CAPITAL UNIVERSITY OF SCIENCE AND TECHNOLOGY, ISLAMABAD

Modeling and Control of Atkinson Cycle VVT Engine for Hybrid Electric Vehicles (HEV)

by

Ghulam Murtaza

A thesis submitted in partial fulfillment for the degree of Doctor of Philosophy

in the

Faculty of Engineering
Department of Electrical Engineering

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This thesis is dedicated to;

The dreams & aspirations of my

Beloved Mother: Basheeran Bibi, my Parents in Law;

and

My sweet wife Fouzia & lovely kids: Amna Murtaza,

Muhammad Sami Ullah and Zahrah Fatima
for their love and care

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List of Publications

Journal Publications

1. Journal Publication 1
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2. Journal Publication 2
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Conference Publications

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3. **G. Murtaza**, A. I. Bhatti, Q. Ahmed and A. Hanif, "LIVC Load Control Strategy for an Atkinson Cycle Engine", Accepted in 36th Chinese Control Conference, held in July 26-28, 2017 in Dalian, China.

4. Athar Hanif, Aamer Iqbal Bhatti, Abdul Rehman Yasin, **Ghulam Murtaza**, and Qadeer Ahmed,"'Sliding Mode Based Robust Observer Design for Field-oriented Control of 3-phase Induction Machine Drive for Applications

in Hybrid Electric Vehicles"', The 33rd Chinese Control Conference, 28-30 July, 2014, Nanjiang, China.

5. **G. Murtaza**, Aamer Iqbal Bhatti and Qadeer Ahmed, "'Modeling of Atkinson Cycle Engine for Hybrid Electric Vehicles"'. Presented in WEC013, NUST, 2013, Islamabad, Pakistan

6. **G. Murtaza** and A.I. Bhatti, "Control of dc motors using sliding mode"', In Applied Sciences and Technology (IBCAST), 2012 9th International Bhurban Conference on IEEE, pages 37 - 42, 2012.

Abstract

The rational usage of fossil fuels and minimization of greenhouse gas emissions, have been the foremost goals of the automotive research and development activities. The innovation of advanced technologies such as variable valve actuation (VVA), variable compression ratio (VCR) mechanisms, use of over-expansion and downsizing have been promising the minimization of fuel consumption and emission reductions of the spark ignition (SI) engines. The integration of these technologies in the conventional SI engines is an active research area. In automotive applications, most of the time the SI engines have to be operated at part load operating conditions resulting in significant increase in part load losses and decrease in the thermal efficiency. These performance degrading aspects arise as a consequence of the throttling effect as well as declining of the effective compression ratio. Thus, performance of the SI engines has to be improved at part load operating conditions, so as to lower the transportation fuel consumption. However, to accomplish the aforesaid objectives and vehicle drivability, availability of accurate and simple control-oriented engine model incorporated with innovative technologies and engine robust controllers are the key contributors. Whereas, SI engine models available in the public literature either have not been incorporated with advanced mechanisms or they suffer from number of limitations, such as complexity and difficulty in analysis and control design. This research frameworks a novel physically motivated control-oriented extended mean value engine model (EMVEM) of the Atkinson cycle engine, wherein the Atkinson cycle, flexible VVA, VCR and over-expansion characteristics have been incorporated. For this purpose, an intake valve timing (IVT) parameter along with its constraints is introduced, which has a vital role, in modeling the inclusive dynamics of the system and to deal with engine performance degrading aspects. The proposed model is validated against the experimental data of a VVT engine during the medium and higher engine operating conditions, to ensure that the model has proficiency, to capture the dynamics of the Atkinson cycle engine and to handle the engine load through an unconventional control strategy. The model encapsulates air dynamics sub-model in which VVA system is unified to cope with the engine performance degrading aspects at part load and physics-based rotational dynamics sub-model, wherein Atkinson cycle is utilized for analysis. The control-oriented model is anticipated for systematic analysis, simulations, development of control system and estimation

strategies to support and enable novel load control strategies, as well as to improve fuel economy of the Atkinson cycle SI engine used in hybrid electric vehicles (HEVs). Besides, a conventional PID control and a robust nonlinear higher order sliding mode (HOSM) control framework for the Atkinson cycle engine with an unconventional flexible intake valve load control strategy instead of the throttle is designed and developed. The control frameworks based on the novel EMVEM model are evaluated by using the notion of VCR in the view of fuel economy for the standard NEDC, FUDS, FHDS and US06 driving cycles. The resulting approaches confirm the significant decline in engine part load losses and improvement in the thermal efficiency and accordingly, exhibits considerable enhancement in the fuel economy of the VCR Atkinson cycle engine over conventional Otto cycle engine during the medium and higher load operating conditions. This research introduces an alternative control scheme resulting in around 6.7% fuel savings.

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Abbreviations

AFR	Air to Fuel Ratio
aBDC	After Bottom Dead Center
aTDC	After Top Dead Center
bBDC	Before Bottom Dead Center
BDC	Bottom Dead Center
bTDC	Before Top Dead Center
BSFC	Brake Specific Fuel Consumption
CAI	Controlled Auto-Ignition
CCEM	Cylinder by Cylinder Engine Model
CI	Compression Ignition
CO	Corbin Monoxide
CO_2	Corbin Dioxide
DEM	Discrete Event Models
ECU	Electronic Control Unit
EEVO	Early Exhaust Valve Opening
EGR	Exhaust Gas Recirculation
EIA	Energy Information Administration
EIVC	Early Intake Valve Closing time
FUDS	Federal Urban Driving Schedule
FHDS	Federal Highway Driving Schedule
GHG	Green House Gases
HC	Hydrocarbons
HEV	Hybrid Electric Vehicle
ICE	Internal Combustion Engine

IVC	Intake Valve Closing Time
IVT	Intake Valve Timing Parameter
LEVC	Late Exhaust Valve Closing
LIVC	Late Intake Valve Closing Time
NEDC	New European Driving Cycle
MEP	Mean Effective Pressure
MVEM	Mean Value Engine Model
PID	Proportional Integral Derivative
SA	Spark Angle
SFC	Specific Fuel Consumption
SI	Spark Ignition
SMC	Sliding Mode Control
SOSMC	Second Order Sliding Mode Control
HOSM	Higher Order Sliding Mode
TDC	Top Dead Center
VCR	Variable Compression Ratio
VSE	Variable Stroke Engine
VVA	Variable Valve Actuation
VVT	Variable Valve Timing
WOT	Wide Open Throttle

Symbols

P_a	Ambient pressure
P_m	Manifold pressure
T_a	Ambient temperature
T_m	Manifold temperature
ϕ_{cl}	Throttle angle at closed position
ϕ	Throttle angle
V_d	Displaced volume
V_{ivc}	Volume of combustion cylinder corresponding to intake valve closing
V_m	Manifold volume
D	Inlet diameter
γ	Ratio of heat capacities
r_e	Expansion ratio
r_c	Compression ratio
R	Specific gas constant
C_D	Throttle discharge coefficient
m	Air mass
m_f	Fuel mass
m_m	Mass of air-fuel mixture
m_i	Air mass flowing into the manifold
m_o	Air mass flowing into the cylinder
$\dot{m}_{fv}$	Fuel vapor mass flow
$\dot{m}_{ff}$	Fuel film mass flow

$\dot{m}_{fi}$	Injected fuel mass flow
$\dot{m}$	Air mass rate
$\dot{m}_i$	Air mass rate flowing into the manifold
$\dot{m}_o$	Air mass rate flowing into the cylinder
λ	Ratio of the expansion ratio to the effective compression ratio
η_{vol}	Volumetric efficiency
η_c	Combustion efficiency
η_{otto}	Thermal efficiency of Otto cycle engine
η_{atk}	Thermal efficiency of Atkinson cycle engine
T_{ind}	Indicated Torque
T_{pump}	Pumping Torque
T_{fric}	Frictional Torque
T_l	Load Torque
ρ_a	Air density
AFR	Air to fuel ratio
J_e	Engine inertia
Q_{LHV}	Heat value of fuel
C_v	Heat capacity at specific volume
ω_e	Engine angular speed
N	Angular speed
β	Pressure ratio
K_P	Proportional gain
K_I	Integral gain
K_D	Derivative gain
B	Engine Bore
S	Engine Stoke
P	Maximum power
v_{cyc}	Vehicle speed
G	Final drive ratio
r	Tire radius

s	Sliding surface
u	Control effort
ω_{eref}	Engine reference speed

Chapter 1

Introduction

1.1 Background

The public has been aware of the issues such as energy savings and environment protection since last many decades especially after the oil crises in 1970's. Both motives walk hand to hand as soon as it is accredited that almost all energy resources, that Western civilized society is utilizing for their living, are not endless and these are creating environmental problems as well. Oil is the leading problematic; amongst the energy means and surprisingly transportation is one of the greatest oil consuming sector. According to a current study conducted by U.S. Energy Information Administration (EIA) [11], in United States 68% of the all available petroleum resources were used in transportation sector in 2006, whereas 66% of these capitals were imported. Correspondingly, in Pakistan (developing country), the oil consumption has grown over the time and averaged 437,000 barrels per day in 2013, while more than 80% of these non-renewable resources are imported as reported by EIA United States [12]. However, Pakistan is using about 48% of the available petroleum resources in transportation sector [13].

Furthermore, the application of several restricting legislations to control the pollutant emissions, specifically the greenhouse gases (GHG) is also essential part of this discussion, as there is a direct relation between the quantity of energy consumed and the need for more fuel proficient devices. For instance, all the countries those have signed the international agreement (Kyoto protocol) have to decrease their whole emissions of the pollutant gases, especially the carbon emissions about 5.2% below 1990 level by 2012 and so on. The research also figured out that the overall energy conversion efficiency in the transportation sector is worst estimated at only 20%. Thus, due to the growing energy demands, limited sources of fossil fuel, poorest energy conversion efficiency and stringent regulations regarding emission [14], the automotive industry is progressively pursuing unconventional propulsion systems (practical technologies) to decrease the reliance of transportation on the non-renewable fuels as well as to improve the fuel economy and the efficiency of the vehicles.

At present, the hybridization of the conventional and the electric power-train, with the intentions of reduction in fuel consumption and noxious exhaust emissions, is one of the viable technologies. These hybrid vehicles are commonly known as hybrid electric vehicles (HEVs), exploit electrical energy system (batteries) and electric drive system frequently in combination (series or parallel configurations) with internal combustion engine (ICE) generally of a smaller size (downsized) than the equivalent one using only the ICE. Dr. Ferdinand P. built the first hybrid vehicle in 1898 and utilizes an ICE to spin the generator which charges the batteries to provide power to the electric motor positioned in the wheel hubs [15]. However, hybrid vehicles have been overlooked primarily because of the evolution of the IC engines technologies and the availability of fossil fuel at a rational price. But today, with continuous increase in the petroleum's cost, growing global population continues to take its toll on the supply and with emergence of the harmful emission legislation, the situation is changing rapidly.

Consequently, to meet the standards of automobile users and requirements of government regulations, the hybrid electric vehicles are promptly entering the market and are produced in several power-train configurations. Commonly, to achieve more fuel economy improvement, an over-expanded engine known as Atkinson cycle engine is used in the hybrid vehicles. Although the fuel consumption is reduced considerably, however to answer the market demands as well as the production of such vehicles and its state of the art are not plentifully good. To resolve the energetic and environmental issues, arising from the transportation demands, the improvement in the existing internal combustion engines is still a significant alternative.

1.2 Motivation and Objectives

The IC engines are used in wide range of applications from electricity production to automotive. So, there are essentially different requirements for the various SI engine applications. For instance, for electricity production an engine works at a constant speed and load, whereas automotive engines have to be operated under continuously varying conditions, during the entire operating range. At any obligatory conditions, the engine can be designed to have its optimal operating point (minimum fuel consumption or maximum power) in the prior case. However, establishment of optimum working conditions to define a point or a most recurrently used range, in the subsequent application is a matter of compromise.

The load is controlled through the quantity of fuel injected during each cycle of the engine in the compression ignition (CI) engines. With this combustion technique, engine efficiency increases as the load reduces. However, in case of the spark ignition (SI) engines load is controlled with the throttle valve, upstream of the intake valve which creates a pressure decline in the intake manifold during the induction stroke, reducing the amount the air-fuel mixture induced in each cycle. As a result, due to the decline in the intake manifold pressure at part load, the pumping losses increase substantially and the effective compression ratio of the engine decreases. In addition, with low peak pressure and peak temperature during the combustion process, the combustion efficiency also deteriorates. Consequently, reducing customarily the part load thermal efficiency of the SI engine and is as shown in Figure 1.1. However, in case of the SI engines, a major impact on the overall efficiency and emissions of the engine put forth by valve events

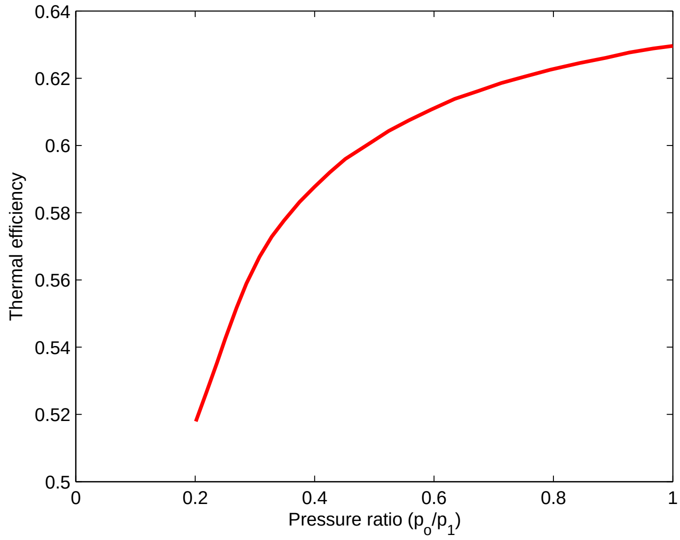

FIGURE 1.1: Part load thermal efficiency of the conventional Otto cycle engine versus pressure ratio [1].

and their timings. The conventional SI engine has compromised results among the engine performance, efficiency and its maximum power during the entire operating range, as a consequence of the fixed valve timing and the synchronization between the crankshaft and camshaft [4]. To cope with these compromises, the innovative mechanisms such as variable valve timing (VVT), variable compression ratio (VCR) etc. domains have seen remarkable research and developments in the last two decades [10, 16–20]. Numerous types of VVT mechanisms [21–25], VCR methods [5, 18, 26–28] etc. have been proposed and designed, then the use of VVT systems have made feasible to control the valve lift, phase and timing at any point of the engine map and VCR mechanism maintains the compression ratio to augment its performance.

With the fast growth in automotive industry, the automotive researchers have tried hard to develop methods and employ the various innovative technologies to accomplish the engine performance as fuel economy, emission reduction and vehicle drivability. With these intentions the modeling and control of the powertrain of vehicle has been of prodigious consequence i.e. the availability of accurate and simple control-oriented engine model incorporated with advanced technologies and the engine robust controllers, are the key contributors [29–32].

From this perspective, the development of comprehensive SI engine model incorporated with advanced technologies significantly benefits the automotive manufacturers for the performance evaluation of virtual prototype and in the testing of different controllers, to lower the fuel consumption as well as noxious exhaust gases. In this view, various modeling approaches specifically empirical methods, mean value engine model (MVEM) as well as cylinder-by-cylinder engine models (CCEM) of the conventional Otto cycle engine have been remained the focus of the research community. However, the automotive researchers [6, 10, 31, 33–35] have developed cylinder-by-cylinder engine models (CCEM) of the over-expanded cycle engine. Although, CCEM models are more physics-based and are more precise than the MVEM approaches and proficient of capturing the details of the engine events but CCEM models suffer from number of limitations, such as complexity and difficulty in analysis and control design. Owing to the inherent limitations of cylinder-by-cylinder models, MVEM is one of the most well-known, suitable system models for analysis and model based control applications and is used broadly by the research community for the engine control strategy development [36–40].

Thus, the limitations of the conventional SI engine as well as CCEM models of the over-expanded engine cycle motivate for the development of the MVEM model of the Atkinson cycle engine incorporated with flexible variable valve timing, variable compression ratio and over-expansion characteristics. The model should have the capability to deal with the conventional engine part load issues and to accomplish fuel economy and emission reduction. Besides, it can also have vital role in implementation of electronically controlled load handling strategy instead of the costly mechanical engine control mechanism. Brief introduction of Atkinson cycle engine is given in the following section.

1.3 Atkinson Cycle Engine

A novel internal combustion engine invented by James Atkinson (British engineer) in 1882, which is known as Atkinson cycle engine [41]. The main diversity of the

Atkinson cycle as compared to the Otto cycle is that, its expansion stroke is longer than its compression stroke which contributes to deliver more energy from the combusted gas to piston due to the over-expansion in the power stroke. Whereas, in the Otto cycle engine both the expansion stroke and compression stroke have identical lengths.

Due to the complex mechanical design, the crankshaft's requirements and piston linkage in order to implement a relatively shorter compression stroke in contrast to the expansion stroke, the novel Atkinson engine is hardly ever used. Modern Atkinson cycle engine can be accomplished by Variable Valve Timing (VVT) with the same crankshaft design as for the Otto cycle engine. With optimum intake and exhaust valves opening and closing timings, longer expansion ratio than compression ratio, the primary characteristics of Atkinson cycle can be established. The PV diagram of the ideal Atkinson cycle is shown in Figure 1.2.

