Paper Number

Development of a Hydro-Mechanical Hydraulic Hybrid Drive Train with Independent Wheel Torque Control for an Urban Passenger Vehicle

James D. Van de Ven†
Department of Mechanical Engineering
Worcester Polytechnic Institute
100 Institute Rd.
Worcester, MA 01609
e-mail: vandeven@wpi.edu

Michael W. Olson and Perry Y. Li‡
Center for Compact and Efficient Fluid Power
Department of Mechanical Engineering
University of Minnesota
111 Church St. SE
Minneapolis, MN 55455
e-mail: {mwo,pli}@me.umn.edu

ABSTRACT

This paper presents a hydraulic hybrid vehicle drive train to improve the fuel efficiency of a passenger car. The developed hydro-mechanical drive train enables independent control of the torque at each wheel. The motivation for developing this drive train is a hydraulic hybrid vehicle test bed for the Center for Compact and Efficient Fluid Power at the University of Minnesota. The hydro-mechanical hybrid drive train is modeled and compared to a series hybrid drive train in operation on the EPA Urban Dynamometer Driving Schedule. The hydro-mechanical system demonstrates excellent fuel economy potential, yet requires development work in the area of pump/motors with high efficiency at low displacement fractions.

INTRODUCTION

A major component of global energy consumption is transportation, which consumes 4.8 billion barrels of crude oil per year. Of the transportation industry, passenger cars consume 2 billion barrels of oil per year with a value of $100 billion, as of 2003[1)]. This substantial fuel consumption is the motivation for the development of a hydraulic hybrid drive train that significantly improves the efficiency of a passenger car. This drive train is being developed for a test bed vehicle in the Engineering Research Center for Compact and Efficient Fluid Power at the University of Minnesota.

A hybrid vehicle contains two sources of power consisting of an internal combustion engine and a second power source that allows for energy storage. The energy storage is used during braking events and other drive train control strategies to minimize fuel consumption. Two auxiliary power sources have been found most practical: electric motor/generators combined with batteries and hydraulic pump/motors combined with hydraulic accumulators.

Electric hybrid vehicles have been the first hybrid technology to be mass produced for the commercial passenger car market. A strength of electric hybrids is the high energy density of electric batteries, allowing for large energy storage in relatively compact and lightweight batteries. A substantial shortcoming of electric hybrids is the relatively low power density of both electric motor/generators and batteries at approximately 30-100 W/kg [2)].

Switching the second hybrid power source to hydraulics realizes benefits in a multiple areas. 1) The power density of hydraulic pumps/motors and accumulators is very high at approximately 500-1000 W/kg[2)]. 2) Hydraulic components are inexpensive when compared with electrical components, especially advanced battery packs. 3) Certain hybrid architectures allow for independent control of the torque at each wheel, which opens numerous possibilities for vehicle dynamics control. 4) Recent and developing technologies such as digital hydraulic valves and high energy density

† Previously with the Center for Compact and Efficient Fluid Power, University of Minnesota.
‡ Corresponding author: Perry Li.

accumulators are improving the future outlook of hydraulic hybrid vehicles.

A prime advantage of certain hydraulic hybrid drive train architectures is leveraging the intrinsically high power density of the hydraulic energy storage system through optimal engine management. Internal combustion engines create power most efficiently at relatively high power levels near the RPM of the peak torque output. Operating at other conditions decreases the energy conversion efficiency. An optimal engine management scheme runs the engine near the peak efficiency, with a portion of the power being transferred to the wheels while the additional power is stored. Once the accumulator is charged to a desired state, the engine is shut off and power for vehicle propulsion is supplied by the accumulator.

There are three commonly distinguished hydraulic hybrid drive train architectures. The parallel system uses a traditional mechanical drive train with a hydraulic pump/motor in-line between the transmission and the axle. The mechanical transmission of the parallel system provides an efficient transfer of power from the engine to the wheels, yet the engine speed is determined by the vehicle speed and the available gear ratios, preventing optimal engine management.

In a series hybrid system, the engine is directly coupled to a hydraulic pump and the wheels are coupled to one or more hydraulic pump/motors. This purely hydraulic transmission allows optimal engine management yet creates a less efficient power transmission path than a mechanical transmission. By placing a hydraulic pump/motor at each of the four wheels, independent wheel torque control is also possible [3)].

The focus of this paper is the hydro-mechanical drive train. This system aims to utilize the highly efficient power transfer of the parallel drive train and the flexible engine management of the series drive train. The hydro-mechanical drive train splits the engine power through two paths, mechanical and hydraulic, which are recombined before the wheels. There are multiple configurations for a hydro-mechanical drive train based on the number of planetary differentials and where they are located to split and recombine the power [4; 5)].

The majority of previous work presented in the literature involving hydro-mechanical drive trains did not include energy storage capability. One exception is work by Bowns, Vaughan, and Dorey performing computer simulations of a regenerative hydro-mechanical drive train for the application of a city bus [4)]. Their simulation predicted an increase in efficiency of just over 20% compared to a conventional drive train. However, it should be noted that their work did not include optimal engine management.

The purpose of this paper is to show the potential of a hydro-mechanical hybrid drive train for a passenger car by comparing this system to a series hydraulic hybrid through a simulation. Following a discussion of the novel innovations on the hydro-mechanical drive train architecture, drive train models will be developed from power flow equations. A rudimentary system operation and control strategy will be used in a simulation operating over the EPA Urban Dynamometer Driving Schedule. The simulation will demonstrate the potential of the system as well as development challenges. The paper will conclude with a discussion of the opportunities for future work on this hydraulic hybrid test bed.

