REAL TIME HARDWARE-IN-THE-LOOP VEHICLE SIMULATION

Sheran Alles, Curtis Swick', Syed Mahmud, Feng Lin
Electrical & Computer Engineering Dept. Wayne State University
Detroit, MI 48202

'Ford Motor Co.
Electrical/Electronic System Engineering Office
Dearborn, MI 48121

ABSTRACT
A computer simulation which simulates an engine, driveline, vehicle and tire/road surface models are described. Along with the models described, hardware of several control units (this specific case) were linked together to provide a generic real time Hardware-in-the-Loop (HTL) simulation. Several reasons exist for developing a generic real time HTL simulation:

- to provide capability to verify analytical and experimental data
- to provide capability to acquire vehicle test parameters to input to HTL simulation
- provides powerful, complex and dynamic real time simulation yet portable (unlike large mainframe systems[3]) and cost effective.

1.0 INTRODUCTION
A realistic simulation of the dynamic behavior of the total vehicle system is needed for:

- vehicle performance under different conditions of driveability
- traction control for various road surfaces
- control law strategy development

This simulation development was such that it produced repeatable simulation runs that could be compared to actual vehicle data parameters. This required that a real-time HTL simulation environment be developed. The environment consists of:

- vehicle model
- driveline model
- user interfaces model
- hardware (i.e. electronic modules and I/O.)

Included also was the need to develop this simulation in a generic form, thus providing the capability to change vehicle dynamics, driveline or hardware with minimal down time.

This paper will outline the major features of the simulation models and the advantages of the HTL simulation design.

2.0 SYSTEM DESCRIPTION
In this section a brief description of the different models used, for this HITL development, will be presented.

2.1 ENGINE MODEL
Since the model developed was for a port fuel injected engine, wall wetting can be neglected. Although an important phase of a vehicle is starting the engine in cold conditions, our specific application (at this time) was for warm engine condition, thus initial transients may be neglected. Therefore, all engine model development was with warm engine data. Refer to Figure 1 for a diagram of the engine model [1,2,4]. The equivalent throttle angle is a combination of the primary throttle (driver controlled) and a secondary throttle (project specific application). Figure 2 is a typical induction process flow for manifold air pressure (MAP). When the manifold pressure is less than about half the atmospheric pressure, a choked or sonic condition exists and the mass air flow rate is a function of the equivalent throttle angle only. When the manifold pressure is greater than half the atmospheric pressure, the mass air flow rate is given by a root pressure relation.

The mass air flow rate can be approximated by the following equations:

\[ m_a = k_n F(\Theta) g(P_m) \]

where:

\[ g(P_m) = 1; \text{ for } p < P_{amb}/2 \]
F(0) is a function of the discharge coefficient and geometric area of the throttle bore, which can be approximated by a high order polynomial. However, this can become extremely complex especially with the two throttle system. Therefore a look-up table or a regressed equation, using actual air flow data, will be used. Air flow rate is assumed independent of the presence of fuel. The properties of air flow in the manifold are assumed uniform with constant manifold temperature. This assumption greatly simplifies the manifold modeling. Generally, changes in temperature influence the condensation and vaporization transients which is important in carbureted engines. The air can be assumed to behave as an ideal gas, with no exhaust gas recirculation (EGR) or manifold leaks present.

Therefore, the equation for manifold pressure rate becomes:

\[ p_m = k_p \left( \frac{m_a}{m_f} - M \right) \]

considering the fuel from the injectors, \( m_f \) becomes

\[ m_f = A_o (P_W - A_1) N_{rpm} \]

The mass flow rate out of the manifold and into the cylinders is obtained by considering the engine as a pump. Generally, port injected systems use the speed density to determine flow rate at the inlet port. Thus the volumetric efficiency defined as:

\[ \eta_v = \frac{\text{Actual mass of air inducted}}{\text{mass of air inducted at intake condition}} \]

for a four-stroke engine,

\[ M = N_{rpm} \left( \frac{V_d}{2} \right) \left( \frac{p_m}{RT_m} \right) n_v \]

For constant intake temperatures and exhaust gas pressure, volumetric efficiency can be regressed as a high order polynomial in speed and manifold pressure from steady state engine mapping data. Thus,

\[ M = f(N_{rpm}, p_m) \]

There is always an induction-to-power stroke lag because the engine torque developed at any time is a function of the mass flow rates sampled one induction event earlier.