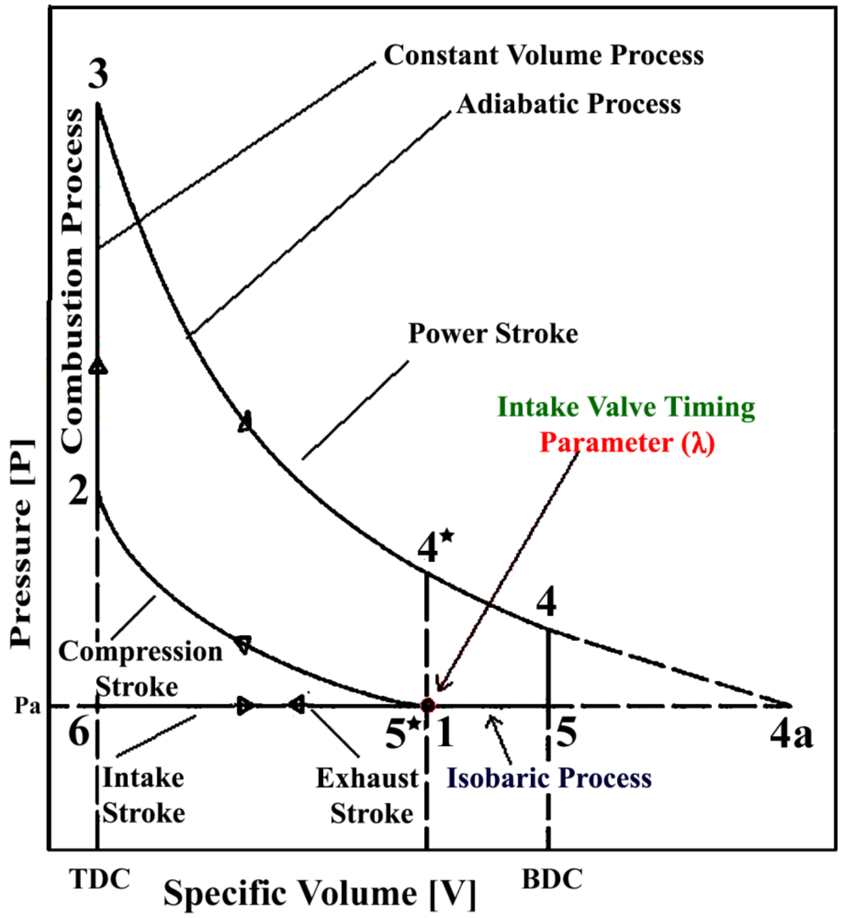

Figure 1.2: Theoretical PV representation of an ideal Atkinson cycle engine [2, 3]

The Atkinson cycle engine, often mentioned as an over-expansion cycle engine, is naturally aspirated spark ignition engine working in four thermodynamic processes named as intake, compression, power and exhaust strokes as illustrated following Figure 1.3.

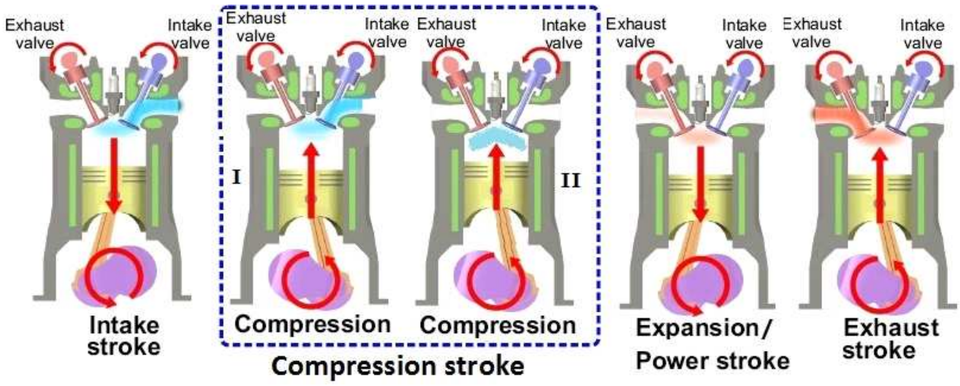

FIGURE 1.3: Four strokes of the Atkinson cycle engine [*courtesy of Wikipedia*]

The process is ongoing with inspiring fresh air and fuel into the combustion cylinder through intake valve to ingest the air-fuel mixture termed as intake stroke. However, the closing of intake valve is retarded towards the end of the compression stroke. As a consequence, some of charge (air-fuel mixture) is expelled back into the intake manifold during the first part of compression stroke. Then, the charge is compressed up to a certain level during the compression stroke (with closed intake and exhaust valves), close to the Top Dead Center (TDC); afterwards the charge is ignited by spark plug. The spark plug gives a high electrical voltage between the two electrodes; consequently the air-fuel mixture in the combustion chamber surrounding plug ignites. The third stroke takes place after the combustion process is responsible for work done by the engine and is known as power/expansion stroke. Finally the exhaust gases are expelled out from the cylinder via exhaust valve.

However, potential benefits of Atkinson cycle are achieved at the expense of power density. The electric motor excels at low speed power but is less efficient at high speeds, whereas the Atkinson engine is on the other way round, with less power at low speed but good power at higher engine speed. So, both the electric motor and the Atkinson cycle engine can contribute as a vehicle propulsion system in their most efficient ranges.

1.4 Contributions

In this work, one solution is proposed which has vibrant role in the reduction of SI engine fuel consumption at part load operation and is also appropriate for its model based control development. This is a control-oriented extended mean value engine model (EMVEM) of an Atkinson cycle engine, wherein the innovated technologies such as variable valve actuation, over-expansion and variable compression ratio systems are integrated. Furthermore, control-oriented EMVEM based robust control framework is designed, developed and evaluated for an Atkinson cycle engine in view of fuel economy. Fuel economy is pursued through combination of over-expansion principal together with the adjustment of compression ratio. The Atkinson cycle i.e. over-expanded cycle is realized with flexible intake valve timing variation in accordance with engine operating conditions. The proposed model comprehend the four strokes in the Atkinson cycle diagram: the intake, compression, power and exhaust. All the four stroke operations are characterized by physics-based modeling method. The aim of this research is always to attain the optimum SI engine efficiency despite the load demands, in consequence improving engine thermodynamic performance at part load operating conditions.

The main contributions of the presented research work are described as:

- Development of control-oriented extended mean value engine model (EMVEM) of the Atkinson cycle engine
- Design, development and evaluation of a robust control framework for the Atkinson cycle engine in fuel economy perspective.
- Introduction of an alternative control scheme resulting in around 6.7% fuel savings.

Flowchart diagram to comprehend the dissertation contributions is as follows:

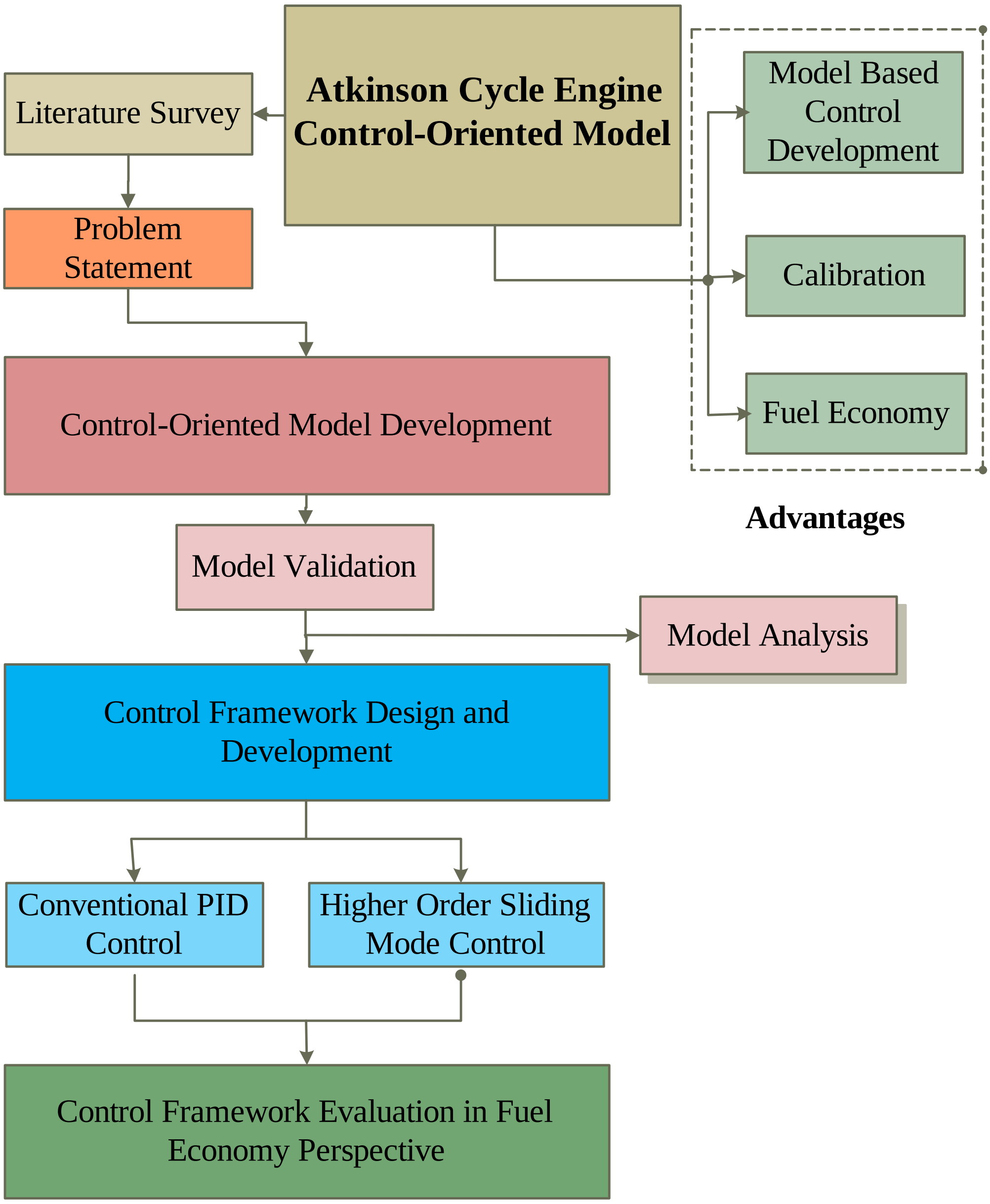

FIGURE 1.4: Thesis contributions illustration

1.5 Organization of the Thesis

Rest of the thesis is organized as discussed below:

Chapter 2 will familiarize the reader with SI engine modeling current avenues, which is organized in two categories: the innovated technologies and the modeling approaches of spark ignition IC engine. In the leading part, various recently invented technique (independently or in combination) playing vital role in enhancement of fuel economy and emission reduction are presented. Whereas, different approaches of the SI engine modeling primarily and focusing finally on the modeling of the Atkinson cycle engine, are discussed comprehensively in the later part of the chapter. The literature review will be exploited for motivation of this research.

Chapter 3 describes the main contribution of this dissertation. Development of control-oriented extended mean value engine model (EMVEM) Atkinson cycle engine established on its manifold pressure and angular speed dynamics. Based on the air flow across the intake manifold incorporated with variable valve timing mechanism, the pressure dynamics is modeled. While, the rotational engine dynamics is described by the torque produced by the combustion process, pumping and frictional accomplishments by taking into account the Atkinson cycle i.e. over-expanded cycle attained through flexible intake valve timing and benefits of the variable compression ratio, instead of the conventional Otto cycle. The developed control-oriented EMVEM model provide a prospect for systematical analysis, fault diagnostics and model based control application of the VVT engine, besides the better fuel economy in comparison with the conventional Otto cycle engine.

Chapter 4 comprises of the design, development and evaluation of a control framework for the Atkinson cycle engine in the view of fuel economy. In this chapter, nonlinear control-oriented EMVEM dynamics is analyzed for its controllability point of view with respect to an unconventional input control channel i.e. late intake valve timing load handing strategy instead of the throttle. A proportional integral derivative (PID) control is employed to evaluate the Atkinson cycle engine for the standard New European Driving Cycle (NEDC), Federal Urban Driving Schedule (FUDS) and Federal Highway Driving Schedule (FHDS) using the notion of variable compression ratio at medium and higher load operation. This chapter also presents a comparative view of fuel economy accomplished through the Atkinson cycle engine EMVEM model against the conventional SI engine.

Chapter 5 contains brief overview of sliding mode control (SMC) specifically higher order sling mode (HOSM) control strategy with its advantages like robustness against the un-modeled dynamics, parametric variations and external disturbances. A nonlinear robust HOSM super-twisting control framework is developed for the Atkinson cycle engine EMVEM model as well as for the conventional Otto cycle engine model to asses the fuel consumption for both the engines during the standard FUDS, NEDC and US06 driving cycles. The results confirms the control-oriented engine EMVEM model as well as robust controller are the key contributors to enhance the SI engine performance.

Chapter 6 concludes the dissertation through outlining the main contributions and list of directions that can be accomplished in the future research.

Chapter 2

Engine Modeling: Current Avenues

This is an introductory chapter, which describes an introduction to spark ignition (SI) internal combustion engine, significance of an engine modeling and advanced technologies need to be incorporated in the conventional SI engine in order to optimize its performance. The leading part of this chapter, Section 2.2 elaborates comprehensively the various innovative mechanisms being exercised by researchers around the world to overcome the compromised solution of the conventional IC engine. Section 2.3 explains an overview of the past efforts of the automotive engineers to the IC engine modeling approaches such as mean value engine modeling as well as cylinder by cylinder engine modeling strategies along with their applications. The research on the Atkinson cycle engine used in hybrid electric vehicles instead of the conventional Otto cycle engine is exhibited in Section 2.4. Atkinson cycle engine modeling efforts and mechanisms to realize the Atkinson cycle to ascertain its better fuel economy and emission reduction are enlightened comprehensively. It is the foremost part of this chapter and still a challenging task for the automotive industry. The conclusion of this chapter, Section 2.5 point towards the motivation and objectives of this research work in view of Atkinson cycle engine modeling and pursuance of its advantages in fuel economy perspective.

2.1 Introduction

Since the Industrial Revolution in Europe, the notion of developing internal combustion engines (ICE) has been started. The objective of the IC engines is to convert the chemical energy stored in the fuel to mechanical energy with an appropriate mechanism. Based on the various criteria, for instance the fuel type, working cycle of the engine and techniques of ignition, these engines can be categorized as, in [42].

The IC engine can also be classified depending on approach of fuel ignition:

- Spark ignition engine (SI): the fuel, petrol, is ignited through spark ignition mechanism (Spark plug), mounted at the top of the combustion cylinder.
- Compression ignition engine (CI): the fuel, diesel, this type of ignition has lower self-ignition temperature as compared to SI engines. With the increase of the in-cylinder pressure during the compression stroke the temperature inside the combustion cylinder also rises, and therefore the fuel starts self-ignition.

The engine under discussion is naturally aspirated spark ignition conventional internal combustion engine, which has been widely used in the commercial vehicles. These engines typically work in four thermodynamic processes termed as intake, compression, expansion and exhaust strokes. However, the charge (air-fuel mixture) is compressed up to a certain level during the compression stroke, as intake valve remains closed from very start of the compression stroke. The PV diagram of the conventional ideal Otto cycle IC engine having length of the expansion stroke equal to that of the compression stroke is shown in Figure 2.1.

Due to ever growing energy demands, limited resources of fossil fuel and stringent regulations regarding emissions [14], the automotive industry is forced to develop practical technologies, which have vital role in the accomplishment of optimum performance i.e. fuel economy improvement and emission reduction. To optimize the internal combustion engine performance in future, the advanced technologies

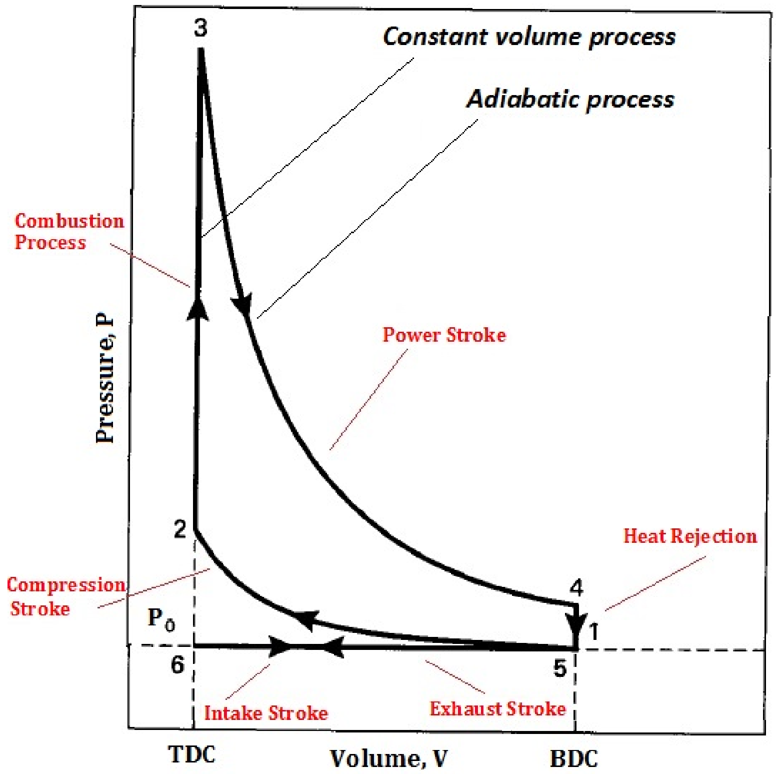

FIGURE 2.1: PV diagram of the conventional ideal Otto cycle IC engine [3].

will have to focus on "variable everything". To improve fuel economy and lower the parasitic losses adding "on-demand" and "adjustable" to almost every system is mandatory[43].

This has been the prime intention of the engine research and advancement in the past few decades. The evolution and the use of technologies like variable valve timing, variable compression ratio mechanisms and over-expansion etc. have made possible to develop and design the engines those work throughout the entire operating conditions, and are playing vibrant role in the improvement of fuel economy. The conventional IC engines where everything is fixed and the engine power can only be controlled with the throttle position, are being generally replaced with the engine that can change the spark timing or control the amount of fuel per cycle through the alteration in the valve timing as well as the engine's effective displacement.

The engines operate at varying speed and load conditions in the automotive applications, the limiting constraint of the fixed configuration, forcing the engine designers to assume compromised solutions for its configuration and working conditions, consequently resulting in poor engine performance concerning the fuel economy. However, the engines equipped with variable parameters can be operated over wide range of operating conditions at the optimum performance to achieve the optimal reduction in fuel consumption along with the thermodynamic improvements.

The causes of losses or inefficiencies in the Spark Ignition IC engines with technical solutions to minimize these losses are presented in the following table.

TABLE 2.1: Various losses Vs Compensating technologies for fuel economy improvement of the IC Engine[10]

Sr. No.	Engine Losses Vs Technologies	Variable Valve Actuation	Variable Compression Ratio	Over-Expansion	Stratified Charge	Downsizing	Variable Displacement
1	Pumping Losses	√		√		√	√
2	Frictional Losses					√	√
3	Incomplete Combustion		√		√		√
4	Exhaust to Ambient	√	√	√	√	√	
5	Exhaust Blow down	√	√	√	√		√
6	Heat transfer			√	√	√	√

The advantages of these methods on their own are significant but the grouping of several techniques consent even better enhancements in the SI engine performance. The research [44] states the advantages such as:

- The variable valve timing and lift play vital role in the reduction of throttling effect leading up to 5% decrease in fuel consumption.
- At part load operation the enhancement in fuel economy can be achieved approximately 7% by variable compression ratio mechanism.

- Advancement in terms of fuel economy goes up to 17% with the use of lean mixtures at part load conditions.
- The overall effect of all the technologies discussed in this research reports the improvement of 24% in fuel economy.

The potential advantages of the three evolving technologies used simultaneously: VVT, VCR and over-expansion are the main emphasis of this section. However, a comprehensive view of the prevailing mechanisms that allow for an engine's thermodynamic improvement is also discussed. An analysis will be made on the technologies to stress on the development got in the recent past apropos the thermal efficiency and supplementary performance parameters of the engine, in the subsequent sections.

