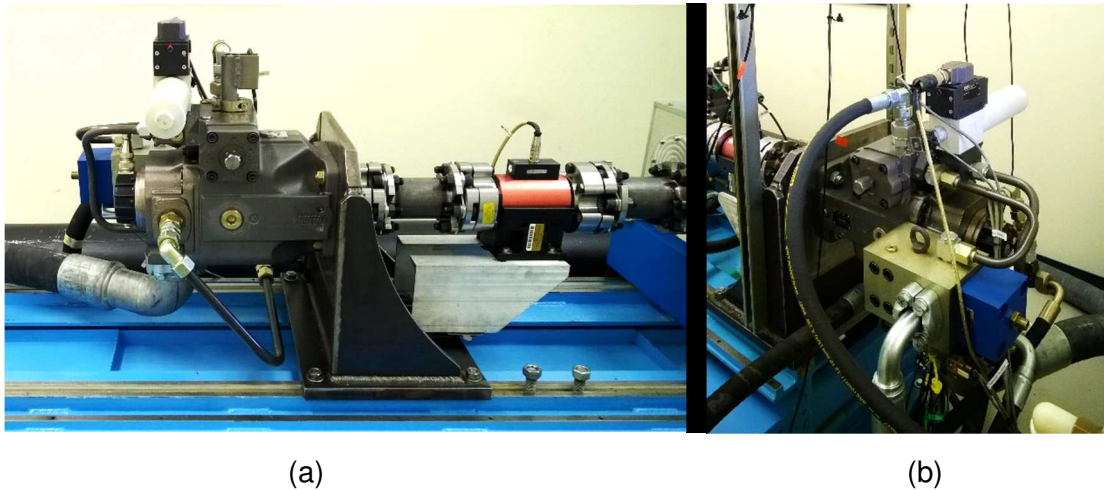


Figure 6.5: a) Engine Simulator, b) Load Simulator.

6.3.2. Hydraulic Unit 1 and Unit 2

Hydraulic Unit 1 is a 28 cc axial-piston unit (Rexroth AA6VM) and it operates as a pump, except when the transmission functions in recirculating mode. The main characteristics of Unit 1 are listed in Table 6.3. Hydraulic Unit 2 is a 55 cc axial-piston unit (Sauer–Danfoss Series 90) and it is capable of functioning over center, operating in motoring as well as in pumping mode. The main characteristics of Unit 2 are listed in Table 6.4. Following the original schematic, the two hydraulic units were connected to the planetary gear train. Both the units are shown in Figure 6.6.

6.3.3. The hydraulic power supply

The Engine and the Load Simulator need an operating fluid power source. Therefore, the power supply for the test rig is delivered by two lines, one for the high pressure and one for the low pressure. The fluid under pressure may be delivered by one or more than five pumps, each capable of delivering 100 l/min, with a maximum flow rate of 500 l/min. The power supply can reach the maximum pressure of 350 bar; therefore, the high-pressure line of the test rig operates under a maximum pressure of 315 bar, whereas the low-pressure line operates under a minimum pressure of 17 bar.

The power supply is shown in the Figure 6.7.

Rexroth mod.AA6VM	
Maximum displacement (Vg)	28 cc
Maximum speed	5,550 rpm (Vg,max) 10,450 rpm (Vg0)
Torque	179 N·m (Vgmax)
Pressure	450 bar (max) 400 bar (nom) 18 bar (min)

Table 6.3: Main characteristics of Unit 1.

Sauer-Danfoss Series 90	
Maximum displacement	55 cc
Rotational speed	4,250 rpm (max) 500 rpm (min)
Torque	350 N·m (Vg,max)
Pressure	450 bar (max) 18 bar (min)

Table 6.4: Main characteristics of Unit 2.

