Hydraulic hybrids

PRITTHVIRAJA A. JAIPAL
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Abstract

Conventional IC Engine powertrain layouts in vehicles tend to consume considerable amounts of fuel and generate emissions that are harmful to the environment. Newer technologies have enabled the development of sustainable vehicle layout designs that favor a reduction in the exhaust emissions and energy consumption without compromising the vehicle’s performance. Hybrid vehicles and electric vehicles are the torch bearers for this development. Although electric vehicles feature high performance at lower emissions, they are generally limited by their range and high battery costs. Development of countries to equip for this electrification is another important factor here as with the development of more electric vehicles comes the problems associated with charging, like charging stations, charge scheduling (from power grid because of the high toll), etc.

Hybrid electric vehicles are energy efficient and reduce the emissions considerably but their costs are substantially higher. Along with the higher efficiency generated by the electric machines, the possibility of regenerating braking energy reduces the energy consumption and increases the energy efficiency of the conventional layouts. Hydraulic hybrids in the recent years have gained recognition for their advantages and are known for being the cheaper alternative for hybridizing heavy vehicles. The ability of storing regenerative braking energy in this fluid form allows for higher cyclic efficiency when compared to that of the electrical means of storing energy.

This thesis focuses on the design and modeling of the hydraulic hybrids using MATLAB/SIMULINK® to construct models depicting the use of the vehicles under the selected drive cycles. Regenerative braking has been one of prime focus for improving the range and minimising the energy consumption of the vehicle along with high operational efficiencies of the operating components.

The thesis takes into account two cases, one with the case of a medium duty vehicle with a conventional IC Engine layout and the other with the case of electric forklifts. The two cases are compared with their hydraulic hybrid layouts along the lines of energy consumption, operational efficiencies and range. Through the design of these simulations, a comparative analysis of the hydraulic hybrid to the electric hybrid is provided for the case of the medium duty vehicle and the benefits of having an electric hydraulic hybrid layout designed for the electric forklift applications are studied.
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1. Introduction

Conventional vehicle layouts with the IC engine are widely developed and well fabricated for use with various applications; however, emissions and fuel source issues are driving vehicle development into a sustainable platform in almost all vehicular applications. Hybrid vehicles and electrification of vehicle layouts are effective ways to pose as a solution for the need of this sustainable development. Most manufacturers have adopted hybrid electric vehicle layouts in order to improve vehicle fuel economy and cost. Research on this development has further led to development of electric vehicles to decrease the dependability on fuel sources while increasing the efficiency of operation in vehicles. Though the development of electric vehicles looks like a promising and viable option for the future, there are still a few hiccups along the way that might interrupt the smooth transition of replacing all fuel based systems with the electric powertrain.

Research has shed light on another means of hybridization, the hydraulic hybrid vehicle. The hydraulic system provides large power capacity and at first glance is suitable for hybridizing heavy duty vehicles. Compared the electric means of hybridization, the hydraulic subsystems are known to recapture more energy during braking which makes even the harshest driving conditions, i.e. with frequent start-stops, operation in urban driving applications more efficient in recuperating energy. These systems have a direct effect on the range and the operation efficiency of the vehicle.

Regarding the transmission of power to the wheels, the transmission provided by the hydraulic subsystems achieves similar characteristics as the conventional manual or automatic transmissions in vehicles with continuously variable speed functions. In principle, the hydraulic pumps convert mechanical energy to fluid hydraulic energy and the hydraulic motors convert the high pressure hydraulic energy to mechanical energy to drive the wheels. The valves on the hydraulic motor and pump allow the continuous variable speed functions and broad gear ratios.

There has been plenty of research done for manual transmissions vehicles, Series and Parallel electric vehicles and electric vehicles. Growing interests by many institutions and manufacturers has led to development of various Series and Parallel hydraulic vehicles in the recent years. However most of the research has focused on the improvement of energy management strategies of individual components such as Hydraulic pumps and motors, batteries, PMSM motors, etc. for certain applications. Energy control strategies have been researched only with insight on improving the developed systems that can be applied only to the particular vehicle layout that has been tailor fitted according to their baseline designs. There has been some research comparing the energy efficiency of the different configurations of hydraulic and electric hybrids.

Through this thesis, the hydraulic systems would be compared to the electric systems through the terms of application. The technical comparisons between the two systems would here be tested on the same platform via simulations using MATLAB/SIMULINK ®. The comparisons mainly look into the potential benefits of the adoption of the two systems for hybridizing the conventional IC engine layout in order to increase the operating efficiency of the overall vehicle while increasing the range and reducing the
energy/fuel consumption. With the development of electric vehicles in the last few years, the research also looks into the effectiveness of having an electric hydraulic hybrid layout in order to reap further benefits of the electric powertrain and reduce its disadvantages.

1.1 Scope and indirect benefits

The design of the simulations would tackle only vehicle propulsion and the benefits that the various layouts have to offer. Energy consumption and increase of vehicular range was an emphasis throughout these simulations and modelling of a characteristic high efficiency system with high fidelity is of the utmost priority. With the vehicle application in mind, vehicle layouts were established with segregation of the two simulation tests, one with the convention IC engine and one for an Electric vehicle layout. The simulations were modelled on MATLAB/SIMULINK under the full vehicle layout in order to receive realistic results.

The modelling of the vehicles has a focus only upon the required components for propulsion, hence auxiliary loads and additional power needs have not been included within the scope. The simulations make use of appropriate drive cycles for the selected applications in order to depict the driving style for the application to avoid region/country specificity.

Some of the indirect benefits of the systems that are not part of the scope of the thesis are:

- The initial costs for the hydraulic hybridization are significantly lower when compared to electric hybrids due to high costs of electric machines and especially batteries.
- The hydraulic hybrids would reduce brake pad wear and increase the life span of the brake pads with reduction of wear due to lesser use of conventional frictional brakes.
- For the case of electric hydraulic hybrid, it could be generalized that the electric battery would have an increase in life span. This would be enabled because of the reduction of charging and discharging of the batteries, not only during acceleration but also during braking as the regenerated braking energy would be stored in the accumulator.
- The thesis does not take into account the variation in emissions from using the hydraulic systems over the electric systems.

1.2 Background and related courseware

The thesis makes use of literature on the various hybrid vehicle developments in both electric and hydraulic means of hybridization. The studies based on configurations, layout design, control theory, and electric and hydraulic machine drives are important sources for information on the design and modeling of the simulation components. The implementation of the vehicle, application wise, along with vehicle dynamics are the required sources for developing a fully functional simulation for the vehicle. Since the simulation was achieved using MATLAB/SIMULINK, various information and literature helped in the creation of different layouts with ease and provided flexibility with the development of the simulations.

For this thesis, apart from the literature work available from the internet, certain courses more than the others were beneficial for the setup and implementation of the simulations. These were courses ranging electrical to machine design to vehicle engineering from the different departments of KTH. The literature
on electrical machines from the Electrical Machines and Drives course was crucial for the development of the PMSM motors in the simulation. The literature and simulations from the Hybrid Vehicle Drives was helpful for the implementation of the vehicle and its layout and configuration. Literature from the Internal Combustion Engines course was helpful for the generation and control of the power characteristics of the modeled IC engine blocks. The various Vehicle Dynamics courses helped in the generation of the vehicle dynamics block that makes the simulation even more complete.

1.3 Hydraulic hybrids

Hybridizing heavier vehicles with hydraulics is one of the viable options to improve the fuel economy of the vehicle. Regenerating and reusing significant amounts of the kinetic braking energy associated with their use in the city, characteristic of their drive cycles, coupled with high system efficiencies enables heavier vehicle architectures to benefit in terms of fuel economy. With higher mass also comes higher power requirement. The hydraulic propulsion and storage subsystems are known for their higher power density in comparison to its electric counterparts. Compared to batteries, the energy storage device in the hydraulic subsystem, the hydraulic accumulator has a significantly higher ability to accept high rates and high frequencies of charging and discharging. It would require several hours to recharge the high energy density batteries. Even under fast charge, the battery cannot be fully charged at a high rate which could by means other than propulsion reduce the range of the vehicle by at least 10%.

When compared to the electrical hybrid system, the HEV cyclic efficiencies have a significant difference which can be seen in Figure 1.

![Figure 1. Efficiencies while braking/accelerating electrically versus hydraulically](image)

One of the major contributions of the higher efficiency in the cycle allows higher amount of regenerative capabilities which is harnessed via the regenerative braking systems. The fuel economy improvement in hydraulic hybrid vehicles is associated with their use and their associated driving cycle. A larger improvement can be seen with worse driving conditions, i.e. more frequent start-stop driving cycle. Shuttle buses, garbage trucks, delivery trucks, etc are some of the vehicles that could benefit with this kind of hybridization.
Hydraulic hybrids also pose as a sustainable option enabled by the fact that the hybrids would pollute less with the engine operating less frequently and optimum efficiency. Hydraulic technology has been around for more than 100 years resulting in lower acquisition and maintenance costs.

Hybridizing of the vehicle hydraulically can be done in two ways, Parallel Hydraulic Hybrid and Series Hydraulic hybrid shown in Figure 2.

![Figure 2. Hydraulic hybrid layouts(2).](image)

The parallel hydraulic hybrid is the most basic layout. It uses one hydraulic pump/motor to provide energy in conjuncture with the IC engine during motoring operation and recover regenerative braking energy in its pumping operation.

The series hydraulic hybrid disconnects the conventional driveline and uses two drive pump motors to drive the vehicle. The primary motor drives the vehicle and regenerates braking energy during pumping operation. The engine pump operation uses the engine’s energy to recharge the accumulators SOC and during its motoring operation acts as a starter for the IC engine.

Electric hybrids have gained success in the parallel layout because current electric technologies cannot handle full power for heavier vehicles. Technologies today have enabled hydraulic machines to harness way higher power per kilogram when compared to electric machines. This allows greater benefits to series hydraulic hybrids over parallel hydraulic hybrids.

**Available technologies**

New developments in the hydraulic hybrids can be seen by various manufacturers. Artemis has developed a direct drive hybrid system that is directly connected to the driving wheel axle without the need of any gearbox.
The hydrostatic machines in the series hydraulic hybrid layout are Artemis’ Digital Displacement motor/pump. Artemis claims that the maximum power which can be transferred here is 200kW. The overall operating efficiency is stated as 85%. This series hybrid system was implemented with success in a BMW 530i shown in Figure 4. With the NEDC driving cycle the fuel consumption was reduced by 27% and 50% for the EUDC driving cycle. The results mainly reflect reduced mechanical losses, regenerative braking with high efficiency and optimal operation (max efficiency) of the IC engine.

Another example of hydraulic hybrid system is by the manufacturer Eaton and the Eaton Hydraulic Launch Assist system is a parallel hydraulic hybrid system.
The Eaton system in Figure 5 focuses on harnessing regenerative braking energy and uses this energy, stored by the high-pressure accumulator, to improve the acceleration of the truck. Test data revealed a reduction in fuel consumption by 17% improvement in performance mode which boosted the acceleration by 26% and 28% fuel reduction in economy mode.

Bosch-Rexroth Hydrostatic Regenerative Braking system is a hydraulic system designed to be fitted in commercial vehicles without much cost or effort. Just for an add-on system they promote a boost in fuel consumption reduction by 25%. The Bosch system works well with commercial vehicles because of its heavy vehicle segment and its high frequency of stop and go drive cycles as in Figure 6.