The literature review chapter emphasizes on SI engine innovative technologies and primarily on the IC engine modeling: First part describes the various evolving technologies, which are being incorporated to overcome the compromised solution of the conventional spark ignition internal combustion engine, in order to enhance its performance in addition to fuel economy and emission lessening. Whereas, in the second foremost part the various modeling approaches of the Atkinson cycle engine as well as the conventional Otto cycle engine essentially are discussed comprehensively, particularly the mean value engine modeling .

2.2 Innovative Technologies of Internal Combustion SI Engine

Several innovated mechanisms to be integrated in the conventional IC engine are elaborated in the following subsections.

2.2.1 Variable Valve Actuation (VVA)

Variable valve actuation systems are the mechanisms which enables an engine to have different valve events in terms of lift, phasing and timing as well as the ability to change the opening area of the valve primarily depending upon the engine load demands [10].

The continuous variable lift with cam phasing and the electromagnetic valve operating mechanism are the most promising system, but simultaneously needs excessive effort for its advancement and production due to its complexity. Depending on the VVA mechanism's configuration, the variation of the valve actuation permits the variation of the subsequent variables:

- Variable valve timing philosophy
- Valve Overlapping
- Valve lift and Velocity

The optimum engine performance can be accomplished by accurately controlling the variation of the different valve events and at the same time VVA system will also be more efficient by curtailing its frictional losses.

2.2.1.1 Variable Valve Timing Philosophy

In this subsection the significance of the variable intake valve timing and variable exhaust valve timing in the improvement of fuel economy during the entire engine operating conditions is explained.

A. Variable Intake Valve Timing

The engine load can also be controlled by variable Intake Valve Closing time (IVC) [45, 46]. The energy losses during the inhaling of sub-atmospheric gases during the intake stroke and expelling of exhaust gases during the exhaust stroke are the foremost shortcomings of the ICE. These pumping losses are dependent on the throttle valve opening and closing position. Moreover, these losses are low at wide open throttle (WOT) and are high when the throttle valve tends to close (at the part load). Therefore, the pumping losses are inversely proportional to the load of the engine [4]. With wide open throttle (WOT), by varying the intake valve-opening duration the control of the air-fuel mixture can be realized. Thus VVT has substantial potential for the reduction of pumping losses, which plays vital role in the improvement of fuel economy during the entire operating conditions.

The fuel must be burnt at a high rate and next to the top dead center (TDC) to attain improved cycle efficiency and the burning rate is profoundly dependent on the in-cylinder mixture's turbulence. The turbulence generated throughout the intake stroke is very low at engine's low speed and it can be overawed with delaying the intake valve opening time so that this happens at the instant while the piston has its higher speed. Thus, the air velocity and the internal turbulence will be improved.

With the valve timing adjustment, a flatter speed torque curve can be produced instead of the compromised one, with peak at its intermediate speed range, as a consequence of the conventional valve train. At engine low speed, delaying of valve closing must be evaded to acquired higher effective compression ratio, whereas the valve closing must be retarded to take the benefit of the ram effects. These strategies can be employed throughout the entire operating range, primarily to the engine with variable compression ratio (VCR) and with the fixed power stroke to attain the fuel economy. The valve timing diagram in relation with PV diagram for the conventional four stroke spark ignition engine is shown in Figure 2.2.

In the conventional SI engine, the intake valve opens during the intake stroke and the air-fuel mixture is admitted into the combustion chamber. The intake valve closes in the compression stroke and the charge gets compressed. Whereas, with the late intake valve closing time (LIVC), the intake valve remains open for longer period during some part of the compression stroke so that some of air-fuel mixture is expelled back into the manifold or simultaneously becomes intake of the induction stroke. Consequently, the pressure of the entrapped charge will be slight in excess of the atmospheric pressure.

The entrapped air-fuel mixture gets readmitted, during the succeeding induction stroke, at a pressure above that of the charge in the conventional SI engines. Thus, the suction pressure declines slightly from the atmospheric pressure line; as a result the pumping losses are reduced. Tuttle [45] has reported that with LIVC system,

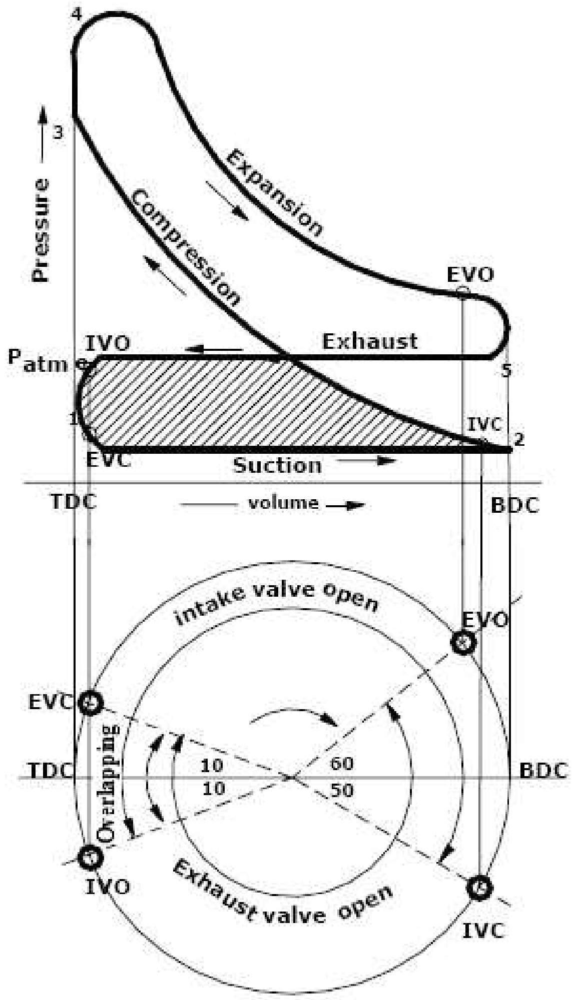

FIGURE 2.2: The PV diagram of the conventional four stroke Otto cycle engine along with its valve timing diagram [4].

there was a 40% decrease in pumping losses. He also testified 7% reduction in fuel consumption and 24% decline in NOx emissions without any change in the hydrocarbons (HC), particularly at part load. At higher speeds the volumetric efficiency will improve with LIVC as the air-fuel mixture high flow momentum continues to fill the cylinder while the piston is traveling upwards [21]. However, with LIVC the volumetric efficiency will be penalized at low speed because of the back-flow of some of the fresh charge into the intake manifold. The work in [47] concluded that the breathing characteristics (volumetric efficiency) of an engine can be enhanced with most promising strategy LIVC and it is beneficial to use VVA devices to change both lift and duration.

A special VVT control mechanism was proposed in [22], through which the intake valve closing time can be varied from 38^o to 78^o after bottom dead center (aBDC) without changing exhaust valve timing and lift. Various engine parameters for instance brake torque, volumetric efficiency, specific fuel consumption (SFC) and emissions were explored for entire speed range at full load or wide open throttle (WOT). It was reported that the brake torque enhanced by 5.1% at low speed and 4.6% at high speed of the engine, whereas SFC is decreased by 5.3% and 2.9% at the low and high speeds respectively with variable intake valve timing. However, the part load operating conditions have not been taken into account.

In [16], a comprehensive analysis was accomplished to explore the optimized valve timings and its effects on the engine performance parameters like power, torque and fuel consumption at full load and extreme engine speeds. In this regard a thermodynamic simulation model was developed and applied to a hypothetical engine cylinder to figure out the optimum arrangement. According to Damasceno, for the best valve timings of the IVO, IVC, EVO and EVC 1.58% and 4.71% enhancement in the volumetric efficiency and torque, whereas 16.52% reduction in residual gas friction for 2000 rpm is reported. On the other way round, there is 4.0% and 8.8% improvement in the volumetric efficiency and torque, while 9.7% decrease in residual gas friction for 5000 rpm is stated.

According to Ahmed and Theobald [48] by using LIVC arrangements, the problem of ignition may occur before the intake valve has been closed, at very low load conditions. Rabia and Kora [49] stated that the knocking onset in the engine occurs to escalate with the reduction of the engine speed in LIVC, as with low charge density, the mixture becomes richer, so the flame speed decreases, consequently the engine knocking increases. Therefore, with LIVC system the engine demands further spark advance than the conventional SI engines, particularly at part load. The air-fuel mixture is allowed enough time to auto ignite, in this way auto ignition can be avoided. The authors in[17, 50] have also figured out that the combination of LIVC with VCR mechanism is more efficient technique for fuel consumption saving over the conventional engines and results in an Otto-Atkinson cycle engine. It is reported that 13% brake specific fuel consumption (**bsfc**) can be attained through LIVC and can be enhanced up to 20% by using LIVC as well as VCR mechanisms, at part load conditions. The former technology is expedient to pumping losses lessening, whereas the later one is worthwhile to increase the expansion ratio and serves to vary the amount of air-fuel charge bestowing to the operating conditions.

B. Variable Exhaust Valve Timing

To optimize the expansion ratio at engine low speed, the option of varying the exhaust valve opening can be used. The thermal efficiency and subsequently the fuel efficiency as well as the torque can be improved by delaying the opening event of exhaust valve particularly at low load conditions. Nevertheless, to avoid interference, the availability of space between valve and piston limits this technique [51]. The control of EGR, eradicating the requirement of EGR extra equipment can be accomplished with the exhaust valve closing time. This is a corresponding approach of load controlling through intake valve closing time. With early exhaust valve opening (EEVO) well before the expansion stroke will provide well scavenging of the burnt gases, consequently the expansion work and thus, the engine's output power is reduced significantly. However, it will cause the reduction of pumping losses. According to Siewert et al. [52], the exhaust hydrocarbons increase as a result of EEVO because it interposes the completion of in-cylinder hydrocarbons reactions. Likewise, the EEVO interrupts the oxidation process in the cylinder reasons an increase in the CO.

2.2.1.2 Valve Overlapping

The valve overlapping is also one of the promising parameter, which can be controlled by valve actuating systems. The period, while both the intake as well as exhaust valves remain open, affects the emissions, the engine's idle speed behavior and full load performance. The optimal adjustment of the valve overlapping to diminish the amount of residual gas inside the cylinder plays vital role in the improvement of the stability during the idle operation. Furthermore, at part load the quantity of residual gas inside the cylinder is increased, by shifting the overlap towards the intake stroke, causes to reduce the NOx and unburned hydrocarbons. These un-burnt hydrocarbons are drawn back into the combustion cylinder before the exhaust valve closing and therefore reducing the pumping work for the duration of the intake [24]. According to Philips and Kramer [51] huge overlap results to greater HC emissions as the excessive amount of residuals retained in the cylinder deteriorate the combustion process and consequences to combustion instability. The substantial reduction in fuel consumption can be attained with proper adjustment of valve overlap and valve lifts variation, allows for a lower idle speed.

2.2.1.3 Valve lift and Velocity

The valve lift variation as a function of engine speed leads to fuel consumption saving in the valve-train motion. At low speed, the air speed crossing the valve can be increased by reducing the lift which may increase turbulence, enhancing the burning condition as a result of superior air-fuel mixture formation. Yet, a prime operating point between the best mixture formation and minimization of the pumping losses can always be found. The exhaust valve lessening can also be employed to throttle the exhaust flow and retain the last portion of the exhaust charge in the cylinder containing the high concentration of unburnt HC, which are later on burnt in the succeeding cycle. Furthermore, the increase of valve lifting velocity for rise and decent time dwell in a shorter time in the entire valve event results in an improved valve area, the significant aspect in enhancement of engine's volumetric efficiency and the torque.

2.2.2 Variable Compression Ratio

The spark ignition engines are designed to have optimal performance at full load, whereas most of the time it has to be operated, at part load. Since the conventional SI engine has fixed compression ratio, its performance is compromised at part load. The effective compression ratio of the engine reduces as a result of less air-fuel mixture intake, consequently its combustion efficiency declines [19]. Thus, to achieve the best engine performance, the compression ratio is required to be variable. Therefore, variable compression ratio (VCR) is the most promising variable which influences the performance/thermal efficiency of the SI engine in terms of fuel consumption [3, 53]. The thermal efficiency of the conventional Otto cycle engine can be expressed as:

$$\eta_{otto} = 1 - \frac{1}{r_c^{\gamma-1}} \tag{2.1}$$

where r_c and γ are the compression ratio and ratio of heat capacity respectively. The simulation diagram of the thermal efficiency is shown in Figure 2.3.

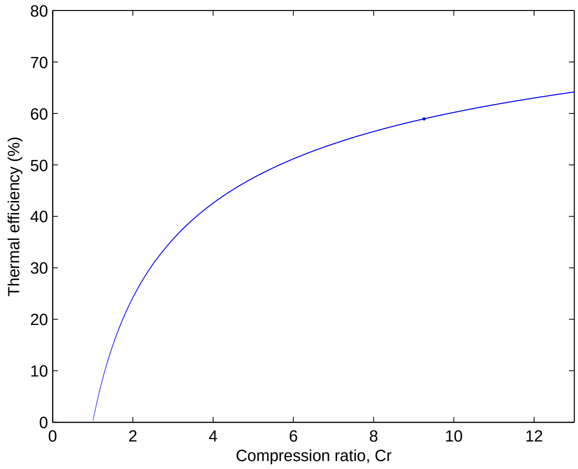

FIGURE 2.3: Thermal efficiency of the conventional Otto cycle engine [3].

But the compression ratio cannot be increased beyond the knocking limit of the fuel. One preference of increasing the compression ratio is the replacement of the fuel. With the increase of compression ratio the heat transfer also increases, however according to Gerty et al. [54], due to the thermodynamic effect of over-expansion, the improvement in the thermal efficiency can be realized.

The compression ratio of the engine may be varied by means of variation in the working conditions for instance load decreased by throttling which allows the compression ratio to increase, whereas low compression ratio is allowed at full load without the knocking problem. The adjustment of the compression ratio according to the working conditions plays a pivotal role to improve the power output as well as the efficiency of the engine. By designing the several engine components the variable compression ratio systems of the IC engine can be developed. The various techniques executing the VCR adjustment systems are reviewed [5, 28] and figured out as follows:

1. Moveable cylinder block
2. Head design variation
3. Piston geometry variation
4. Eccentric crank shaft bearing
5. Eccentric con rod bearing/ length
6. Force transmission by rack gear
7. Adjustable pivot point for con rod etc.

The summary of the VCR mechanisms is depicted in Figure 2.4.

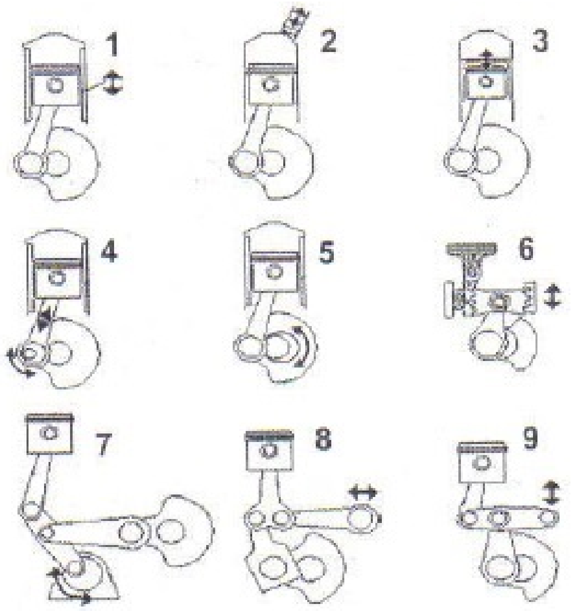

FIGURE 2.4: Variable Compression Ratio variants [5].

2.2.3 Over-Expansion

The engine having expansion/power stroke longer than the compression stroke is known as an over-expanded engine. The PV-diagram of the theoretical over-expanded engine cycle is as shown in Figure 1.2. As it can be seen that its compression is starting from point 1 instead of the BDC (Point 5) as in Otto cycle, which makes the compression shorter than the expansion as it is made in the same way as in the Otto cycle, by using all stroke length. The over-expanded cycle can be realized through various techniques, for instance it can be attained by crank-train mechanism (with a different arrangement of the traditional crankshaft/connecting rods arrangements), the over-expansion was achieved through this way [42].

However, the over-expanded cycle can also be accomplished with variable valve timing [55] by using the conventional crankshaft rod configuration. In this strategy, the intake valve is kept open for longer duration than the conventional valve timing depending on load conditions [56]. As a consequence, some of the inducted charge flows back into the intake manifold, which results in increasing the manifold pressure. Therefore, it reduces the throttling effect and plays vibrant role in reduction of pumping losses at part load. The attaining of good part load fuel economy in addition to avoiding the engine knocking, through reducing the effective in-cylinder compression ratio by utilizing the high geometric compression ratio/ expansion ratio, usually around 14:1, are main intentions of the LIVC strategy.

Toyota Prius model introduced in the market is fitted with hybrid system, its propulsion system comprises of an electric motor and an IC engine linked through a planetary mechanism [57]. The IC engine has 1497cc displacement and 13.5: 1 as its expansion ratio, whereas the effective compression ratio is limited from 4.8:1 to 9.3:1 by means of LIVC system and its intake valve closing between 80^o to 120^o after BDC. The engine speed is also lowered in order to minimize the friction losses. Moreover, the reduction of engine vibration during start and stop has also been accomplished with LIVC scheme. Another, approach for appreciating the over-expansion is through crankshaft offset as shown in Figure 2.5. With this strategy, the piston speed is reduced at TDC, managing the combustion near to constant volume and reducing the engine friction as well [58].

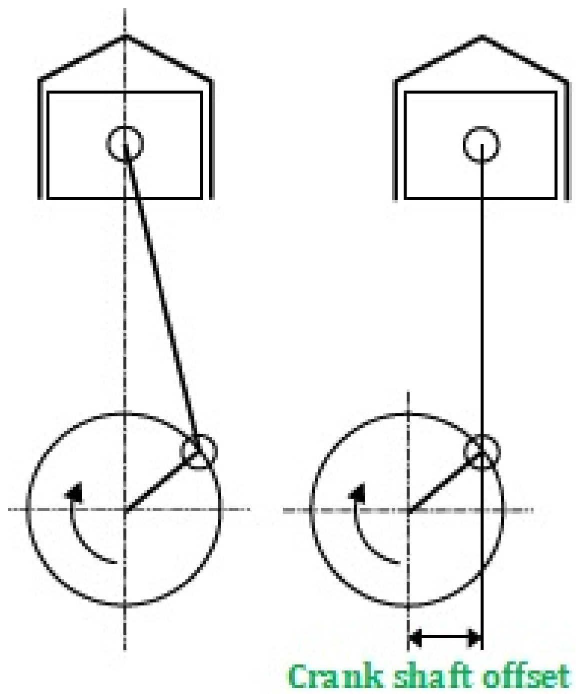

FIGURE 2.5: Crankshaft offset [1].

