A Thesis

Entitled

Improving the Energy Density of Hydraulic Hybrid Vehicle (HHVs) and Evaluating Plug-In HHVs

By

Xianwu Zeng

Submitted as partial fulfillment of the requirements for

The Master of Science in Mechanical Engineering

Advisor: Dr. Mohammad H. Elahinia

College of Graduate Studies

The University of Toledo

May 2009

The University of Toledo

College of engineering

I HEREBY RECOMMEND THAT THE THESIS PREPARED UNDER MY

SUPERVISION BY: Xianwu Zeng

ENTITLED: Improving the Energy Density of Hydraulic Hybrid Vehicle (HHVs) and Evaluating Plug-In HHVs

BE ACCEPTED IN PARTIAL FULLFILLMENT OF THE REQUIREMENTS FOR

THE DEGREE OF Master of Science in Mechanical Engineering

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Thesis Advisor: Dr. Mohammad H. Elahinia

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An abstract of

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Hydraulic hybrid vehicle (HHV) is a new technology being developed in order to improve fuel economy for road vehicles. This technology also has limitations for example: low energy density, no power grid plug-in capability. This research is on the evaluation of a new concept for improving the HHV technology. With an added air system to HHV, the air system can be charged through grid plug-in or by the internal combustion engine (ICE). The new scheme has the potential to significantly improve the energy density of the hydraulic hybrid vehicles and also provide plug-in capability for these vehicles.

Basing on a symbolic program developed in MATLAB/Simulink, a parallel hybrid simulation model for the new system is developed in this thesis. The simulation model includes all the system components such as the vehicle, the air tank, the accumulators, the pressure exchangers, the hydraulic pump/motor, the compressor and the ICE. The power management is implemented based on using all the available

hydraulic power. The main objective of this model is to evaluate the average fuel economy (FE) for the HHV with the added compressed-air system. This model is tested basing on the federal urban drive schedule (FUDS). The simulations results with various configurations have not shown significant improvement in the fuel economy. This thesis provides a detailed analysis about the results from the system structure and the energy loss. In this system, there are two alternating accumulators. Every time the accumulator switches to reservoir, energy will be lost. When the engine drives the compressor to recharge the air system, a large engine would be needed to power such a compressor. These are the main reasons for the poor fuel economy of the proposed HHV system.

Acknowledgements

Firstly, I want to show my thanks to my advisor Dr. Elahinia and co-advisor Dr. Walter W. Olson. Dr. Elahinia gave me the opportunity to work with him on this project. He always gave me good suggestions and direction. They were very helpful for my master degree studies. I would like to thank Dr. Walter W. Olson who accordingly managed the Hydraulic Hybrid research group where I experienced team collaboration. I also would like to express my thanks to Dr. Maria R. Coleman for being in my committee for taking her time to review my thesis.

Secondly, I would like to thank Dr. Mark Schumark. He has very good experience on hydraulic hybrid vehicle system and MATLAB/Simulink program. He contributed his parallel hydraulic hybrid system model with detailed model specification to me. I can share his work. This is great help.

I also would like to thank my wonderful friends in Hydraulic Hybrid vehicle group and Dynamic and Smart Systems Laboratory with whom I spent my important time in the University of Toledo.

Finally, I would like to thank my wife and my daughter. They are my spiritual support. My wife's encouragement was a great motivation in my graduate studies. I haven't seen my daughter in about two years. Hopefully, she can remember what I look like.

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Nomenclature

a	Vehicle acceleration (m^2/s)
A	Frontal area of the vehicle (m^2)
c_v	Constant volume specific heat of the gas
C_D	Aerodynamic drag coefficient
D	P/M volumetric displacement (m^3)
D_e	Engine volumetric displacement (m^3)
EPA	Environmental protection agency
f	Rolling resistance coefficient
FE	Fuel economy (mile/gallon)
FE_{avg}	Average Fuel economy (mile/gallon)
HEV	Hybrid Electric Vehicle
HH	Hydraulic Hybrid
HHV	Hydraulic Hybrid Vehicle(s)
ICE	Internal combustion engine
k	Specific heat ratio
$\dot{m}_f$	Fuel mass flow (kg/s)
mpg	Mile(s) per gallon
M	Vehicle full loading mass (kg)
M_r	Equivalent mass of the rotating components in the vehicle (kg)
N_f	Gear ratios for the final drive (differential)
N_t	Gear ratios for the transmission
PHH	Parallel hydraulic hybrid
P/M	Pump/motor
Q	Volumetric flow rate through the pump/motor (m^3/s)
Q_{LHV}	Lower heating value of the fuel
r	Compression ratio

r_c Cutoff ratio.

R Specific gas constant [J/(kg.K)]

R_L Road load

SHH Series hydraulic hybrid

T Absolute temperature (K)

T_h Pump/motor torque

V Vehicle speed

$\forall$ Volume

$\forall_a$ Accumulator volume

$\forall_{N2}$ N_2 volume

$\forall_o$ Oil volume

$\forall_{PE}$ Pressure Exchanger volume

$\forall_t$ Air-tank volume

W Vehicle full loading weight

$\dot{W}_c$ Compressor required power

$\dot{W}_e$ Engine output power

$\dot{W}_{efric}$ Friction power produced by engine

$\dot{W}_{fric}$ Power dissipated by friction brakes

$\dot{W}_h$ Hydraulic power

$\dot{W}_{req}$ Vehicle required power at wheels

θ Grade angle of the road

ρ Air density

ρ_f Fuel density

η_c Compressor overall efficiency

η_{ce} Combustion efficiency

η_{ideal} Engine ideal efficiency

η_f	Differential efficiency
η_t	Transmission efficiency
ω_h	Pump/motor angular speed

Chapter 1

Introduction

1.1. Hybrid Vehicle

When a vehicle uses multiple propulsion systems to provide motive power, it's called a hybrid vehicle. Normally the vehicle is equipped with an internal combustion engine (ICE) using fossil fuels to generate power. The power is transmitted to the wheels via a mechanical drive train. When an additional power source is added to the traditional vehicle, it is termed a "hybrid". The additional power source can be electrical, chemical, hydraulic, fly wheel operated or any other form of power storage and production.

There are many reasons for hybridization. Recently, the main reasons are environment pollution and gasoline savings. In big cities the exhausts of the conventional vehicles pollute the air badly thus requiring an alternate source. In recent years, the gasoline price has been on the rise due to the fluctuation in the crude oil price. As a solution, hybridization is implemented to achieve better overall fuel economy of the vehicle and to reduce the demand for fossil fuels.

1.2. Hydraulic Hybrid Vehicle

A hybrid vehicle, in addition to its main engine, has a drivetrain that can recover and reuse energy. A hydraulic hybrid vehicle (HHV) has a transmission that can recover, store and reuse power hydraulically which is different with hybrid electric vehicle

(HEV). HHV stores energy in hydraulic accumulators and the drive unit is hydraulic pump/motor. HEV stores energy in batteries and/or ultra-capacitors and the drive unit is electric generator-motors.

Today, almost all commercially available hybrid passenger cars and light trucks are HEV. Over 90% of commercialization effort has focused on electric hybrid vehicles. Hydraulic hybrids have not received the visibility of electric hybrids. There is very little funding available for development of HHV. Nearly all R&D funding has been allocated to electric hybrids. EPA has pioneered the research efforts on HHVs in the US [1].

1.3. Statement of Problem and Objectives

As an energy storage device, a hydraulic accumulator has the ability to accept high rates and high frequencies of charging/discharging. It can supply very high density of power. Figure 1-1 shows different hybrids power density and energy density.

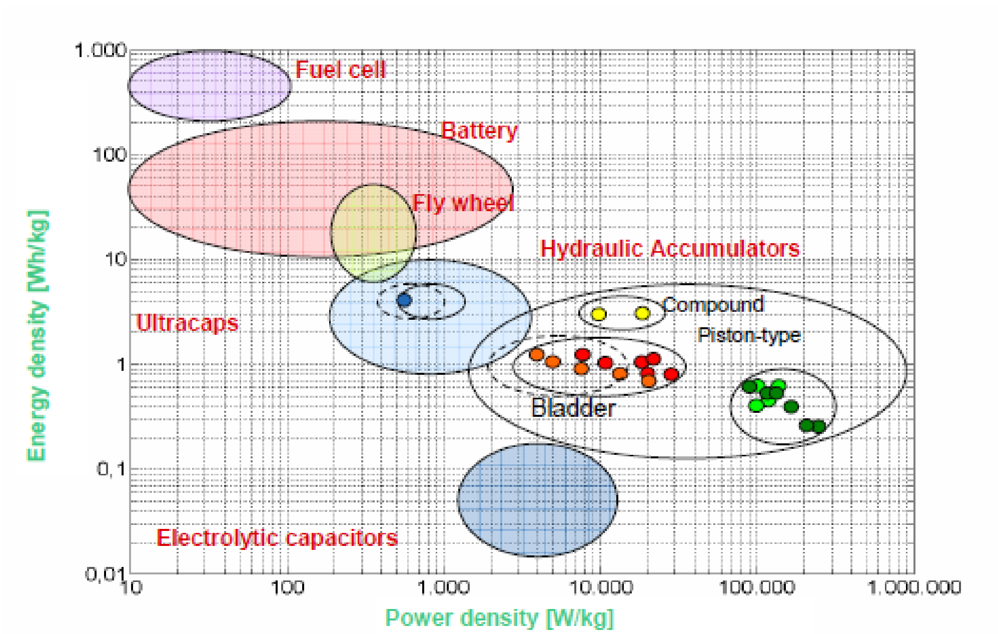


Figure 1-1 Energy Storage Device Comparison – Power vs. Energy [1]

The hydraulic hybrid system can easily capture the braking energy. Because of the high efficiency of the system components such as the accumulator and the pump/motor, the operational efficiencies can exceed 70% which is far better than any other form of hybridization [1, 2]. These properties are very suitable for the trucks on city drive condition. However, hydraulic hybrid technology has two limitations. Firstly, hydraulic hybrids have limited storage of energy which restricts the time of continuous operation and secondly, they currently lack grid plug-in capabilities. This study seeks to improve the hydraulic hybrid technology by addressing these two issues.

1.4. Approach

The following chapters of this thesis are to address the two limitations in the hydraulic hybrid system.

Chapter 2 presents the background of the research. It includes the contents as follows: a brief history of hybrid vehicles, hydraulic hybrid vehicle (HHV) benefits, parallel and series HHVs, and the energy management systems (EMS) of HHV.

Chapter 3 includes the proposed model and the control method. The detailed MATLAB/Simulink models including all the components in the proposed model are presented.

Chapter 4 is devoted to the simulation results and analysis. The results are compared with a conventional vehicle.

Chapter 5 is the conclusion of the thesis where the results are discussed and conclusions are made.

Chapter 2

Background

This chapter reviews the work on hybrid vehicles with an emphasis on hydraulic hybrids and also including some work on electric hybrids and compressed air vehicles.

2.1 Hybrid Vehicle

A definition of the entity hybrid vehicle is given by Michael Tamor at Ford Motor Company: “A Hybrid vehicle is a conventionally fueled and operated vehicle that has been equipped with a power train capable of implementing at least the first three of the following four hybrid functions:

1) Engine shutdown when power demand is zero or negative;

2) Engine down-size for improved thermal efficiency;

3) Regenerative braking for recovery and re-use of braking energy;

4) Engine-off propulsion at low power (when engine is inefficient).”[3]

Hybrid technology is not a new idea. It has the same long history as the cars. In early time the hybrids were gasoline-electric vehicles. However, at that time the hybrid was not for fuel savings and emission reducing. The function of electric motor was to help the engine to drive the vehicle faster. More recently, as gasoline prices started rising because of the diminishing oil supplies and concerns on environment and pollution increase, manufacturers have begun to seriously look into these systems again and hybrid cars are gaining popularity and acceptance [4,5].

Some of important events contributed to hybrid technology research are listed in the following lines in a chronological order [4]:

1).In order to reduce air pollution, in 1966, U.S. Congress passed first bills recommending use of electric vehicles [4].

2).The oil crises of 1973 and 1979, the oil embargoes and the Electric and Hybrid Vehicle Research, Development & Demonstration Act of 1976 made more and more companies develop hybrid vehicles [4].

3).In 1990, California adopted rules requiring car companies to sell certain percentage of “Zero Emission Vehicles" (ZEVs), which significantly influenced on the advancement of electrics and hybrids vehicle [4].

The commercial hybrid vehicles came forth in 1997. That year Toyota began offering a hybrid automobile, the ***Prius*** to the Japanese home market. In USA market, Honda introduced its two-seat ***Insight*** hybrid and Toyota provided the Prius in 1999. In 2002 Honda began offering the ***Honda Civic*** Hybrid. In 2004, Ford released the ***Escape*** Hybrid, the first SUV hybrid. Figure 2-1 shows a few of current hybrid vehicles. Until now, most of the main automakers follow the hybrid trend and popularity. As a result there are more options than ever including Sedan, SUVs, Trucks, Luxury vehicles, etc.

Toyota Prius 2007: MPG: 60 City/51 Hwy **Nissan Altima 2007 MPG: 42 city/36 Hwy**

Honda Civic 2007 MPG: 49 city/51 hwy **Ford Escape 2007 MPG: 36 city/31 Hwy**

Figure 2-1 Electric hybrid vehicles and mpg [6]

There are many kinds of hybrid technology. According to power source, there are gasoline-electric, gasoline-flywheel, gasoline-chemistry, gasoline-hydraulics, etc. Today in the market, most of the commercial hybrid vehicles are electric hybrids. According to the structure, there are parallel designs, series designs and a combination of both. Most of the new hybrid vehicles today are parallel Hybrid vehicles [7].

2.2 Hydraulic Hybrid Vehicle

Hydraulics (often called fluid power) offers the best solution for hybridizing heavier vehicles such as SUV's, trucks and buses to improve fuel economy. Large mass associated with heavier vehicles enables regenerating and reusing significant amounts of braking energy in hybrid powertrain configurations. Consequently, the power flows through the hybrid subsystem can be very high. This makes hydraulic propulsion and storage components very attractive for heavier vehicles applications, since they are characterized by much higher power density compared to their electric counterparts. As the energy storage device, a hydraulic accumulator has the ability to accept high rates and high frequencies of charging or discharging, both of which are not favorable for electric batteries. Normally, it needs one to several hours to recharge a high-energy-density battery.

Hydraulic hybrid system has many benefits [7]. These benefits are listed below:

- *Greatly improved fuel economy (FE).* In an electric hybrid system, if fast charging happens, the battery cannot be fully charged at a high rate, which further limits the range of the hybrid electric vehicles (HEV) by 10-20%. Normally, HEV have a wheel to wheel efficiency of less than 40%, hydraulic hybrids are able to produce efficiencies about 70% (Figure 2-2). One of the significant benefits of a HHV is its ability to capture a large percentage of the energy lost in vehicle braking. In frequent urban stop-and-go driving, as much as half of all the energy available at the vehicle wheels is lost in braking, while a parallel hydraulic hybrid design can capture and reuse up to 80% of this wasted energy. The specific fuel economy improvement associated with a HHV is dependent upon vehicle driving cycle, i.e., there will always be a larger improvement for those vehicles with a more frequent of stop-and-go driving cycle such as shuttle buses and delivery trucks.

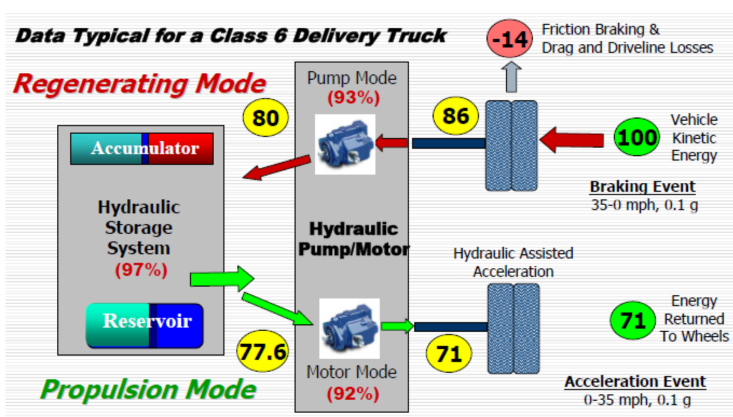


Figure 2-2 Analysis courtesy of Automotive Research Center – University of Michigan [8]

- *Less pollution*. Hydraulic hybrids offer more than 40% reduction in carbon dioxide emissions because of the lesser need to operate the engine.

- *Improved vehicle performance*. Hydraulic fluid power is easily delivered. In the truck applications, the accumulator will accept exceptionally high rates of charging and discharging. A combination of high efficiency and high charging/discharging rates enables effective energy capture and regeneration vehicles.

- *Lower incremental costs and less vehicle maintenance.* Hydraulic technology has been in industrial use for more than 100 years. All the hydraulic components are readily available. As a result, the acquisition costs are low.

There are also two types of hydraulic hybrid vehicles namely Parallel Hydraulic Hybrid (PHH) and Series Hydraulic Hybrid (SHH).

-***Parallel Hydraulic Hybrids (PHH)***: The PHH scheme is shown in Figure 2-3. The main components of this system are the accumulators which are the energy storage units, the pump/motor which is the drive unit and the reservoir. In this system, the engine and the hydraulic pump/motor are mechanically connected to the same drive shaft. This implies that the internal combustion engine (ICE) and the hydraulic pump/motor can power the vehicle individually or simultaneously. The work process of the vehicle is as follows:

When the vehicle is in braking, the hydraulic unit is in pump mode, the vehicle drive the pump to push the oil from the reservoir to the accumulator. This is the energy absorbing process.

When the vehicle is in acceleration, the hydraulic unit is in motor mode, the high pressure oil in the accumulator pushes the motor to drive the vehicle, if the power of the motor is not enough to drive the vehicle, the engine will make up the difference. This is the energy regeneration process.

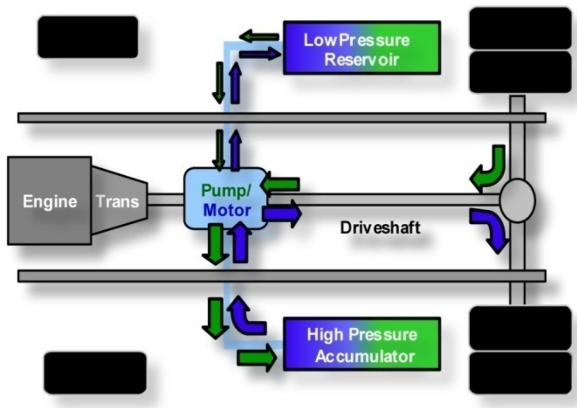


Figure 2-3: A parallel hydraulic hybrid configuration [7]

The PHH is effective in regenerative braking and capturing energy. This system has the capability, with conventional engine technology, to realize an improvement of fuel usage between 20% and 40% depending on vehicle usage. Because the pump/motor must follow the speed of the road which is geared up through the transmission, higher efficiencies cannot be achieved. The structure of this system is simple and it is very cost-effective [2].

- ***Series Hydraulic Hybrids (SHH)***: A more efficient system is SHH. The structure of SHH is shown in Figure 2-4. The main components are the accumulator, two pump/motor units and the reservoir (low pressure accumulator). The accumulator is the energy storage unit. Pump/motor 1 is attached to the ICE to refill the accumulator. Pump/motor 2 is connected to the drive shaft to drive or to slow down the vehicle. Since the ICE is decoupled from the drive shaft, higher efficiencies can be gained by operating the engine only at the torques and speeds necessary to achieve maximum efficiencies. The work process is as follows:

When the vehicle is in acceleration, pump/motor 2 is in motor mode. The accumulator pushes the motor to drive the vehicle. When stopping the vehicle, the pump/motor 2 is in pump mode, the vehicle drives the pump to push the oil from the reservoir to accumulator. Normally, the ICE is shut off or in idling. Only when the accumulator has no enough energy to drive the vehicle, the ICE drives the pump/motor 1 to charge the accumulator.

SHH is one of the most efficient powertrain technologies in the world. In laboratory tests, the city fuel economy of the hydraulic hybrid UPS vehicle has increased by 60% to 70% with respect to miles per gallon compared to a conventional UPS truck. The CO_2 emissions of the demonstration UPS vehicle are more than 40% lower than a comparable conventional UPS vehicle. The hydraulic hybrid vehicle also achieves approximately 50% lower hydrocarbon emissions and 60% lower particulate matter in laboratory tests. Hydraulic hybrids are able to capture and reuse 70-80% of the otherwise wasted braking energy [9].

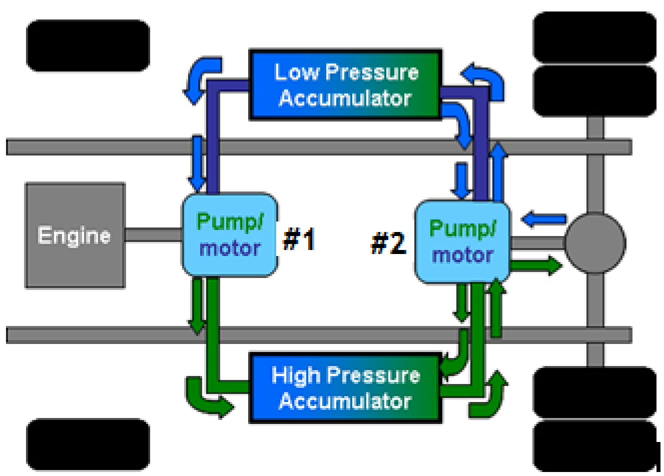


Figure 2-4: A series hydraulic hybrid configuration [7]

In summary, there are three key design features enabling an HHV to provide maximum fuel efficiency [40]:

-***Regenerative Braking***: When slowing down the vehicle, the vehicle drives the #2 unit, to pump fluid from the reservoir into the high pressure accumulator. When the vehicle starts accelerating, this stored energy is used to accelerate the vehicle. This process recovers and reuses over 70% of the energy normally wasted during braking.

-***Optimum Engine Control:*** In the SHH, the ICE is decoupled from the drive shaft. The vehicle speed has no relationship with ICE. The engine is free from the vehicle which means it can be operated at its maximum efficiency to achieve optimum vehicle fuel economy.

-***Shutting Engine Off When Not Needed***: The SHH structure enables the engine to be completely shut off during certain stages of operation when the hydraulic power is enough to drive the vehicle. As a result, in stop and go urban city driving engine use is cut almost in half.

The differences between the two HHV architectures are the following: the parallel system is simple, and easy to be realized, but the energy savings is limited; the series can get more energy savings, but is more complex.

All the HHV studies have focused on heavy-duty trucks and buses, as the heavy mass of these vehicles creates a large amount of energy which can potentially be reused. When this system is used on small vehicles, such as city delivery vans and taxis, the result is not so attractive. It has only 7-10% improvement in fuel efficiency [11].

2.3 Compressed air vehicle

In 2007, MDI and Tata supplied a "zero pollution" car to market. This is the first commercial compressed air car [43]. Compressed air vehicle is not a new idea. This technology uses high pressured air which comes from an air tank to run an air engine to drive the vehicle. There is no fuel, hence the name zero pollution. MDI has recently claimed that an air car will be able to travel 140km in urban driving , and have a range of 180 km with a top speed of 110km/h on highways, when operating on compressed air alone[44].