At a speed of \( N_{rpm} \), the rate of change of crank angle is \( 6N_{rpm} \) deg/sec (i.e. \( 1/6N_{rpm} \) sec/deg). To turn one combustion cycle (720 degs) it will take \( 120/N_{rpm} \) secs. For a \( n \) cylinder engine, the time delay is \( 120/n N_{rpm} \) secs. The torque generated from the combustion process primarily depends on the air, fuel, residual gas, and spark angle. By regressing dynamometer engine mapping data, engine torque \( T_e \) becomes:

\[ T_e = f(A/F_{delay}, EGR_{delay}, M_{delay}, spark, N_{rpm}) \]

### 2.2 DRIVELINE MODEL

The detailed driveline model[5] has been modified to meet the preliminary requirements for the HITL simulation. Assume power from the engine is transmitted to the tires through rigid shafts and gears with no frictional loss. The automatic transmission is assumed to be first gear, which is sufficient for low speed Traction Assist (TA). The engine is assumed to be rigidly mounted to the chassis. At present, the model is operational for a lumped gear ratio (Refer to Figure 3).

\[ \omega_e = N/2(\omega_{fl} + \omega_{fr}) \]

Neglecting the inertia of the differential angular acceleration of the engine becomes:

\[ \dot{\omega}_e = N/2(\dot{\omega}_{fl} + \dot{\omega}_{fr}) \]

From Newton’s laws of motion, for left wheel (similar for right wheel)

\[ J_{fl} \dot{\omega}_{fl} = T_{shaft} - T_{flb} - T_{fl} \]

Assuming equally split shafts the torque of the engine, \( T_e \), is

\[ T_e = (J_e + J_{trans}) \dot{\omega}_e + (2/N) T_{shaft} \]

### 2.3 TIRE AND VEHICLE MODEL

It is important to have a realistic non-linear tire model for this simulation as Traction Assist (TA) studies depend mainly on the road and tire surfaces. In maneuvering the vehicle, the lateral and longitudinal motions of the vehicle are strongly coupled with the tire forces, and large slips occur simultaneously. For simplicity, the tire model represents a vehicle traveling on a straight road with no influence
due to tire pressure. For future simulations tire modeling will need to include the effects of longitudinal forces, lateral forces and tire deformation (i.e. tire inflation). This becomes more apparent when the vehicle slips while cornering. Figure 4 describes the wheel rotation dynamics (e.g. left wheel).

The difference between the tires circumferential speed and the vehicle speed is modeled as a continuous deformation, \( s \). Therefore the following is true:

\[
\frac{r \cdot \omega_d - v_{veh}}{v_{veh}} = s
\]

Influenced by the load on the wheel, \( F_z \), the tires produce a circumferential force, \( F_y \), which accelerates the vehicle. When braking, wheel speed < \( v_{veh} \) thus slip becomes negative, while during acceleration the slip is positive.

\[
u_{\text{peak}} = \frac{F_y}{F_z}
\]

\[
u = (1 - A \cdot v_{veh}) u_{\text{peak}}
\]

normalized slip \( s = \frac{C_s \cdot s}{u_{F_z}(1+s)}\)

Refer to [7] for a detailed tire model description. At present, steering system dynamics, effects of the lateral load transfer during cornering, and braking distribution on the vehicle system dynamics are not accounted for, although they are important in many different types of simulations.

2.4 BRAKE MODEL

Figure 5 shows a typical brake system. In this implementation there are two ON/OFF solenoids per brake caliper. The brake pressure is increased by opening the inlet valves and decreased by opening the outlet valves which sends the brake fluid back to the reservoir. This simple model adequately simulates the main behavior of a typical electro-hydraulic brake system [6]. The time delay of the pump is modeled as a first order system. The brake pressure is obtained by regressing data for the volume of the brake fluid passed to the caliper, including the activation delays. The caliper torque is obtained from a look-up table as it is a function of many variables, that is, brake pressure, contact surface area of brake shoe, friction between shoe and rotor, and temperature. The line dynamics, solenoid dynamics and various leakages in the hydraulic path have not been considered.

3.0 IMPLEMENTATION

An IBM PS/2 486, 33 MHz computer was chosen for developing the mathematical models. The I/O cards were chosen such that it was not necessary to always utilize the PC interrupts (i.e. generation of wheel speed signals). Refer to Figure 6 and 7 for a complete diagram of the computer layout and HITL bench system. This I/O hardware was chosen to meet requirements for user intervention.

The system requirement for the HITL simulation is to be able to run "real-time". This has been achieved by several methods. First, actual hardware was chosen for the Electronic Control Unit (ECU), TA, and Throttle Control Units (TCU). This allows the control units to utilize the internal processors and strategies without interference. Also, this allows the computer to off-load some extremely intense calculations. Without this, the simulation could not have been contained in the PC. Therefore, a mainframe would have been necessary and this would have removed the scope of this design. That is, its portability and ability to be used as a simulator and data acquisition system. Secondly, the mathematical models are scheduled according to the needed update rates. That is, the modules that have to be updated at the fastest rate are scheduled first and progress to the slowest update rate. Thus, the simulation to date can achieve a frame rate of 5ms. The goal is 1ms, however this has not been tested. It might be noted here that if the PC cannot attain the 1ms frame rate, then transputers will be utilized.