Parker Hannifin designed one of the first parallel hydraulic hybrids and soon after came up with the first series hydraulic hybrid. The Cumulo Brake Drive developed in 1982 and the Cumulo Hydrostatic Drive are shown in Figure 7.
Parker has continued with further development of Advanced Series Hydraulic Hybrid system shown in Figure 8. The advanced series hybrid is based on a power split concept with one hydraulic path and one mechanical path in parallel. At low speed the vehicle is propelled using hydraulic energy and at high speed the mechanical path is used to power the wheels, disconnecting the hydraulics. Disconnecting the hydraulic system from the mechanical path at high vehicle speed is necessary because of the low loading torque at high speeds resulting in low efficiencies of the hydraulic motor.

Figure 8. Parker advanced series hydraulic hybrid (7) and Figure 9. Schematic layout of the split hybrid (2).

### 1.4 Electric hybrid vs. hydraulic hybrid

Hybrid vehicle research in the recent years has focused on the electric subsystems based on the fact that higher efficiencies are obtained with ease of control with electric power. The clean source of electric power has multiple advantages and goes in sync with the development of sustainable vehicles for the future.

One of the differences in the layout of the driveline between the two subsystems is that in the hydraulic hybrid layout, the gearbox between the hydraulic motor and driving wheel is unnecessary. This brings lesser mechanical losses in the driveline. More importantly, this means lesser losses in harnessing regenerative braking energy. Having this as an advantage can be crucial during design phases as losses in the energy transfer both during accelerating and braking can be minimized.
1.4.1 Energy recovery

To calculate the energy consumption for a vehicle, the number of components connected and the efficiencies of each component play an important factor for achieving higher energy recovery.

\[ \frac{E_b}{E_d} = \eta_{es} \cdot \eta_{pc}^2 \cdot \eta_{mg}^2 \cdot \eta_{p/m}^2 \]

Equation 1

Where,

- \( \eta_{es} \) = Round trip efficiency of the energy storage device
- \( \eta_{pc} \) = Control power efficiency
- \( \eta_{mg} \) = Mechanical gear efficiency
- \( \eta_{p/m} \) = Pump/Motor efficiency

Upon calculation, the value for the energy recovering efficiency is 53% for electric system and 69% for the hydraulic system. The values for the individual components of efficiencies however vary with the speed of operation. Hence these values are recorded values of component efficiency at a particular instance. The higher energy recovering efficiency potential of the hydraulic systems is because of the higher component efficiencies for the energy storage devices and also because of the lower mechanical losses generated by lower gear ratios.

According to Parker Hannifin in their Cumulo systems, the hydraulic accumulators show case a 94% round trip efficiency whereas the reported round trip efficiency for a Li-ion battery is about 81% (90x90%)(8).
1.4.2 Electric machines vs. hydraulic machines

Electric machines by manufacturers UQM and Toshiba have been globally sold and have known to have motor designs for various applications in sizes up to 150kW.

Toshiba’s permanent magnet motor with a nominal power rating of 38kW has a maximum efficiency of 97%. High maximum efficiencies similar to Toshiba’s motors are achieved by other manufacturers as well. However, this range of max efficiency is limited. In comparison, the hydraulic motors have a higher efficiency rating at lower speeds. This permits a different design approach for different vehicle applications. The Artemis Digital hydraulic motor is designed to meet the requirements on high overall efficiency even at partial loading.

Comparison of a 100 kW UQM PowerPhase motor and the Artemis digital hydraulic motor is shown in the table below.

<table>
<thead>
<tr>
<th>Performance (max power: 100 kW)</th>
<th>El-motor</th>
<th>Hyd-motor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power density</td>
<td>UQM PowerPhase</td>
<td>Artemis DD</td>
</tr>
<tr>
<td>Efficiency at 20% load</td>
<td>90%</td>
<td>93%</td>
</tr>
<tr>
<td>Cost</td>
<td>1 / kW</td>
<td>0.3 / kW</td>
</tr>
</tbody>
</table>

The requirements for the design of a motor apart from torque and speed characteristics depend on efficiency, size, and weight and in order to compete with the research on electric motors, these are important factors of consideration. With hydraulic machines, axial piston machines with variable displacement are machines that can be controlled easily but their efficiency at low displacement settings is not impressive. The figure below, Figure 11, shows the variation of efficiency with speed for variation in displacement from 0.25 to 0.5 to 1.0.

From the figure, it is easy to see the efficiency variation at low displacement. The variation drops to about 72% from 90% for the in line hydraulic motor at constant power of 90 kW. This makes the creation of the hydraulic systems hard.

Research in the last few years has allowed development of newer hydraulic machine technologies which surpass the efficiency of the earlier developed machines. The Digital Displacement Pump/Motor from Artemis and Floating Cup from Innas(9) are few of the promising concepts that not only boast higher operating efficiencies but also have lesser hydro-mechanical loses and idling loses.
1.4.3 Hydraulic accumulators vs. batteries

An important improvement of hydraulic accumulators over the years is the development of light weight accumulators, something which Parker systems have been working on constantly. Table 2 shows the comparison of the two storage systems used in the hydraulic and electric hybrids respectively.

Table 2. Comparison of storage systems.

<table>
<thead>
<tr>
<th>Energy recovering performance</th>
<th>Hydraulic accumulator</th>
<th>Electric battery, Li-ion</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power density</td>
<td>5 kW/kg</td>
<td>0.5 kW/kg</td>
</tr>
<tr>
<td>Energy density</td>
<td>4-11 kJ/kg</td>
<td>150 kJ/kg</td>
</tr>
<tr>
<td>Round-trip effic.</td>
<td>94%</td>
<td>81%</td>
</tr>
</tbody>
</table>

Characteristic advantages here of the hydraulic accumulators include their high power density with the high efficiency of operation. This has a direct effect on the energy recovering potential when compared to batteries. The only advantage of electric batteries is the high energy density, which enables the development of full electric vehicles. This however is limited only to light weight vehicles. For a heavy truck about 20 tons of battery weight would be required for 1000 km of driving (10). In the case of heavier vehicles, the energy savings potential is an important aspect for design, characteristic of their frequent braking in their driving cycles. Greater possibility of energy recovery is brought about by stronger breaking and increased mass. Hydraulic accumulators have an inherent advantage here because
of the quicker round trip cycle time in the hydraulic system with the acceptance of higher power densities at high efficiency.

### 1.5 Related work: Electric hydraulic hybrids

There is a lack of research done on this segment of having electric as well as hydraulic systems propel vehicles. Upon searching the internet for related works, two different reports were found that dealt with a similar parallel electric hydraulic hybrid layout. The first, a paper published by Xiaobin Ning and Yangyan Guo on decrease of energy consumption for an electric bus (11). The paper looks into developing a simulation to target the regenerative braking energy of an electric bus and a means to increase the regenerative energy by introducing the hydraulic energy recovery system in parallel. The testing was done under the ECE-15 drive cycle and the results were compared. The results showcased an improvement of 11 km on the single charge range of 145 km. The system tested to have an increase in 8.2% in range. Figure 12 shows the variation of depth of discharge generated by the results.

![Figure 12. Depth of discharge for battery(11).](image)

Another paper, (a journal entry under mdpi.com/energies) published by Jia-Shiun Chen compares the energy efficiency comparison between hydraulic hybrids and hybrid electric vehicles (12). This paper tests the simulations of various layout configurations of a 1500 kg vehicle (simulating a car). One segment of this paper compares the electric and electric hydraulic hybrid vehicle under the NEDC and EUDC drive cycles. The energy consumption reduction reported in this paper was 23.38% for NEDC and 4.34% for EUDC.

In new systems development, KersTech Inc. has devised a setup for the hydraulic system in electric vehicles. The company has developed an electric-hydraulic motor called the TwinTorq to employ hydraulic systems for low speeds and the electric power for higher speeds(13).
Figure 13. KersTech operation (13).

Figure 14 shows the efficiency curves of typical hydraulic and electric motors and according to the company, the resultant higher efficiencies and reduced demand on the battery could increase the driving range by 45% depending on the drive cycle.

Figure 14. Combined efficiencies KersTech (13).
2. Subsystem modeling

This section looks into the modeling of the hydraulic and electric subsystems required for the simulation setup. The modeled blocks have been executed using MATLAB/SIMULINK ©.

2.1 Hydraulic hybrid subsystem modeling

2.1.1 Hydraulic accumulator

The hydraulic accumulator is an energy storage device that acts as a power source, functions the same way as batteries in electric vehicles. The hydraulic hybrid subsystem requires a both high-pressure accumulator and a low-pressure reservoir for its application. The low-pressure reservoir here helps to avoid cavitations in the hydraulic pumps. It is important here to have a difference in pressures between the accumulator and the reservoir as the pressure difference and the pump/motor displacement determines the torque of the pump/motor.

The storage device is a hydro-pneumatic accumulator and is split into the inert gas chamber and a fluid chamber (14)(15). The chambers are separated by a bladder with mixed with flexible open celled foam mixed with the inert gas to increase the thermal capacity and reduce heat losses. The energy is stored in inert gas chamber, usually compressed nitrogen gas and is stored inside a movable barrier which can be in the form of a rubber bladder or metal lined plastic bags. This separates the gas from the oil in the fluid chamber while being all snug inside the accumulator (16)(17).

In Figure 15, the labels: -

1: Elastic Bladder barrier (bl); 2: Nitrogen gas mixed with polyurethane foam; 3: Accumulator shell (sh); 4: Hydraulic fluid (Oil); 5: Oil input/output; 6: Nitrogen charging port
In order to design the hydro-pneumatic energy storage unit, its volume and sizing needed calculation. The accumulator, in Figure 16, consisting of the movable bladder barrier that stores the nitrogen gas has a bladder volume defined by:

\[
V_{bl} = \frac{4}{3} r_{bl}^3 \pi + r_{bl}^2 \pi h_{bl}
\]

Equation 2

Here the ratio of length and radius is introduced as:

\[
a_{bl} = \frac{h_{bl}}{r_{bl}}
\]

Equation 3

Also, the bladder’s surface area:

\[
A_{bl} = 4r_{bl}^2 \pi + 2r_{bl} \pi h_{bl}
\]

Equation 4

From the previous equations, bladder surface area now:

\[
A_{bl} = \left( \frac{V_{bl}}{\pi \left( \frac{4}{3} + a_{bl} \right)} \right)^{\frac{2}{3}} (2 + a_{bl})
\]

Equation 5

The accumulator shell surface area and volume similarly:

\[
V_{sh} = \frac{4}{3} r_{sh}^3 \pi + r_{sh}^2 \pi h_{sh}
\]

Equation 6

\[
A_{sh} = 4r_{sh}^2 \pi + 2r_{sh} \pi h_{sh}
\]

Equation 7

The energy conservation equation for inert gas (nitrogen) inside the bladder is expressed as:

\[
m_g \frac{du}{dt} = -p_g \frac{dv}{dt} - m_f c_f \frac{dT}{dt} - k A_{bl}(T_g - T_{oil})
\]

Equation 8

Where,

\( m_g \) and \( m_f \) are nitrogen gas mass and foam gas mass respectively,
\( k \) is global heat transfer coefficient,
\( T_g \) and \( T_{oil} \) are nitrogen gas and oil temperatures respectively,
And \( c_f \) is the specific heat capacity.