An over-expansion cycle engine is often mentioned as an Atkinson cycle engine [55]. James Atkinson (1846-1914) invented the Atkinson cycle and characterized by different lengths between expansion/exhaust and intake/compression strokes.

However, due to recent technological advancements a modern Atkinson cycle engine can be realized through the innovative mechanisms with the same crankshaft design as for the Otto cycle engine [55, 59, 60].

2.2.4 Engine Downsizing

The improvement in fuel consumption and the decrease in CO_2 emission can be accomplished with the reduction of engine size by keeping nearly the same engine performance, the most robust technique of economizing is termed as engine downsizing. It can be attained by several technologies [61], including the controlled auto-ignition (CAI), switching from four-strokes to two-strokes, boosting and biofuels. With this technique, the part load operating point shifts to the higher specific loads with decrease of engine displacement. Consequently, the engine performance enhances significantly because of decrease in pumping losses, upturn in effective compression ratio and the friction mean effective pressure also lessens. Furthermore, with less thermal inertia, fewer energy and time is required to warm up the cold starts engine, and due to the smaller engine bore, the flame propagation distance reduces and the engine knocking suppression stability improves. Fuel proficient downsizing concepts can also be accomplished by the application of variable compression ratio [5]. The Atkinson cycle is also one of these engine downsizing technologies. However, to enhance efficacy of the downsizing idea, the use of direct injection of fuel technology is more imperative instead of the port injection of fuel, subsequently increases the compression ratio.

2.2.5 Variable Stroke Engine

The Variable Stroke Engine (VSE) is an engine capable of variation in its displacement, by keeping the compression ratio same. Various VSE mechanisms are presented to perform this engine thought [62–65]. Alsterfalk et al. [66] proposed a stroke variation system to control engine load, to eliminate the need of throttle. The engine always operates at WOT, during the entire operating range, so the pumping losses almost constant and negligible. Moreover, with this technique, 33% reduction in NO emissions and 20% improvement in the brake specific fuel consumptions (**bsfc**) at typical loads can be achieved. Though, there are some limitations of the VSE, like the requirement of additional throttle valve for idling, engine braking and low load operations. Furthermore, less inner cylinder turbulences are added to the geometrical proportions of the combustion chamber, due to the short strokes, which slow down the fuel burning rate subsequently.

So far, various developed innovated mechanisms, their pros and cons, required to be integrated in the conventional spark ignition IC engine, to resolve the compromised solutions concerns and augment the performance throughout the operating conditions, have been described comprehensively. With the fast growth in automotive industry, the automotive researchers have tried hard to develop methods and employ various innovative technologies to accomplish fuel economy enhancement and emission reductions of the vehicles. For these intentions the comprehensive modeling and simulations of the IC engine, and subsequently of the vehicle powertrain, have been of prodigious significance for the performance evaluation [67]. It enables the investigative studies to be conducted with low cost, in short time and in the designing/testing of different robust controllers, to realize aforesaid intents such as declining of fuel consumption and noxious exhaust gases. With this perspective, various approaches of IC engine modeling have been presented since 1970. In the following section, literature available on the SI engine modeling is revealed comprehensively.

2.3 Modeling of Internal Combustion SI Engine

A spark ignition IC engine is extremely complex nonlinear system comprising of several sub systems for instance, air dynamics, fuel dynamics, thermodynamic and chemical phenomenon of combustion, engine inertial and rotational dynamic subsystems and pollution formation subsystem etc. and has been modeled in several ways established on the level of the model complexity. To reveal the engine's phenomenon through establishing cause and effect, dynamic relations between its essential inputs and outputs are the foremost emphasis in the engine modeling. The dynamic relations are the differential equations attained from the laws of conservation of mass and energy. The key challenge in engine modeling is to find the relation between its input and output variables that best describe the engine model and prophesy the output throughout the operating range of the engine [68] and schematic diagram of possible inputs and outputs of MVEM model is as presented in Figure 2.6.

Based on the reciprocating nature, the SI engine can be modeled under Discrete Event Models (DEM) and if the discrete cycles of the engine are neglected and all the engine processes are considered as of continuous nature, then the approach of modeling is known as Mean Value Engine Modeling (MVEM). However, the assortment of the model depends on the control purposes, for instance for mis-fire detection DEMs are most commonly preferred, whereas MVEM models are used for the control of slow engine processes [69].

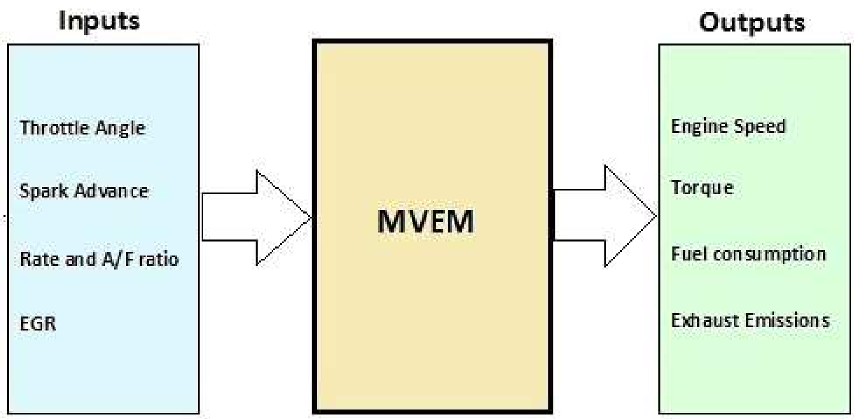

Figure 2.6: Schematic diagram of possible Inputs and Outputs of MVEM.

There are three significant spark ignition IC engine modeling approaches that are well renowned in the public literature [36, 37, 40, 70–74]. The engine models developed from experimental data either in the form of dynamic models, fitting polynomials of engine variables or phenomenological models heavily fitted with experimentally determined constants and augmented with empirical relations subject to be specific for an engine under study. Whereas, modeling approach based on first principle/phenomenological basis yields a reasonably accurate and generic model. The engine modeling methodology with physical meaningful equations and parameters is commonly named as physics-based modeling and it gives the detailed understanding of the operations as well as its components. Tracking the effects of the physical meaningful parameters on the system operation is the foremost advantage of using physics-based models. For illustration, various method of IC engine modeling are shown in the schematic Figure 2.7, it comprises of progressively more physical parameters as approaching from left to right. The first

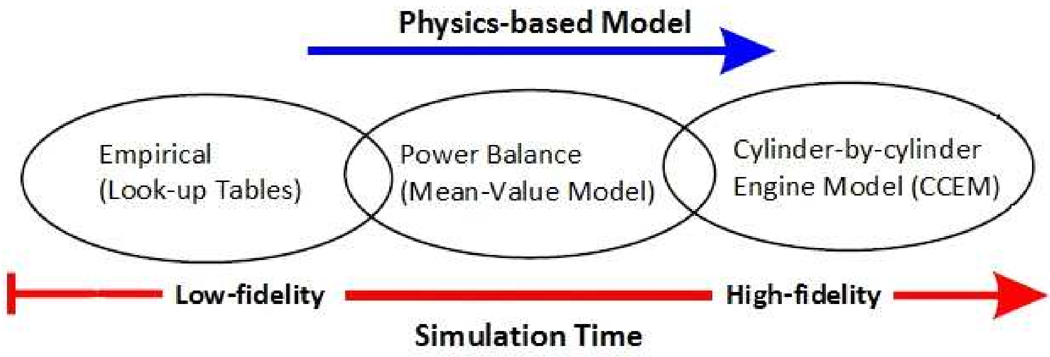

FIGURE 2.7: Different physics-based IC engine modeling approaches [6].

technique represents the empirical methods, wherein experimental data is used to develop and simulate the engine model and this approach of modeling is commonly described with a group of look-up tables. Second engine modeling approach is mean value engine modeling (MVEM) comprising of the physics-based equations (e.g. laws of mass and energy conservations) and the empirical data (e.g. volumetric efficiency). MVEM model uses time scale domain and time scales in MVEM are in excess of a single engine cycle and less than a time required to warm up a cold engine. The third methodology is the cylinder-by-cylinder engine modeling (CCEM). These models are more physics-based and are more precise than the MVEM models and proficient of capturing the details of the engine events, for instance in-cylinder (individual) pressure and temperature variations. CCEM models are usually employed to evaluate fuel injection, engine diagnostics, emissions as well as fuel consumptions. As CCEM model uses a crank angle domain, so the parameters like area, volume, temperature, pressure of the combustion cylinder besides the fuel mass burning rate, valve lift and the power of the engine can be found and associated to a specified crankshaft angle.

In the following subsections of literature review, different modeling and simulation approaches of SI conventional Otto cycle internal combustion engine are presented thoroughly. The reviews in these sections are categorized into two modeling approaches. The leading approach is the mean value engine modeling and the subsequent is the cylinder by cylinder engine modeling approach of the SI engine. Later on, the critic view point on the Atkinson cycle engine modeling and the prime objective of this research will be discussed comprehensively.

2.3.1 Mean Value Engine Modeling of the Spark Ignition IC Engines

Mean value engine models (MVEM) are intermediate level IC engine models which comprise of more physical details than simplistic linear transfer function/ Empirical (Look-up table) models, but considerably simpler models than large complex cylinder by cylinder engine models [6, 71, 75]. MVEMs are low frequency models neglect the discrete events of the engine and describe the dynamics of engine with limited bandwidth, likewise to considering the mean/average behavior of state variables over few engine cycles. The operating time scale is supposed longer than the engine cycle in this modeling approach. The MVEM has capability to capture the dynamics of main engine components and their dynamics can be mathematically formulated. It can predict an engine's foremost external variables such as crankshaft speed and manifold pressure, and significant internal variables, such as volumetric and thermal efficiencies. Typically, the differential equations are used to represent the fuel film flow, manifold pressure, and crankshaft speed in the MVEM models.

The MVEM models have been well-known as a suitable system model for the model based control applications and have been broadly used for the engine control strategy development [37]. The combination of the physics based components, which allow the physical parameter effects to be evaluated, controlled and look-up table models, with less computational effort, make the MVEM models the most appropriate for the control applications [36, 38, 40, 72, 76]. The schematic diagram of the MVEM modeling interfaces is presented in Figure 2.8.

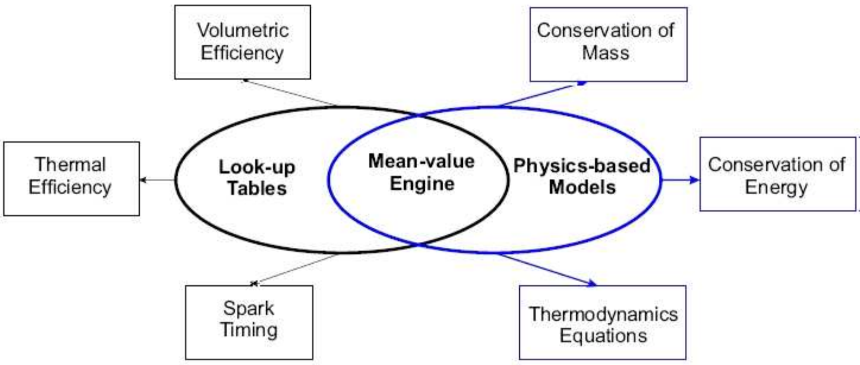

FIGURE 2.8: Schematic diagram of the MVEM modeling interfaces [6].

Usually, IC engine models which are expressed by empirical data for instance look-up tables, represent the engine characteristics variables as a function of engine rotational speeds and torques or through static math-based models [77, 78]. These IC engine models could not capture the transient phenomena.

Since 1970, the dynamic math-based models of IC engine have been studied and have been considerably expedient to study the transient occurrences for instance; manifold pressure, fuel dynamics, air-fuel ratio variations, temperature changes and effects of throttle angles [7, 79]. Later on, with the advances in computer technology and further restrictions on fuel emissions have led to engine dynamic models fetching a universal technique for engine modeling.

Dobner and Fruechte [7, 36] introduced a dynamic mathematical model with the carburetor model for IC engine. It was a generic model, only with few changes in input parameters; it can be used for other engines as well. The carburetor model, which is used to control the air and fuel flow into the engine, is described using empirical (look-up tables) as a function of the throttle angle, the air-fuel ratio and the manifold pressure. The model's dynamics was presented by time delays integration of dynamic equations. Dobner's model was a discrete model and its computations were done per each engine firing. The schematic diagram of the Dobner's model is illustrated in Figure 2.9.

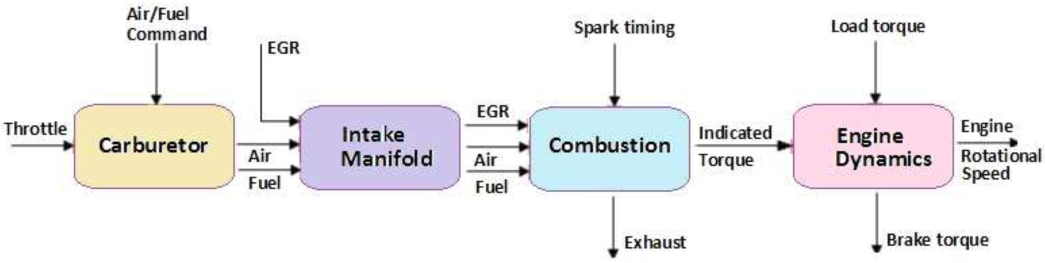

FIGURE 2.9: Schematic diagram of the IC engine model [7].

Aquino [80] developed a math-based fuel dynamics model by using volumetric efficiency and engine speed. He was among the first researchers who presented fuel dynamics model and it was a continuous flow model contrasting Dobner's model. Aquino presented the wall-wetting phenomena with assumptions that a fraction of the injected fuel is left on the port walls and evaporated after some time delay and mixed with incoming air and flows into the combustion cylinder. In Aquino's model the fuel dynamics variables are expressed as a function of manifold pressure and engine speed. The model can be used for other engine as well through adjustment of portion of injected fuel parameters and time constant.

Powell [81] introduced a rudimentary nonlinear model of dynamics system of the IC engine, which comprises of engine dynamics such as throttle body dynamics, intake manifold dynamics, fuel injection dynamics and exhaust gas-recirculation (EGR) system. He used a regression model for mass flow rate as a function of manifold pressure and engine speed, instead of using volumetric efficiency for intake air mass calculations which has significance in transient conditions. Powell fitted a second degree polynomial for throttle angle function and obtained throttle mass flow from an expression for pressure ratio at choked and non-choked conditions. He also used another regression equation as a function of speed to obtain engine output torque.

Crossley and Cook [39] developed a nonlinear mathematical engine model and employed a methodology alike to Powell's however replaced the throttle angle approximated function by a third degree polynomial. They also used engine output torque as a function of spark angle (SA), Air-fuel ratio (AF), throttle mass flow, engine speed and engine mass flow variables. In the model throttle body and inlet air flow, engine pumping and torque generation were modeled as nonlinear algebraic relations established on empirical data, while intake/exhaust manifold and rotational dynamics were modeled using differential equations. However, fuel dynamics were not conferred besides graphical simulation software was not employed.

Yuen and Servati [82] introduced a math-based engine model with analogous approach used in [7], by using a carburetor instead of throttle body. To make the model more generic, they used a normalized function for pressure as a substitute of throttle area characteristics and one more normalized function by using Harington and Bolt's [79]] estimation technique. In this model the intake manifold temperature variations and their effects on the fuel evaporation and manifold pressure in transient conditions have been considered unlike Dobner's. The emission gases for instance CO and NO_x are formulated as a function of throttle mass flow and A/F ratio of the combustion mixture. The air mass flow is diligently related to the throttle area characteristics and geometry.

Moskwa [70] presented throttle body area as a function of throttle pin diameter, throttle angle and throttle bore. In the model, the output engine torque expressed in terms of SA, A/F ratio and friction look-up tables. He used manifold pressure, compression ratio and engine cylinder geometries to describe the thermal energy of the IC engine.

Hendricks and Sorenson [71] introduced the most well-known description of the MVEM in 1990 for the first time and described in additional details in [75]. The four cylinders SI engine model consists of three main states (three differential equations): the air mass flow rate, manifold pressure and engine rotational speed. In the model, they used the main state variables to express engine instantaneous variables (parametric equations) such as look-up data as a function of the manifold pressure and the engine speed have been used to describe the volumetric efficiency and the thermal efficiency. Hendricks and Sorenson developed fuel dynamics model, which is based on the fuel flow in the intake manifold unlike Aquino's fuel-mass based calculations. A realistic agreement with the experimental data throughout the entire operating range of the SI engine, with the accuracy of most of the variables within 2% of the experimental results is shown by the simulation results. The major developments of the MVEM have been accomplished by Hendricks since 1990 [83, 84].

Adibi Asl et al. [67] described the significance of the spark timing (ignition timing) through the thermal efficiency in the MVEM proposed by Saeedi [85]. According to Adibi et al. the thermal efficiency of the IC engine is a product of the four thermal efficiencies such as engine speed, the air fuel ratio, manifold pressure and the spark timing angle contributions, and then the spark timing has a pivotal role to enhance the performance. He suggested that at higher engine speed, the advanced spark angle is more advanced, to attain better combustion and to retain the combustion duration in a reasonable range as at higher speed the piston velocity increases, consequently the time duration of one of the stroke process reduces.

Another significant physics-based engine modeling scheme known as cylinder-by-cylinder engine modeling (CCEM), has been presented in the following subsection which is more proficient in capturing the details of the engine events, such as in-cylinder pressure and temperature variations [76]. CCEM models are usually employed to evaluate the engine performance.

2.3.2 Cylinder by Cylinder SI Engine Modeling

The cylinder-by-cylinder engine (CCEM) models are more physics-based and are more precise than the MVEM models and capable of capturing the inclusive dynamics of the engine events. CCEM models foresee the instantaneous torque and in-cylinder peak pressure at a given crank angle, whereas control can only affect the plant on per-cycle basis. In this IC engine modeling technique, the cycle average torque is computed by integrating the net integrated torque over the engine cycle [35].

To predict the engine performances, for instance torque and power, a mathematical modeling of the spark ignition engine with cylinder-by-cylinder modeling technique has a great significance. With the models for each of the four operations/processes of the conventional Otto cycle engine as shown in Figure 2.1, a complete cycle simulation can be built up and investigated. The characteristics of each process can be described by these ideal models [42].

