Fuel potential and prediction sensitivity of a power-split CVT in a wheel loader

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Abstract: Wheel loader transmissions are commonly based on a torque converter and an automatic gearbox. This solution is mechanically robust and well suited for the typical operation of the machine, but the fuel efficiency is low at some modes of operation. One proposed improvement is to replace the present transmission with a multi-mode power-split CVT (MM-CVT). This paper compares the fuel saving potential of the MM-CVT to the potential of the present transmission under different assumptions on the prediction of future loads. A load cycle with a probability distribution is created from a measurement including 34 short loading cycles. Trajectory optimization is performed both against this, probabilistic, and three deterministic load cycles with the two concepts. The optimization shows that the MM-CVT transmission has at least 15% better fuel saving potential than the present transmission, and that this difference is not sensitive to the quality of the prediction or the smoothness or length of the load case.

Keywords: Continuously variable transmission, Hydraulic actuators, Optimization, Uncertainty, Vehicles

1. INTRODUCTION

1.1 Background

The common wheel loader operation is characterized by its highly transient nature and the periods of high tractive effort at low speed. The engine also has to deliver power both to the transmission and the working hydraulics pump. The most common general transmission layout of heavy wheel loaders is presented in figure 1. The engine is connected to the variable-displacement working hydraulics pump and a hydrodynamic torque converter. The torque converter is connected to an automatic gearbox, which connects to the drive shaft.

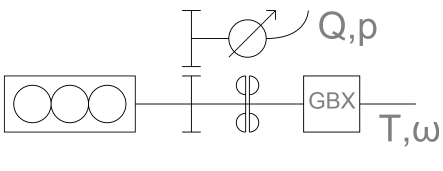

Fig. 1. Reference vehicle drivetrain setup

In this setup the torque converter is a crucial component, since it provides some disconnection between the engine and transmission speeds. This disconnection makes the system mechanically robust and prevents the engine from stalling if the bucket gets stuck. But the solution is also prone to high losses. Thrust at low speed is achieved by high torque converter slip, which produces losses. High hydraulic flow requires high engine speed, which also produces transmission torque which has to be balanced by the brakes, producing losses in both torque converter and brakes. This lack of efficiency is the reason for a desire to find other transmission concepts for wheel loaders.

1.2 On the choice of a hydraulic multi-mode CVT

Any alternative transmission has to enable increased efficiency at the typical operation conditions mentioned above. The low speeds at which the machine often operate makes it impractical to use a stepped gearbox without a torque converter. One alternative is to consider infinitely variable transmissions, such as the diesel-electric used in Filla (2008) or the hydrostatic used in Rydberg (1998). The drawback with this type of transmission is that the repeated power conversions reduce the efficiency. This is addressed by power-split constructions such as those described by Carl et al. (2006) and by Gramattico et al. (2010), in which some part of the power is mechanically transmitted. In such transmissions the efficiency depends on the power-split ratio, the proportion of power in the mechanical and CVT branches, which in turn depends on the gear ratio. If high efficiency at widely spaced gear ratios is required the power-split CVT has to be expanded. Multi-mode CVTs are constructed so that several power-split layouts can be realized with the same device. The continuously variable component in the device can be of any type, including belt, electric or hydraulic. Just like in Savaresi et al. (2004) a hydrostat is used, since this solution has a favorable cost and torque rating.

1.3 Optimal control

When the torque converter is replaced with a less flexible component the demand for active control increase (see for example Zhang (2002)). The quality of the control is also critical for the successful implementation of the concept. Evaluation of transmission controllers for wheel loaders is not trivial since the hydraulic and transmission loads are mutually dependent while filling the bucket, as described in Filla (2008). Evaluating controllers would therefore require rather complex environment models. Since control development is far from trivial, it is not desirable to spend time and effort developing control algorithms unless the fuel saving potential of the concept is high. The potential of a concept, excluding control, can be determined by state and control trajectory optimization.

It is not uncommon to use optimization to evaluate mechanical or control concepts. This has for example been done in Paganelli et al. (2000), Pfiffner (2001) and Sciarretta and Guzzella (2007). Since wheel loaders are off-road vehicles with highly transient operating patterns, access to accurate prediction in any form is improbable. In this paper this is address by examining the sensitivity of the fuel saving potential to uncertainties in the prediction. This is done by comparing the minimum fuel potential of the present drivetrain to that of the concept under deterministic and stochastic driving cycles.