Now we need to define the pressure of the stored nitrogen gas in the accumulator. Although the adiabatic equation is helpful to understand relative sizing of the components, it does not take into account the thermal losses and deviation of the gas from an ideal gas situation. In order to do justice to the design of
the accumulator and model the overall dynamics well, the Benedict-Webb-Rubin equations of state for real gas is used (18). The BWR equations are derived from both fundamental thermodynamics and empirical data sets.

\[ p_g = \frac{RT}{v} + \left( B_0RT - A_0 - \frac{C_0}{T^2} \right) + \frac{(bRT - a)}{v^3} + \frac{a\alpha}{v^6} + \frac{c\left(1 + \frac{\gamma}{v^2} \right)e^{\frac{-T}{v^2}}}{v^3T^2} \]  
Equation 9

Where the equation coefficients:
\[ A_0 \text{ is } 136.0436 \]
\[ B_0 \text{ is } 1.454397 \times 10^{-3} \]
\[ C_0 \text{ is } 1.040558 \times 10^{-6} \]
\[ a \text{ is } 1.156984 \times 10^{-1} \]
\[ b \text{ is } 2.966165 \times 10^{-6} \]
\[ c \text{ is } 7.3806 \times 10^{-5} \]
\[ \alpha \text{ is } 5.786149 \times 10^{-9} \]
\[ \gamma \text{ is } 6.753738 \times 10^{-6} \]

The differential equation here that determines the nitrogen gas’ temperature:

\[ \tau = \frac{m_g c_v}{k_{bl}} \]

Here; \( \tau = \frac{m_g c_v}{k_{bl}} \)

The above equation represents the nitrogen gas constant (14). In order to increase the efficiency of the accumulator in the system, elastomeric foam added to the gas helps to increase the thermal capacity of nitrogen, which in turn reduces the heat losses. The oil’s temperature can also be determined here as the equation below takes into account the heat transfer between the nitrogen gas and the hydraulic oil, dissipated heat based on the ambient temperature from the mechanical friction of the hydraulic pump/motor.

\[ m_{oil}c_v \frac{dT_{oil}}{dt} = k_{bl}A_{bl}(T_g - T_{oil}) - k_{sh}A_{sh}(T_{oil} - T_{amb}) + Q_{hyd} \]  
Equation 11

Where, \( k_{bl} \) and \( k_{sh} \) are heat transfer coefficients for oil and nitrogen and nitrogen and ambient respectively.

The State of Charge (SOC) for the hydraulic accumulator is defined on the ration between instantaneous volume of fluid inside accumulator and the maximum shell volume (18). The linearity of the SOC depends on the ratio of the volumes. Here,

\[ SOC = SOC(r) \]  
Equation 12
Also;

\[ SOC(r_{\text{max}}) = SOC_{\text{max}} \]

And

\[ SOC(r_{\text{min}}) = SOC_{\text{min}} \]

While the ratio, \( r \):

\[ r = \frac{V}{V_{\text{max}}} \]

Equation 13

Based on the selection of the minimum and maximum SOC values, the ratio varies from the minimum to the maximum SOC of the accumulator.

Here another important factor that needs addressing is the Initial State of Charge:

\[ r_{\text{initial}} = r_{\text{min}} + \left( \frac{r_{\text{max}} - r_{\text{min}}}{SOC_{\text{max}} - SOC_{\text{min}}} \right) \times (SOC_{\text{initial}} - SOC_{\text{min}}) \]

Equation 14

Initially the volume of nitrogen gas:

\[ V_{N_2(\text{initial})} = V_{\text{max}} - r_{\text{initial}}V_{\text{max}} \]

Equation 15

The hydraulic subsystem now has to have an additional container to store the fluids when not in the accumulator. The Low-pressure reservoir gets its name here as it stores this fluid at significantly low pressures and in turn provides sufficient inlet pressure for the hydraulic pump while avoiding cavitations (15). Figure 17 shows the accumulator block modeled in Simulink.

Figure 17. Accumulator model.
2.1.2 Hydraulic pump and motor

The bent axis type piston pump and motor is selected for the modeling and simulation of the hydraulic pump and motor due to its variable displacement and high efficiency characteristics. The motor and pump is connected to the high-pressure accumulator and the low-pressure reservoir. During pumping, the mechanical energy is supplied to the pump shaft to increase the pressure in the high-pressure accumulator by transferring hydraulic fluid from the low-pressure reservoir. During its operation as a motor, it converts hydraulic energy to mechanical energy by reducing the pressures in the high-pressure accumulator and supplies this to the drivetrain. Ideally, the volumetric flow rate and the torque expressions can be defined as (14):

\[ Q_i = x\omega D \quad \text{Equation 16} \]

\[ T_i = x\Delta p D \quad \text{Equation 17} \]

Where, \( x \) is the displacement factor of the hydraulic motor pump, \( \omega \left( \frac{\text{rad}}{\text{s}} \right) \) is the angular velocity, \( \Delta p \) is the pressure difference across the machine and displacement per radian given by \( D \left( \frac{\text{m}^3}{\text{rad}} \right) \).

For the hydrostatic machines to be more accurate, the losses need to be calculated. Based on Wilson’s hydraulic pump theory (19)(20)(21), the overall efficiency of the hydrostatic machine:

\[ \eta_0 = \eta_v \eta_{hm} \quad \text{Equation 18} \]

Where \( \eta_v \) and \( \eta_{hm} \) are the volumetric and hydro mechanical efficiencies of the machines respectively.

The volumetric efficiency has direct dependency on laminar and turbulent losses, compressibility of the hydraulic fluids used and losses due to dead volume (20). The volumetric efficiency for the hydraulic machines can be defined as:

\[ \eta_v(\text{pump}) = \frac{Q_a}{Q_i} = 1 - \frac{C_s}{|x|S} - \frac{\Delta p}{\beta |x|\sigma} - \frac{C_{st}}{|x|\sigma} \quad \text{Equation 19} \]

\[ \eta_v(\text{motor}) = \frac{Q_i}{Q_a} = \frac{1}{1 + \frac{C_s}{|x|S} + \frac{\Delta p}{\beta |x|\sigma} + \frac{C_{st}}{|x|\sigma}} \quad \text{Equation 20} \]

Where, \( \beta \) is the oil’s bulk modulus of elasticity, \( C_s \) is the leakage coefficient, \( C_{st} \) the turbulent leakage coefficient and \( S \) and \( \sigma \) are dimensionless values described by:

\[ S = \frac{\mu \omega p}{\Delta p} \quad \text{Equation 21} \]
\[
\sigma = \frac{\omega_p D \rho^{1/3}}{\sqrt{2\Delta p/\rho}}
\]

Equation 22

Where \( \rho \) is the oil density.

The torque efficiency on the other hand has a direct dependency on the three types of friction, dry friction, viscous friction and the hydrodynamic friction. The dry friction is directly proportional to the load pressure, the viscous friction is proportional to the viscosity and the speed of operation and the hydrodynamic friction is the friction generated by the sealing (20)(14).

The torque efficiency can be defined as (14):

\[
\eta_{t(pump)} = \frac{T_i}{T_a} = \frac{1}{1 + \frac{C_s S}{|x|} + \frac{C_f}{|x|} + C_h x^2 \sigma^2}
\]

Equation 23

\[
\eta_{t(motor)} = \frac{T_a}{T_i} = 1 - \frac{C_s S}{|x|} - \frac{C_f}{|x|} - C_h x^2 \sigma^2
\]

Equation 24

The overall efficiency of the pump/motor:

\[
\eta = \frac{E_{output}}{E_{input}} = \eta_{t(pump)} \times \eta_{t(motor)}
\]

Equation 25

The mechanical and volumetric efficiencies however also depend on pressure difference, displacement ratio and rotational speed other than the individual parameters of the motor/pump. The other parameters are assumed to be constant.

Figure 18. Pump's displacement ratio vs. efficiency and Figure 19. Pressure difference versus efficiency.
After the consideration of the parameters and the figures above, it is easy to conclude that the efficiencies of the pump significantly depend on the related individual variables. According to the first figure, Figure 18 mechanical efficiency is more sensitive to displacement ratio than volumetric efficiency. With increase of the displacement ratio, the mechanical and volumetric ratios increase. It can be concluded hence that pump operation at higher loads generates higher efficiencies.

The pump pressure difference across the two chambers/sections also affects the pump efficiencies as shown in Figure 19. The volumetric efficiency decreases with increase of pressure difference, whereas the mechanical efficiency increases.

In Figure 20 variation of the pump efficiencies with variation to pump speeds, interesting conclusions can be drawn. At higher speeds of operation, the mechanical efficiency decreases when compared to low speed operation and the volumetric efficiency in high speed pump operation is higher than low speed operation. The increase of pump leakages at low speeds pump operation could be a reason for this. Figure 21 shows the modeled pump/motor block in Simulink.

![Figure 20. Pump speed versus efficiency.](image1)

![Figure 21. Pump/Motor model.](image2)
2.1.3 Losses in hydraulic circuits

In order to improve the accuracy of the hydraulic subsystem and its components, the concentrated and distributed pressure drops need to be taken into account. The pressure drop losses are calculated for losses in pressure in the connection of the high-pressure accumulator and the low-pressure reservoir. The connection of these two components can be further segregated into two sections, the high pressure and the low-pressure section respectively.

From Figure 22 the high-pressure section between the high-pressure accumulator and the hydraulic machine can be quantified as $H-1$. The low-pressure section between the hydraulic machine and the low-pressure reservoir can be quantified as $2-L$.

During its motor operation, the fluid is discharged from high pressure accumulator to the low-pressure reservoir, $HtoL$. During the pumping operation, it follows a reverse pattern, $LtoH$. The pressure drops between the two sections:

Pressure drop between H and 1:

$$\Delta p_{1H} = \frac{\rho v_{1H}^2}{2} \left( \varepsilon_H + \lambda_H \frac{L_{1H}}{D_{1H}} \right)$$  \hspace{1cm} \text{Equation 26}

And between section 2 and L:

$$\Delta p_{2L} = \frac{\rho v_{2L}^2}{2} \left( \varepsilon_L + \lambda_L \frac{L_{2L}}{D_{2L}} \right)$$  \hspace{1cm} \text{Equation 27}

The pressure drop hence across the hydraulic motor can be:

$$\Delta p_{\text{discharge}} = p_{HPA} - p_{LPR} - \Delta p_{1H} - \Delta p_{2L}$$  \hspace{1cm} \text{Equation 28}

The pressure drop across hydraulic pump:

$$\Delta p_{\text{charge}} = p_{HPA} - p_{LPR} + \Delta p_{1H} + \Delta p_{2L}$$  \hspace{1cm} \text{Equation 29}
In reality, there is heat exchange with the hydraulic fluid and the ambience through the walls of the hydraulic lines but these losses can be disregarded here in this model due to low temperature of the fluids and hence low impact on the results.

2.1.4 Pressure relief valve

The proposed hydraulic system has pressure limitations which should not be exceeded. A safety feature needs to be introduced here which would help release pressure if the safety of the system is under compromise. The system designed is tailored around the peak pressures of the motor and pump. The design of the accumulator is able to hold higher pressures than the pump/motor. The max pressure that the accumulator was designed for is 5000 psi. To keep the line pressure less than maximum operational pressure of the valve, the difference between the input and output fluid flow in the high-pressure section must be equal to or anything less than the maximum flow permitted by the valve. A basic direct operated pressure relief valve is used by the system here that consists of a poppet, spring, housing and adjusting screw as depicted in the figure.