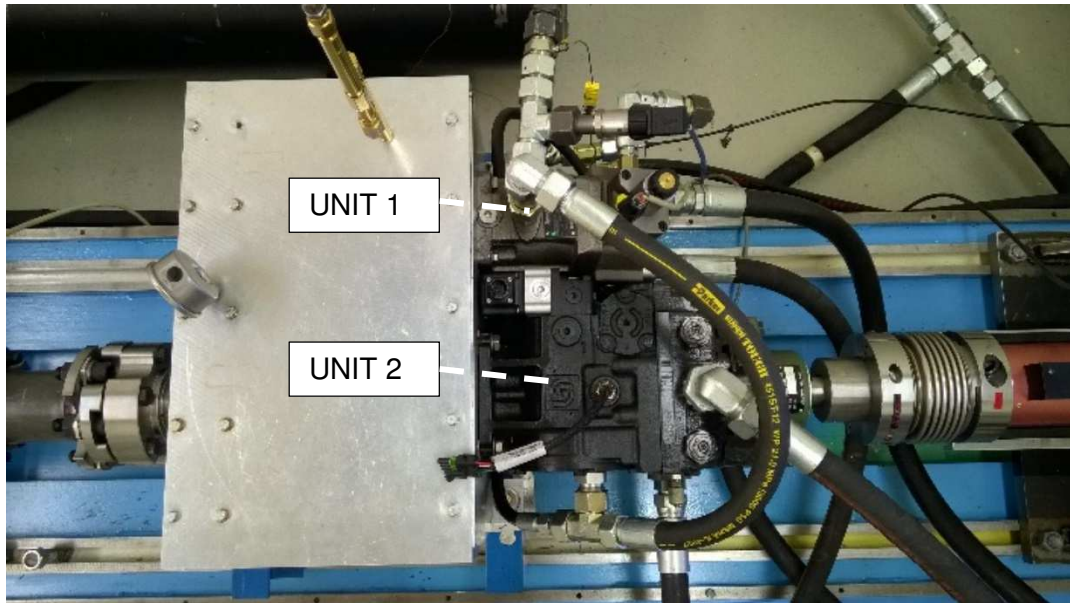


Figure 6.6: Unit 1 and Unit 2 connected to the planetary gear train.

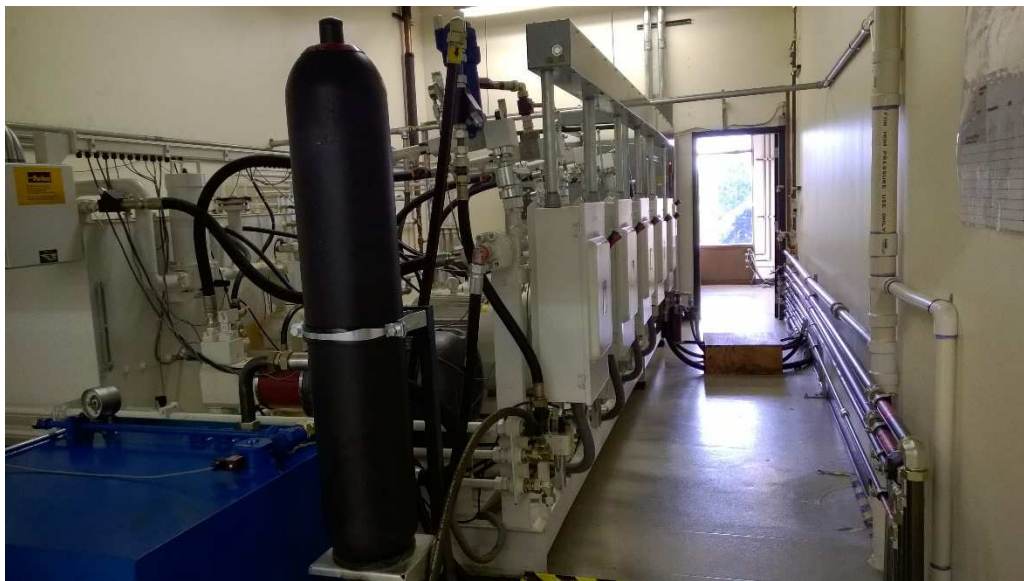
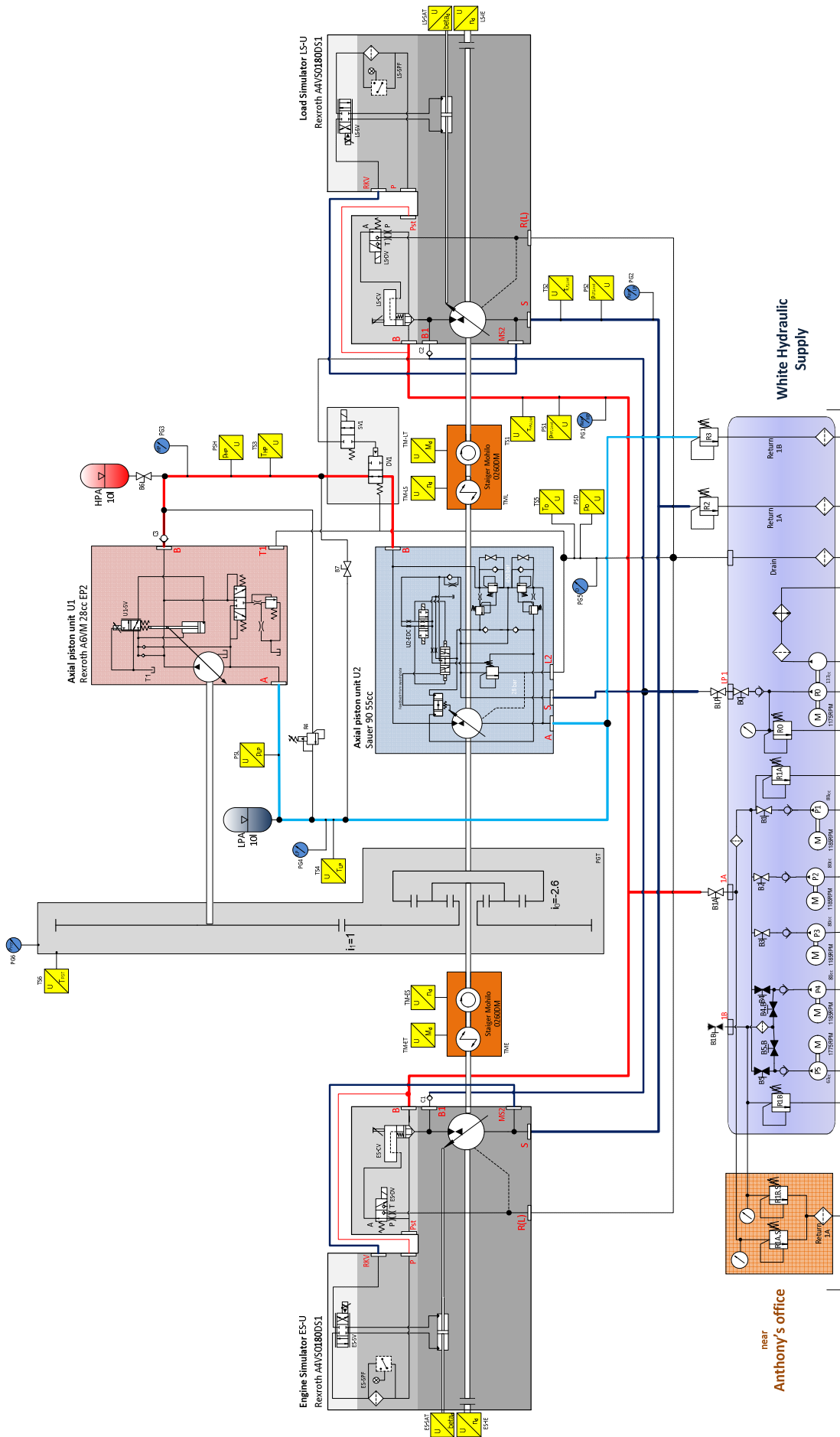