Figure 2-5: MDI's Compressed Air-Powered Car-MiniflowAir [44]

Figure 2-5 shows one of MDI's commercial compressed air car. The advantages and disadvantages of compressed air car are as follows [45]:

Advantages:

- Zero emission. (Air Engine).
- Low cost of operation. (Using electricity for filling air tank)
- Compressed air engines reduce the cost of vehicle production by about 20%, because there is no need to build a cooling system, spark plugs, starter motor, or mufflers. It also makes the vehicle very simple. Lighter vehicles would result in less wear on roads.

- Refueling can be done at home using a small air compressor or at service stations.
- Expansion of the compressed air lowers its temperature; this may be exploited for use as air conditioning.

Disadvantages:

- Limited storage capacity of the tanks. Compressed air is a heavy. As an example a 300 liters (11 cu ft) air tank at 30 Mpa (4,500 psi) contains about 16 kWh of energy (the equivalent of 1.7 liters gasoline). This technology therefore can only be used in small car.
- For compressed-air cars, energy is lost when electrical energy is converted to compressed air. (Compressor efficiency is very low).
- Safety problem. High pressured air tank (30Mpa) could be dangerous.
- When expands, the engine cools dramatically. It needs a heater to heat the engine. When refilling the air tank, the tank becomes very hot.
- Refueling the compressed air container using a home or low-end conventional air compressor may take as long as 4 hours.

2.4 Optimal Energy Management Strategy (EMS)

The automotive industry is facing increasing challenges of fuel consumption and emission pollution. In order to meet these challenges, it is essential to optimize the architecture and various devices and components of the vehicle drivetrain, at the same time, the energy management strategy (EMS) is important to efficiently control the energy flow through a vehicle system. The EMS is much more important for the hybrid structure, because this design with two power sources is more complex than conventional vehicle. In the HHV, hybridization raises the question of coordinating the operation of the primary power source (ICE) and the assistant power source (hydraulic motor) to maximize fuel economy while satisfying performance constraints [12]. The goal of energy management is to lower the fuel consumption by controlling the power flow in the hybrid vehicle.

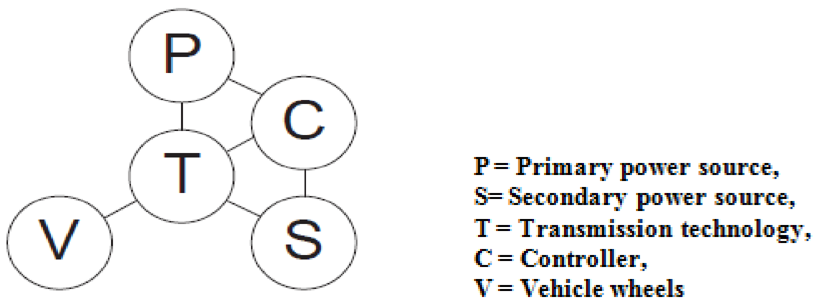


Figure 2-6: Schematic representation of a Hybrid Vehicle [14]

Hybrid systems are able to reach high fuel economy as well as low emissions without compromising drivability. As shown in Figure 2-6, a hybrid vehicle has a primary power source (P), which in most cases is ICE, and a secondary power source (S) e.g. the pump/motor. S connects to the system P with the help of a Transmission (T). The Controller (C) distributes the power flow between P, S and the Vehicle wheels (V). In the case of the HHV, the pump/motor retrieves its energy from a hydraulic accumulator and is able to assist the engine. When the car brakes, the pump works as a generator and the recovered energy is stored in the accumulator. This energy can be used later to assist the primary power source [13].

The function of the control system in hybrid vehicles is to regulate the power split between the two power resources and to maximize fuel economy. Normally the control system is used to operate the vehicle at peak efficiency. (See Figure 2-7).

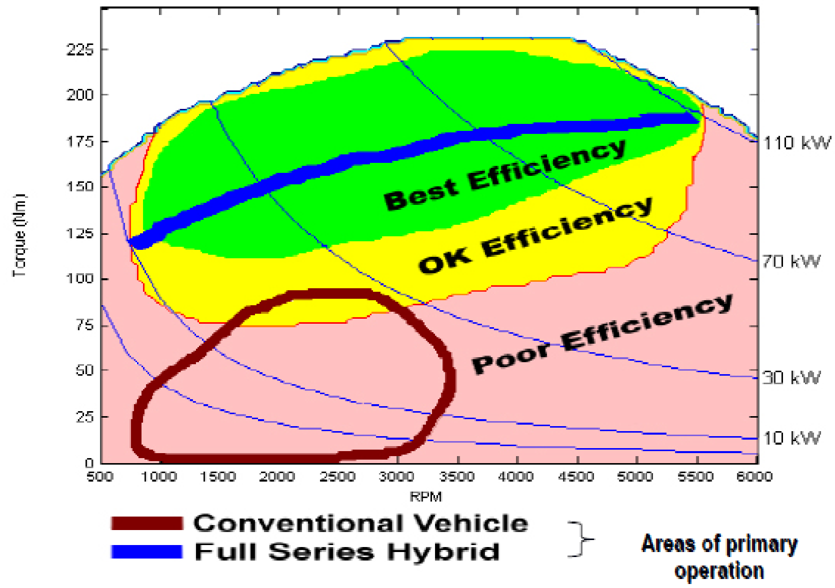


Figure 2-7: Power Map for a Typical Engine for Series Hybrids [8]

The hybrid vehicle power management system has been an active research area in the past two decades, and numerous energy management strategies have been developed to control the driving modes and define the power split for hybrid system. Most of these approaches were developed based on mathematical models, the knowledge gained from these simulations, or human expertise. The application of optimal control theory to power distribution and management has been the most popular approach. This includes linear programming, optimal control, and especially dynamic programming (DP). Optimal control methods have been widely studied and applied to a broad range of vehicle models [13, 14, 15, 16]. In recent years, various intelligence systems approaches such as neural networks, fuzzy logic, genetic algorithms, etc., have been applied to vehicle power management systems. Research has shown that driving condition has strong influence over fuel consumption and emissions [15].

For a parallel hydraulic hybrid system, two control systems will be presented: Rule-based power management strategy and Dynamic Programming (DP) algorithm [12,18]. These two power management strategies are also widely used on parallel electric hybrid vehicles [13, 14, 17].

-Rule-based power management strategy: Many existing hybrid electric vehicles' (HEV) power management algorithms are rule-based, because of the ease in handling switching operating modes [14]. Rule-based power management strategy is designed with a primary goal of shifting engine operating points to a more efficient region.

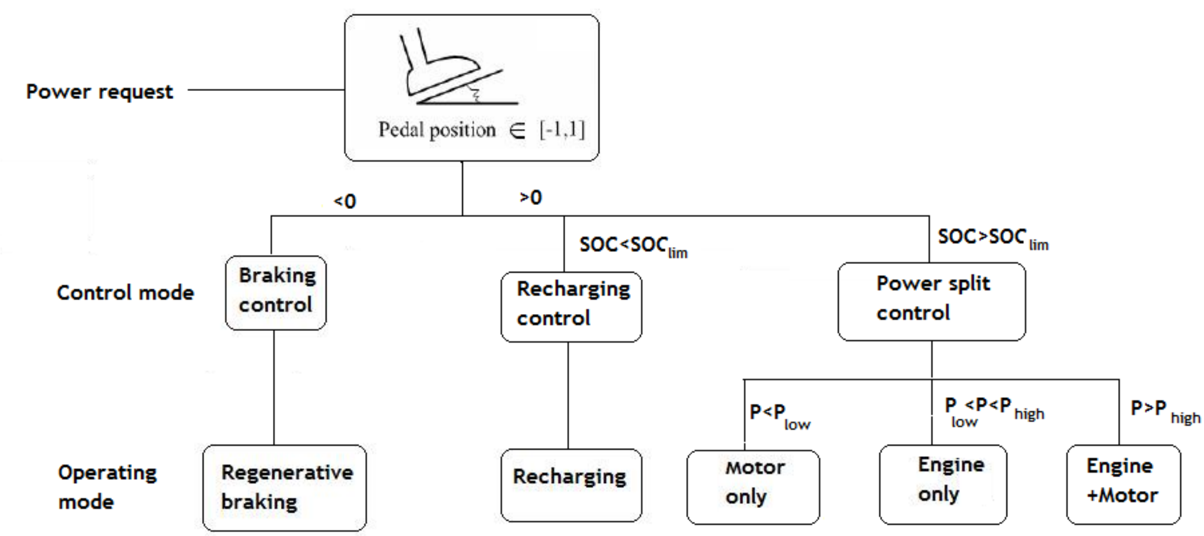


Figure 2-8: PHH rule-based control algorithm

Figure 2-8 shows the PHH rule-based control algorithm. In PHH vehicles, there are three control modes such as braking control, recharging control and power split control and five possible operating modes such as motor only, engine only, power assist (engine plus motor), recharging (engine charges the accumulator) and, regenerative braking (pump only). The control command comes from the driver's pedal motion as a power request. According to the power request and the vehicle status, the operation of the controller is determined the three control modes. If request power is negative, the Braking Control is applied to decelerate the vehicle. If request power is positive, either the Power Split or the Recharging Control will be applied, depending on the SOC of the accumulator. Thus, in PHH structure, the SOC is important to control the power distribution. When SOC drops below the lower limit, the controller will switch to the Recharging Control until the SOC reaches the upper limit, and then the Power Split Control will take over. The basic logic of each control rule is described below:

-*Power Split Control*: Figure.2-9 illustrates the basic power splitting idea on the engine Brake Specific Fuel Consumption (BSFC) map. The engine operating range is divided into three zones such as Motor Only, Engine Only and Power Assist with two constant power lines, the low power (P_{low}) and high power (P_{high}) limit, respectively. In the Engine Only zone, the engine has good efficiency. When the power demand is less than P_{low}, the hydraulic motor provides all the power, as long as there is energy available in the accumulator. When the needed power exceeds the P_{low}, the engine replaces the motor and provides all the power and the Motor stops. Once the power requirement exceeds the P_{high}, the hydraulic motor will supply the power deficit. In case the deficit is overreaches the motor available output, the engine will make up. The purpose of above strategy is to force the engine to operate in the more efficient region [12].

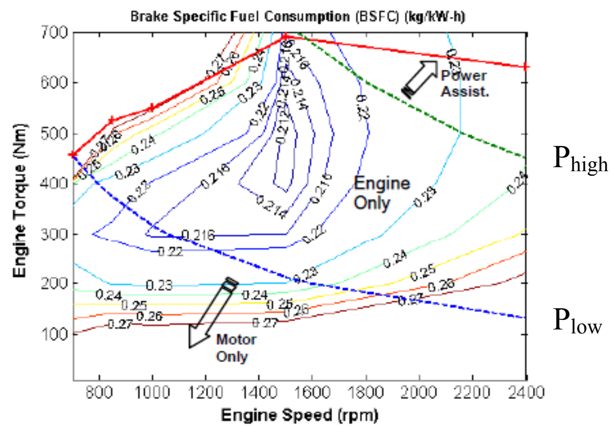


Figure 2-9: Power Split Control rule. [12, 14]

-*Recharging Control*: In the recharging control mode, the engine needs to provide additional power to charge the accumulator in addition to powering the vehicle [14].

-*Braking Control*: A simple regenerative braking strategy is used to capture as much regenerative braking energy as possible. If the accumulator is full, the pump will stop working, and friction brakes will assist the deceleration [14].

-Dynamic Programming (DP): Traditional rule-based algorithms are popular because they are easy to understand. However, when the control system is multi-variable and/or multi-objective, as the case in HHV control, it is difficult to come up with rules that capture all the important trade-offs among multiple performance variables. Optimization algorithms such as Dynamic Programming (DP) can help to understand the deficiency of the rules [15].

Dynamic Programming is a mathematical technique developed by R. Bellman in the 1950's. It is a powerful tool to solve general dynamic optimization problems. The main advantage is that it can easily handle the constraints and nonlinearity of the problem while obtaining a globally optimal solution. The DP technique is based on Bellman's Principle of Optimality, which states that the optimal policy can be obtained if we first solve a one stage sub-problem involving only the last stage and then gradually extend to sub-problems involving the last two stages, last three stages, …etc. until the entire problem is solved. In this manner, the overall dynamic optimization problem can be decomposed into a sequence of simpler minimization problems [14].

A DP algorithm is a multi-variable optimal control strategy used for HHVs. The objective is to search for optimal trajectories of control signals, u(k) , including engine command, hydraulic pump/motor command and gear shifting commands to minimize the fuel consumption of the HHV truck over the whole driving cycle, i.e.[12,13,14,17,18]:

$$\min J = \min_{u} \sum_{k=0}^{k=N-1} L(x(k),u(k)) + G \qquad (2\text{-}1)$$

$$G = \alpha(SOC(N) - SOC(0))^2$$

Where L is fuel consumption over a time segment, N is driving cycle length and x and u are the vectors of state variables and control signals respectively. In order to match the final value of accumulator SOC with its initial value, a penalty term G is added. After simplification, only two state variables (x) remain: the transmission gear number and the accumulator SOC. Based on Bellman's principle of optimality, the Dynamic Programming algorithm is presented as follows [14]:

$$J^*_{N-1}(x(N-1)) = \min_{u(N-1)}[L(x(N-1),u(N-1)) + G(x(N)) \qquad (2\text{-}2)$$

$$J^*_{k}(x(k)) = \min_{u(k)}[L(x(k),u(k)) + J^*_{k+1}(x(k+1)) \qquad (2\text{-}3)$$

After the recursive equation is solved backwards from step N -1 to 0, an optimal, time-varying, state-feedback control policy can be obtained. The resulting optimal control trajectory is then used as a state-feedback controller in the simulations to generate the fuel economy result. The optimal results based on the federal urban drive schedule (FUDS) is in Figure 2-10 [12,18]:

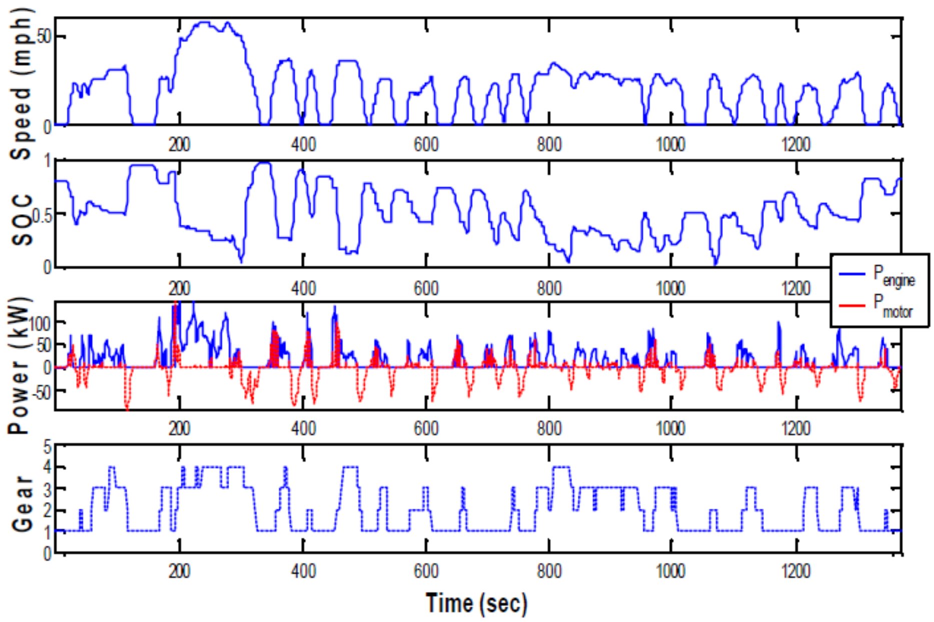


Fig. 2-10: Dynamic Programming results obtained over the FUDS driving schedule [12, 18].

DP cannot be used online because of two reasons. First of all, the trajectory has to be known beforehand and secondly, the computational time is too extensive for online calculation. A rule based control law has to be derived from the offline results obtained with DP which then can be used for online implementation. A great disadvantage is that the computational time scales are exponentially different. When more state or control variables are implemented or the grid size is being enlarged, the time needed to compute the DP problem grows explosively [14].However, the optimal control signal trajectories provide a benchmark for evaluating applicable strategies. By analyzing the DP results, we can get improved strategies that can be practically implemented.

In practice, the braking energy is "cost-free", the general splitting rule during braking is to use regenerative braking whenever possible, i.e. whenever the hydraulic pump can supply sufficient negative torque and the accumulator is not full. Friction brakes are activated when braking torque requirement exceeds what pump can provide. Thus the improved rule is as follows: When there is energy available in the accumulator, the controller will call upon the motor to satisfy the total power demand. If the power requirement is more than what motor can provide, the engine will make up the deficit. If the accumulator is empty the engine becomes the sole power source. These rules should capture the main features of the Dynamic Programming results in a very simple and easily implementable way [12].

Different vehicles with different road conditions can get different simulation results. Table 2-1 is the simulation result for an International 4700 series, Class VI 4x2 truck running on FUDS.

Table 2-1 Summary of Simulation Results [12, 18]

Configuration	Conven-tional	Initial Rule		Initial Rule + Improved Shifting		Improved Rule		Dynamic Programming	
P/M Efficiency	NA	High	Low	High	Low	High	Low	High	Low
mpg	10.39	13.75	12.01	14.08	12.40	15.32	13.28	18.37	14.34
mpg Improve.	NA	32.3%	15.6%	35.5%	19.3%	47.4%	27.8%	76.8%	38.0%
Regen. Energy (kJ)	NA	9748	9700	9652	9736	9459	9656	10013	10458
Reused Energy (kJ)	NA	6034	3187	5963	3229	7476	4524	8491	5134
Reused / Regen.	NA	61.9%	32.9%	61.8%	33.2%	79.0%	46.9%	84.8%	49.1%

High-High Efficiency Pump/Motor   Low-Low Efficiency Pump/Motor

Results: 1). Improved rule can get FE improvements of 47.4% (High) and 27.8% (Low).

2). Initial Rule can get FE improvements of 32.3% (High) and 12.01% (Low).

3). Total regenerated energy captured during braking reaches similar levels in all cases, except for DP calculations. However, reused energy varies significantly with both efficiency of components (High vs. Low) and power management. Reused energy ratio is from 61.9% to 79%.

Regeneration of braking energy is a major factor in improving fuel economy of hydraulic hybrid trucks. Hydraulic components can accomplish this very effectively, provided that power management system is optimized to take full advantage of their characteristics [12].

Series hydraulic hybrid system power management strategy: In the SHH system the engine is decoupled from the wheels. This places strong emphasis on the power management strategy. The main task of the power management system is maximizing the fuel economy. This can be accomplished through effective regenerative braking and optimization of engine operation. In this system the engine easily can be made to operate "sweet spot", the point in the eye of the brake specific fuel consumption (BSFC) map with lowest specific fuel consumption (Figure 2-11) [19].

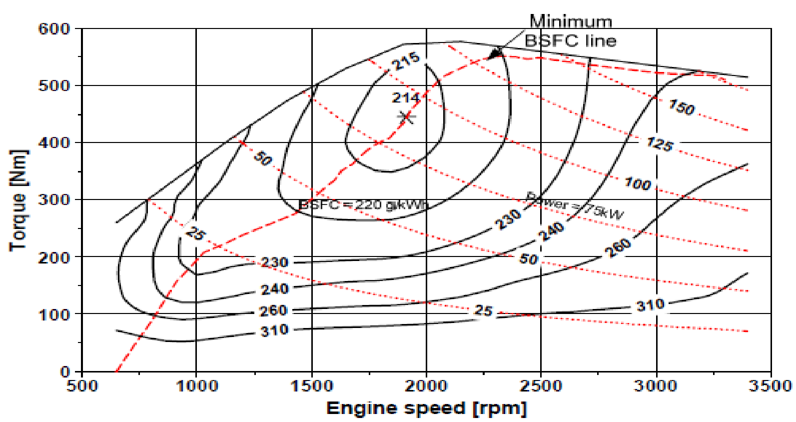


Figure 2-11 The engine BSFC map with a best BSFC trajectory (dashed line) [19]

Young Jae Kim and Zoran Filipi [19] gave a power management strategy based on the threshold SOC. The basic four parameters are the threshold SOC, the threshold power, the dead band and the SOC at which power command becomes 100%. This strategy does not involve the engine "sweet spot", because the fuel economy of the vehicle depends on the whole system efficiency, not only on one component efficiency.

The basics idea of the thermostatic SOC control is shown in Figure 2-12.

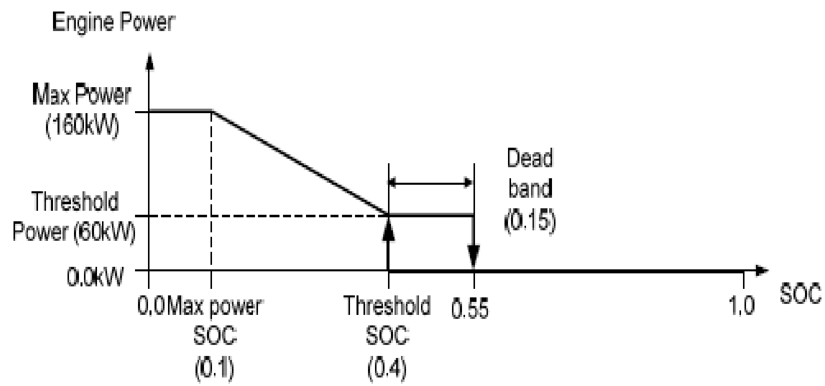


Figure 2-12 Schematic illustrating the thermostatic SOC power management concept [19].

As long as the SOC is above the threshold SOC (0.4), the engine shut down (or idling). When SOC drops to the threshold value, the engine starts charging the accumulator with the threshold power (60kW), until SOC reaches the upper threshold (0.55). The dead band (0.15) is for preventing frequent engine on-offs. If the power required for propulsion exceeds the threshold level, the SOC will drop below the lower limit and the engine power will be progressively increased. The lower threshold SOC can supply enough spare capacity for energy regeneration when stopping vehicle. However, it should be high enough to provide quite a number of energy when needed [19].

After using a simulation to study all the parameters respectively, a midsize truck simulation result can be obtained. Figure 2-13 and Table 2-2 show the simulation results.

Figure 2-13 shows the threshold SOC control algorithm can force the engine to operate in the more efficient area. Notice, the best fuel efficiency is not in the "sweet spot" (90kw@1800rpm) [19].

Table 2-2 shows [19]:

1).The engine can be downsized from V8 to V6.

2).Based on FUDS, with engine idling, the HHV fuel economy can get 49.2% improvement, with engine shut-down, the HHV fuel economy can get 68.2% improvement compared with conventional vehicle.

