

Hydrostatic Wheel Drives for Vehicle Stability Control

2010-01-0105

Published
04/12/2010

Sisay Molla, Justin Sill and Beshah Ayalew
International Center for Automotive Research, Clemson University

Copyright © 2010 SAE International

ABSTRACT

Hydrostatic (hydraulic hybrid) drives have demonstrated energy efficiency and emissions reduction benefits. This paper investigates the potential of an independent hydrostatic wheel drive system for implementing a traction-based vehicle lateral stability control system. The system allows an upper level vehicle stability controller to produce a desired corrective yaw moment via a differential distribution of torque to the independent wheel motors. In cornering maneuvers that require braking on any one wheel of the vehicle, the motors can be operated as pumps for regenerating energy into an on-board accumulator. This approach avoids or reduces activation of the friction brakes, thereby reducing energy waste as heat in the brake pads and offering potential savings in brake maintenance costs.

For this study, a model of a 4×4 hydrostatic independent wheel drive system is constructed in a causal and modular fashion and is coupled to a 7 DOF vehicle handling dynamics model. The integrated system model is then used to first verify component selection and hybrid control threshold settings for the independent drive system. Then, a vehicle stability controller is set up as a cascade of a yaw controller and a torque distribution strategy. The overall system is evaluated by simulating a reduced handling test maneuver. The energy recovery attributes of the independent drive system are clearly shown as changes in accumulator state of charge during the maneuver.

1. INTRODUCTION

Concerns about resource depletion and global warming have accelerated the exploration of alternative and energy efficient vehicle propulsion systems. One of these proposed alternatives are hydrostatic (hydraulic hybrid) drives, which have been shown to improve fuel efficiency over

conventional drivetrains, particularly for urban vehicles characterized by frequent stop and go motion [1,2,3]. The high specific power (peak power per unit mass) of hydrostatic components opens up the possibility of using multiple but smaller pump/motors for a per-axle or per-wheel drive system.

On the other hand, vehicle stability control (VSC) systems help reduce accidents by minimizing driver's loss of control of the vehicle during emergency/aggressive maneuvers. Central to these VSC (also referred to as vehicle dynamics control (VDC)) systems is the generation of a corrective yaw moment on the vehicle through controlled proportioning of the longitudinal force on each tire of the vehicle. Most VSC systems available on the market today are brake-based and merely extend the functionality of mature hardware in ABS systems to affect the differential braking of individual wheels of the vehicle and generate the required corrective yaw moment [4,5,6]. However, this strategy could involve too much slow down of the vehicle against driver intent and lead to energy waste in the brakes and accelerated wear of brake-linings. The alternative approach for generating the corrective yaw moment for VSC is to distribute the tractive force between driving wheels and eliminate the listed drawbacks of brake-based VSC strategies [7,8,9,10]. Current solutions employing this strategy are the so-called torque-vectoring systems involving active differentials within conventional IC engine powertrains [11, 12].

In this paper, we study a solution where the traction force of each individual wheel can be controlled by modulating the torque output of the individual wheel motors [8,9,10]. The goal is to exploit the arrangement and integrate both the demonstrated energy efficiency benefits of hydraulic hybrids and the safety of vehicle stability control. We focus the discussion on a series hydraulic hybrid vehicle featuring independent wheel drives, but the VSC strategy can be

implemented with similar independent wheel drives for series electric hybrids and pure electric and fuel cell vehicles.

