COMPARATIVE STUDY OF MULTIPLE MODE POWER SPLIT TRANSMISSIONS FOR WHEEL LOADERS

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ABSTRACT

To increase energy efficiency and lower emissions in construction machines, the use of hydromechanical power split drive trains shows high potential. The possibility of using multiple gear speeds without losing traction force makes the power split architecture especially suitable for heavier wheel loaders. This paper analyses two known concepts of multi-mode power split transmissions suitable for the wheel loader application and compares the solutions based on energy efficiency. The concepts are scalable in the sense that additional modes can be used without necessarily adding complexity to the transmission. Simulations are made with respect to steady-state transmission losses and the relation between number of modes and transmission efficiency is shown for each of the proposed concepts. The operational characteristics of the hydraulic displacement machines strongly affects the transmission efficiency and the design choice of number of transmission modes.

KEYWORDS: Power split, energy efficiency, transmission, wheel loader

NOMENCLATURE

Quantity	Description	Unit
a	Loss Model Parameter	-
b	Loss Model Parameter	-
C_m	Speed Constant	m rev$^{2/3}$/s
D	Displacement	cm^3/rev
F_{max}	Maximum Traction Force	N
f_{vmech}	Mode Shift Speed Ratio	-
i	Gear Ratio	-
i_s	Mode Gear Ratio	-
m	Number of Modes	-
n	Rotational Speed	rpm

n_{ICE}	Engine Speed	rpm
n_{carr}	Carrier Speed	rpm
n_{ring}	Ring Speed	rpm
n_{sun}	Sun Speed	rpm
n_0	Nominal Speed	rpm
p_{boost}	Boost Pressure	bar
p_H	High Pressure	bar
p_L	Low Pressure	bar
P_{ICE}	Engine Maximum Power	kW
P_{max}	Maximum Load Power	kW
P_{wheel}	Wheel Power	kW
Q_m	Motor Flow Loss	l/min
Q_p	Pump Flow Loss	l/min
R	Planetary Gear Ratio	-
T_m	Motor Torque Loss	Nm
T_p	Pump Torque Loss	Nm
v_{mech}	Speed at Full Mechanical Point	km/h
v_{shift}	Speed at Mode Shift	km/h
β_e	Bulk Modulus	bar
δ	Loss Model Parameter	-
Δp	Pressure Difference	bar
η_{hm}	Hydromechanical Efficiency	-
η_{vol}	Volumetric Efficiency	-
η_{trans}	Transmission Efficiency	-
ε	Swivel Angle	-
γ	Loss Model Parameter	-
μ	Dynamic Viscosity	Ns/m^2

1 INTRODUCTION

Interest in hydromechanical power-split (PS) drives for construction machines has grown in recent years. Numerous academic publications have shown their potential for reducing energy losses and increasing control flexibility, e.g. [1], [2], [3] and [4]. Industry have been active, displaying a rapidly increasing number of patents concerning transmission architecture, components and control strategies. PS transmissions have been used in agricultural tractor transmissions for some time [5] and today are more or less state-of-the-art technology [6]. Recently, some PS concepts have also been commercially introduced for construction machinery and wheel loaders in particular, e.g. [7] and [8]. A pure hydrostatic transmission (HST) is not rarely seen in compact wheel loader drive trains where relatively small hydraulic machines are enough to cover the torque and speed demands. Such transmissions are dimensioned for the corner power of the traction force requirements and are thus unnecessarily large. However, hydraulic machines used in pure hydrostatic drives lack the operating range, in terms of maximum speed and pressure, to meet the demands of heavier wheel loaders. A conventional solution is to use a torque converter in series with a powershift gearbox to ensure sustained traction force between gear shifts. The price of the powershift gearbox and the high power losses of the torque converter will, however, make this option unattractive in modern drive trains. A more advanced system design is thus desirable with a wide range continuously variable speed

ratio and reasonably sized hydraulic units [9]. One important feature of the PS architecture is the possibility of using coupling elements to engage new gears without losing traction force. The HST can consequently work within its operating range and a wider speed ratio can be achieved with multiple modes. Three basic PS configurations can be distinguished: input coupled (IC), output coupled (OC) and variable bridge (VB) [10]. Different combinations of these configurations can then be assembled into one multiple mode concept. One common principle of designing the PS architecture is to assemble the planetary gears resulting in several output shafts from the PS part and alternating the coupling to the transmission output between them. This principle requires the total speed ratio to be increased with increased variator speed ratio in one mode and with decreased variator speed ratio in the next [10]. This configuration offers a very scalable PS transmission where additional modes can be used without drastically adding further complexity to the system. A greater number of modes enables less power flow through the HST, smaller hydraulic machines and higher overall efficiency.

