Dynamic Modeling and Control of Port Fuel Injection Engines

A dissertation submitted in partial fulfillment

of the requirements for the degree of

Master of Technology

in

Mechanical Engineering

(Design Engineering Specialization)

by

Pushkaraj A. Panse

(03310023)

Supervisor:

Professor Shashikanth Suryanarayanan

Department of Mechanical Engineering

Indian Institute of Technology Bombay

July 2005

Approval Sheet

Dissertation entitled "**Dynamic Modeling and Control of Port-Fuel Injection Engines**" submitted by **Pushkaraj A. Panse** (03310023) is approved for the degree of **Master of Technology** under Design Engineering Specialization in **Mechanical Engineering**.

Supervisor:

____

Professor Shashikanth Suryanarayanan

Internal examiner:

____

Professor Anuradda Ganesh

External examiner:

____

Mr. Jagannath

Larsen & Toubro Ltd., Powai

Chairman:

____

Professor Kavi Arya

Date : _____________

Place: Mumbai

Indian Institute of Technology Bombay

July 2005

Abstract

In this dissertation, we study dynamic modeling and control of gasoline powered, four stroke, spark ignited, port fuel injection engines. Control of automotive engines is important because it yields benefits on several fronts such as fuel efficiency, exhaust emission reduction, better power delivery. To achieve this, an automotive engine is equipped with a control system architecture that comprises of a microcontroller, several sensors and actuators. The microcontroller, also called as the engine control unit (ECU), continually identifies what the driver is demanding out of the engine. Accordingly, it continually selects the engine control inputs such as air flow rate, fuel flow rate, and spark timing. The microcontroller also processes the information carried by several sensors for feedback control as well as for diagnostics. The methodology for designing the controller involves modeling the engine and developing the controller in simulated environment as a first step.

In this dissertation, we describe different control issues in an engine and a typical control architecture as a solution to tackle them. Then we model different engine subsystems and the engine phenomena of relevance in "control-oriented sense". Integrating them, we develop a dynamic, control oriented model of a port fuel injection engine. Simulation results for the model in are presented. A feedback controller for idling operation of the engine has been developed and demonstrated to work well through simulations. The model developed could be useful for controller development for other modes of engine operation as well such as transient operation, steady state running etc. The controller development exercise and results can be useful in further study for other similar mode of engine operation, such as cruising.

Contents

List of Figures

List of Abbreviations

BMEP	Brake Mean Effective Pressure
CO	Carbon Monoxide (emission)
DBW	Drive By Wire
EGR	Exhaust Gas Recirculation
FMEP	Frictional Mean Effective Pressure
HC	Hydrocarbon (emission)
IMEP	Indicated Mean Effective Pressure
MBT	Maximum Brake-torque Timing
MEP	Mean Effective Pressure
NO_x	Nitric Oxides (emission)
PFI	Port Fuel Injection
PID	Proportional Integral Derivative (controller)
SI	Spark Ignited
SISO	Single Input Single Output
TWC	Three Way (catalytic) Converter

Nomenclature

Notation	Explanation	Unit
rpm	Revolutions per minute	min^{-1}
η_{ind}	Indicated thermal efficiency	
η_{mech}	Mechanical efficiency	
$\eta_{brake\text{-}thermal}$	Brake thermal efficiency	
T_e	Engine torque	N m
ω	Engine speed	rad/s
$\dot{m}_f$	Mass flow rate of fuel	kg/s
$BMEP$	Brake Mean Effective Pressure	N/m^2
MEP_{fuel}	Fuel Mean Effective Pressure	N/m^2
$IMEP$	Indicated Mean Effective Pressure	N/m^2
V_d	Displaced volume of cylinder	m^3
M_f	Mass of fuel inducted during one stroke	kg
H_l	Calorific value of fuel	J/kg
$\dot{m}_\alpha(t)$	Mass flow rate of air past throttle	kg/s
$\dot{m}_\beta(t)$	Mass flow rate of air inducted into cylinder	kg/s
R	Universal gas constant	J/kg ^oK

Notation	**Explanation**	**Unit**
T_m	Intake manifold pressure	^oK
V_m	Intake manifold pressure	m^3
α	Throttle angle	radians
d_{th}	Diameter of throttle section	m
P_{atm}	Atmospheric pressure	N/m^2
T_{atm}	Atmospheric temperature	^oK
C_d	Coefficient of discharge	
γ	Ratio of specific heats for air, C_p/C_v	
ρ_m	Density of intake manifold pressure	kg/m^3
η_{vol}	Volumetric efficiency	
V_c	Clearance volume	m^3
P_{ex}	Exhaust pressure	N/m^2
P_m	Intake manifold air pressure	N/m^2
m_{ff}	Mass of intake manifold fuel film	kg
τ_{ff}	Time constant for fuel film evaporation	s
M_a	Mass of air inducted in one suction stroke	kg
λ	Air-fuel ratio	
σ	Spark advance	radians
$\delta_{suct-pow}$	Suction to power stroke delay in time	s
$\delta_{spk-pow}$	Sparking to power stroke delay in time	s
T_L	Load torque	N m
J	Mass moment of inertia for crankshaft	kg m^2
P_{cr}	Critical pressure for compressible flow	N/m^2

Chapter 1

Introduction

The subject of this dissertation is design of controller for an internal combustion engine. In particular, this dissertation deals with feedback control of gasoline powered, four stroke, single cylinder, port fuel injection (PFI) automotive engines. An automotive engine is a typical multiple-input multiple output system that has to satisfy a number of performance criteria under different operating conditions. Controller development for an internal combustion engine is a challenging engineering problem that requires background of different disciplines such as thermal engineering, dynamics, control theory. This dissertation presents a control oriented, dynamic model for a port fuel injection engine prototype and discusses development of an idle speed controller for the same. Simulation results for the model and the controller performance are also documented.

An important motivation behind this study is that automatic control of internal combustion engines leads to several benefits such as reduction in emissions, improvement in fuel efficiency, and power delivery. Perhaps the most significant benefit is emission reduction especially when it comes to developing countries like India. The air quality in Indian cities falls considerably below the standards set by the World Health Organization as well as the National

Ambient Air Quality Standards, set by the Indian Central Pollution Control Board [8]. It is also estimated that vehicular exhaust emissions contribute about 63% of total air pollutants emitted in Indian cities such as New Delhi [8]. [2] indicates that in India, the gasoline powered engines contribute to more than 50% to vehicular carbon monoxide (CO) emissions, and more than 80% to vehicular hydrocarbon (HC) emissions. In recent years, the Indian market has been experiencing a large increase in sales of gasoline powered vehicles, mainly motorcycles and cars [6]. This will likely lead to further deterioration of air quality. In light of the situation, there clearly exists an urgent need of efforts to reduce emissions from gasoline engines.

1.1 Problems Addressed

As mentioned earlier, in this dissertation we are interested in control oriented modeling and control of gasoline port fuel injection engines. This dissertation considers following two problems:

1. Problem 1: Given a gasoline port fuel injection engine, develop a control oriented, dynamic model to represent the phenomena occurring in the engine. The model should closely resemble behavior of a warmed up port fuel injection engine. The solution to this problem involves study of various engine phenomena followed by laboratory experiments to identify various parameters in the model, as well as to test validity of the model.
2. Problem 2: Given a control oriented, dynamic model for a gasoline port fuel injection engine at idling condition, develop a simulation based controller so as to satisfy certain performance requirements. In this dissertation, idle speed control is formu-

lated as a SISO (single input single output) problem. A PID control loop is shown to work successfully. Idling mode requires constant speed operation in the event of sudden unknown disturbances. The work done in this dissertation on satisfying idling mode requirements can be easily extended to other engine modes such as cruising.

This dissertation addresses these problems by looking at methods proposed in the literature, and through simulations. MATLAB/SIMULINK was used as a tool for simulation studies.

1.2 Outline of the Dissertation

The remainder of the dissertation is organized as follows. Chapter 2 provides background and preliminary material on gasoline port fuel injection engines as a control system plant. Various ideas to improve performance of these engines have also been discussed in brief. Chapter 3 presents a mean value model for a warmed up gasoline port fuel injection engine. Chapter 4 presents linearized, state space form of the mean value model for idling condition. Simulation results for the two models are presented. Chapter 5 presents design of controller for idle speed control purpose. Simulation results for performance of the engine are also presented. Chapter 6 contains a summary of the dissertation and scope of future work.

Chapter 2

Preliminaries and Literature Survey

The purpose of this chapter is to introduce the reader to control issues in a typical gasoline port fuel injection engines. In Section 2.1, the performance expectations from an automotive engine are first put forward in. Role played by control engineering in meeting those expectations is explained. In Section 2.2, different modes of engine operation and objectives of the engine control system in each of them are stated. Out of the modes that are listed, chapter 5 focuses on controlling a port fuel injection engine in idling mode. In Section 2.3, important control problems in port fuel injection engines are listed. Section 2.4 presents an important dichotomy of control oriented models for automotive engines in general. Out of the two types of models that are discussed, chapter 3 presents a mean value type model for a port fuel injection engine.

2.1 Role of Control Engineering in a Gasoline Port Fuel Injection Engine

Most of the four wheelers today are equipped with gasoline port fuel injection (PFI) engines. Typically these engines are four stroke type, spark ignited (SI), and are based on the Otto cycle to convert chemical energy of the fuel into mechanical work.

The main job of an automotive engine is to produce power as per the driver's demand. As a by-product of power production, PFI engines also produce harmful gases - the most significant being the unburnt and partially burnt hydrocarbons (HC), carbon monoxide (CO) and nitrogen oxides (NO_x). Governments all over the world have enforced stringent norms that have forced automobile manufactures to design engines such that the tailpipe emissions are kept within specified limits during standard emission compliance tests. In a typical emission compliance test called a "drive cycle test", the vehicle is made to follow a certain speed vs time profile. Tailpipe emissions over the entire drive cycle are recorded. The vehicle passes the test only if the emissions are less than prescribed levels (emission norms). Also automobile users expect vehicles to be fuel efficient.[1] In order to help the engine perform on all the three fronts described above, it is equipped with a control unit (Figure 2.1). The job of the engine control unit is to continually choose the engine inputs (air and fuel flow rates, spark timing) so that engine outputs (torque, emissions, and fuel efficiency) meet targets.

In this dissertation, it is assumed that the PFI engine under discussion is equipped with "drive-by-wire" (DBW) technology. This involves actuation of the throttle valve by the engine control unit, and not directly by the driver. The driver's demand is communicated to the controller through a sensor that measures the depression at the accelerator pedal. It is also assumed that the engine does not possess an exhaust gas recirculation (EGR) system.