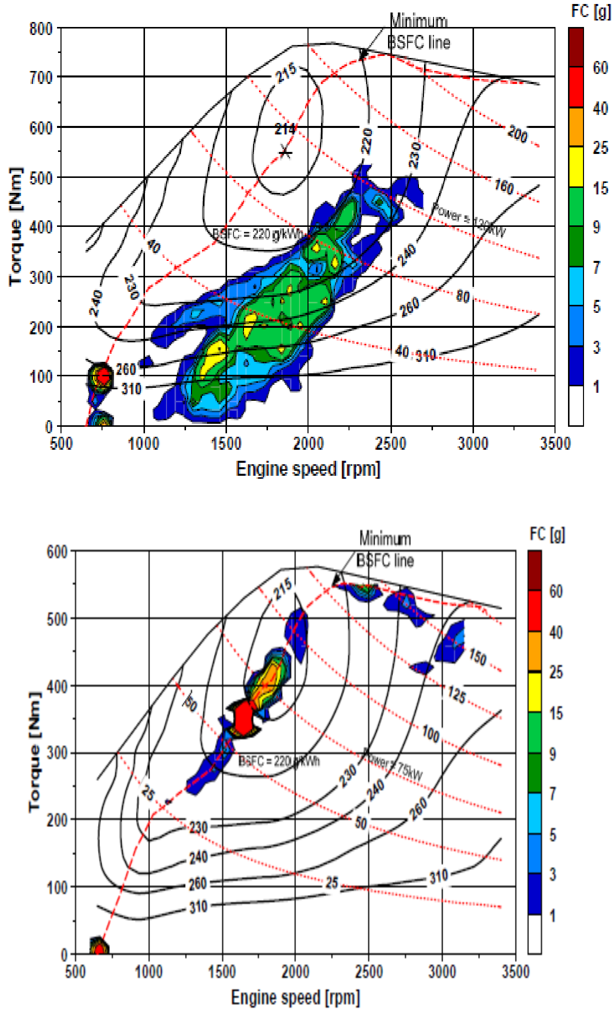


Figure 2-13 Engine visitation points on the BSFC map. Comparison of engine operation in: a) a conventional vehicle (V8 engine), and b) a series hydraulic hybrid (V6 engine). Color scale indicates the relative amount of fuel consumed at a given zone during FUDS. [19].

4x4 Vehicle, Conventional Vehicle's Total Weight 5112 kg		Conventional vehicle with a 4-speed automatic V8 6L Diesel	Series Hydraulic Hybrid, V6 4.5L Diesel
		F.E. [MPG]	Fuel Economy. improvement [%]
With Engine Idling	City Driving (FUDS)	12.4	49.2
	Highway Driving (HWFET)	14.3	11.2
With Engine Shut-downs	City Driving (FUDS)	-	68.2
	Highway Driving (HWFET)	-	12.5
0 – 50 MPH (80.5 km/h) Acceleration Time[sec]		10.8	10.3

Table 2-2 Summary of the Series Hydraulic Hybrid Vehicle predicted performance and fuel economy improvements over the conventional baseline [19]

Chapter 3
Modeling

This chapter introduces the proposed concept for the hydraulic hybrid system in detail and presents the MATLAB/Simulink models for each of the components of the system. The MATLAB/Simulink program is based on M. Schumack's work [20].

3.1 Proposed Concept Structure

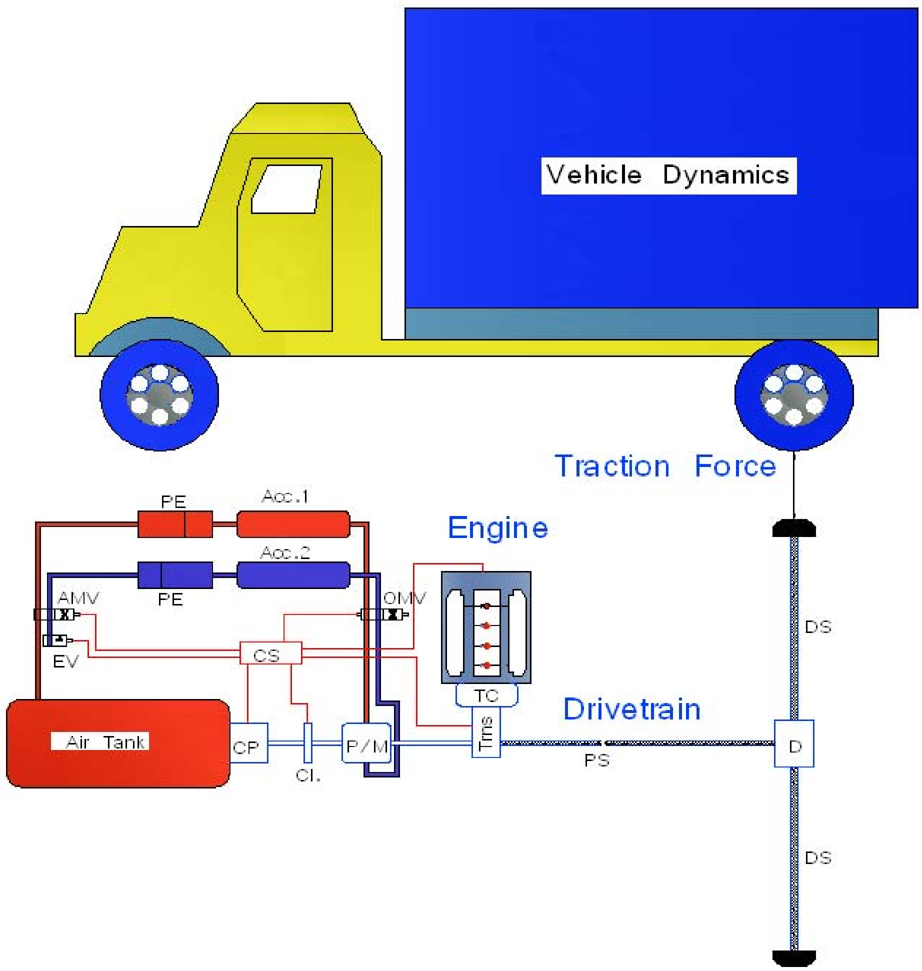
The hydraulic hybrid system has two limitations:

1) Hydraulic hybrids have limited storage of energy which restricts the time of continuous operation. For example, if the vehicle is braking for a long time, the hydraulic accumulator which absorbs the braking energy will become full of high pressure oil. There is no more room for capturing the braking energy. Once the charge of oil has been moved from the high pressure accumulator to the low pressure side, the motor must be used to serve as a pump to recharge the system. In other words, when the high pressure accumulator is discharged, there is no energy to be used, the hydraulic motor cannot continue operating. In this condition, the demand is transferred to the main vehicle ICE.

2) Hydraulic hybrids currently lack grid plug-in capabilities. In electric hybrids, the energy storage batteries can be fully recharged by grid plug-in when the vehicle is not in use. This means the system have two kinds of energy: electric power and gasoline. But for current hydraulic hybrids, the accumulator only works as braking energy storage. All

the energy (including the braking energy) comes from engine. This means there is no other power source.

The objective of the proposed concept structure, as shown in Figure 3-1, is to address the two limitations.



PE-Pressure Exchanger
Acc.-Accumulator
AMV-Air Mode Valve
OMV-oil Mode Valve
EV-Exhaust Valve
CS-Control System
CL-Clutch

CP-Compressor
P/M-Hydraulic Pump/Motor
TC-Torque Converter
Trns-Transmission
PS-Propeller Shaft
D-Differential
DS-Driveshaft

Figure 3-1 The proposed concept hydraulic hybrid system and the integrated vehicle structure

The main components of the proposed concept structure include two accumulators, two pressure exchangers, the hydraulic pump/motor, the air tank, the compressor and the control valves.

The specification of this system (Figure 3-1) is as follows: The high pressure air is fed to a pressure exchanger that transfers the pressure to the nitrogen gas in the accumulator. The purpose for this exchange is to prevent air entrained with water and other airborne contaminants from forming and reaching in the accumulators. The Air Mode Valve controls which accumulator receives the high pressure air. In Figure 3-1, the Accumulator 1 is acting as the high pressure accumulator and is driving the hydraulic pump/motor through the Oil Mode Valve. The Accumulator 2 acts as the low pressure accumulator accepting the spent oil from the motor. Accumulator 2 is connected to a second pressure exchanger that is currently exhausting air back to the atmosphere. When all oil in high pressure accumulator (Accumulator 1) goes to low pressure accumulator (Accumulator 2), the system will switch: air mode valve and oil mode valve switch at the same time. After switch, the air tank fills the Accumulator 2 and Accumulator 2 becomes the high pressure accumulator to drive the pump/motor, and the Accumulator 1 becomes the low pressure accumulator to receive the spent oil. The air tank is high pressure air storage. It can be filled by a compressor. The compressor is fitted with an electric motor. It can be powered by the electric motor from the electric grid or by the ICE. When the vehicle is out of service the air tank can be fully filled by the electric motor. If the vehicle is running, the air pressure in the air tank is too low, the clutch will be engaged and the engine will charge the air tank.

The air tank can save much high pressured air which can supply energy to the accumulator. This system can potentially improve the energy density of the hydraulic hybrid vehicles. There are several additional benefits which accrue from this system. This system has the same ability to capture the braking energy as the HHV system. The air tank can be recharged to full capacity while the vehicle is not in use. This would be done with an onboard electric air compressor that is plugged into a conventional electric outlet while the vehicle is parked and not in use. This provides the benefit of using an energy that is typically less polluting. In addition, the recharging of the accumulators from the air compression system results in the formation of very cold air (-15^0C). This air can be bused to cool the hydraulic oil and the hydraulic motor and eliminate or reduce the need for supplemental oil cooler.

The proposed concept for the hydraulic hybrid vehicles is to elongate the service time, improve energy (density) and provide the plug-in feature. The proposed HHV scheme detailed operation control can be found in Figure 3-2. The pressure exchangers and the accumulators are piston type. When all the oil is in Accumulator 1 (Acc1), valves #1, #2, #5, #6 are open, and valves #3, #4 are closed, the high pressure air from air tank fills the pressure exchanger 1 (PE1) and pushes Nitrogen (N_2) to Acc1. Immediately all N_2 goes into Acc1, valves #1, #2 are closed, valve #3 is open, the high pressured air in PE1 vents. Until all the oil goes to Accumulator 2 (Acc2) from Acc1, a switch occurs: valves #2 , #4, #5 are open, valve #6 is closed, Air tank fills PE2, when all N_2 goes to Acc2, valves #5, #4 are closed, valve #6 is open, the Acc2 is at high pressure and Acc1 is at low pressure. At each switch the high pressure air in pressure exchanger will be lost. When the pressure in the air tank becomes too low, the compressor will recharge it. When switch occurs, the direction control valve also switches to maintain the P/M rotating direction.

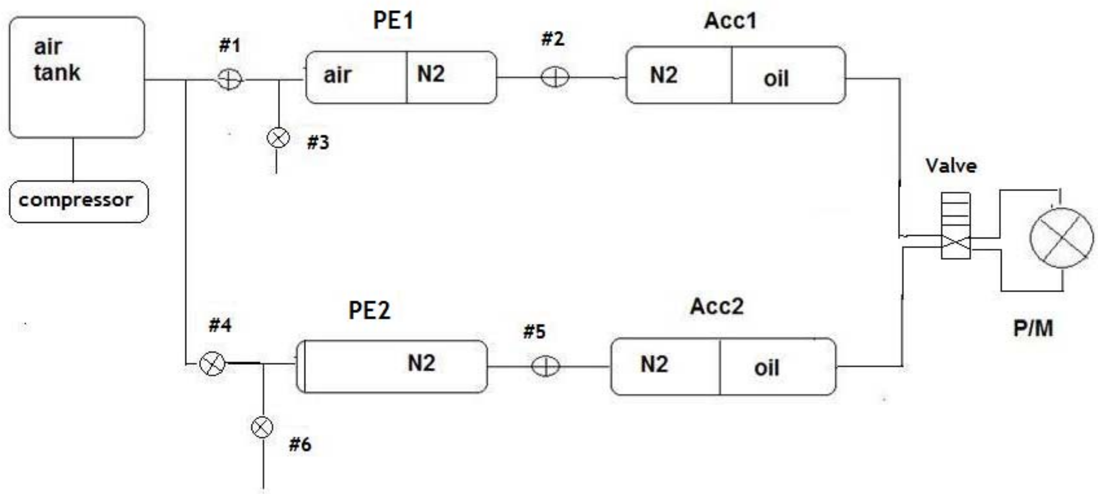


Figure 3-2: The Operation Control of the proposed system

3.2 Vehicle System

An International 4700 series, Class VI 4x2 truck is selected as the baseline for this work. In cities, medium-size trucks like the 4700 series are frequently used for delivery which includes frequent stops-and-goes. [12]. The conventional truck has a mass of 7340 kg when fully loaded. When the proposed HHVs is added in this truck, the mass will change as follows:

1). Average air mass: 1000kg (2m^3,50Mpa)

2). Air-tank mass (Carbon Fiber): 200kg (2m^3):

3). Exchanger mass (Carbon Fiber): 100x2=200kg.

4).Accumulator (Carbon Fiber): 100x2=200kg

5).Pump/Motor: 100kg

6).compressor: 100kg

Total added mass is about 1800kg; Total added volume of this system is about $2.4m^3$. Total vehicle mass: M=7340+1800=9140kg.

The vehicle system includes the hydraulic system, engine, wheels/tires, axles, suspensions and body of the vehicle, as shown in Figure 3-1. The engine is connected to the torque converter (TC), whose output shaft is then coupled to the transmission (Trns). The coupling at the transmission output side engages the Propeller Shaft (PS). The hydraulic pump/motor also engages the Propeller Shaft. Thus, the transmission and pump/motor can be linked to the propeller shaft (PS), differential (D) and two drive shafts (DS), coupling the differential with the driven wheels. The engine and the pump/motor all connect to the PS. This structure is a parallel hydraulic hybrid system. The main components on this structure are the vehicle body, the air tank, the pressure exchangers, the accumulators, the compressor, the hydraulic pump/motor model and the engine.

3.3 Components Modeling

3.3.1 Vehicle Dynamics Model:

Vehicle dynamics describes the motion of the selected rigid bodies (wheels, axles and body), that are allowed to move in space with respect to forces/moments and rigid constraints. The forces/moments are physical elements, which act at a specific point on the two bodies. In addition, two bodies can be restrained to move only in specific directions by a rigid constraint, e.g., a wheel is allowed only to rotate around the axis of the axle. The connection of the vehicle with the driveline is at the driven wheels where the drive-axle is connected to the rim of the wheels [21].

A number of approaches can be used to model vehicle dynamics depending on the overall simulation objectives. In this project, the model assumes that the vehicle mass is lumped at the center of wheelbase. Such an approach can give sufficiently accurate predictions of vehicle acceleration and speed on "smooth" or flat roads. If needed, other conditions can also be added to the model such as road roughness, steering, braking, etc.

The studies in this work consider only the acceleration of the vehicle on a flat and straight road (grade angle of the road, θ=0), where the excitation does not generate significant pitch motion. Therefore, the point mass model adequately predicts the interactions between the powertrain and vehicle dynamics. For simplification, the side force, lift force, pitching moment, yawing moment and rolling moment are all ignored.

The vehicle dynamics model [20] is used to calculate the required power of a vehicle. Power required at the wheels:

$$\dot{W}_{req} = \left[R_L + \left(M + M_r\right)a\right]V \qquad (3\text{-}1)$$

$\dot{W}_{req}$ is the power required at the wheels to accelerate the vehicle and overcome drag, rolling resistance, and climbing forces. The vehicle speed is V and the acceleration is a. The road load is:

$$R_L = \frac{1}{2}\rho V^2 C_D A + fW + W\sin\theta \qquad [22] \qquad (3\text{-}2)$$

Where the first part is aerodynamic drag, the second part is the rolling resistance force and the third part is the climbing force.

M is vehicle full loading mass and the effective mass. The equivalent mass of the rotating components M_r can be obtained from the following equation:

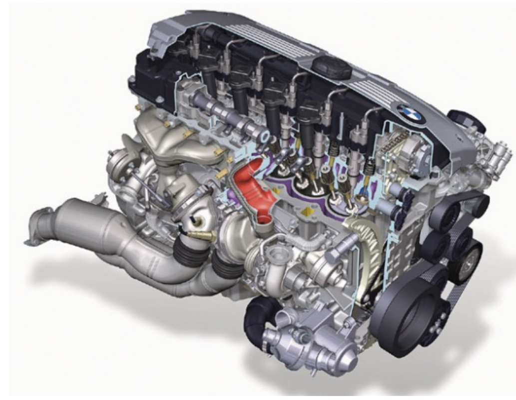
$$M_r = M(1 + 0.04 N_t N_f + 0.0025 N_t^2 N_f^2) - M \quad [22] \qquad (3\text{-}3)$$

N_f and N_t are the gear ratios for the final drive (differential) and transmission. (The added mass term associated with rotating hydraulic components and compressor components is neglected; this assumption is reasonable because the expression for effective mass is conservative). If the vehicle is being powered by the hydraulic motor only (or absorbing power through the pump), $N_t = 0$. Since the vehicle power is largely governed by the acceleration loads in the urban drive cycle, the simulation is particularly sensitive to the equivalent mass: $M + M_r$.

Appendix 5 shows the vehicle dynamics MATLAB/Simulink model.

3.3.2 Internal Combustion Engine Model

The International 4700 series, Class VI, 4x2 delivery truck is powered by a V8 turbocharged, intercooled, 7.3L diesel engine with rated power of 157 kW@2400 rpm. Although parallel hybrids offer the opportunity for engine downsizing, it is not adopted here. Because in this proposed concept system there is the condition that the engine runs the compressor to recharge the air tank and also run the vehicle. In this state, the engine will supply more power than a conventional vehicle. Figure 3-3 shows two type of diesel engines.



GM Diesel Engine 6.5L [23] **BMW Diesel Engine [24]**

Figure 3-3: Diesel engine

Dennis Assanis and Zoran Filipi [21] developed a high fidelity, thermodynamic diesel engine model for conventional truck. The high fidelity engine model was comprised of multiple cylinder modules linked with external component modules for manifolds, compressors and turbines, heat exchangers, air filters, and exhaust system elements. In this project, the engine module is not complex, composed of a thermodynamics model for calculating the efficiency, a friction model to calculate the friction power and a torque model to calculate the torque. The purpose of the engine model is to calculate the fuel consumption and fuel economy [20].

Fuel consumption:

The mass flow rate of fuel to the engine is determined from [20]:

$$\dot{m}_f = \frac{\dot{W}_e + \dot{W}_{efric}}{\eta \eta_{ce} Q_{LHV}} \quad \text{(kg/s)} \tag{3-4}$$

Where $\dot{W}_e$ is the engine output power, $\dot{W}_{efric}$ is the friction power produced by the movement components inside the engine. η is the engine thermal efficiency, η_{ce} is the combustion efficiency, and Q_{LHV} is the lower heating value of the diesel fuel. In order to obtain the fuel mass flow rate in kg/s, $\dot{W}_e$, $\dot{W}_{efric}$ must be in watts and Q_{LHV} must be in J/kg.

$$\dot{W}_{efric} = \frac{f(rpm)D_e N}{2} \text{ (w)} \quad [20] \qquad (3\text{-}5)$$

Where D_e (m^3) is volumetric displacement (per revolution) of the engine, N is the engine angular speed in rev/s. The empirical quantity $f(rpm)$ accounts for engine friction, accessory power, and engine pumping losses. For diesel engines, the quantity $f(rpm)$ can be expressed as [25]:

$$f(rpm) = C_1 + 48\left(\frac{rpm}{1000}\right) + 0.4\bar{S}_p^2 \text{ (kPa)} \qquad (3\text{-}6)$$

Where C_1 is a constant in kPa and $\bar{S}_p$ is the mean piston speed in m/s. The mean piston speed is obtained from:

$$\bar{S}_p = 2LN \quad \text{(m/s)} \qquad (3\text{-}7)$$

Where L is the stroke (m), and N is the engine angular speed in rev/s. The unit for $f(rpm)$ is kPa.

The thermal efficiency (η) calculation:

The thermal efficiency is $\eta = 0.87\eta_{ideal}$ [26], where η_{ideal} is the ideal thermal efficiency. For diesel engines, the ideal efficiency is calculated as follows [27]:

The diesel ICE differs from the gasoline powered Otto cycle by using a higher compression of the fuel to ignite the fuel rather than using a spark plug ("compression ignition" rather than "spark ignition"). In the diesel engine, air is compressed adiabatically with a compression ratio typically between 15 and 20. This compression raises the temperature to the ignition temperature of the fuel mixture which is formed by injecting fuel once the air is compressed. The ideal air-standard cycle (Figure 3-4) is modeled as a reversible adiabatic compression (a—b) followed by a constant pressure combustion process (b—c), then an adiabatic expansion as a power stroke (c—d) and an isovolumetric (constant volume process) exhaust (d—a). A new air charge is taken in at the end of the exhaust, as indicated by the processes a-e-a on the diagram.

Since the compression and power strokes of this idealized cycle are adiabatic, the efficiency can be calculated from the constant pressure and constant volume processes. In this process, the efficiency can be described [27]:

$$\eta_{ideal} = 1 - \frac{1}{r^{k-1}}\left(\frac{r_c^k - 1}{k(r_c - 1)}\right) \qquad (3\text{-}8)$$

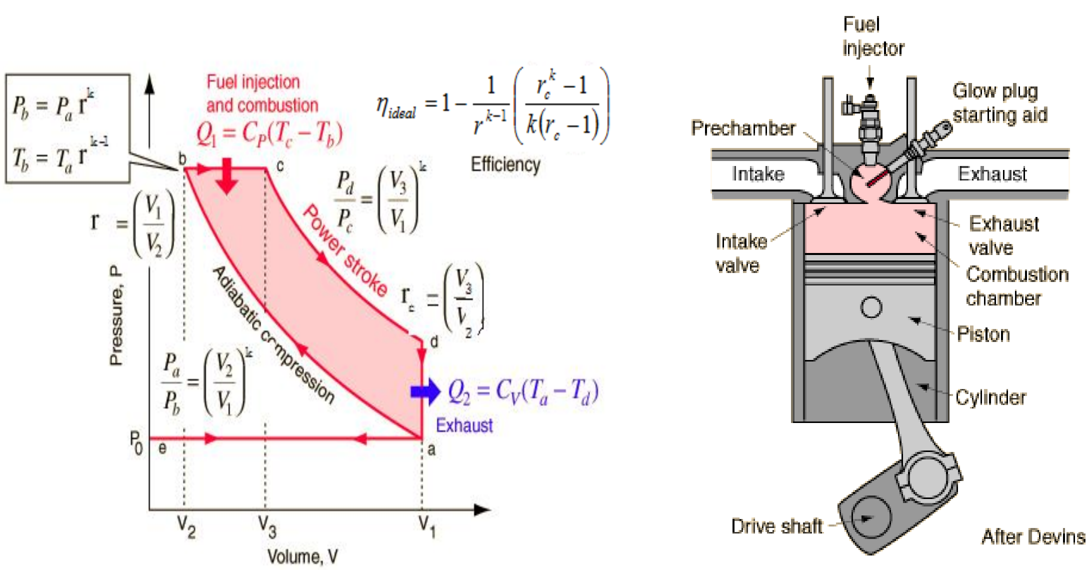


Figure 3-4: Air standard diesel engine cycle [28]

Here r is the compression ratio: $r = V_{\max} / V_{\min} = V_1 / V_2$

r_c the cutoff ratio: $r_c = \dfrac{volume\ at\ end\ of\ heat\ addition}{volume\ at\ start\ of\ heat\ addition} = \dfrac{V_3}{V_2}$

k is the specific heat ratio: $k = C_P / C_V$, Once the fuel mass flow rate is determined, the instantaneous fuel economy is determined from:

$$FE = \frac{V}{\dot{m}_f / \rho_f} \quad \text{(mpg)} \qquad (3\text{-}9)$$

Where ρ_f is the density of the fuel. The average fuel economy for the trip ($0 \le t \le tf$) is determined from [20]

$$FE_{avg} = \frac{\int_0^{t_f} V dt}{\frac{1}{\rho_f}\int_0^{t_f} \dot{m}_f dt} \quad \text{(mpg)} \qquad (3\text{-}10)$$

When the engine power $\dot{W}_e$ is zero (which is when the vehicle is not moving, the motor is handling the full load, or when the vehicle is braking), the engine angular speed is set to an idling speed of 750rpm. From the equation for $\dot{m}_f$, it can be seen that the engine still consumes fuel (due to total friction loads) even when $\dot{W}_e$ is zero.