The system’s pressure from the high-pressure section acts on the poppet and for pressure values larger than valve set pressure, the poppet is pushed back. The initial force of the spring can be adjusted manually by the adjustment screw. The minimum pressure can hence be set to whatever the system uses and hence be tuned for every application. The model here consists of the poppet dynamics and the equivalent valve flow rate. The equations:

\[ m_p \ddot{x}_p = p_l A_p - F_0 - k_s x_p - d_s \dot{x}_p \]

Equation 30

Where \( m_p \) denotes the mass of the poppet, \( x_p \) the position of the poppet, \( A_p \) poppet area, \( F_0 \) which is the pretension force, \( k_s \) and \( d_s \) the spring and damping coefficient respectively.

Here the dynamics of the poppet are not only dependent on the valve parameters but the bypass flow through the valve:

\[ Q_v = C_d A_v x_p \sqrt{\frac{2 p_l}{\rho}} \]

Equation 31

Where, \( Q_v \) is the bypass flow through the valve and \( A_v \) is the valve’s throttling area. This all however varies with the position of the poppet and the dimensions of the valves.
2.2 Electrical subsystem modeling

2.2.1 Battery modeling

In order to design a battery, the first requirement was to identify which battery is suitable for the application as the electric power source. After the process of literature review it was decided that a lithium ion battery would be most suitable as relatively to its small size, it has higher energy densities and has a higher discharge capacity compared to a Lead Acid battery or a Nickel Cadmium (NiCd) battery.

For Li-Ion battery packs, battery management is essential. Due to the nature of the Li-Ion technology, the cells need to be controlled individually. For example, without a Battery Management Unit (BMU), a Li-Ion cell could easily be discharged under its voltage limit and be irreversibly damaged. Individual control over the cells also enables optimization of the stored energy, improving lifetime, safety and cost of the battery pack. In an EV, a BMU is needed to keep track of the battery’s capacity status and also to increase the safety and reliability of the vehicle.

Determining the State of Charge (SOC) of the battery accurately is one of the major functions of the designed battery model.

The BMS monitors and calculates the SOC of each individual cell in the battery to check for uniform charge in all of the cells in order to verify that individual cells do not become overstressed. The SOC indication is also used to determine the end of the charging and discharging cycles. Over-charging and over-discharging are two of the prime causes of battery failure and the designed battery must maintain the cells within the desired Depth of Discharge (DOD) operating limits.

Electric vehicle batteries require both high power charge capabilities for regenerative braking and high power discharge capabilities for launch assist or boost. The lower limit is set to optimize fuel economy and also to prevent over discharge, which could shorten the life of the battery. Accurate SOC information is therefore needed for EVs to keep the battery operating within the required, safe limits.

**HEV Battery operating range**

A typical EV works under operating conditions between 80%-20%. This is to make sure that the battery does not undergo over charging where at times due to the excess thermal runaway a battery breaks down. It is important to reduce this aspect as it can lead to explosion of the battery which can be harmful to the vehicle and the passenger. The headroom also allows for technologies like regenerative breaking and solar panels to increase the charge of the battery, without damaging the battery.

The battery used in the project is a generic preset battery from SimScape library in MATLAB®. Figure 23 shows the predefined battery.

An important factor here to be noted is the initial SOC level. The maximum upper value of 100% is considered here for two reasons. The primary reason here being the maximum range calculation. Through the simulation, an attempt was made to simulate the vehicle running with a fully charged battery. The second reason here is that with this higher-level estimation; the goal of having replaceable batteries can be simulated with ease if the battery originally has a 100% charge. Apart from this, the vehicle would
function up to the point where the batteries are not harmed, that is under the safe region of operation as mentioned above.

![Figure 23. Predefined Battery.](image)

Nominal Discharge Characteristics of the lithium ion battery, shown in Figure 24, based on input values:

![Figure 24. Discharge Characteristics.](image)

The nominal area is the area under the graph where the rate of discharge is close to constant providing a smoother line. This is optimum condition for the battery to work efficiently as the rate of discharge is constant.

Followed by the battery is a DC-DC Buck Boost converter which has a multi-function, denoted in Figure 25. It boosts the voltage to the DC line that leads to the inverter and the buck converter converts the high voltage to low voltage to store in the battery.

![Figure 25. Electric power source layout.](image)

A variant to the predefined battery model is its cell variant, shown in Figure 26. To depict more realistic operation, the battery has been depicted with a 10-cell branch, connected in series. The advantage here is
that monitoring of each individual cell is possible as well as adding a basic heat management solution to the cells in a realistic level.

![Battery: Cells variant](image)

**Figure 26. Battery: Cells variant.**

The cells here however have been modeled with similar characteristics from initial charge and charge and discharge characteristics. The modeling of the electrical circuit of the cell has been achieved with various circuits over the years and the resistive and capacitive terms of the batteries has been established with different variations. Developing the right electrical circuit for the need of the application is very important as the cell operation would vary, so very slightly. Allowing accurate depictions allows depth of the battery cell modeled here.

The cell circuit, Figure 27, has been modeled with a 1-RC circuit model with additional resistances R2 and Rp to depict realistic operations. The variations of the cells modeled 1-RC to 2-RC circuit and other variations with other circuits have not been studied here and hence could be a task for future work.

![Equivalent RC circuit](image)

**Figure 27. Equivalent RC circuit.**
2.2.2 Motor modeling

Modeling in simulation is done by considering the electrical equation of motors. There are two general types of motors based on the working principle, one is synchronous motors and another one is asynchronous motors. The motor type is permanent magnet synchronous motor (PMSM) so the equation used for modeling are based on the synchronous machines. The basic equation used in the design of synchronous machine based on the input frequency and output speed is written below:

\[ p = \left( \frac{60 \times f_1}{n} \right) \]  

Equation 32

Where,

\( p \) - Number of poles  
\( f_1 \) - Input frequency  
\( n \) - Required speed in rpm

The no of poles has a relation with the dimension, torque and speed. The equation of torque and Back EMF is the two most important equations which are used in the modeling of motors. The basic equation of torque is given below:

\[ T = \frac{3}{2} p l m (\psi_s^* \cdot i_s) \]  

Equation 33

Where,

\( T \) - Torque  
\( i_s \) - Stator current  
\( \psi_s \) - Stator flux

The above equation is further written in d and q-axis with the help of park transformation. The reason for transforming it to d and q axis is to have better control over the torque, which would be useful in the field oriented towards control of motors.

\[ T = \frac{3}{2} p (\psi_{sd} i_{sq} - \psi_{sq} i_{sd}) \]  

Equation 34

Where,

\( \psi_{sd} \) - Stator Flux in d - axis  
\( i_{sq} \) - Stator current in q - axis  
\( \psi_{sq} \) - Stator Flux in q - axis
\[ i_{sd} - \text{stator current in d-axis} \]

The basic equation for Back EMF is written as follows:

\[ E_f = w_1 \psi_m \]

Equation 35

Where,

\[ w_1 - \text{angular speed} \left( \frac{\text{rad}}{\text{sec}} \right) \]

\[ \psi_m - \text{magnetising flux} \]

The equations above give a clear indication that the torque is controlled with the current and the speed is controlled with voltage. The current and voltage also has certain limitation hence it is always preferred to run at rated operation.

The input voltage is 3 Phase AC current, the motor in three phase is a bit difficult to control hence a vector control method is used to control the motor by transforming the 3-phase to 2-Phase system. The torque and flux can be controlled very quickly to follow the reference value with less response time with the help of Vector control method. It also gives high accuracy on flux estimation, but in scalar control method a flux sensor is needed to estimate the flux which eventually increases the cost of the system and provides less accuracy and faults. This gives immense motivation to use vector control method. For better understanding the coordinate transformation can be shown in the figure below:

Figure 28. Co-ordinate Transformation.

The figure above shows how the transformation is related to stator reference frame and rotor reference frame. In motor modeling the 3-phase voltage which are \( V_a, V_b, \text{and} V_c \) are transformed to \( V_d \) and \( V_q \) this transformation is known as park transformation. The equation used for modeling in rotating reference frame i.e. dq-axis is shown in the equation below:
\[ \begin{bmatrix} V_d \\ V_q \end{bmatrix} = \begin{bmatrix} 1 & -1 & -1 \\ 0 & -\frac{2}{\sqrt{3}} & -\frac{2}{\sqrt{3}} \end{bmatrix} \begin{bmatrix} V_a \\ V_b \\ V_c \end{bmatrix} \]

Equation 36

The equation of voltage has a relation with current and flux in dq-axis which is shown below:

\[ u_d = R_s i_d + \frac{d\psi_d}{dt} - w_1 \psi_q \]

Equation 37

\[ u_q = R_s i_q + \frac{d\psi_q}{dt} + w_1 \psi_d \]

Equation 38

\[ \theta = \frac{dw}{dt} \]

Equation 39

The d-axis is known as real axis and q-axis is known as imaginary axis. With the help of above equations the torque and speed of the motor can be calculated. This part of modeling in Simulink involves park transformation, which helps to calculate theta (\( \theta \)), Electrical torque (\( T_e \)) speed (\( w_e \)) and generates pulse which would be required in inverter for power conversion. The diagrammatical representation of Simulink model is shown in the figure below:

**Figure 29. Vector control method of modelling PMSM motor.**

Figure 29 has three blocks whereas in the first block the conversion of 3-phase system to 2 phase system takes place. The reason for converting 3 phase into 2 phase system is to have better control over torque and speed. The middle block uses the equations mentioned above to calculate the angle, torque and speed and respective currents in dq-axis. Whereas, in the third block, the current in dq-axis is converted to abc-axis to generate pulses which would be used in inverter.
The next step involves the motor controller and inverter modeling in conjunction with reference torque and reference speed from driving cycle. The DC/AC inverter is provided with pulse input from the controller block to maintain the voltage and current. The main task of the inverter is to control the flows of current in different direction based on the nature driving cycle. When the vehicle is decelerating the current flows in opposite direction and saves the energy recovered during braking into the battery and acts as a generator, similarly when it is accelerating it acts as the motor and propels the wheel. The overall motor design with controller is shown in Figure 30.

![Figure 30. Complete PMSM model with inverter and controller.](image)

In the motor controller block, the input of vehicle speed is conveyed from the driving cycle, the feedback speed from the rotor is also given as an input to the controller to generate pulses for inverter. The Inverter is built with Mosfet Bridge which is directly taken from the Simulink block to provide four quadrant operations. Four quadrant operation means forward braking, forward motoring, reverse motoring and reverse braking.

### 2.2.3 Heat management

Electric vehicles require highly optimized thermal management systems to oversee its efficient operation. Both the efficiency and range are directly affected by heat and this requirement increases with cabin heating and/or cooling. Due to shortage of waste heat, cabin heating can be hard to achieve and so is cooling as the toll on the batteries is high due to the high-energy demand. Compared to a conventional vehicle, electric vehicles need different methods based on the application and in most cases more efficient methods of heat management.

An attempt was made at developing a basic heat management strategy for the modeled battery and motor blocks. These represent the main heat sources in the running simulation and developing an integrated heat management solution for these blocks increases accuracy.

Battery cells as an energy source have a strict requirement on the working environment. They are sensitive to temperature and to ensure a good working environment, a Battery Thermal Management System (BTMS) will normally be integrated with battery cells. The control requirements are hence essential for the performance and durability of the battery pack.

