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Advanced Real Time Powertrain Systems Analysis

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ABSTRACT

This paper describes a combined analytical and experimental hardware-in-the-loop powertrain systems analysis methodology. Central to the implementation of this methodology is a real time dynamic system simulation computer such as the high-speed Applied Dynamics Model AD10. For automotive engine control system studies, wide bandwidth in-cylinder combustion pressure sensor signals are input to the AD10 computer. Control commands are calculated and communicated at high data rates to throttle valve, spark ignition, and fuel injector actuators. Both simulation and experimental results are presented. Using this approach, the functional improvements associated with various control philosophies can be determined.

INTRODUCTION

For many powertrain optimization and control problems neither testing nor computer simulation provides satisfactory solutions. Testing is reliable but is handicapped as a design tool because of the difficulty in performing design parameter variation studies. With computer simulation, design parameter studies are readily performed, but results may be unreliable due to inaccurate mathematical models of complex systems. A combination of real time process simulation and hardware-in-the-loop testing may be the most effective approach to many powertrain analysis and vehicle control problems (1). The ultimate simulation of on-board computer control functions linked to either powertrain hardware or a powertrain model would allow real time analysis of potential control, sensor, actuator and on-board computer requirements (2).

With the advent of on-board microcomputer controllers, a simulation of computer control functions initially linked to a simulation of the process to be controlled, allows debugging of the real time controller functions in a synchronous and, if desired, nonrandom environment where experimental replication is possible. This is a significant aid to powertrain control analysis and controller sysnthesis (3). To be an effective tool, the process model must contain the key or dominant dynamic cause-and-effect relationships germane to real time information processing and control system synthesis (4-6). In the work presented here, the research deals with the concept of in-cylinder combustion pressure sensor based dynamic control (7,8). Consequently, in order to study a controller simulation that uses cylinder pressure information, it is necessary to develop a dynamic thermodynamic engine model that reflects the cause-and-effect relationships appropriate for in-cylinder pressure analysis.

Generally, models of the thermodynamic portion of the engine dynamic process are quasi-static, often single cylinder representations, and have execution times much slower than real time making such models impractical as a hardware-in-the-loop analysis and design tool (9,10). On the other hand, many earlier engine dynamic models (11-17), though suitable for real time simulation, did not reflect the high frequency thermodynamic characteristics necessary to simulate cylinder combustion pressure behavior. The high-speed, test hardware linkable Applied Dynamics Model AD10 computer (6) provides the necessary execution speed for advanced real time powertrain simulation and control. In conjunction with a hybrid laboratory, a real time nonlinear dynamic model of the powerplant process was developed. It is a four cylinder thermodynamic model with manifold dynamics, port injected fuel, and mass concentration dynamics, as well as sensor and actuator models. Model calibration and validation are conducted on-site. With a real time process model, the on-board microcomputer controller representation can either control the real powerplant or the thermodynamic model, the concept of which is depicted in Figure 1.

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and its associated controller are inadequate.

Extensive laboratory quality engine mounted sensors, including in-cylinder combustion pressure transducers, are interfaced to an Applied Dynamics Model AD10 high-speed computer through signal conditioners, low noise trunklines, and 12-bit analog-to-digital (A/D) converter inputs. Engine mounted actuators, including throttle valve (stepper motor), spark ignition, and fuel injectors (port injection) are similarly interfaced to the AD10 digital-to-analog (D/A) converter outputs. Although only one is shown, the AD10 is actually interfaced to two dynamometer test cells to facilitate concurrent engine only, transmission only, or combined engine with transmission experiments.