Sitthiracha et al. [34] developed a mathematical model of Spark Ignition Engine with cylinder-by-cylinder method by combining both the physical and empirical formulations. The objective of this study was to investigate the engine performances; power and torque by integrating the in-cylinder pressure over one engine cycle and the model has significance for improvement of engine efficiency as well. The research also emphasize on prediction of the volumetric efficiency in order to minimize its dependence on testing data by using zero dimension model. Later on, the proposed model was validated with the data from eight different engine models. The model was comprising of set of tuning parameters such as physical geometry of engine, air/fuel ratio, ignition advanced, heat transfer etc. and it can capture torque and power characteristics verses speed accurately. Furthermore, it was used for simulations in order to evaluate the burning duration of the alternative fuels and studied its effects on volumetric efficiency. This CCEM model was based on Otto cycle, wherein valve timing and lift have been lumped in the discharge coefficient to improve the engine performance. However, for analysis and simulation of complete engine cycle, exact cycle's states data will be needed which is hardly possible and throttling effect cannot be reduced considerably.

To develop and demonstrate a control technique to estimate cylinder charge online and used that data to update a feed forward map between charge demand, valve lift and duration. Nevertheless, the exhaust process dynamics and effect of combustion on cylinder pressure was not included in this model. Lawrence et al. [86] proposed a more complete cylinder-by-cylinder model which includes the exhaust valve, manifold and breathing process and in-cylinder combustion process as well.

Mianzo and Peng [35] developed a 24 states CCEM model of the VVT engine which includes manifold mass, pressure, temperature, burned gas residual, combustion as well as the exhaust process dynamics. As CCEM predicts instantaneous torque at a given crank angle and control can only effect the plant on per-cycle basis, therefore cycle-averaged mapping between torque and intake valve timing for the engine operating conditions has been obtained to make it appropriate for control applications.

Karlsson and Fredriksson [76] offered an interesting comparison of both the physics based modeling approaches i.e. mean value engine MVEM model and cylinder-by-cylinder engine CCEM model. Both models were based on the spark ignition conventional Otto cycle engine and comparison was based on their possible application in model based power train control design and their utility. The need of the engine model complexity for use in power-trian control applications was foremost objective of this investigation. The studies [36, 71, 83, 87] concluded that complexity of the model depends on its utilization and MVEM model is sufficient for nonlinear engine control and state estimation as same dynamic response of both MEVM and CCEM models come out in simulations. However, for the study of backlash phenomena in power train these models could not be found to be reasonably accurate. In dynamical equations form, complete description of both MVEM and CCEM models was not given in this research. For both the models, sub models consist of induction manifold as well as throttle body and the engine inertia/rotational dynamics were described. For both MVEM model in time domain and CCEM model in crank angle domain delays peculiar to the nature of four stroke reciprocating machines were modeled. The equations of power train model were not given and driveline/power train model was taken from a previous work of the author(s). The simulation results point towards the conclusion that MVEM model was a better option for power train controller design, as it was not accurate in the study of backlash phenomenon. However, a CCEM model was suggested as a better candidate for backlash in drive-line study.

However, almost all of the past efforts have exclusively considered modeling of conventional Otto cycle based SI engine which has compromised performance throughout the entire engine operation as a consequence of fixed valve timing, thermal efficiency confines and throttling effect, particularly at part load operation. In the following section inclusive research conducted on the Atkinson/over-expanded cycle engine is presented.

2.4 Research on the Atkinson Cycle Engine

The striking dissimilarity of the Atkinson cycle engine in comparison with the Otto cycle engine lies in the fact that its expansion (power) stroke is longer than its compression stroke. Consequently, it delivers more energy from combusted gas to piston throughout the over-expanded power stroke. Whereas, the prospective advantages of Atkinson cycle are accomplished by compromising the power density, precisely at low-speed [59]. To implement a relatively longer expansion stroke in contrast to the compression stroke with the crankshaft's necessities, intricate mechanical design and piston linkage, the innovative Atkinson cycle engine is barely ever used [55]. On the other hand, a modern Atkinson cycle engine can be attained with VVT by using the similar crankshaft scheme as for the Otto cycle engine owing to the recent technological developments [59, 60]. With optimal engine intake and exhaust valves timing, the primary characteristic of Atkinson cycle engine, the longer expansion ratio than compression ratio, can be realized.

In recent times, variable valve timing engines have gained numerous attentions due to their capability of controlling the valve events independent of crank shaft rotation, reducing the pumping losses particularly at part load operation and increasing the torque performance over wider range in comparison with the conventional SI engine. The VVT has implication in controlling the EGR, through the control of valve overlap, consequently in controlling the NO_x emissions. To realize the benefits of the VVT engines, many studies [25, 45, 88] has been accomplished and several VVT systems [89, 90] have been proposed. The automotive researchers have also focused on the control oriented modeling of the VVT engines. In this regards, Ashhab et al. [91] developed a cylinder-by-cylinder model of the breathing dynamics of the camless engine and later on, introduced a cycle average model between valve timing and lift and mean- valued cylinder charge. Moreover, Ashhab et al. has also used the cycle average model.

By using the innovative mechanisms, optimum efficiency of the Spark Ignition Engine can be accomplished during the entire engine range, particularly at part load. With the late intake valve closing (LIVC) load handling strategy, the phenomena of back-flow of air-fuel mixture contributes in increasing the manifold pressure; subsequently plays a vibrant role in the reduction of throttling effect. Thus reduction in the pumping losses and improvement in volumetric efficiency can be attained. Besides, the enhancement in the thermal efficiency can be accomplished through over-expanded cycle achieved as a result of LIVC. According to Martins et al. [1] an enormous quantity of high pressure gas is freed to the atmosphere, without any useful work, when the exhaust valve of the conventional SI engine opens at the end of expansion stroke. An engine might have better thermal efficiency by using this lost energy as well.

In this context, several investigations have been worked out. The research in [4, 16, 18, 19, 22, 53, 92] elaborates comprehensively the significance of Variable Valve Actuation (VVA) and VCR on the SI engine performance, specifically benefits of the late intake valve closing to eliminate the throttling effect. Besides the studies [1, 55, 59, 60] investigated the thermal efficiencies of various engine cycles and figured out the over-expanded cycle, partaking the optimal thermal efficiency.

The authors in [19], emphasized on investigating the advantages of using a continuously variable valve timing and lift (VEL) system and VCR on fuel economy, power output, emission decline and improvements on deterioration of combustion, the inherent disadvantage of early intake valve closing. The configuration of VEL system, variable compression ratio mechanism, its issues and effects of early intake valve closing over-expanded cycle are presented comprehensively. The VCR engine used for this study is based on 2.0 liter inline four-cylinder and experimental results shows that 4.5% fuel economy is achieved due to VCR mechanism alone and 14.4% as a result of VCR in combination with variable valve timing (VEL) system.

In [55], the analysis of the Atkinson cycle engine with different expansion ratios is revealed thoroughly in-comparison with benchmark having same expansion and compression ratios. Its objective is to comprehend the characteristics of over expansion and its affect on performance regarding fuel economy and emissions (in-cylinder CO and NOx concentration). By using fully variable valve lift and timing system, 13% augmentation in the fuel economy have been achieved. In this research the main drawback, the deterioration of combustion process due to reduction of in-cylinder pressure and temperature caused by LIVC strategy have not been addressed.

Martins et al. [1] has addressed the part load concerns of the Otto cycle comprehensively and enlightened the dependance of its thermal efficiency on engine operating conditions. The authors have derived and compared the expressions for the thermal efficiency of various engine cycles for instance Otto cycle, Otto cycle with Charge Injection (Stratified charge), Diesel cycle and Dual cycle at part load etc. and concluded that Atkinson cycle is the most efficient SI engine working cycle at part load. The comprehensive study based on over-expanded cycle has been carried out by means of CCEM model, wherein flexible VVA load controlling strategy has been employed, theoretically and experimentally, using different cams having different dwell angles [10]. Around 6% fuel consumption of the VCR Over-expanded engine cycle using CCEM model over the Otto cycle engine has evaluated using standard New European Driving Cycle (NEDC) [93]. The specific fuel consumption (SFC) of this engine cycle, realized by VVT as well as VCR mechanism has been improved around 19% over the conventional Otto cycle SI engine, at medium and higher loads.

2.5 Conclusion

In the so far research, most of the researchers [59, 60, 88, 94, 95] have focused on how Atkinson cycle can be accomplished with different innovative mechanisms and have been devised various strategies to resolve its issues such as deterioration of the combustion efficiency etc. However, reasonable studies have also been reported on cylinder-by-cylinder Atkinson cycle engine modeling and in the perspective of the Atkinson cycle engine fuel economy and emission lessening using CCEM model. It should however be noted that cylinder-by-cylinder models suffer from number of limitations such as complexity, difficulty in analysis and control design etc.

In consequence of the inherent limitations of cylinder by cylinder models, MVEM model is one of the most suitable system's model for analysis and engine control strategies development [36, 38, 40, 72, 76]. Besides, the control-oriented (MVEM) model is anticipated for systematic analysis, simulations, control based applications and estimation strategies to support and enable novel load control strategies as well as improve fuel economy of the Atkinson cycle SI engine used in hybrid electric vehicles. "However, MVEM models of the SI engine present in the literature, have not been incorporated VVT or Atkinson cycle".

Hence, the part load concerns of the conventional engine, realization of the potential advantages and control design aspects of an unconventional Atkinson cycle VVT engine used in HEVs, aspire toward the development of a physically motivated control-oriented model i.e. extended mean value engine model (EMVEM) of the Atkinson cycle engine. Wherein, the innovative mechanisms have to be incorporated. The proposed control-oriented (EMVEM) model of an Atkinson cycle VVT engine has great implications on the reduction of throttling effect, augmentation in the thermal efficiency, fuel economy as well as control design, is presented in the succeeding chapter.

Chapter 3

Mathematical Modeling of Atkinson Cycle Engine

Things of this world cannot be made known without Mathematics.
Roger Bacon (1220-1292), Opus Majus, Transl. R. Burke 1928.

This chapter is dedicated to the development of a novel mathematical model of the Atkinson cycle engine having leading applications in hybrid electric vehicles (HEVs). It also comprises of the brief background details required for understanding the automotive engine from a dynamical system perspective. Owing to the advancement in the innovated technologies, the performance compromising aspects of spark ignition (SI) engine can be coped and its optimum efficiency can be appreciated during the entire engine operating range. In this framework, a novel control-oriented extended mean value engine model (EMVEM) of the Atkinson cycle engine is proposed, wherein the Atkinson cycle, variable valve timing (VVT), over-expansion, and variable compression ratio (VCR) characteristics are incorporated. With these intents, an intake valve timing (IVT) parameter is introduced, which has a pivotal role in modeling the inclusive dynamics of the VVT engine system and to deal with engine performance degrading features. The control-oriented Atkinson cycle engine dynamics modeling is appreciated comprehensively in Section 3.1. Whereas, summary of the EMVEM model along with EMVEM model parameters description and nominal values, are given in Section 3.2. In Section 3.3, the proposed EMVEM model is validated with the experimental data of a VVT engine, obtained from literature, to ensure that the proposed model has the capability to capture the dynamics of the Atkinson cycle engine, and engine load can be controlled by an alternative input i.e. IVT parameter, instead of the conventional throttle. The potential benefits of the developed model with its intrinsic late intake valve closing (LIVC) strategy and copious integrated characteristics in perspective of reduction in throttling effect as well enhancement in thermal efficiency, besides the constraints on LIVC parameter, are exhibited in Section 3.4. These advantages have a great significance in the engine performance improvements, fuel economy, and emissions reduction. Section 3.5 concludes the modeling work and highlight the significance of the proposed model in the control design and fuel economy point of view.

3.1 Atkinson Cycle Engine Dynamics Modeling

Model based control strategies heavily rely on the model accuracy of the dynamical system [3, 40, 69]. To cope with the intrinsic shortcomings of the conventional Otto cycle engine such as throttling effect, fixed valve timings and the constraints on thermal efficiency, the innovated mechanisms [4, 16, 18, 22, 60, 92, 93] have to be employed having vital role in the development of a more realistic mean value engine model of the Atkinson cycle engine [19, 38, 53, 55]. With the unification of these systems the actual engine behavior can be produced from MVEM, based on ideal physical laws for the entire engine operating range.

The modeled dynamics comprise of key modeling processes, such as air dynamics (inlet air path), which is further divided into throttle body, manifold dynamics with VVT incorporated to realize Atkinson cycle, induction of air into the engine cylinders, combustion and rotational dynamics based on the following assumptions.

- In the throttle body one dimensional steady compressible flow is considered [40].
- For capturing the air dynamics in the intake manifold filling and emptying approach is employed [37], [42].

Moreover, for the analysis and modeling of in-cylinder dynamics Atkinson cycle is exploited [3]. Whereas, the calculation of engine torque and rotational dynamics is based on Newton's law [3, 42, 96].

The dynamics of the Atkinson cycle VVT engine that consists of manifold pressure and rotational speed is described in-detail as in the ensuing subsections.

3.1.1 Manifold Pressure Dynamics

The intake manifold pressure is responsible for delivering air to the engine through manifold to each cylinder. The intake manifold of the SI engine can be visualized in Figure 3.1. The part of the air intake system that starts after throttle body and finally connects to the piston, named as runners /receivers. These runners are supposed to have fixed volume for which the thermodynamic states (temperature, pressure etc.) remain same all the way through the volume (lumped parameter approach)[69]. The manifold pressure dynamics can be modeled by applying laws of conservation of mass and energy, ideal gas law and Dalton's principle of partial pressure [3, 40, 71]. The rate of change of air mass flow is the difference between the air mass flow flowing into the intake manifold and out of the manifold/ flowing into the combustion cylinder through intake valve in the conventional Otto cycle SI engine, can be mathematically described as [71]:

$$\frac{dm}{dt} = \dot{m}_i - \dot{m}_o \tag{3.1}$$

where,
m_i is the air flowing into the manifold
m_o is the air flowing into the combustion cylinder.

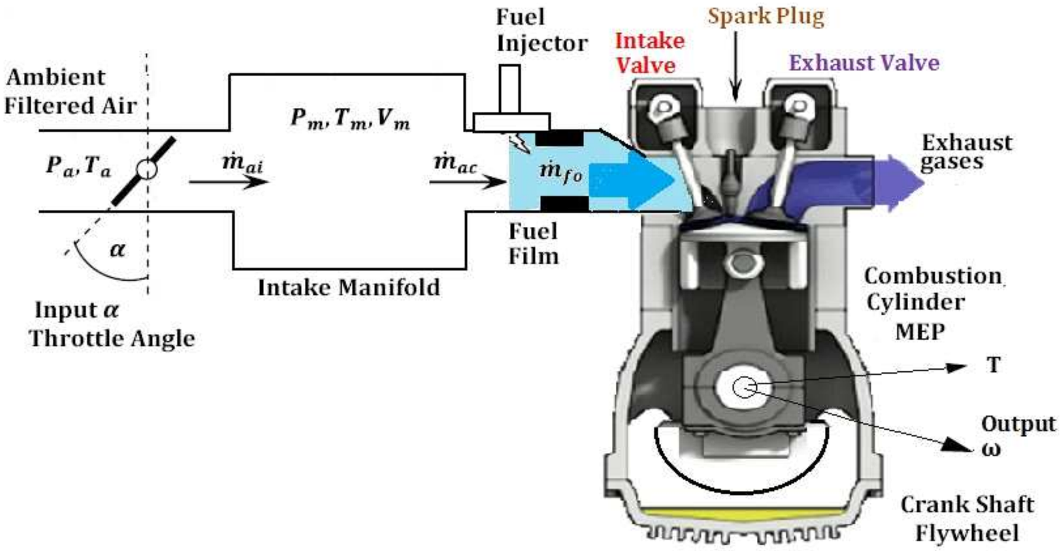

FIGURE 3.1: Diagram of components involved in the MVEM of the SI engine.

3.1.1.1 Incorporation of Variable Valve Timing (VVT) Mechanism

In the conventional IC engine the intake valve opens during the induction stroke and air-fuel mixture (charge) is admitted into the combustion chamber. The intake valve closes and the charge gets compressed during the compression stroke. However, the closing of intake valve is retarded (remains open for longer time) towards the end of the compression stroke in the Late Intake Valve Closing (LIVC) strategy. As a consequence, some of charge is expelled back into the intake manifold during the first part of compression stroke. The entrapped air-fuel mixture gets readmitted during the subsequent intake stroke, at the pressure slightly above that of the charge in conventional Otto cycle engine. This means suction pressure line deviates slightly the atmospheric pressure and less work is required to complete the intake stroke [4]. It plays a vital role in the reduction of the engine performance degrading aspect, enhancement of the engine thermal efficiency and lowers the fuel consumption, particularly at part load. The PV-diagram of an ideal Atkinson cycle engine as depicted in Figure 1.2 is shown here.

In the Atkinson cycle engine, the valve closing time is essentially dependent on the engine operating conditions and it can be accomplished by VVT system.

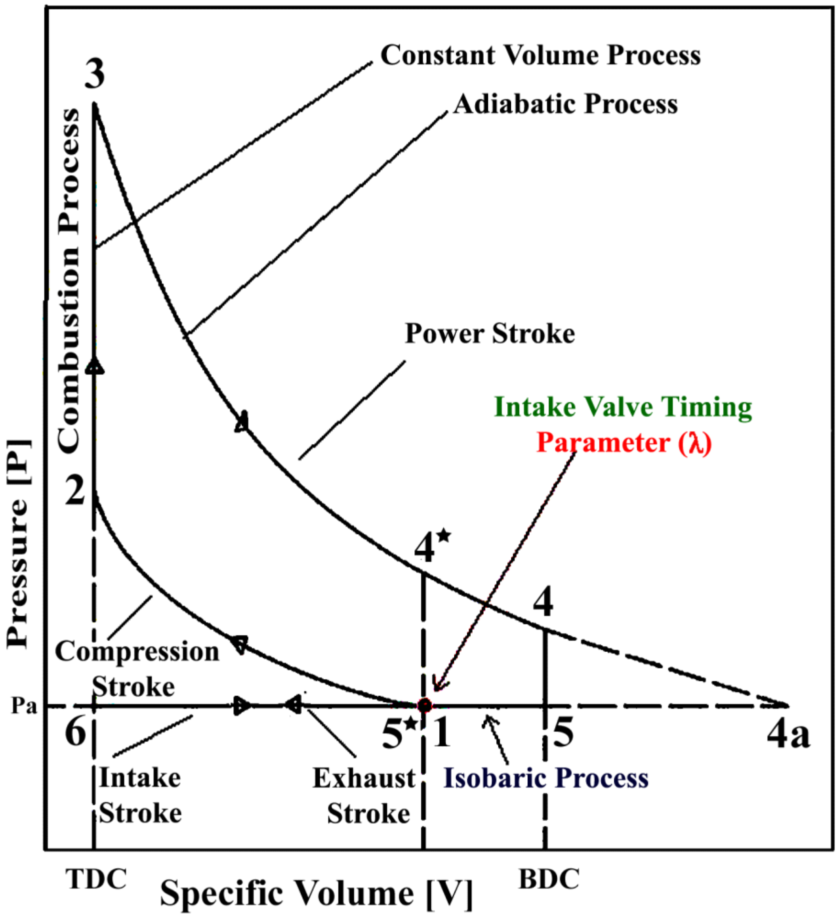

FIGURE 3.2: PV-diagram of an ideal Atkinson cycle engine [2, 3]

Intake Valve Timing Parameter (λ):

To realize the Atkinson cycle and incorporate variable valve actuation mechanism in the conventional SI engine model, a novel Intake Valve Timing (IVT) parameter "λ" is introduced, which is defined as a ratio of the expansion ratio r_e to the effective compression ratio r_c. It is used to describe the intake valve closing time. It has great significance to encapsulate the dynamics of the Atkinson cycle engine and furthermore, to accomplish the advantages of the variable compression ratio and over-expansion characteristics. Consequently, the fuel economy of the VVT engine used in HEVs can be enhanced. Mathematically, it can be expressed as:

$$\lambda = \frac{r_e}{r_c}, \quad \lambda \geq 1 \tag{3.2}$$

with,

$$\begin{aligned} r_e &= \frac{V_c + V_d}{V_c} \\ and & \\ r_c &= \frac{V_c + V_{ivc}}{V_c} \end{aligned}$$

where,

r_e is the expansion ratio

r_c is the effective compression ratio

$V_c = V_2$ is the clearance volume

$V_{ivc} = V_1$ is the displaced cylinder volume in accordance with the intake valve closing time as depicted in Figure 3.2 and is given as:

$$V_{ivc} = \frac{\pi B^2}{4}\left[r + a - \left\{acos\theta + \sqrt{r^2 - a^2 sin^2\theta}\right\}\right] \tag{3.3}$$

while,

B is the engine bore

a is crank offset

r is the connecting rod length

θ is engine crank angle.