1 Recently "fuel economy norms" also have been introduced by governments in some parts of the world, e.g. the Europe.

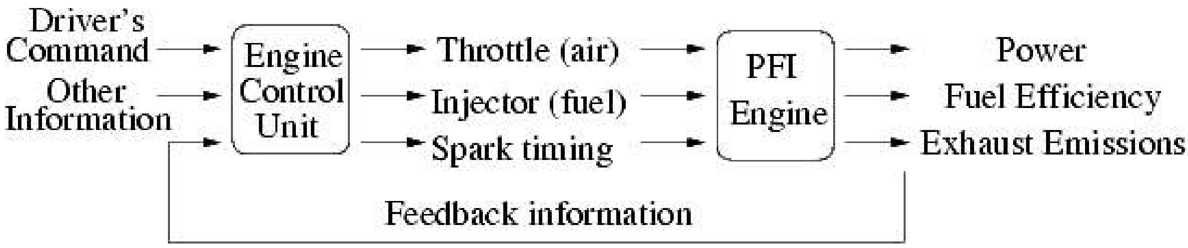

Figure 2.1: Schematic of a typical port fuel injection engine control system

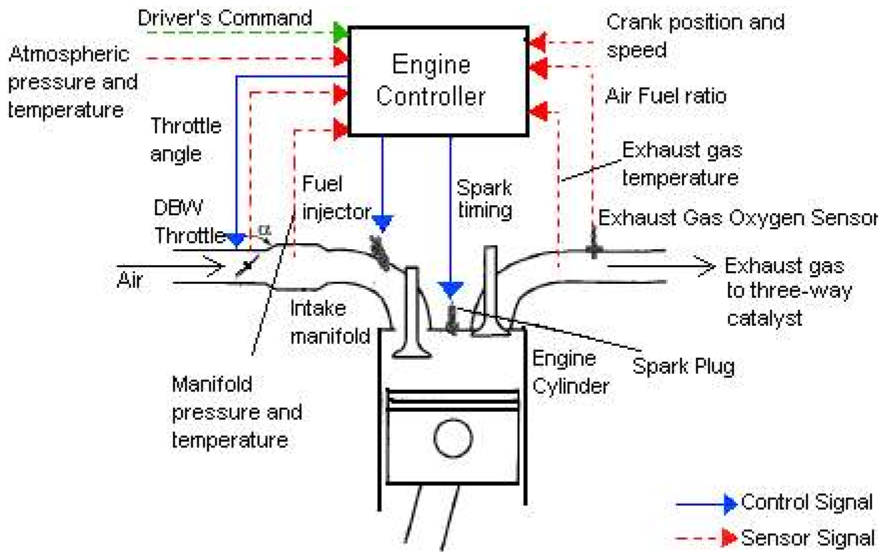

Figure 2.2: Sensing and actuation in case of a drive-by-wire port fuel injection engine

It must be noted that a PFI engine by itself is not an inherently unstable system. Therefore, even without exercising control engineering practices, it is still possible to "run" a typical PFI engine. But under uncontrolled operation, it would most likely not comply with current emission norms nor is likely to be fuel efficient. In PFI engines, the role of control engineering is that of "enabling technology" and not of "core technology". It has been utilized mainly to improve performance.

2.2 Operating Modes: Objectives and Identification

In its life cycle, an automotive engine runs in different operating conditions such as cold start, idling, normal driving etc. Performance objectives during each of these conditions are usually different than the other. Therefore, the engine control system must continually identify what is expected out of the engine and act accordingly. To accomplish this, different operating modes are defined in the engine controller. The controller switches from one mode to another considering the driver's demand - characterized by the states of clutch and accelerator pedal. The choice of identification of operating modes is left to the designer. Important modes for PFI engine operation and a typical mode identification scheme are described below are [9, 3]:

1. **Cold start**: This mode represents a period of rapid transient experienced by the engine from the moment it is just started till it warms up. The cold start mode is characterized by low engine coolant temperature, the clutch being disengaged, and the accelerator pedal not being pressed. It is observed that the HC & CO emitted during the cold start period contribute more than 80% to the total HC & CO emitted in typical drive cycle tests [9]. Also the engine frictional losses are large in comparison with other modes, due to high lubricant viscosity at low temperatures. In line with the cold start behavior of engines, the control objectives during this period are:
    - to minimize the HC & CO emissions emitted during the cold start period in the standard emission compliance tests; and
    - to generate torque that is enough to overcome frictional resistance.

   To solve the cold start control problem, model based dynamic optimization techniques are often used. Trajectories of the engine inputs during the cold start period are sought after as the solution [9].

2. **<u>Idling</u>**: The engine operates in this mode if the clutch is disengaged, the accelerator pedal is not pressed by the driver and the engine is in the "idle speed range" (typically 800-1500 rpm). A separate idle speed controller takes over control of the engine. The reference speed for this mode is set equal to minimum possible speed such that the engine does not shut off. When the clutch is in disengaged condition, the net power-train inertia is very low. As a result, the cyclic fluctuations in torque may cause undesirably large speed fluctuations. Also the engine load during idling is of variable nature and could be a source of sudden drop or rise in the engine speed. The control objectives during idling mode are:
    - to maintain the mean engine speed equal to the reference value; and
    - to minimize speed fluctuations.
3. **<u>Transition into idling</u>**: The engine switches to this mode in either of the following two situations [3]:
    - the engine speed is in transition speed range (typically from 1500 rpm to 2500 rpm) with the accelerator pedal not pressed by the driver and the clutch disengaged; and
    - the engine speed is in the idle speed range with the accelerator pedal not pressed by the driver, and the clutch engaged.

    The control objective is to match the engine speed to the upper limit of the idle speed range i.e. equal to 1500 rpm. In the first situation, the controller "interprets" that the driver may soon want the engine to idle. So, it slows the engine down in order to enter the idling mode soon. In the second situation, the controller "interprets" that the driver may soon want to drive the vehicle (if already in the idling mode) or may soon want the engine to idle (if not in idling mode already). The controller takes decision to maintain the engine in close vicinity of the upper speed limit of the idle speed range. This helps the controller to satisfy the driver demand in any of the two scenarios.

4. **Transition out of idling**: The engine is in this mode if it is in the idle speed range with clutch disengaged and the accelerator pedal is pressed by the driver [3]. The controller interprets that the driver wants to drive (i.e. to end idling). The control objective is to increase the engine speed to the upper limit of the idle speed range. This makes the vehicle ready to "take off".
5. **Drive**: Engine operates in this mode if it is not in any of the above modes. The main control objective is to follow the driver's demand. It is interpreted from how much pressed the accelerator pedal is.

   In modern PFI engines, the drive mode may include additional submodes such as "cruising". While cruising, the main objective is to maintain a steady cruising speed provided the driver's demand is over a certain limit and "not fluctuating much". Basically both, the idle speed control problem and the cruise control problem, are engine speed regulation problems. Historically the problems have been "solved" using centrifugal governors. Modern electronic control systems serve the same purpose while offering considerable freedom in design combined with improvement in performance.

2.3 Important Control Problems

Performance of an automotive engine is often characterized by how it performs during standard drive cycle tests. In a typical drive cycle, the engine has to function in different operating conditions. The torque demand from the engine is often of varying nature and the engine must meet it as quickly as possible. Whereas the other performance deliverables, viz. tailpipe emissions and fuel economy, are recorded and averaged out over the complete cycle (rather than measured in real time). The engine must meet imposed standards on these "averaged deliverables" as well. Developing an engine that performs well in all these three areas requires efforts on both of the following fronts: the mechanical design of the engine as well as the engine controller design.

On the controls side, the usual practice is to design a controller that follows real time requirement of torque. While doing so, the controller also needs to ensure that certain engine inputs are always within their prescribed ranges. It is hoped that such a regulation of inputs would ensure that the averaged out deliverables do not exceed the imposed limits. This approach reduces the degrees of freedom in the engine control problem. For example, out of the three inputs in case of a PFI engine system, the air-fuel ratio and the spark timing are almost always regulated around certain fixed values, whereas the intake air flow is controlled such that real time torque demand is met. In this way, the degrees of freedom for the control problem reduces from three to one. The regulation of air-fuel ratio leads to satisfaction of the emission norms, and that of spark timing yields benefits on the fuel economy front.

1. <u>Air-fuel ratio Control</u>: In order to reduce emissions, PFI engines are equipped with exhaust after-treatment systems that chemically process exhaust gases. The most widely used after-treatment systems consist of a three-way catalytic converter

(TWC) in the exhaust manifold. The TWC considerably reduces quantities of HC, CO and NO_x simultaneously. However, as figure 2.1 shows, it works well only if the air-fuel ratio of the exhaust gas is in close vicinity of the stoichiometric value.

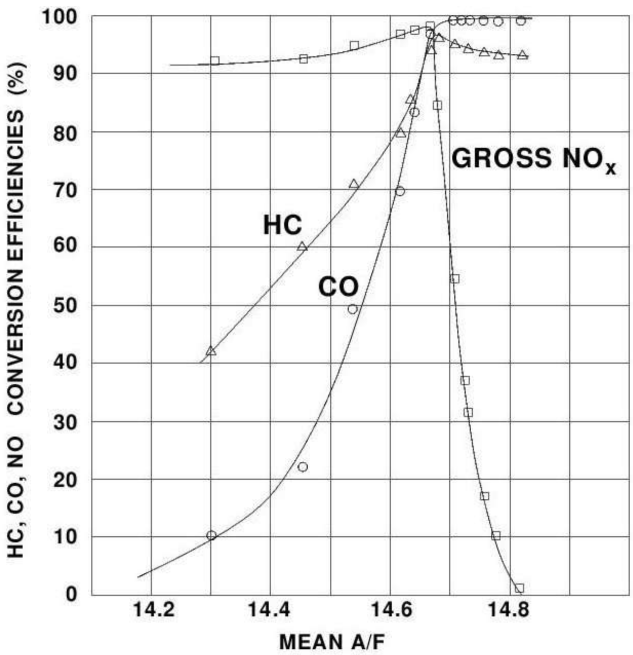

Figure 2.3: Typical three-way catalytic converter efficiency curves [10]

Non-stoichiometric operation not only leads to a failure in the emission compliance tests, but also causes damage to the catalytic converter [4]. Over the years, feedforward and feedback schemes have been devised as means of precise air-fuel ratio control. Following facts make the air-fuel ratio control problem a challenging one.

- Feedforward schemes depend on precise estimates of the intake air flow. These estimates are made by suitable processing of measurements from appropriate sensors. The controller then issues commands to the fuel injector so that proportionate quantity of fuel is injected.[2] However, the estimates for

2 While actuating the fuel injector, the controller must also consider dynamic effects in the fuel delivery path.

the air flow that are usually developed, are often valid only for steady state operation [5].[3] Reducing air-fuel ratio excursions during transient engine operation thus becomes a challenging task.

- Feedback schemes rely on measurement of air-fuel ratio made by an oxygen sensor placed in the exhaust manifold (figure 2.2). Measurement process is based on diffusion and includes delay of the order of tens of milliseconds. Another source of delay is the physical location of the sensor itself. The delay time equals the time taken by the air fuel mixture at intake to reach the sensor at exhaust, which is slightly more than the time taken by a complete engine cycle. At idling condition this delay becomes large. This, coupled with low powertrain inertia at idling, could prove to be a major problem to contend with. Another issue comes up due to the cost of the sensor. A "continuous" type air-fuel ratio sensor adds considerably to the total cost of the emission control system. "Switching" type sensors that convey whether the mixture is lean or rich of stoichiometric, are in use due to their low cost. Designing a controller that uses switching type feedback information is a challenging problem.

Often, a combination of both schemes is used. The feedforward scheme helps mainly to increase the speed of response. The feedback scheme acts slow in comparison, but helps to nullify the effect of disturbances.

2. Ignition Control: For other engine inputs held constant, the brake torque produced varies with spark advance or spark timing. At a particular value of spark advance, the brake torque maximizes; this value is called as "Maximum Brake-torque Tim-

3 More accurate estimates would require complex, detailed thermodynamic simulations. Such simulations take considerable time and also require a computer with lot of memory. Therefore, such estimates are unsuitable for real time control purpose.

ing" (MBT). The MBT value varies with engine speed and load. At MBT, fuel efficiency also peaks because same amounts of air and fuel produce maximum work. However, it may not be possible to operate the engine always at the MBT setting. This is because, the tendencies of knocking and NO_x production also peak at MBT (due to higher peak pressures and temperatures in the engine cylinder) [4]. This makes ignition control a trade-off between fuel efficiency and knock prevention. As a solution, a combination of feedforward and feedback schemes is often used.

The usual practice is to carry out extensive laboratory testing on the engine prototype in order to find out values of spark advance (as function of engine speed and load) that produce best torque while avoiding knocking and excessive NO_x production. The spark advance values, called "nominal spark timings", are stored in the engine control unit. A feedforward scheme is then devised to produce nominal spark advance by sensing engine speed and load (usually inferred from intake manifold pressure). To ensure that the engine does not knock in actual working, a feedback scheme is deployed. It involves placing a knock sensor (typically a vibration sensor) and causing an offset (retardation) in spark advance in case the engine knocks [4].

3. Torque Control: With air-fuel ratio and spark timing values always dictated by the considerations of emissions and fuel efficiency, air flow rate is the only input left to control. The controller must choose the air flow rate such that the driver's torque demand is met. Recognizing what the driver wants the engine to do, is crucial especially in case of drive-by-wire or electronically throttled engines. While actuating the throttle, it is important to maintain the driving feel for the driver to be analogous to what he/she experiences with other engines (that do not possess drive-by-wire

technology).

