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Design and Modeling of a Parallel Hydraulic Hybrid Bus

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***Abstract*— In the present paper a hydraulic hybrid propulsion system for an urban bus has been designed, modeled and simulated. The propulsion system consists of a diesel Internal Combustion Engine (ICE) as the primary and a hydraulic pump/motor (P/M) as the secondary power generation components. The hydraulic accumulators have been used as the energy storage components. The two power generating components have been connected to the driven shaft through a post-transmission torque coupling. For designing of hybrid propulsion system, the initial sizing of two power generation components has been performed by using MATLAB software. Then, the hydraulic hybrid propulsion components have been modeled in the MATLAB/Simulink and are combined to form a feed-forward model of the hydraulic hybrid bus. In addition, a control strategy has been designed so as to distribute appropriate torque among the hybrid power generation components. This control strategy also includes regenerative braking. After the designing and modeling of the hydraulic hybrid bus, its operation has been simulated in the Tehran and the Nuremburg drive cycles and assessment of fuel consumption and performance are obtained. In comparison to the conventional bus, at least 26 percent reduction in the fuel consumption of the hydraulic hybrid bus has been achieved by the simulation results.**

Keywords: Hydraulic hybrid bus, modeling, design, regenerative braking

I. INTRODUCTION

In recent years, rising consideration about the fossil fuel consumption and increasing air pollution has led to development of innovative propulsion systems such as hybrid propulsion for vehicles. A hybrid propulsion system consists of two power sources in which one of them has the ability to restore energy. Reduction of fuel consumption and air pollution are achieved in hybrid propulsion systems by using a smaller engine size, operating the engine near its optimum BSFC[1] region, turning it off whenever possible and regenerating braking energy. Moreover, hybrid vehicles do not need any additional fleet infrastructure. Hybrid propulsion systems are divided in two main branches: electrical hybrid which has an electric unit as the secondary power source and mechanical hybrid that uses a mechanical component to generate and store energy. Furthermore, there exists three types of mechanical hybrid powertrain i.e., flywheel, hydraulic and pneumatic. Among the mechanical hybrid propulsion systems the hydraulic one is most attractive due to extended field of application in vehicle industry. In comparison with hybrid electric vehicle (HEV), a hydraulic hybrid vehicle (HHV) has larger power and lower energy densities. This characteristic of HHV is useful for transient driving modes of vehicle like acceleration and braking. In addition, the round-trip efficiency of hydraulic hybrid propulsion system is much higher than the electric one [1].

Urban buses are the best choice for implementing a hybrid propulsion system due to the low speed and highly frequent stop and go nature of their driving cycle. In addition, the braking energy that is wasted during operation in an urban bus is much larger than that of a passenger car. Therefore,

[1] Braking Specific Fuel Consumption

implementing a hydraulic hybrid on an urban bus would be a reasonable decision [2].

The hydraulic hybrid propulsion system contains one or two hydraulic Pump/Motor (P/M) as the power generating component. Also Hydro-pneumatic accumulators are the energy storage components in the HHV. Hydraulic hybrid vehicles can be divided into three configurations: serial, parallel and power-split.

In serial hydraulic hybrid, ICE acts as the charger of accumulators and a hydraulic P/M provides the required power of the vehicle. But in Parallel hydraulic hybrid architecture both hydraulic P/M and Internal Combustion Engine (ICE) are mechanically connected to the load. Since there are few changes in the mechanical components of the conventional vehicle in parallel HHV, its implementation is easier and has lower cost than the series structure. A power-split hydraulic hybrid has both parallel and serial driveline and can switch between them. A parallel architecture is much simpler than a power-split one. As a result, the parallel HHV is more suitable for vehicle manufacturing companies. The disadvantage of parallel HHV is that the fuel consumption of engine is higher than series HHV, because the ICE speed is depended to wheel speed in parallel configuration. Parallel hydraulic hybrid powertrains were previously demonstrated for buses in Japan in late 1980s and early 1990s, and are currently developed by Eaton, Permo-Drive and Bosch Rexroth as described in [2].Also some studies on modeling and simulation of parallel HHV have been organized in last decade [3, 4]. Guo-Qing Liu et al. studied the implementation of a parallel hydraulic hybrid powertrain for an urban bus from 2008 to 2010 [2, 5 and 6]. They have published a model for the hybrid powertrain in their last paper. The model of hydraulic circuits has been created in AMESim and linked with the complete MATLAB/Simulink bus model. The efficiencies of hydraulic P/M are not taking into account and this is the main drawback of the proposed model.