Figure 6.7: View of the hydraulic supply room.

6.4. Test rig new schematic

The new schematic of the test rig is proposed in Figure 6.8. It differs from the old version owing to certain additional details and general improvement. In fact, after the first analysis of the circuit, certain issues arose.

Figure 6.8: Test rig schematic



6.4.1. The Engine and Load Simulator

To allow the calibration of the mechanical sensor of the fractional displacement for the Engine and the Load simulator, certain changes in the original schematic were required. In Figure 6.8, the old and the new version of the circuit are shown.

The major difference between the two schematics is the connection via the lines. In fact, for the first calibration of the fractional displacement, the old schematic, as is, could lead the unit into overspeeding. In both versions, when the low-pressure line was open, the hydraulic unit would not deliver flow because the differential pressure between the two ports was $\Delta p = 0$.

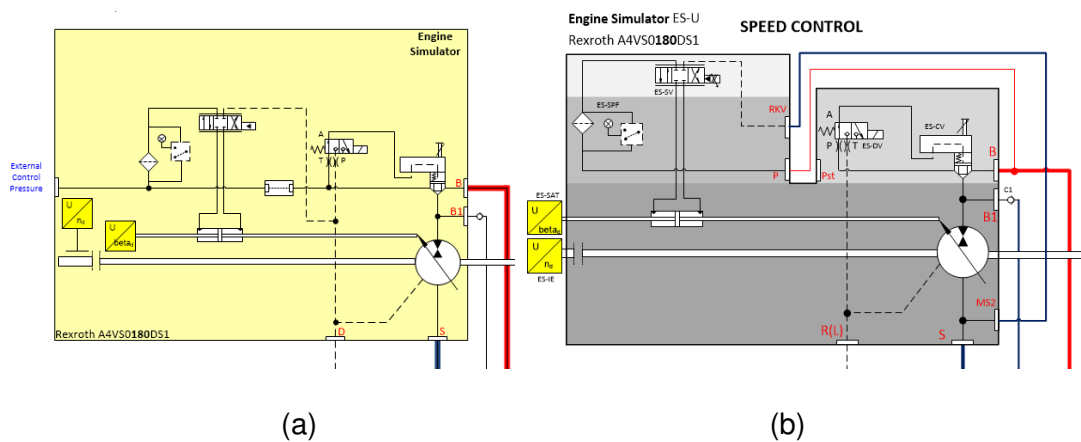


Figure 6.8: Old version of the Engine Simulator schematic (a), and the new version (b).