In order to obtain a quick torque response, sometimes a multivariable strategy is used that involves controlling the spark advance and the throttle [4]. When the driver's torque demand increases, the spark timing is advanced to that corresponding to MBT value from the nominal value. This causes immediate increase in the torque produced.[4] The throttle is also actuated so that air flow rate increases. As the air flow rate nears the desired value, spark timing is gradually retarded to its nominal value.

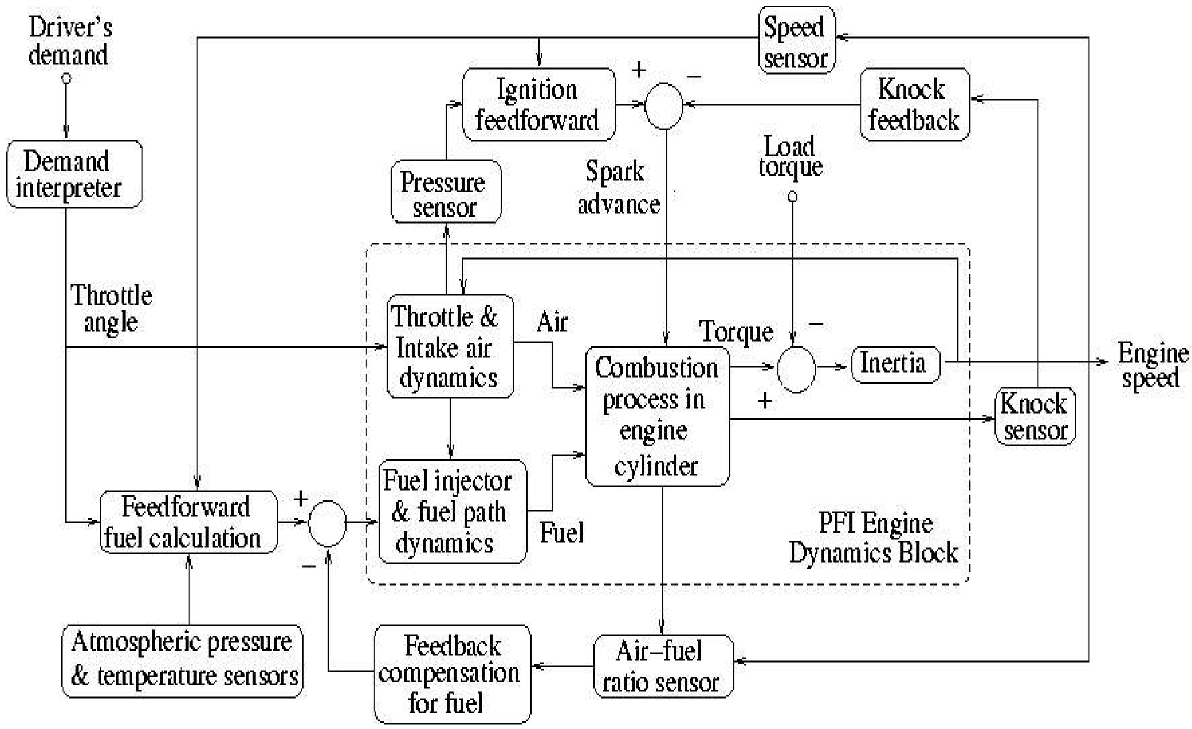

Figure 2.4: Control substructure for a drive-by-wire port fuel injection engine [4]

Various control loops discussed above are shown in figure 2.4. Out of various blocks

4 Spark timing is considered to be the fastest actuator of all engine actuators. In comparison, throttle-to-torque-production exhibits significant delay due to manifold air dynamics and suction-to-expansion stroke delay. Fuel-injection-to-torque-production also exhibits a considerable delay.

that are shown, the later part of the dissertation focuses on: the PFI engine dynamics block (engine model), the feedforward fuel calculation block, the feedback fuel compensation block, and throttle angle control for idling mode.

2.4 Control Oriented Modeling of Engines

Reciprocating engines differ from continuously operating thermal engines in two important aspects: they exhibit highly transient combustion with large & rapid temperature and pressure variations and the thermodynamic boundary conditions that govern the combustion process in them (e.g. intake pressure, composition of air fuel mixture) are not constant. Combustion phenomena in reciprocating engines involve rapid reaction kinetics of the order of few milliseconds. The models used to describe these phenomena in detail are complex, computationally expensive, and are not much useful for design of real-time control systems. The models that have been used traditionally for control purposes are less detailed in comparison. These control-oriented models describe the evolution of thermodynamic boundary conditions as dynamic phenomena and assume the actual combustion process to be static. In other words, they assume that for same boundary conditions imposed on combustion, the combustion will evolve in identical fashion each time. Such models do not necessarily reflect all phenomena in an engine (e.g. random cylinder pressure variations). Control-oriented models for reciprocating engines can be classified into the following two categories:

1. Discrete Event Based Models: This class of models takes into account reciprocating, discrete behavior of engines. Phenomena that occur during each stroke are modeled separately. Usually they consider crank angle as the independent variable.

That is, they describe evolution of all other variables with respect to crank angle. This dissertation does not focus on this class of models.

2. Mean Value Models: This class of models assumes the engine phenomena to be continuous (as in a gas turbine for example). They only consider the mean values of relevant properties (such as torque, fuel flow rate) over a cycle that actually vary considerably during the cycle. To capture the reciprocating behavior, these models include delays in several cause and effect relationships. Most of the mean value models are "lumped parameter" in nature in the sense they do not consider spatial distribution of variables. Therefore, they have only one independent variable (time) and are represented by ordinary differential equations. In the next chapter, this dissertation presents a mean value model for typical PFI engine subsystems.

2.5 Efficiencies and Mean Effective Pressures for Reciprocating Engines

The mechanical energy obtained at the engine crankshaft is much less than the chemical energy of the fuel combusted. This discrepancy may be attributed to thermodynamic losses, heat transfer and the mechanical energy losses in an engine. Following efficiencies are defined for an engine.

1. **Indicated thermal efficiency (η_{ind}):** It is the ratio of work done on the piston to chemical energy of the fuel. It considers the thermodynamic and heat transfer losses.

2. **Mechanical efficiency (η_{mech}):** It defines how much efficient the mechanical power

transmission path is. It considers the mechanical energy losses alone.

3. **Brake thermal efficiency ($\eta_{brake\text{-}thermal}$):** It defines how much efficient the net chemical-to-mechanical energy conversion is. It considers all kinds of energy losses mentioned above.

Consider a reciprocating engine modeled in mean value fashion. In other words, the engine is assumed to be a continuous (in time) system that consumes fuel continuously and produces torque continuously. For a mean value engine formulation, the energy balance per unit time may be written as

$$T_e . \omega = \dot{m}_f . H_l . \eta_{brake-thermal}$$

where T_e, ω, $\dot{m}_f$, H_l, and $\eta_{brake-thermal}$ denote engine torque, engine speed, fuel mass flow rate, calorific value of fuel, and brake thermal efficiency respectively.

For reciprocating internal combustion engines, it is sometimes advantageous to use the notion of mean effective pressure (MEP) instead of usual notions of energy or power[5]. "Balance" of MEP can be established for the engine instead of energy or power balance. Following MEPs are defined for later use in this dissertation.

1. **Brake Mean Effective Pressure (*BMEP*):** It is a constant hypothetical pressure that is assumed to act on the piston during the expansion stroke such that it produces same amount of work that the real engine does in two crank revolutions.

   That is,

5 Concept of "mean effective pressure" can be thought of to be analogous to the concept of "pressure head" in case of hydraulic turbines.

$$BMEP = \frac{Constant\ hypothetical\ force\ acting\ on\ piston}{Piston\ area}$$

$$= \frac{\left(\frac{Work\ produced\ during\ one\ cycle}{Piston\ displacement\ during\ expansion\ stroke}\right)}{Piston\ area}$$

$$= \frac{(Brake\ Torque)\times(Angular\ duration\ of\ one\ cycle)}{Stroke\ volume}$$

$$= \frac{T_e.(4\pi)}{V_d} \tag{2.1}$$

2. **Fuel Mean Effective Pressure (*MEP*$_{fuel}$):** It is the *BMEP* that an engine with brake thermal efficiency of unity would produce. That is,

$$MEP_{fuel} = \frac{\begin{pmatrix} Constant\ hypothetical\ force\ acting\ on \\ piston\ if\ the\ engine\ were\ 100\%\ efficient \end{pmatrix}}{Piston\ area}$$

$$= \frac{\left(\frac{Work\ produced\ with\ 100\%\ efficiency\ during\ one\ cycle}{Piston\ displacement\ during\ expansion\ stroke}\right)}{Piston\ area}$$

$$= \frac{\begin{pmatrix} Mass\ of\ fuel\ inducted \\ during\ one\ cycle \end{pmatrix}\times\begin{pmatrix} Calorific\ value \\ of\ fuel \end{pmatrix}}{Stroke\ volume}$$

$$= \frac{M_f.H_l}{V_d} \tag{2.2}$$

3. **Indicated mean effective pressure (*IMEP*):** It is the *BMEP* that an engine that does not incur any mechanical energy loss would produce.

The *BMEP* and the *IMEP* for a real engine are always less than the *MEP*$_{fuel}$. The difference between the $BMEP$ and the MEP_{fuel} corresponds to the thermodynamic and mechanical energy losses that occur in the engine. Whereas the difference between the $IMEP$ and the MEP_{fuel} corresponds to the thermodynamic energy losses alone. Following set of equations establishes the balance of MEPs:

$$IMEP = MEP_{fuel} - MEP_{thermodynamic\,losses}$$

$$BMEP = MEP_{fuel} - MEP_{thermodynamic+mechanical\,losses}$$

Also, the efficiencies defined earlier in case of an engine may be redefined in the context of MEPs as follows:

$$\eta_{ind} = \frac{IMEP}{MEP_{fuel}}$$

$$\eta_{mech} = \frac{BMEP}{IMEP}$$

$$\eta_{brake\,thermal} = \frac{BMEP}{MEP}$$

Chapter 3

Mean Value Model of a Port Fuel Injection Engine

In this chapter, a dynamic, lumped parameter, mean value model of a PFI engine is developed. Modeling an engine in control oriented sense is an important preliminary step towards engine controller development. The model presented in this chapter captures most of the relevant cause and effect relationships in a port fuel injection engine. Sections 3.1 and 3.2 present models that describe how the intake air and fuel are transported into the engine cylinder. Section 3.3 presents a static model for torque production as an effect of the engine inputs-the masses of intake air, fuel and spark timing.

3.1 Intake Air Path Dynamics

In this section, a mean value dynamic model for intake air path in PFI engines is developed. The goal is to derive mathematical equations that describe evolution of air flow entering the cylinder, given the throttle angle, and the engine speed. Figure 3.1 shows schematic of the intake air path. Important subsystems to be modeled are throttle body, intake manifold, and cylinder air induction.

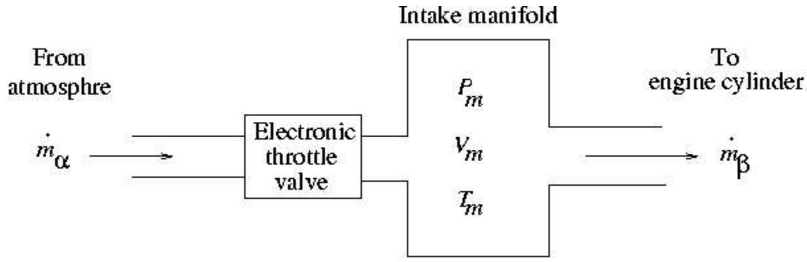

Figure 3.1: Schematic of intake air path

3.1.1 Dynamic Model for Intake Manifold

The intake manifold is modeled using the mass conservation equation. The intake manifold exhibits filling-emptying phenomena of the intake air mass. The throttle admits air from one end, while the engine cylinder draws air out from the other. Since the volume of the intake manifold is fixed, the filling-emptying process causes the manifold air pressure to vary. Let the filling and emptying rates of air mass be represented by $\dot{m}_\alpha(t)$ and $\dot{m}_\beta(t)$. Let the gas constant for air, the manifold temperature and the manifold volume be represented by R, T_m, and V_m respectively. Then the rate of change of manifold pressure, $dP_m(t)/dt$ may be expressed as follows:

$$\frac{d}{dt}P_m(t)=\frac{R.T_m}{V_{m.}}[\dot{m}_\alpha(t)-\dot{m}_\beta(t)] \quad (3.1)$$

Following assumptions are made while writing the above expression:

- air is a perfect gas;

- manifold conditions are homogeneous;
- there is no leakage flow entering/leaving the manifold; and
- the intake manifold air temperature is fixed.