HYDRO-MECHANICAL DRIVE TRAIN DEVELOPMENT

The hydro-mechanical system discussed in this paper, and shown in

Figure 1, consists of a hydraulic pump/motor mounted between the engine and a mechanical transmission. The output of the transmission and a second hydraulic pump/motor are combined in a planetary differential. The output of the differential powers the wheels. The two variable displacement pump/motors are connected to a high pressure accumulator for energy storage. The goal during operation is to transmit as much power as possible through the highly-efficient mechanical transmission, while using the hydraulics to allow the engine speed and vehicle velocity to be independent.

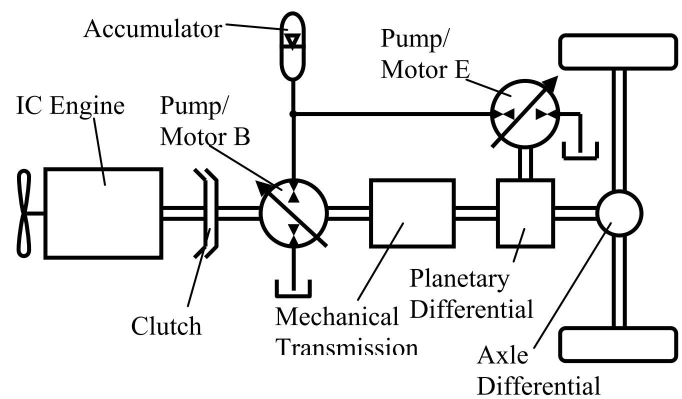

Figure 1. Hybrid hydro-mechanical drive train with a single planetary differential that recombines the split power after the mechanical transmission. Double lines represent shaft coupling while single bold lines represent hydraulic coupling.

An innovation of the hydro-mechanical drive train developed by the authors is a method to control the torque at each wheel independently. Independent control of the torque at each of the wheels opens avenues to an array of vehicle dynamic controls. In conventional drive trains, active vehicle dynamic controls, such as traction control, are accomplished by applying brakes to individual wheels. This method is inefficient and imprecise. As a solution to this issue, the authors developed a drive train that allows direct control of the torque at each wheel through varying the displacement of hydraulic pump/motors.

The hydro-mechanical drive train with independent wheel torque control is functionally similar to the standard hydro-mechanical drive train presented in Figure 1, yet the power split is recombined in planetary differentials for each wheel. Figure 2 shows the concept for independently controlling the torque at both rear wheels. The output of the mechanical transmission enters a right-angle gear set which contains two directly coupled outputs. These outputs enter planetary differentials which combine the mechanical branch with one of two hydraulic branches. While Figure 2 is a representation of a two-wheel-drive implementation for clarity, this drive train can also be implemented for all-wheel-drive by adding the front drive shaft, right-angle gear set, and differentials with pump/motors for each front wheel.

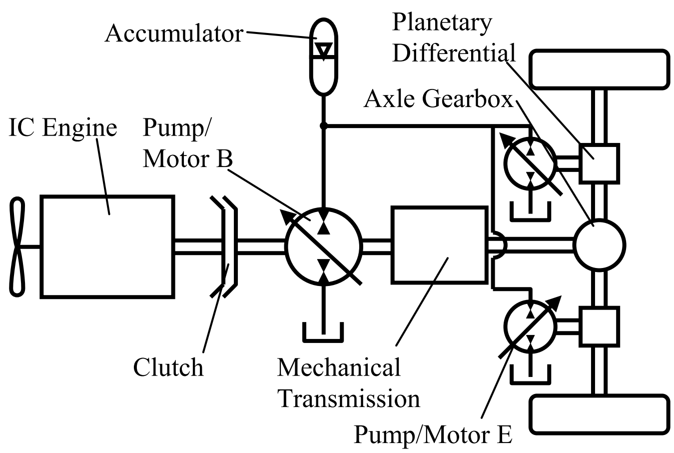

Figure 2. The novel hydro-mechanical drive train with independent wheel torque control. This figure demonstrates the concept for two-wheel drive. The drive train can also be implemented for all wheel drive with independent torque control of each wheel.

DRIVE TRAIN MODELING

To demonstrate the potential of the hydro-mechanical drive train, it will be compared to a series hydraulic hybrid architecture. This comparison will be made by modeling both drive trains and simulating their performance on the EPA Urban Dynamometer Driving Schedule. To build from simple to more complex, the series drive train model will be developed first, followed by the hydro-mechanical architecture.

SERIES DRIVE TRAIN MODEL - To aid in analysis of the series drive train, a diagram of the system is provided in Figure 3. The hydraulic series drive train consists of an internal combustion engine (A), a hydraulic pump (B), an accumulator (C), a pump/motor (D), and a differential (E) at the axle to transmit torque to each wheel. The figure is further annotated with numbers in balloons to describe power flow between components. The arrows associated with each balloon set a sign convention for the direction of positive power flow. For example, when pump/motor D is acting as a motor, power flow at 4 is positive; when it is acting as a pump, power flow at 4 is negative.

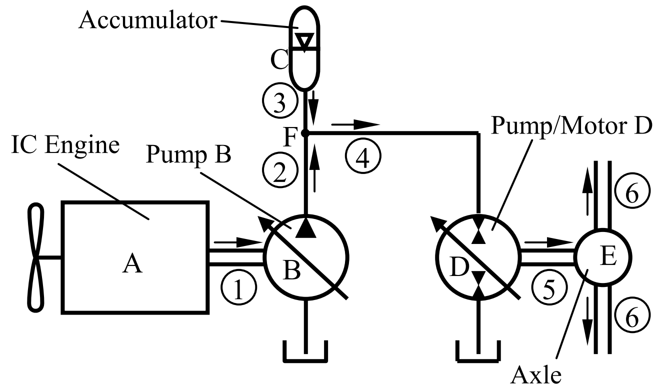

Figure 3. Series drive train. Double lines represent shaft coupling while single bold lines represent hydraulic coupling.