The aim of this study is to give an overview of the benifits of using multiple modes in PS transmission architectures. Two concept architectures are considered: multiple input coupled modes (MIC) with a pure hydrostatic start mode and multiple variable bridge modes (MVB) with an OC start mode. Each represents two opposite levels of complexity of a wide range of multi-mode PS architectures that fulfils the transmission requirements. Both concepts are scalable in the sense described above and could be suitable for a variety of applications. The concepts are simulated with varying number of modes and evaluated with the focus on energy efficiency. The reference vehicle is a wheel loader with an operational weight of 31 tonnes, a maximum speed of 50 km/h and the principle load characteristics according to figure 1. The engine speed is considered to be constant at 1500 rpm which would be a favorable speed when it comes to fuel consumption for the diesel engine. The transmission alone must consequently meet the necessary speed ratio to achieve maximum vehicle speed. This is a reasonable requirement since it would allow independent power management of the combustion engine. The presented reference vehicle is today typically equipped with torque converter and powershift gearbox and is a suitable example of a machine size where a more advanced transmission design is necessary to fulfil the requirements.

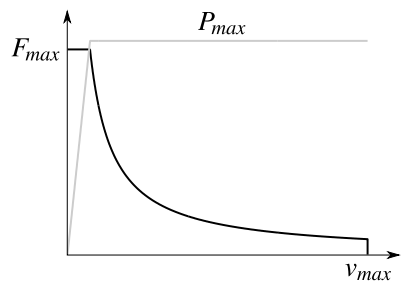

Figure 1. Reference vehicle load characteristics

2 MULTIPLE INPUT COUPLED MODES

The concept of alternating between input coupled modes is commonly seen in tractor transmissions, see [5]. The considered layout follows the principle patented by Jarchow [11] and includes a pure hydrostatic first mode and subsequent input coupled modes. Figure 2 shows the basic layout consisting of two planetary gears and an arbitrary number of modes. The unnamed gear ratios in the figure are considered to have $i = 1.0$ since no

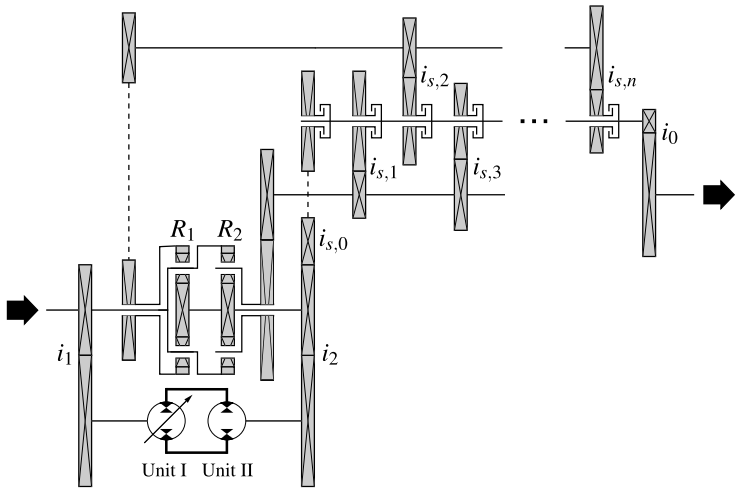

Figure 2. MIC transmission layout

extra design freedom is obtained by changing them. The hydrostatic first mode reduces the high control effort of the IC configuration during stall and allows switching between forward and reverse drive smoothly by controlling unit I over centre. This is particularly suitable for the wheel loader application due to the frequent reversing in load cycles. The second unit is a fixed displacement machine which makes the total speed ratio proportional to the HST speed ratio in every mode. The IC modes are described kinematically in figure 3. Each mode contains one power additive and one power recirculative operation

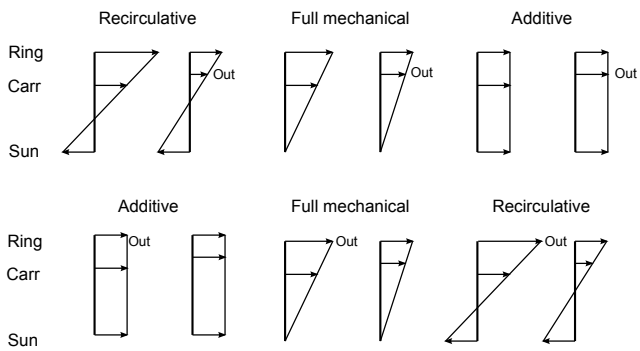

Figure 3. Kinematics of the planetary gears in the MIC concept

range. At the full mechanical points, the transmission efficiency reaches its peak since no power is fed through the HST. The possibility of positioning the full mechanical points in the vehicle's speed range is an important concept design problem. On the one hand, it is desirable to have high transmission efficiency throughout the entire speed range, which is fulfilled by an equal distance between each full mechanical point. On the other hand, many low speed mode shifts, where the highest load demands are, would require smaller

hydraulic units. Yet another option is to optimise the energy efficiency specifically to the working points for certain operating cycles such as the wheel loader's short loading cycle. However, this is problematic since it is also desirable to avoid too many mode shifts within the speed range of the loading cycle. In this study, each full mechanical point is placed with a constant relation to the one above. Equation (1) shows how the fraction is assumed to vary with the number of modes to obtain a reasonable distribution while avoiding many low speed mode shifts.