2. PROBLEM

2.1 Problem statement

The problem studied in this paper is the minimization of the total amount of fuel needed for completing a prespecified driving mission, or load case. No deviation from the load case is allowed.

Since no deviations from the load case are allowed, it is assumed that a prediction of the load case exist, just as in Pfiffner (2001) and in Nilsson et al. (2011). Since perfect prediction is unrealistic, especially for heavy off-road equipment, uncertainties in the predictions are introduced based on statistics from measurements. Load cases without uncertainties are analyzed with deterministic dynamic programming (DDP) and load cases with uncertainties are analyzed with stochastic dynamic programming (SDP).

This paper compares two transmission concepts; the present torque converter/automatic gearbox and the multi-mode CVT. Models of similar complexity are used in both cases for a fair comparison. Identical diesel engine and hydraulic pump models are used in the two concepts.

2.2 Load cases

Wheel loader usage can typically be characterized as loading cycles, as in Filla (2008) and in Fengyuan et al. (2012). For efficient measurement data treatment, an automatic loading cycle identifier has been developed. This starts by detecting events such as bucket fillings and driving direction changes. The identifier then searches the sequence of detected events, utilizing automata theory, as described in Kelley (1998), for patterns corresponding to loading cycles. The measurement sequence used in this work consists of 34 loading cycles with durations between $21.5s$ and $30.6s$.

The load cases are specified by the wheel speed ω_w, wheel torque T_w, hydraulic pressure p_H and hydraulic flow q_H. In the deterministic cases, ω_w, T_w, p_H and q_h are deterministic. In the stochastic case p_H and q_H are stochastic and, since rapidly varying vehicle speed is unrealistic, ω_w is deterministic and T_w is only partly stochastic. The wheel torque is divided into two parts; $T_w = T_A(\frac{d\omega_w}{dt}) + T_D$ where T_A depends on the acceleration and is deterministic, while T_D is stochastic and includes the rolling resistance and longitudinal bucket forces.

When creating the stochastic cases the time scale in each detected cycle is altered so that the driving direction changes in each cycle coincide. The cycle durations are changed to $25s$, divided into $10s$ forward and filling of the bucket, $5s$ reversing, $5s$ forward toward, and including, bucket emptying and finally $5s$ reversing, so that the direction changes occur at the exact same times in each cycle. At each time instant, the mean E and standard deviation σ of each of the stochastic components are calculated. The new load has three alternatives ($W_Y = [E - \sigma, E, E + \sigma]$) of each of the three independent stochastic components $Y \in [T_D, p_h, q_h]$. Each of the alternative loads also has a corresponding probability P_Y. These probabilities are assumed to be independent.

$$P(W_Y(t_k)|W_Y(t_{k-1})) = P(W_Y(t_k)) \tag{1}$$

One of the possible trajectories is also used as actual load when the fuel consumption is calculated. The stochastic cycle is labeled the 'SDP *mc*' cycle. The load trajectory which is actually applied in the 'SDP *mc*' cycle is also used as a deterministic case, labeled the 'DDP *mc*' cycle. Since the averaging in the creation of the 'DDP *mc*' smoothen the cycle, optimization is also performed with one individual cycle from the set; the 'DDP *sc*' ($27s$). One longer ($137s$) cycle, the 'DDP *lc*', is also included in the set of load cases used. In this way the impact of uncertainties, of the averaging and of the cycle length can be studied.

Figure 2 shows the 'SDP *mc*' and 'DDP *mc*' load cases. The dotted lines are the alternative loads, while the continuous lines are the load alternative which was actually applied and is also used as the 'DDP *mc*'. Figure 3 shows the 'DDP *sc* load case. Note the sharp fluctuations, especially in T_w, which supports the independent probability assumption of Equation (1).

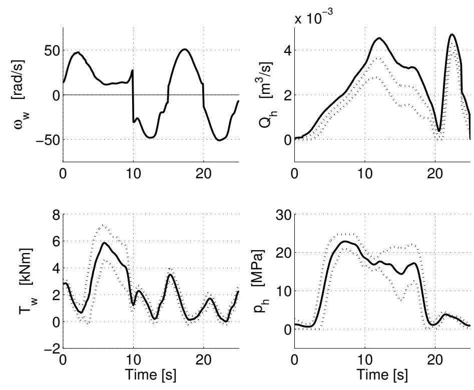

Fig. 2. The SDP *mc* load case

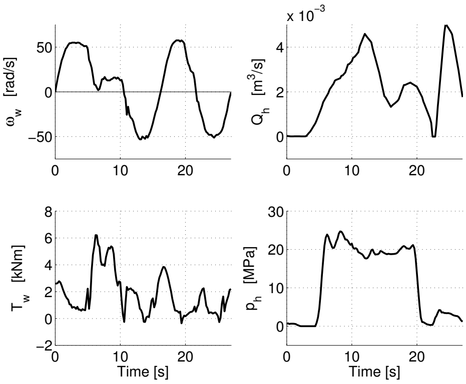

Fig. 3. The DDP *sc* load case

2.3 Engine model

The engine is modeled as an inertia I_e which is affected by the engine torque T_e, the transmission torque T_T and the hydraulic pump torque T_H.