In the old version, Figure 6.8(a), even when the high-pressure line was open, the piloted check valve allowed the high-pressure fluid to reach the servo valve. The four-way servo valve connects and compares two pressures, namely the high pressure and the pressure of the drain line. The calibration of the circuit was therefore possible, however, the problem lies in the first instant of operation because when the piloted check valve was open and the unit would sense the pressure difference ($\Delta p \geq 0$), and it would immediately start to rotate. If the correct position of the mechanical sensor of the fractional displacement is unknown, and considering the worst case (displacement angle $\beta = \beta_{\max}$ and no torque applied), the unit may overspeed.

In the newest version of the circuit, a high-pressure line was added near port B, which links port B with port P, bypassing the piloted check valve. Moreover, the servo valve is no longer linked to the drain line; it is directly linked to the low-pressure line.

Therefore, when the high-pressure line opens, the unit does not sense the pressure difference (because the check valve is still closed); hence, it does not speed up. However, it is possible to calibrate the displacement angle β using electronic devices.

6.4.2. Unit 2

Hydraulic Unit 2 has been redrawn with its hydraulic circuit, merely for improved clarity. The old and the new schematics of Unit 2 are illustrated in Figure 6.9.

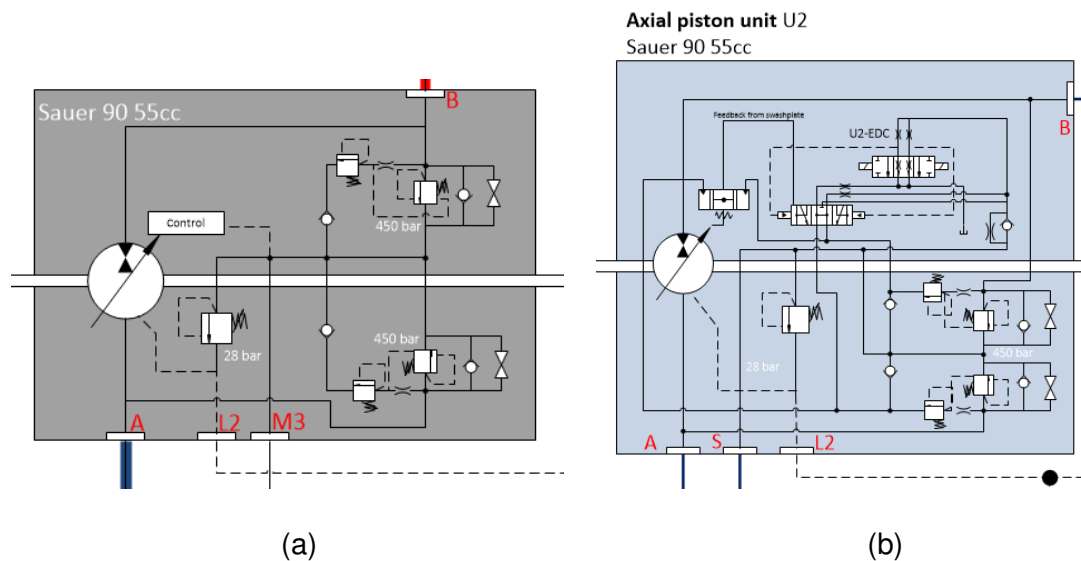


Figure 6.9: Old Unit 2 schematic (a), and new Unit 2 schematic (b).

6.4.3. The isolating valve and the bleeding valve

The purpose of the isolating valve in the hydraulic circuit is to close the line between Unit 1 and Unit 2. However, in the old schematic, the position of the valve (normally closed) is dangerous if an electrical problem occurs. For instance, a voltage drop in the electrical circuit may occur when the transmission is in operation, and may accidentally close the isolating valve.

The old version of the valve is shown in Figure 6.10(a) and the new version of the valve is shown in (b).

The bleeding valve, which connects the high-pressure line with the low-pressure line, is presented in Figure 6.11(a). However, the valve is not present in the test rig, replaced with a manual on-off valve (Figure 6.12). The new schematic is, thus, adjusted with the right valve (Figure 6.11 (b)).