$$f_{vmech} = \frac{v_{mech,i+1}}{v_{mech,i}} = \frac{v_{shift,i+1}}{v_{shift,i}} = 5 + 4(m^{-1} - 1) \tag{1}$$

This relation allows an independent comparison of transmission concepts with different numbers of modes. Table 2 shows the vehicle speeds at which the mode shifts occur: The

Table 2. Vehicle speed during modeshifts

m	Mode Shifts $[km/h]$						
2	16.7						
3	9.18	21.4					
4	6.25	12.5	25.0				
5	4.76	8.57	15.4	27.8			
6	3.89	6.48	10.8	18.0	30.0		
7	3.32	5.22	8.20	12.9	20.2	31.8	
8	2.93	4.39	6.58	9.88	14.8	22.2	33.3

standing planetary gear ratios R_1 and R_2 are given by equations (2) and (3):

$$R_1 = \frac{n_{sun1} - n_{carr1}}{n_{ring1} - n_{carr1}} = \frac{2}{1 - f_{vmech}} \tag{2}$$

$$R_2 = \frac{n_{sun2} - n_{carr2}}{n_{ring2} - n_{carr2}} = \frac{1 + f_{vmech}}{1 - f_{vmech}} \tag{3}$$

The gear ratios corresponding to each mode, are dimensioned according to equation (4) to assure gap free mode switching:

$$i_{s,i} = \begin{cases} \frac{n_{shift,i+1}}{n_{ICE}} \frac{1}{i_0} & \text{for i = odd} \\ \frac{n_{shift,i+1}}{n_{ICE}} \frac{1}{i_0\left(1 - \frac{2}{R_1}\right)} & \text{for i = even} \end{cases} \tag{4}$$

The gear ratios of the hydraulic units i_1 and i_2 are dimensioned according to equations (5) and (6) resulting in the hydraulic units running at maximum speed during each mode shift:

$$i_1 = \frac{n_{I,max}}{n_{ICE}} \tag{5}$$

$$i_2 = \frac{n_{II,max}}{n_{ICE}} \tag{6}$$

Taking into consideration the properties of different hydraulic machine sizes, the following conclusions can be drawn according to [12]: The maximum pressure can be considered constant independent of the machine size and the maximum speed varies according to equation (7):

$$n_{m,max} = C_m D^{-1/3} \tag{7}$$

The parameter C_m is constant for a geometrically uniform machine series and can be physically interpreted as the maximum relative speed between the movable parts in a machine. After studying modern transmission machines the following speed constants have been achieved:

- Variable in-line machine: $C_m = 2.7\, m\, rev^{2/3}/s$
- Fixed bent-axis machine: $C_m = 3.0\, m\, rev^{2/3}/s$
- Variable bent-axis machine: $C_m = 4.5\, m\, rev^{2/3}/s$

The variable bent-axis motor tolerates higher speeds at lower displacement while the in-line machine normally reaches lower maximum speeds. The maximum pressure is assumed to be $\Delta p = 400\, bar$ for all displacement sizes and machine designs. For the MIC concept unit I is assumed to be an over-centre in-line machine and unit II a fixed displacement bent-axis machine. The gear ratio corresponding to the hydrostatic mode is given by equation (8) to match the first IC mode:

$$i_{s,0} = i_{s,1} i_2 \frac{R_2 + 1}{R_2 - 1} \frac{n_{ICE}}{n_{II,max}} \tag{8}$$

The size of unit II is dimensioned to the maximum startup traction force:

$$D_{II} = \frac{20\pi F_{max} r_{tire}}{\Delta p_{max}} \frac{i_0 i_{s,0}}{i_2} \tag{9}$$

Unit I is sized to cover the flow demand of unit II:

$$D_I = \frac{n_{II,max}}{n_{I,max}} D_{II} = \left(\frac{C_{m,II}}{C_{m,I}}\right)^{3/2} D_{II} \tag{10}$$