Engine power and torque curve calculation:

In order to get the torque-speed graph and power-speed graph, we assume the engine torque (T) is a polynomial function of engine angle speed (ω):

$$T = C1 + C2 * \omega + C3 * \omega^2 \qquad (3\text{-}11)$$

C1, C2, C3 are coefficients. The routine puts a parabola through the max torque and power points, with zero slope at the max torque point. So,

$$\begin{cases} C1 + C2 * \omega_{T\max} + C3 * \omega_{T\max}^2 = T_{\max} \\ C1 + C2 * \omega_{p\max} + C3 * \omega_{p\max}^2 = \dot{W}_{\max} / \omega_{p\max} \\ C2 + 2C3 * \omega_{T\max} = 0 \end{cases} \tag{3-12}$$

in matrix format:

$$\begin{bmatrix} 1 & \omega_{T\max} & \omega_{T\max}^2 \\ 1 & \omega_{p\max} & \omega_{p\max}^2 \\ 0 & 1 & 2\omega_{T\max} \end{bmatrix} \begin{pmatrix} C1 \\ C2 \\ C2 \end{pmatrix} = \begin{pmatrix} T_{\max} \\ \dot{W}_{\max} / \omega_{p\max} \\ 0 \end{pmatrix} \Rightarrow \begin{pmatrix} C1 \\ C2 \\ C2 \end{pmatrix} = \begin{bmatrix} 1 & \omega_{T\max} & \omega_{T\max}^2 \\ 1 & \omega_{p\max} & \omega_{p\max}^2 \\ 0 & 1 & 2\omega_{T\max} \end{bmatrix}^{-1} \begin{pmatrix} T_{\max} \\ \dot{W}_{\max} / \omega_{p\max} \\ 0 \end{pmatrix} \tag{3-13}$$

For the 7.3 L diesel engine: 210HP@2400rpm, 520lb-ft@1500rpm [29]:

$$\begin{cases} T_{\max} = 705Nm \\ \dot{W}_{\max} = 157kw \\ \omega_{T\max} = 1500rpm = 157rad/s \\ \omega_{p\max} = 2400rpm = 251rad/s \end{cases} \tag{3-14}$$

C1,C2, C3 can be calculated.

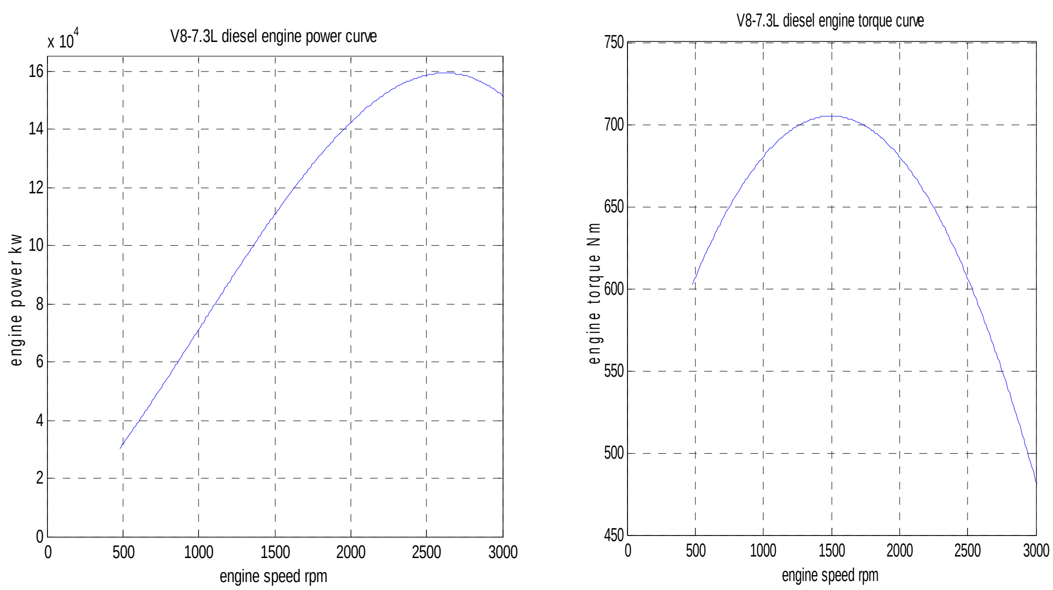


Figure 3-5: Engine power and torque curve

Figure 3-5 shows the engine power and torque respect to the engine speed. In the Simulink model, at each time step , the program can calculate the limiting torque with this equation and compare it to the actual required torque. The actual torque T_e :

$$T_e = \frac{\dot{W}_e}{\omega_e} \quad \text{(Nm)} \qquad (3\text{-}15)$$

Here ω_e is the engine angular speed:

$$\omega_e = N_f N_t \omega_w \qquad (3\text{-}16)$$

ω_w is the wheel angular speed:

$$\omega_w = \frac{V}{r_w} \qquad (3\text{-}17)$$

r_w is the wheel radius.

Appendix 11 shows the ICE MATLAB/Simulink model.

3.3.3 Hydraulic Pump/Motor (P/M) Model

Hydraulic pump/motor (P/M) units are two directional energy conversion devices. In the pump mode, the hydraulic P/M converts the kinetic energy from vehicle braking motion into hydraulic energy stored in the high pressure accumulator. In the motor mode, the hydraulic P/M converts this hydraulic energy into kinetic energy to assist vehicle acceleration. The hydraulic pump/motor is an axial, variable displacement design. The piston travel and displacement are varied by changing the swash plate angle. The modeling approach follows fundamentals analyzed by Pourmovahed et al. [30,31]. Modeling starts with the equation for the ideal volumetric flow rate through a pump or motor [32].

The pump/motor power is:

$$\dot{W}_h = \mathrm{T}_h \omega_h \text{ (w)} \qquad (3\text{-}18)$$

Where ω_h is the P/M angular speed.

$$\omega_h = N_f N_h \omega_w \qquad (3\text{-}19)$$

T_h is the P/M torque.

$$\mathrm{T}_h = \Delta p D \quad \text{(Nm)} \qquad (3\text{-}20)$$

Where Δp is the pressure difference across the pump/motor (P/M),

$$\Delta P = P_{high} - P_{low} \qquad (3\text{-}21)$$

P_{high} is the pressure in the pressure accumulator.

P_{low} is the pressure in the low pressure accumulator (the reservoir).

D is the pump/motor displacement (m3). It is in the range $(-D_{\max} \sim D_{\max})$. $D_{\max}$ is the maximum displacement of the pump/motor.

The volumetric flow rate (Q) through the pump/motor is:

$$Q = \omega_h D \quad (\mathrm{m^3/s}) \qquad (3\text{-}22)$$

The differences between the real volumetric flow (Q_{act}) and real torque (T_{act}) and ideal quantities calculated above are accounted for by the volumetric and torque efficiencies. The volumetric efficiencies (η_v) and torque efficiency (η_T) of the P/M are defined by the following equations [30]:

$$\eta_v = \frac{Q_{act}}{Q} \qquad \eta_T = \frac{T_{act}}{T} \qquad (3\text{-}23)$$

Pump versus motor:

$$\eta_{v_{motor}} = \frac{1}{2-\eta_{v_{pump}}} \quad \eta_{T_{motor}} = \frac{1}{2-\eta_{T_{pump}}} \qquad (3\text{-}24)$$

In this model, the displacement D can be positive (motor model) and negative (pump model). Thus, T_h and Q can be positive and negative. Figure 3-6 shows the P/M unit.

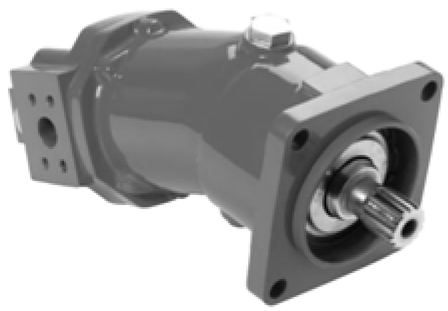


Figure 3-6. A bent axis variable displacement hydraulic pump/motor unit [33]

The P/M is driven by accumulator, in this model, the pressure and volumetric flow come from accumulator. Appendix 6 shows the MATLAB/Simulink model of the P/M.

3.3.4 Accumulator model

Accumulators are used as energy storage devices. These accumulators are based on the principle that gas is compressible and oil is incompressible. Oil flows into the accumulator and compresses the gas by reducing its storage volume. Energy is stored by the volume of hydraulic fluid that compressed the gas under pressure. If the fluid is released, it will quickly flow out under the pressure of the expanding gas. Compared to electric batteries, the hydraulic accumulator has relatively low energy density (Wh/kg), but it can be charged and discharged very frequently, so it has very high power density.

The hydraulic accumulator is better suited for short bursts of power rather than sustained energy delivery [1].

Some common types of accumulators are depicted in Figure 3-7. The main difference between them is in how the gas is confined to a specific volume.

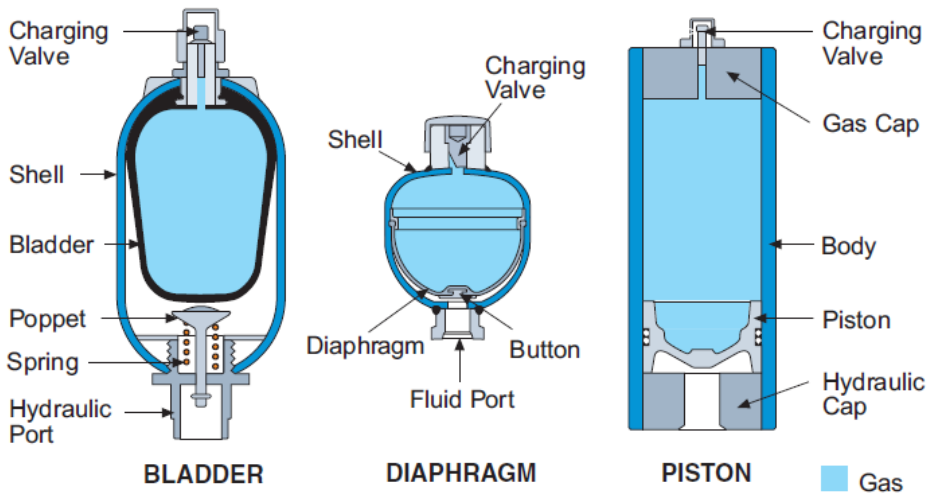


Figure 3-7 Types of accumulators [34]

Normally, a gas is considered in ideal state. The basic state parameters are pressure (P), volume ($V\!\!\!\!-$) and temperature (T). The rate at which compression and expansion of the gas takes place affects the gas state. If the rate is very slow and the gas temperature doesn't change, this process is called as isothermal process. If the rate is so fast that the gas temperature changes but not the surroundings (no gain or loss of heat), this process is known as adiabatic process [35]. In this project, the gas in the accumulator is considered in isothermal process because the foam in the accumulator acts as a heat sink [30], and the gas follows the ideal gas law:

$$pV\!\!\!\!- = mRT \qquad (3\text{-}25)$$

Here m is the gas mass, R is the specific gas (Nitrogen) constant.

The time rate of gas volume change is determined from the physical constraint. Figure 3-8 shows that the change in gas volume is equal to the change in liquid volume. The volume flow rate and the volume change can be figured out from the following equation:

$$Q = \frac{d\cancel{V}}{dt} \qquad (3\text{-}26)$$

$$\cancel{V} = \int Q dt \qquad (3\text{-}27)$$

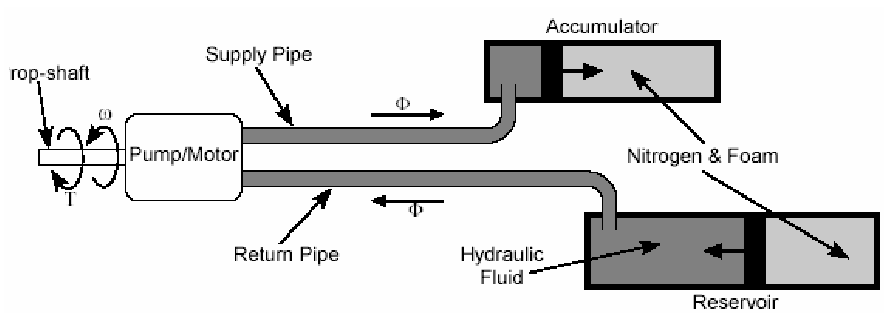


Figure 3-8: P/M and accumulator relationship [32]

In the MATLAB/Simulink model, there are two accumulators. When one accumulator works as high pressure accumulator, another one works as reservoir. When the all oil flows to low pressure accumulator from high pressure accumulator, this means $\cancel{V} = \cancel{V}_o$ ($\cancel{V}_o$ is the oil volume in the system), the switch occurs, the low pressure accumulator becomes high pressure and the high pressure accumulator becomes low pressure.

Appendix 7 shows the MATLAB/Simulink model of the accumulator.

3.3.5 Air-tank Model

The air tank is an energy storage unit. It contains the high pressure air (50Mpa) which is supplied by the compressor and will be sent to the pressure exchanger. The assumption is that the tank is an isothermal process when it acquires high pressure air from the compressor or when it delivers the high pressure air to the pressure exchanger. The assumption is valid because the air tank is very big. The stored energy in the air tank can be calculated as follows equations (ideal gas in isothermal process) [36]:

$$E = mRT \ln \frac{P}{P_0} \qquad (3\text{-}28)$$

The parameters are defined as follows: E -the stored energy; m - the air mass in the air tank; P - the air pressure; P_0 -the minimum pressure, here is the atmosphere pressure; T -temperature; R - the specific gas (air) constant.

When the system runs, the switch from the high pressure accumulator to the low pressure accumulator occurs and the air in the pressure exchanger will be lost. The total loss mass is M_{loss} (comes from pressure exchanger model). When the pressure in the air tank is too low, the compressor will be started to fill the air tank, the filled air mass is M_c.

Then the air pressure in the air tank can be calculated from the following equations:

$$m = M_{airo} - M_{loss} + M_c \qquad (3\text{-}29)$$

$$PV_t = \left(M_{air0} - M_{loss} + M_c\right)RT \qquad (3\text{-}30)$$

$$P = \left(M_{air0} - M_{loss} + M_c\right)\frac{RT}{V_t} \qquad (3\text{-}31)$$

Here, m is the air mass in the air tank, V_t is the volume of the air tank; M_{air0} is the initial air mass in the air tank.

$$M_{air0} = \frac{P_1 V_t}{RT} \qquad (3\text{-}32)$$

P_1 is the initial air pressure in the air tank (in this model, it is set to 50Mpa).

Appendix 9 shows the MATLAB/Simulink model of the air tank.

3.3.6 Pressure Exchanger Model

The pressure exchanger is used for transferring high pressure from the air tank to the accumulator. When the switch from the high pressure accumulator to the low pressure accumulator occurs, the air in the pressure exchanger will be lost. The lost air mass is:

$$m_{loss} = \frac{P V_{PE}}{RT} \qquad (3\text{-}33)$$

Here P is the air pressure in the pressure exchanger (in the MATLAB/Simulink model, it is set to 35Mpa); V_{PE} is the volume of the pressure exchanger. The total loss mass is:

$$M_{loss} = \sum_{i=1}^{N} m_{loss} \qquad (3\text{-}34)$$

Where N is the switch times.

The lost high pressure air leads energy loss. Just for simplification, assume the process that the pressure exchanger receives the high pressure air is isothermal. The lost energy can be calculated from the following equation (isothermal process):

$$E_{loss} = \sum_{i=1}^{N} m_{loss} RT \ln \frac{P}{P_0} \qquad (3\text{-}35)$$

Appendix 8 shows the MTALB/Simulink model of the pressure exchanger.

3.3.7 Air Compressor Model

In the proposed system, the compressor is used to recharge the air tank when the pressure in the air tank is low ($P < P_{\min}$). The compressor in this system can be driven by an electrical motor when the vehicle is out of use or by the ICE when the vehicle is running. Figure 3-9 shows a commercial high pressure compressor.

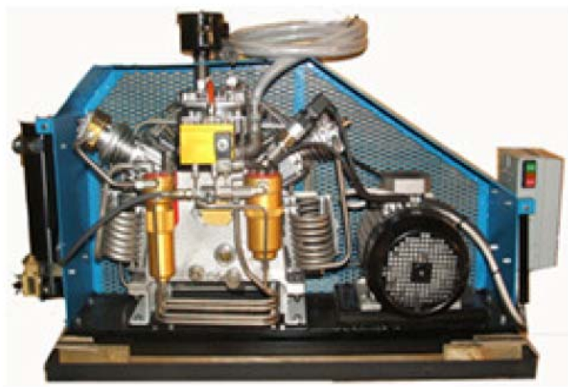


Figure 3-9: High pressure air compressor [37]

Normally the reciprocating compressors are used for high pressure. When air at high pressure is required, a multi-staged compression cycle is more efficient than using a single stage compressor. Also, single stage compressors delivering high pressures result in high gas temperatures which effect the lubrication. In the MATLAB/Simulink model the compressor has three stages. Figure 3-10 is the P-V diagram for a three -stage reciprocating compressor, which is adopted in this work.

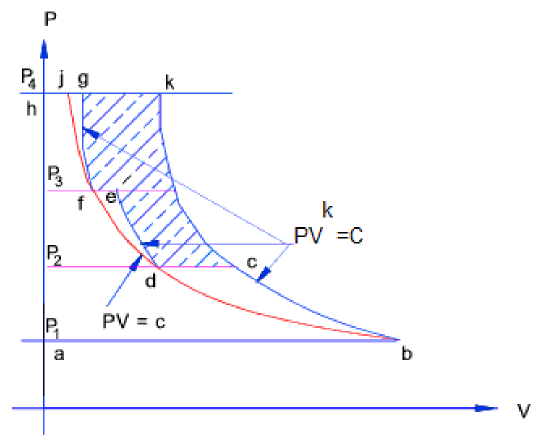


Figure 3-10: P-V diagram for a three-stage reciprocating compressor without clearance [38].

In Figure 3-10, P_1 is the input air pressure, P_4 is the output pressure. In the Simulink model, P_1 is atmosphere pressure and P_4 is air tank pressure. The route of a-b-c-k-h shows a single stage compression which is adiabatic process and a-b-j-h shows an ideal isothermal compression. The area enclosed by the curves indicates the work done per cycle respectively. It is clear that the work done in the ideal isothermal process is far less than that done in the single stage compression.

In a three stage compressor process, the compression process is as follows: First, the air is compressed from P_1 to P_2 (a → c) and the air is transferred into a receiver and cooled to its original temperature (c→d). Second, the air is then transferred from the receiver to a second cylinder and compressed to P_3 (d → e). The air is then transferred to a second receiver and cooled back to its original temperature (e → f). Third, the air is transferred to a third cylinder and compressed to P_4 (f → g).

The overall process is the curve a-b-c-d-e-f-g-h. The cooling brings the process closer toward the ideal isothermal (constant temperature) curve. The savings in work done per cycle is identified by the shaded area.

The Simulink model only analyzes the condition that the compressor is driven by the ICE. For this condition the compressor and the ICE have the same angular speed ω_e. The compressor displacement per revolution is V_{cs}. The compressor output air mass (M_c) can be calculated from the following equations:

$$m_c = \rho V_{cs} \qquad (3\text{-}36)$$

$$M_c = \int_0^{t_f} \omega_e m_c dt \qquad (3\text{-}37)$$

ρ is the air density, m_c is the air mass per revolution. t_f is the filled time.

For an approximate calculation we only consider the compressor without clearance. For a three-stage compression process, the basic rule is [27]:

$$\frac{P_4}{P_3} = \frac{P_3}{P_2} = \frac{P_2}{P_1} \qquad (3\text{-}38)$$

For simplification, the compressor is considered in the adiabatic process, the required power is [27]:

$$\dot{W}_C = \frac{3k}{k-1} M_c RT \left[\left(\frac{P_4}{P_1} \right)^{\frac{k-1}{3k}} - 1 \right] \frac{1}{\eta_c} \qquad (3\text{-}39)$$

Where:

k is the specific heat ratio (k=1.4). η_c is the compressor overall efficiency (80%), it includes the mechanical and thermodynamic efficiency [42].

The compressor required energy (E_c) can be attained from the equation:

$$E_C = \int \dot{W}_c dt \qquad (3\text{-}40)$$

Appendix 10 shows the MATLAB/Simulink model of the compressor.

3.3.8 Power Management in the Model

Different rules for power distribution between the P/M and the ICE are implemented for each of the power delivery modes, as explained in this section.

-Power delivery mode. In power delivery mode ($\dot{W}_{req} > 0$) the motor attempts to take the entire load. The actual required volumetric flow rate and motor output power can be attained through the following equations:

$$Q_{act} = \omega_h D_{act} \quad (3\text{-}41)$$

$$\dot{W}_{h,act} = \Delta p Q_{act} \quad (3\text{-}42)$$

Where D_{act} is the actual displacement $(D_{\min} \le D_{act} \le D_{\max})$, ω_h is the angular speed of the P/M.

If the power output ($\dot{W}_{h,act}$) meets the required demand, the engine idles or the engine only drives the compressor and all vehicle power is supplied by the motor. If $\dot{W}_{h,act}$ is less than the demand, the engine will make up the difference.

The power required at the propeller shaft is $\dfrac{\dot{W}_{req}}{\eta_f}$. ($\eta_f$ is the differential efficiency).

The power delivered to the propeller shaft by the engine and the P/M unit is

$$\eta_h \dot{W}_h + \eta_t (\dot{W}_e - \dot{W}_c) .$$

Where η_h is the hydraulic transmission efficiency; η_t is the transmission efficiency (which depends on the transmission gear ratio N_t); $\dot{W}_h$ is the hydraulic motor power output, and $\dot{W}_e$ is the engine power output, $\dot{W}_c$ is the compressor required power. This leads to:

$$\frac{\dot{W}_{req}}{\eta_f} = \eta_h \dot{W}_h + \eta_t(\dot{W}_e - \dot{W}_c) \qquad (3\text{-}43)$$

-Power absorption mode In power absorption mode ($\dot{W}_{req} < 0$), the engine idles or the engine only drives the compressor. The hydraulic unit operates in pump mode, which is subjected to the same displacement limitation as in motor mode, now filling and pressurizing the accumulator. If the braking load is beyond the pump's capability (which is the case if the maximum displacement magnitude is reached or the accumulator is full), the remaining braking power is absorbed by friction brakes.