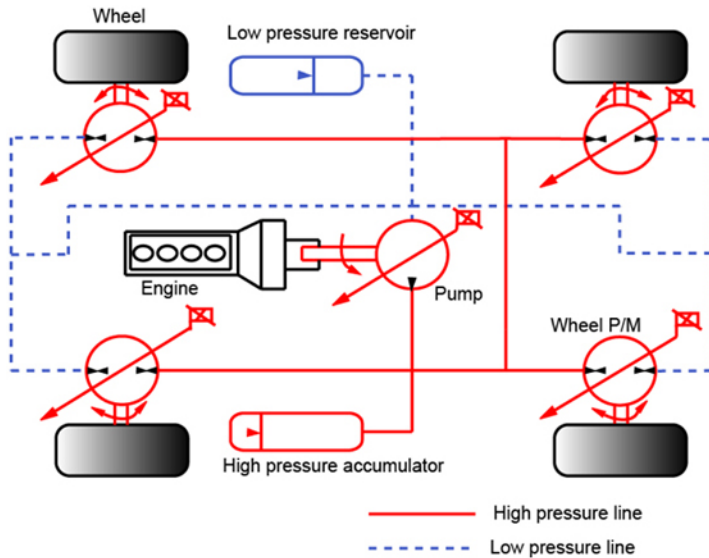


Figure 1. Schematic of a 4x4 Independent Hydrostatic Wheel Drive

Figure 1 shows the schematic of the proposed system. The engine (ICE) is directly connected to the pump. The fluid from the pump either charges the accumulator or directly flows to the individual wheel-end pump/motor (P/M). The ICE can be turned off when the vehicle comes to a full stop or when the SOC of the accumulator is greater than a minimum threshold value. The wheel-end P/M can be operated as a motor in drive mode or as a pump when regenerative braking is initiated and/or when the vehicle stability control dictates that it operate as a pump.

The rest of the paper is organized as follows. Section 2 details the system modeling adopted for the complete hydrostatic system as well as for the vehicle dynamics. Section 3 details the supervisory control strategies for powertrain performance and lateral stability control. Section 4 presents some results and discussions based on computer simulation of the developed models. Section 5 summarizes the conclusions of the work and highlights directions for further research.

2. SYSTEM MODELING

Figure 2 shows the high level system model architecture. It comprises of the subsystems of the engine and hydrostatic powertrain, the supervisory controller, the driver and vehicle dynamics models. The individual components of each of these subsystems are developed on the basis of forward-facing models interconnected by enforcing strict physical causality. The details of the component models are briefly outlined in the following subsections.

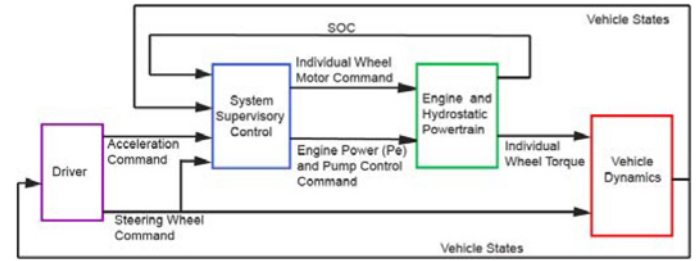


Figure 2. Schematics of high level system model

2.1. ENGINE AND HYDROSTATIC POWERTRAIN

Figure 3 shows the causal interconnection adopted for integrating the engine model, hydrostatic component models and the vehicle dynamics model.

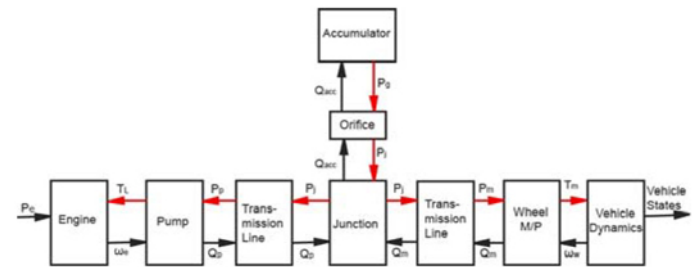


Figure 3. Input-output (Causality) relationships of hydrostatic powertrain component models

2.1.1. Engine Subsystem Model

The engine is implemented as a lookup table from the fuel consumption map of the engine and incorporating the dynamics of the engine/pump rotational inertia. The engine power (P_e) determined by the power management strategy in the supervisory controller and the minimum BSFC line of the engine intersect at a point on the engine torque-speed map which gives the desired engine torque T_e and desired engine speed ω_{e_des} . The engine output torque is considered to match the desired (i. e., torque generation delays are neglected). However, the actual speed of the engine/pump ($\omega_{e/p}$) is determined from the dynamics:

$$T_e - T_L = J_{eq.} \cdot \dot{\omega}_{e/p} \quad (1)$$

where T_L is the load torque (pump torque) on the engine, and $J_{eq.}$ is of the engine/pump rotational inertia. The actual speed of the engine/pump from Eq. (1) is controlled via the displacement of the pump (through its displacement factor x defined below) to track the desired engine speed, ω_{e_des} . A PI controller is tuned to minimize the speed error.

2.1.2. Hydrostatic Powertrain Model

As shown in Figure 3, the hydrostatic powertrain connects the engine and the vehicle dynamics models.