In this study, the sizing of components in a parallel hydraulic hybrid (PHH) powertrain for an urban bus has been performed at first. Then a feed-forward model for the bus powertrain has been generated in MATLAB/Simulink. In this model, the efficiencies of hydraulic P/M are taken into account by use of look-up tables. Finally, the model is simulated in two different drive cycles by using a rule-based control strategy. The conventional bus is also modeled similarly for purpose of comparing the results.

II. DESIGNING OF PROPULSION HYBRID SYSTEM

The PHH powertrain is performed on O457 urban bus. This bus is manufactured by IKCO [2]. The required power in conventional O457 bus is provided by 220 kW OM457LA engine which has maximum torque of 1250 Nm. The design specification of the conventional bus is shown in Table I.

The proposed PHH propulsion system can be seen in Fig. 1. Internal combustion engine, hydraulic P/M, transmission and accumulators are the main components of the system.

2 Iran Khodro COmpany

TABLE I. SPECIFICATION OF O457 BUS

Vehicle specifications	Value
Gross Weight (kg)	9820
Weight considering the passengers (kg)	14720
Wheel radius (m)	0.466
Visible surface (m^2)	2.0
Rolling resistance	0.01
Drag coefficient	0.55
The Efficiency of transmission (%)	0.85
Gear ratio of final drive	4.2
Air density (kg/m^3)	1.202
Rotational inertia	1.05

Among the main components, simulation results are needed for sizing of accumulators. In another words, generation and consumption of energy in a drive cycle is needed in order to determine the required volume and number of accumulators. Thus, the initial volume and number of accumulators has been specified according to former studies on parallel HHV. Eight 20 liter accumulators have been chosen. The accumulators are of bladder type and the pressurized gas is Nitrogen.

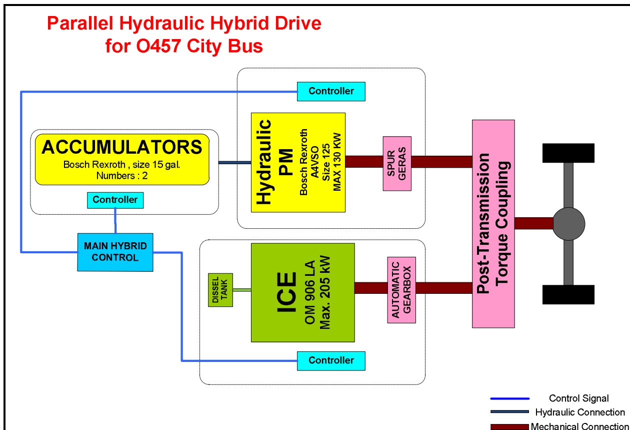

Figure 1- The schematic diagram of PHH powertrain

As can be seen from Fig. 1, the hybrid system has a post-transmission torque coupling. The advantage of post-transmission torque coupling over the pre-transmission one is that in the former torque of each power sources are appropriately improved according to its nature.

In order to design the initial sizes of ICE and hydraulic P/M, a code has been generated by MATLAB. For an urban bus in that code, gradeability constraint is set as cruising in 35 km/h on 10 percent grade and acceleration constraint is reaching the 65 km/h from stationary in 60 seconds. The maximum powers of both power sources have been determined while satisfying the performance constraints. For easier implementation of PHH on the bus, the designing process is performed considering the least possible changes in the powertrain. Finally the initial specification of PHH powertrain components is as mentioned in Table II.