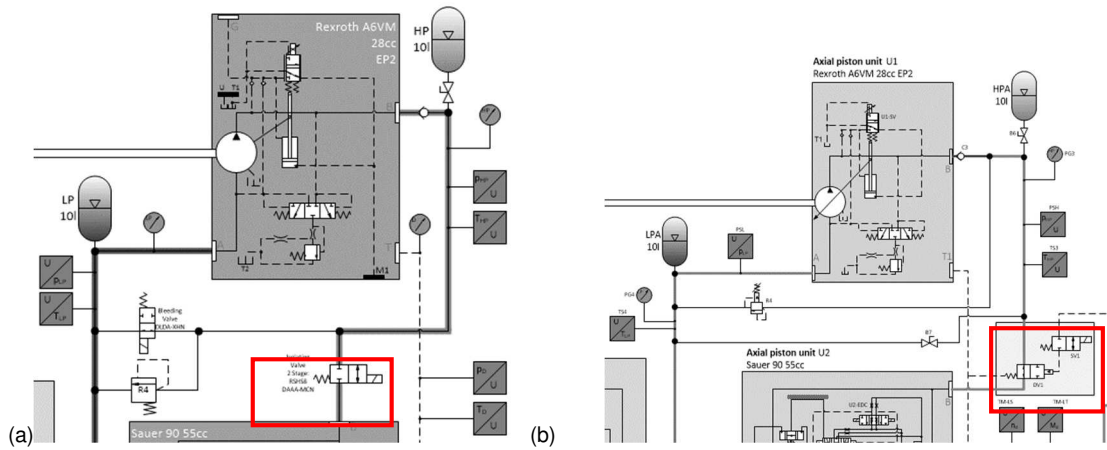


Figure 6.10: The difference between the old version of the isolating valve (a), and the new version (b).

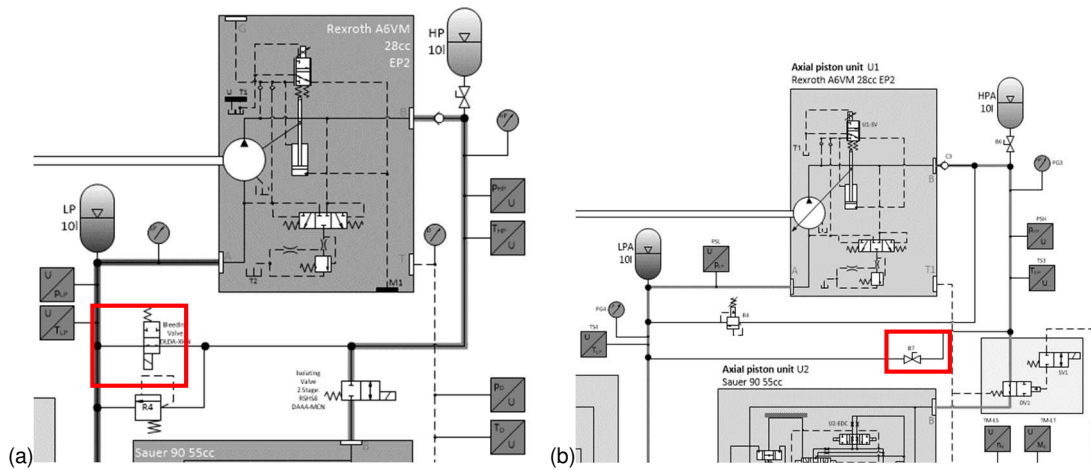


Figure 6.11: The difference between the old version of the bleeding valve (a) replaced with an on-off manual valve,

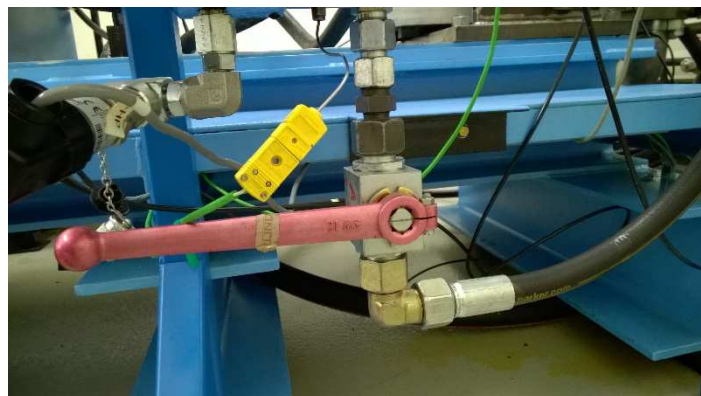


Figure 6.12: The on-off manual valve.

6.4.4. Check valves

Regarding the Engine and the Load Simulator, the check valves C1 and C2 do not allow the fluid to return to the low-pressure line of the hydraulic power supply. However, in the test rig on the Engine simulator side, there are two check valves in series, installed along the same line (LP1). Although this double presence of the check valves does not cause significant issues, it was decided that one of them be removed.