COMPONENT EFFICIENCY EQUATIONS - Equations will now be developed for the power flow through the drive train. Power flow from the engine and pump B is either positive or zero, as the engine is cycled on and off. For all of the other components in the drive train, the power flow can reverse during operation, requiring unique efficiency equations based on the direction of power flow. For example, pump/motor D either propels the vehicle when acting as a motor or slows the vehicle when acting as a pump. When acting as a motor, the power flow at 5 is equal to the power flow at 4 multiplied by the motoring efficiency of the unit. When acting as a pump, the power flow at 4 is equal to the power flow at 5 multiplied by the pumping efficiency of D. In the interest of brevity, the power flow equations for all components can be found in Table 1.

HYDRO-MECHANICAL DRIVE TRAIN MODEL - To aid in the analysis of the hydro-mechanical drive train, a diagram of the system is provided in Figure 4. The figure is annotated with numbers in balloons and arrows to define the positive power flow direction between components, which are labeled with letters.

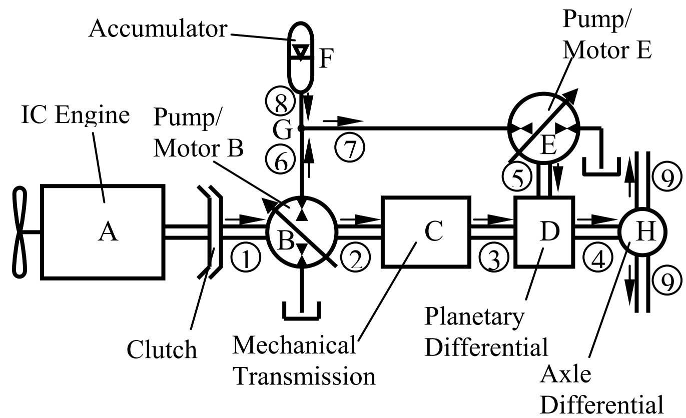

Figure 4. Diagram of hydro-mechanical drive train. Double lines represent shaft coupling while single bold lines represent hydraulic coupling.

Table 1. Power flow equations for the series hybrid drive train. The efficiency equations depend on the direction of power flow through the components, which is labeled in the table with the positive or negative port designation.

Component	Port		Mode	Power Equation
Pump B	1	2		
	+	+	Pumping	$P_2 = P_1 \eta_{B,pump}$
Accumulator C	3			
	+		Discharging	$E_C = E_{C,previous} - \dfrac{P_3}{\eta_C} \Delta t$
	-		Charging	$E_C = E_{C,previous} - P_3 \eta_C \Delta t$
Pump/Motor D	4	5		
	+	+	Motoring/Generating	$P_5 = P_4 \eta_{D,motor}$
	-	-	Pumping/Regenerating	$P_5 = \dfrac{P_4}{\eta_{D,pump}}$
Axle Differential E	5	6		
	+	+	Generating	$P_6 = P_5 \eta_E$
	-	-	Regenerating	$P_6 = \dfrac{P_5}{\eta_E}$

Symbols:
P_i – Power flow through into or out of each component where i refers to the balloons in Figure 3.
η_j – Power conversion efficiency of component j.
E_k – Energy contained in component k.
Pumping and motoring refer to the operating of the hydraulic units in sourcing or sinking flow respectively. Generating and regenerating refers to applying positive or negative power to the wheels. Finally, charging and discharging refers to adding or removing energy from the accumulator.

COMPONENT EFFICIENCY EQUATIONS - As a first step to modeling the hydro-mechanical drive train, equations for power flow through each component are developed. During a driving cycle involving acceleration and regeneration, the power flow through every component of the hydro-mechanical drive train, other than the engine, will reverse. Based on the direction of power flow through the various ports of each component, unique power flow equations are required. Power flow equations of each component in the non-independent wheel torque form of the hydro-mechanical drive train are presented in Table 2. Recognize that the analysis of the independent wheel torque control hydro-mechanical drive train is very similar.

The hydro-mechanical drive train enters recirculative modes of operation in which the power in the mechanical transmission is higher than the input or output power. Power recirculation can occur when the power flow direction in shafts 3 and 5, in reference to Figure 4, are opposite. One unavoidable instance of power recirculation occurs when the vehicle is near zero speed and requires a large power output to the wheels. If the engine is on and operating at a fixed angular velocity , pump/motor B acts as a motor and adds power to the drive train. Simultaneously pump/motor E rotates in a positive direction while supplying negative torque. Because the direction of the angular velocity and the torque are opposite for pump/motor E, it is acting a pump, sinking power from the engine. During these recirculative modes, the power through the transmission is higher than the power reaching the wheels. Due to the circulating power, the power loss due to component efficiency increases. A strategy to avoid power recirculation modes will be discussed below.

Table 2. Component power equations for the hydro-mechanical drive train. The application of the component efficiency in the power equation depends on the direction of power flow, as indicated in the table by the positive or negative port designation.