Pump/Motor Model

The 4 pumps/motors (P/M) considered in this work are of the bent-axis design and are mechanically coupled to the wheels of the vehicle through a single gear ratio speed reduction unit. The P/M units convert available hydraulic power from the engine-mounted pump or the accumulator in to mechanical power for vehicle propulsion (motor mode), or convert the kinetic energy of the vehicle to hydraulic energy for storage in the accumulator (pump mode) during regenerative braking. Both the motor mode and pump mode can be activated for any one of the P/M units to generate a corrective yaw moment for vehicle stability control (VSC).

The P/M units (either at the engine or at the wheel-end) are modeled here as depicted in Figure 4. The main approach is to use a 3-D look up table using data from steady state measurements. The torque and the flow rate through the variable displacement P/M can be controlled by varying the displacement factor, x , which is defined as the ratio of the prevailing displacement to the maximum displacement of the machine. The causality of the model adopted is such that x , ω , and ΔP across the P/M are inputs and the volumetric efficiency η_v and the mechanical efficiency η_m of the P/M are interpolated for as outputs. Knowing the volumetric and mechanical efficiency values, the flow rate and the torque of the P/M can be computed by using the following sets of equations[13, 14]. First, the ideal flow rate of the P/M is given by:

$$Q_i = x\omega D \quad (2)$$

Then, the actual flow rate of the pump and the motor are calculated, respectively, by:

$$\begin{aligned} Q_{ap} &= Q_i * \eta_v \\ Q_{am} &= \frac{Q_i}{\eta_v} \end{aligned} \quad (3)$$

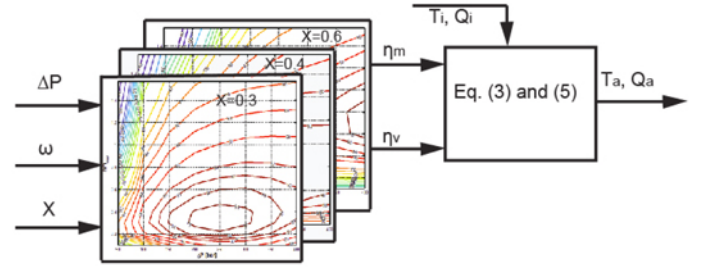


Figure 4. Steady-state lookup table based modeling of pumps and motors

Similarly, the ideal torque of the P/M is given by:

$$T_i = x\Delta PD \quad (4)$$

Then, the actual torque of the P/M is given by:

$$\begin{aligned} T_{ap} &= \frac{T_i}{\eta_m} \\ T_{am} &= T_i * \eta_m \end{aligned} \quad (5)$$

The look up table approach we used here avoids the need for the numerous dimensionless numbers and loss coefficients frequently used in pump/motor modeling following the so called Wilson's pump theory[15, 16].

Accumulator/Reservoir Model

An accumulator is a pressure vessel that contains a hydraulic fluid and a pressurized gas (mostly nitrogen) where the two sides are separated by a bladder, a diaphragm or a piston. When hydraulic fluid is pumped in, the gas is compressed, resulting the pressure to increase and store energy. When the fluid is discharged through the P/M (in motor mode), the pressure in the gas decreases while delivering propulsion energy. A reservoir (or low pressure accumulator) is a hydraulic accumulator working at much lower pressure than the high pressure accumulator, but just enough to prevent the occurrence of cavitation in the pumps[3].

Detailed modeling of hydraulic accumulators has been undertaken earlier by Pourmovahed[14, 17]. Considering the use of elastomeric foam on the gas side of the accumulator (to reduce irreversible heat losses) and taking energy balance on the gas side, it can be shown that the temperature evolution is given by[14, 18]:

$$\left[1 + \frac{m_f c_f}{m_g c_v} \right] \frac{dT}{dt} = \frac{T_w - T}{\tau} - \frac{T}{C_v} \left(\frac{\partial p_g}{\partial T} \right)_v \frac{dv}{dt} \quad (6)$$

where τ is the thermal time constant and is defined as $\tau = m_g c_v / h A_w$. The pressure in the accumulator is related to the gas temperature and the specific volume through a real-gas equation of state. Here, we adopted the Beattie-Bridgeman (BB) equation of state given by [19]:

$$p_g = \frac{RT(1-\varepsilon)}{v^2} + (v+B) - A/v^2 \quad (7)$$

where,

$A = A_0(1-a/v)$, $B = B_0(1-b/v)$, $\varepsilon = c/(vT^3)$ and A_0, B_0, a, b, c are constants in the BB equation of state.