$$\frac{d\omega_e}{dt} \cdot I_e = T_e - T_T - T_H \tag{2}$$

The relation between injected fuel and engine torque is described by a quadratic Willan's efficiency model, as presented in Rizzoni et al. (1999)

$$T_e = e(\omega_e, m_f) \cdot \frac{q_{lhv} n_{cyl}}{2\pi n_r} \cdot m_f - T_L(\omega_e) - T_{pt} \tag{3}$$

in which m_f is fuel mass per injection, ω_e is engine speed, e and T_L are efficiency functions, q_{lhv}, n_{cyl} and n_r are constants and T_{pt} is torque loss due to lack of air intake pressure $p_{off} = p_t - p_{set}(\omega_e, m_f)$. Here p_t is the actual pressure and p_{set} is a static setpoint map. The turbo is modeled as a first order delay for the intake air pressure

$$\frac{dp_t}{dt} \cdot \tau(\omega_e) = -p_{off}(\omega_e, m_f) \tag{4}$$

The torque loss from low intake pressure is described by

$$T_{pt} = \begin{cases} k_1(\omega_e) \cdot p_{off}^2 - k_2(\omega_e) \cdot p_{off} & \text{if } p_{off} < 0 \\ 0 & \text{if } p_{off} \geq 0 \end{cases} \tag{5}$$

Figure 4 presents the efficiency map of the engine used. The gray lines indicate allowed operating region (minimum speed and maximum torque) and the black line indicates the static optimal operating points for each output power. The figure also shows efficiency levels and output power lines with kW markings.

2.4 Reference transmission model

The transmission of the reference vehicle consists of a torque converter and a four speed forward/reverse automatic gearbox. The main source of losses in this transmission is the torque converter, which is modeled according to Equations (6), (7) and (8).

$$\nu = \frac{\omega_t}{\omega_p} \tag{6}$$

$$T_p = M_P(\nu) \left(\frac{\omega_p}{\omega_{ref}}\right)^2 \tag{7}$$

$$T_t = \mu(\nu) T_p \tag{8}$$

in which index-p is pump, or engine, side and index-t is turbine, or gearbox, side. $M_P(\nu)$ and $\mu(\nu)$ are maps measured at the reference speed ω_{ref}.

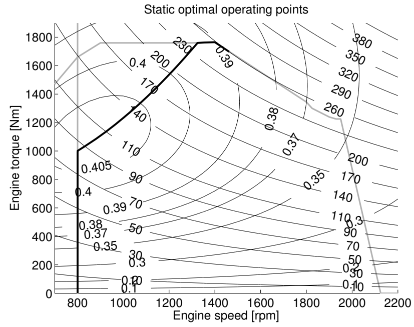

Fig. 4. Engine map with static optimal operation line, speed and torque limits, efficiency curves and output power lines with kW markings

2.5 Multi-mode CVT (MM-CVT) model

The concept transmission is the three mode ($m_T \in [1, 2, 3]$) CVT which is described in the patent Mattsson and Åkerblom (2012) and has a structure similar to those used in Savaresi et al. (2004) and Lauinger et al. (2007). The proposed layout is presented in Figure 5 in which the dark gray box represents a Ravigneaux planetary gearset and the light gray box represents a regular planetary gearset. The CVT functionality is provided by the two hydraulic machines *H1* & *H2* which together is called a variator. Changing gear ratio within a mode is done by altering the ratio between the displacement of the hydraulic machines. The clutches 1, 2 & 3 are used to select the mode by applying that which correspond to the desired mode. Mode shifts are performed at the extremals of the variator displacement, and mode shifts at these points do not change the overall gear ratio for a lossless transmission.

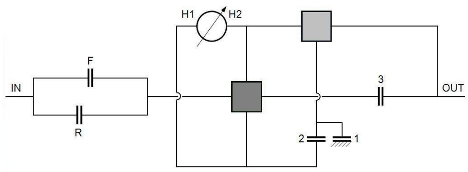

Fig. 5. Multi-mode CVT layout.