The intake manifold control block is shown in figure 3.2.

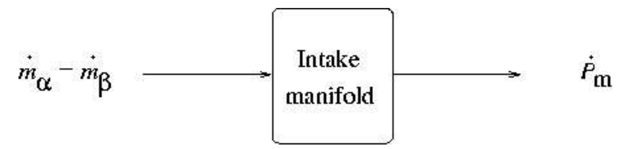

Figure 3.2: Intake manifold control block

3.1.2 Static Model for Throttle Body

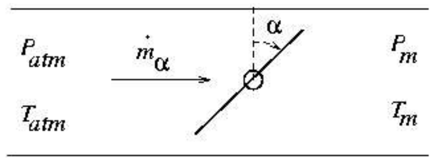

Figure 3.3: Air flow past the throttle plate

Air mass flow rate past the throttle section, $\dot{m}_\alpha(t)$, is a function of following two variables: area available for the flow, and pressure ratio across the throttle section. Area available for the flow equals the cross-sectional area of the channel less the area blocked by the throttle plate. The blocked area depends on the throttle angle α, measured from fully closed position as shown in figure 3.3. Let the channel and the throttle plate be circular in shape with diameter d_{th}. Then the available flow area can be expressed as:

$$Area(t)=\frac{\pi}{4}.d_{th}^{2}.[1-\cos(\alpha(t))] \tag{3.2}$$

Let the atmospheric pressure and temperature be represented by P_{atm}, T_{atm} respectively. Let C_d be the coefficient of discharge for the flow. Let γ denote the ratio of specific heats of air. Considering the air flow to be compressible, the following expression for the air mass flow rate $\dot{m}_{\alpha}(t)$ may be written using thermodynamic relationships for isentropic expansion

$$\dot{m}_{\alpha}(t)=C_d.Area(t).\frac{P_{atm}}{\sqrt{R.T_{atm}}}.\Psi\left(\frac{P_{atm}(t)}{P_m(t)}\right) \tag{3.3}$$

where the flow function $\Psi(.)$ is given by

$$\Psi\left(\frac{P_{atm}(t)}{P_m(t)}\right) = \begin{cases} \sqrt{\gamma\left[\dfrac{2}{\gamma+1}\right]^{\frac{\gamma+1}{\gamma-1}}} & \text{for } P_m(t)<P_{cr}(t) \\ \left(\dfrac{P_m(t)}{P_{atm}(t)}\right)^{\frac{1}{\gamma}}.\sqrt{\dfrac{2.\gamma}{\gamma-1}.\left[1-\left(\dfrac{P_m(t)}{P_{atm}(t)}\right)^{\frac{\gamma-1}{\gamma}}\right]} & \text{for } P_m(t)\geqslant P_{cr}(t) \end{cases} \tag{3.3a}$$

and where

$$P_{cr}(t)=\left[\frac{2}{\gamma+1}\right]^{\frac{\gamma}{\gamma-1}}.P_{atm}(t) \tag{3.3b}$$

is the critical pressure where the flow reaches sonic conditions in the narrowest part. The following assumptions are made:

- all flow phenomena are zero dimensional i.e. no spatial effects need be considered;
- completely isolated conditions;
- no inertial effects in the flow (steady state assumption, the throttle model therefore becomes a “static” model);
- no losses in the accelerating part up to the narrowest point (all the potential energy stored in the fluid gets converted isentropically into kinetic energy); and

- fully turbulent flow after the narrowest point with no pressure recuperation (all of the kinetic energy gained in the first part is dissipated into thermal energy).

The consequences of this are that the pressure at the narrowest point of valve is (approximately) equal to the downstream pressure, $P_m(t)$, and that the temperature of the flow before and after the orifice is (approximately) the same i.e. $T_{atm}(t) = T_m(t)$. The throttle body control block is shown in figure 3.4.

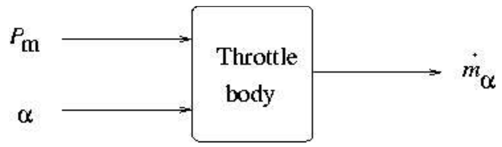

Figure 3.4: Throttle body control block

3.1.3 Static Model for Cylinder Air Induction

Regarding modeling air induction into the cylinder, the engine can be considered to be a volumetric pump, i.e. a device that enforces a volume flow approximately proportional to its speed. In this case, a typical formulation for mass flow rate for is as follows:

$$\begin{aligned}\dot{m}_\beta(t) &= \rho_m(t).\dot{V}(t)\\ &= \rho_m(t).\eta_{vol}(P_m(t),\omega(t)).V_d.\frac{\omega(t)}{2\pi}\end{aligned} \tag{3.4}$$

where the new variables defined are as follows:

$\rho_m(t)$ density of intake manifold air;

$\omega(t)$ engine speed in rad/s;

$\eta_{vol}(t)$ volumetric efficiency of the engine, a function of $P_m(t)$ and $\omega(t)$; and

$V_d(t)$ stroke volume of the engine cylinder.

The following assumptions are made:

- volumetric efficiency is a static function of manifold pressure and engine speed; and
- other engine variables do not affect the volumetric efficiency.

Cylinder air induction control block is shown in figure 3.5.

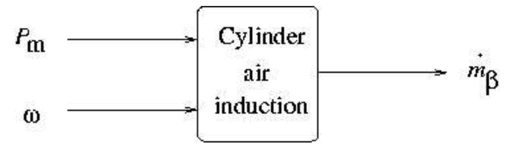

Figure 3.5: Cylinder air induction block

Sometimes it is useful to use the following multilinear formulation for the volumetric efficiency term for the purpose of identification [4]:

$$\eta_{vol}(P_m(t),\omega(t))=\eta_\omega(\omega(t)).\eta_{P_m}(P_m(t)) \tag{3.4a}$$

In the above equation, the effects of the engine speed and the intake manifold pressure on the volumetric efficiency are modeled separately (in an “individualistic manner”). The usual forms of $\eta_\omega(\omega(t))$ and $\eta_{P_m}(P_m(t))$ that are used are

$$\begin{gathered}\eta_\omega(\omega(t))=\gamma_0+\gamma_1.\omega(t)+\gamma_2.\omega^2(t)\\ \eta_{P_m}(P_m(t))=\frac{V_c+V_d}{V_d}-\frac{V_c}{V_d}.\left(\frac{P_{ex}}{P_m(t)}\right)^{1/\gamma}\end{gathered} \tag{3.4b}$$

where the new defined variables are as follows:

V_c clearance volume; and

P_{ex} exhaust gas pressure.

Finally, the mass flow rate entering the cylinder can be expressed as a static function of the type :

$$\dot{m}_\beta(t)=\dot{m}_\beta(P_m(t),\omega(t)) \tag{3.4}$$

Figure 3.6 shows implementation of complete air path in MATLAB/SIMULINK environment.

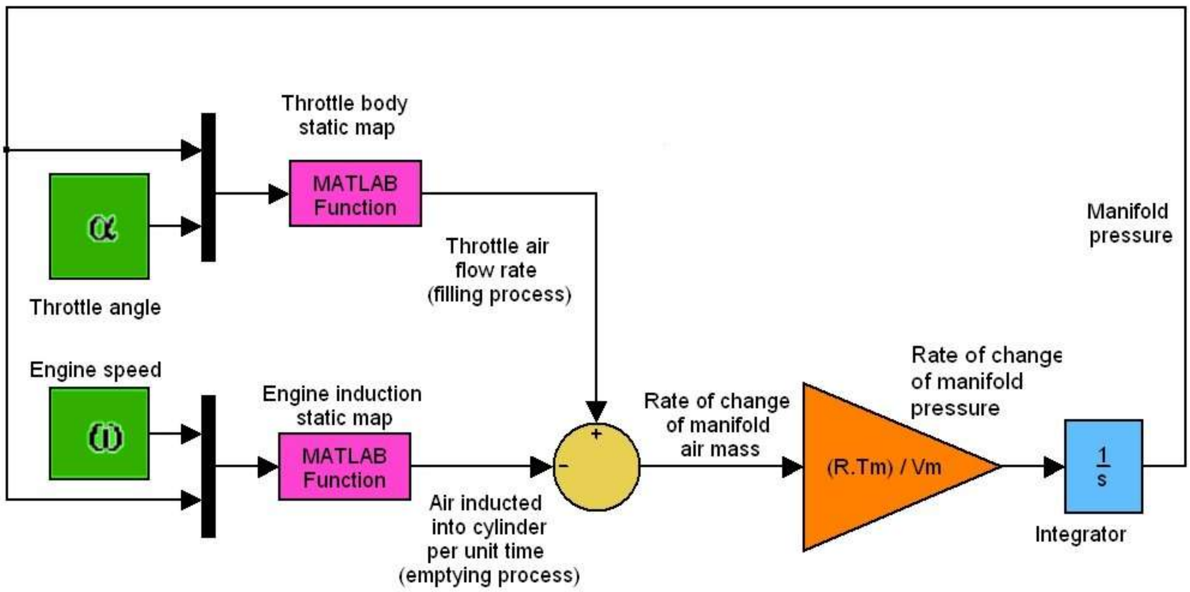

Figure 3.6: Intake air path model in MATLAB/SIMULINK environment

3.2 Fuel Path Dynamics

In this section, a mean value dynamic model for fuel delivery in PFI engines is developed. The goal is to derive mathematical equations that describe evolution of fuel flow entering the cylinder, given fuel flow coming out of the fuel injector. In PFI engines, the fuel injector is placed in the intake manifold just upstream of the cylinder intake valves. The fuel injector injects fuel in pulses when the intake valve is not open.[6] The pulse width is determined by how much fuel the controller wants to inject, which is result of the feedforward and feedback calculations mentioned in the last chapter. For the purpose of developing a mean

6 This gives enough time for the fuel to evaporate and mix thoroughly with the air before entering the cylinder during the next suction stroke. Fuel injection, if made while the intake valves are open, causes high unburnt HC emissions due to improper air-fuel mixing [4].

value model, we neglect the pulsating nature of fuel flow.

Let $\dot{m}_{fi}$ be the mass flow rate of fuel delivered by the fuel injector. A part of the fuel injected by the injector hits the manifold wall and forms a thin film on it [1]. Let X denote the fraction of the fuel that impinges on the wall, and m_{ff} denote the mass of the manifold fuel film. Fuel from the manifold fuel film continuously evaporates. Therefore, the net fuel flow entering the cylinder, $\dot{m}_f$, constitutes of the non-impinging part (1-X) of the injected fuel flow and the fuel flow evaporating from the wall (m_{ffout}). The evaporation rate (m_{ffout}) is assumed to be directly proportional to the mass of the fuel film (m_{ff}). Let τ_{ff} denote time constant for the first order evaporation model. Then the proportionality constant for the evaporation process becomes $1/\tau_{ff}$.

Parameters X and τ_{ff} of the model vary with operating condition. For a given engine, the fraction X usually assumes a fixed value once the engine warms up [1]. The evaporative time constant τ_{ff} is mainly influenced by the intake manifold temperature amongst other engine variables [1]. Value of τ_{ff} drops as the intake manifold temperature rises. The following set of equations expresses the dynamics of fuel path around an operating point (i.e. for fixed values of X and τ_{ff}).

$$\begin{aligned} \dot{m}_f(t) &= \dot{m}_{fi}(t).[1-X] + \frac{m_{ff}(t)}{\tau_{ff}} \\ \dot{m}_{ff}(t) &= \dot{m}_{fi}(t).\,X - \frac{m_{ff}(t)}{\tau_{ff}} \end{aligned} \tag{3.5}$$

Figure 3.7 shows MATLAB/SIMULINK implementation of the fuel path dynamics.