Component	Port			Mode	Power Equation
Pump/Motor B	1	2	6		
	+/0	+/-	+	Pumping	$P_6 = (P_1 - P_2)\eta_{B,pump}$
	+/0	+/-	-	Motoring	$P_6 = P_1 - \frac{P_2}{\eta_{B,motor}}$
Mechanical Transmission C	2	3			
	+	+		Generating	$P_3 = P_2\eta_C$
	-	-		Regenerating	$P_3 = \frac{P_2}{\eta_C}$
Planetary Differential D	3	4	5		
	+	+	+	Generating+Standard	$P_4 = (P_3 + P_5)\eta_D$
	+	+	-	Generating+Recirculating	$P_4 = P_3\eta_D + P_5$
	-	-	-	Regenerating+Standard	$P_4 = \frac{P_3 + P_5}{\eta_D}$
	-	-	+	Regenerating+Recirculating	$P_4 = \frac{P_3}{\eta_D} + P_5$
Pump/Motor E	5	7			
	+	+		Motoring	$P_7 = \frac{P_5}{\eta_{E,motor}}$
	-	-		Pumping	$P_7 = P_5\eta_{E,pump}$
Accumulator F	8				
	+			Discharging	$E_F = E_{F,previous} - \frac{P_8}{\eta_F}\Delta t$
	-			Charging	$E_F = E_{F,previous} - P_8\eta_F\Delta t$
Axle Differential H	4	9			
	+	+		Generating	$P_9 = P_4\eta_H$
	-	-		Regenerating	$P_9 = \frac{P_4}{\eta_H}$

Symbols:

P_i – Power flow through into or out of each component where i refers to the balloons in Figure 4.

η_j – Power conversion efficiency of component j.

E_k – Energy contained in component k.

Pumping and motoring refer to the operating of the hydraulic units in sourcing or sinking flow respectively. Generating and regenerating refers to applying positive or negative power to the wheels. Standard and recirculating refers to the operation mode of the hydro-mechanical drive train. Finally, charging and discharging refers to adding or removing energy from the accumulator.

SYSTEM OPERATION AND CONTROL STRATEGY

The control strategy for the series and hydro-mechanical drive trains is divided into three main sections: energy management, pump/motor operation and shifting management, and commanding the wheel torque of individual wheels based on controlling the vehicle dynamics. Because the focus of this paper is demonstrating the potential efficiency improvements of a hydro-mechanical drive train, the first two sections will be further developed, while the vehicle dynamics control will be left for future development.

ENERGY MANAGEMENT – A hybrid vehicle requires careful management of the kinetic and stored energy to minimize fuel consumption. The role of the energy storage device is to capture and release the kinetic energy of the vehicle, while the role of the engine is to provide the power to overcome the parasitic losses such as aerodynamic drag and rolling resistance. Furthering this concept reveals that the total energy of the vehicle, defined as the sum of the energy stored in the accumulator, the kinetic energy, and the gravitational potential energy, should remain approximately constant in an ideal hybrid operation.

Due to the energy storage and high power density of hydraulics, the engine operation is drastically different than a conventional drive train. The engine is not required to produce large amounts of power for rapid vehicle acceleration. Instead, the engine is sized to overcome the average parasitic losses and system inefficiencies at the maximum cruising velocity as well as the ability to climb a grade at a desired velocity. This means that the engine can be dramatically downsized from a conventional drive train without a negative impact on performance.

As previously discussed, to maximize the efficiency, the engine is only operated at or near the minimum brake specific fuel consumption. Operating at a fixed angular velocity and torque not only creates the most efficient conversion of fuel to shaft work, but also allows for other engine optimization options. Current automobile engines are required to operate across a wide range of speed and torque conditions and are not optimized for a specific operating condition. By designing an engine for a specific operating condition, the emissions, economy, vibrations, and acoustics can be optimized.

Because the engine is only operated at one torque and angular velocity condition, it must be cycled on and off according to the charge state of the accumulator. When the accumulator charge drops below a specified level, the engine is turned on. The engine continues to run, propelling the vehicle and/or storing energy in the accumulator, until the accumulator charge exceeds a specified high level, at which point the engine is turned off. The vehicle is then propelled solely by the accumulator until the charge drops below the low charge level, causing the engine to restart.

Using this control strategy, the engine is started and stopped quite often. A sensible method of restarting the engine is using the hydraulic system. By always keeping an adequate level of charge in the accumulator when the engine is off, the engine can always be started with the hydraulic unit coupled to the engine, eliminating the need for a conventional electric starter.

An exception to this engine on-off operation is avoiding modes of power recirculation in the hydro-mechanical drive train. This will be further discussed in the hydro-mechanical sub-section.

SERIES DRIVE TRAIN - The operation of the series drive train is relatively straight-forward due to the decoupling of the engine and the wheels through the purely hydraulic transmission. When the engine is on, pump B, in reference to Figure 3, is operated at the angular velocity and torque that result in the maximum engine efficiency. To operate at a constant torque and angular velocity while the output pressure is fluctuating, the displacement of the variable displacement pump B is varied.

The role of pump/motor D is to provide the desired torque at the wheels at the angular velocity dictated by the current vehicle velocity. This unit must provide high torque at low angular velocity when starting the vehicle from a stop and operate at high angular velocity during highway operation.

HYDRO-MECHANICAL DRIVE TRAIN - The hydro-mechanical drive train control strategy can be divided into engine on and engine off modes. For simplicity, this discussion will focus on the hydro-mechanical drive train without independent wheel torque control. However, the operation is the same for the independent wheel torque control system with the addition of having unique torque commands at each wheel based on the vehicle dynamics.