3 MULTIPLE VARIABLE BRIDGE MODES

A more complex transmission architecture is the MVB concept, which uses two infinitely variable hydraulic units and three planetary gears. The starting mode is an OC mode to achieve good controllability under stall conditions. Two additional clutches are needed to complete the switch from the first to the second mode. Figure 4 shows the basic layout with an arbitrary number of variable bridge modes. As opposed to the MIC concept, switching modes occur at the full mechanical points as shown in figure 5. Each mode is power additive which results in good efficiency throughout the speed range. The mode shifts and consequently the full mechanical points are distributed according to equation (1). The hydraulic units are both considered to be of bent-axis design since over-centre capability is not necessary. The standing planetary gear ratios R_1 and R_2 are given by:

$$R_1 = -2.5 \tag{11}$$

$$R_2 = \frac{1}{1 + \frac{f_{ymech}}{\frac{1}{R_1} - 1}} \tag{12}$$

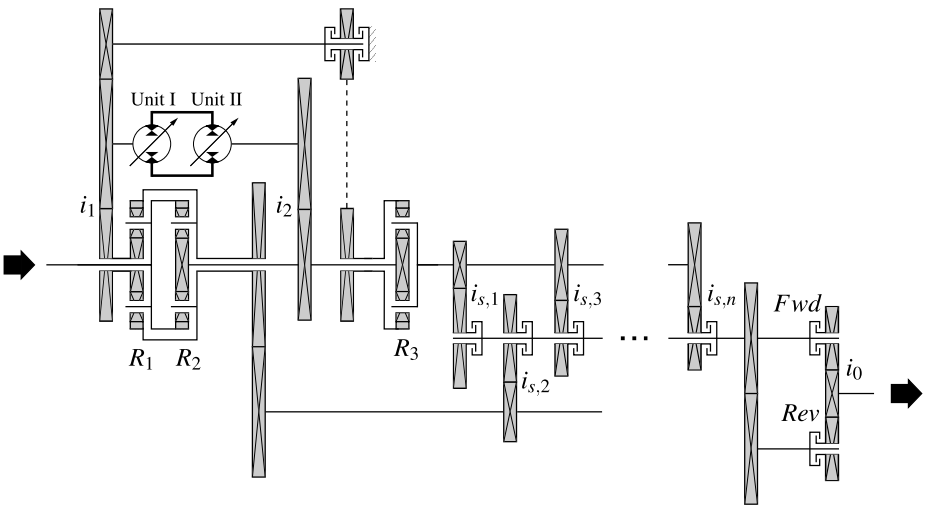

Figure 4. MVB transmission layout

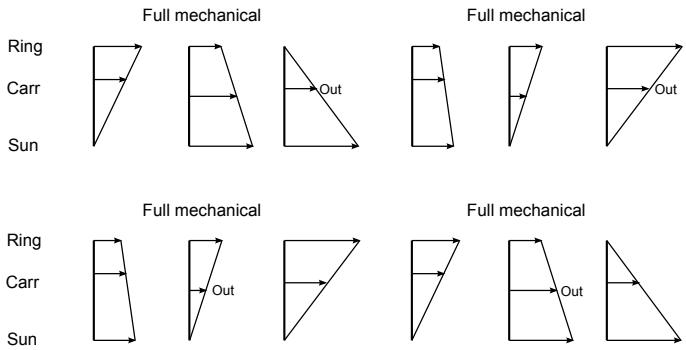

Figure 5. Kinematics of the planetary gears

The gear ratios corresponding to each mode are dimensioned according to equation (13) to ensure gap free mode switching:

$$i_{s,i} = \begin{cases} \frac{n_{shift,i+1}}{n_{II,max}} \frac{1-\frac{1}{R_2}}{i_0} & \text{for i = odd} \\ \frac{n_{shift,i+1}}{n_{ICE}} \frac{1}{i_0\left(1-\frac{1}{R_1}\right)} & \text{for i = even} \end{cases} \tag{13}$$

The gear ratios to the hydraulic units i_1 and i_2 are dimensioned according to equations (14) and (15), resulting in the hydraulic units running at maximum speed during each mode shift:

$$i_1 = \frac{n_{I,max}}{n_{ICE}} \frac{1}{1 - R_1 + \frac{R_1}{1-\frac{1}{R_2}}} \tag{14}$$

$$i_2 = \frac{n_{II,max}}{n_{ICE}} \frac{1}{1 - \frac{1}{R_1} + \frac{R_2}{R_1}} \tag{15}$$

The third planetary gear ratio R_3 is given by:

$$R_3 = -\frac{f_{vmech}}{i_2} \tag{16}$$

Unit I is sized to cover the flow demand of unit II:

$$D_I = \frac{n_{II,max}}{n_{I,max}} D_{II} = \left(\frac{C_{m,II}}{C_{m,I}}\right)^{3/2} D_{II} = D_{II} \tag{17}$$

Unit II is sized for the maximum startup torque:

$$D_{II} = \frac{20\pi F_{max} r_{tire}}{\Delta p_{max}} \frac{i_0\ i_{s,1}}{(1-R_3)\left(i_2 - \frac{i_1}{R_1(1-R_2)}\right)} \tag{18}$$