The main source of losses in this concept is the variator, which is modeled according to Lennevi (1995). Equations (9) and (10) describe the hydraulic machines.

$$\varepsilon_1 D_v \omega_1 \pm p_v(C_a + (\omega_1 + \omega_2)C_b) - \varepsilon_2 D_v \omega_2 = C_{vT} \dot{p}_v \tag{9}$$

$$\varepsilon_n D_v p_v - T_n \pm (C_c \omega_n + C_d p_v) = 0 \tag{10}$$

The index $n = 1, 2$ denotes the two machines, D_v is maximum displacement, $\varepsilon \in (0, 1)$ is relative displacement, ω is axle speed, p_v is variator hydraulic pressure, T is torque and C_a,C_b,C_c and C_d are efficiency parameters. The sign in the equations depends on the power flow direction. Equation (9) describes hydraulic fluid flow and Equation (10) describes torque at each machine. The variator is constructed so that $\varepsilon_1 + \varepsilon_2 = 1$. The variator pressure dynamics is assumed to be fast compared to the engine dynamics, that is; it is assumed that the time constant C_{vT} can be set to zero. At mode shifts the speed differences over the involved clutches are, ideally, zero. Therefore there will be no power losses in the clutches.

2.6 Hydraulics model

The bucket and boom are hydraulically driven. Pressure and flow of the hydraulic fluid are supplied by a hydraulic pump connected to the engine axle. This pump has variable displacement, so that the same pressure and flow can be provided at different engine speeds. Equations (11) and (12) describe the hydraulic pump

$$q_H = \varepsilon_H D_H \omega_e \tag{11}$$

$$q_H p_H = \eta_H T_H \omega_e \tag{12}$$

D_H is maximum displacement, $\varepsilon_H \in [0, 1]$ is relative displacement and η_H is pump efficiency.

3. METHOD

3.1 Dynamic programming

Denote the applied load w and the discretized state $x \in X$, controls $u \in U$ and time t_k with $k = 0, 1, ..., N$. The optimization problem can then be stated as

$$\min_{u \in U} \left(J_N(x_N) + \sum_{k=0}^{N-1} g(x_k, u_k, w_k) \right) \tag{13}$$

The deterministic dynamic programming (DDP) algorithm, which is described in detail in Bellman (1957); Bertsekas (2005), that recursively solves this problem can be formulated as

$$J_k(x_k) = \min_{u \in U} \Big(g(x_k, u_k, w_k) + \tilde{J}_{k+1}(x_{k+1}(x_k, u_k, w_k)) \Big) \tag{14}$$

in which $\tilde{J}_{k+1}$ is found by interpolating over J_{k+1}. The algorithm, as applied here, is described in more detail in Nilsson et al. (2011, 2012). In stochastic dynamic programming (SDP) there is some uncertainty in w_k. The algorithm is then expressed as the minimization of the expected value

$$J_k(x_k) = \min_{u \in U} E\Big[g(x_k, u_k, w_k) + \tilde{J}_{k+1}(x_{k+1}(x_k, u_k, w_k)) \Big] \tag{15}$$

Here the uncertainty is expressed as w_k being w_m with the probability p_m. Then g and $\tilde{J}$ are calculated for each w_m, weighted with the probability p_m and summarized

$$E\Big[g(x_k, u_k) + \tilde{J}_{k+1}(x_{k+1}(x_k, u_k)) \Big] = \sum_m p_m \Big[g(x_k, u_k, w_m) + \tilde{J}_{k+1}(x_{k+1}(x_k, u_k, w_m)) \Big] \tag{16}$$

This method is well known, straight-forward, does not require an initial guess and guarantees global optimum. The problem is that the number of simulations and interpolations grows rapidly with the density of the discretization. Further details about SDP can be found in Ross (1983).

Here the terminal cost is defined by Equation (17) and the cost $g(x_k, u_k)$ at each stage is the fuel mass used.

$$J_N = \begin{cases} 0 & \text{for } x_N \geq \Omega \\ \infty & \text{else} \end{cases} \tag{17}$$

The dynamics of this vehicle is described by (2) and (4), regardless of transmission. The controls available in the MM-CVT vehicle are fuel injection, variator displacement ratio and choice of gear mode. The controls available in the reference vehicle are fuel injection and choice of gear.