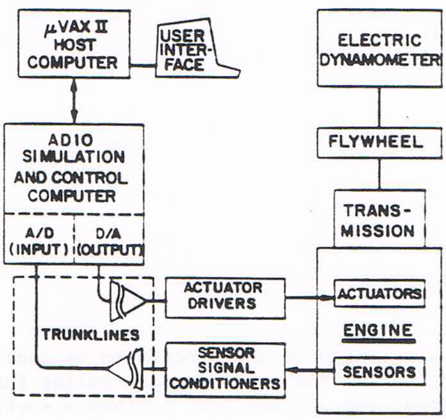

Fig. 3 Real Time Control Systems Laboratory

While implementing the laboratory plan shown in Figure 3, considerable care was exercised in instrumentation practices, particularly grounding, shielding, and power conditioning. The critical trunklines that interface test cell sensor and actuator hardware with the AD10 computer (24 inputs, 24 outputs) utilize low-noise triaxial cable and include wide bandwidth (100 KHertz maximum) isolation amplifiers (flux coupled to avoid potential ground loops) with adjustable gain and bandwidth control. Power conditioners are provided on all computer and instrumentation systems to minimize the effects of undesirable AC power line regulation and noise (SCR controlled electric dynamometers).

The Applied Dynamics International Model AD10 digital computer is a crucial component of this hardware-in-the-loop simulation facility. Generally, the primary AD10 application is in aerospace hardware simulation, development and testing, being used for tasks in which analog or classical hybrid computers have previously been employed. Its advantage over these lies in the precision and repeatability of its numeric results, a characteristic that is due to its digital processors.

An AD10 operating under an MPS10 (Modular Programming System 10) software system and a Digital Equipment Corp. uVAX II under uVMS represent the computer environment in which the simulation runs are performed and the simulation embedded or stand-alone controllers are tested. The AD10 is a specialized, high speed, multiple processor digital machine with analog (+/- 10 volt) input/output channels designed for interfacing with test hardware. It achieves high computational throughput with separate parallel pipelined processors for arithmetic operations, numerical integration, table look-up, decision making, and input/output.

Program development for the AD10 is done on the VAX host and information structures consisting of object code modules and data tables are downloaded prior to program execution. Control over execution of an AD10 program is also retained under the operators interactive supervision via a VAX host resident "relate" program written in FORTRAN. At the end of or during a run, resulting data may be shipped back to the host for monitoring by the operator, either directly or via postprocessing graphic report generating data reduction programs. During AD10 operation, instructions are executed at a 20 to 30 MIPs rate on 16 bit fixed-point operands (48 bit operands within the Numerical Integration Processor) permitting significant size programs (500 to 1000 lines of code) to be executed at frame-times of 100 microseconds, for 10 KHz update rates.

The engine data of primary interest is combustion pressure inside the cylinder, especially during the latter stages of charge compression and the early phase of the power stroke. The sensor employed to measure this property is a flush mounted Kistler pressure transducer based on the piezoelectric effect of quartz. The sensor output is processed by a charge amplifier giving the computer a signal that has both a high S/N ratio and wide bandwidth. This signal is a voltage (-/+ 10 v.) analog of pressure and which in turn is converted to the digital domain in high speed (100 KHz.), dedicated (non-multiplexed) 12 bit A-to-D converters that are a part of the AD10's I/O peripheral.

The intended purpose of the laboratory facility is the exploration of the complete spectrum of possible powertrain control alternatives without regard to the limitations imposed by current automotive sensors, actuators, control strategies, or microprocessors.

CONTROLLER REQUIREMENTS AND STRUCTURE

Combustion Quality Feedback Control

Current automotive microprocessor based engine controllers implement primarily scheduled (open loop) and feedforward control strategies, with very little feedback and adaptive control. Next-generation engine controllers may have available (through new sensors) more and higher quality information about the combustion process in each cylinder, enabling improved feedback and adaptive control, as well as facilitating on-board functional diagnostics.