Problems to batteries with temperature can be due to both lack of heat and excess heat. Chemical reaction rates have a linear relation to temperature. Lack of appropriate heat will reduce reaction rate and the
capacity of carrying current during charging or discharging. This reduction of power capacity also causes reduction of reaction rate, which results in the reduction of power. Excess heat increases the reaction rate with higher power output; however, it also increases the heat dissipation and generates even higher temperatures. Unless heat is dissipated quicker than heat is generated, the temperature will be higher and finally a thermal runaway will result.

![Figure 31. Operating temperature cells(22).](image)

Generally, the temperature must be controlled between 20°C and 40°C to ensure the performance and cycle life. The cells operation based on temperature is shown in Figure 31. Lithium battery models with thermal effects are an essential part in the workflow for battery management system design. A battery model should capture the nonlinear dependencies associated with charge and temperature for specific battery chemistry.

The modeling of the heat management in the battery pack has been achieved in the cells variant of the Li ion battery. The heat dissipation is only in a basic level where only convective heat transfer between the cells has been considered. Figure 32 shows the architecture of the designed battery heat management.

![Figure 32. Battery heat management.](image)
Power density and reliability specifications for permanent magnet motor drives in traction applications are important constraints in terms of performance. Heat management plays a vital role in this domain. The vehicles traction motor’s peak torque is limited by the switching temperatures and must be maintained for accuracy in performance and reliability of the motor.

In order to meet the thermal performance analysis accuracy requirements of high density permanent magnet synchronous motor (PMSM), the model incorporates heat management of the motor with both conductive and convective heat transfer, as shown in Figure 33.

**Figure 33. PMSM heat management.**

The conductive heat transfer has been modeled consisting of heat transfer between the three stator windings, A, B and C represented by HA, HB and HC respectively. Convective heat between these three windings and each individual winding has been modeled by a 1-by-3 row vector defining the temperature of the A, B, and C thermal ports. Convective heat transfer of the rotor individually and its convective heat transfer with the three stator windings have been modeled to complete the thermal circuit. The thermal model for the motor accounts for Thermal mass for each stator winding, Initial stator winding, Rotor thermal mass, Rotor initial temperature and Percentage of magnetizing resistance associated with the rotor.

**Figure 34. Heat management block.**
The heat management block in the schematic of the powertrain layout allows the coupling of the two heat management solutions, the battery and the motor. It involves the use of a basic thermal management line consisting of two radiator hoses, expansion tank and pump, and a radiator. Based on the thermal settings of the fluid used, the heat management solution has been established which helps in depicting a basic heat management solution for the vehicles powertrain as shown in Figure 34.

### 3. Simulation setup

The simulations have been modeled in accordance with the scope of the thesis to focus on the optimized efficiencies of operation for reducing the energy consumed for each of the applications. The modeled hydraulic and electric subsystems from the last section are developed into two separate cases, where Case I looks into the potential benefits of adopting the hydraulic hybridization into a vehicle with the conventional IC engine layout and Case II is a further developed simulation to tackle the potential benefits of hydraulically hybridizing an electric vehicle. In both of the cases, the setup looks into the selection of the vehicle parameters and its intended use with its application along with the sizing of the components and the various control parameters in addition to the information provided with modeling of the subsystems in the previous section.

#### 3.1 Case I: Hydraulic hybridization with conventional IC engine layout

Figure 35 shows the diversity of vehicles based on the Gross Vehicle Weight Rating (GVWR).

![Vehicle segregation by weight](image)

Classes 1 and 2 are lighter vehicles which use spark-ignition internal combustion engines and 80% of these vehicles are used for personal use. Class 3 and above are usually commercial vehicles. A mix of...
gasoline and diesel engines is used in this class. Class 3 through class 6 are the medium duty vehicles with single rear axles and class 7 through class 8 are heavy duty vehicles with two or more rear axles.

The test vehicle parameters were chosen here with the vehicle’s application in mind along with its associated mass and driving characteristics in the city. The hydraulic system needs space to be installed and with its size comes added mass. From preliminary perspective, a garbage/refuse truck was selected for the vehicle simulations. However, the garbage/refuse trucks come in different sizes for different applications. Also, the other vehicles in the similar weight category like delivery vans, bucket trucks, shuttle busses, tow trucks, etc. can benefit from this hybridization. To increase the generality of the vehicle simulations, a class 6 truck with 9000 kg mass vehicle was selected. This represents the gross vehicle weight of medium duty vehicles or the test/curb vehicle weight for a heavy-duty vehicle. Having the vehicles run in the same scenario within the cities allows the results to be analysed for a medium duty truck which in this case could be the gross vehicle weight of a delivery truck or the test vehicle weight of the shuttle bus. Similarly, it can also be analysed as the gross vehicle weight of a light/small garbage truck or the test vehicle weight of a large garbage truck. For the vehicle simulations, the parameters of the truck are similar to a medium duty International truck: Durastar® and the table below shows the basic specifications. More details on the latest model are available here (24).

**Table 3. Modeled specifications.**

<table>
<thead>
<tr>
<th>Engine</th>
<th>Cummins 6.0 L V8</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Transmission</strong></td>
<td>Allison transmission</td>
</tr>
<tr>
<td>Maximum power at 3300 rpm (kW)</td>
<td>230</td>
</tr>
<tr>
<td>Maximum torque at 2000 rpm (kW)</td>
<td>770</td>
</tr>
<tr>
<td>Maximum speed (rpm)</td>
<td>4500</td>
</tr>
<tr>
<td>Transmission gear ratios and relative efficiencies:</td>
<td></td>
</tr>
<tr>
<td>1&lt;sup&gt;st&lt;/sup&gt; gear</td>
<td>3.44; 0.9883</td>
</tr>
<tr>
<td>2&lt;sup&gt;nd&lt;/sup&gt; gear</td>
<td>2.23; 0.967</td>
</tr>
<tr>
<td>3&lt;sup&gt;rd&lt;/sup&gt; gear</td>
<td>1.41; 0.994</td>
</tr>
<tr>
<td>4&lt;sup&gt;th&lt;/sup&gt; gear</td>
<td>1; 1</td>
</tr>
<tr>
<td>Differential gear ratio and efficiency</td>
<td>3.212; 0.96</td>
</tr>
<tr>
<td>Vehicle mass (kg)</td>
<td>8500 kg*</td>
</tr>
<tr>
<td>Frontal area (m&lt;sup&gt;2&lt;/sup&gt;)</td>
<td>6.767</td>
</tr>
<tr>
<td>Aerodynamic drag coefficient</td>
<td>0.5</td>
</tr>
<tr>
<td>Rolling resistance coefficient</td>
<td>0.005</td>
</tr>
<tr>
<td>Wheel radius (m)</td>
<td>0.413</td>
</tr>
<tr>
<td>Axle configuration and wheels on rear axle</td>
<td>4x2; 4 wheels</td>
</tr>
</tbody>
</table>
The vehicle in the simulations has an updated mass of 9000 kg to accommodate the masses for the hydraulic subsystem components.

### 3.1.1 Component sizing

#### 3.1.1.1 Hydraulic components

The modeled hydraulic components now are sized for the application in this segment.

**Hydraulic Pump/Motor**

The conventional series hydraulic system makes use of two motor/pumps. To increase the hydraulic machine’s efficiency during operation, three hydraulic machines are used in the layout.

The engine pump/motor is mainly influenced by sizing on the capability of it absorbing the maximum engine power, performing the pumping operation at minimum pressure (1000 psi). The system will remain pressurized above the minimum pressure almost at all times. The engine motor is also used here as a starter for the engine.

The drive motor integrated with the transmission and differential has been designed with two drive motors. The primary motor provides most of the acceleration to the vehicle while the secondary motor having designed with a gear ratio of 1.6 runs at higher speeds or during hard acceleration with gradient performance. If the primary motor is capable of providing the power required, the secondary motor is set to idle. The splitting of the motors ensures higher efficiencies of use with each of the motors, thus improving the operating efficiency at even high speeds. The engine pump motor has a maximum displacement of 200 cm³/rev and the primary and secondary drive motors have a maximum displacement of 160 cm³/rev and 120 cm³/rev respectively.

**Hydraulic accumulators**

The pressure in the low-pressure reservoir is between 50 and 250 psi and the pressure in the high-pressure accumulator is between 1000 and 5000 psi, both representing the minimum and maximum pressures respectively. The accumulators need to store the regenerative braking energy and have excess energy stored at all times for the engine restarts. A smaller accumulator would lead to a problem with more frequent engine restarts but designing a larger system would add weight and the resulting space problems. The designed series hydraulic hybrid will have a capacity of 110 litres, out of which 70 litres is the maximum fluid capacity. The high-pressure accumulator could store 1050 kJ of energy at maximum operational pressure.

#### 3.1.1.2 Electrical subsystem

To have a comparison, the series hydraulic hybrid vehicle is compared to its electrical rival, a series electric and a series parallel hybrid vehicle. Having a fuel consumption analysis here allows further insight into the potential of hybridizing vehicles. Having the two configurations in the electrical
subsystems diversifies the comparisons by having fuel consumption as a standard to measure the scaling of the hydraulic hybrid and its performance.

The series electric hybrid was designed based on similar characteristics as the BAE systems that make series hydraulic buses. It uses a 120 kW motor and a 25 kWh battery with a nominal voltage of 330 V.

The parallel electric hybrid was scaled to the Eaton electric systems fitted in parallel. It uses a 55 kW motor and a 4 kWh battery with a nominal voltage of 340 V.

3.1.2 Power management and control

The strategy for the series hydraulic hybrid is to reduce the usage of the engine and use the engine at its most fuel-efficient point during usage. During its start operation, the engine would operate on the optimum efficiency line by controlling engine pump displacement and throttle position. The engine has two tasks to execute. Either deliver power to drive the motor or fill the accumulator. During its operation at low power demand, the engine power would drive the motor and the surplus power would be redirected to charge the accumulators. The accumulator can always provide energy as the secondary power source is there is a peak power source required during running. In the event that the driver requires more power, the engine would operate at a higher power range along the same trajectory of the optimum efficiency line. In the extreme cases only is the engine allowed to deviate from the optimum trajectory line to produce maximum power. In the series hydraulic hybrid, two hydraulic drive motors would provide energy to the wheels efficiently. Along with the conventional transmission and differential, the primary motor provides most of the propulsion and the other motor is an assistant motor tuned with a fixed gear ratio of 1.5 to provide power during hard accelerations. If the primary motor is capable of providing power on its own, the secondary motor can be set to idle. This split of motors allows the primary motor to operate at higher efficiencies and allows the secondary motor to focus on power filling which improves the efficiency when compared to a single large motor.

3.1.3 Operation modes

Hydraulic power mode

In this mode the power driving the wheels comes only from the high pressure accumulator until either the SOC drops to the lower set limit of SOC threshold or the displacement of the motor reaches its maximum value. The maximum displacement refers to the sum of the displacements of both the primary drive motor and the secondary drive motor with its fixed gear ratio. In the former case where the fluid in the high-pressure accumulator is close to depletion, the engine needs to be switched on and deliver power before the fluid is depleted to oversee smooth transition. In the latter case, where the maximum displacement of the motors is met, the engine must be switched on to increase the power delivered to the wheels. The limit value of SOC threshold plays an important role here as a higher SOC threshold can improve vehicle performance but a lower SOC threshold would mean decreased engine restarts. This is discussed in detail in the results section.