4 SIMULATION

Modelling and simulations of transmission concepts are performed using LMS Imagine.Lab AMESim [13]. AMESim supplies acausual system simulation with a wide range of component libraries making it easy to build up system models similar to their schematics. The predefined hydraulic and powertrain component models are used to some extent in order to complete the functionality of the system. To evaluate the energy efficiency of a transmission concept, more precise models of the key components are needed however. The predominant power losses of the transmission are caused by the hydraulic machines in the HST and must be modelled in detail. A mathematical model developed by Rydberg [14] where the power losses vary with pressure, speed and displacement is used in this study. The model describes the torque and flow losses with a polynomial expression according to equations (19), (20), (21) and (22).

$$Q_p = \varepsilon Dn - a_0 \varepsilon Dn - (a_1 + a_2\varepsilon) Dn \frac{\Delta p}{\beta_e} - a_3 \frac{D\Delta p}{2\pi\mu} - a_4 D\Delta p^2 \tag{19}$$

$$\begin{aligned} T_p = {} & \varepsilon \tfrac{D\Delta p}{2\pi} + (b_0 + b_1\varepsilon)\tfrac{D\Delta p}{2\pi} + (b_2 + b_3\varepsilon)\tfrac{D}{2\pi} p_L + \\ & b_4 \tfrac{|p_H + \delta p_L|}{1+\left(\frac{n}{n_0}\right)^{\gamma}} \tfrac{D}{2\pi} + b_5 \mu Dn + b_6 \varepsilon^3 \tfrac{D}{2\pi} n^2 \end{aligned} \tag{20}$$

$$Q_m = \varepsilon Dn + a_0 \varepsilon Dn + a_1 Dn \frac{\Delta p}{\beta_e} + a_2 \frac{D\Delta p}{2\pi\mu} + a_3 D\Delta p^2 \tag{21}$$

$$\begin{aligned} T_m = {} & \varepsilon \tfrac{D\Delta p}{2\pi} - (b_0 + b_1\varepsilon)\tfrac{D\Delta p}{2\pi} - (b_2 + b_3\varepsilon)\tfrac{D}{2\pi} p_L - \\ & b_4 \tfrac{|p_H + \delta p_L|}{1+\left(\frac{n}{n_0}\right)^{\gamma}} \tfrac{D}{2\pi} - b_5 \mu Dn - b_6 \varepsilon^3 \tfrac{D}{2\pi} n^2 \end{aligned} \tag{22}$$

The coefficients a, b, δ and γ are machine specific parameters. This model has been proved valid for axial piston machines with both in-line and bent-axis design over a wide operating range. The model parameters are adapted, using linear regression, to 3D efficiency maps supplied by the manufacturer throughout a limited range of operating points. The mathematical model becomes useful when predicting the power losses outside of this operating range. Hardware measurements have been made on a hydrostatic transmission with a $110\,cm^3/rev$ over centre in-line machine and a $150\,cm^3/rev$ bent-axis machine to validate the calculated efficiciency models. The resulting volumetric and hydromechanical efficiencies for $\varepsilon = 0.6$ can be seen in figures 6 and 7.