Whereas, the engine displaced volume for one cylinder engine, V_d is defined as:

$$V_d = \frac{\pi}{4}B^2 S \tag{3.4}$$

while S describes the engine stroke.

By incorporating back flow notion, air-fuel mixture flows back into the intake manifold or at the same time becomes the intake of the other cylinder, during the first part of compression stroke. The suction pressure remains slightly above the atmospheric pressure, as a consequence of late intake valve closing (LIVC)[4]. At part load operating conditions, it plays a pivotal role in the reduction of throttling effect. With integration of the flexible intake valve actuation mechanism in the conventional SI engine in order to realize the Atkinson cycle, the air mass flow rate through the intake manifold can be expressed as [2]:

$$\frac{dm}{dt} = \dot{m}_i + (\lambda - 1)\dot{m}_o - \dot{m}_o \tag{3.5}$$

where the 2nd term $(\lambda - 1)\dot{m}_o$ is representing the back flow notion as a consequence of the late intake valve closing (load handling) strategy and it can become the part of intake for the subsequent engine cycle.
Therefore, the proposed air mass flow rate, with the VVA mechanism incorporated in the conventional SI engine, can be stated as:

$$\frac{dm}{dt} = \dot{m}_i - (2 - \lambda)\dot{m}_o \tag{3.6}$$

By applying ideal gas law to model the fluids [3], and assuming that the temperature of gases flowing in and flowing out from the manifold, remain the same [40]. The Eq (3.6) can be described as:

$$\frac{dP_m}{dt} = \frac{RT_m}{V_m}\left[\dot{m}_i - (2 - \lambda)\dot{m}_o\right] \tag{3.7}$$

where,
P_m is the Manifold pressure
V_m is the Manifold volume
R is the Specific gas constant
m is the air mass
T_m is the manifold temperature

3.1.1.2 Air Flow Across Throttle Body

The fluid flow between the two reservoirs is determined by valves and the basis of the flow of fluids at the inlet of throttle body (manifold) is one dimensional compressible flow. The intake manifold model can be developed by applying law of conservation of mass and energy, ideal gas and Dalton's principle of partial pressure [3, 38, 40, 71]. Using adiabatic flow process, isentropic process property and Bernoulli law [3, 40], the air/fluid mass flow rate, across the throttle body, can be determined as a function of atmospheric and manifold pressure

$$\dot{m}_i = A_e \rho_a \sqrt{2 C_p T_a (1 - \frac{P_m}{P_a})^{\frac{\gamma-1}{\gamma}}} \tag{3.8}$$

and

$$\rho_a = \frac{P_a}{R T_a} (\frac{P_m}{P_a})^{\frac{1}{\gamma}} \tag{3.9}$$

where,

C_p is the Specific heat at constant pressure

P_a is the Atmospheric pressure

A_e is the Throttle effective area

T_a is Ambient temperature

γ is ratio of heat capacities

ρ_a is density of fluid.

With $C_p = \frac{\gamma R}{\gamma - 1}$, the air mass flow rate across the throttle valve can be determined as:

$$\dot{m}_i = A_e P_a \sqrt{\frac{2\gamma}{R T_a (\gamma - 1)}} \sqrt{(\frac{P_m}{P_a})^{\frac{2}{\gamma}} - (\frac{P_m}{P_a})^{\frac{\gamma+1}{\gamma}}} \tag{3.10}$$

The air mass flow passing through the narrowest point is termed as choked flow and that can analytically be determined by critical pressure i.e., $P_m \leq P_c r$, where,

$$P_{cr} = P_a (\frac{2}{\gamma + 1})^{\frac{\gamma}{\gamma+1}} \tag{3.11}$$

For control, estimation and diagnosis objectives, the piecewise function in the Eq. (3.10) is not ideal as the flow reaches sonic conditions in the narrowest path at high pressure differences. It can be changed by a single non-switching nonlinear function $\chi(p)$ as follows [97]:

$$\dot{m}_i = A_e P_a \sqrt{\frac{\gamma}{R T_a} (\frac{2}{\gamma + 1})^{\frac{\gamma+1}{\gamma-1}}} \chi(p) \tag{3.12}$$

where,

$$\chi(p) \quad = \quad 1 - \exp\left(9\frac{P_m}{P_a} - 9\right) \tag{3.13}$$

To elaborate the approximation error of changing the piecewise functions by non-switching $\chi(p)$, piecewise functions from Eq (3.10) and non-switching $\chi(p)$ from Eq (3.13) are simulated as function of pressure ratio as shown in Figure 3.3. The

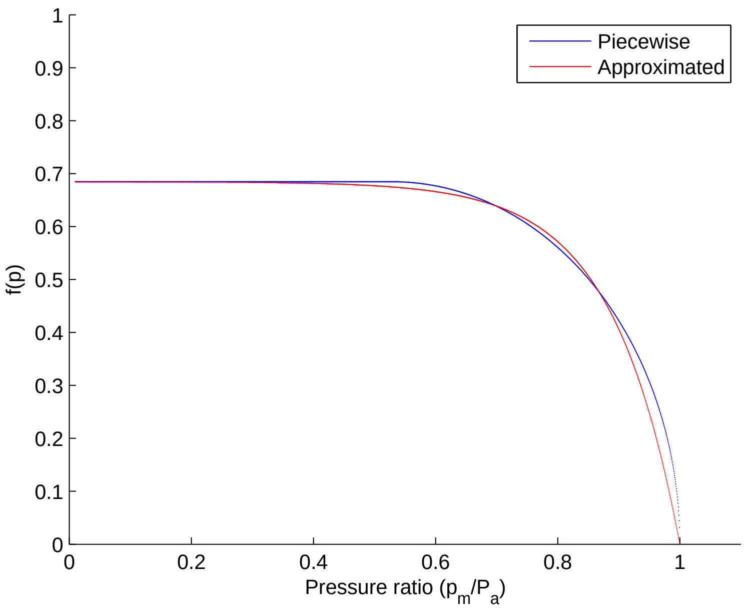

FIGURE 3.3: Comparison of the piecewise and non-switching $\chi(p)$ functions

simulation result exhibits that the replacement of piecewise function by $\chi(p)$ has no substantial effect.

Thus, Eq (3.12) is the inflow of air across the throttle body which is represented by the gas equation of isentropically compressible flow of air mass through an orifice under ideal conditions which are essentially hardly ever possible. It is purely theoretical and gives the ideal amount of the air mass flow. To accommodate the inaccuracies and assumptions, a coefficient of discharge defined as a ratio of actual to the ideal air mass flow rate is incorporated, which is used to measure the air flow restrictions as a consequence of throttle body. Therefore, the realistic throttle air mass flow rate is described as:

$$\dot{m}_i = C_D A_e P_a \sqrt{\frac{\gamma}{RT_a}(\frac{2}{\gamma+1})^{\frac{\gamma+1}{\gamma-1}}}\chi(p) \tag{3.14}$$

where C_D is throttle discharge coefficient and its effect can be analyzed through various approaches. Most of the researchers have been considered it as a constant parameter in the MVEM model under the steady state conditions [36, 71, 97, 98]. Whereas, it can also be taken as a varying parameter and several expressions have been developed that need engine other variables as an input such as manifold pressure, engine speed, throttle area, air temperature, etc [99]. Besides, C_D can be estimated by sliding mode observer as well [100, 101].

3.1.1.3 Throttle Effective Area

To control the amount of air flow required by SI engine, a butterfly valve called as throttle is mounted at the intake of air path (manifold). The state of the throttle valve is termed as Wide Open Throttle (WOT) when the pedal, connected to the throttle plate, is fully pressed, otherwise it is named as part open throttle. The area of the throttle pipe through which the air can pass is called the effective throttle area. To attain the desired effective area the throttle angle is either controlled through Bowden cable or by electric motor. One-dimensional compressible flow of air is assumed from ambient to past throttle. In the throttle body, this air flow can be considered as an isentropic flow as a result of its adiabatic and reversible nature The accuracy of the model of throttle effective area plays a vital role to measure the air flow into the engine cylinder. The effective throttle area in simple mathematical form can be expressed as [102]

$$A_e = \pi\frac{D^2}{4}(1-\cos\phi), \qquad 0 < \phi_o \leq \phi \leq 90^0 \tag{3.15}$$

where,

A_e is the throttle effective area.

$\pi\frac{D^2}{4}$ is throttle plate's cross-sectional area.

ϕ is the throttle angle.

D is inlet diameter.

ϕ_o is the minimum opening throttle angle required to keep the engine operational at a lowest speed termed as engine idle speed.

The precise model of throttle effective area can be presented in the form [97, 103].

$$A_e = \pi \frac{D^2}{4}(1 - \cos(\frac{(\phi + \phi_{cl})}{\phi_{cl}})) \tag{3.16}$$

where,

$\phi_{cl} = 9.8^0$ is the closed position throttle angle.

The assortment of the throttle effective area expression depends upon the application type and the required model accuracy.

3.1.1.4 Air Flow Across Engine

The air mass flowing into the cylinder is owing to the engine's reciprocating motion and which resembles with the reciprocating pump. By using speed density formula, the air mass flow entering into the combustion cylinder, through the fixed intake valve, can be described as [40, 42]:

$$\dot{m}_o = \eta_{vol} \beta_m P_m \tag{3.17}$$

where,

$\beta_m = \left(\frac{V_d}{RT_m}\right)\left(\frac{\omega_e}{4\pi}\right)$

ω_e is engine speed in rad/sec,

η_{vol} is the volumetric efficiency and is a look up table function of the engine speed and intake manifold pressure in fixed valve timing engines. Generally experimental setup is designed to determine the η_{vol} as a function of the manifold pressure and engine speed. The optimum air mass flow into the combustion cylinder depends on the volumetric efficiency η_{vol} and is the most promising parameter that governs the performance of engine throughout the entire operating range. According to

Pulkrabek [3], it can be mathematically expressed as

$$\eta_{vol} \quad = \quad \frac{\dot{m}_{actual}}{\dot{m}_{ideal}}$$

where,

$\dot{m}_{actual}$ is the flow of air mass leaving the manifold.

$\dot{m}_{ideal}$ is the flow of air mass leaving the manifold.

To measure the volumetric efficiency several attempts have been employed. It has been considered constant term [37], therefore ignoring the dynamics induced by η_{vol} while manipulating the throttle angle. It can also be calculated by modeling of the valve timings [97]. Typical values of the volumetric efficiency for SI engine at wide-open throttle (WOT) are in the range of 80 to 90 percent, going down to much lower values as throttle is closed [3, 42].

For VVT engines, this simple relation can no longer be used, as flexible VVT actuators radically affect the in-cylinder air mass. The optimum air mass flow into the combustion cylinder depends essentially on the volumetric efficiency (efficiency parameter) which is the most promising parameter that governs the performance of engine throughout the entire operating range. The realistic and optimal volumetric efficiency can be achieved by incorporating the variable valve timing. Based on engine geometrical parameters (using the positions of VVT actuators), the aspirated air mass in the cylinder can be expressed as [33]:

$$m_{asp} = \alpha(P_m, \omega_e)\frac{P_m V_{ivc}(\Theta_{int})}{RT_m} \tag{3.18}$$

where,

$\alpha(P_m, \omega_e)$ is a look up table function of the engine speed and intake manifold pressure.

V_{ivc} defines the volume of the combustion cylinder at intake valve closing (a function of intake valve timing) as described in Eq (3.3).

Therefore, using Eq (3.18) a model of the volumetric efficiency for the Atkinson cycle VVT engine can be derived as:

$$\eta_{vol}^{atk} = \frac{m_{asp}}{\frac{P_m V_d}{RT_m}} \tag{3.19}$$

Eventually, by substituting Eq (3.19) in Eq (3.17), the realistic and optimal air mass flow entering the engine's combustion cylinder through the intake valve for the VVT engine is:

$$\dot{m}_o = \eta_{vol}^{atk} \beta_m P_m \tag{3.20}$$

It can also be expressed as:

$$\dot{m}_o = \alpha(P_m, \omega_e) \frac{\rho_m \omega_e V_{ivc}}{4\pi} \tag{3.21}$$

where ρ_m is manifold air density.

3.1.1.5 Complete VVA Incorporated Manifold Pressure Dynamics

Finally, by substituting Eq (3.15) and Eq (3.21) in Eq (3.7), the complete variable valve actuation mechanism incorporated air dynamics of the Atkinson cycle VVT spark ignition engine can be described as:

$$\begin{aligned} \frac{dP_m}{dt} &= \frac{T_m R}{V_m} \left\{ C_D A_e P_a \sqrt{\frac{\gamma}{RT_a} (\frac{2}{\gamma + 1})^{\frac{\gamma+1}{\gamma-1}}} \chi(p) \right\} \\ &\quad -(2 - \lambda) \frac{V_{ivc}}{4\pi V_m} \alpha(P_m, \omega_e) P_m \omega_e \end{aligned} \tag{3.22}$$

3.1.2 Rotational System Dynamics

The production of mechanical power is the prime objective of an automotive engine. The amount of generated torque can be determined from its speed which depends on in-cylinder air fuel mixture and its composition. In engine MVEM, the rotational torque can be considered as nonlinear function of the variables like air to fuel ratio, mass of fuel, ignition timing and engine speed etc. Various approaches have been used to model the rotational dynamics, Willan's Approximation is utilized in [40] which states mean fuel pressure and Mean Effective Pressure (MEP) relationship to approximate a realistic behavior of engine. Basically the mean fuel pressure is brake mean effective pressure produced by an the engine with mass of fuel burnt per engine cycle with 100% efficiency. The torque generation concept, that the total pressure developed due to the thermal energy released by fuel minus the pressure produced as a consequence of the frictional and other losses, has been exploited [97]. Crossley and Cook [39] proposed that the torque generation is a function of the engine speed, air-fuel mixture, EGR mass and spark advance.

The torque generation model of Atkinson cycle has been developed by its thermodynamics analysis, using the pressure volume diagram as depicted in Figure 3.2. Based on the over-expansion cycle, the in-cylinder mean effective pressure (mep) expression is derived and afterward employed to generate torque, which has subsequently been used for evaluation of engine's angular speed. By using mechanics principles of angular motion, the rotational dynamics are modeled. In MVEM model the relationship between the net produced torque and angular speed can be expressed as [71, 96]:

$$\frac{d\omega_e}{dt} = \frac{1}{J_e}(T_{ind} - T_{pump} - T_{fric} - T_{load}) \tag{3.23}$$

where,

T_{ind} is the Indicated torque (Nm).

T_{pump} is the Pumping torque (Nm).

T_{fric} is the frictional torque (Nm).

T_{load} is the Load torque (Nm).

ω_e is Engine angular speed rad/sec

J_e is the engine inertia (kgm^2).

3.1.2.1 Indicated Torque of the Atkinson cycle

A theoretical engine torque, produced as a result of air-fuel mixture burning inside the combustion chamber, is termed as indicated torque. It can be estimated by any of the several estimation approaches. However, in this work it is presented as a function of manifold pressure and Atkinson cycle is utilized for the analysis of in-cylinder dynamics. Indicated torque in mathematical form is given as:

$$T_{ind} = \frac{V_d}{4\pi}\eta_{atk}\, mep \tag{3.24}$$

where,
η_{atk} is the Thermal efficiency of Atkinson cycle.
mep is the Mean Effective Pressure.

Essentially, the average pressure which acts on the piston head all the way through the power stroke is characterized as mean effective pressure [3, 96]. Mathematically, it can be expressed as:

$$mep = \frac{W_{percycle}}{Swept\ volume} \tag{3.25}$$

The combustion process in a four stroke SI engine is almost a constant volume process. In the modeling of rotational dynamics, a constant specific heat assumption of an ideal gas working fluid is considered throughout the cycle [3, 42]. By using Figure 3.2, the net work done during the complete over-expanded cycle is described as:

$$Wpercycle = W_{34} - W_{12} - W_{51} \tag{3.26}$$

Then we have

$$Wpercycle = \frac{P_3V_3 - P_4V_4}{\gamma - 1} - \frac{P_2V_2 - P_1V_1}{\gamma - 1} - (P_1V_1 - P_5V_5) \tag{3.27}$$

In the above equation the leading two expressions are isentropic expansion and compression processes respectively, whereas the latter one is the isobaric process.

With $P_5 = P_1$ and $V_4 = V_5$ and by approximating the actual engine processes for the analysis of the Atkinson engine cycle with ideal processes [3, 104], we have $(\frac{V_1}{V_2})^\gamma = \frac{P_2}{P_1}$ and $(\frac{V_4}{V_2})^\gamma = \frac{P_3}{P_4}$ we have

$$W_{percycle} = \frac{P_1 V_1}{\gamma - 1}[(r_c^{\gamma-1} - \lambda)(\frac{P_3}{P_2} - 1) + (\lambda - 1)(\gamma - 2)] \tag{3.28}$$

where P_2 and P_3 are the in-cylinder pressures before and after the combustion process.