In particular, the concept of in-cylinder combustion pressure sensor based engine control (Figure 4) offers promise for a substantial improvement in the ability to control engine systems. The idea (Figure 4) is that a dedicated signal processor combines in-cylinder combustion pressure sensor signals with crankshaft position information to produce combustion process parameters, such as 0-90% mass fraction burn duration, work done per cycle, or "indicated" torque. These combustion parameters are then sent to an engine control computer (an AD10 in this figure) at a rate of once per engine cycle. The engine control computer then implements feedforward, feedback, and adaptive lowest level are end-of-burn modules, one for each cylinder and effectively synchronized with each cylinder. Engine cycle related data collection (especially for graphic presentation) is performed by some modules at this place in the program structure.

Controller Functions

The function of the engine controller is to regulate fuel and air-flow into the combustion chamber and to ignite this charge via an appropriately timed firing of the spark plug. All this is done to meet the changing speed and torque demands of the vehicle's operator. The experimetal set-up employed for testing controller operation consists of procedures to run the engine / dynamometer at a fixed rpm over a series of regulated torque level changes. For example, starting from a low level (20 ft-lbs) the A/F charge and spark setting are modified to rapidly transit to a higher load level (e.g. 70 ft-lbs) which is maintained for a number of seconds before applying fuel flow, air flow and spark advance changes to bring the engine back to its former load setting.

The primary control variable that is directly altered in response to load chages is throttle angle setting to control air flow. For precise and rapid throttle plate motion control a stepper motor driver unit is cog belt geared at a 5:1 ratio to the throttle plate shaft. The AD10 control program issues direction and step signals, the latter at up to a 2000 Hz rate. As A/F ratio alteration induces changes in charge burn duration, the secodary control variables of fuel quantity and spark setting come into action. The controller operates the fuel injector pulse width setting to maintain charge burn duration at a predetermined value. This target burn duration value may be selected by the operator to be either fixed or a function of engine speed and load. The overall effect is considerable latitude in A/F ratio control.

A REAL TIME THERMODYNAMIC MODEL

Model Structure

The structure of the dynamic thermodynamic model was synthesized primarily from two physically oriented substructures or models. The first substructure defines a mechanically linked four cylinder pump with a pressure and two zone temperature and time dependent characterization for each cylinder as well as breathing into finite intake and exhaust plenums. The basic cycle analysis is modeled after that of Blumberg and Kummer (20); Blumberg, Lavoie and Tabaczynski (21); and Tobler (22). The second substructure or model, consists of a throttle body with a dual plenum four cylinder pocket manifold, fuel injectors and condensation dynamics, common exhaust plenum, and load and rotational dynamics. This primarily lumped parameter characterization is based on the models of Powell (14), Morris and Powell (2), and Dobner (16).

The Thermodynamic Process

The model used for the present studies is not currently a comprehensive thermodynamic representation. In particular, several simplifying assumptions are made with respect to heat transfer, mass fraction burn behavior, and thermodynamic properties of burned and unburned gasses, which are all at the heart of a rigorous thermodynamic representation. There are two primary reasons for this; (i) the necessity for developing a real time cylinder pressure based control design model implies functional dependence on control variables such as in-cylinder air charge, fuel charge, and spark advance, and on estimates of residual faction, burn duration, mean effective pressure, and so on, and (ii) the lack of an emission requirement for the initial investigation, the modeling of which would also be dependent on accurate estimates of cylinder composition and temperature, perhaps in several temperature zones.

The main governing equations for the open system for each cylinder consists of the first law of thermodynamics

$$E = \Sigma_i h_i m_i + Q - W + \Sigma_e h_e m_e$$

where E is the internal energy rate change, Q is the rate of heat added, W is the work transfer rate by the changing volume, m_i is the mass flow rate into the system from the intake manifold and $_{me}$ is from the exhaust manifold. The quantity h is the entholpy per unit mass. In addition, the formulation employs the ideal gas law with uniform cylinder pressure. The charge is assumed to be a homogeneous mixture of air, vapor fuel and residual gas during intake and before combustion. During combustion there exists a burned and unburned zone, with uniform composition and temperature, separated in volume by a burn propagation or mass fraction burn combustion rate relation. This is represented pictorially in Figure 6.