The power delivered to propeller shaft by the differential is $\eta_f \dot{W}_{req}$, the power at the propeller shaft to drive the pump is $\frac{\dot{W}_h}{\eta_h}$, and thus

$$\eta_f \dot{W}_{req} = \frac{\dot{W}_h}{\eta_h} + \dot{W}_{fric} \qquad (3\text{-}44)$$

$$\dot{W}_e = \dot{W}_c \qquad (3\text{-}45)$$

Where $\dot{W}_{fric}$ is the power dissipated by friction brakes.

These power rules can be summarized as follows:

For $\dot{W}_{req} > 0$, $\frac{\dot{W}_{req}}{\eta_f} = \eta_h \dot{W}_{h,act} + \eta_t(\dot{W}_e - \dot{W}_c)$ with $\eta_t \dot{W}_e = \frac{\dot{W}_{req}}{\eta_f} - \eta_h \dot{W}_{h,act} + \eta_t \dot{W}_c$.

For $\dot{W}_{req} < 0$, $\eta_f \dot{W}_{req} = \frac{\dot{W}_{h,act}}{\eta_h} + \dot{W}_{fric}$ and $\dot{W}_e = \dot{W}_c$.

Chapter 4

Results

In chapter 3 all the sections of the models were presented. This model is constructed in MATLAB/Simulink. In this chapter, the detailed numerical results obtained from the model will be presented. This chapter includes simulation results for the conventional truck, the hydraulic hybrid truck, and the proposed system with the compressed air system.

4.1 Duty Cycle

For vehicle tests, the duty cycle is a critical element in any analysis of fuel economy (FE). Its importance is emphasized in hybrid applications, where complex interaction in the system and the control strategy might play out very differently depending on the driving scenario [12]. EPA has developed many driving schedules for vehicle testing. The Federal Urban Driving Schedule (FUDS) is the most used schedule for medium trucks FE testing [12,17,18,19].

The FUDS includes the vehicle speed as a function of time as shown in Figure 4-1.

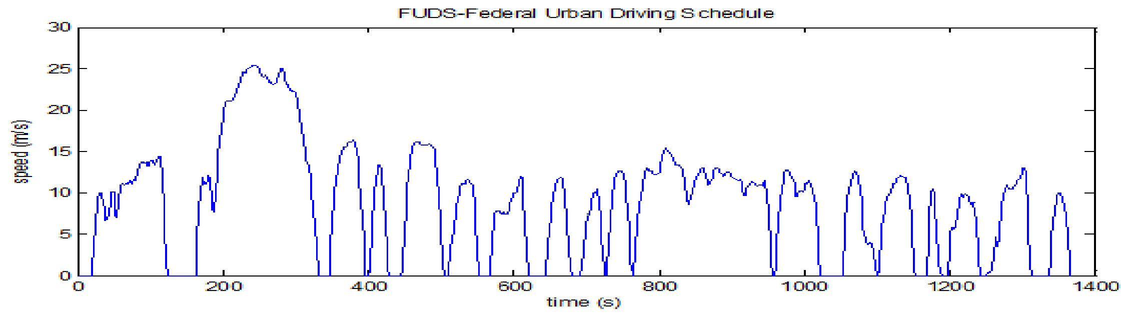


Figure 4-1: FUDS-Federal Urban Drive Schedule [39]

4.2 Conventional Truck Results

For a conventional vehicle, the model [20] only includes the vehicle dynamics and internal combustion engine. The vehicle that was studies is the International 4700 series, Class VI 4x2 truck. The vehicle mass is 7340kg when fully loaded. All the energy is supplied by the engine. When the vehicle is braking the required power is zero, therefore the engine is idling.

Appendix 2 shows the conventional vehicle MATLAB/Simulink model. Figure 4-2 is the simulation result.

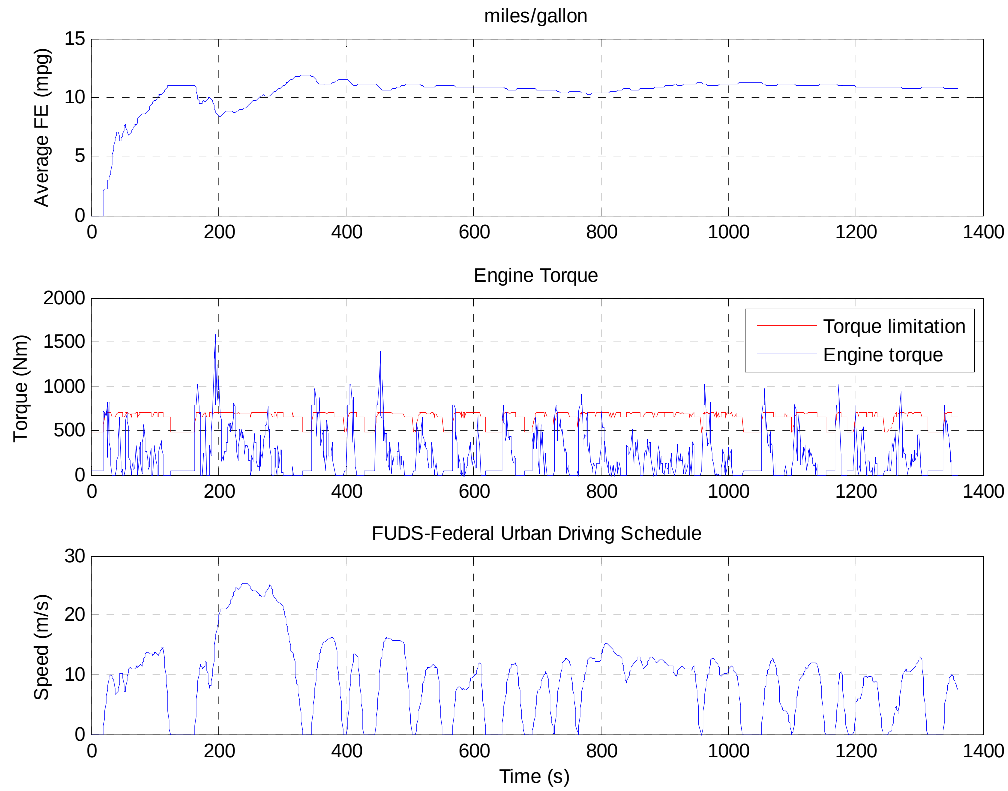


Figure 4-2: The conventional truck simulation result based on FUDS

The average fuel economy (FE) is:

$$FE_{avg} = \frac{\int_0^{t_T} V dt}{\frac{1}{\rho_f}\int_0^{t_f} \dot{m}_f dt} = \frac{journey\ distance(miles)}{fuel\ consumption(gallon)} \qquad (4\text{-}1)$$

Where t_T is the running time; ρ_f is the fuel density; $\dot{m}_f$ is the fuel mass flow; V is the vehicle speed.

At the end of running time the FE_{avg} is 10.78mpg. Figure 4-2 shows in some short time the vehicle is overload (the engine torque is bigger than the torque limitation). The overload happens in the acceleration time. Overload time is very short, which is acceptable for this simulation study as it does not change the calculated fuel economy.

Figure 4-3 is the power curve about the vehicle required power (+), braking power (-), and engine power (+).

Figure 4-3: The conventional truck power curve

Figure 4-3 shows that the engine power can cover all the required power except the braking power. Since the braking power is negative, this implies that no additional power comes from the engine when braking.

After integrating the power, the energy can be attained:

$$E_{engine} = 36.3MJ$$

$$E_{required+} = 36.3\ MJ$$

$$E_{brake} = 13.7MJ$$

The engine energy (E_{engine}) can cover the drag energy ($E_{required+}$). All of the braking energy is lost, this is the energy that would be absorbed through the use of HHVs. In a conventional truck, the lost energy is about $38\%\left(\frac{E_{brake}}{E_{required+}}\times 100\%\right)$ of the required energy based on FUDS.

4.3 Hydraulic Hybrid Truck Results

For the hydraulic hybrid vehicle, the parallel system is studied [20]. The vehicle studied is the International 4700 series, Class VI 4x2 truck, for consistency. The vehicle mass is 7612kg when fully loaded.

The models used to simulate this vehicle include vehicle dynamics model, engine model, pump/motor model, accumulator model.

The power management rule is as follows [12]:

Whenever there is energy available in the accumulator, controller will call upon the motor to satisfy the total power demand. If the power requirement is more than what motor can provide, the engine will supplement the motor power. If the accumulator is empty then the engine becomes the sole power source.

Appendix 3 shows the parallel hydraulic hybrid system MATLAB/Simulimk model. The solution can be seen in Figure 4-4.

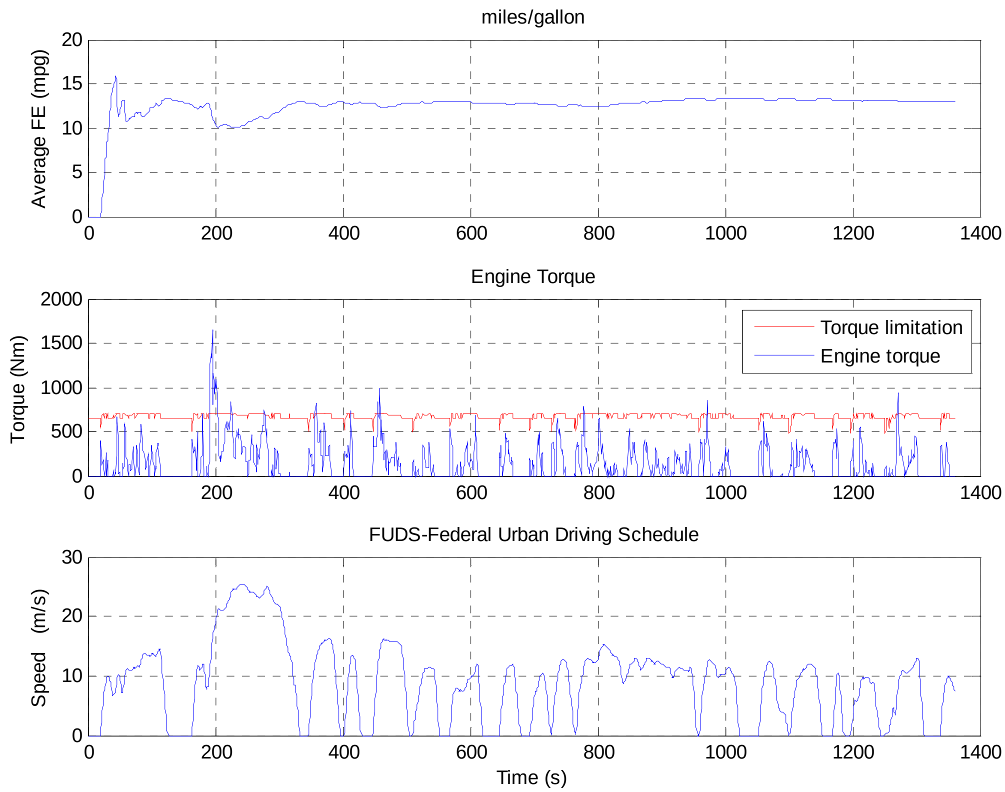


Figure 4-4: Average FE and engine torque curve based on FUDS

At the end of running time the FE_{avg} is 13.06 mpg, So, the improvement in FE is:

$$\frac{13.06-10.78}{10.78}\times 100\% = 21\%$$

Figure 4-4 the engine torque is better than Figure 4-3 the engine torque. This means the hydraulic hybrid system can make the engine work in a comparative mild state.

Figure 4-5 is the power curve for the vehicle required power, the hydraulic power, and engine power.

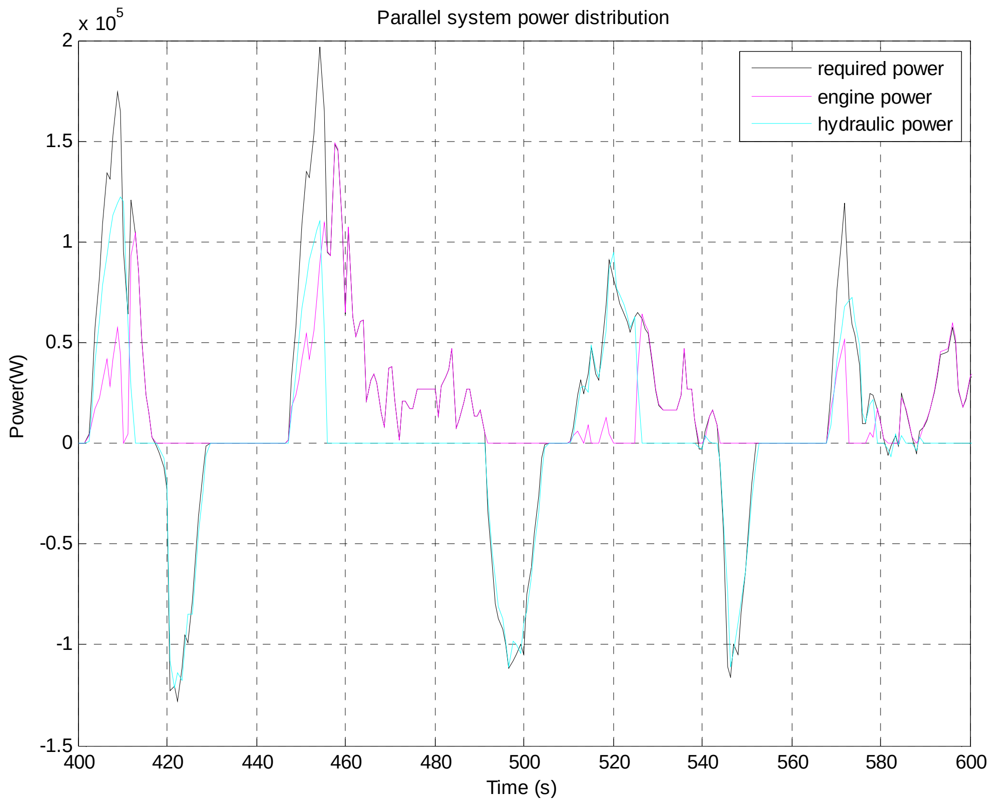


Figure 4-5: Comparison of the required power of the truck for the FUDS, the hydraulic power, and the power provided by the engine.

When braking, the required power and the hydraulic power are negative; therefore the engine power is zero. The hydraulic unit operates as a pump; it pumps hydraulic fluid from a low pressure reservoir to the accumulator. The braking energy will be absorbed and stored in accumulator. When the vehicle is accelerating or constant-speed driving, the required power is positive, the hydraulic unit operates as a motor to allow pressurize fluid to flow in the opposite direction. The hydraulic power will supply the required power. If the hydraulic power is not enough to meet the requirement set by the vehicle operator, the ICE power will make up the deficit. So, at that time:

$$required\ \ power = Hydraulic\ \ power + Engine\ \ power$$

After integrating the power, the energy can be attained:

Engine output energy to the propeller shaft: $E_e = 27.4\ MJ$

Required energy (+): $E_{r+} = 37.7\ MJ$, this energy is required for driving the vehicle.

Braking energy (Required energy (-)): $E_{r-} = 13.9 MJ$

Hydraulic energy (+): $E_{h+} = 10.8\ MJ$, this energy is the hydraulic system output energy.

Hydraulic energy (-): $E_{h-} = 12.8 MJ$, this energy is the hydraulic system absorbing energy.

So, based on FUDS :

$72\% \left(\frac{E_e}{E_{r+}} \times 100\% \right)$ driving energy is supplied by engine.

$28\% \left(\frac{E_{h+}}{E_{r+}} \times 100\% \right)$ driving energy is supplied by hydraulic system.

$92\% \left(\frac{E_{h-}}{E_{r-}} \times 100\% \right)$ braking energy is absorbed by the hydraulic system and stored in accumulator.

$84\% \left(\frac{E_{h+}}{E_{h-}} \times 100\% \right)$ absorbed energy is supplied to drive the vehicle.

So, wheel to wheel efficiency is $74\% \left(\frac{E_{h+}}{E_{r-}} \times 100\% \right)$

When calculating the hydraulic energy, the thermodynamics process of the accumulator is assumed isothermal.

4.4 Proposed Hydraulic Hybrid Truck Results

The proposed hydraulic hybrid system can be found in Figure 3-3. The assumption is that when all N_2 and oil is in Acc, the pressure is $P_{N2\max}$=35Mpa (about 5000psi). Since the high pressure air in the air tank can push all the N_2 to accumulator, the pressure should be very high. The initial value is given $P_{\max}$=50Mpa (about 7250psi).

Other assumption:

The air-tank volume: V_t=2 m^3 ;

The Oil volume: V_o=0.06m^3 (2 $feet^3$);

The accumulator volume: V_a=0.08m^3;

The minimum volume of N_2: $V_{N2\min}$=$V_a - V_o$=0.02m^3.

The minimum N_2 pressure is $P_{N2\min}$ (1.7Mpa or 250psi) when the N_2 volume is in maximum volume

$$V_{N2\max}(V_{PE}+V_a).$$

The volume of pressure exchanger is V_{PE}, so $V_{PE}=\dfrac{P_{N2\max}V_{N2\min}}{P_{N2\max}}-V_a=0.32m^3$

The initial air mass in the air tank is $M_{air0}=\dfrac{P_{\max}V_t}{RT}=1154kg$

Minimum air pressure $P_{\min}=\dfrac{35Mpa(V_t+V_{PE})}{V_t}=40.6Mpa\ (5888\,psi)$

The pressure difference between the two accumulators (ΔP) can be calculated as follows (isothermal process):

The pressure of the high pressure accumulator (P_1) can be attained from:

$$P_{N2\max}V_{N2\min}=P_{high}(V_{N2\min}+V) \qquad (4\text{-}2)$$

$$P_{high} = \frac{P_{N2\max}\cancel{V}_{N2\min}}{\cancel{V}_{N2\min} + \cancel{V}} \qquad (4\text{-}3)$$

The pressure of the low pressure accumulator (P_{low}) can be attained from:

$$P_{N2\max}\cancel{V}_{N2\min} = P_{low}(\cancel{V}_{PE} + \cancel{V}_a - \cancel{V}) \qquad (4\text{-}4)$$

$$P_{low} = \frac{P_{N2\max}\cancel{V}_{N2\min}}{\cancel{V}_{PE} + \cancel{V}_a - \cancel{V}} \qquad (4\text{-}5)$$

So, $\Delta P = P_{high} - P_{low}$ (4-6)

$\cancel{V}$ is the oil or N_2 volume change, from equation (3-27).

The low pressure accumulator pressure (P_{low}) range can be calculated: 1.75-2.06Mpa (since the N_2 volume range is $V_{PE} + V_{N2\min} \sim V_{PE} + V_a$). The high pressure accumulator pressure (P_{high}) range is 8.75-35Mpa (since the N_2 volume range is $V_{N2\min} \sim V_a$). So, ΔP range is 6.69-33.25Mpa.

Each switch the lost mass is: $m_{loss} = \frac{35Mpa \times \cancel{V}_{PE}}{RT} = 134kg$.

Appendix 4 shows the MATLAB/Simulimk Model of the proposed parallel hydraulic hybrid system. The input data about engine, pump/motor, air compressor and the vehicle is in attached table, see Appendix 1: Simulink model input data table.

<u>Simulation results:</u>

(with the compressor displacement 3000X10^{-6}m^3)

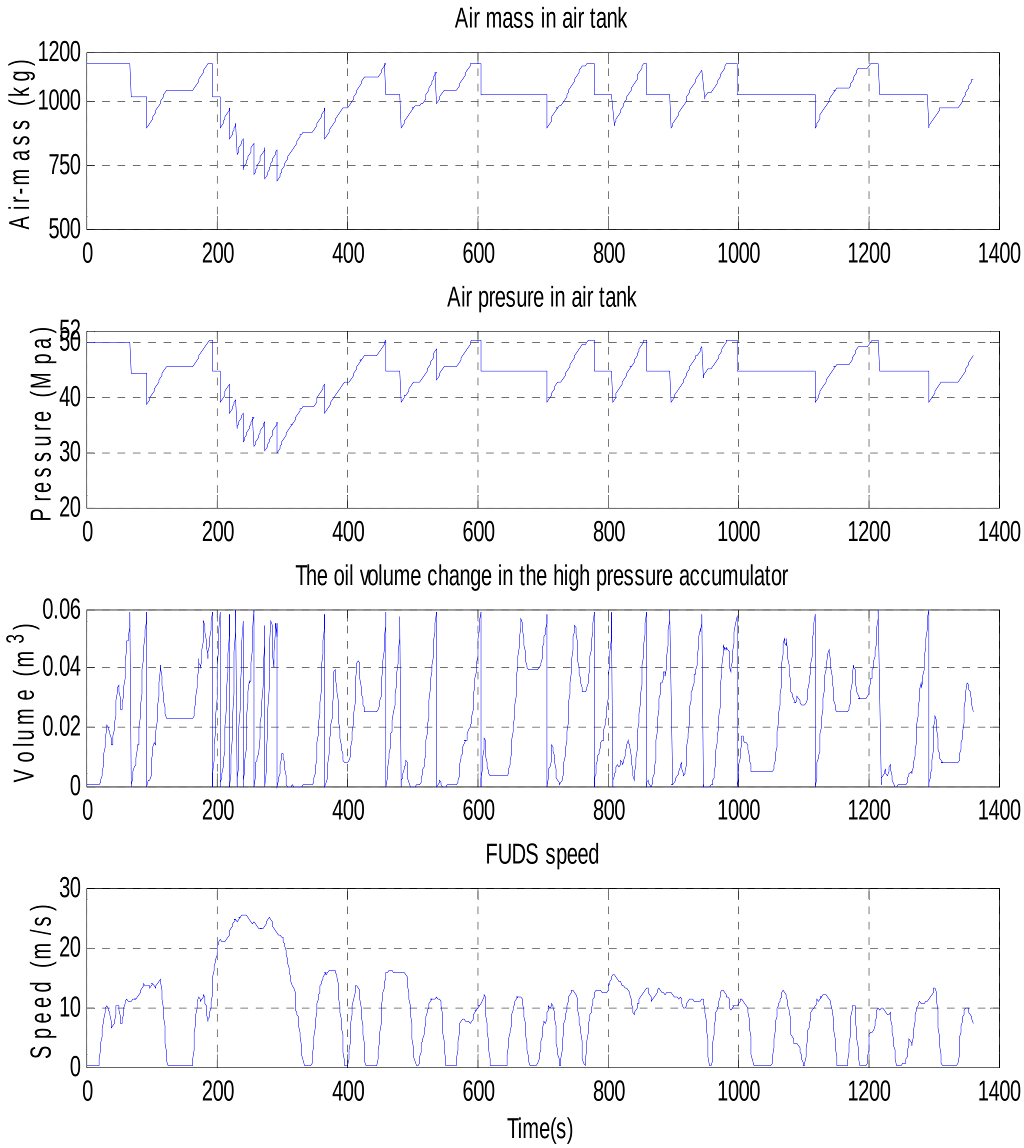


Figure 4-6: Air mass, pressure in air tank change along with the accumulator switch

Figure 4-6 shows when the switch occurs, the air mass and the air pressure in the air tank will decrease. For a fully loaded air tank, after switching the accumulators only two times, the pressure drops to P_{min}. At this point the compressor begins to recharge the air tank.