The state of charge (SOC) of the accumulator is defined as the ratio of the instantaneous oil volume in the accumulator to the maximum oil capacity. However, measuring the instantaneous oil volume is not straightforward in a real-time application. As long as the temperature variation in the accumulator is kept low, the more directly measurable fluid/gas pressure can be used as an indicator of the SOC of the accumulator provided appropriate margins are considered [20].

Hydraulic Transmission Lines

The frequency of interest in the present study is rather low (a <10 Hz or so) that the dynamic effects of transmission lines are negligible compared to the dynamics of the other components. Therefore, a one dimensional lumped parameter resistive model, considering laminar or turbulent losses depending on the computed Reynolds number [14], is used for the transmission lines instead of more elaborate distributed parameter models [21].

2.1.2. Vehicle Dynamics Model Integration

The above hydrostatic powertrain model is integrated with a 7-DOF handling dynamics model [9, 10]. The model adopted includes the degrees of freedom of lateral, longitudinal and yaw motions of the vehicle body, as well as the rotations for each of the four tires. The schematic and equations for the adopted model are listed in the [Appendix](#). Detailed derivations and discussions of the model are given in [9, 10, 22]. The tire/wheel dynamics connect the hydraulic wheel-end P/M torque to the dynamics of the vehicle via the longitudinal tire forces. The tire/wheel dynamics are given by:

$$I_w \dot{\omega}_i = T_{m,i} - F_{x,i} R_w \quad \text{for } i = LF, RF, LR, RR \quad (8)$$

where LF, RF, LR and RR stand for left front, right front, left rear, and right rear tires. In the present study, since tractive forces are intended to influence the lateral dynamics, a proper tire model that considers combined slip conditions

(longitudinal and lateral) must be used. However, tire data are often available as separate functions of the tire longitudinal slip ratio and normal load for longitudinal force (as $F_{x,\kappa} = F_x(\kappa, F_z)$) and the tire slip angle and normal load for the lateral force (for $F_{y,\alpha} = F_y(\alpha, F_z)$). The tire slip ratios and slip angles are computed from:

$$\kappa_i = \frac{\omega_i R_w}{V_{xi}} - 1 \quad \text{for } i = LF, RF, LR, RR \quad (9)$$

$$\alpha_{LF} = \tan^{-1} \left(\frac{V_y + l_f \dot{\psi}}{V_x + \frac{T_f}{2} \dot{\psi}} \right) - \delta, \quad \text{and, } \alpha_{LR} = \tan^{-1} \left(\frac{V_y - l_r \dot{\psi}}{V_x + \frac{T_r}{2} \dot{\psi}} \right) \quad (10)$$

where the RF, RR side slip angles are defined similarly (See the [Appendix](#) for definition of the variables). In this work, we considered the normal load changes that occur due to lateral and longitudinal accelerations as well as differences in the front and rear roll-stiffness distributions. The equations for updating the normal loads are given in the [Appendix](#). The required combined slip model is arrived at by assuming that the contours of constant slip angle and constant slip ratio (in a F_y vs. F_x map) can be approximated by elliptical curves whose equations are given, respectively, by:

$$\left(\frac{F_x}{F_{x,\max}} \right)^2 + \left(\frac{F_y}{F_{y,\alpha}} \right)^2 = 1 \quad (11)$$

$$\left(\frac{F_x}{F_{x,\kappa}} \right)^2 + \left(\frac{F_y}{F_{y,\max}} \right)^2 = 1 \quad (12)$$

These are rather good approximations for actual tire data as can be seen in [23,24,25,26]. The set of ellipses for the constant slip angle contours (Eq.(11)) have a common point at the maximum longitudinal force. $F_{y,\alpha}$ is the lateral force for the pure lateral slip condition under a given slip angle. Therefore, when maximum longitudinal force is observed ($F_x = F_{x,\max}$) there is no lateral force capacity. Conversely, the set of ellipses for the constant slip ratio contours (Eq.(12)) have a common value at the maximum lateral force, where $F_{x,\kappa}$ is the longitudinal force for the pure longitudinal slip condition under the given slip ratio. As F_y approaches $F_{y,\max}$, the tire's longitudinal force capability reduces. Given the family of curves given by Eqs.(11) and (12), the combined tire forces for a given pair of slip angle and slip ratio can be found at the intersection of the ellipses. It can be shown that