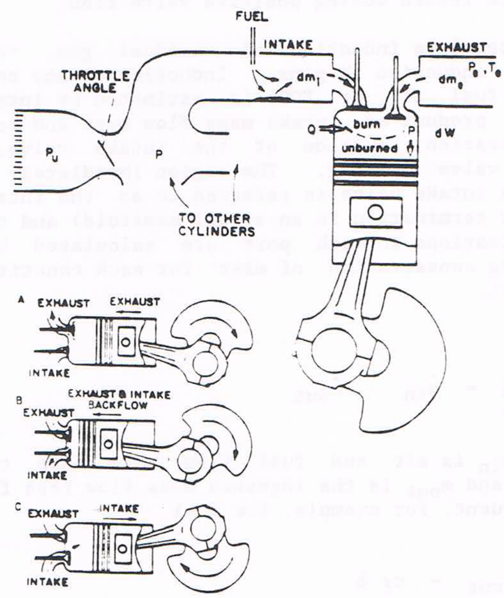

Fig. 6 Intake and Exhaust Flow Events

For burned and unburned gasses the differential internal energy is

$$dE = dE_u + dE_b \ , \quad u = \text{unburned}, \quad b = \text{burned}$$

$$= C_v{}^u \, d(m_u T_u) + C_v{}^b \, d(m_b T_b)$$

where C_v is a constant volume specific heat, m is burned or unburned mass and T is burned or unburned temperature. It is assumed that temperature and mass changes per unit time are larger than constant volume

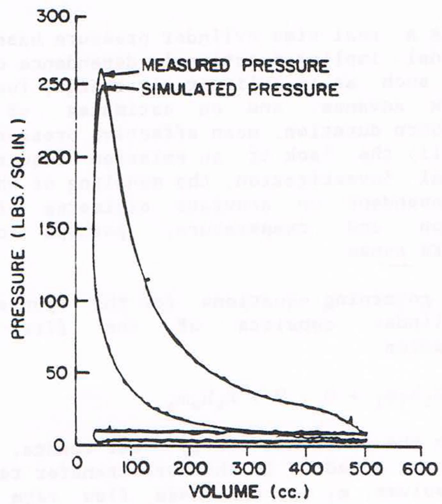

Fig. 8 Pressure Indicator Comparison after Exhaust Valve Adjustment

The Intake Process

During the valve overlap period, mass backflow in both exhaust and intake systems for each cylinder is assumed to be a plug type flow. Only burned (residual) gas crosses the exhaust valves whereas the charge entering the intake manifold during intake valve opening is the first to return during positive valve flow.

After complete induction of residual gas, the fresh charge induction begins. Induction of any component (vapor fuel, air, or EGR) is estimated by integration of the product of intake mass flow rate and component concentration, outside of the intake valve, during intake valve opening. The region immediately outside of each intake valve is referred to as the intake port (runner termination in an actual manifold) and the mass concentrations in each port are calculated by first assuming conservation of mass for each constituent in the port.

Thus,

$$\frac{dm}{dt} = \dot{m}_{in} - \dot{m}_{out}$$

where m_{in} is air and fuel flow rate into the port volume and m_{out} is the ingested mass flow rate for that constituent. For example, for fuel

$$\dot{m}_{fout} = c_f \dot{m}$$

where C_f is the changing ratio of the mass of fuel in the port divided by the total mass in the port and m is the total changing ingested mass flow rate for a particular cylinder. The integration to obtain the constituent masses is a continuous process independent of crankangle. It is assumed that all fuel is burned during the combustion process and all fresh charge becomes residual. A pictorial representation of the process for one cylinder is shown in Figure 9.
It is assumed that there is some fuel condensation near each cylinder port and the fuel condensation model employed for the warm engine simulation is that of Aquino (26). A simulation result for two different injector pulse widths (the same for all cylinders) is