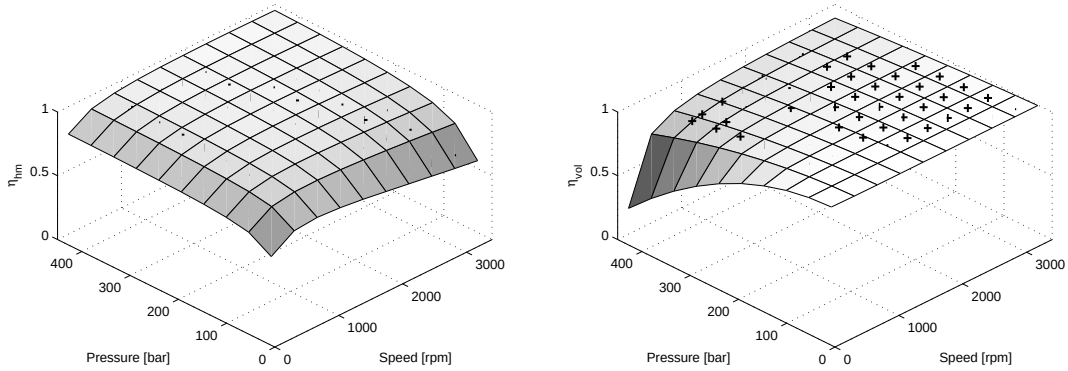

Figure 6. Efficiency models for in-line machine

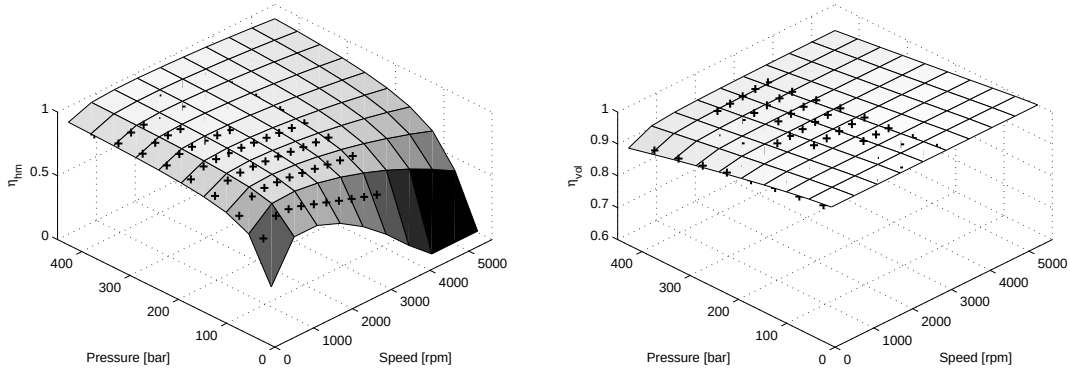

Figure 7. Efficiency models for bent-axis machine

The charge pump of the HST is mounted to the input shaft of the transmission for both considered concept architectures. This configuration produces a constant but not negligible power loss of the transmission. The displacement is considered to be $D_{cp} = 0.2max(D_I, D_{II})$ and the boost pressure is set to $p_{boost} = 2.0\,MPa$. The mechanical part of the transmission stands for a smaller part of the total power losses and a simpler efficiency model is used for this purpose. Each spur gear and planetary gear is modelled with a simple friction model that produces a speed dependent torque loss corresponding to figure 8. More detailed modelling of the gear stages and the planetary gears in PS transmissions are described in [15]. Additional steady-state losses in the mechanical part originated from bearings, clutches and sealings are considered sufficiently small to be ignored in this level of detail.

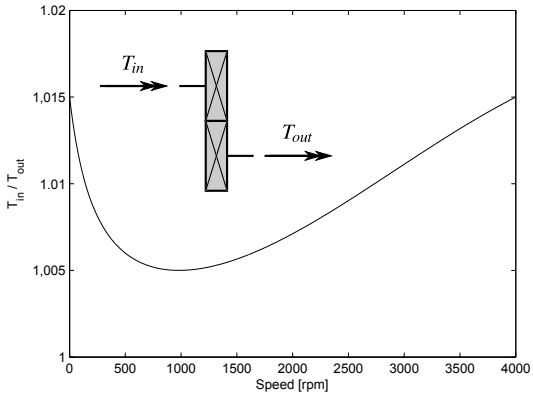

Figure 8. Model of torque losses in spur gears and planetary gears

5 RESULTS

Figure 9 shows the machine displacements as a function of number of modes for each of the proposed concepts. It is clear that the MIC concept generally requires larger machines. It can, however, be observed that the dimensioned displacement of the hydraulic units

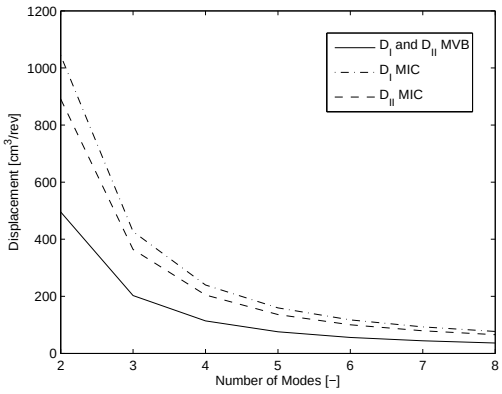

Figure 9. Displacements of the hydraulic machines with respect to number of modes

heavily depends on the speed constant C_m which is considerably higher for the variable bent-axis machines used in the MVB concept than for the in-line machine used in the MIC concept. Figure 10 shows the machine displacements vs. speed constant of both concepts for the 4 mode transmissions. The achieved curves are rather alike since the OC start mode in the MVB concept initially has all power flowing through the HST similar to the MIC starting mode. The HST dimensions are obviously also highly dependent on the tolerated maximum pressure of the machines. To evaluate the energy efficiency of the concepts, the transmissions are simulated for a case where they are performing an acceleration of the vehicle from standstill to maximum speed. Only positive speeds are simulated since both concepts behave similarly independently of the vehicle speed direction. The total efficiency is understood to be:

$$\eta_{trans} = \frac{P_{ICE}}{P_{wheel}} \tag{23}$$

Figure 11 exemplifies the results of the MIC concept for 2, 4 and 6 modes. The efficiencies peak at the full mechanical points, described in table 2 above.