3.3 Torque Production

In this section, a mean value static model for torque production is presented. The model is proposed by [4] and requires less calibration effort in comparison with other static models for torque production.

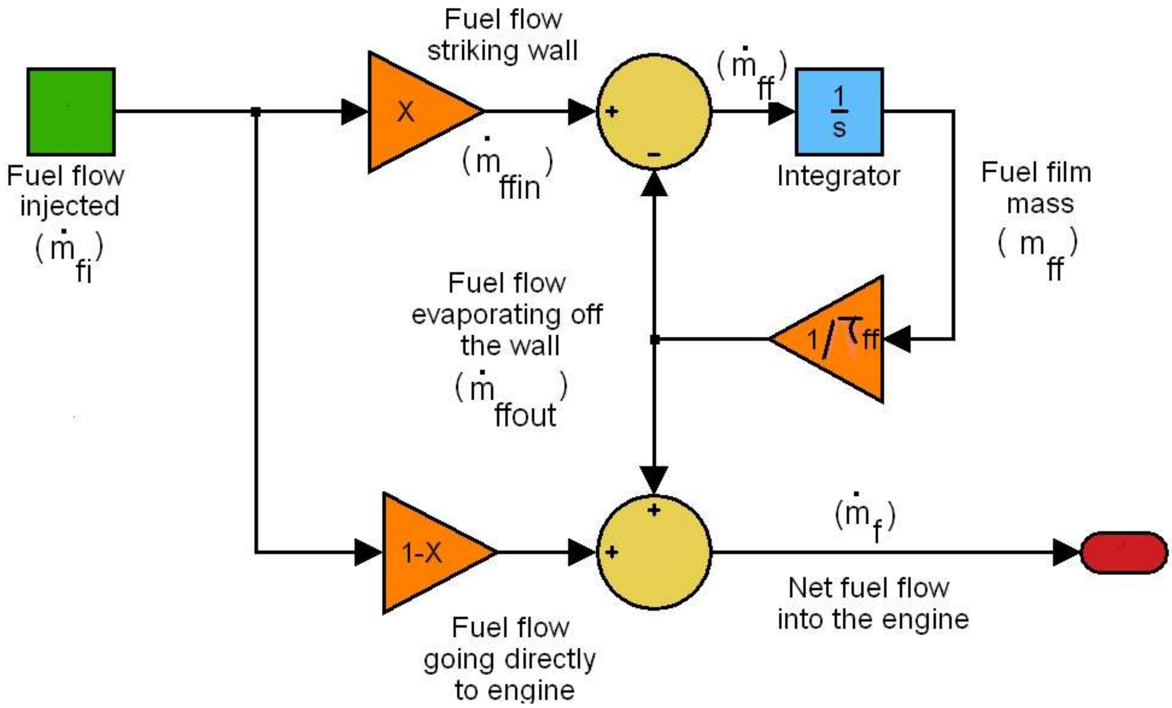

Figure 3.7: Fuel path dynamics model in MATLAB/SIMULINK environment

Work obtained at the engine crankshaft during the expansion stroke is mainly affected by the following variables amongst others: mass of air (M_a) and the air-fuel ratio (λ) for the charge mixture taken in during earlier suction stroke, mass & composition of the residual gas at the end of earlier exhaust stroke, spark advance (σ) during the earlier compression stroke, and the engine speed (ω). Engine speed affects the work obtained in several ways. At higher engine speeds, heat transfer loss during each expansion stroke is less duel to reduced available time. This contributes to rise in indicated efficiency. However, at very high engine speeds the time available for chemical reaction kinetics during the expansion stroke reduces and may not be sufficient. This contributes to fall in indicated efficiency. Also, the frictional losses in an engine are (approximately) proportional to square of the engine speed. This causes mechanical and brake thermal efficiencies to fall at high engine speeds.

Precise estimation of torque will require detailed, three dimensional thermodynamic simulation of the combustion process. For the purpose of control-oriented modeling, the engine torque is usually formulated as a static function of the influencing variables. In other words, once all the (assumed) influencing variables are fixed at the start of the combustion process, it is assumed that the combustion process will evolve in identical fashion every time.

Various approaches have been proposed to obtain the static torque function. One popular approach is to map torque for the whole grid of influencing variables. This requires extensive lab experimentation on the engine, which may not be affordable. Other approaches use some physical insight to model influence of some of the input variables, and reduce the experimentation required. Such torque models are of great interest. One such approach that uses the notion of mean effective pressures is described below.

It can said that the BMEP depends on the same variables that influence the engine torque. If an expression for *BMEP* could be developed then the torque function will just be an extension of it. The *BMEP* may be expressed as follows:

$$\begin{aligned} BMEP(t) &= IMEP(t).\,\eta_{mech}(t) \\ &= IMEP(t) - MEP_{mechanical\,losses}(t) \\ &= \eta_{ind}(t).\,MEP_{fuel}(t) - MEP_{mechanical\,losses}(t) \end{aligned} \tag{3.6}$$

Using "Willans Approximation" the indicated efficiency, η_{ind}, can be expressed as a static function of the following variables: engine speed (ω), air fuel ratio (λ), and spark advance (σ) [4]. That is,

$$\eta_{ind}(t)=\varphi(\omega(t),\lambda(t),\sigma(t)) \tag{3.7}$$

If the air-fuel ratio is regulated tightly around stoichiometric, and the spark timing is maintained around the MBT value, the structure of function $\varphi(.)$ is usually of the following form [4]:

$$\varphi(\omega(t))=\eta_0+\eta_1.\omega(t) \tag{3.7a}$$

Using (2.2) the MEP_{fuel} can be expressed as:

$$MEP_{fuel}(t)=\frac{\dot{m}_\beta(P_m(t),\omega(t))}{\lambda(t).\omega(t)}.\frac{H_l.4\pi}{V_d} \tag{3.8}$$

[4] proposes a friction model for an SI engine. The $MEP_{mechanical\,losses}$ is expressed as a static function of engine speed, and the intake manifold pressure.

$$MEP_{mechanical\,losses}=\psi(\omega(t),P_m(t)) \tag{3.9}$$

$MEP_{mechanical\ losses}$ mainly constitutes of frictional losses that are represented by "frictional mean effective pressure" (FMEP). An additional an MEP term for intake-exit gas exchange losses may also be included in $MEP_{mechanical\ losses}$. It mainly consists of the pumping losses during suction. Thus, $\psi(.)$ is of the form:

$$\psi(\omega(t),P_m(t))=\beta_0+\beta_2.(\omega^2(t)).\frac{4\pi}{V_d}+(P_{atm}-P_m(t)) \tag{3.9a}$$

Experiments must be performed on a given engine prototype in order to calibrate the static functions $\varphi(.)$ and ψ (.) in (3.7) and (3.9) respectively. Substituting (3.7), (3.8) and (3.9) into (3.6) gives following static formulation for the *BMEP*:

$$BMEP(t)=\frac{\dot{m}_\beta(P_m(t),\omega(t)).\varphi(\omega(t))}{\lambda(t).\omega(t)}.\left[\frac{H_l.4\pi}{V_d}\right]-\psi(\omega(t),P_m(t)) \tag{3.10}$$

Putting (2.1) into (3.10) gives

$$T_e(t) = \frac{\dot{m}_\beta(P_m(t), \omega(t)) . \varphi(\omega(t))}{\lambda(t) . \omega(t)} . H_l - \psi(\omega(t), P_m(t)) . \frac{V_d}{4\pi} \tag{3.11}$$

In case of an actual engine, the torque developed during expansion stroke depends on masses of air and fuel taken in the earlier section stroke and spark timing in earlier compression stroke. To capture this behavior in the mean value model, we include following two delays:

1. **Suction to power delay (*$\delta_{suct\text{-}pow}$*):** This approximates the time delay between suction phenomenon and torque production. It is approximately equal to the time taken by the piston for traveling through two strokes (i.e. through 2π radians for a four stroke engine).

$$\delta_{suct-pow}(t) = \frac{2\pi}{\omega(t)} \tag{3.12}$$

2. **Sparking to expansion delay (*$\delta_{spk\text{-}pow}$*):** This approximates the time delay from sparking to torque production. Maximum value of this delay equals the time taken by the piston to travel through one stroke (i.e. π radians for a four stroke engine).

$$\delta_{spk-pow}(t) = \frac{\pi}{\omega(t)} \tag{3.13}$$

Incorporating the above two delays into (3.11) gives

$$\begin{aligned} T_e(t) = \dot{m}_\beta\left(t - \frac{2\pi}{\omega(t)}\right) . \varphi(\omega(t)) . \frac{H_l}{\omega(t) . \lambda(t)} \\ - \psi(P_m(t), \omega(t)) . \frac{V_d}{4\pi} \end{aligned} \tag{3.14}$$

3.4 Pollutant Formation

Precise prediction of emissions would require detailed, complex simulation of combustion process. Instead, the usual practice is to record emission generation rates for all three of HC, CO and NO_x obtained in steady state tests for a given engine prototype [11]. The test inputs (air flow rate, air-fuel ratio, and spark timing) cover their operating ranges completely. Regression of the multidimensional data obtained is useful for model based optimization. However [4] reports that the emission generation rates during transient operation are much different than those predicted by the above technique. Moreover, ageing and other, not easily modeled effects, cause large deviations in pollutant formation such that, in general, prediction errors are quite large. Control-oriented engine out emission models are therefore often used as a means to predict the relative impact of a specific controller structure or algorithm on pollutant emission. In this dissertation, we do not take a look into regression based technique for pollutant formation.

3.5 Engine Speed Dynamics

The crank shaft is assumed to be acted upon by two torques viz. the engine torque and the load torque. The difference between the two causes the crankshaft to accelerate or decelerate as per Newton's Second Law. Let T_L and J represent the load torque and rotational inertia of the engine. Following equation expresses the dynamics of the crankshaft:

$$\frac{d\omega}{dt}=\frac{(T_e-T_L)}{J} \tag{3.12}$$

Load torque on the engine is not known a priori. It is due to aerodynamic resistance, road gradient load, and rolling load on the tyres that the vehicle has to overcome.

Chapter 4

Linearization and Simulation Results

In this chapter, we take a look at linearized form of the mean value model developed in earlier chapter. Simulation results for the nonlinear and linearized engine models have been presented.

4.1 Nonlinear and Linearized Models for a Candidate PFI Engine at Idling Condition

In this section, the mean value model developed in the last chapter is adapted for the idling mode. As mentioned before, the main objective during the idling mode is to maintain a constant engine speed irrespective of engine loading. It is assumed that the air-fuel ratio is held at stoichiometric value by a separate controller. It is also assumed that the spark advance is effected by a separate controller (figure 2.4). Therefore, for the purpose of controlling idling speed, throttle angle (α) is the only control input that influences engine torque production, and thus engine speed. The model has two state variables manifold pressure (P_m) and engine speed (ω).

4.1.1 Expression for Rate of Change of Intake Manifold Pressure

We begin with intake air manifold dynamics with the goal of obtaining an equation for rate of change of manifold pressure ($\dot{P}_m(t)$) using (3.1).

Equation (3.2) for the throttle area is modified to the following form:

$$Area(t)=\frac{\pi}{4}.d_{th}^2.\left[1-\frac{\cos(\alpha(t))}{\cos(\alpha_0)}\right]+A_{leak} \tag{4.1}$$

where α_0 is the throttle angle when the throttle is fully closed and A_{leak} is the flow area when the throttle angle is α_0.