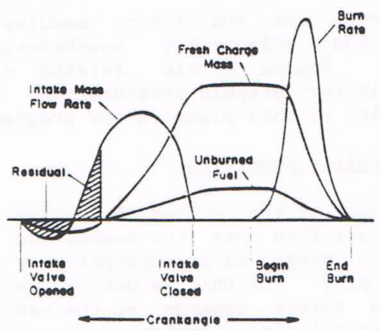

Fig. 9 General Intake Flow Rate and Constituant Behavior During Combustion

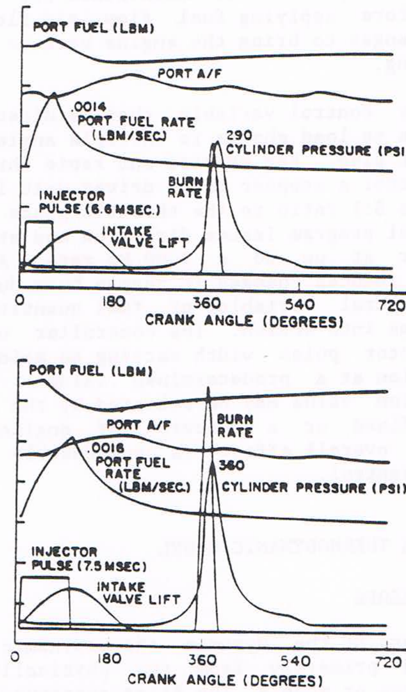

Fig. 10 Examples of Parameter Variation on Engine Model Operation

shown in Figure 10. The injection timing is shown as injection into an open valve which is not the general operating mode for the control study results presented herein. The figure shows the general behavior of the port ingestion fuel rate (characteristic of first order fuel evaporation rate model), the port vapor fuel (which continually increases after injector firing is stopped due to local evaporation), and the varying local port A/F changing during the breathing of each cylinder.

The Induction Process

The induction process consists of a throttle body embedded between an upstream single channel plenum with air filter (or air measuring restriction) and a downstream intake plenum with a local port region near each intake valve. Manifold pressure is assumed uniform (no acoustic propagation) and the total air

There are some additional considerations in dynamic spark and fuel control. For the spark advance control it is desirable to determine predicted burn to obviate knock and misfire, and to provide levels of ability to control for some operating modes (i.e spark margin). This will generally provide a weak control loop (weak spark-fuel interaction at MBT) and liberal compensation gains can be chosen to affect a stability versus response tradeoff. An additional consideration is that the controller actually uses an "average" spark over four cylinder as there is one continually changing spark advance rather than a different advance for each cylinder. For fuel control, the burn duration command is based on engine speed, implicit emissions considerations from experimentally determined set point functions of engine speed and IMEP, and compensation for IMEP variability as well as inequities in work per cylinder.

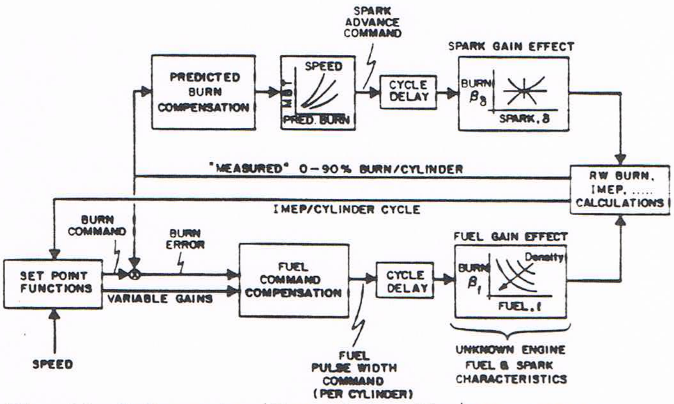

Fig. 13 Information Flow of 0 - 90% Mass Fraction Burn Duration Feedback Control for Each Cylinder

The fuel command compensation shown in Figure 13 is fundamentally a proportional and integral (PI) control. The inner loop control gains (variable gains) are maximized for nonoscillatory dynamic performance which is dependent on burn rate sensitivity to air charge and fuel. The integral error function of the fuel loop is necessary in order to command and achieve 5 percent error resolution in burn.