TABLE II. THE SPECIFICATION OF INITIAL HYDRAULIC HYBRID POWERTRAIN COMPONENTS

Component	Specifications
ICE	**OM906LA** Max power : 205 kW Max torque : 1100 Nm
Hydraulic P/M	**A4VSO** Max power : 131 kW Max torque : 696 Nm Maximum displacement : 125 cc/rev
Gearbox	Automatic transmission (the one used in conventional drivetrain)
Accumulators	8 bladder type Maximum operating pressure : 345 bar Nominal Volume : 20 liters (each one)

III. MODELING OF PARALLEL HYDRAULIC HYBRID BUS

In the feed-forward model of PHH bus, the driver block creates appropriate command signal according to the difference between the drive cycle and the actual vehicle speeds. This signal is sent to the Hybrid Central Control (HCC) block. The HCC block is the heart of the hybrid model and computes the torques of each power sources so as to satisfy the driver demand. The performance of this block is based on the designed control strategy. The output signals of the HCC block is sent to the ICE and hydraulic P/M blocks. These two blocks generate the demanding torques. Finally, both output torques are coupled in torque coupling and final signal is sent to the vehicle dynamic block. The equations for dynamics of bus considering the resistance forces are modeled in the vehicle dynamic block.

A. Hydraulic P/M Model

The main consideration in modeling of components of a powertrain is their efficiencies. In 1950, Wilson proposed a model that can determine the hydraulic machines' efficiencies with the use of volumetric and torque losses. Some experiments should be conducted in order to use the Wilson's model which is a constraint for the use of the model. Using the efficiency map is the alternative choice. If we use the efficiency map, there is not any constraint to model the hydraulic machine.

There is an efficiency map for hydraulic P/M in the manufacturer's catalogue (Fig. 2). The volumetric and total efficiencies of hydraulic P/M in the pump mode operation are determined by use of this map. The efficiency map is implemented by use of look-up tables.

The following equations

$$\eta_v = \frac{q_v \times 1000}{v_g \times n} \tag{1}$$

$$\eta_t = \frac{q_v \times \Delta p}{p_{max} \times 600} \tag{2}$$

$$\eta_{mh} = \frac{\eta_t}{\eta_v} \tag{3}$$

have been used to determine the efficiencies of the hydraulic P/M. In these equations p_{max} and q_v are determined by the map of Fig. 2. It should be noted that these efficiencies are for the pump mode operation of hydraulic P/M. However, the above efficiencies can be used for motor mode operation with high accuracy. This claim would be true according to comparison between the efficiencies map of available hydraulic axial piston P/M's in their both pump and motor operation.

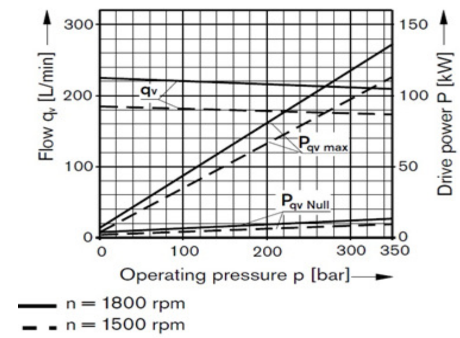

Figure 2 - The Map for determining the efficiencies of hydraulic P/M

In addition to hydraulic P/M efficiency, we need to model the torque generation using the following equations:

$$T_{pump} = \frac{v_g \times \Delta p}{20 \times \pi \times \eta_{mh}} \tag{4}$$

$$T_{motor} = \frac{v_g \times \Delta p \times \eta_{mh}}{20 \times \pi} \tag{5}$$

It should be noted that the torque which the hydraulic P/M can be produced is dependent on the pressure difference in the hydraulic circuit and the P/M displacement. There is a separate block in the P/M model that computes this torque.