In Figure 6.13, the check valves in the same line are shown: we kept the first one at the same location (a) and removed the other (b).



Figure 6.13: Check valves in the same low-pressure line.

6.4.5. Pressure and temperature sensors

New pressure and temperature sensors were added on each line to gain a full overview of the circuit when in operation. In particular, in the high pressure-line we added pressure sensors PS1 and PS2, the TS6 thermocouple, and the PG6 pressure gauge.

6.5. Engine and Load Simulator test

For the transmission to be fully functional, the first objective is to test all the components in safe mode. In this section the project of the first test is explained, which comprises the test of the Engine and the Load simulator, because they required a very similar procedure. The other tests, for instance the Unit 1 and Unit 2 test, haven't been included in this section, because they still need the final approval.

The test of the Engine and Load Simulator test is divided into certain sections: first, the schematic and the calculation of the test will be explained. Then the procedure of the test will be described systematically, and finally, the calibration of the hydraulic machine and the check of the rotational direction of the shaft will be addressed.

6.5.1. Schematic and calculations

The test schematic for the Engine Simulator in Figure 6.14 comprises the unit, linked to three hydraulic lines, namely the high-pressure line (1A), the low-pressure line (LP1), and the anticavitation line (from port LP1 to port B1). Furthermore, the fluid was delivered to the tank through the drain line (port R(L)) and the Return 1A after the relief valve.

To measure the rotational speed and the fractional displacement, the Unit was equipped with an incremental encoder (speed sensor ES-IE) and a swivel angle transducer (displacement sensor ES-SAT).

In this specific test, the Engine Simulator was not secondary-controlled, and no load was applied on the unit. Thus, the only forces that the unit has to overcome are the friction losses.

The maximum displacement of the Engine Simulator is 180 cc, and the pressure of the high-pressure line is assumed 50 bar, while the pressure of the low-pressure line is assumed 20 bar. Thus, the differential pressure between the two lines is $\Delta p = 30 \text{ bar}$.

The flow is delivered to the Engine Simulator through one of the fixed-displacement pumps from the hydraulic power supply, P1 (80 cc and 1,185 rpm of fixed rotational speed).

The flow that P1 can deliver is calculated using the following relation:

$$q_{P1} = V_{g,P1} n_{P1} \eta_v = \frac{80 * 1185 * 0.95}{1000} = 90 \text{ l/min} \quad (6.16)$$

The maximum rotational speed of the Engine Simulator, considering a volumetric efficiency of 0.98, is calculated via (6.17).

$$n_{max,ES} = \frac{q_{P1} * \eta_v}{V_{g,max,ES}} = \frac{90 * 1000 * 0.98}{180cc} = 510 \text{ rpm} \quad (6.17)$$

The maximum torque that the Engine Simulator can achieve is given by (6.18).

$$T_{ES} = \frac{V_{g,max,ES} * \Delta p}{20\pi} = \frac{180 \text{ cc} * 30 \text{ bar}}{20\pi} = 84 \text{ Nm} \quad (6.18)$$

6.5.2. Modifications of the original circuit

The original test rig comprised the Engine and the Load simulators assembled on the same shaft of hydraulic Unit 2. To avoid the difficult process of realigning the shafts when the components would be disassembled, we planned to perform the lowest number of mechanical operations.

Therefore, the transmission is disconnected upstream the torque meter (TM-E), as shown in Figure 6.14, together with the hydraulic line, which linked the PST port and the port.

A new T connection has to be developed which connect the High pressure (port B side) and the P port.

In addition, it was essential to disconnect and close:

- the high-pressure line of the Load Simulator side,
- the low-pressure line of the Load Simulator side and the Unit 2 side (port S), and
- the drain and the anticavitation line of the Load simulator side.

6.5.3. The test procedure

In this section, the procedure of the test will be explained. The names of the components are included in Figure 6.14.

Before starting the test, the ball valves B1B, B2, B3, B4, B4.B, B5.B, and B5 had to be closed, and to remain closed for the entire duration of the test.

- a) To calibrate the relief valve of pump P1 (R1A), ball valves B1A and BLP were temporarily closed, and valves B1 and B0 were opened.
- b) Pump P1 was set in operation, and the relief valve R1A was set to 70 bar, which was higher than the maximum pressure assumed in the previous section. Then, the relief valve R1A,S was set to 52 bar (high pressure incremented with 2 bar, considering the losses in the hydraulic lines).
- c) Pump P0 was set in operation and the relief valve R0 was set to 30 bar, slightly higher than the low pressure assumed in section 6.5.1.
- d) Ball valve BLP was opened and the relief valve R2 was set to 22 bar. This was the low pressure previously assumed, increased by 2 bar, considering the losses in the hydraulic lines.
- e) Checking of the pressure (and temperature) on the LP line.