Controller/Model Developments

While initial developments of the thermodynamic engine model and the combustion pressure based engine controller passed through several stages independent of each other, more recent work on both has been done in concert and with information and experience derived from controller - engine hardware experiments. The model - controller combination has proved very useful in implementing and testing modifications to the controller by providing a safe environment for exercising new code modules destined to control engine - dynamometer hardware in operating regions where a "simple bug" might have catastrophic consequences. At the same time, the relative costs, in time, personnel and facilities, of operating the model versus engine - dynamometer hardware encourages use of the former as long as it closely mirrors the performance of the latter.

Conversely, the development of the model has benefited from repeated comparisons with the "real hardware". The controller which they have in common has here been employed to examine functional differences. Thus when the end-of-burn detection procedure that sucessfully observed this event in the engine was unable to do so in the model, the comparison pointed to an incomplete accounting for pressure changes in the latter. By including an additional term for thermal loss to the cylinder walls in the model, the end of burn could be detected using the identical controller logic.

Results

While both engine model and controller development is continuing at the facility, there are a series of results that indicate the utility and support the validity of the methodology here described. For example, closed loop control was applied to the actual engine hardware and this (and the model) was exercised over a series of ramped load trajectories as shown in Figure 14 for the model and Figure 15 for the engine.

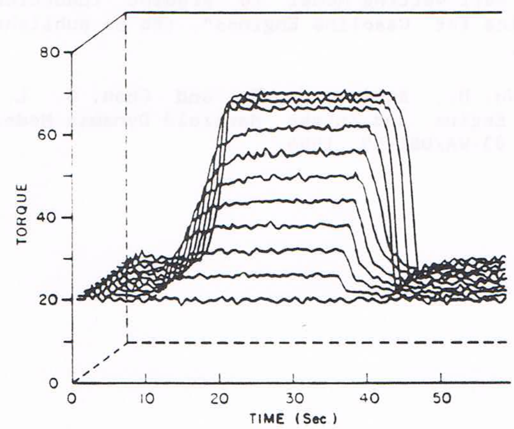

Fig. 14 Thermodynamic Engine Model Multi-Run Torque Output. Composit of Eleven 60 Second Test Runs of Torque Ramps from 20 to 70 FT-LBS.

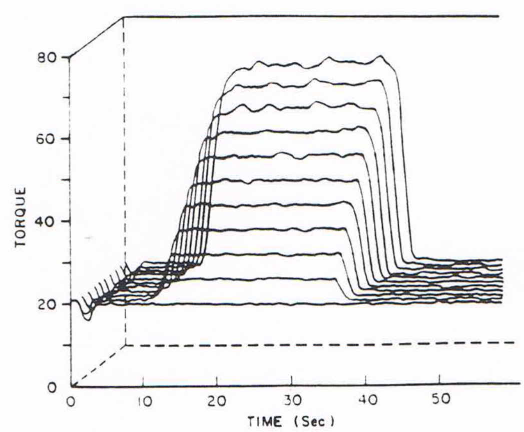

Fig. 15 Engine Hardware Multi-Run Torque Output. Composit of Eleven 60 Second Test Runs of Torque Ramps from 20 to 70 FT-LBS.

There are differences in the high torque regions that may be ascribed to non-linearities in this area of the model. The dynamics of the transition from one load level to another are also an area of difference, but one that points to more basic disagreements between the model and engine that yet remain to be resolved. Despite these and other, less apparent differences, the

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