4.0 APPLICATION

A specific application for the hardware-in-the-loop (HITL) simulation was for Traction Assist. To obtain stability, steerability, and maximum traction, the effective torque on the wheels should be maintained at about 0 - 15% slip. This can be best achieved in three ways:

1. the application of brake torque to control spin
2. effectively reducing the engine torque
3. combination of 1 and 2

The braking solution is effective for reducing speed in a slip condition but repeated use of the brakes will cause excessive wear and enormous heat, which could deteriorate other brake components. This is similar to driving with the hand brake on. The engine managed torque control technique is good for uniform u-surfaces but will not suffice when the wheels are on split u-surfaces. The engine managed torque reduction technique can be achieved by the following methods or combinations of them:

1. reducing mass air flow into cylinders
2. cutting off injectors
3. retarding spark

Generally, fuel injection control and spark retardation have the fastest response for torque reduction but may create problems for emission control and may degrade the catalytic converter. Throttle only control is suitable for higher u-surfaces (i.e. u>0.25) however, it may not be able to control effectively depending on the response time of the throttle. Therefore a combination of engine managed torque reduction and brake control is the most effective strategy for all speed traction assist.

With the onset of wheel slip, the ECU controls the throttle, spark and fuel while the Traction Assist module controls the braking function. Figures 8a-c (engine control only) shows simulations for a vehicle moving on uniform u-surface, Figures 9a-b (brake only) shows a simulation for split u-surface (i.e. one wheel on ice(u=0.1) and the other on dry pavement(u=1.0)) and Figure 10 shows the combination of engine and brake control. The driver has demanded a wide-open-throttle (WOT). The traction assist becomes operational after a certain threshold in the wheel slip is reached. Also when starting the vehicle on an inclination and even in "stop-go" situations, the wheel slip threshold may need to be adjusted dynamically. This simulation allows the user to input desired parameters on-line to develop optimum performance, stability and maneuverability. The performance of the different traction assist strategies can be better studied. The different control strategies could be saved in files and later compared or even re-run depending on the requirements.

The real-time-simulation was used, in this instance, for developing the Traction Assist System. However, it should be noted that this was just one specific application. The development of the HITL simulation was designed and developed for several purposes. One, was to provide a simulation that would be generic such that it would not be limited to one specific application (i.e. Traction Assist)[8]. By developing the models as shown in Figure 6, many other applications are possible. Secondly, it was desirable to provide a low cost, yet accurate, HITL simulation.

SUMMARY

A sufficiently detailed real-time HITL simulation has been developed. The HITL simulation structure and models were designed generically such that it can be utilized for a variety of control system developments. The simulation shall be a very useful tool for the software and hardware development engineer. The simulation will reduce the amount of time for hardware testing, now done in the vehicle, and provide a more analytical approach to optimizing control system design. Also, the software strategy engineer will have the advantage of having a dynamic real time simulation over the present static testing now performed.

Future development for the simulation is to incorporate higher order models (i.e. lateral forces in tire models, suspension models for driveline models, etc.). This may require the use of transputers to maintain the real time environment desired. The future goals for the HITL simulation is to incorporate transputers for each mathematical model.

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REFERENCES


[6] D. Hrovat, "Personal Correspondence"


NOMENCLATURE

- $A_0$ flow rate from injector
- $A_1$ lag in injector opening, s
- $A/F$ air fuel ratio
- $A_s$ tire slip stiffness, Nm/slip
- $C_s$ slip coefficient
- $F$ force on tire
- $EGR$ exhaust gas recirculation
- $J_e$ engine inertia
- $J_{fl,fr}$ front left, right inertias
- $J_{trans}$ transmission inertia
- $k$ characteristic coeff. of throttle, g/s
- $k_{dp}$ prop. const. for ideal gas
- $M$ mass out of manifold
- $M_t$ traction momentum
- $m_{af}$ mass of air, fuel
- $N$ gear ratio
- $N_{nj}$ number of injectors ON
- $N_{rpm}$ engine speed in rpm
- $\eta_v$ volumetric efficiency
- $P_{atm}$ atmospheric pressure, psi
- $P_{man}$ manifold pressure, psi
- $PW$ command to injectors, s
- $r$ average radius of tire
- $T_{fl,fr}$ left, right wheel torque
- $T_{bfl,fr}$ left, right brake torque
- $T_{shaft}$ shaft torque
- $u_l$ coefficient of friction
- $V_d$ engine displacement, cc.
- $V_{veh}$ vehicle speed, mph
- $\omega$ engine angular speed, rps
- $\omega_{fl,fr}$ front left, right angular spd.

Figure 1 Engine Model

Figure 2 A Typical IC Engine Induction MAP

Figure 3 Driveline Model

Figure 4 Tire Model
Figure 5  Brake Model

Figure 6  Test Bench Hardware Interconnection

Figure 7

Figure 8a  Brake Control Only

Figure 8b  Engine Control Only

Figure 9a  Brake Control Only

Figure 9b  Engine & Brake Control

Figure 10