<u>ENGINE ON OPERATION –</u> As discussed above, the general engine control strategy is to cycle the engine on when the energy in the accumulator drops below a specific threshold. The threshold should ideally decrease with increasing engine speed to maintain approximately constant total vehicle energy. The engine control strategy is modified slightly for the hydro-mechanical drive train to avoid power recirculation modes. Power recirculation occurs when the power transmitted through the transmission is greater than the input or output power due to the two pump/motor units acting in opposition. Referring to Figure 4, the torque in shafts 4 and 5 of the differential is equal. The potential for recirculation only occurs when the angular velocity of shaft 5 is rotating the opposite direction of shaft 4. The angular velocity of the differential shafts is described by:

$$\omega_5 = 2\omega_3 - \omega_4 \tag{1}$$

where ω_i is the angular velocity of shaft i. Assuming that ω_4 is positive, meaning the vehicle is driving forwards, ω_5 must be greater than 0 to avoid recirculation:

$$0 > 2\omega_3 - \omega_4 \tag{2}$$

$$\omega_4 > 2\omega_3 . \tag{3}$$

The angular velocity ratio through the mechanical transmission is described by:

$$\omega_2 = R_C \omega_3 \tag{4}$$

where R_C is the reduction in the transmission. Substituting Eqn. 4 into Eqn. 3 yields the criteria for possible power recirculation:

$$\omega_2 < \frac{R_C}{2}\omega_4 . \quad (5)$$

The engine control strategy will be modified to minimize power loss in drive train components by decreasing the engine speed to maintain the relationship in Eqn. 5. There are two exceptions to this engine control strategy modification. First, the engine will not drop below idle speed, as otherwise dictated by Eqn. 5 as the vehicle approaches zero speed. Second, when the accumulator is below the low energy threshold and the power required at the wheels exceeds the engine power output, the engine speed will be raised to the standard operating speed to increase the output power.

Another primary goal of the hydro-mechanical control strategy is to maximize the ratio of the power transmitted through the mechanical branch to the hydraulic branch. In reference to Figure 4, the percentage of total power contributed by the mechanical branch versus the hydraulic branch can only be modified by changing the relative angular velocity of shafts 3 and 5. To minimize the power transmitted through the hydraulics, the gear ratio in the mechanical transmission C is selected to minimize the angular velocity of pump/motor E, while maintaining a positive value to minimize recirculation.

The gear ratio in the mechanical transmission sets the operating angular velocity of pump/motors B and E. The displacement of pump/motor E is calculated by using the required torque at the wheels and the current accumulator pressure. The displacement of pump/motor B is set by comparing the torque required at the input to the transmission with the torque output of the engine and the current accumulator pressure.

ENGINE OFF OPERATION - When the engine is cycled off, the system is reduced to an accumulator and two pump/motors connected to the drive wheels through a gear train. For engine off operation, only one of the pump/motors is operated while shaft of the other pump/motor is fixed. This removes the power lost to mechanical friction resulting from rotating an additional pump/motor.

SIMULATION

This section will present the simulations for the hydro-mechanical and series drive trains using a constant value for the efficiency of the pump/motors. From the simulation, the operating speed and fractional displacement of the pumps will then be discussed. These operating conditions will then be compared to a model of the pump efficiency as a function of the speed, fractional displacement, and pressure.

The simulation implements the above component equations and control strategy to complete a drive cycle. The computer simulation reads the drive cycle from a file, calculates the tractive force required at each time step, applies the control strategies to determine the drive train operation, and finally calculates the power flow based on the component efficiency equations. Through post-processing, the fuel economy is calculated and the initial energy content of the accumulator is compared to the final energy content to provide a valid fuel economy calculation.

Because the target application for the drive trains discussed in this paper is a passenger car operating in an urban environment, the EPA Urban Dynamometer Driving Schedule (UDDS) was selected as the input to the model. The UDDS, depicted in Figure 5, consists of a velocity profile as a function of time that covers 12 km in nearly 23 minutes. This driving schedule is designed to represent city driving conditions for a light duty vehicle [6].

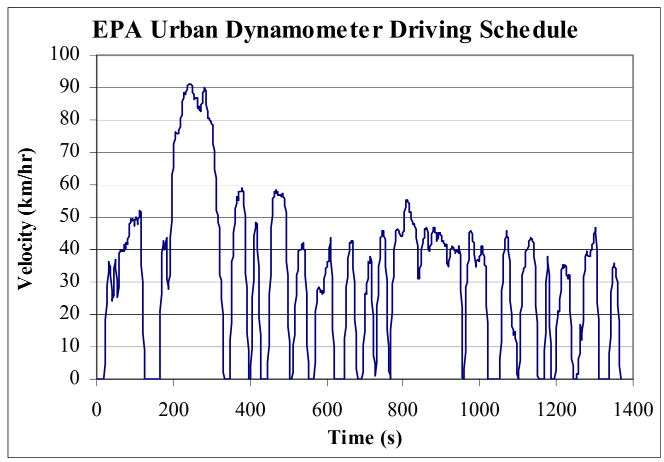

Figure 5. The EPA Urban Dynamometer Driving Schedule.

TRACTIVE FORCE - To simulate the vehicle operation across the drive cycle, the tractive force required at each time step is calculated using properties of the vehicle. The tractive force at the wheels is comprised of four components, the aerodynamic drag, rolling resistance, road grade, and the inertial forces of acceleration. The aerodynamic drag is calculated as follows:

$$D_A = \frac{1}{2}\rho V^2 C_D A \quad (6)$$

where D_A is the aerodynamic drag, ρ is the mass density of the air, V is the vehicle velocity, A is the frontal area, and C_D is the drag coefficient of the vehicle [7].

The rolling resistance of the entire vehicle is described by:

$$R_X = f_r W \quad (7)$$

where R_X is the rolling resistance, f_r is the rolling resistance coefficient, and W is the weight of the vehicle [7]. There are multiple mechanisms that contribute to the rolling resistance of the tires, requiring a precise rolling resistance model to include dynamics of the vehicle. For an approximate model, the following equation for the rolling resistance coefficient of tires on concrete is used:

$$f_r = f_o + 3.24 f_s \left(\frac{V}{44.7} \right)^{2.5} \tag{8}$$

where f_o is the basic rolling resistance coefficient and f_s is the speed effect rolling resistance coefficient [7)]. Note that the velocity is in units of meters per second in the above equation.