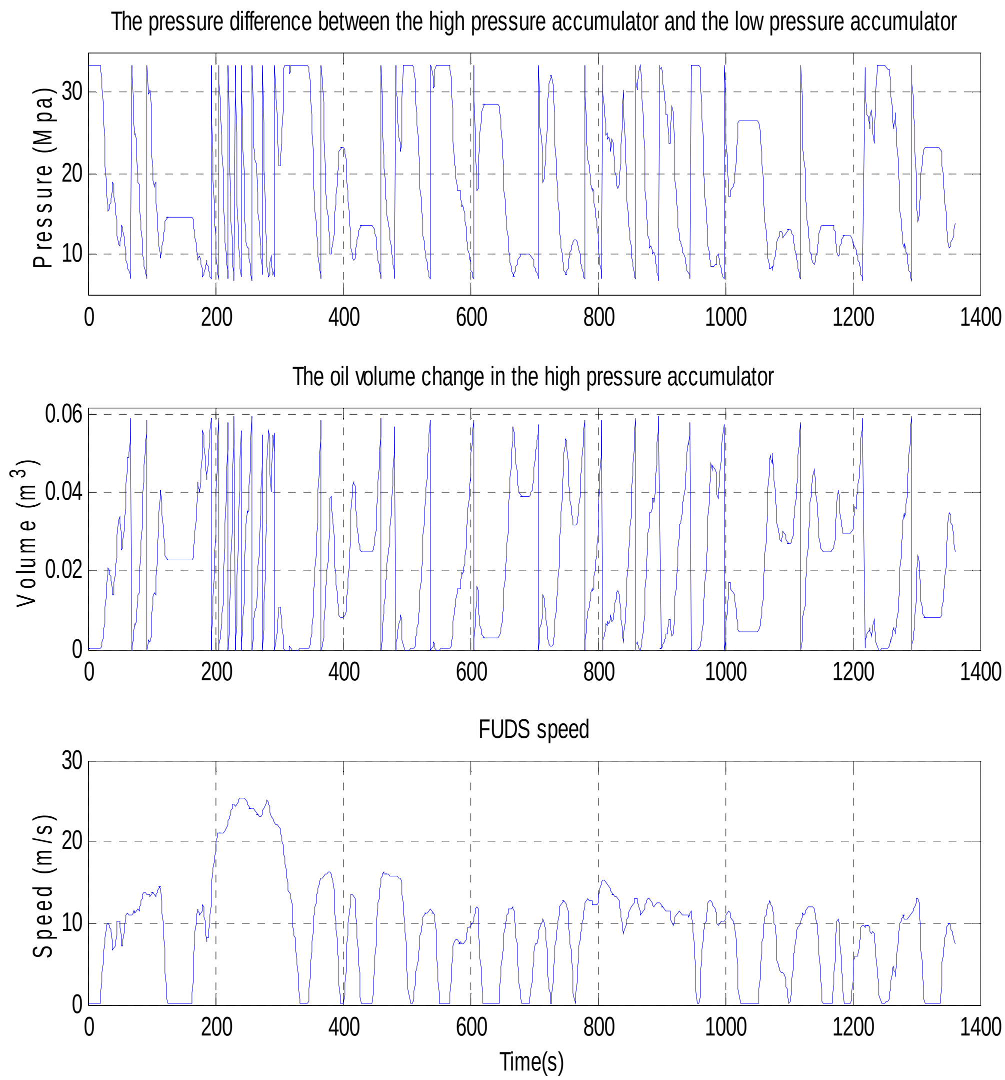


Figure 4-7: The P/M pressure change along with the accumulator switch

Figure 4-7 shows when the switch happens, the P/M pressure change. With the increasing oil volume the pressure decreases, when switch happens the pressure becomes highest again. As analysis before the pressure range is 6.69-33.25Mpa.

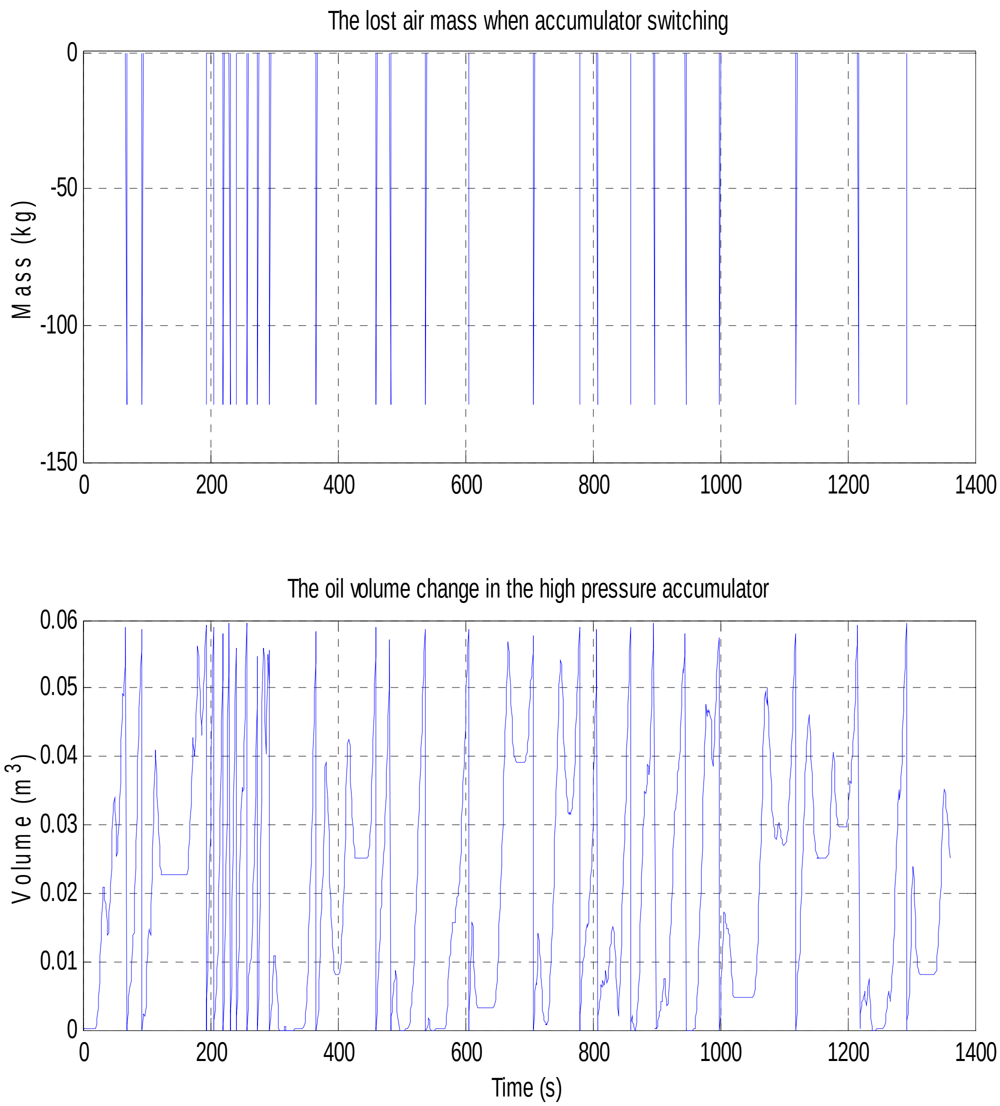


Figure 4-8: The lost mass along with the accumulator switch

Figure 4-8 shows the lost air mass when each switch occurs. All of the value for the lost mass are the same (134kg), because valves #1 or #4 (Figure 3-3) can be controlled. When the pressure of the pressure exchanger becomes 35Mpa, all the N_2 is pushed to accumulator, then #1 or #4 is closed.

Fuel economy analysis:

Figure 4-9 shows that after about 100s, the compressor begins to work. Before the compressor starts, the FE is about 8-12mpg. When the compressor starts, the FE decreases very quickly. At the end of the running time, FE is about 0.2 mpg. The reason for this is that before the compressor starts, the hydraulic units can supply most of the required power to run the vehicle and the engine is only supplemental energy source. In most of that time the engine is idling, only a little fuel is consumed. However, when the air pressure is too low, the engine must work as the only power source to run the vehicle and also run the compressor to recharge the air tank. So, at this condition the engine must supply more power than the conventional vehicle. Even after the air tank is recharged, the FE does not recover quickly because the FE is calculated by the average value.

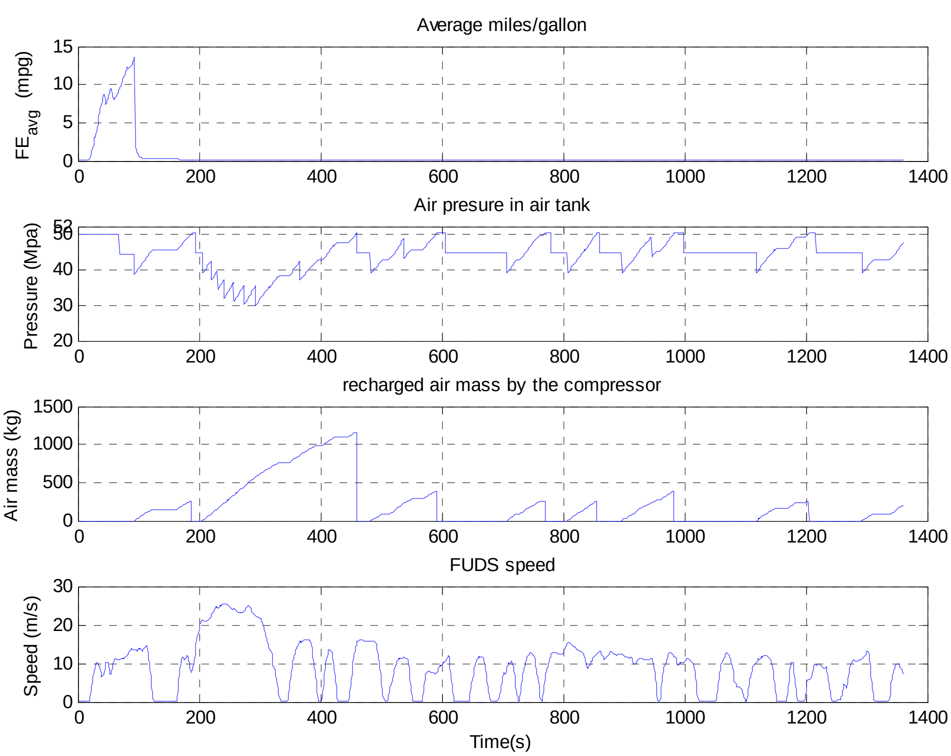


Figure 4-9: FE, air pressure and compressor supplied mass curve.

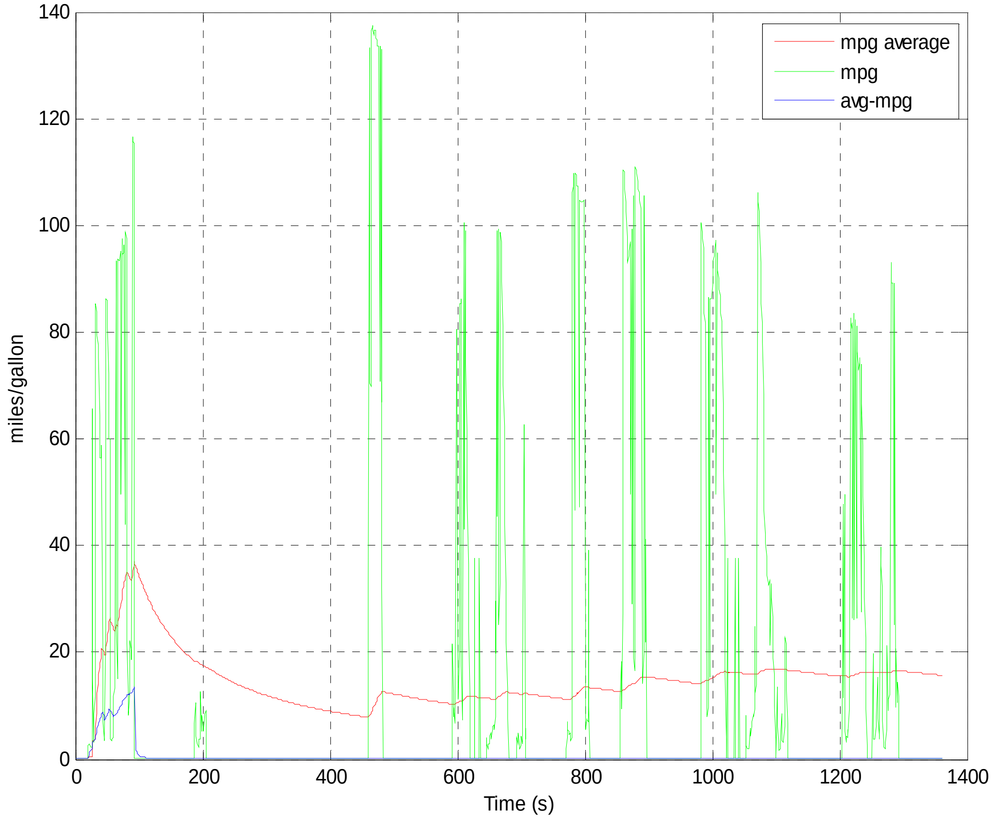


Figure 4-10: FE curve

Figure 4-10 shows the FE change in FUDS cycle.

In Figure 4-10, $mpg = \frac{V}{\dot{V}} = \frac{miles/s}{gallon/s} = miles/gallon$, it is instantaneous value.

$$mpg\_average = \frac{\int \frac{V}{\dot{V}} dt}{t} = miles/gallon$$, it is average value of mpg.

$$avg\_mpg = FE_{avg} = \frac{\int V dt}{\int \dot{V} dt} = miles/gallon$$, FE_{avg} shows the most important information about the system.

(t - running time; V -vehicle speed, miles/s; $\dot{V}$ -fuel consumption volumetric flow rate)

Figure 4-11 shows the energy change in the air tank and the required compressor energy.

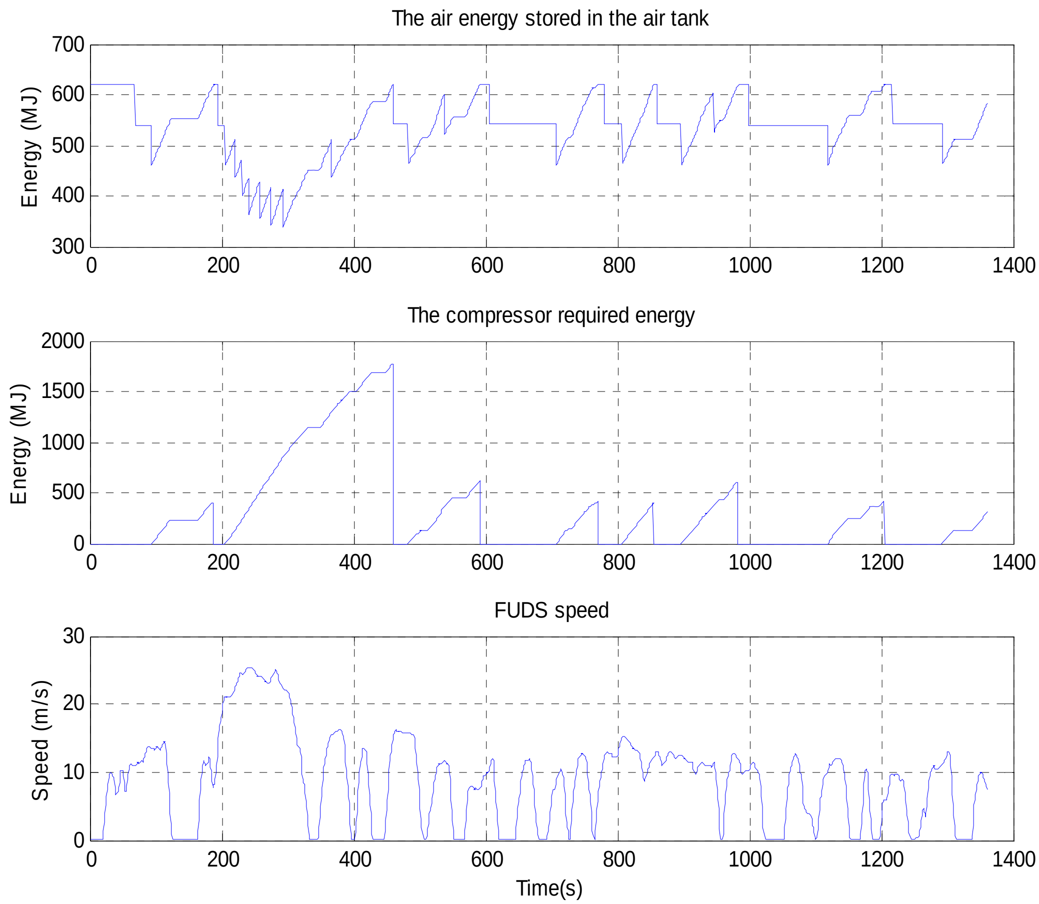


Figure 4-11: The air tank energy change and compressor required energy

While the vehicle is running, the energy in the air tank decreases. When there is not enough supplemental energy, the compressor will recharge the air tank, so, the energy in the air tank will increase, until the air tank is full, and the energy storage is at its maximum.

Figure 4-11 shows that the compressor required energy is much more than the energy stored in the air tank. This is because when recharging, the air tank still works as power source, the switch still happens. With each switch, the energy stored in the pressure exchanger will be lost. The lost energy can be seen in Figure 4-12:

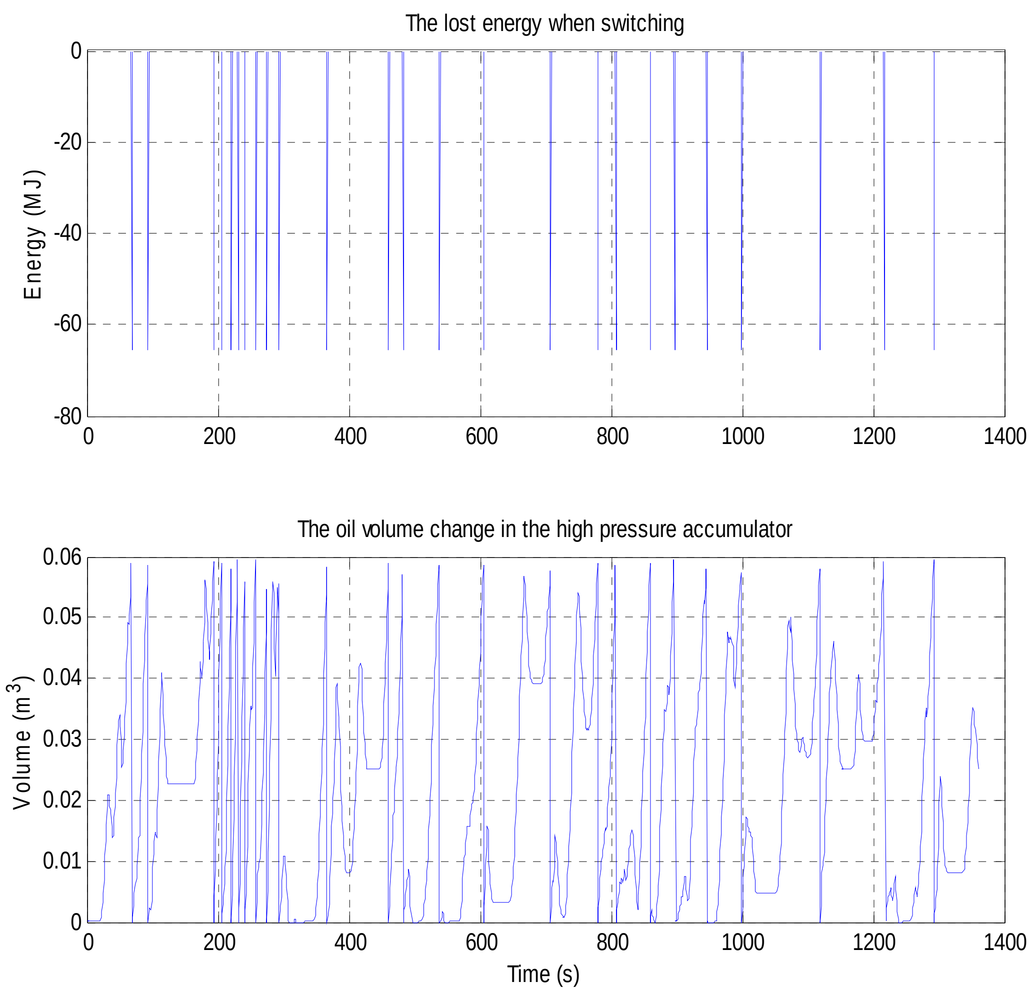


Figure 4-12: The lost energy when switch occurs

Figure 4-12 shows the lost energy when each switch happens. With each switch the lost energy is the same. The simulation results as follows:

The compressor required energy is 4987MJ.