Since $\frac{T_2}{T_5} = (\frac{P_2}{P_5})^{\frac{\gamma-1}{\gamma}}$ So the temperature before combustion by using isentropic transformation condition for ideal gases is:

$$T_2 = T_5 r_e^{\gamma-1} \tag{3.29}$$

3.1.2.2 Combustion Process

The fresh air fuel mixture's mass is converted into the burnt gases by deducting the same mass quantity at a defined rate of conversion from the fresh mixture and expelling it to the exhaust gas, during the combustion process. The air fuel mixture is always considered stoichiometric in the proposed model; accordingly it can be expressed as [105]:

$$C_x H_y + a(O_2 + 3.76N_2) \rightarrow xCO_2 + (y/2)\, H_2O + 3.76aN_2 \tag{3.30}$$

where

$$a = x + \frac{y}{4}$$

However, in the combustion process, the total heat released by the injected fuel does not correspond to the total heating value of the fuel. Thus some combustion imprecision is taken into account. The heat released is stated by:

$$Q_{in} = m_f Q_{LHV} \eta_c \tag{3.31}$$

where η_c is the combustion efficiency and is defined as [106]:

$$\eta_c = \eta_{cmax}\left(-1.6082 + 4.6509\sigma - 2.0764\sigma^2\right) \tag{3.32}$$

where 0.75<σ<1.2 and usually η_{cmax} is considered 90 to 95 percent for a spark ignition engine [3, 42].

As a consequence of the in-cylinder combustion process, the in-cylinder temperature and pressure upsurge, thus by using Eq (3.29) the temperature at the end of isochoric process can be given as:

$$\frac{T_3}{T_2} = 1 + \frac{m_f Q_{LHV}\eta_c}{m_m C_v T_2} \tag{3.33}$$

where,
m_m is the mass of air-fuel mixture inside the cylinder.
m_f is the mass of fuel.
Q_{LHV} is heating value of fuel.
η_c is the combustion efficiency.

By using perfect gas law at constant volume and $Swept\ volume = V_2(r_e - 1)$. The concluding expression of mean effective pressure (mep) proposed for the Atkinson cycle engine is:

$$mep = \frac{r_c}{(\gamma-1)(r_e-1)}[\frac{\zeta(r_c^{\gamma-1}-\lambda)}{r_e^{\gamma-1}} + (\lambda-1)(\gamma-2)]P_m \tag{3.34}$$

$$where,$$

$$\zeta = \frac{Q_{LHV}\eta_c}{T_m C_v(AFR+1)}$$

where,
C_v is Heat capacity at constant volume.
AFR is Air to fuel ratio.
γ is Ratio of heat capacities.
λ is the Ratio of the expansion ratio to the effective compression ratio.

With late intake valve closing (LIVC), the in-cylinder temperature and pressure reduce as a consequence of lower effective compression ratio (CR). Thus, with LIVC the mean effective pressure (mep) may declines. However, as intake valve closing time is essentially dependent on the engine operating conditions and can be physically realized for medium and higher loads, to avoid the ignition delays and combustion duration, subsequently the combustion process. So, LIVC strategy would not have substantial impact on the engine load (mep). Besides, to maintain the combustion process the CR is required to be maintained by variable compression ratio (VCR) mechanism which is inversely proportional to the engine load [60].

3.1.2.3 Thermal Efficiency of the Atkinson Cycle Engine

From an analysis of the ideal Atkinson cycle as shown in Figure 3.2, the effect of over-expansion on the efficiency can be estimated. Using the laws of thermodynamics, the thermal efficiency of the over-expanded cycle can be described as [42]:

$$\eta_{atk} = 1 - \frac{Q_{4-5} + Q_{5-1}}{Q_{2-3}} \tag{3.35}$$

where, Q_{2-3} and Q_{4-5} are the heat added and released at constant volume respectively, while Q_{5-1} is the heat released at constant pressure.

Eventually, the thermal efficiency of Atkinson cycle VVT engine derived is as:

$$\eta_{atk} = 1 - \frac{1}{r_e^{\gamma-1}} - \frac{\{(\gamma-1)\lambda^{\gamma} - \gamma\lambda^{\gamma-1} + 1\}}{\zeta\lambda^{\gamma-1}} \tag{3.36}$$

In detail derivation of thermal efficiency of the Atkinson cycle $'\eta'_{atk}$ is given in the Appendix A.

3.1.2.4 Pumping and Frictional Torque

The mathematical forms of pumping torque required to perform pumping action and engine frictional torque can be written as [42]:

$$T_{pump} = \frac{V_d}{4\pi}(Pa - Pm) \tag{3.37}$$

and

$$T_{fric} = \frac{V_d}{4\pi}\left\{(0.97 + 0.15\frac{N}{10^3} + 0.05\frac{N^2}{10^6}) \times 10^5\right\} \tag{3.38}$$

where N is the angular speed in RPM.

Moreover, some of the parameters used for the physics based EMVEM model are derived from first principal basis and several of these are used as given in the concerned references. For instance the parameters C_D, $\alpha(P_m, \omega_e)$ are taken from [10] and the coefficients in Eq (3.38) are used from [42], have been determined experimentally, whereas η_{vol}^{atk} is physics based and is modeled by incorporating the valve timings.

In the proposed EMVEM model, the IVT parameter "λ" is essentially the control parameter and its value depends primarily on the engine operating conditions. It has a vibrant role in handling the engine load, encapsulating the thorough dynamics of the Atkinson cycle engine and to work out the benefits of the control-oriented model. In the following sections the model validation as well as the simulation results of the anticipated EMVEM model, are presented.

3.2 Complete Control-Oriented Atkinson Cycle Engine EMVEM Dynamics

The novel nonlinear control-oriented extended mean value engine model (EMVEM) dynamics of the Atkinson cycle engine [2], is given as follows:

$$
\begin{aligned}
\dot{P}_m &= \Psi_1 \chi(p) - (2-\lambda)\Psi_2 P_m \omega_e \alpha(P_m, \omega_e) \\
\dot{\omega}_e &= \frac{1}{J_e}\left(T_{ind} - T_{pump} - T_{fric} - T_{load}\right)
\end{aligned}
\tag{3.39}
$$

where,

$$
\begin{aligned}
\Psi_1 &= \frac{T_m R}{V_m} C_D A_e P_a \gamma_c \\
\Psi_2 &= \frac{V_{ivc}}{4\pi V_m} \\
\chi(p) &= 1 - \exp\left(9\frac{P_m}{P_a} - 9\right) \\
A_e &= \pi \frac{D^2}{4}(1 - \cos(\frac{\phi + \phi_{cl}}{\phi_{cl}})) \\
\gamma_c &= \sqrt{\frac{1}{RT_a}}\sqrt{\gamma(\frac{2}{\gamma+1})^{\frac{\gamma+1}{\gamma-1}}} \\
T_{ind} &= \frac{V_d}{4\pi}\eta_{atk}\ mep \\
mep &= \frac{r_c}{(\gamma-1)(r_e-1)}[\frac{\zeta(r_c^{\gamma-1} - \lambda)}{r_e^{\gamma-1}} + (\lambda-1)(\gamma-2)]P_m \\
\eta_{atk} &= 1 - \frac{1}{r_e^{\gamma-1}} - \frac{[(\gamma-1)\lambda^{\gamma} - \gamma\lambda^{\gamma-1} + 1]}{\zeta\lambda^{\gamma-1}} \\
where, & \\
\zeta &= \frac{Q_{LHV}\eta_c}{T_m C_v(AFR+1)} \\
\eta_c &= \eta_{cmax}\left(-1.6082 + 4.6509\sigma - 2.0764\sigma^2\right)
\end{aligned}
$$

with $0.75<\sigma<1.2$ and usually η_{cmax} is considered 90 to 95 percent for a spark ignition engines [3, 42].

$$
\begin{aligned}
T_{pump} &= \frac{V_d}{4\pi}(Pa - Pm) \\
T_{fric} &= \frac{V_d}{4\pi}[(0.97 + 0.15\frac{N}{10^3} + 0.05\frac{N^2}{10^6})10^5]
\end{aligned}
$$

Load Torque (T_{load}):

An external loads on engine include all except pumping and frictional losses that the engine have to rotate/pull are termed as T_{load} . For instance, for a vehicle, the entire engines rotating parts and its driven subsystems, including air conditioner, cam and valve timing system, electrical generating set etc. vehicles weight and everything in it is the load. Whereas, a generator is the load in case of electrical generating set. The descriptions and parameters values of the gasoline engine used for simulations are illustrated in the following Table 3.1.

TABLE 3.1: Description of EMVEM model parameters and their nominal values

Symbol	**Description**	**Value/Units**
P_a	Ambient pressure	101325 $Pascal$
P_m	Manifold pressure	$Pascal$
T_a	Ambient temperature	298 K
T_m	Manifold temperature	325 K
ϕ_{cl}	Throttle angle at closed position	9.8 deg
V_m	Manifold volume	dm^3
D	Inlet diameter	19 mm
C_D	Throttle discharge coefficient	0.345
γ	Ratio of heat capacities	1.4
r_e	Expansion ratio	11.5
R	Specific gas constant	287 $J/Kg.K$
$\alpha(P_m, \omega_e)$	Volumetric efficiency	
η_c	Combustion efficiency	0.9
AFR	Air to fuel ratio	14.7
J_e	Engine inertia	0.20 $kg.m^2$
Q_{LHV}	Heat value of fuel	44 MJ/Kg
C_v	Heat capicity at specific volume	717 J/Km^2
ω_e	Angular speed	rad/sec
N	Angular speed	RPM

3.3 EMVEM Model Validation

With the intention to demonstrate the capability of the developed model to accurately capture the dynamics of the Atkinson cycle VVT engine and its load control proficiencies, the proposed EMVEM model is validated against 28 steady state operating points through the experimental data of the VVT YANMAR L48AE engine, obtained from literature, for each of the three different late intake valve closing (LIVC) timings. Which has the following specifications [10] as shown in Table 3.2.

Table 3.2: Specifications of VVT Engine

Sr	Parameter	Value
1	No. of cylinders	1
2	Bore -Stroke (mm)	70 x 55
3	Displacement (dm^3)	0.21166
4	Inlet diameter (mm)	19
5	Air to Fuel ratio	14.5
6	Compression ratio	11.5
7	Throttle discharge coefficient	0.345
8	Maximum power (kW)	3.5 @ 3000rpm
9	Maximum torque (N-m)	18 @ 2500rpm
10	IVO [CA]	20 BTDC
11	IVC [CA]	32 ABDC
12	EVO [CA]	40 BBDC
13	EVC [CA]	12 ATDC

The experimental data, such as air mass flow rate, volumetric efficiency, load torque against rotational speed ranging from 1000 rpm to 3670 rpm for LIVC with three cams having dwell angles 20^o, 40^o and 60^o, have been used for the validation purpose. With this strategy, the valve timing variations have been attained and the cam profiles used to achieve different loads are shown in Figure 3.4. The volumetric efficiencies of each camshaft which have been used for this purpose are revealed in Figure 3.5.

The comparison of the proposed EMVEM model's simulation results, using the given data and the experimental data by way of LIVC load controlling strategy, are demonstrated in Figure 3.6. The dashed and dotted lines correspond to the simulation model results and solid lines correspond to the experimental data. Wherein, at the outset the load has been controlled by incorporating the given air mass flow rate $\dot{m}_i$ (experimental data) in accordance with volumetric efficiency and load torque, whereas angular speed has been considered as an output in the course of entire operating range. In case of the conventional Otto cycle engine, this air mass flow rate $\dot{m}_i$ can only be controlled through throttle (the foremost cause of pumping losses at part load).

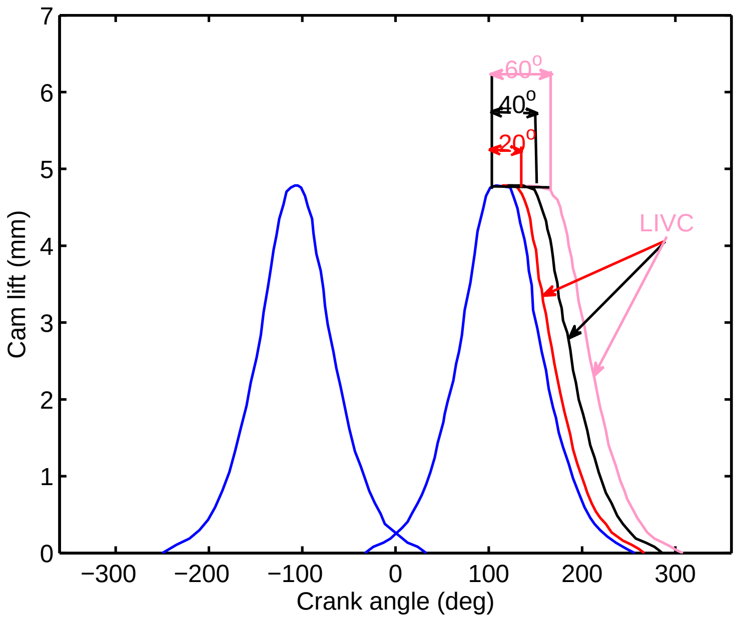

FIGURE 3.4: Camshaft profiles used for LIVC strategy

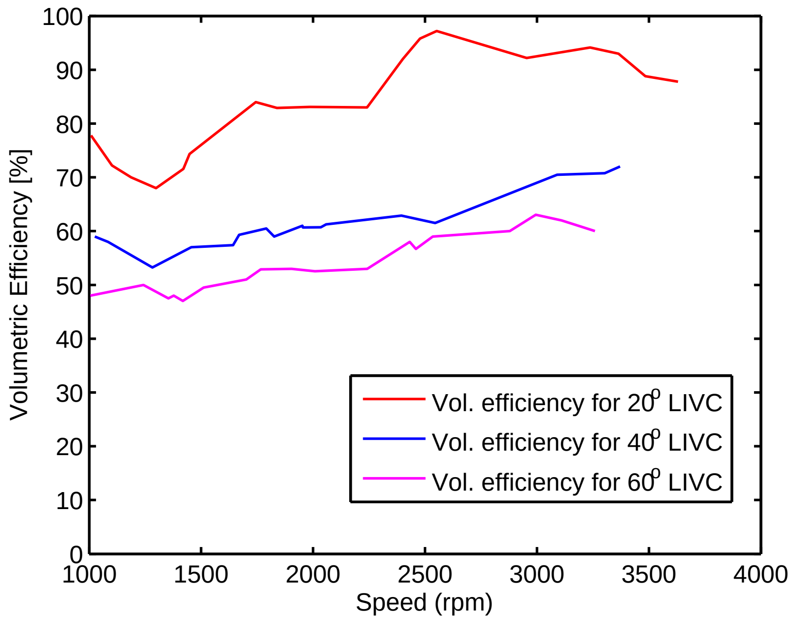

FIGURE 3.5: Volumetric efficiency against speed at various dwelling angles

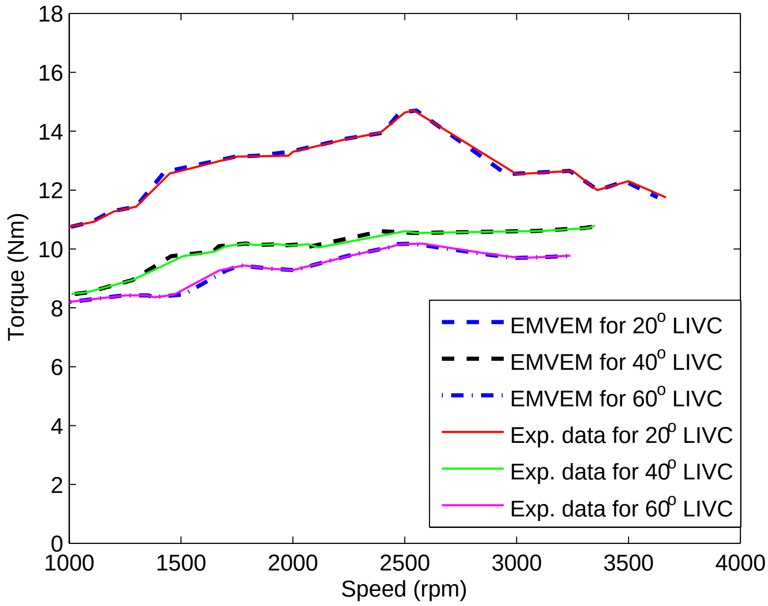

FIGURE 3.6: Comparison of the EMVEM model simulation and experimental torques at different LIVC timings

Afterward, with the similar engine operating conditions the late intake valve closing strategy has been used for controlling engine load (through the proposed IVT parameter $'\lambda$') for authentication of the anticipated EMVEM model at Wide Open Throttle (WOT). These results indicate that the developed model has proficiency to apprehend the essential dynamics of the over-expanded cycle engine and provides an appropriate means of handling the engine load. EMVEM model has a pronounced significance in the reduction of the throttling effect (the leading source of engine's performance degradation), at part load.

The intake valve opening and exhaust valve timings have been kept fixed during the analysis. With proper VVA mechanism design and control, a VVT engine can accomplish near zero pumping losses, optimum volumetric efficiency, enhancement in the thermal efficiency, subsequently specific fuel consumption (SFC) reduction and minimization of exhaust pollutions [4, 19, 22, 92]. The proposed EMVEM model simulation results and the experimental data have a good agreement; however, the discrepancies at some points are due to using the engine frictional torque [42], instead of frictional torque specified in [10]. The given frictional torque expression has exponential growth beyond the angular speed 1700 rpm. Generally, the model has applications in the development of estimation and control algorithms, impact of small modeling discrepancies can be mitigated through the benefits from the use of sensor feedback.

3.4 Results and Discussions

The analysis of the proposed EMVEM model of the Atkinson cycle engine is performed, by using the parameters of aforesaid engine, to figure out the advantages and ascertain the effectiveness of the EMVEM model in the declining of the throttling effect and improvement in the thermal efficiency. It is shown that the proposed EMVEM model demonstrates the same benefits/features as commonly attributed to the Atkinson cycle engine. These observations are tantamount to indirect validation of the proposed EMVEM model.

3.4.1 Engine Part Load Losses and IVT Parameter Constraints

The part load losses for Atkinson cycle engine with developed EMVEM model and MVEM of the conventional Otto cycle engine are studied under similar operating conditions. Figure 3.7 reveals the comparison between pumping losses of the EMVEM model and the conventional model for the entire engine load range while keeping the speed constant at 3000 rpm. It shows the significance of the proposed model by exhibiting considerable reduction in pumping losses with EMVEM model as a consequence of its load handling LIVC strategy over the conventional throttled Otto cycle. The decline in pumping losses with LIVC strategy is predominantly the result of back-flow notion of the A/F charge into the intake manifold of the engine or it simultaneously turns into the intake of another combustion cylinder. Thus, suction pressure rests considerably above the atmospheric pressure during the first part of compression stroke, which is a key contributor in lessening of pumping losses.