For controlling the hydraulic P/M produced torque, the only controllable parameter is the P/M displacement, v_g as seen from equation (4). Therefore, an appropriate control signal should be sent to the P/M block in order to achieve the desired torque at the P/M shaft. The control signal is generated by a PID controller according to the difference between the desired P/M torque and its real one. The controller model for P/M displacement is shown in Fig. 3.

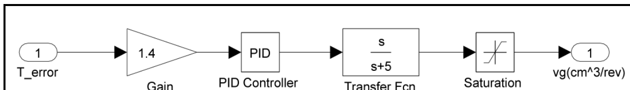

Figure 3 - Model for controller of hydraulic P/M Displacement

B. ICE model

The efficiency of ICE should be considered in order to model the engine accurately. The efficiency of an ICE can be represented by BSFC map. There is a look-up table in the model of ICE that computes the BSFC of engine in every speed and torque. Thus, fuel consumption of ICE can be computed by using this look-up table. Also, there is a block in the ICE model that determines the torque output of the engine. The provided engine torque is constrained by its upper band. The model for ICE is shown in Fig. 4.

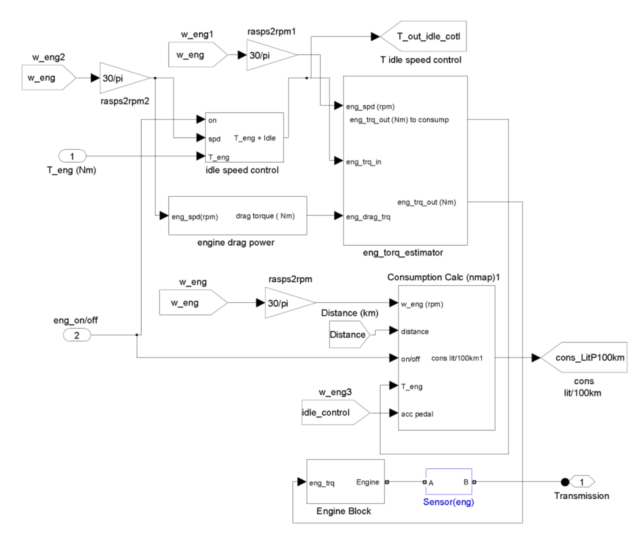

Figure 4 - Model for ICE

C. Accumulator Model

An accumulator in hydraulic hybrid drivetrain plays the role of battery in an electric hybrid. Bladder accumulators store energy in form of pressurized gas. The thermodynamic process of the gas behavior should be specified in order to model the accumulator. In reality, the process that gas acts on happens so fast that there is no heat transfer between gas and its environment. Speed of the process depends strongly on the frequency of torque command signal which is sent from HCC block. This frequency is normally very high for most hybrid control strategies. As a result the gas process is assumed to be adiabatic. The corresponding equation for gas behavior is as follows:

$$p_0 \times {v_0}^n = p_1 \times {v_1}^n = p_2 \times {v_2}^n = p_x \times {v_x}^n \qquad (6)$$

In the above equation the initial pressure, the minimum and the maximum pressures of accumulator gas are shown by p_0, p_1 and p_2, respectively. Also v_0, v_1 and v_2 represent the volumes of corresponding situations. The values of pressure and volume of gas in an unknown situation is shown by p_x and v_x. Also, n represents the gas specific heat ratio (for N2 is 1.4).

In addition to the gas process, the values of maximum and minimum operating pressures of the hydraulic system as well as the pre-charge pressure of the accumulator should be determined. The maximum operating pressure of the hydraulic P/M is 345 bars and pre-charge pressure of accumulator is 140 bars. The minimum operating pressure of the hydraulic circuit can be specified using these values by following equation.

$$p_{min} \approx 0.9 \times p_{pre} \qquad (7)$$

A look-up table has been used in order to compute the accumulator block outputs. This look-up table is based on the provided map in the accumulator catalogue. The map, Fig. 5, determines the available volume of fluid that can be stored in the accumulator for different pressures. Actually, this map considers the efficiency of accumulator operation. The most important output variable of the accumulator block is its State-of-Charge (SoC). SoC represents the ratio of fluid volume to its maximum by following equation

$$SoC(\%) = \frac{v_x}{v\_\max} \times 100 \qquad (8)$$

The maximum fluid volume in the accumulator corresponds to the maximum gas pressure situation.