Application to the MM-CVT vehicle In the ideal case, without losses, $\omega_e(\varepsilon, m_T)$ is invertible. This means that the choice of either ε and m_T, or ω_e, as states is equivalent. The main losses in the transmission are leakages in the variator. These are load dependent, which means that the variator speed ratio, and thereby the total gear ratio, depends on the load. It can no longer be assumed that $\omega_e(\varepsilon, m_T)$ is invertible for all loads. Since $\omega_e(\varepsilon)$ is always invertible for the concept at hand, m_T and either ω_e or ε should be used as states. The possibility of restrictions on $\Delta\varepsilon$ during mode-shifts points to using ε as state. Since one of the hydraulic machines will speed up when ε gets close to 0 or 1, the losses in these regions increase. Because of this it is desirable to have higher state grid density near the extremes of ε, which also points toward using ε as state. The dynamics however are described in ω_e, so using ε as state implies the following computational scheme:

$$\varepsilon_k \xrightarrow{W_k} \omega_{e,k} \xrightarrow{\frac{d\omega_e}{dt}} \omega_{e,k+1} \xrightarrow{W_\kappa} \varepsilon_{k+1}$$

In the first and last step the load is required, since $\omega_e(\varepsilon)$ depends on the load. At the last step a choice has to be made whether to use $\kappa = k$ or $\kappa = k + 1$. Using $\kappa = k$ is equivalent to making a variable change in (2) from $\frac{d\omega_e}{dt}$ to $\frac{d\varepsilon}{dt}$. This choice of κ does not guarantee continuity in ω_e, due to simulation errors, and makes it possible for the optimizer to draw a net power from the engine inertia. $\kappa = k + 1$ on the other hand guarantees continuous ω_e, but causes a quadratic increase in load combinations in the stochastic case since $g(x_k, u_k, w_k) + \tilde{J}_{k+1}(x_{k+1}(x_k, u_k, w_k))$ would have to be evaluated for all combinations of W_k, W_{k+1}. Even though a small load variation space (3^3 combinations) is used, this would cause an unacceptable increase in calculation time. This means that for SDP it is not practical to use ε as a state, and that instead ω_e is used. Since $m_T(\omega_e)$ may not be well defined only in small regions near $\varepsilon = 0$ or 1, the state m_T is omitted and in ambiguous cases the m_T which give highest efficiency is used. Mode changes are only allowed at the extremes of the variator ratio, and since mode changes can no longer be explicitly controlled, this has to be checked in the optimization results. In DDP ε, m_T are used as states. The turbo speed obviously also has to be used as a state in both DDP and SDP.

Table 1. MM-CVT vehicle states and controls

	DDP	SDP
states	ε,m_T,p_t	ω_e,p_t
controls	m_f,Δm_T	m_f

Application to the reference vehicle The output torque from the torque converter depends on the input/output speed ratio. Since the drive shaft speed and torque are given by the load case, the engine speed will be implicitly given by the transmission speed and load. This means that rapidly varying load requires a rapidly varying engine speed. Since the power loss caused by slow change of engine speed is expected to be small compared to the loss in the torque converter, and since this model is only used as a reference for the MM-CVT vehicle, it is assumed that $I_e = 0$. Since there is a lower limit on the engine speed, there will also be a lower limit on the output torque at low speeds. The hydraulic flow requirement might raise the minimum engine speed due to the maximum displacement of the working hydraulics pump. If the transmission load is less than what is given by the minimum engine speed and acceleration is not desired, a brake torque has to be applied. Therefore in this model the states are gear g_T and turbo pressure and the controls are injected fuel m_f, brake torque T_b and gear change Δg_T. A minor cost ($\sim$1% of $\max(m_f)$) for changing gear is introduced to prevent excessive gear changes, though the clutch losses are assumed to be small by not associating any fuel usage to the gear changes.

Table 2. Reference vehicle states and controls

	DDP	SDP
states	g_T,p_t	g_T,p_t
controls	m_f,T_b,Δg_T	m_f,T_b,Δg_T

4. RESULTS

Table 3 summarizes the fuel usage in the eight cases that have been analyzed. The load cases are the deterministic and stochastic cases presented in Figure 2 (labeled *mc*), one non-averaged short loading cycle (labeled *sc*) and one similar long loading cycle (labeled *lc*). All load cases have been applied to both the reference- and the CVT-vehicle. The transmission induced fuel saving potential is just above 15%. The similar savings values for the 'SDP *mc*' case and the 'DDP *mc*' case suggest that the CVT vehicle is at least not more sensitive than the reference vehicle to load prediction uncertainties.