Refill mode
In this mode, the engine would act as the primary power source for the vehicle. Here the objective is to increase/refill the high-pressure accumulator’s SOC while maintaining optimum operation of the engine. The accumulator SOC is steadily increased here until the predetermined SOC upper threshold is met. The engine switch off can be set by either this threshold limit and if the power supplied by the two drive motors, is lesser than the maximum displacement of the two motors. Here it is important to set the predetermined upper SOC threshold limit. If the limit is set too high, the accumulators would not have enough capacity to capture the regenerated braking energy during the braking events. Similarly, a low SOC upper threshold would lead to more frequent engine restarts, hence having a negative impact on fuel economy.

**Regenerated energy mode**

The drive motor/pump operates as a pump here in this mode to recapture braking energy until the max limit of SOC is met (SOC=1). When the maximum SOC of the accumulator is met; the mechanical brakes can be engaged.

**Mechanical braking mode**

If the SOC of the accumulator reaches its maximum value, 1, the accumulator would be disconnected from the drive line and the traditional mechanical brakes are applied. Also, when the vehicle speed is close to zero, the mechanical brakes of the system are engaged. It is at these points that the displacement of the drive pump is set to zero.

**Engine start mode**

Having the hydraulic subsystem as an auxiliary power source, it is capable of acting as a starter for the vehicle. This would eliminate the need of a traditional electric starter motor. Here, the engine pump would act as a motor and is used to start the engine by using energy from the accumulator. The initial SOC is also important here as it cannot be zero for starting the engine.

3.1.4 **STATEFLOW Logic**

For the execution of the operation modes within the simulation in MATLAB/SIMULINK®, STATEFLOW® allows the strategy to be implemented with ease as shown in Figure 36. The control strategy implantation allows this realistic visualization to be easily downloadable to the electronic control unit (ECU) of the vehicle. The logic implemented monitors the pedal position from the driver along with the other input variables which is the SOC, vehicle speed, engine speed, primary and secondary motor/pump displacements.
The actuation controls for the hydraulic machines used are executed within the actuation controls block shown in Figure 37.

The Testing of these models with the simulated conditions under the various drive cycles is discussed in Case I of the Simulation Results section.
3.2 Case II: Hydraulic hybridization of an electric vehicle

The hybridization of vehicles in the last few years have tackled vehicle hybrids with IC engines to generate better results in terms of higher efficiencies and lesser fuel consumption. With development of newer technologies, increased focus on the development of electric vehicles in terms of sustainable vehicle transport has been a trend among major manufacturers. The benefits of electrification of vehicles surpass all other means of future development in vehicles due to its ease of energy management. However in the near future, such developments require additional facilities like charging stations, charge scheduling in cities due to high loads required by power grids, etc to facilitate smooth transition into seeing the vehicles developed now to be implemented on the roads.

From the characteristics of the two hybridization systems discussed in the previous sections, namely the electric and hydraulic hybridization, one cannot help but notice the advantages of having both the systems developed into vehicles. At first glance, the two systems, even though have similar roles in the vehicle layouts, have quite a lot of difference in operational characteristics. Having reached this day and age where the cost, efficiencies of operation, energy consumption, maintenance, etc play a vital role in future vehicle development. In certain vehicular applications there could be scope for development of these systems under one layout, thus attaining the benefits of the two systems add up to deliver better results. With the differences in the two layouts comes the possibility where the two systems have certain synergy that could be productive for certain vehicle types, where only development of one of the systems would seem obsolete. Examples of such areas have been specified in Section 1.

Battery electric forklifts

Forklifts in the past had a preference towards IC engine propelled forklifts due to its unparalleled performance in terms of lift and travel speed. With improvement in AC motor technology and batteries and battery charging methods, for a large majority of forklift applications, electric forklifts have now the means to surpass its IC engine counterpart. Electric forklifts that rely on batteries perform well without producing any exhaust emissions. This reduces the damage caused not only to the environment but any goods/products that the forklifts work with. A plus here would be improved air quality at the area of operation as well as reduced noise pollution. The lower noise and vibration levels are another benefit to the operators by reducing fatigue. Apart from the operation, the electric forklifts are known to have significant cost advantages through their life cycle operation. The economic benefits are directly influenced by lower fuel costs, longer operational life, and lesser costs spent on maintenance. According to Yale forklifts, Figure 38 shows the operational costs for both an electric and IC engine forklift.

![]() Figure 38. Potential savings(25).
The total acquisition cost is an important factor for making the choice between the various technologies available, instead of just comparing the initial costs. The total acquisition costs involve initial costs, maintenance costs, replacement part costs and all the indirect costs related to the purchase. The involvement of operational costs for AC PMSM driven electric forklifts could be beneficial if a time period of 5 years is considered. The return of investment would begin after a year or two based on the operation. Data from 2012 reveals this cost difference in ownership shown in Figure 39. The initial investment for electric forklifts is significantly higher than its IC engine alternative. The increases in costs are a direct resultant of increased costs of electrical components, mainly the battery and charger. In addition to having purchase the electric forklift, two additional batteries and charger may be needed depending on the number of hour shifts that the forklift is meant to operate.

![Figure 39. Total cost of ownership(26).](image)

The savings per vehicle here is substantial and as most operators own a fleet, the savings for converting to electric forklifts is a real deal. These are benefits apart from the obvious higher efficiencies of operation of electric motors and batteries versus the IC engine counterpart. Figure 40 shows a comparison of the three types of forklifts technologies.

![Figure 40. Overall comparison of technologies(26).](image)

With development of AC PMSM motors, the electric forklifts are comparable to IC engine forklifts in terms of acceleration, gradeability, lift speed etc. Having electric braking helps reducing brake wear which would be caused due to mechanical braking in a conventional forklift. Battery operated Electric
forklifts use electrical energy for driving and to lift loads. This is done using two electric motors, one for driving and one for lifting loads. Additionally, with the electric layout, the forklift is now able to recover regenerative braking energy. It is also possible here to recover potential energy when loads are lifted and/or lowered. The massive battery here acts as a counterweight for stability. Electric forklifts, as in Figure 41, only comprise of the lower class forklifts (by weight) and can handle a variety of lift capacities; however most of them have lift capacities in the range of 3000-6000 lbs or 1300-2700 kilograms. They are known for their applications indoor with narrow aisles and indoor environments with no tailpipe emissions, where ICE powered forklifts cannot compete.

<table>
<thead>
<tr>
<th>Class</th>
<th>Type of Preparation and Operation</th>
<th>Lift Code</th>
<th>Description</th>
<th>Picture</th>
</tr>
</thead>
<tbody>
<tr>
<td>I</td>
<td>Electric Motor Rider</td>
<td>4</td>
<td>Counterbalanced; sit-down, 3-wheel</td>
<td>![Picture]</td>
</tr>
<tr>
<td></td>
<td></td>
<td>5</td>
<td>Counterbalanced; sit-down, cushion, lifted</td>
<td>![Picture]</td>
</tr>
<tr>
<td></td>
<td></td>
<td>6</td>
<td>Counterbalanced; sit-down, pneumatic</td>
<td>![Picture]</td>
</tr>
<tr>
<td>II</td>
<td>Electric Motor Walker</td>
<td>2</td>
<td>Low lift pallet</td>
<td>![Picture]</td>
</tr>
<tr>
<td></td>
<td></td>
<td>5</td>
<td>High lift reach type</td>
<td>![Picture]</td>
</tr>
<tr>
<td></td>
<td></td>
<td>7</td>
<td>High lift counterbalanced</td>
<td>![Picture]</td>
</tr>
<tr>
<td>III</td>
<td>Internal Combustion Engine Rider</td>
<td>3</td>
<td>Counterbalanced; sit-down, cushion, solid</td>
<td>![Picture]</td>
</tr>
<tr>
<td>IV</td>
<td>Internal Combustion Engine Rider</td>
<td>4</td>
<td>Counterbalanced; sit-down, pneumatic</td>
<td>![Picture]</td>
</tr>
<tr>
<td>V</td>
<td>Rough Terrain</td>
<td>1</td>
<td>All types</td>
<td>![Picture]</td>
</tr>
</tbody>
</table>

Figure 41. Forklift class segregation(27).

One of the main disadvantages of battery powered forklifts is the long charging hours. The batteries designed in the forklifts today are capable of handling long shifts of 6-8 hours, it takes long to charge the batteries which is valuable time wasted for its application. In most cases to operate tight shifts, like 24/7 operation, there is a requirement of three battery packs per forklift. In many a case, the 8:8:8 rule is followed which is 8 hours to deplete the battery, or one shift, 8 hours to charge the battery and 8 hours to properly cool the batteries. Even in this case, time is lost for changing batteries, 15 minutes for automatic change out and 45 minutes if done manually with the presence of a queue (28). Newer technologies have shortened the 8:8:8 timings by fast charging, charging batteries at quicker rates which could be incorporated during breaks. There still would be a disadvantage here because there is a limit to how far this technology can be adopted. Such fast charging would for instance, require weekly monitored equalizing charge to prevent harm to the battery.

Another disadvantage of battery powered forklifts is the power declination as the battery discharges. This along with battery ageing has known to decrease the productivity of the forklift with time. The problems could also be linked with additional factors such as facilities like charging stations, charge scheduling in cities due to high loads required by power grids or insufficient power supply from the grid by the city, etc. One way to reduce the problems is by increasing the range or battery discharge time. This can be
achieved by reducing the energy consumption of the forklift by maintaining higher efficiencies of the operating systems.

3.2.1 Electric hydraulic hybrid layout

Hydraulic hybrids popularly have two designed layouts: the series and parallel layouts. These are layouts that have been tried and tested on IC engine (conventional) vehicles. There are however other possible layouts, just like with the electric hybrids, such as complex or series-parallel hybrid, through the road hybrid etc. When it comes to the case of forklifts, the electric forklifts have started to make a segment for themselves with the many advantages that they bring to the operator. Adding the hydraulic system to the electric forklift would only make sense because of its low speed applications and constant stop and go operation. As hydraulic systems excel in this domain, having it installed in the forklift could probably help in decreasing the load on the battery and electrical systems. Hydraulic systems, characteristic of their high-power density could pose as an added benefit with certain applications that the electric forklift has to deal with. The high energy density electrical systems are known for their higher efficiencies of operation but at low speeds of operation the hydraulic systems boast their higher efficiencies. As seen from the results of the simulation from Case I, the hydraulic hybrid systems also perform better with their higher cyclic efficiencies resulting in greater energy capture with regenerative braking. Forklifts in certain applications also require additional hydraulic energy for lifting purposes, which can be blended with the hydraulic hybrid layout. The advantages rack up for having these systems installed into forklifts and a possible positive outcome could be expected with only a few drawbacks such as increased noise and maintenance.

Since there is a lack of literature on this subject, only conventional layout configurations (i.e. series or parallel) were considered to be adopted to the electric forklift. The series hydraulic hybrid could be a good option here due to its low speed application and it generating higher efficiencies with more frequent stop and go driving styles than the parallel hydraulic hybrids. But having the electric forklift performing well in its segment and having a parallel hydraulic subsystem to focus as an assistance system would make sense. Also, an influencing factor here is that the work circuit is powered in parallel to the driving subsystems. Same is the case with the hydraulic assistance systems for the work circuits with it being powered in parallel with the main electric layout. Hence instead of going into deeper concepts of layout configurations for the electric hydraulic hybrid, an even more basic layout, i.e. the parallel layout was selected for the forklift.