- f) Opened the high-pressure line using ball valve B1A (ensuring that the piloted check valve ES-CV (or LS-CV) is closed, and that there is no current).
- g) Checking of the pressure (and temperature) on the HP line.
- h) Calibration of the Engine simulator displacement angle β (from 0 to 100%).
- i) Checking of the rotational direction of the shaft.

6.5.4. The calibration of the linear variable differential transformer (LVDT)

When the Unit was connected to the high-pressure line, and the check valve was still closed, the calibration of the position transducer (LVDT) and the servo valve could begin.

The Engine Simulator swash plate angle was normally centered. The first voltage reading (it should be zero) on the position transducer is the center position signal, and can be visually checked from the pump (Figure 6.15).



Figure 6.15: A view of the mechanical displacement feedback of the Engine Simulator.

Then, by the PC-controller, a positive step was fed as input to the servo valve. Observing through the Engine simulator LVDT, when the value of the current would stabilize, that would correspond to the maximum displacement value. At that point, the relationship between the voltage and the fractional displacement β was documented.

The same procedure was repeated with a negative step input.

We obtained the following:

- The calibrated range of the LVDT.

- The relationship between the input signal and the fractional displacement β .

6.5.5. The rotational direction of the shaft

After calibrating the displacement sensor, the rotational speed direction of the shaft should be checked.

According to the Engine Simulator datasheet, the unit may operate as a motor in two cases, as shown in Figure 6.16.

Flow direction				
Swivel range ³⁾	Rotation direction ⁴⁾		Pressure in	Operating mode
	Clockwise	Anti-clock-wise		
Clockwise	$B \Rightarrow S$	-	B	Motor
Clockwise		$S \Rightarrow B$	B	Pump
Anti-clock	-	$B \Rightarrow S$	B	Motor
Anti-clock	$S \Rightarrow B$	-	B	Pump

³⁾ Compared to the swivel angle indicator
⁴⁾ Viewed on the shaft

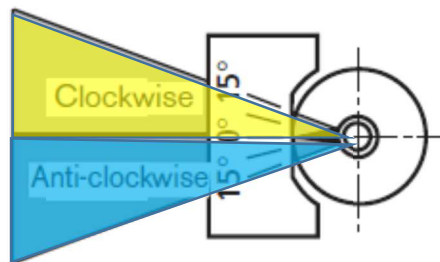


Figure 6.16: Flow direction of the Engine Simulator.

To remain in agreement with the kinematics of the transmission, the Engine Simulator should to perform a clockwise rotation of the shaft, in such a manner that the displacement feedback in Figure 6.16 should be within the yellow area.

Based on the description provided in the present section, the technical documentation that would allow starting the tests has been developed.

7. CONCLUSIONS

In this thesis, two design procedures for the hydromechanical transmission have been proposed, in order to find the optimum average efficiency of the system as a function of the design parameters, the configurations, and the layouts.

In the first work, the design of the hydromechanical transmission was approached as an optimization problem, the goal of which was the simultaneous optimization of the layout and the components. This dual level of optimization constitutes a strong element of novelty because in traditional design, the layout is assumed a priori based on prior knowledge; the size of the components, instead, is calculated based on the design data. In this way, the result of the design cannot be considered optimal.

The problem of the mathematical representation of the configuration was treated by employing graph theory; the representation of the transmission was assigned to a simulator. In the models of hydraulic machines, appropriate efficiency functions were included that were derived from experimental data provided by the manufacturers. The resolution of the optimization problem was accomplished through a "direct search" algorithm based on the particle swarm method.

The procedure to obtain an optimized design was applied to the case of the transmission of a high-power agricultural tractor. The optimum configuration was the Input Coupled, with the carrier being connected to the wheels, and the ring connected to the hydrostatic group; its efficiency is slightly higher than the standards proposed in the literature.

In the second work, a different optimization approach was proposed. In the previous procedure in fact, the global efficiency was a function of the layout and the planetary gear train ratio. However, the efficiency changes by changing another important design variable: the speed of the full mechanical point, which was considered in the procedure in question.

The global efficiency was evaluated by employing the "manual search" optimization algorithm, which calculates the efficiency for all the combinations of the design variable. The result will be an efficiency matrix as function of the design variables. The solution of the design equations were implemented in MATLAB, whereas the solution of the transmission was assigned to the simulation model.

The procedure was applied to a city bus, pointing out that the best efficiency point was realized for an fmp value of 0.3 value, and with a layout in which the engine is connected to the sun gear of the planetary gear train, the hydrostatic transmission is

connected to the ring gear, and the wheels were connected to the carrier, in agreement with the previous study.