In idling condition, the manifold pressure (P_m) is always less than the critical pressure limit (P_{cr}) defined in (3.3b). In this case, (3.3) for mass flow rate across the throttle section is approximated by the following simpler equation.

$$\dot{m}_\alpha(t)=Area(t).\frac{P_{atm}}{\sqrt{R.T_{atm}}}.\frac{1}{\sqrt{(2)}} \tag{4.2}$$

Substituting (4.1) into (4.2) gives the following static relationship between the mass flow rate and the throttle angle.

$$\begin{aligned}\dot{m}_\alpha(t)&=\frac{P_{atm}}{\sqrt{R.T_{atm}}}.\frac{1}{\sqrt{(2)}}.\left\{\frac{\pi}{4}.d_{th}^2.\left[1-\frac{\cos(\alpha(t))}{\cos(\alpha_0)}\right]+A_{leak}\right\}\\&=m_1(\alpha(t))\end{aligned} \tag{4.3}$$

From (3.4), (3.4a), and (3.4b), the mass flow rate of air entering the cylinder may be expressed as a static function as follows.

$$\dot{m}_{\beta}(t)=\frac{P_m(t)}{R.T_m}.\eta_{P_m}(P_m(t)).\eta_{\omega}(\omega(t)).V_d.\frac{\omega(t)}{2\pi}$$

$$=m_2(P_m(t),\omega(t)) \tag{4.4}$$

Thus, from (3.1), (4.3), and (4.4), we can express the rate of change of manifold pressure ($\dot{P}_m(t)$) as a nonlinear function of $\alpha(t)$, $P_m(t)$ and $\omega(t)$ as follows:

$$\dot{P}_m(t)=f_1(P_m(t),\omega(t),\alpha(t)) \tag{4.5}$$

4.1.2 Expression for Rate of Change of Engine Speed

Evolution of the engine speed happens as expressed by (3.12). It is assumed that the air-fuel ratio is held tightly at the stoichiometric value. The time delay from suction to power stroke is neglected. Then from (3.11) the engine torque can a expressed as a nonlinear static function function of engine speed (ω) and the intake manifold pressure (P_m). This in-turn allows rate of change of engine speed to be expressed in the following form:

$$\dot{\omega}(t)=f_2(P_m(t),\omega(t),T_L(t)) \tag{4.6}$$

4.1.3 State-space Representation of the Engine Model at Idling Condition

State matrix is chosen to have two elements viz. manifold pressure (P_m) and engine speed (ω). The input to the control system is throttle angle (α). Load torque (T_L) is the disturbance for the system. The nonlinear system expressed by (4.5) and (4.6) is linearized and expressed in the following state-space form:

$$\begin{bmatrix} \Delta \dot{P}_m \\ \Delta \dot{\omega} \end{bmatrix} = \begin{bmatrix} \dfrac{\partial f_1}{\partial P_m} & \dfrac{\partial f_2}{\partial P_m} \\ \dfrac{\partial f_1}{\partial \omega} & \dfrac{\partial f_2}{\partial \omega} \end{bmatrix} . \begin{bmatrix} \Delta P_m \\ \Delta \omega \end{bmatrix} + \begin{bmatrix} \dfrac{\partial f_1}{\partial \alpha} \\ 0 \end{bmatrix} . \Delta \alpha + \begin{bmatrix} 0 \\ \dfrac{\partial f_2}{\partial T_L} \end{bmatrix} . \Delta T_L \tag{4.7}$$

The coefficient matrices must be evaluated at a given operating point defined by combination of (P_m, ω) or (α, T_L).

4.1.4 Engine Parameters and Coefficient Matrices at Operating Point

For the purpose of simulations, engine specifications for a 2.8 liter port fuel injection engine are taken from [4]. These specifications and values of various other engine parameters for the engine at idling state are mentioned in Appendix A.

The desired idling speed is chosen as 100 rad/s. The initial load torque is taken as 5 Nm. Required throttle angle and the intake manifold pressure were found using (4.5) and (4.6). They came out to be 0.1598 radians and $(0.2566).10^5$ N/m^2 respectively. The coefficient matrices in (3.7) were evaluated at the operating point to give following state-space equation.

$$\begin{bmatrix} \Delta \dot{P}_m \\ \Delta \dot{\omega} \end{bmatrix} = \begin{bmatrix} -2.7848 & -713.3291 \\ 0.0105 & 1.4387 \end{bmatrix} . \begin{bmatrix} \Delta P_m \\ \Delta \omega \end{bmatrix} + \begin{bmatrix} 1.733\text{e+6} \\ 0 \end{bmatrix} . \Delta \alpha + \begin{bmatrix} 0 \\ -5 \end{bmatrix} . \Delta T_L \tag{4.8}$$

4.2 Simulation Results

This section presents simulation results for both nonlinear and linear models for idling condition. Responses have been simulated for step change in throttle angle as well as in load torque.

4.2.1 Step Change in Throttle Angle

Both nonlinear and linear models were given a step change in throttle angle from 0.1598 radians to 0.1605 radians at 10 seconds. We can see that responses of the linearized model resembles that of the nonlinear model (figure 4.1).

4.2.2 Step Change in Load Torque

Both nonlinear and linear models were given a step change in load torque from 5 N m to 7 N m radians at 10 seconds. We can see that responses of the linearized model resembles that of the nonlinear model (figure 4.2).

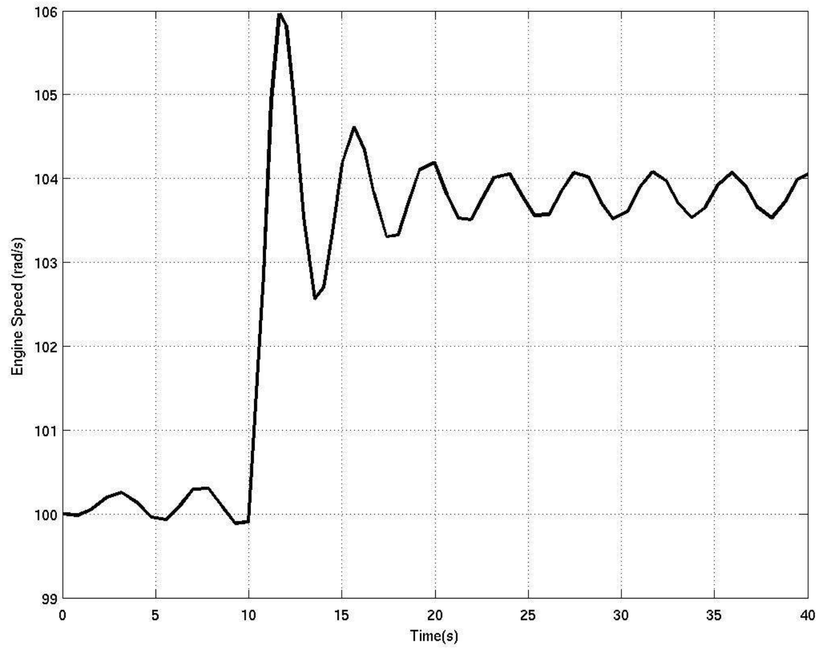

Figure 4.1 (a): Speed response of nonlinear model to a step in throttle angle

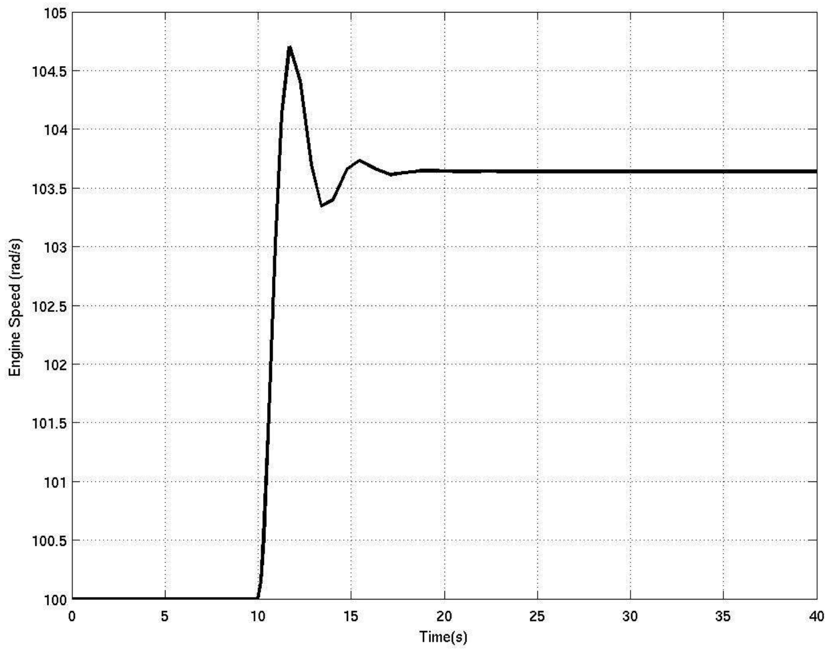

Figure 4.1 (b): Speed response of linearized model to a step in throttle angle

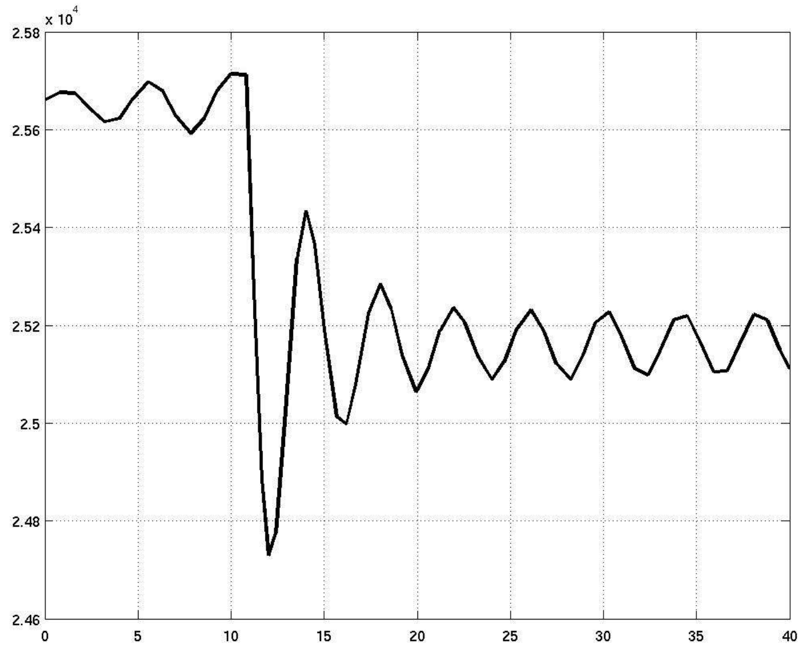

Figure 4.1 (c): Manifold pressure response of nonlinear model to a step in throttle angle

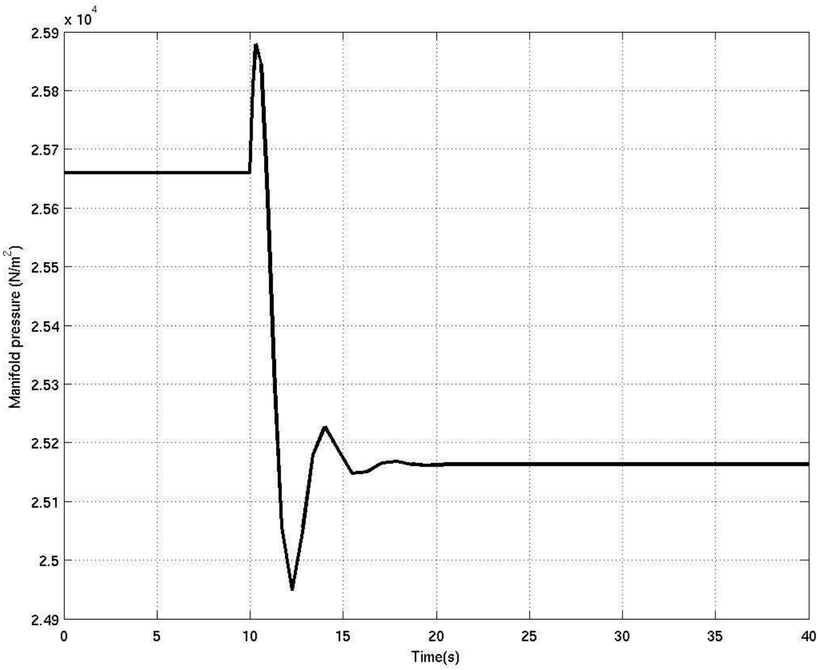

Figure 4.1 (d): Manifold pressure response of linearized model to a step in throttle angle