The parallel hydraulic subsystem with the electric system could focus on increasing the cyclic efficiencies involved with the system without the need of complication. Having it designed in the parallel layout also has a design benefit where the hydraulic motors could be designed for propulsion and/or lifting and lowering, i.e. powering the work circuit, without complications.

The forklift specifications and the component sizing are discussed in the next section under Case II of the Simulation Results.
4. Simulation results and analysis

The results of the simulation tackle the segregated cases individually to better understand the hybridization of the vehicles. In order to avoid confusion, the results have been segregated into two cases as were described in the previous section.

4.1 CASE I: Medium duty truck with conventional IC engine and hybrid

In order to be completely analysed, the Series Hydraulic Hybrid for the class VI truck will be tested on two driving cycles. The EPA Urban Dynamometer Driving Cycle (UDDS) and the Highway Fuel Economy (HWFET) driving cycle would be used to interpret the performance of the medium duty vehicle in the urban cities and highway respectively. The driving cycles also captures the ideal situation’s driving characteristics of the selected vehicle’s use in the city and highway, and would be interesting to see the Series Hydraulic Hybrid’s performance in comparison to the Series and Parallel type Hybrid Electric Vehicle.

![Figure 42. Test driving cycles(29).](image)

The importance of setting the limits for the SOC threshold plays a key role in controlling the initial SOC of the accumulator per cycle. The initial SOC in turn has an effect on the fuel consumption of the vehicle. As mentioned in the operation modes section, control of the vehicles components is devised around the SOC threshold limits. For a selected SOC_lower_threshold, selecting a higher SOC_higher_treshold would result in less frequent restarts of the engine and longer engine running times. However, for the same SOC_higher_treshold, selecting a larger value for the SOC_lower_treshold would mean lowering the average engine running times but with more frequent engine restarts. A high SOC_higher_treshold could lead to decrease the benefits of the system caused by lack of capacity to capture regenerative braking energy.

The difference between the limits of the SOC threshold has a direct influence on the engine’s duty cycle. In order to completely capitalize the on the potential of capturing regenerative braking energy and acceptable increase in engine restarts, the selected SOC thresholds:

SOC_lower_treshold: 0.2
SOC\_higher\_treshold: 0.5

The initial SOC varies the fuel consumption of the vehicle too. The variation in the initial SOC generates fairly small variations with the fuel consumption. The max initial SOC, SOC =1, is used for UDDS and HWFET driving cycles.

### 4.1.1 The urban dynamometer driving cycle

The UDDS driving cycle is mostly used for light duty vehicles. The aggressive slopes of acceleration may not be as realistic as light duty vehicles but the medium duty trucks can have similar driving characteristics. In the following simulations vehicle speed error has been significantly reduced.

![Figure 43. UDDS driving cycle.](image)

![Figure 44. Accumulator SOC variation with UDDS driving cycle.](image)

![Figure 45. Engine speed variation with UDDS driving cycle.](image)

Figure 44 shows the accumulator’s SOC variation with the UDDS driving cycle. Figure 45 shows the engine’s speed (rpm) variation with the UDDS driving cycle. The peaks generated here by the engine show the need of the engine to provide additional power that cannot be met with the hydraulic drive motors. It is at this time that the controller designed allows the engine pump’s displacement to its maximum to increase the SOC of the accumulator. This obviously also depends on the SOC of the accumulator at that point of time.

Figure 46 shows the displacement as a factor of the engine’s pump/motor, primary and secondary drive pump/motors respectively.
Focus on the displacement characteristics around the 195 second mark allows important performance analysis. During this period, the engine’s pump reaches the maximum displacement factor. During this point, the pump harnesses the full engine power. The primary drive motor is used most of the time with high efficiency and the secondary drive motor kicks in only when it is required. This allows higher efficiencies when compared to a single larger motor in use. There does however lay a drawback in this design. Electrically, there would be an unsmooth torque curve followed by a mechanical delay when switching to the secondary motor. This does also create a vehicle speed error within the simulation.

The fuel consumption of the medium duty truck with the conventional IC engine layout gives 25.135 L/100 km. The series hydraulic hybrid consumes 13.86 L/100 km. The series electric hybrid vehicle consumes 16.83 L/100 km. The parallel electric hybrid vehicle consumes 14.25 L/100 km.

### 4.1.2 The highway fuel economy driving cycle

The HWFET driving cycle represents driving for light and medium duty vehicles. Due to lack of complete braking, the engine runs closer to the sweet spot of operation, resulting in lesser fuel consumption. Because the hydraulic machines operate at high efficiencies at lower speeds, the hydraulic subsystem is used less frequently in this driving cycle. Figure 48 and Figure 49 show the accumulator’s SOC and engine speed variation with the HWFET driving cycle respectively.
The fuel consumption for the convention IC engine layout is 20.1 L/100 km. The series hydraulic hybrid consumes 16.8 L/100 km. The benefits of this system are limited because of lesser use of the hydraulic machines. It can be concluded that the series hydraulic hybrid is better suited for in city use.

The series electric hybrid consumed 17.87 L/100 km and the parallel electric hybrid consumed 14.84 L/100 km.

### 4.1.3 Downsizing potential

In the series hydraulic hybrid vehicle, the accumulator, the secondary power source provides sufficient power density to ensure that the vehicle’s acceleration performance isn’t compromised. The conventional engine in the vehicle is the Cummins 6.0 Liter V8 diesel engine. Based on the application, there is a possibility of exploiting the downsizing potential in the series hydraulic hybrid. An assumption made here for the downsized engine is that the 4.8 Liter V6 diesel engine scales proportionally to the conventional vehicle.
The size of the engine has a direct relation with the vehicle’s acceleration capability. The acceleration capability of the vehicle with the series hydraulic hybrid with its downsized V6 engine is compared with the traditional vehicle without any hybridization. The acceleration benefits from the hydraulic system however the benefit varies on the initial SOC of the accumulator. Figure 51 shows the comparison of the series hydraulic hybrid with the components sized in the previous section to the traditional IC engine layout with the initial SOC of the accumulator set to max (SOC=1). This control of the initial SOC of the vehicle is completely characterized by the set values of the SOC threshold limits. The difference in the acceleration times are given in Table 4.

![Figure 51. Acceleration timing with downsized engine.](image)

<table>
<thead>
<tr>
<th>Initial SOC</th>
<th>Conventional V8</th>
<th>Series Hydraulic Hybrid with V6</th>
</tr>
</thead>
<tbody>
<tr>
<td>TIME (Sec)</td>
<td>23.7</td>
<td>28.9</td>
</tr>
<tr>
<td></td>
<td>27.2</td>
<td>25.8</td>
</tr>
<tr>
<td></td>
<td>24.1</td>
<td>23.6</td>
</tr>
</tbody>
</table>

Table 4. Acceleration times with variation in initial SOC.

Unless the SOC reaches the upper limit threshold (SOC=1) by regenerative braking energy, the drive pump would recharge the accumulator SOC above 0.8 after complete stop. However, this downsizing possibility is limited to the specificity of the vehicle design. Downsizing would make sense if the vehicle was designed with the gross vehicle weight in mind and not the test/curb vehicle weight. This is because some vehicles would make better use of the excess power, so it would be sensible for those vehicles to keep the existing conventional engine and hybridize hydraulically. So, the potential of downsizing comes down to the vehicles application and specific design of that vehicle.

### 4.1.4 Analysis

The series hydraulic hybrid model for a medium duty truck of class 6 weight category generated the best results for in city use over the UDDS driving cycle when compared to the other layouts with the electric hybrids. The series electric hybrid would have higher system acquisition costs due to larger sized motors
and batteries required for this layout. Hence having a parallel electric hybrid would make more sense in layout adoption with decreased fuel consumption in the case of parallel versus the series electric hybrid.

The results of the simulation however are only as true as the driving cycle. The accuracy of the driving cycle would enable more accurate results. Like in the case of garbage trucks, the driving cycle characteristics of in city use would not be similar to the UDDS cycle used. Also the fact that the design of the medium duty vehicle has been left at a versatile stage, for the various applications, like shuttle buses, delivery trucks, box trucks, etc causes problems with the specificity of design. From the results generated in this section, it is clear that the benefits of having the vehicle hybridized hydraulically are plentiful. This system however does not hold the best results with the HWFET driving cycle. This is due to lack of operation of the hydraulic machines at high speeds under low efficiency conditions.

The series hydraulic hybrid performed with higher efficiency when compared to the parallel electric hybrid but at a limitation of use at lower speeds. With certain applications like in the case of garbage trucks the adoption of the hydraulic components would make sense as it would out shine the other layouts with worse conditions, i.e. more frequent stop and go drive style. Through these simulations the higher efficiencies at lower speeds and higher power density was clear in the case of hydraulic hybrids. The hybrids however had poor energy density and at higher speeds of travel the hydraulic hybrid systems performed under lower efficiencies comparatively.
4.2 Case II: Electric forklift and hybrid

Through the simulation of the designed forklift layouts, the main focus here is to decrease the energy consumption used for traction/propulsion. This section also looks into the potential energy recovery possibility briefly. Plenty of research has already been done in this field with different technologies used for the potential energy recovery with lifting/lowering loads. The conventional electrical layout is compared here to an electric hydraulic hybrid layout with the main focus on the reduction of energy consumption for traction while utilizing maximum efficiencies with each layout. With the reduction of energy consumption comes an added bonus of optimizing the application of the forklifts by increasing the productivity and decreasing down time.

4.2.1 Potential energy recovery

Recovering energy while lifting or lowering loads is now possible with available technologies. In one such paper(30), research has been done on the electric and hydraulic regeneration methods for the recovery of potential energy. Here two similar forklifts equipped with electric or direct hydraulic energy storage are compared. The forklifts selected for the testing were Class 1, three wheeled forklifts; from the manufacturer Humanic and model HS-16F5400 for the electric setup and HX-16 for the hydraulic setup.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Electric recovery</th>
<th>Hydraulic recovery</th>
</tr>
</thead>
<tbody>
<tr>
<td>Theoretical volumetric displacement of the pump [m³/rev]</td>
<td>13.3·10⁻⁴, manufactured by Erker</td>
<td>19·10⁻⁴, manufactured by Parker</td>
</tr>
<tr>
<td>Motor</td>
<td>10 kW CFM112M PMSM manufactured by Sew-eurodrives</td>
<td>14 kW IM by Danaher</td>
</tr>
<tr>
<td>Converter</td>
<td>ACSM1-04x90, 0.014A-4 by ABB</td>
<td>MHI 16A70-04020, A001</td>
</tr>
<tr>
<td>Piston: cross-sectional area of the free lift cylinder [m²]</td>
<td>0.0026</td>
<td>0.0033</td>
</tr>
<tr>
<td>Maximum stroke of free lift cylinder [m]</td>
<td>0.38</td>
<td>1.35</td>
</tr>
</tbody>
</table>

Figure 52. Differences in specifications(30).

The test setups were designed to test payloads of 0, 500 and 1000 kg at different motor speeds. The velocities of the forks were set to 0.2, 0.3 and 0.4 m/s. The travel height here was 1.6 m.
At 0 kg load, there was no recovery observed in the electric and during lowering of the loads the electric machine was operating as a motor instead of generator mode. Similarly, with no load, the system pressure levels remain too low for energy recovery.