Table 3. Reference and MM-CVT vehicle fuel usage

	Reference [ml]	MM-CVT [ml]	saving [%]
DDP *mc*	200	169	15.5
SDP *mc*	221	184	16.7
DDP *sc*	222	187	15.8
DDP *lc*	930	771	17.1

Figures 6 and 7 shows the optimal state and control trajectories for the reference vehicle in the 'SDP *mc*' load case. The states are gear and turbo pressure. The turbo pressure set point is indicated with a dotted line.

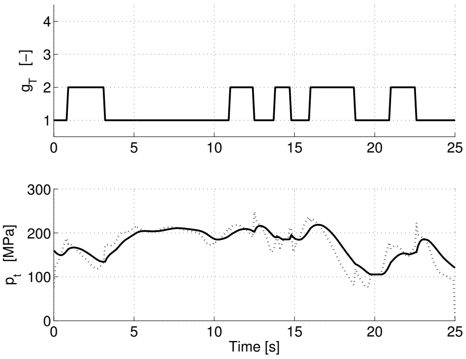

Fig. 6. Reference vehicle state trajectories (SDP *mc*)

Figure 8 presents the optimal engine operation for the reference vehicle in the 'SDP *mc*' load case. The minimum engine speed, given by the hydraulics flow and pump

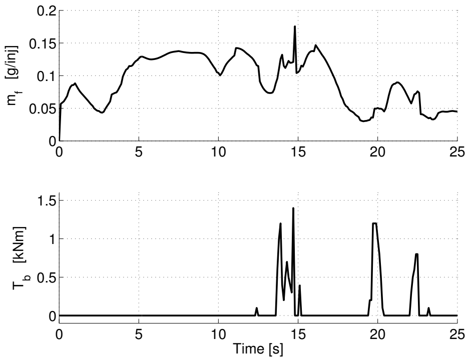

Fig. 7. Reference vehicle control signals (SDP *mc*)

maximum displacement, is indicated with a dotted line.

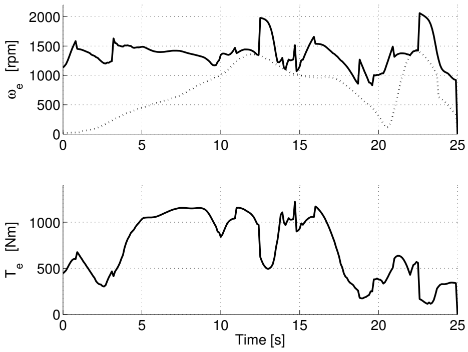

Fig. 8. Reference vehicle engine trajectories (SDP *mc*)

Figure 9 presents the optimal torque converter operation in the 'SDP *mc*' load case. The continuous lines are input, or engine, side and dotted is output, or gearbox, side. The power is always positive since there is no engine braking during this cycle. The ratio between the input and output powers indicate the transmission efficiency.

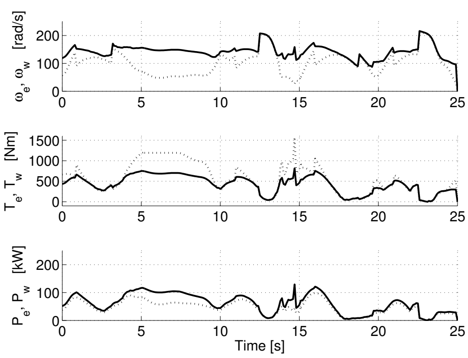

Fig. 9. Ref. vehicle torque converter operation (SDP *mc*)

Figure 10 presents the optimal engine operation for the MM-CVT vehicle in the 'SDP *mc*' load case. The minimum engine speed given by the hydraulics flow is indicated with a dotted line.

Figure 11 presents the internal view of the optimal CVT operation in the 'SDP *mc*' load case. The figures to the left show gear mode and variator displacement ratio, while the three figures to the right show clutch input/output speed

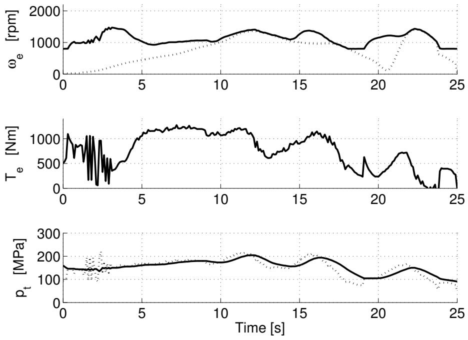

Fig. 10. MM-CVT vehicle engine operation (SDP *mc*)

differences. Due to the low speeds in the SDP load case the vehicle mainly operates in the first gear mode, and never in the third. The clutch speed difference figures show that the speed differences at mode changes are near zero, and therefore the clutch losses are also close to zero.