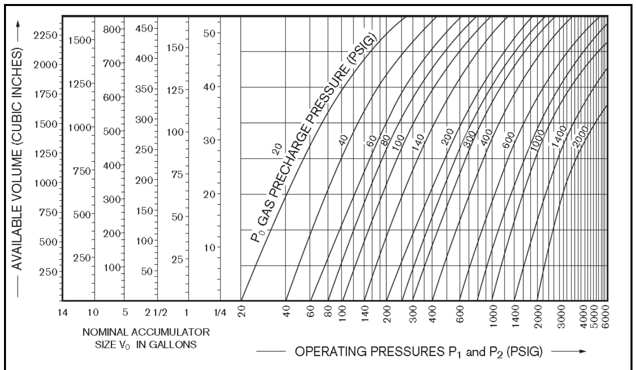

Figure 5 - Map of Volume-Pressure for Bladder Type Accumulator

D. Transmission Model

The transmission of the PHH powertrain has a post-transmission torque coupling. This coupling contains two spur gears on the shafts of two power sources. Each of the power sources has its own gearbox. According to the controllable nature of hydraulic P/M, it has only a one-speed gearbox. But a four-speed automatic gearbox is used for improving the output torque of ICE. This gearbox is the one that was used in the conventional powertrain of O457 bus. Model of the four-speed automatic transmission contains three basic parts:

- *Gear shifting control*. This block computes the gear number needed according to the vehicle speed and a map from the gearbox catalogue. A state-flow block has been used for implementing the control scheme.
- *Gearbox*. This block contains four gears with different ratios. The appropriate gear number is chosen according to the control signal which is received from gear shifting control block.
- *Torque Convertor*. A torque convertor is a hydraulic set that transmits the power from engine to the gears. Model for this component has been created according to its catalogue, either.

All connections in the transmission block are made by using SimDriveline toolbox in MATLAB. The model of gearbox for ICE is shown in Fig. 6.

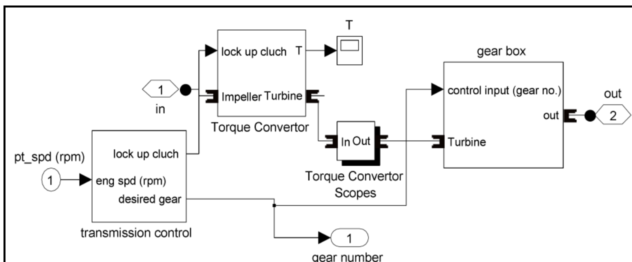

Figure 6 - Model of Automatic transmission

IV. HYBRID CENTRAL CONTROL UNIT

The proposed control strategy for the PHH powertrain has been implemented in HCC block. The driver demanding torque, vehicle speed and the accumulators SoC are input variables for the HCC block. In this block, the desired torque of two power sources is determined according to the control strategy and the input variables. The control strategy is as follow:

- *First mode*. The bus is in acceleration mode, driver demanding torque is below T_0, vehicle speed is below V_0 and the SoC is above the minimum allowable value. The demanding torque from driver is provided by hydraulic P/M alone and the ICE is off. T_0 is 500 Nm and V_0 is 5 km/h in the control strategy.
- *Second mode*. The bus is in acceleration mode, driver demanding torque is above T_0, vehicle speed is above V_0 and the *SoC* is above the minimum allowable value. In this mode the ICE turns on and operates as the main power source of the PHH powertrain. Also the ICE would be helped by the hydraulic P/M in situation that the driver demanding torque is above the maximum torque generated by the engine. Accumulator charging is not considered in this mode.
- *Third mode*. The bus is in acceleration mode and the *SoC* of accumulator is not above the minimum allowable value. In this mode the ICE is the only power source in PHH powertrain. In order to keep the ICE in its most efficient region, the torque generated by ICE could be more than the driver demand torque for most of situations. In these situations additional engine torque is stored in accumulators by hydraulic P/M.
- *Fourth mode*. The bus is in braking mode. In this mode the hydraulic P/M is working as a pump and generates the main braking torque for the bus (Regenerative Braking). The mechanical braking system of the bus would be used if the generated torque by the hydraulic P/M is not sufficient.

In the proposed control strategy, the values of desired torque of both power sources have been determined in such a way that the command signal frequencies are not high. The reliability of model would be reduced when high frequencies exist in these signals. The implementation of this control strategy has been performed by a state-flow block.

V. ACCUMULATOR FINAL SIZING

As mentioned before, the energy storage sizing of proposed PHH powertrain needs some information from simulation of the model. After the complete bus modeling and its simulation in the Nuremburg drive cycle, the appropriate volume and number of accumulators are determined. The designing volume choices for nominal accumulators are 20, 35 and 50 liters and total volume ranges between 60 to 300 liters. The designing procedure is as follow: At first number of accumulators are determined by assuming 20 liters for its volume. The complete model has been simulated in the Nuremburg drive cycle for this reason. In this stage, the least fuel consumption with the less number of accumulators is chosen. After determination of total required volume, the accumulator type is specified. To do this, the equal volume which should be reached by accumulators from 35 and 50 liters type are determined. Finally the model is simulated for all three types of accumulators with equal total volume. The selected accumulators for the PHH powertrain are two 50 liters accumulators. Every accumulator has the gross weight of 220 kilograms.

VI. SIMULATION RESULTS

After designing and modeling stages, the complete O457 urban bus model has been simulated in two different drive cycles that each of them contains a serial of Tehran and Nuremburg drive cycles. Use of serial drive cycles improves the simulations comparison validity. The simulations have been performed with step size of 0.01 seconds. Results for speeds of hybrid bus and the drive cycle, the torques of two power generation sources and the *SoC* of accumulator have been shown in figures 7-10. For a better resolution, the speeds and torques results are shown in a short period of the Nuremburg drive cycle. Moreover, in order to evaluate the effect of hybridization, a model of conventional O457 urban bus has been created in MATLAB/Simulink and its simulation results are compared to those of hybrid bus model. The values of fuel consumptions for both conventional and hybrid bus are presented in Table III. As can be seen, by implementing the PHH powertrain the fuel consumption of O457 urban bus is reduced 26.1% in the Nuremburg drive cycle and 28.8% in the Tehran drive cycle.

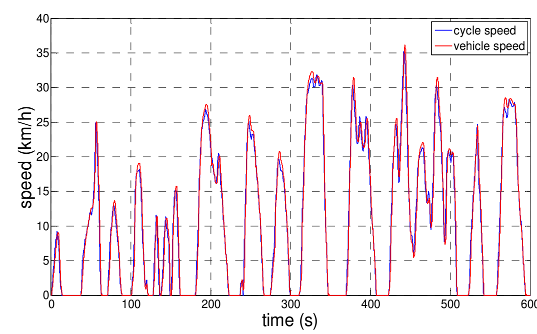

Figure 7 - bus and drive cycle speeds

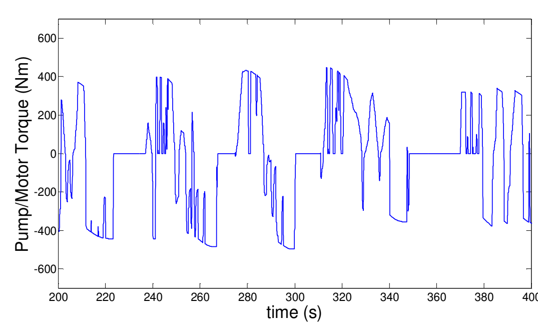

Figure 8 - Hydraulic P/M Torque (Nm)