The final contribution to the tractive force is the inertial force. The drive cycle provides the vehicle velocity at discrete time steps. An approximation of the acceleration at each time step is calculated as follows:

$$A^t = \frac{V^{t+1} - V^t}{\Delta t} \tag{9}$$

where A^t is the average acceleration of the time step, V^{t+1} is the velocity at the next time step, V^t is the velocity at the current time step, and Δt is the time step. The inertial force is then calculated as follows:

$$F^t_{inertial} = mA^t \tag{10}$$

where $F^t_{inertial}$ is the inertial force for the time step and m is the mass of the vehicle. Finally, the total tractive force F^t_{total} is calculated:

$$F^t_{total} = D_A + R_X + F^t_{inertial}\,. \tag{11}$$

Note that the inertial force can be positive or negative based on whether the vehicle is accelerating or decelerating. An additional component of the total tractive force not accounted for above is the change in elevation due the grade of the road. The Urban Dynamometer Driving Schedule does not include any changes in road grade.

CONSTANT EFFICIENCY SIMULATION - The vehicle properties for the simulation emulate a four-door passenger car. The vehicle specifications and the primary assumptions of the model are presented in Table 3.

Table 3. Vehicle and model inputs for the constant pump/motor efficiency simulation.

Specification	**Value**
Vehicle Mass	1325 kg
Engine Type	Diesel
Engine Power Output	13.7 kW @ 2200 RPM
Engine Efficiency	32.1% thermal efficiency
Mechanical Transmission Efficiency	97%
Mechanical Differential Efficiency	97%
Drag Coefficient (C_D)	0.26
Frontal Area	2.16 m^2
Air Density	1.20 kg/m^3
Accumulator Volume	30 liters
Max Accumulator Pressure	35 MPa
Accumulator Expansion Ratio	2:1
Mechanical Transmission Ratios	2

For the simulation, the accumulator is assumed to operate isothermally. Thus the volume of oil and the pre-charge pressure determine the system operating pressure.

To demonstrate the potential of the hydro-mechanical system, constant values for the pump/motor efficiency were used in the simulation. While the efficiency of a pump/motor unit varies greatly with displacement, pressure, and speed, as will be further discussed below, a constant efficiency simulation develops understanding of the drive train operation. Figure 6 displays the fuel economy versus the constant pump motor efficiency for both the hydro-mechanical and series drive trains. The value of the constant efficiency is applied to both units when acting as both pumps and motors.

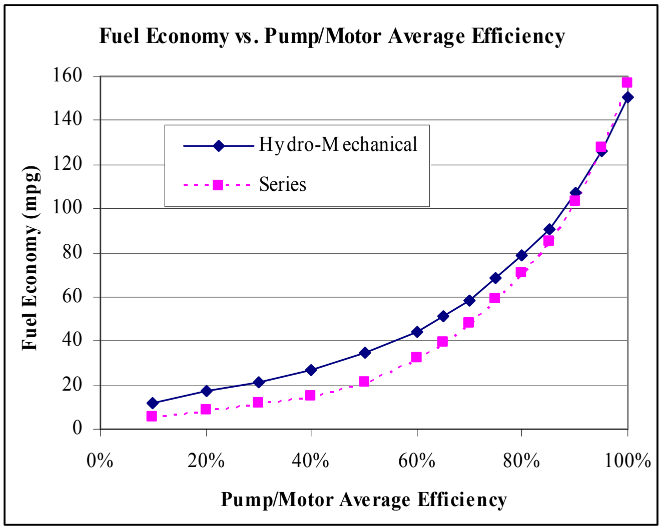

Figure 6. The fuel economy predicted by the simulation for the series and hydro-mechanical transmissions as a function of the average pump/motor efficiency.

Figure 6 shows that at pump/motor efficiency values greater than 93%, the series system achieves better fuel economy than the hydro-mechanical system. This effect is due to the additional efficiency losses through the mechanical components. However, when the value of the constant pump/motor efficiency decreases, the hydro-mechanical system demonstrates improved economy over the series system. The reason for this improved economy is that a significant portion of the power is transmitted through the higher efficiency mechanical branch.

Because the efficiency of hydraulic pump/motors is a function of their operating conditions, it is beneficial to know how the pump/motors in the drive train are functioning during the driving cycle. Figure 7 is a plot of the system pressure and fractional displacement of pump/motor B at every time step of the UDDS. Similarly, Figure 8 is a plot of the angular velocity and fraction displacement of pump/motor E at every time step of the UDDS. Note that the majority of operation is at fractional displacements below 20%. These low operating displacements result from sizing the pump/motors to meet the maximum acceleration and braking power requirements.

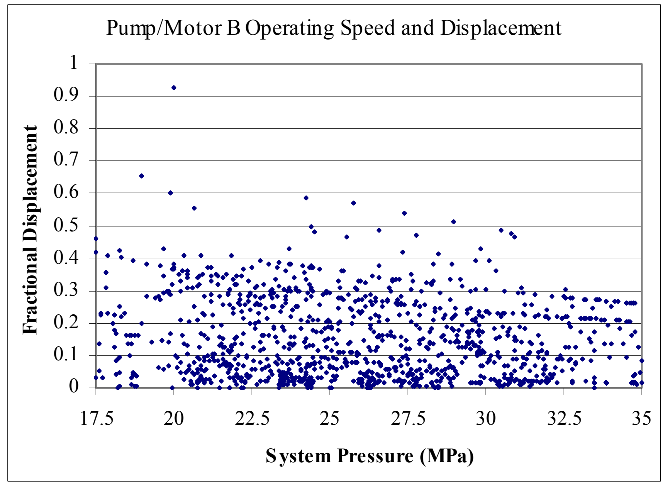

Figure 7. The system pressure and fractional displacement of pump/motor B at each time step in the UDDS.