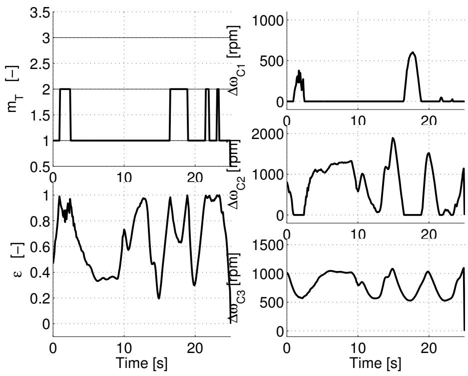

Fig. 11. MM-CVT vehicle CVT operation (SDP *mc*)

Figure 12 presents the external view of the optimal CVT operation in the 'SDP *mc*' load case. The continuous lines are input, or engine, side and dotted is output, or drive shaft, side. The power is always above zero since there is no engine braking during this cycle. The ratio between the input and output powers indicates the transmission efficiency.

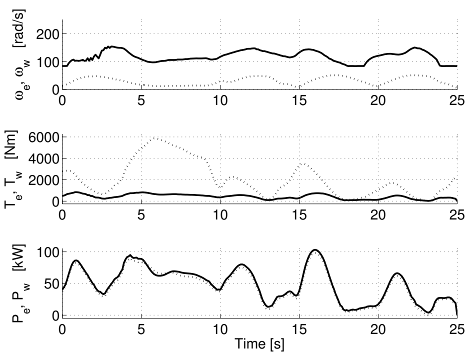

Fig. 12. MM-CVT vehicle CVT operation (SDP *mc*)

Figure 13 compares the optimal engine operation of the reference (dotted) and CVT (continuous) vehicles. The figure shows engine speed, engine torque and turbo pressure.

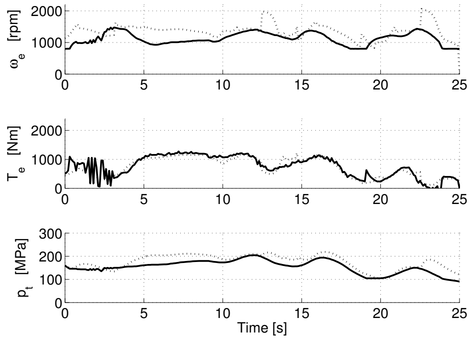

Fig. 13. Ref./MM-CVT vehicle engine operation (SDP *mc*)

Figure 14 compares the average efficiencies and power losses of the reference (black) and CVT (gray) vehicles, divided into engine including turbo losses (E), transmission (T) and hydraulics pump (H). In all three components the efficiencies are higher and losses lower for the CVT vehicle.

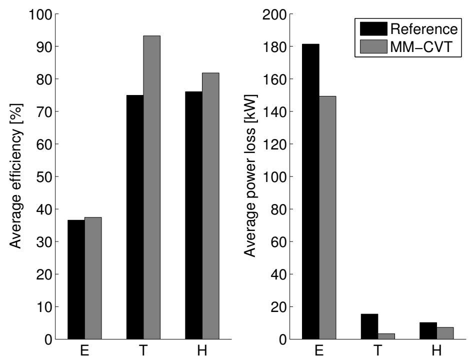

Fig. 14. Reference and MM-CVT vehicle power losses (SDP *mc*), divided into engine (E), transmission (T) and working hydraulics pump (H)

Figure 15 compares optimal engine and variator operation with $J(t, X)$ from the deterministic DDP *mc* (dotted) and stochastic SDP *mc* (continuous). In general the engine speed is lower, occasionally giving a higher CVT mode, in the DDP case.