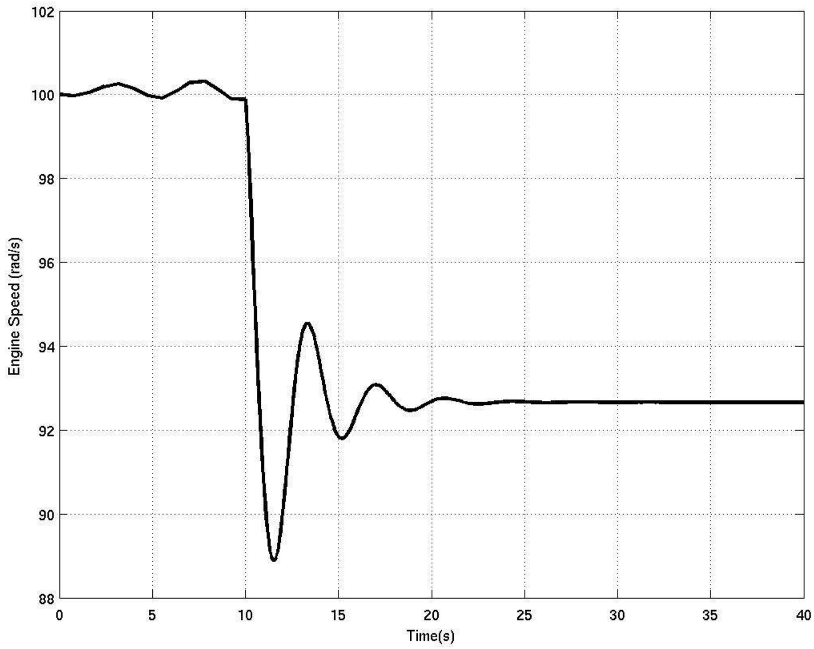

Figure 4.2 (a): Speed response of nonlinear model at idling condition to a step in load torque

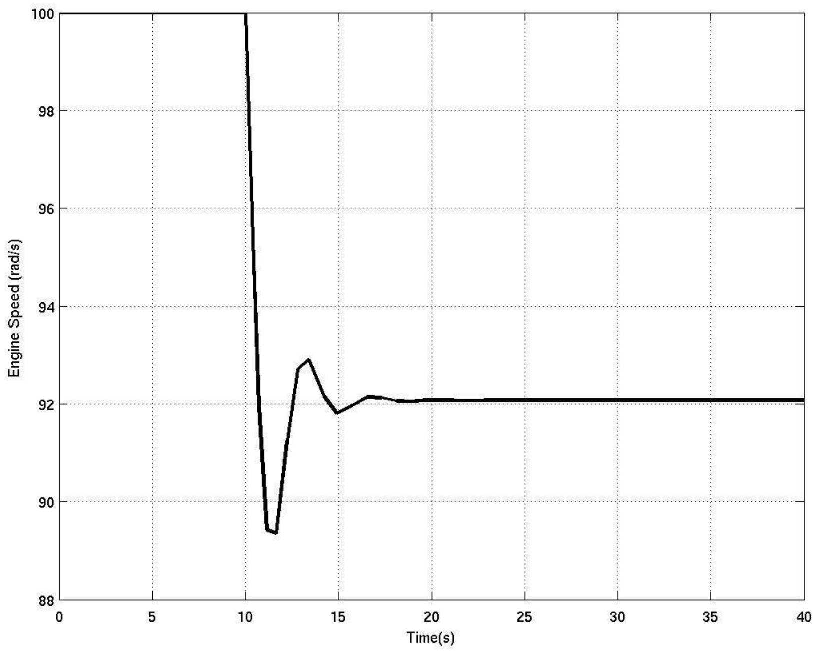

Figure 4.2 (b): Speed response of linearized model at idling condition to a step in load torque

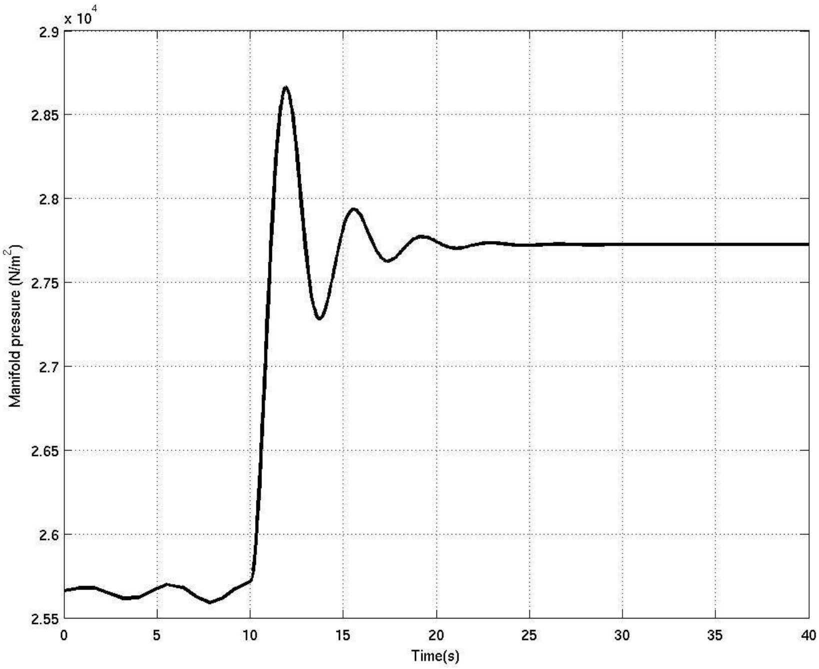

Figure 4.2 (c): Manifold pressure response of nonlinear model at idling condition to a step in load torque

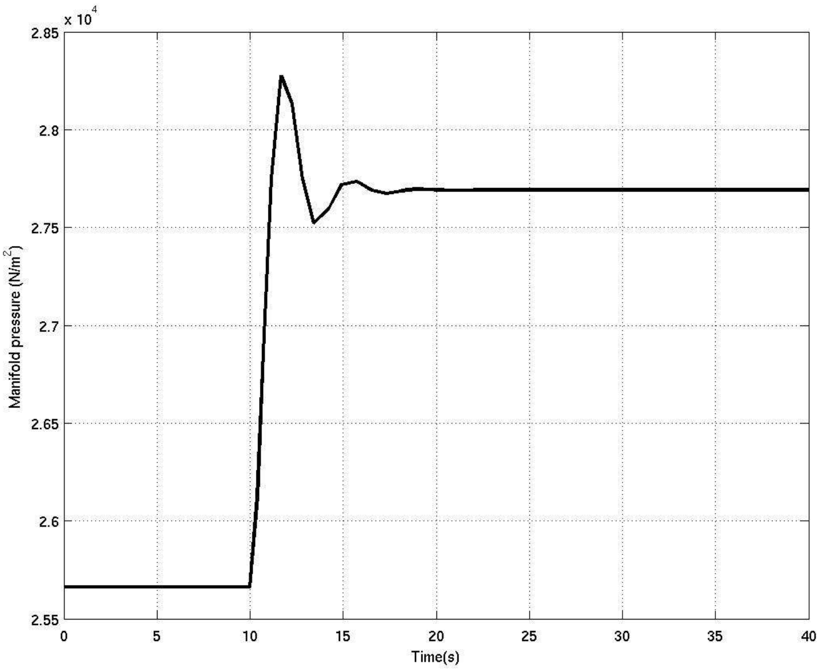

Figure 4.2 (d): Manifold pressure response of linearized model at idling condition to a step in load torque

Chapter Five

A PID Controller for Idle Speed Regulation

In this chapter, a PID controller is developed for the idle speed control problem. Simulation results for the performance of both linear and nonlinear models are presented. The idle speed controller controls only the throttle angle in order to maintain the desired idling speed in the event of unknown load disturbances. While developing this controller, it is assumed that the air-fuel ratio is tightly held at the stoichiometric value by a separate controller. It is also assumed that a separate controller is used for deciding the spark timing. Section 5.1 describes the development the PID controller. Section 5.2 presents simulation results for both linear and nonlinear models.

5.1 Development of Idle Speed PID Controller

For calculating suitable PID gains, the transfer function from throttle angle to engine speed is computed from the state-space formulation in (4.8). The transfer function evaluated is as follows:

$$TF_{\Delta\alpha \rightarrow \Delta\omega} = \frac{1.827e{+}4}{s^2 + 1.3461 . s + 3.5122} \tag{5.1}$$

The above transfer function represents a BIBO stable plant. To proceed with the computation of the PID gains, another transfer function was computed for a the same system, but this time with PD gains as K_p and K_d respectively. The transfer function evaluated in terms of K_p and K_d is of the following form:

$$TF_{\Delta\omega - ref \rightarrow \Delta\omega} = \frac{(1.827e{+}4) . K_d . s + (1.827e{+}4) . K_p}{s^2 + [1.3461 + (1.827e{+}4) . K_d] s + 3.5122 + (1.827e{+}4) . K_p} \tag{5.2}$$

A first approximation for the PD gains was obtained for above second order system such that it follows the following time domain specifications:

- the transfer function in (5.2) is always stable; and
- the maximum overshoot is small enough (target laid was: less than 3% for a step change in reference speed); and
- the settling time is small enough (target laid was: less than 3 s for a step change in reference speed).

In order to reduce the settling time, an integrator term (with gain *Ki*) was introduced in the controller. After few trials, the following values of the PID gains were finalized:

Gain	Value
K_p	0.0017
K_d	4.3633e-4
K_i	9.1629e-4

The gains are rather small in magnitude because of relatively large coefficients associated with them in (5.2).

5.2 Simulation Results for Idle Speed Control System

This section presents simulation results of the PID controller in case of both, nonlinear and linear models, for idling condition. Responses have been simulated for change in reference idling speed as well as in load torque. The reference idling speed may be set as a function of atmospheric conditions or considering the driver's demand as mentioned before. Therefore, it is important for the idle speed control system to maintain the desired idling speed whatever it may be. Significant torque loading during idling condition is due to vehicle accessories such as hydraulic power steering system, air conditioning system etc. It is important to test the response of the system for change in reference idling speed as well as sudden change in load torque to make sure that the engine does not stall.

5.2.1 Step Change in Reference Idling Speed

Both nonlinear and linear models were given a step change in reference idling speed from 100 rad/s to 110 rad/s radians at 10 seconds. Responses of both models are shown in figure 5.1.

5.2.2 Step Change in Load Torque

Both nonlinear and linear models were given a step change in load torque from 5 N m to 15 N m (typical load due to air conditioner in a car) at 10 seconds. Responses of both models are shown in figure 5.2.