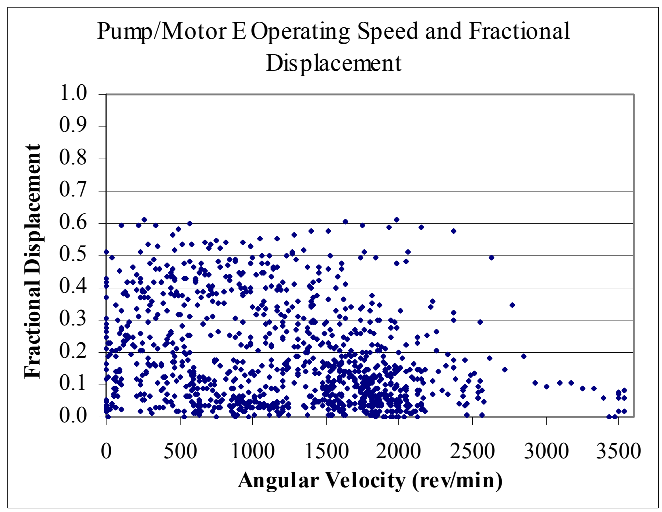

Figure 8. The angular velocity and fractional displacement of pump/motor E of the hydro-mechanical system during operation on the UDDS.

PUMP/MOTOR EFFICIENCY MODELS - For variable displacement hydraulic pumps and motors, efficiency is parameterized by displacement, angular velocity, operating pressure, and oil viscosity. The method used to model the volumetric flow rate and torque was first presented by Wilson [8)] as a linear model, and then further developed by McCandlish and Dorey to account for non-linearity in the data [9)]. The modeling technique described requires known volumetric flow and torque data, which is then mathematically fit to the following equations.

The model for volumetric flow through a variable displacement hydraulic pump unit is:

$$Q = x\omega D\left[1 - \frac{C_s}{x}\left(\frac{p}{\mu\omega}\right) - \left(\frac{p}{xB}\right)\left(V_r + \frac{1+x}{2}\right)\right] \tag{12}$$

The torque model for a variable displacement hydraulic pump unit is:

$$T = xpD\left[1 + \frac{C_v}{x}\left(\frac{\mu\omega}{p}\right) + \frac{C_f}{x}\right] \quad (13)$$

where:
p = system pressure
μ = viscosity of hydraulic fluid
ω = rotational speed of pump/motor
D = displacement of hydraulic unit
x = fraction of maximum displacement
B = Bulk modulus of hydraulic fluid
Cs = coefficient of slip
Cv= coefficient of viscous drag
Cf = coefficient of coulomb (dry) friction
Vr = volume ratio of hydraulic unit
Cs, Cv, Cf, Vr can be determined by a least squares fit of pump data

The first term in Eqn. 12 is the ideal flow rate through the unit, the second term in this equation accounts for the flow leakage through the pump, while the third term in the equation models the effect of fluid compressibility on flow rate. The first term in Eqn. 13 describes the ideal torque produced by the hydraulic unit, the second term models the effect of viscous drag on torque, and the third term represents torque loss due to coulomb friction.

A plot of the efficiency predicted by the model for a typical variable displacement axial piston pump can be found in Figure 9. This specific plot is at a pressure of 25 MPa, which is the approximate average operating pressure for the simulation presented above. In the figure, note the drastic decrease in overall efficiency with decreasing pump displacement. Because the power output of the pump at low displacements is small, the mechanical friction loss dominates the efficiency. This trend is troublesome for the hydro-mechanical drive train, which operates at low levels of displacement for the majority of the UDDS, as shown in Figures 7 and 8.

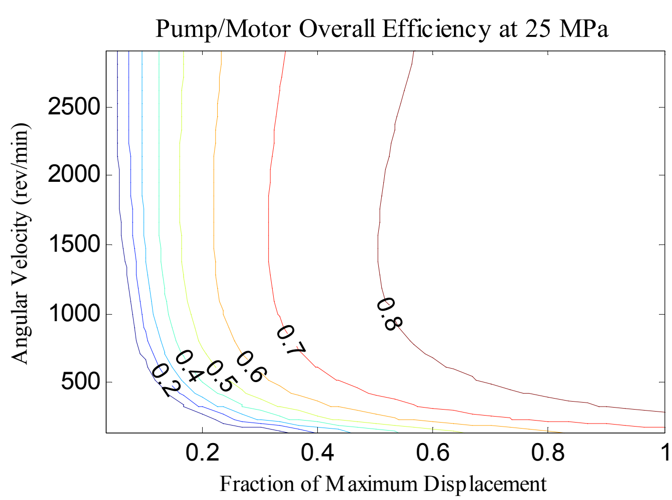

Figure 9. Overall efficiency of a typical variable displacement axial piston pump as a function of fraction of maximum displacement and angular velocity.

DISCUSSION/CONCLUSION

The hydro-mechanical hybrid drive train has the potential for excellent fuel efficiency combined with the ability to independently control torque at each wheel for vehicle dynamic control, yet numerous areas for further work exist. Two prime areas for future work are developing hydraulic pump/motor units that are efficient in the operating conditions seen in the hydro-mechanical system and developing the system control strategy to maximize the efficiency of the hydro-mechanical architecture and implement the independently control the torque at each wheel.