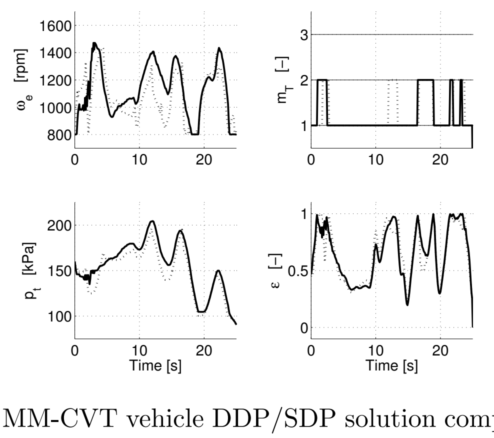

Fig. 15. MM-CVT vehicle DDP/SDP solution comparison

5. DISCUSSION

5.1 Drivetrain

The result of this study shows that the concept provide the potential for more than 15% reduction of fuel consumption. This number is for the optimal operation of both reference and CVT-transmission vehicles. The present vehicles are in general operated in a fashion quite far from the optimal; changing driving direction for example is often done by changing gear direction and braking with engine torque. This means that the fuel saving potential is much larger than the indicated 15%, but since this work does not attempt to evaluate the control it would not be fair to include control inflicted losses. The mentioned 15% is the increase in potential, when changing transmission.

Figure 13 shows that the engine output power is, as expected, higher for the reference vehicle both during the loading phase (around $5s$-$10s$), when high thrust gives high torque converter losses, and during low speed combined with bucket raising ($10s$-$17s$) or tilting ($22s$-$23s$) since this requires that a brake torque is applied at the output side of the torque converter.

Figure 14 show that there are three individual effects that give the fuel saving. First, the CVT provide an increased freedom in the choice of engine operating point. This means that the mean engine efficiency can be increased. Since the engine used has a rather flat efficiency map, the mean efficiency increase is quite small (36.7% to 37.4%), but due to the high average output power this correspond to more than $30kW$ average loss reduction. Second, the removal of the torque converter reduces the average transmission losses from about $16kW$ to $3.4kW$; an efficiency increase from 74% to 93%. The third effect is that the average displacement of the hydraulics pump can be increased, raising the average efficiency from 76% to 82%, reducing the average loss by about $3kW$.

The CVT-vehicle response to uncertainty in future load is, just as for the reference vehicle, to increase engine speed. In Figure 15 this is most visible at $12s$-$14s$ and $21s$-$22s$, where the vehicle use the second gear mode ($m_T = 2$) only in the deterministic case.

5.2 Formulation and method

This paper is focused on the loading cycle, since this is typical wheel loader usage and the repetitiveness can be used for prediction. Prediction is however inherently uncertain, a fact that is addressed by introducing the stochastic cycle. Due to the calculatory effort required for performing an SDP analysis a choice was made to focus on short cycles. The data treatment performed to make the SDP-load cycle essentially give a low-pass filtering. As indicated in Table 3, two separate cycles were analyzed with DDP. These are one short cycle, to investigate the effect of the low-pass filtering, and one long cycle, to investigate the effect of different cycle lengths and vehicle speeds. It turned out that the average power losses and efficiency levels, as presented in Figure 14, were only marginally affected. The only exception was the average hydraulic pump power loss, which was lower in the long cycle due to the lower average hydraulics use (longer transportation distance). It is concluded that the load cases analyzed with SDP adequately represent repetitive wheel loader usage.

The choice of uncertainties was restricted by the optimization method, since dynamic programming is not practical when many states are required. A stochastic vehicle speed with independent probability would cause extreme and time discretization dependent accelerations. A stochastic speed derivative would be a more natural description, but this would only cause inertia forces unless the vehicle speed is a state. The same argument can be made for using bucket height as a state, with a stochastic derivative that give the hydraulic flow. The difference is that vehicle speed is required in the loss model while bucket height is not. Therefore it can be assumed that on average, the height will stay within the feasible region, the state is not needed and a stochastic flow can be used.

6. SUMMARY AND CONCLUSIONS

The transmission concept analyzed in this paper has more than 15% better fuel consumption potential than the present drivetrain. That is, the best possible control of the new concept would give over 15% lower fuel consumption than the best possible control of the present drivetrain. This potential is not sensitive to prediction errors, cycle smoothness or cycle length. The savings are made in better choice of engine operating point, better hydraulics pump operating point and higher transmission efficiency due to the ellimination of the torque converter. The main power saving is in the choice of engine operating point while the biggest efficiency improvement is in the transmission. If future machines should be powered by diesel engines, we recommend that this type of transmission be further analysed due the high improvement in transmission efficiency.

The performance of the concept depends on the controller, to a much higher degree than for the present transmission. Today the torque converter transmission performs far from the optimal presented in this paper, due to lack of active control and prioritization of driveability over fuel consumption. The MM-CVT concept on the other hand will not be operable without active control, regardless of prioritizations. Due to the decisive importance of the controller, any further analysis of the new drivetrain should include investigation of performance, requirements and applicability of different controller concepts. The work presented here can be used for benchmarking of such controllers.

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