The totally lost energy is $E_{loss} = 1637\ MJ$ (isothermal process)

The vehicle required energy: $E_{r+} = 44\ MJ$

The braking energy: $E_{brake} = 17\ MJ$

Hydraulic energy: $E_{h+} = 22.7\ MJ$

Absorbed energy: $E_{h-} = 10\ MJ$

Internal combustion engine power analysis:

In the Simulink model, the power relationship of the vehicle required power, the hydraulic power, the compressor required power and the engine power is:

When the vehicle required power is positive,

$$engine\,power = vehicle\,required\,power - hydraulic\,power + compressor\,required\,power \quad (4\text{-}7)$$

When the vehicle required power is negative,

$$engine\ power = compressor\ required\ power \quad (4\text{-}8)$$

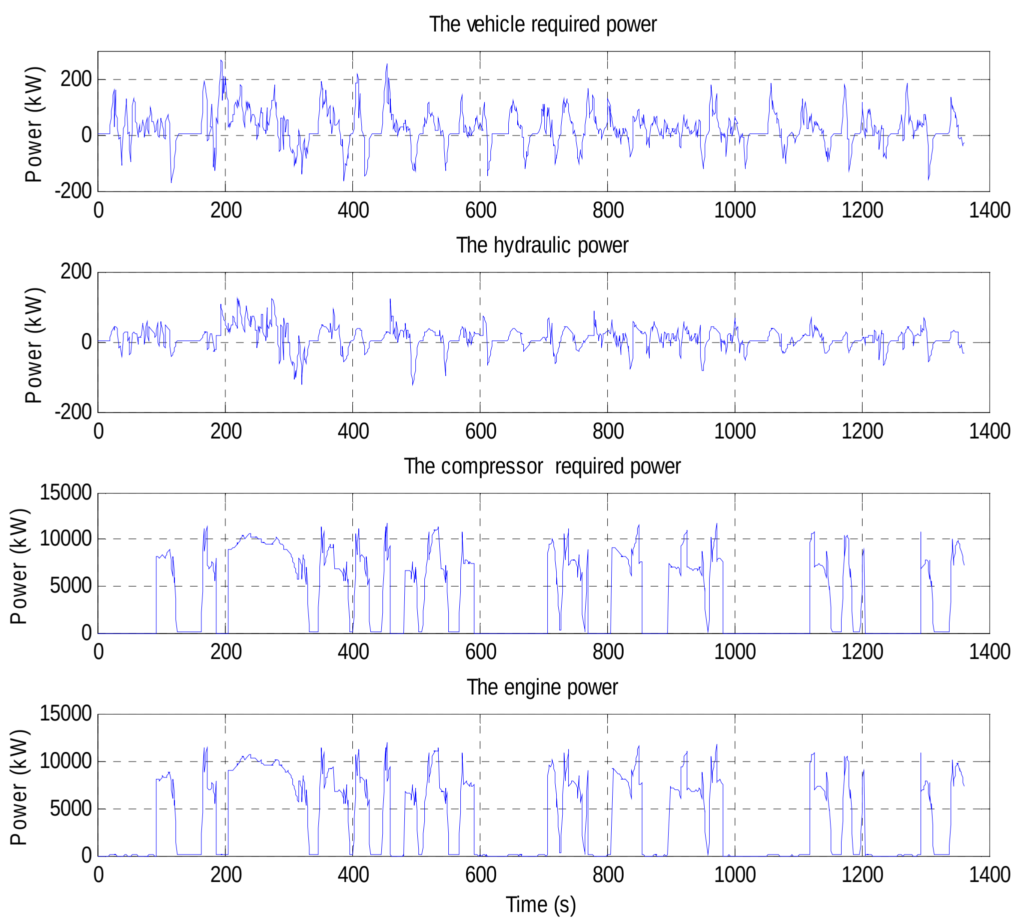


Figure 4-13: The power relationship

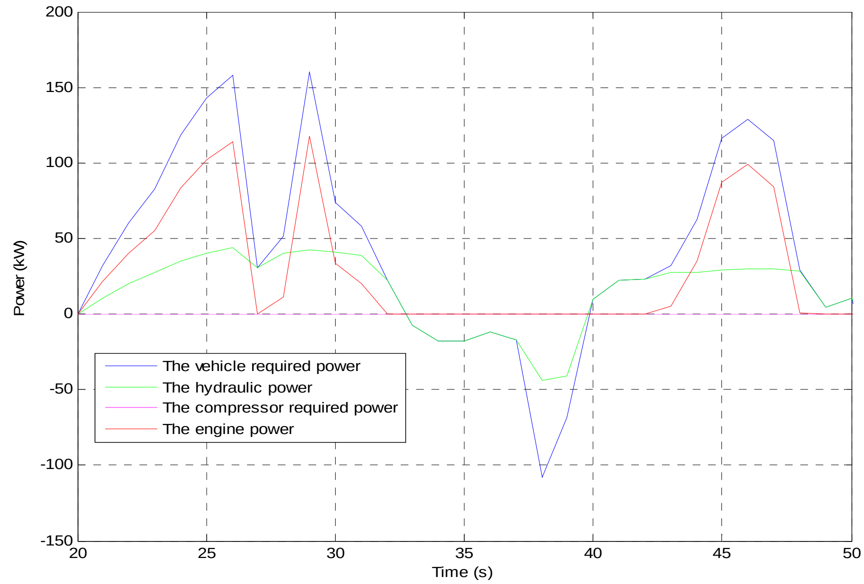


Figure 4-14: The power relationship before compressor works

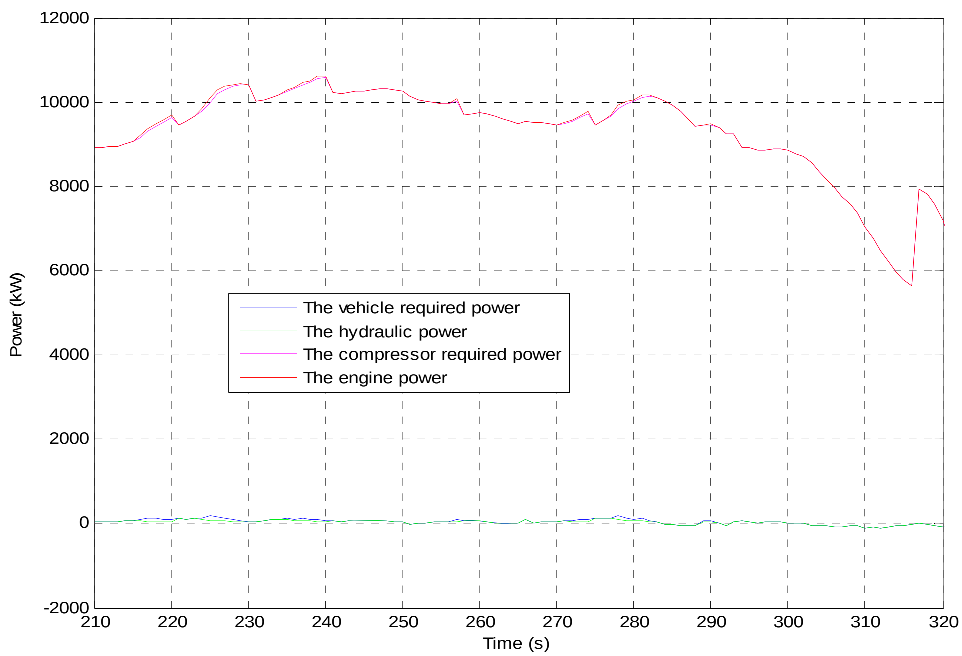


Figure 4-15: The power relationship when compressor works

Figure 4-13, 14, 15 shows the relationship of the powers above. When the compressor works the engine power is very large. At this time the engine power is almost the same with the compressor required power. The engine power and compressor power are more than 10000kw.

Based on FUDS , the engine output energy to driveshaft:

$E_e = 5008MJ$

The engine supplied $99.5\%\left(\frac{E_c}{E_e}\right)\times 100\%$ energy to the compressor.

About $60\%\left(\frac{E_{h-}}{E_{r-}}\right)\times 100\%$ braking energy was absorbed by the hydraulic system.

About $52\%\left(\frac{E_{h+}}{E_{r+}}\right)\times 100\%$ required energy was supplied by the hydraulic system. This energy comes from the air tank and also the absorbed energy.

When fully loaded, the air tank stored energy is 620 MJ ($E_a = mRT\ln\left(\frac{P_{\max}}{P_0}\right)$)

So:

The engine output energy 5008MJ→the compressor required energy 4987MJ →effective energy 2244MJ (efficiency 0.45)→lost energy 1637MJ→607MJ left

most of the energy is lost.

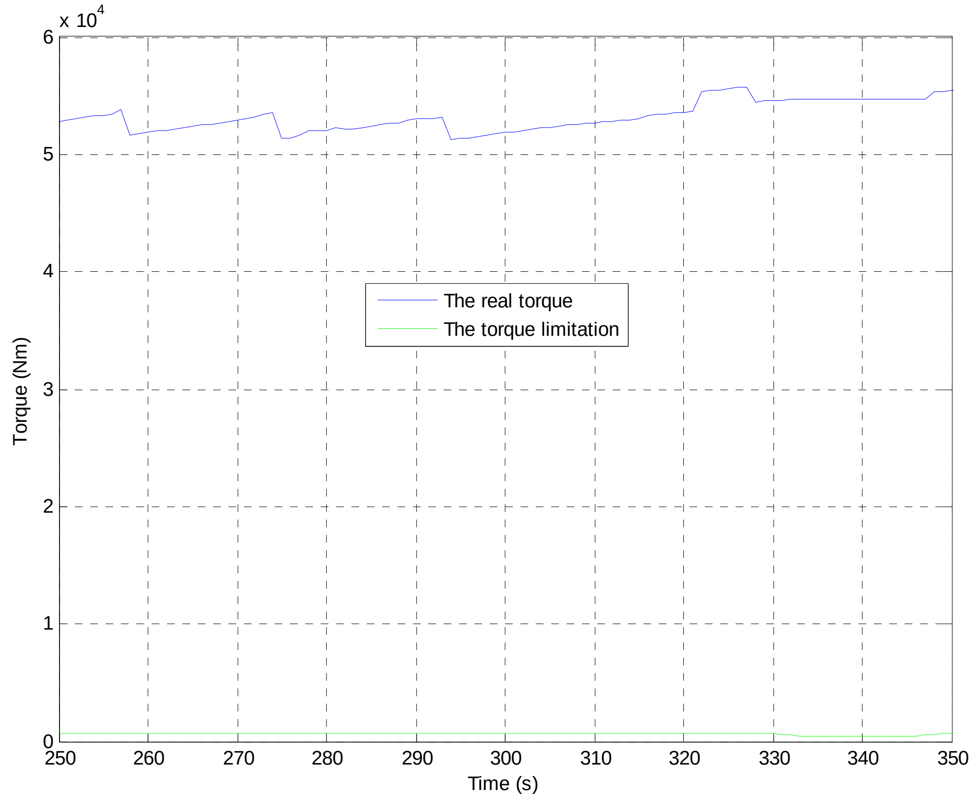


Figure 4-16: The engine torque curve

Engine torque curve (Figure 4-16) shows that if this system can work, the engine must be changed a very large one. The engine is more than 50 times the torque limitation.

Other solutions:

Given small compressor displacement (for example $500x10^{-6}m^3$), the result is as follows:

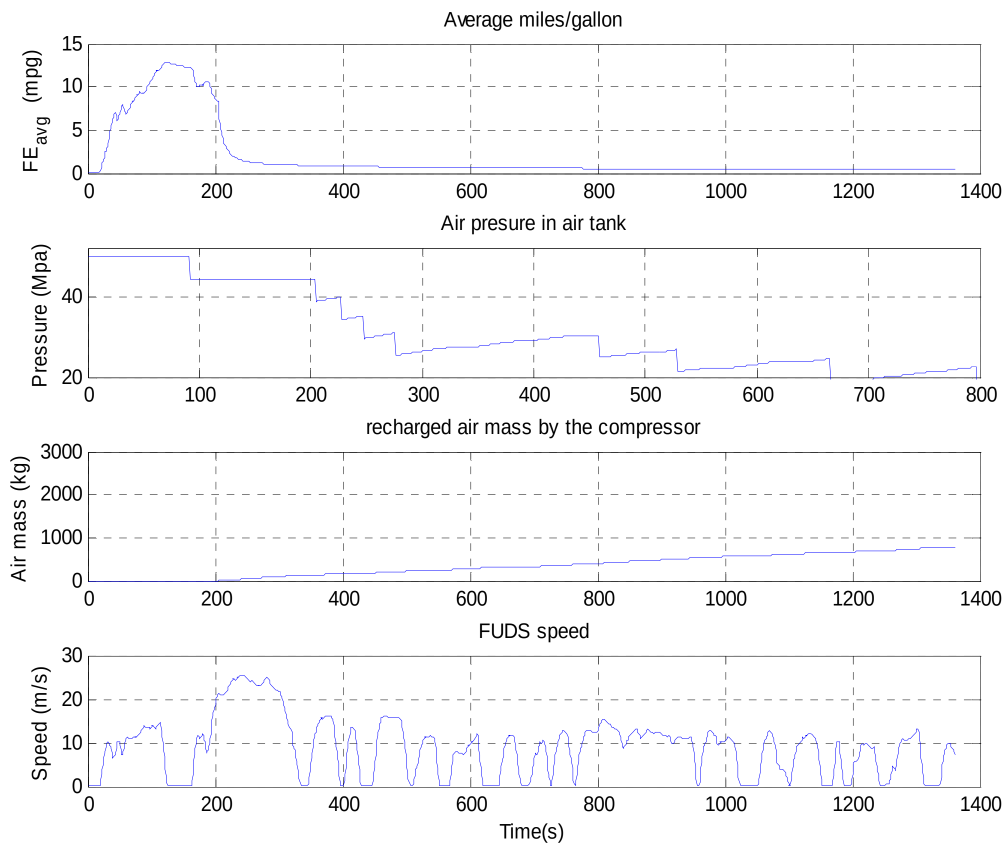


Figure 4-17: Compressor displacement is $500X10^{-6}m^3$

For a small compressor displacement, it would require a long time for the compressor to fully charge the air tank. Even the compressor cannot fully charge the air tank in the running time. For the fuel efficiency, the results are not favorable.

Chapter 5

Conclusion

This research was about the hydraulic hybrid system. In order to improve the energy density and also add plug-in capability on the hydraulic hybrid system, an air system is added to this hybrid system. The high pressure air can supply energy to drive the hydraulic motor. This system can improve the energy density and has the plug-in capability. The main purpose of the hybrid system is to improve the fuel economy as well as reducing emissions. This research adopted the MATLAB/Simulink tool to simulate the system. Based on FUDS, the Simulink model is tested.

For the standard International 4700 series, Class VI 4x2 truck, the average fuel economy is 10.78mpg. All the required energy is supplied by the engine, and all the braking energy is lost. The lost energy is 38% of the required energy.

The hydraulic hybrid truck fuel economy is 13.06 mpg. 72% driving energy is supplied by engine. 28% of the driving energy is supplied by hydraulic system. 92% braking energy is absorbed by the hydraulic system and stored in accumulator. 84% absorbed energy is supplied to drive the vehicle. Compared with conventional truck, there is 21% improvement on the fuel economy.

The proposed hydraulic hybrid system has some problems. The proposed system cannot get any improvement on the fuel economy. The first reason is in this system the two accumulators work in high or low pressure by turns. At each switch lots of energy will be lost from the pressure exchanger. The main work of the engine (99.6% engine energy) is to run the compressor to recharge the air tank, but most of the energy is lost. The second reason is the compressor overall efficiency is too low (45%). In order to run the compressor, the engine needs to be changed to a very large one. In addition, the added mass of this system is large when compared with the mass of a medium truck and the addition of this much mass adversely affects the fuel economy of the vehicle. Therefore, it can be concluded that the addition of this system to a medium truck will not provide an economic advantage.

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Appendix 1: SIMULINK model input data table

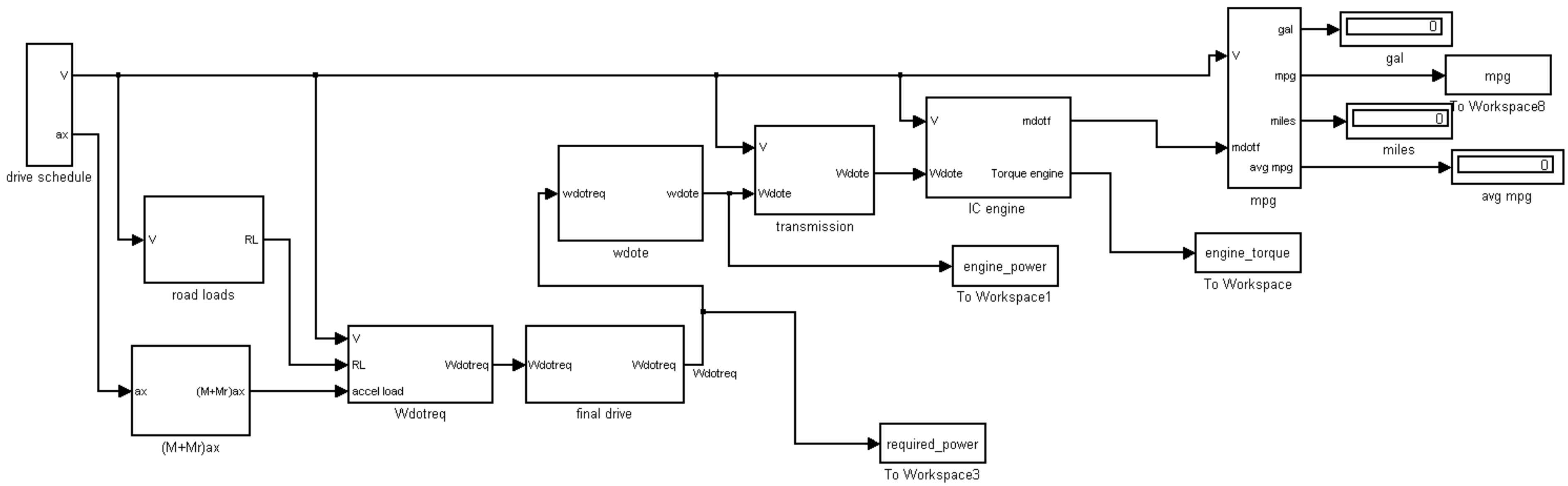
Table 1: Input Data[20, 21]

vehicle specifications	
vehicle mass: M (kg)	10340
radius of vehicle wheel: rw [m]	0.4131
Transmission :	
Transmission - 1st gear ratio: N1	3.45
Transmission - 2nd gear ratio: N2	2.24
Transmission - 3rd gear ratio: N3	1.41
Transmission - 4th gear ratio : N4	1
Transmission - 1st Gear efficiency :η_1	0.9893
Transmission - 2nd Gear efficiency :η_2	0.966
Transmission - 3rd Gear efficiency :η_3	0.9957
Transmission - 4th Gear efficiency:η_4	1
Propshafts/Differential - Differential drive ratio N_f	3.21
Propshafts/Differential - Differential efficiency :η_f	0.96
Hydraulic systems	
pump/motor ratio: Nh=2	2
pump/motor efficiency: η_h	0.9
pump displacement:(m^3) Dmax	$35X10^{-6}$
pump/motor torque efficiency: η_t	0.95
pump/motor volumetric efficiency: η_v	0.95
gas system	
air tank volume [m^3]: vt	2
initial pressure of air tank [Mpa]: P_{max}	50
residual air pressure in air tank [Mpa]: P_{min}	40.6
the gas constant of air [J/(kg.K)]: Rair	287
the air specific heat of constant volume [J/(kg.K)]: cvair	716
accumulator full fill air volume [m^3]: va	0.06
compressor displacement [m^3]: vcs	$4X10^{-3}$
specific heat ratio: k	1.4
the gas constant of Nitrogen [J/(kg.K)]: Rconst	296.8
The Nitrogen specific heat of constant volume [J/(kg.K)]:cv	743

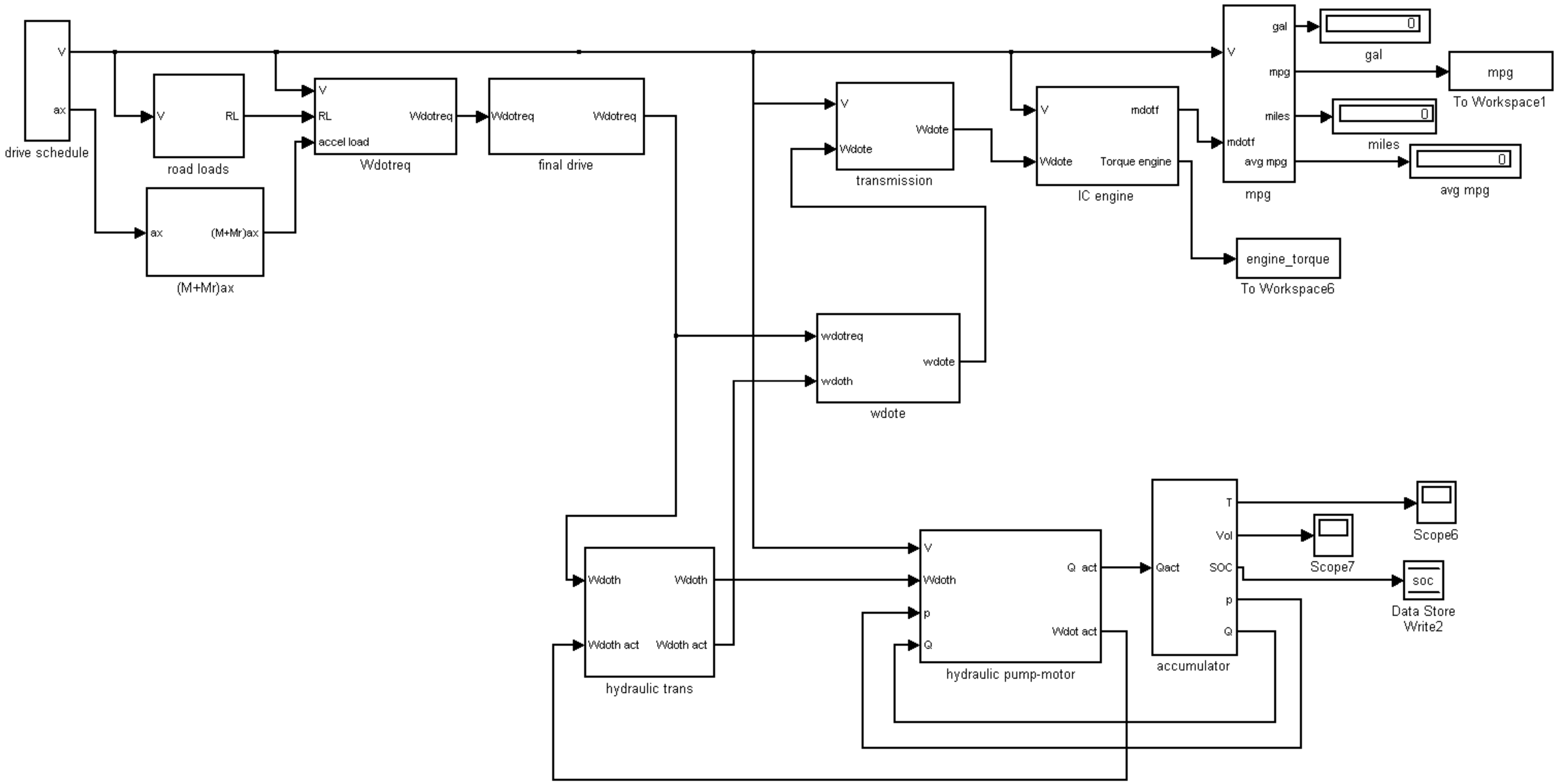
Table 2: DI Diesel Engine Specifications [20,21]

Configuration	V8, Turbocharged, Intercooled
Displacement [L]	7.3
Bore [cm]	10.44
Stroke [cm]	10.62
Connecting Rod Length [cm]	18.11
Compression Ratio :r_c	17.4
cutoff : r_{cut}	2
combustion efficiency: η_c	1
Rated Power [HP]	210HP @ 2400 rpm 520lb-ft@1500rpm
heating value of diesel QLHV [J/kg]	43000000
fuel density: ρ_f [kg/m3]	800