The simulation demonstrated that for average pump/motor efficiency values below 93%, the hydro-mechanical drive train outperforms the series architecture in fuel economy. However, due the strategy of transmitting the majority of the power through the mechanical branch, during routine driving, replicated by the UDDS, the pump/motor units operate at low displacements for the majority of the cycle. Current pump/motors demonstrate a significant decrease in efficiency at low fractional displacements due to the low power output and dominating mechanical coulomb friction. This mismatch between operating conditions and the efficiency profiles demonstrates a research need, for an organization such as the Center for Compact and Efficient Fluid Power (CCEFP), to develop hydraulic pump/motors with high efficiency down to low fractional displacements.

This paper has presented a rudimentary system control strategy that requires significant improvement. Using a formalized optimization of the system controls will create drastic improvements in the system performance. Furthermore, advances in predictive control, such as minimizing the energy stored in the accumulator when expecting a braking event, would be very beneficial to hybrid vehicles. Predictive control could be advanced further by integrating a GPS and communication system into the control computer that would allow the system to predict braking or acceleration events based on upcoming changes in road grade or traffic congestion.

The novel innovation presented in this paper that enables independent wheel torque control needs to be exploited for advances in the control of vehicle dynamics. While a good deal of research has previously explored these types of vehicle dynamic controls, they have primarily relied on applying braking systems to certain wheels, which is inherently inefficient. Furthermore, the high torque capability of hydraulics allows a much greater torque differential between wheels, even allowing anti-lock braking without friction brakes. This drive train architecture has the potential to spawn completely new vehicle dynamic control options.

Beyond advances related to optimizing and exploiting the presented hydro-mechanical drive train, a number of development areas could drastically improve the potential of hydraulic hybrid vehicles. One primary difficulty in implementing a hydraulic hybrid drive train in a passenger car is the large volume required for the conventional hydraulic accumulator. Improving the energy density of accumulators, with such concepts as the open accumulator [10), also currently being pursued by the CCEFP would greatly improve the adoption of this technology.

As discussed above, the efficiency of the hydraulic pump/motors has much room for improvement, especially when operating at low displacements. These improvements could come from optimizing existing designs, including the friction surfaces, or with new designs specifically aimed at high efficiency across a wide operating range. Another approach to improving efficiency is to implement digital valves to create virtually variable pump/motors from fixed displacement units [11)]. One area where digital valves would be especially beneficial is in operating a hydraulic unit in both directions and as a pump and a motor. With a typical hydraulic unit, this requires compromises in the valve plate design, which could be eliminated with digital valves.

In summary, the hydro-mechanical drive train with independent wheel torque control has the potential for excellent fuel economy in a passenger vehicle operated in an urban environment. The drive train combines a highly efficient power transmission through the mechanical branch and infinite speed variation through the hydraulic branch. Through modeling it was demonstrated that the pump/motor units typically operate at low displacements, creating poor efficiency in the current generation of pump/motors. This reveals a significant research and development project to develop units that operate efficiently in this regime. Through continued development and optimization, this drive train offers an attractive alternative to curb increasing energy consumption.

ACKNOWLEDGEMENT

This work was performed within the Engineering Research Center for Compact and Efficient Fluid Power (CCEFP), supported by the National Science Foundation under Grant No. EEC-0540834.

REFERENCES

1) "Department of Energy, Annual Energy Review 2003," *DOE/EIA-0384*, 2004, Energy Information Administration (EIA), Washington, D.C.
2) Krivts, I. L., and Krejnin, G. V., Pneumatic Actuating Systems for Automatic Equipment : Structure and Design, CRC/Taylor & Francis, Boca Raton 2006.
3) Fronczak, F. J., and Beachley, N. H. "An Integrated Hydraulic Drive Train System for Automobiles," Fluid Power, R. Heron, ed., Elsevier Applied Science, London, 1988, pp. 199-215.
4) Bowns, D. E., Vaughan, N. D., and Dorey, R. E., "Design Study of a Regenerative Hydrostatic Split Power Transmission for a City Bus," I Mech E Hydrostatic Transmissions for Vehicle Application, Coventry, Engl, 1981, pp 29-38.
5) Kress, J. H., "Hydrostatic Power-Splitting Transmissions Fo Wheeled Vehicles - Classification and Theory of Operation," Society of Automotive Engineers, No. 680549, 1968.
6) "Environmental Protection Agency Urban Dynamometer Driving Schedule," <http://www.epa.gov/nvfel/testing/dynamometer.htm>, August 9, 2007.
7) Gillespie, T. D., Fundamentals of Vehicle Dynamics, Society of Automotive Engineers, Inc., Warrendale, PA, 1992.
8) Wilson, W. E., "Rotary-Pump Theory," ASME Transactions, Vol. 68, 1946, pp 371-384.
9) McCandlish, D., and Dorey, R. E., "The Mathematical Modelling of Hydrostatic Pumps and Motors," Proceedings of the Institution of Mechanical Engineers, Part B, Vol. 198, No. 10, 1984, pp 165-174.
10) Li, P. Y., Van de Ven, J. D., and Sancken, C., "Open Accumulator Concept for Compact Fluid Power Energy Storage," Proceedings of the ASME International Mechanical Engineering Congress, Seattle, WA, 2007, pp 42580.
11) Tu, H., Rannow, M., Van de Ven, J., Wang, M., Li, P., and Chase, T., "High Speed Rotary Pulse Width Modulated on/Off Valve," Proceedings of the ASME International Mechanical Engineering Congress, Seattle, WA, 2007, pp 42559.