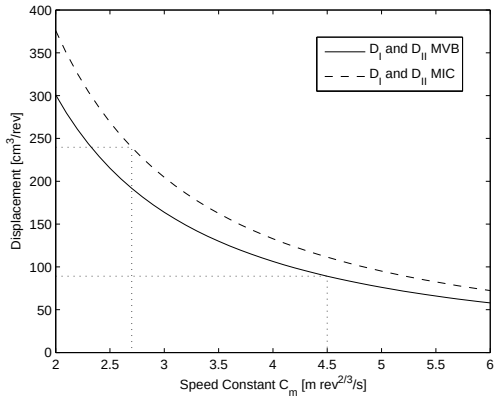

Figure 10. Displacements of the hydraulic machines for $m = 4$

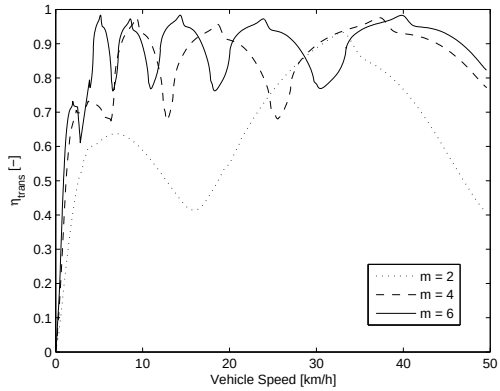

Figure 11. MIC transmission efficiency for $m = 2$, $m = 4$ and $m = 6$

The reason for the varying values for the top efficiencies is the high boost pump power needed for the transmissions with a low number of modes. A greater number of modes results in less power flow through the HST and increased efficiency. At the mode shifts, the transmission reaches its highest HST power ratio and thus the lowest efficiency. Figure 12 shows the corresponding results for the MVB concept. The efficiency is generally higher throughout the speed range due to less power flow through the HST. The peaks are distributed at the mode shifts, which coincide with the full mechanical points.

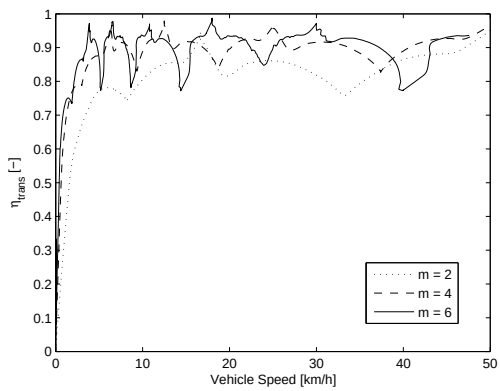

Figure 12. MVB transmission efficiency for $m = 2$, $m = 4$ and $m = 6$

It is also observed how close the 4-mode transmission comes to the 6-mode transmission efficiency. For a better understanding of how the efficiency changes with the number of modes, the total power losses are compared. Figure 13 shows a normalised graph of the transmission energy losses vs number of modes.

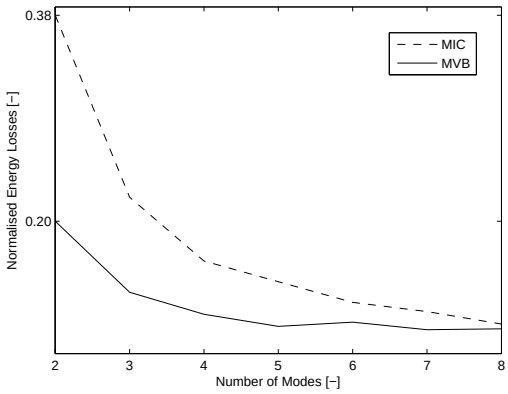

Figure 13. Energy losses vs number of modes for a maximum load acceleration

It is seen that the total energy losses are only marginally decreased at $m > 4$ for the MVB concept. At this point the total efficiency is only vagely improved with additional modes. Figures 14 and 15 show efficiency maps for partial loads for both concepts with $m = 4$. The dashed line encloses the operating range of a measured short loading cycle for the wheel loader. The loading cycle is performed with the reference vehicle equipped with a torque converter and a powershift gearbox.

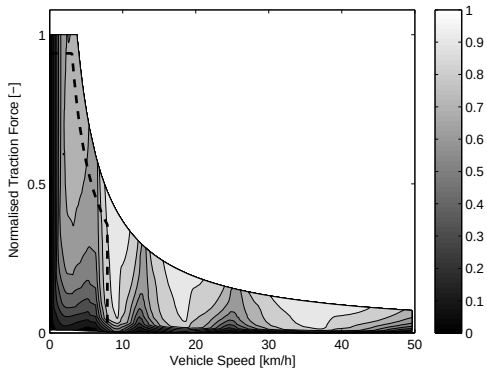

Figure 14. Efficiency map for the MIC concept with $m = 4$

The transmission efficiency is hence far from its optimal points during the loading cycle, although the transmission efficiency is high during maximum load conditions. Due to low pressures at the hydraulic machines, both concepts suffers from a poor HST efficiency during situations with lower loads.