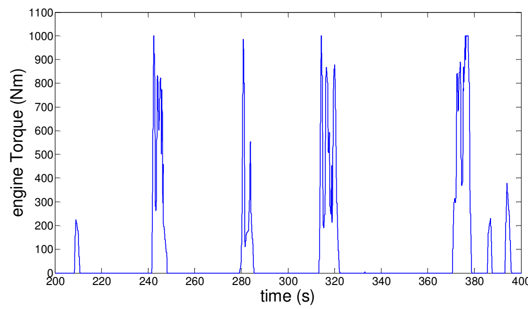

Figure 9 - ICE Torque (Nm)

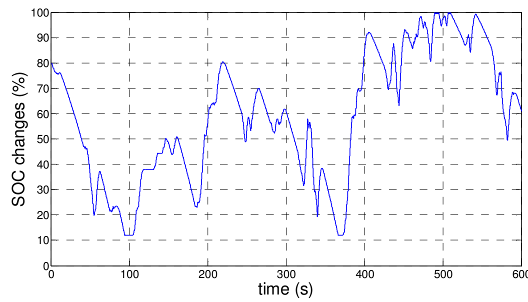

Figure 10 - *SoC* changes

TABLE III. COMPARISON OF FUEL CONSUMPTION FOR CONVENTIONAL AND PHH BUSES

Drive Cycle	Fuel Consumption of Conventional Bus (lit in 100 km)	Fuel Consumption of Hybrid Bus (lit in 100 km)	Reduction of Fuel Consumption (%)	Mean absolute error (km/h)
Tehran	69.5	49.5	28.8	0.9
Nuremburg	70.6	52.2	26.1	0.4

In simulation stage, the performance of the hybrid bus is tested, too. The ability of bus to accelerate and its gradeability are two parameters which are known as the performance of bus. The results of performance simulations for both conventional and hybrid bus are presented in Table IV. As can be seen the ability of hybrid bus in acceleration is the same as the conventional bus. But the gradeabilty of hybrid bus is lower than the conventional one.

TABLE IV. COMPARISON OF PERFORMANCES FOR CONVENTIONAL AND PHH BUSES

Performance Criteria	Constraint	Conventional Bus	PHH Bus
Acceleration	0 to 25 km/h	3.3 s	4.6 s
	25 to 35 km/h	2.5 s	2.4 s
	35 to 40 km/h	1.4 s	1.2 s
	40 to 44 km/h	1.3 s	1.1 s
	0 to 65 km/h	17.8 s	18.3 s
Gradeability	4% slope	94.5 km/h	76.5 km/h
	7% slope	80 km/h	51 km/h
	10% slope	50 km/h	39 km/h
	14% slope	35 km/h	21 km/h

VII. CONCLUSION

In this paper, a parallel hydraulic hybrid powertrain for an urban bus has been designed, modeled and simulated. The powertrain consists of a diesel ICE as the primary and a hydraulic P/M as the secondary power sources. Also, the hydraulic accumulators have been used as the energy storage components. The two power sources have been connected to the driven shaft through a post-transmission torque coupling. The modeling of hybrid powertrain components has been performed considering their efficiencies which are modeled by use of look-up tables. The feed forward model for the hybrid powertrain has been created in MATLAB/Simulink. Simulation of the model has been performed using a rule-based control strategy. Comparison with the conventional model shows at least 26 percent reduction of fuel consumption for hydraulic hybrid technology. Moreover, the performance of hybrid bus in comparison with those of conventional bus is acceptable. The generated model of parallel hydraulic hybrid bus can be used for performing further simulations and also implementing more advanced control strategies.

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