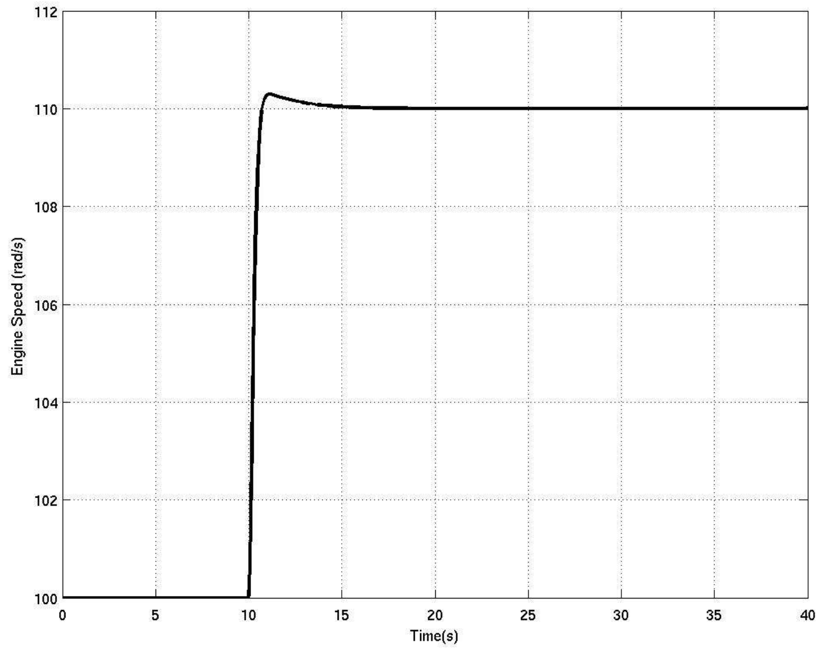

Figure 5.1 (a): Response of PID controller for step change in reference idling speed in case of linearized model

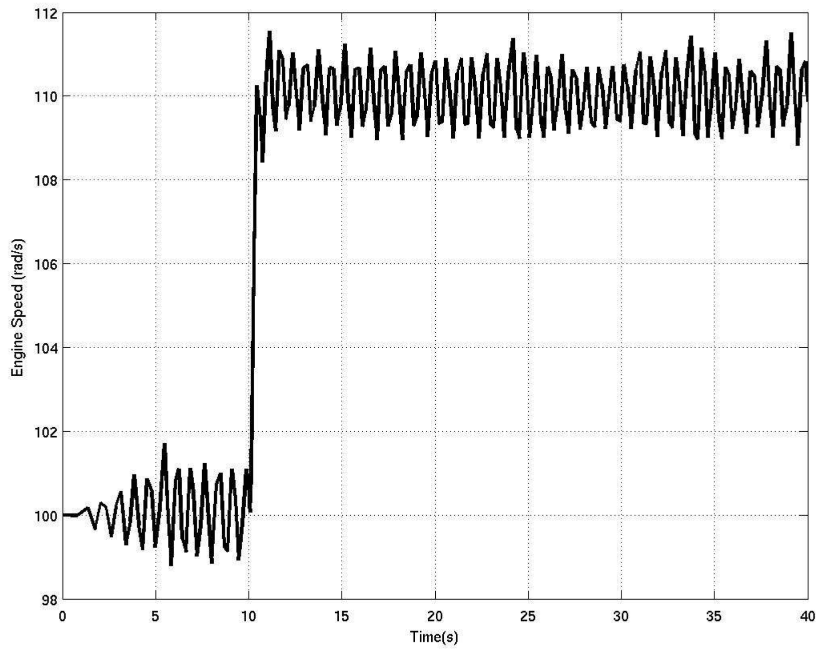

Figure 5.1 (b): Response of PID controller for step change in reference idling speed in case of nonlinear model

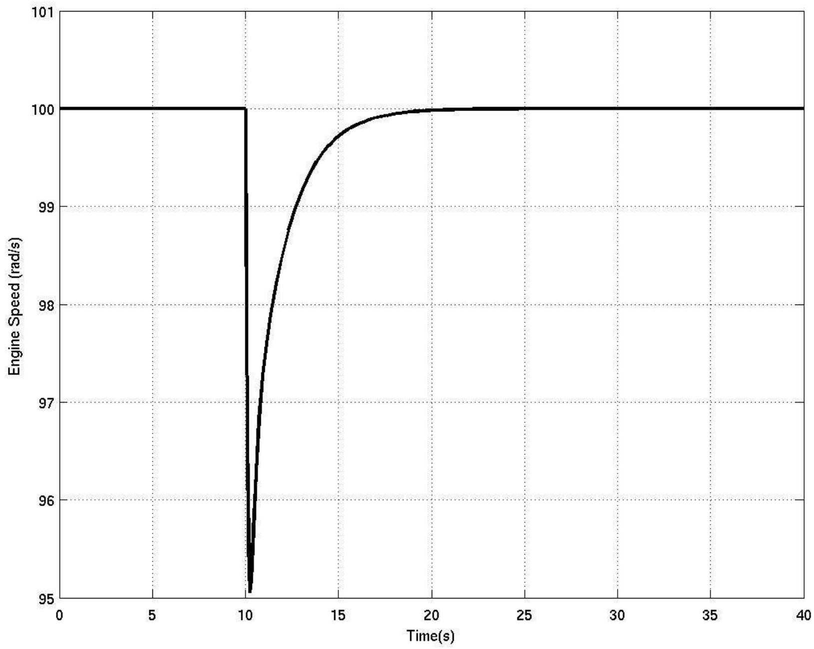

Figure 5.2 (a): Response of PID controller for step change in load torque in case of linearized model

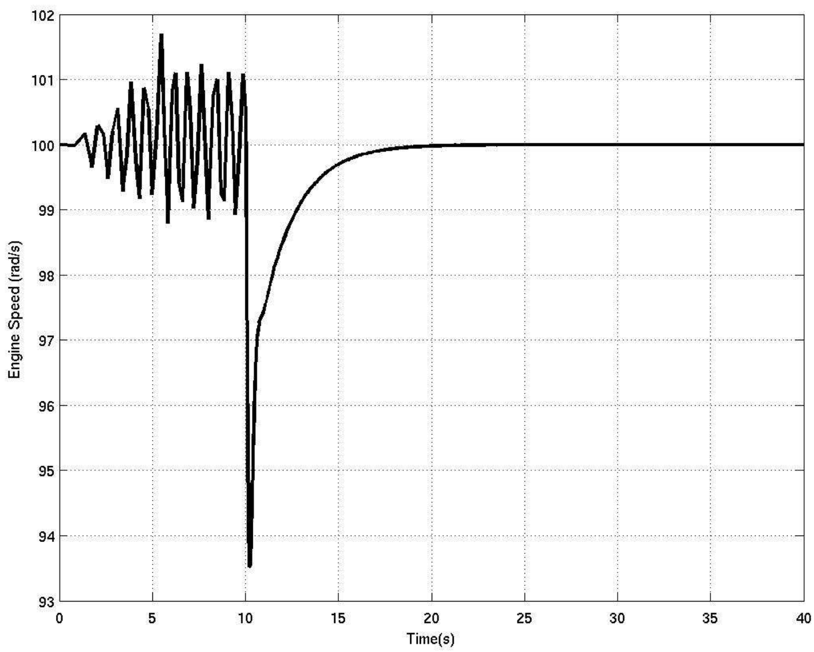

Figure 5.2 (b): Response of PID controller for step change in load torque in case of nonlinear model

Chapter 6

Summary, Conclusions and Scope of Future Work

To summarize the work, it can be said that we studied various control problems in a port fuel injection engine and developed a controller to solve one of them. In Chapter 2, we identified control objectives for different operations of engine operation. In Chapter 3, we developed mathematical equations to describe behavior of various engine subsystems in control oriented sense. In Chapter 4, we linearized the model in order to aid in the process of controller development. In Chapter 5, we developed a PID controller for idling operation of the engine. The PID controller successfully maintains a steady state idling speed in simulated environment.

The simulation results for the model show that the model has captured peculiarities of engine behavior reasonably well. Also the simulation results for the nonlinear and linearized models are close to each other. However, small oscillations observed in case of the nonlinear system are not very well captured in the linearized version. Also the steady-state values of engine speed are slightly different for the two models, as expected.

One of the important reasons for discrepancy in transient results is lack of consideration of suction-power stroke delay in the model. As an effort towards improvement in the linearized model, the delay could be considered. However this would increase order of the model and render the described method for PID gain tuning ineffective.

One more reason for discrepancy is that approximation for the throttle mass flow rate made in case of the linearized version. This approximation makes the throttle mass flow rate independent of the manifold pressure. This in-turn makes the transfer function from throttle angle to engine speed stable and the linearized model does not show steady oscillations.

As a step towards building more precise model, a discrete event based model formulation mentioned would be helpful. An additional benefit of the discrete-event based model could be its utility in developing a more precise air flow estimator. Such an estimator would also play an important role in the feedforward fuel computation for air-fuel ratio control. The next problem of immediate interest in the idle speed control is development of an air-fuel ratio controller. This would involve calibration of fuel delivery model and design of feedforward and feedback schemes for fuel delivery. Spark timing control is another area that can be looked into.

To improve the idle controller performance, a prediction scheme can be developed for sensing the load disturbances through vehicle on-board sensors. An example will be a sensor to recognize when the air conditioner is turned on and with what cooling rate. This will allow the controller to “foresee” the load torque and in a way to “be prepared” to deliver.

Appendix A: Engine Specifications and Mean Value Model Parameters

Block	**Parameter**	**Value**
Throttle Body	Angle, α_0	0.1379 radians
	Diameter, d_{th}	$(58.7).10^{-3}$ m
	Leakage area, A_{leak}	$(5.3).10^{-6}$ m^2
	Gas constant, R	287 J/kg K
	Ambient temperature, T_{amb}	298 K
	Ambient pressure, P_{amb}	$(0.98).10^{5}$ N/m^2
	Isentropic exponent, γ	1.35
Intake Manifold	Volume, V_m	$(5.8).10^{-3}$ m^3
	Air temperature, T_m	340 K

Block	**Parameter**	**Value**
Engine Air Induction	Coefficient γ_0	0.45
	Coefficient γ_1	$(3.42).10^{-3}$ s
	Coefficient γ_2	$(-7.7).10^{-6}$ s^2
	Stroke volume, V_d	$(2.77).10^{-3}$ m^3
	Clearance volume, V_c	$(0.277).10^{-3}$ m^3
	Exhaust gas pressure, P_{ex}	$(1.08).10^5$ N/m^2
Torque Production	Parameter, η_0	0.16 J/kg
	Parameter, η_1	$(2.21).10^{-3}$ J s/kg
	Parameter, β_0	15.6 N m
	Parameter, β_2	$(0.175).10^{-3}$ N m s^2
	Transport delay from suction to power stroke	0.125 s
Engine Speed Dynamics	Inertia, J	0.2 kg m^2

Bibliography

[1] C. Aquino, “Transient A/F Control Characteristics of a 5 Liter Central Fuel Injection Engine”, SAE Technical Paper Number 810494, 1981

[2] R. Bose, The Energy and Resources Institute, New Delhi, “Automotive Energy Use and Emissions Control: A Simulation Model To Analyze Transport Strategies For Indian Metropolises”, 1997

[3] R. G. Ford, “Robust Automotive Idle Speed Control in a Novel Framework”, Ph. D. Dissertation, Darwin College, University of Cambridge, 2000

[4] L. Guzzella, C. Onder, “Introduction to Modeling and Control of Internal Combustion Engine Systems”, Springer Verlag, Berlin, 2004

[5] E. Hendricks, and T. Vesterholm, "The Analysis of Mean Value SI Engine Models", SAE Technical Paper No. 920682, 1992

[6] Ministry of Petroleum & Natural Gas, Government of India, “Auto Fuel Policy Report”, 2002

[7] K. Patton, R. Nitshke, J. Heywood, “Development and Evaluation of a Friction Model for Spark Ignited Engines”, SAE Technical Paper Number 890836, 1989

[8] P. W. Reidhead, S. Gupta, D. Joshi, The Energy and Resources Institute, New Delhi, “State of India's Environment: A Quantitative Analysis”, Report Number 95EE52, 1996

[9] J. Sun & N. Sivashankar, “Issues in Cold Start Emission Control for Automotive IC Engines”, Proceedings of the American Control Conference, 1998

[10] A. Stefanopoulou, “Modeling and Control of Advanced Technology Engines”, Ph.D. Dissertation, University of Michigan, 1996

[11] W. Yuen and H. Servati, “A Mathematical Engine Model Including the Effect of Engine Emissions”, SAE Technical Paper Number 840036, 1984

Acknowledgments

I sincerely thank Professor Shashikanth Suryanarayanan for all he did and is doing for we students. “Shashi”, as we in the Robotics Lab call him, has been a constant source of enthusiasm, encouragement, support, and learning. Working together, and of course the discussions during frequent pizza parties with him, have lead to development of enormous interest inside me (and I believe - some maturity also!) about technology in general. I think that I am a changed man now!

I thank all my friends at IIT. Riessom, Manish, Manisha, Hailu, Amit have been great friends for me. I will cherish the memories of the time spent with you guys.

And lastly, I thank my parents, my grandmother, and my sister who always supported me wholeheartedly and let me continue my education.