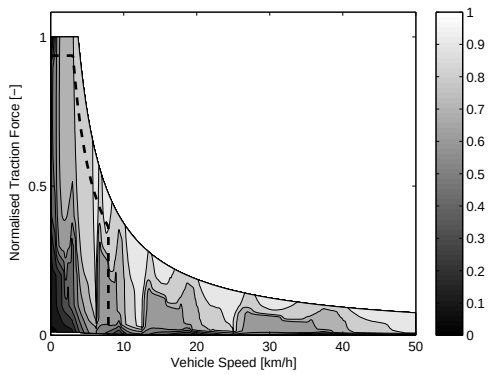

Figure 15. Efficiency map for the MVB concept with $m = 4$

6 CONCLUSIONS

Two scalable power split transmission architectures suitable for heavy wheel loaders have been evaluated and compared with respect to energy efficiency and transmission dimensions. The multiple input coupled concept is comparatively simple and still offers the desired power split advantages. The assembly requires fairly few mechanical gears and would result in a low installation volume and weight of the wheel loader gearbox. The multiple variable bridge concept is a more complex system resulting in higher energy efficiency and also smaller hydraulic machines being required. The necessary dimensions of the hydrostatic transmission are highly dependent on the machine operating range, which varies widely for different machine design types. The effects on transmission efficiency of increasing the number of modes are shown for each of the concept architectures. The resulting energy efficiency is, however, strongly coupled to the power losses in the hydrostatic transmission. During partial loads and at the limits of their working range, the hydraulic machines are forced to work at bad operating points, which affects total efficiency. To further increase the efficiency, it is necessary to dimension the mechanical part of the transmission with consideration to the efficiency maps of the hydraulic machines. Although there are demands for high transmission efficiency throughout the speed range, it is also of interest to optimise the operation points of standardised working cycles, such as the short and long loading cycle, for the highest energy savings. Further savings for the complete drive train is achievable with power management of the combustion engine by adjustment to the most favourable engine speed for a certain load situation.

REFERENCES

[1] B. Carl and M. Ivantysynova. Comparison of Operational Characteristics in Power Split Continuously Variable Transmission. *SAE 2006 Commercial Vehicle Engineering Congress and Exhibition, Chicago, IL, USA*, (SAE 2006-01-3468), 2006.

[2] J. Liscouet, J. C Ossyra, M. Ivantysynova, G. Franzoni, and H. Zhang. Continuously Variable Transmissions For Truck Applications - Secondary Control Versus Power Split. *5th International Fluid Power Conference (IFK), Achen*, 2006.

[3] R. Kumar, M. Ivantysynova, and K. Williams. Study of Energetic Characteristics in

Power Split Drives for on Highway Trucks and Wheel Loaders. *SAE 2007 Commercial Vehicle Engineering Congress & Exhibition*, (SAE 2007-01-4193), 2007.

[4] T. Fleczoreck, S. Kemper, and H. Harms. Partially Continuously Variable Power-split Transmission for Wheel Loaders. *6th FPNI - PhD Symposium, West Lafayette, IN, USA*, 2010.

[5] K. Renius. Continuously Variable Tractor Transmissions. *ASAE Distinguished lecture series, No 29*, 2005.

[6] Heinz Aitzetmüller. Hydrostatic - Mechanical Power Split Transmission for Locomotives. *International Conference on Gears, Munich*, October 2010.

[7] Bosch Rexroth AG. Hydromechanical Variable Transmission HVT. Website, 2009. http://www.boschrexroth.com/business_units/brm/en/products_and_solutions /hydraulic-systems/hvt-system/index.jsp.

[8] ZF Friedrichshafen AG. ZF cPOWER. Website, 2010. http://www.zf.com/corporate/en/products/product_range/construction_vehicles/ product_highlights/cpower/cpower.html.

[9] T. Kohmäscher, H. Jähne, and H. Deiters. Comparison Of Selected Fluidtechnical Drive Line Concepts For Off-Highway Machines. *5th International Fluid Power Conference (IFK), Achen*, 2006.

[10] Per Matsson. *Continuously Variable Split-Power Transmissions with Several Modes*. PhD thesis, Chalmers University of Technology, 1996.

[11] F. Jarchow. Continuous Acting Hydrostatic-Mechanical Power-Shift Transmission with Toothed Clutches. United States Patent 5,052,986, 1991.

[12] Mikael Sannelius. *On Complex Hydrostatic Transmissions - Design of a Two-Motor Concept using Computer Aided Development Tools*. PhD thesis, Linköping University, 1999.

[13] LMS International. *LMS Imagine.Lab AMESim Rev 10 User manual and software*, 2011.

[14] Karl-Erik Rydberg. *On Performance Optimization and Digital Control of Hydrostatic Drives for Vehicle Applications*. PhD thesis, Linköping University, 1983.

[15] T. Kohmäscher and H. Murrenhoff. Advanced Modeling of Hydro-Mechanical Power Split Transmissions. *6th FPNI - PhD Symposium, West Lafayette, IN, USA*, 2010.