At the 500 and 1000 kg loads, if the hydraulic recovery system is tuned to load optimized pre-load pressures, the hydraulic energy recovery systems do better. It can be concluded that the test arrangement favoured the direct hydraulic recovery system, but it also shows that the electric recovery system has numerous advantages. The electric energy recovery systems do not need a pre-load setting and any tuning for specific load or lifting height.

Considering the applicability of such a hydraulic recovery system with pre-load optimised pressures, it is evident that loads should remain relatively constant for sufficient durations. Warehouse and material handling tasks are one of the examples that have relatively constant loads. In mixed goods warehouses, i.e. variable loading, it would not be advisable to alter the preload pressure between each lifting/lowering (because of additional energy consumption of pressurised gas) and hence need to be optimised for a particular load setting. In order to fix this an alternative recovery circuit would be required based on a hydraulic transformer. Results of that have not been included in this paper. But in conclusion with proper tuning higher energy recovery is achievable with the hydraulic systems.

4.2.2 Braking energy recovery

Forklifts, through the years, have maintained and undergone a general test procedure for testing the energy consumed as per the VDI 2198 cycle. VDI 2198 is an important test used by different manufacturers for the comparison of technical specifications along with its use over the various layouts available (IC engine, Electric, LPG, etc.). Figure 55 and Figure 56 show the details of the VDI 2198 drive cycle. The cycle consists of segregation into four stages where each stage comprises of the various sub steps of forklift operation. The simulation here tests the energy consumed for propulsion purposes only with a constant pay load of 1000 kg tested at a constant lifting/lowering lift speed.
The selected forklift for this simulation is a Class 1, three wheeled Hyster A25 XNT series forklift. This model was chosen as it is in the similar weight class with similar specifications as the Humanic models used to compare the potential energy recovery in the forklift. Certain important specifications of the Hyster A25 XNT series forklift are included in Table 5 and the more details of the forklift specifications can be found on their website (32).
Table 5. Forklift specifications.

<table>
<thead>
<tr>
<th>Manufacturer and Model</th>
<th>Hyster A25XNT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type and Operation</td>
<td>Electric Counterbalanced and Seated operator</td>
</tr>
<tr>
<td>Rated capacity</td>
<td>1134 kg</td>
</tr>
<tr>
<td>Truck weight without battery</td>
<td>2068 kg</td>
</tr>
<tr>
<td>Travel speed maximum</td>
<td>13 Km/h</td>
</tr>
<tr>
<td>Lift speed maximum</td>
<td>.51 m/s</td>
</tr>
<tr>
<td>Traction motor 60 minute rating</td>
<td>4.7 kW</td>
</tr>
<tr>
<td>Lift pump motor 15 minute rating</td>
<td>8 kW</td>
</tr>
<tr>
<td>Battery Voltage and Capacity</td>
<td>24 V and 600 Ah</td>
</tr>
<tr>
<td>Battery weight maximum</td>
<td>700 kg</td>
</tr>
<tr>
<td>Wheelbase</td>
<td>1168 mm</td>
</tr>
<tr>
<td>Wheels: Number Front/Rear (X=driven)</td>
<td>2/1X</td>
</tr>
<tr>
<td>Tire size</td>
<td>18 x7.8 in</td>
</tr>
</tbody>
</table>

**The weight of the electric forklift is at 2770 kg and the electric hydraulic hybrid is modelled at 3000 kg. The load for the lifting/lowering in VDI 2198 was selected to be 1000 kg.

4.2.3 Hydraulic component sizing

The hydraulic subsystem here has been modelled in a similar manner as Case I and is described in modelling section. The difference however lies in the layout and the control logic in accordance to the parallel hybrid architecture. The sizing of the motor/pump was designed in accordance with two major factors. The size of the motor/pump design needs to tend to the needs of accumulator charging with regenerative braking energy. The other important factor being the flexibility of design where the current hydraulic system could tend to the needs of lifting and lowering loads. The current design with this simulation however uses this hydraulic machine for propulsion purposes only.

The size of the hydraulic machine chosen is 20 cm³/rev and the accumulator is sized at 12 liters.

4.2.4 Acceleration benefits

Having a limited maximum speed of 13 km/h, the forklift would have to have faster acceleration timings in order to increase productivity. The rate at which the forklifts complete the task is directly related to its operational shifts. The quicker the tasks are completed, the lesser is the number of hours the vehicle is operated and hence an increase in productivity. Figure 57 shows the acceleration times for the electric forklift and the proposed electric hydraulic hybrid.
Figure 57. Acceleration times for forklift.

The forklifts have a general test procedure for testing the acceleration times of the forklifts. The time taken to accelerate to 10 m of distance is tested and measured in order to have a comparison between the forklifts from different manufacturers. The electric forklift modelled completes this in 5.8 seconds and the electric hydraulic hybrid achieves this at 4.93 seconds. The increase in power density plays a major role and can be seen in the results.

The acceleration times and the speed limit have a trade-off between safety and productivity. This trade-off is a direct relation to internal transport development. For safety reasons a lower speed limit is recommended but an increase in productivity is related to the forklift’s performance decreasing the costs of by the customer per shift. The speed limit and the forklift performance would in normal cases be determined by the storage location and the warehouse sizes, where the driving distances can vary considerably. Having this being said, the performance characteristics for even large warehouses could be executed at a maximum driving speed of 13 km/h. If the operators choose to further limit the maximum speed to say 10 km/h, logically this would induce safer operating conditions. But this is a deceiving factor as the operators focus and alertness is the underlying factor for both set maximum speeds. Another important factor here would be the decrease in energy consumption led due to higher performance requirements of the forklift. Compared to the 13 km/h speed limit, the 10 km/hr would have a loss of productivity by about 8% and the energy gain would be about 25% (33). Either way having a greater acceleration time would increase productivity in the electric hydraulic hybrid design without much loss in the energy consumption.

4.2.5 Energy consumption

The two forklifts, namely, the electric forklift and the electric hydraulic hybrid forklift, were tested and compared according the versatile VDI2198 drive cycle designed for forklifts. The drive cycle tests the energy consumed for both traction and lifting. In this simulation however only the energy consumed for traction/propulsion is tested with the change of vehicular mass because of addition of the payload only at the intervals when specified according to the test parameters in Figure 56. Figure 58 below shows the drive cycle generated for the tests.
The energy consumed in the drive cycle is 81.8 kJ. This is the case where no regenerative braking energy was recovered. Energy dissipated during braking here is about 45.6 kJ which amounts to about 56% of the energy used.

The electric forklift with regenerative braking consumes 68.12 kJ of energy whereas the electric hydraulic hybrid consumes 57.92 kJ of energy. With the possibility of harnessing regenerative braking energy, the electric forklift decreases energy consumption by 17% and the electric hydraulic hybrid reduces energy consumption by about 29%. There is a direct 12% decrease in energy consumption here with the electric hydraulic hybrid layout.

### 4.2.6 Analysis

The parallel electric hydraulic layout generated higher performance characteristics under the acceleration tests characteristic with its increase in power density. When compared to the its electrical counterpart, the electric hydraulic hybrid had a further decrease in energy consumption as the hydraulic system prevented the use of the electrical motors in its low efficiency operating regions at lower speeds. The regenerated energy via braking was also higher in the case of the electric hydraulic hybrid as the hydraulic machines operated at higher efficiencies at lower speeds. There was also an added benefit in the performance characterised by the high-power density of the hydraulic systems which has a direct influence on productivity of the forklift.

The electric hydraulic hybrid however would have an increase of mass in the forklift as well as generate more noise during operation. There would also be an increase in maintenance for the vehicle. The increase of mass however is not a problem in the case of forklifts because of the need for counter balance for lifting loads. As for the increase in maintenance, there would be a decrease in maintenance and increase in battery life of the electrical battery due to lesser charging/discharging of the battery. The electric hydraulic hybrid could hence potentially increase the productivity and decrease the energy consumption in the case of forklifts at the expense of more noise generated during its operation.

Alternatively, the electric forklift could now be redesigned. This could lead to smaller batteries for the equivalent workload which could in turn reduce the initial costs. Also having lesser to worry on the power delivery, the system could focus on delivering energy with focus on energy density which the electric systems are known for and a possibility of redesigning the electric machines.
5. Conclusions

Through this thesis, investigation of the hydraulic hybridization of vehicles and its incentives were tested through simulation using MATLAB/SIMULINK®. Energy consumption and increase of vehicular range was an emphasis throughout these simulations and a characteristic high efficiency system was modelled with high fidelity. With the vehicle application in mind, vehicle layouts were established with segregation of the two simulation tests.

With the first set of simulations, the conventional IC engine was hybridized and compared with a series hydraulic hybrid layout and further compared with the electric hybrids. The application selected here was for that of a medium duty vehicle which could have vehicles designed for various applications. The series hydraulic hybrid showcased better results with reduced fuel consumption and higher acceleration possibilities. The series hydraulic hybrid performed with higher efficiency when compared to the parallel electric hybrid but at a limitation of use at lower speeds. With certain applications like in the case of garbage trucks the adoption of the hydraulic components would make sense as it would out shine the other layouts with worse conditions, i.e. more frequent stop and go drive style. Through these simulations the higher efficiencies at lower speeds and higher power density was clear in the case of hydraulic hybrids.

The second set of simulations tackled a different vehicle application with a different base used, i.e. electric energy for propulsion. The case of electric forklifts and their advantageous use over the IC engine forklifts was discussed along with its limitation of higher initial costs with batteries and charging and downtime. Through the simulations, a basic parallel hybrid architecture was designed and tested with the versatile drive cycle, VDI 2198 used for testing forklifts. The energy consumption was reduced by using the parallel hybrid layout by operating the electric motor at higher efficiencies and an increased regenerative braking energy recuperated by the hydraulic motors characterised by braking at low speeds. There was also an added benefit in the performance characterised by the high-power density of the hydraulic systems which has a direct influence on productivity of the forklift.
6. Future work

The proposed vehicle systems have been designed to meet the needs of the application in the near future. All of the components and layouts have been selected and scaled based on available technologies and hence the concepts can be visualised easily. The selected designs have been compared to various alternatives and designed with high fidelity yet more iterations with further testing would be required for the applications.

Having auxiliary loads (such as air conditioning systems, lighting, etc.) modelled in the simulations would depict more accurate results of energy consumption.

The heat management model in the simulation is a basic model in the case of the electric vehicle simulations. A more accurate heat management model would depict more realistic vehicle operation.

A possibility here into further investigations on various kinds of layouts could be tested and the results hence compared. Having the electric hydraulic hybrid designed in the conventional parallel layout is a basic way of achieving a new kind of hybrid vehicle but further comparisons with other possible layout designs (for example with a series layout, series-parallel layout, through the road hybrid, etc.) could be a more efficient system and could bore better results.

The second set of simulations, with the case of electric hydraulic hybrid forklifts, could be investigated further in terms of use and application of the hydraulic system. With greater iterations of the designed system comparisons could be established to check if using the hydraulic systems for lifting purposes instead of propulsion could bore better results. For this of course, the lifting and lowering of loads and the energy required for it needs modelling.

Comparisons could also be established on the use of different hydraulic machines with greater efficiencies and newer technologies that could increase the benefits of use of the hydraulic system. Some of the newer technologies have been mentioned in Section 1.

Furthermore, the control strategy used along with the layout configurations could be upgraded in terms of the application in order to reap more benefits.
7. Works cited


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