An important aspect to consider in the design of the hydromechanical transmission is the noise emitted by the main system components, namely the engine and the hydrostatic transmission. In the third work, the noise emitted from a dual-stage hydromechanical transmission of a bus was studied using a simulation model. The aim was to assess the relative contribution of the noise produced by the Dual Stage out of the total noise generated by the engine transmission group.

The engine noise was described according to an empirical model in the existing literature; the noise model of the hydromechanical transmission was inferred by means of a linear regression technique applied on experimental data provided by a manufacturer. The output model values may be slightly overestimated because the new-generation engines are modern and more acoustically isolated.

The simulation results indicated that the hydraulic pressure is the most important parameter affecting the sound emission of the transmission.

The pressure also influences the gearshift phase, where a sensible increment of noise can be observed. During the full mechanical point, a strong decrease of pressure sound level occurs, because of the cancelling of the displacement value of the first hydraulic unit.

Regarding the gas engine, the main contribution of the emission is the housing, followed by the exhaust system; less important is the effect owing to the air-intake system.

Although the transmission is equipped with two hydraulic machines, it only slightly contributes to the total noise generated by the engine transmission group.

Regarding the advanced design methods explained in the first section of the thesis, certain future steps could be pointed out. First, the method should be extended to complex hydromechanical transmissions, such as the dual-stage or the compound, because certain heavy vehicles that reach medium-high speeds (i.e., buses and certain trucks) are equipped with these systems.

Another step ahead would be to consider the optimized efficiency as one of the objective functions within the same problem. A multiobjective optimization would be, thus, realized, which would compare the efficiency to other global parameters, such as the total bulk of the transmission, rather than the noise and the emissions.

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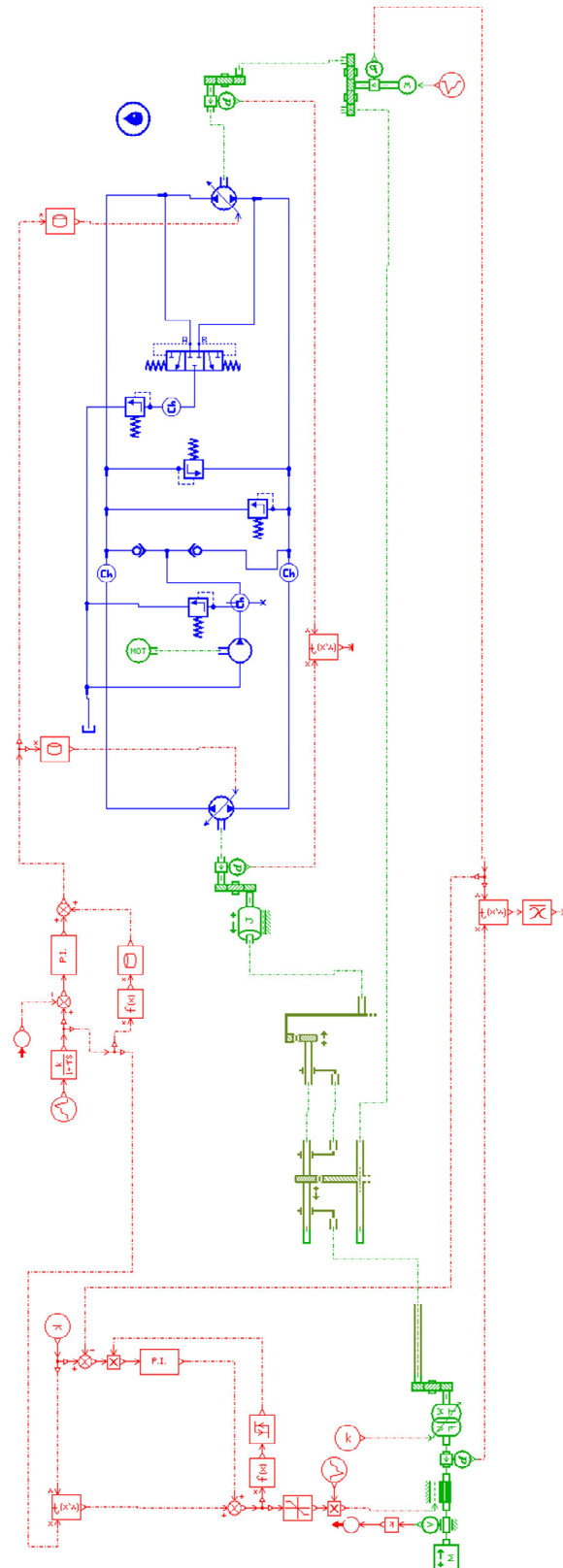
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APPENDIX

Amesim simulation model of HMT (Layout VI)



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