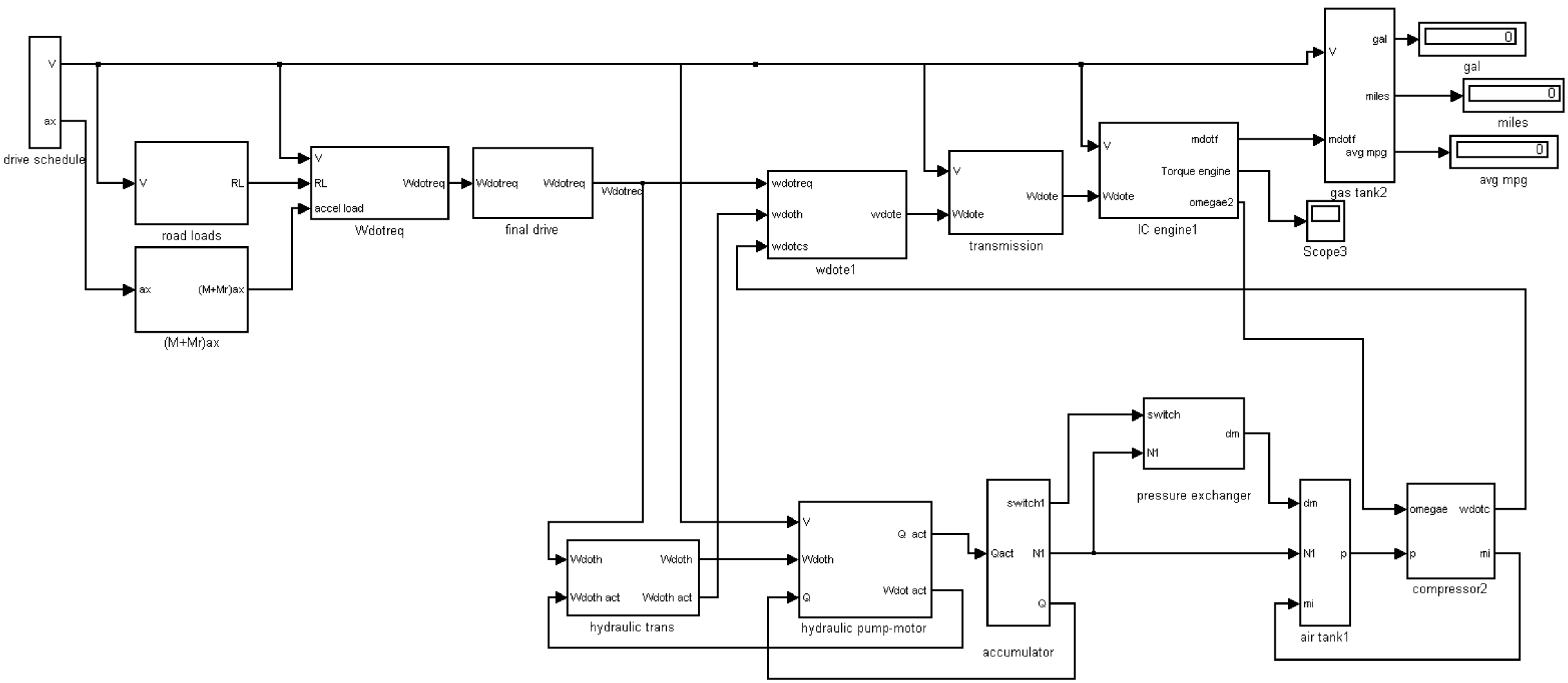
Appendix 2: Conventional Vehicle MATLAB/Simulimk Model



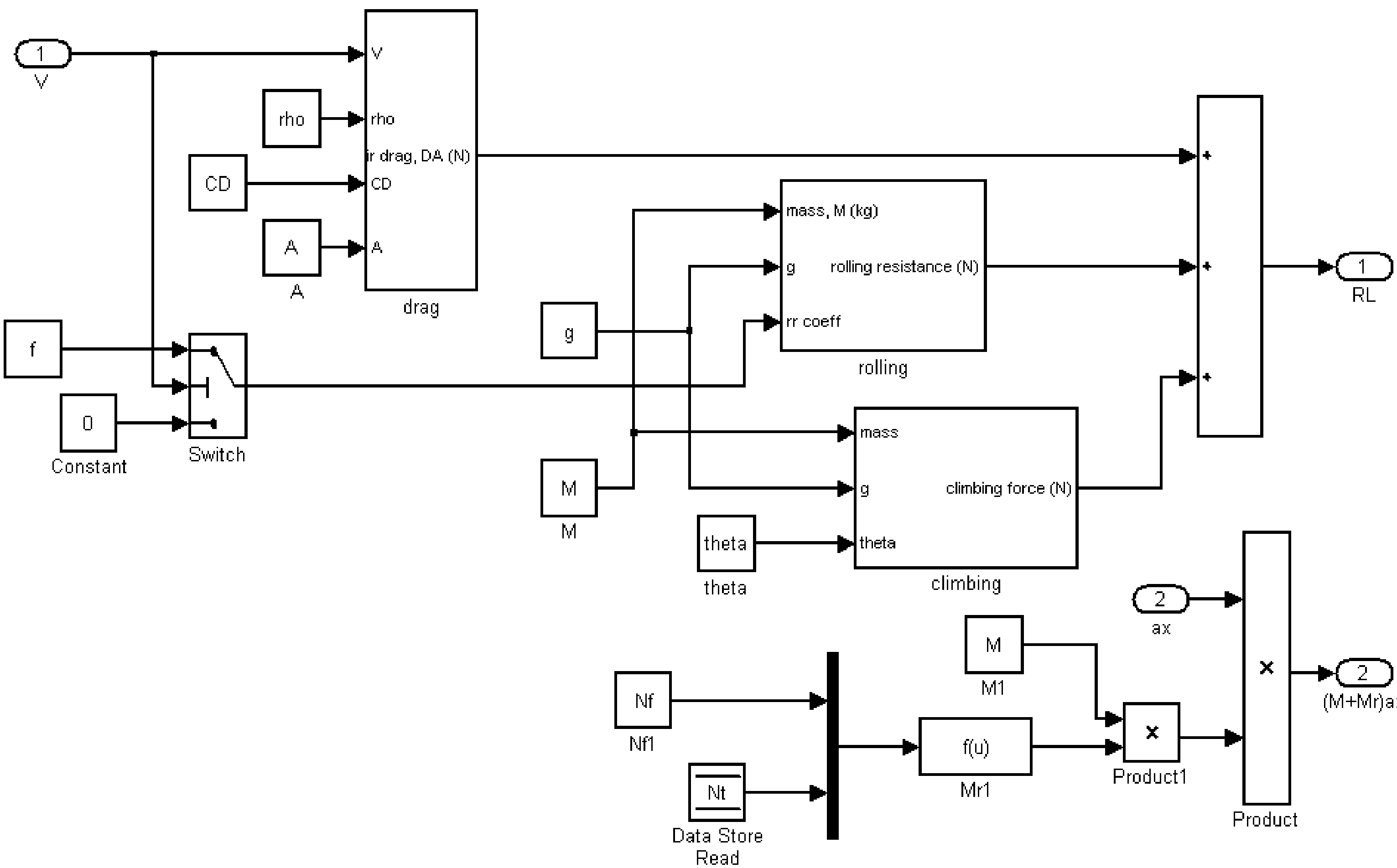
Appendix 3: Parallel Hydraulic Hybrid System MATLAB/Simulimk Model



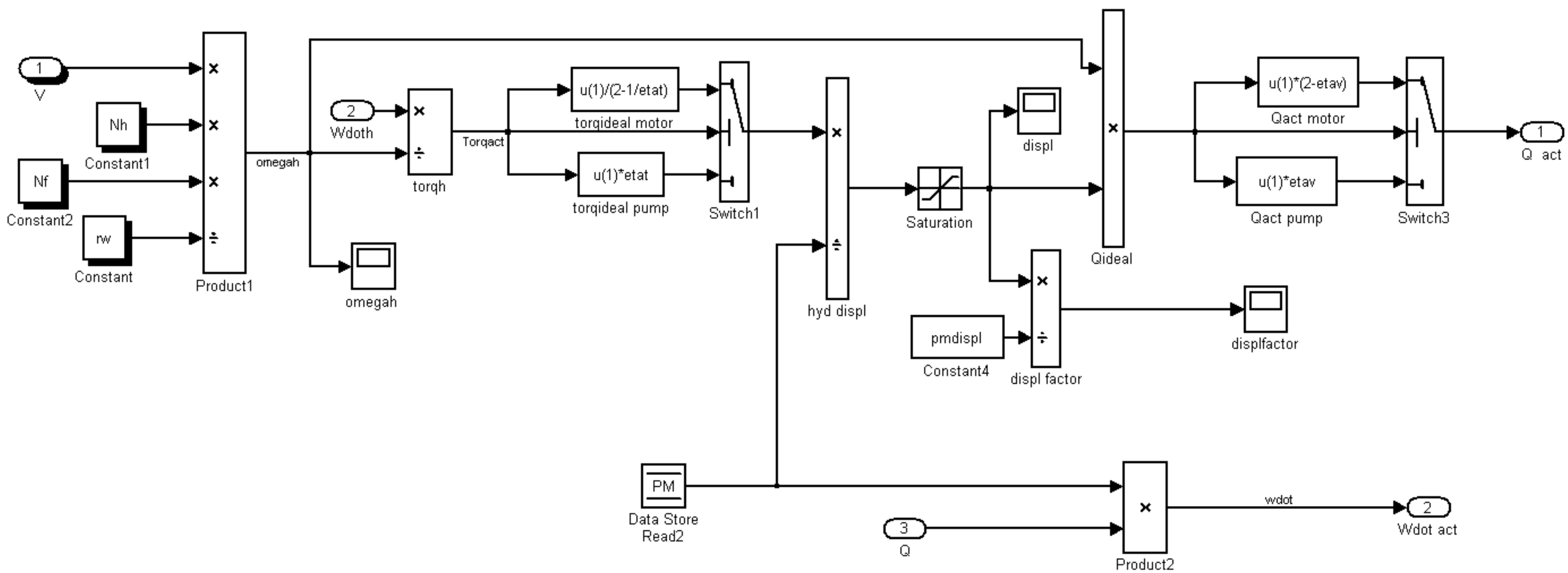
Appendix 4: Proposed Parallel Hydraulic Hybrid System MATLAB/Simulimk Model



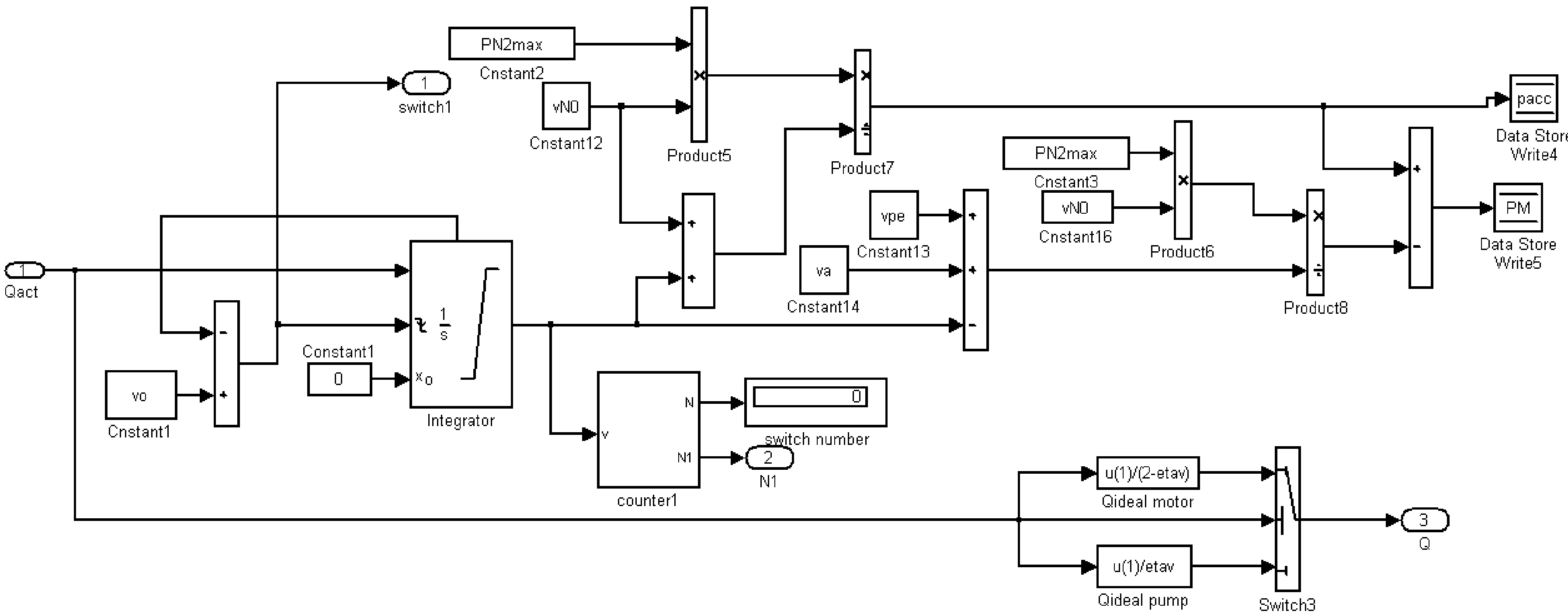
Appendix 5: Vehicle Dynamics Model



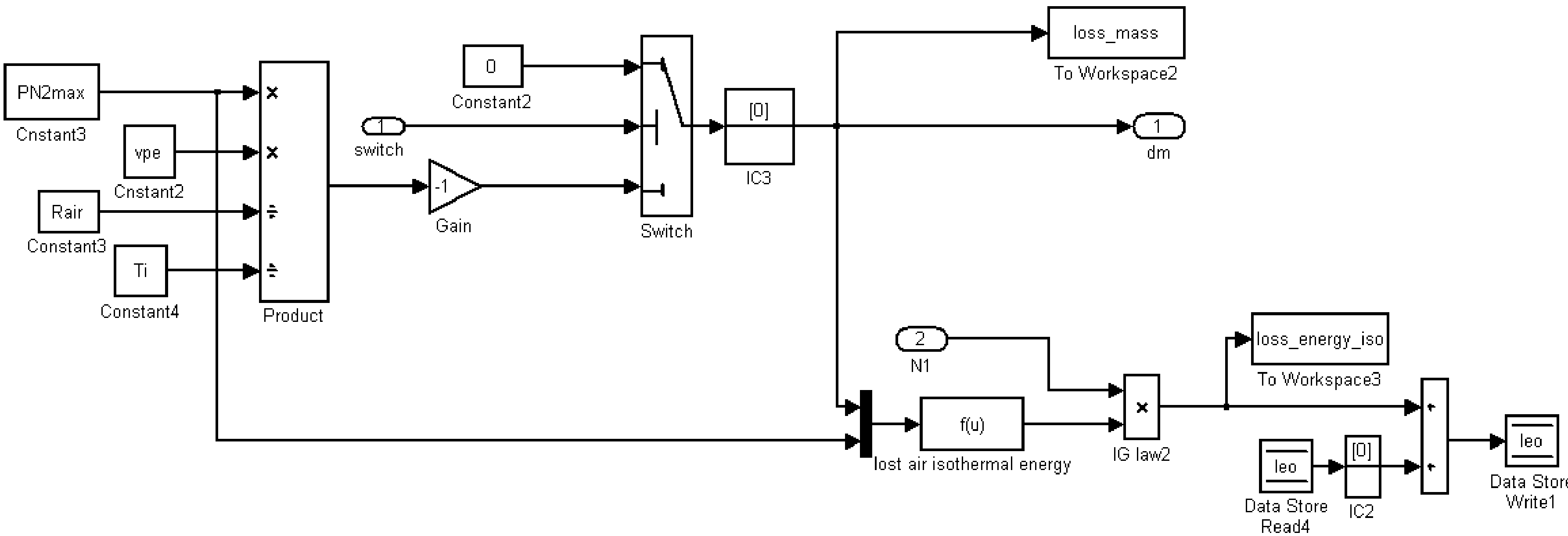
Appendix 6: P/M Model



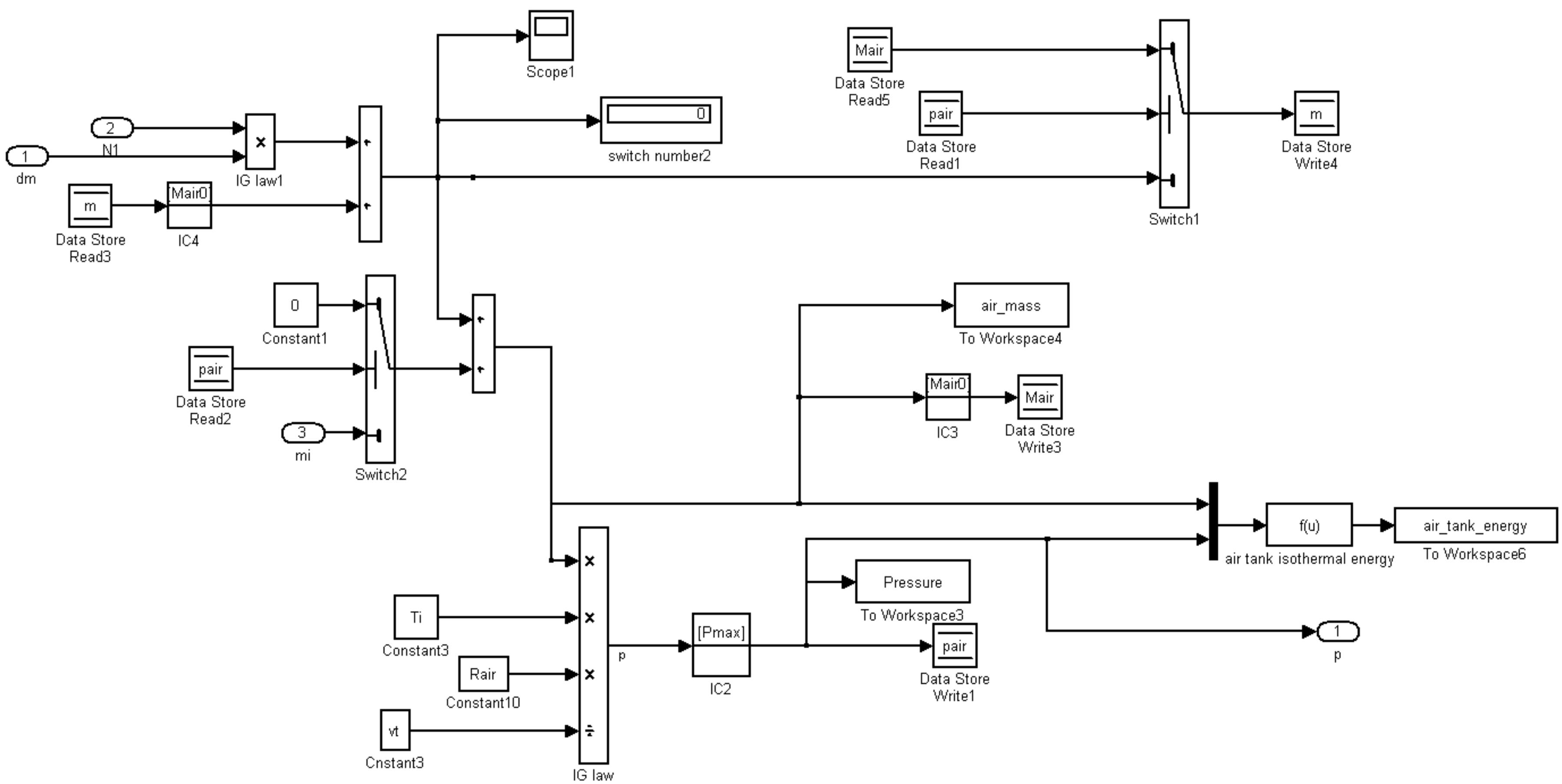
Appendix 7: Accumulator Model



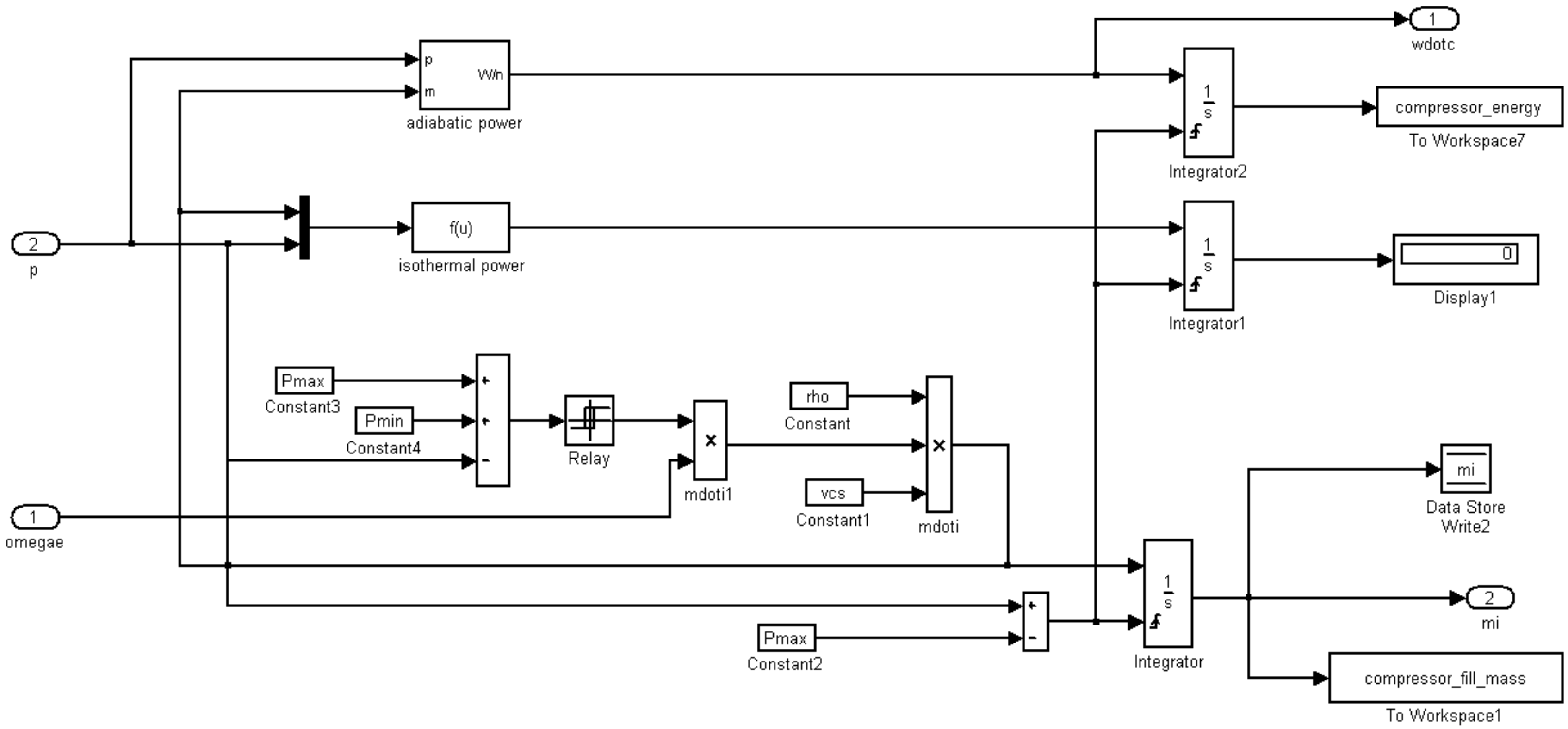
Appendix 8: Pressure Exchanger Model



Appendix 9: Air Tank Model



Appendix 10: Compressor Model



Appendix 11: Engine Model