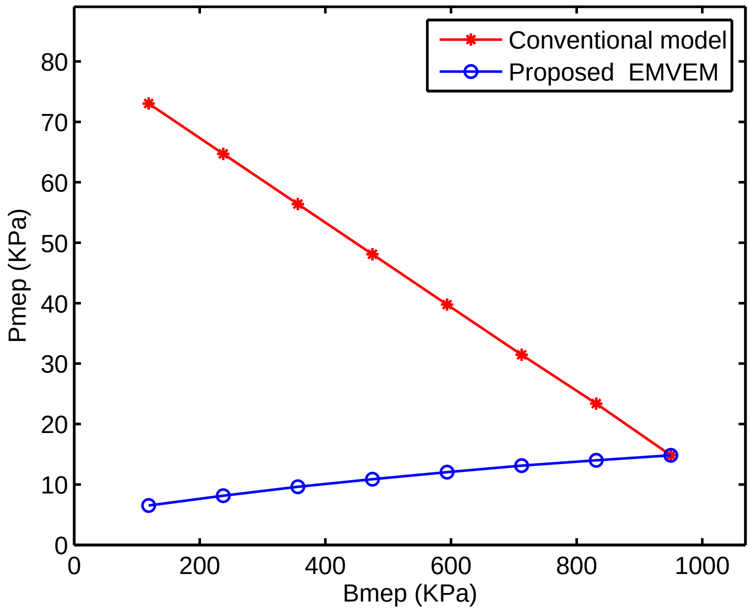

FIGURE 3.7: Comparison of Pumping losses Vs Brake mean effective pressure of the EMVEM model and conventional model at 3000 rpm

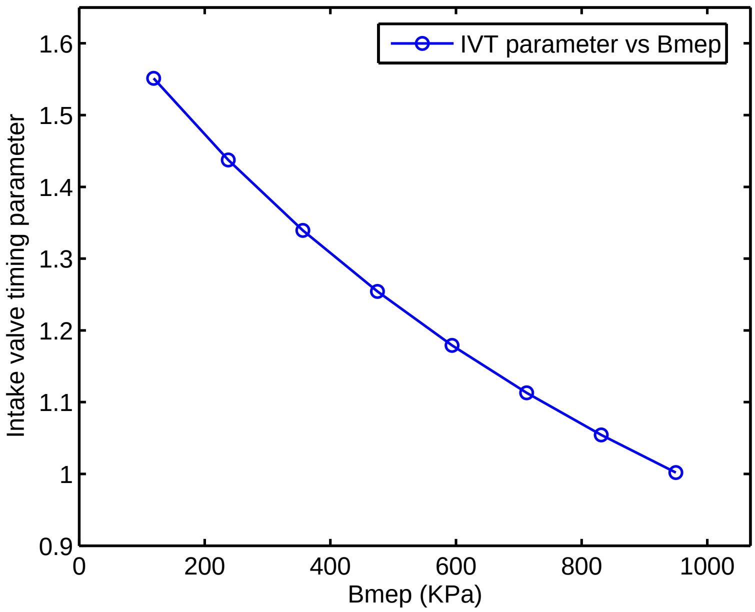

FIGURE 3.8: Constraints on the IVT parameter of the EMVEM model against load

In addition, the simulation results also figured out the bounds i.e. $1 \leq \lambda \leq 1.55$ of the IVT parameter "λ" during engine operating conditions. The intake valve closing time is essentially engine operating conditions dependent as shown in Figure 3.8 and can be physically realized for medium and higher loads, to avoid the affect on ignition delay and combustion duration. Instead, the use of throttle may be required below 40% of the full engine loads without losing the advantages of the incorporated VVT mechanism [53]. Moreover, it is also figured out that as the "λ" upturns beyond 1, the effective compression ratio (CR) decreases due to LIVC. Consequently, the in-cylinder temperature and pressure decline, which degrades the combustion efficiency [18, 19] and to improve it the compression ratio is required to be maintained with the variable compression ratio mechanism, which varies dynamically the clearance volume depending on the engine load [53].

3.4.2 Thermal Efficiency Augmentation

The enhancement in the thermal efficiency with the developed EMVEM model of the Atkinson cycle VVT engine in comparison with the conventional Otto cycle MVEM model predominantly at part load, is demonstrated in Figure 3.9, which depicts that augmentation in the thermal efficiency is rather significant further down the WOT. Essentially, with LIVC strategy, as load decreases it allows comparatively for greater value of the IVT parameter (λ), therefore the pressure ratio ($\frac{p_m}{Pa}$) improves as a result of the air-fuel mixture back-flow phenomenon which plays vital role in the reduction of losses at part load. Consequently, the improvement in the thermal efficiency can be attained.

Whereas, the results in Figure 3.10 point out the promising aspects of over-expanded cycle that the thermal efficiency of the EMVEM model can be enhanced approximately up to 18.0%. The optimum increase in the thermal efficiency corresponds to value of $\lambda = 3.5$, which can be attained through the expansion stroke, where the in-cylinder pressure becomes equal to the atmospheric pressure, which is hardly ever possible otherwise. However, with the proposed EMVEM model 9.0% improvement in the thermal efficiency has been achieved as compared to the

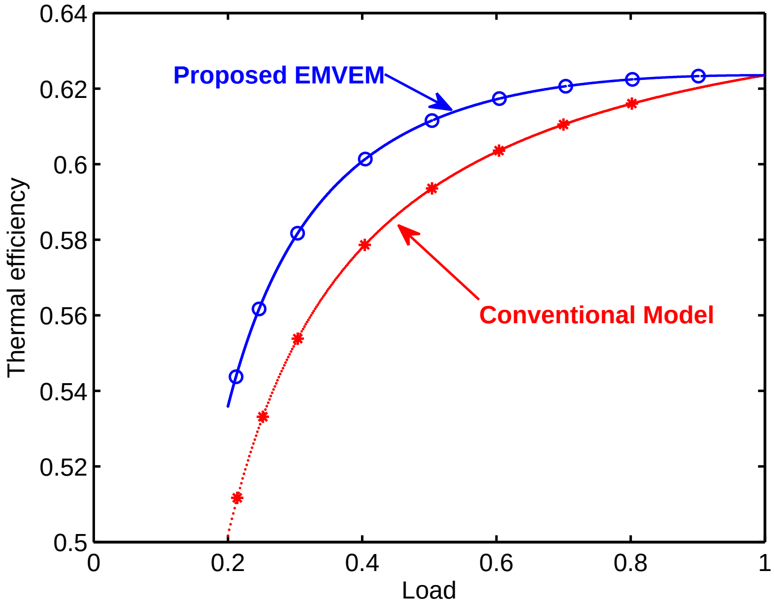

Figure 3.9: Proposed EMVEM model's thermal efficiency in comparison with that of the conventional Otto cycle MVEM as a function of Load

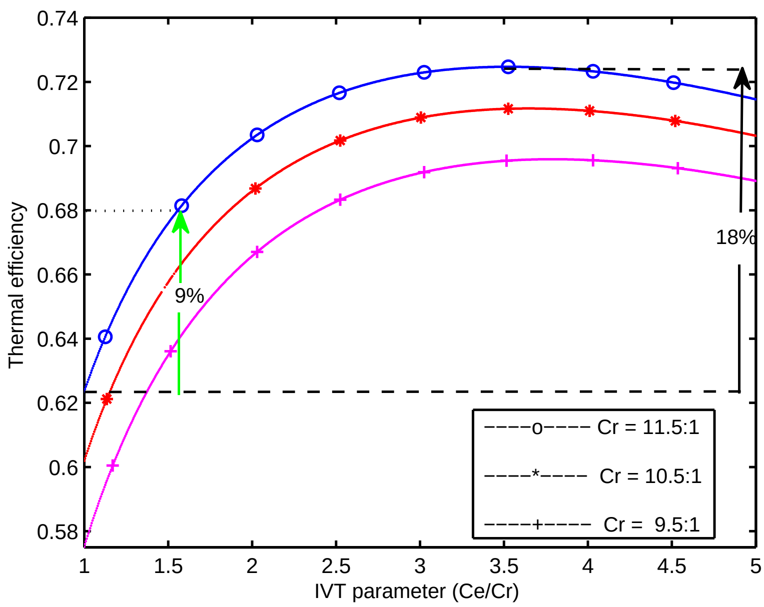

Figure 3.10: Thermal efficiency of EMVEM Versus IVT parameter for various values of compression ratio

conventional MVEM model. The decline in the thermal efficiency beyond $\lambda = 3.5$ happens as a consequence of the attendant increase in losses, such as unburned fuel and cooling losses [60]. The enormous benefits of the EMVEM model in terms of the reduction of throttling effect and augmentation in the thermal efficiency, furthermore the constraints on IVT parameter "λ" i.e on LIVC and the significance of variable compression ratio are strongly endorsing the proposed EMVEM model. Moreover, aforesaid advantages, besides the control based applications of the developed model of the Atkinson cycle VVT engine, have great significance in the improvement of fuel economy.

Besides, the notion of variable compression ratio that stems from the EMVEM modeling philosophy is beneficial for turbocharge engines as well, as a turbocharged engine need variable compression ratio to avoid overheating and over-stressing the cylinder head, otherwise it may cause knocking or even damages. When the engine runs off-boost, a weak output will be produced as a consequence of lower compression. With variable compression ratio mechanism, using a higher ratio before the turbo gets into operation and a lower ratio under boost, a perfect turbocharged engine, can be realized.

3.5 Conclusion

This chapter frameworks a physically motivated extended mean value engine model of the Atkinson cycle VVT engine, wherein the Atkinson cycle, flexible variable valve actuation, variable compression ratio and the over-expansion characteristics have been incorporated. The proposed model has been validated against the experimental data of the YANMAR L48AE engine during the entire engine operating conditions, to ensure that the EMVEM model has proficiency, to capture the dynamics of the Atkinson cycle engine and to handle the engine load not only through the conventional actuator (throttle position) but with intake valve timing over the useful engine operating range as well. The model encapsulates air dynamics sub model in which variable valve actuation system is unified to cope with the engine performance degrading aspects as throttle valve, fixed valve timings and physics based rotational dynamics sub model, wherein Atkinson cycle is utilized for analysis of in-cylinder dynamics. The EMVEM model has great implications on the reduction of throttling effect and augmentation in the thermal efficiency, primarily at part load, furthermore constraints on IVT parameter 'λ' i.e. on LIVC and the significance of variable compression ratio, are strongly endorsing the model. The control-oriented model is anticipated for systematic analysis, simulations, the development of closed-loop control and estimation strategies to support and enable novel load control strategies as well as improve fuel economy of the Atkinson cycle SI engine used in hybrid electric vehicles.

To investigate fuel economy of an Atkinson cycle Engine in comparison with the conventional Otto cycle engine, the proposed novel EMVEM model is analyzed with respect to unconventional LIVC control input and an alternate control framework for an Atkinson cycle VVT Engine is developed in the ensuing chapter.

Chapter 4

Design, Development and Evaluation of a Control Framework for an Atkinson Cycle Engine

To pursue the prospective benefits of the novel control-oriented EMVEM model of the Atkinson cycle engine [2], in view of fuel economy, this chapter deals with the design and development of a control framework for the Atkinson cycle engine. This chapter also comprehends the detailed analysis of the Atkinson cycle engine EMVEM model with an alternative flexible intake valve load handling strategy from control design as well as fuel economy point of view. The key performance degrading aspects such as pumping losses and thermal efficiency of the conventional Otto cycle SI engine have been investigated and concluded that the efficiency of the spark ignition (SI) engine degrades while working at part loads [10]. It can be optimally dealt with a slightly different thermodynamic cycle termed as an Atkinson cycle. The control framework based on the already developed control-oriented EMVEM model of the Atkinson cycle engine is evaluated for the standard New European Driving Cycle (NEDC), Federal Urban Driving Schedule (FUDS) and Federal Highway Driving Schedule (FHDS) driving cycles.

To demonstrate, the advantages of the VCR Atkinson cycle VVT engine, in the presence of auxiliary loads and uncertain road loads, its EMVEM model is simulated by using a precisely tuned controller having similar specifications as that of the conventional gasoline engine. In this context, a standard framework (typical driving cycles) is adapted. The simulation results point towards the significant reduction in engine part load losses and improvement in the thermal efficiency over the wide operating range. Consequently, considerable enhancement in the fuel economy of the VCR Atkinson cycle VVT engine is achieved over conventional Otto cycle engine during the NEDC, FUDS and FHDS driving cycles.

4.1 Performance Degrading Aspects of Conventional SI Engine

The potential benefits of the Atkinson cycle engine as compared to the Otto cycle engine, in terms of the fuel economy improvement and exhaust emissions decrease stem from the fact that the standard SI engines that are used in the automotive cars are designed to have optimum efficiency at the full load. Whereas, most of the time the automotive engines have to be operated at part loads (light loads), their load is being controlled through the conventional throttle, as a result the intake manifold pressure drops considerably. Accordingly, aggravation of pumping losses (throttling effect) leads to substantial inefficiency [4].

Likewise, the thermal efficiency degrades meaningfully at engine light loads as the conventional SI gasoline engines have fixed compression ratio ranges between 8 to 12 and are limited by fuel quality and knock. Compression ratio is most the promising variable that influences the thermal efficiency [3]. Martine and Ribeiro [1, 53] examined the part load concerns of the Otto cycle engine and have derived the thermal efficiency of the conventional Otto cycle engine described as :

$$\eta_{otto} = 1 - \frac{1}{r_c^{\gamma-1}} - \left(\frac{1}{B}\right)\left(1 - \frac{1}{\beta}\right)\left(1 - \frac{1}{r_c}\right) \tag{4.1}$$

where,

$$B = \frac{\eta_c Q_{LHV}}{RT_m(AFR+1)}$$

and

$$\beta = \frac{P_m}{P_a}$$

The research in [4, 16, 19, 22, 92, 107, 108] investigates comprehensively the importance of variable valve actuation (VVA), variable compression ratio (VCR) systems and the vital role of the accurate engine control on the SI engine's fuel consumption. Moreover, the research [1, 55, 59, 60, 109**?**] examined the thermal efficiencies of several engine cycles and concluded that the Atkinson cycle (over-expanded cycle) is optimum thermal efficiency contributing thermodynamic cycle. Vincent Knop and Leonardo Mattioli [109] have analyzed the pros and cons of the VVT engine, part load efficiency enhancements using VVA and VCR mechanisms and determined the improvement in fuel consumption.

Thorough studies have been accomplished on the over-expanded engine cycle by using cylinder-by-cylinder (CCEM) models with LIVC load handling approach, theoretically as well as experimentally. About 6% improvement in the fuel economy of the VCR Over-expanded engine cycle using CCEM model over the Otto cycle engine has been evaluated during the standard New European Driving Cycle [10, 93]. The specific fuel consumption (SFC) of this engine cycle, realized by VVT as well as using VCR mechanism has been improved around 19% over the conventional Otto cycle SI engine, at medium and higher loads[1, 53, 55]. Sugiyama et al. [19] has reported 16.2% reduction in the fuel consumption as a result of the innovated mechanisms.

Therefore, to cope with the SI engine performance degrading aspects i.e. engine throttling effect and the thermal efficiency, in order to enhance the fuel economy and emission reduction, hybrid electric vehicles are using the over-expanded Atkinson cycle engine appreciated through VVT and incorporated with VCR mechanism

[1, 55, 59]. Around half of the fuel economy claimed by HEVs is achieved from the Atkinson cycle engine.

4.1.1 Modeling of Variable Compression Ratio Effect

The research [10, 19, 93, 109] has also investigated inclusively, the causes of the innovative mechanisms incorporated in the Atkinson cycle VVT engines, on the combustion process ensuing from decline in the engine effective compression ratio as a consequence of the delaying of IVC timing, besides the provision of additional degree of freedom to optimize the engine efficiency. LIVC load control strategy leads to less efficient combustion process (lower the in-cylinder peak temperature and pressure) and surpassed the gain in terms of fuel consumption. However, this inadequacy can be better dealt with Atkinson cycle VVT engine, wherein VCR mechanism is incorporated, which can adjust compression ratio to pre knock onset conditions [1, 10, 18–20] and so the efficiency of SI engine can be enhanced significantly at part loads specifically. Therefore, to model the variable compression ratio (VCR) effect, the combustion chamber's height is changed at each engine load simulation in order to accomplish around the same maximum in-cylinder temperature and pressure [93].

Another significant aspect to accomplish the engine performance (fuel economy, emission reduction and vehicle drivability), the availability of accurate and simple control-oriented engine model incorporated with innovative technologies and the engine robust controllers, are the key contributors [29–31, 46]. SI engine speed performance has significant impact on the vehicle design attributes as fuel economy, emission and comfort i.e. drivability, especially during the traditional vehicle operations [110, 111]. Engine speed control is a demanding nonlinear issue and has led to many approaches to tackle it. In this association, Puleston et al. [112] have used a dynamic sliding mode design approach for speed control of the conventional SI engines, whereas [113] has employed 2-sliding super-twisting algorithm for the conventional engine speed control based on MVEM model. The work in

[29] has also addressed speed control problem of the SI engine by using a proportional feedback controller scheme as well as the stability analysis of the proposed control system by considering its transient performance. Leroy et al. [114] have studied the air-path control problem of a variable valve timing actuators equipped turbocharged SI engine to look out for the benefits on its torque output and on the air to fuel ratio.

Most of the research in the appraisal of the over-expanded cycle engine, incorporated with the innovative mechanisms regarding fuel economy, has considered either the conventional Otto cycle MVEM model or CCEM models of the VVT engines only. Development and evaluation (in the view of fuel economy) of any control strategy based on EMVEM model of the Atkinson cycle engine through standard outline, has not been studied so far.

In the research [8], a control framework for the Atkinson cycle engine with flexible late intake valve load control strategy is designed and developed. The control framework is based on EMVEM model of the Atkinson cycle engine and is evaluated for the standard NEDC, FUDS and FHDS driving cycles.

The organization of this chapter is structured as follows: The brief description of the control-oriented EMVEM model of an Atkinson cycle engine is presented in Section 2. EMVEM model based control framework for the Atkinson cycle engine is developed in Section 3. In Section 4, Control framework evaluation, in the context of fuel economy is examined and finally, presented the conclusion.

4.2 Atkinson Cycle Engine Model Description

The model accuracy of the SI engine with flexible advanced technologies as well as model based control strategies play key role in the improvement of the engine performance, fuel economy and emission decline [3, 40, 69, 98, 100, 107]. The complete control-oriented Atkinson Cycle Engine EMVEM model [2], in which the characteristics such as variable valve timing, variable compression ratio, over-expansion and realization of the Atkinson cycle have been integrated, and, established based on

- Fluid Dynamics Principles
- Thermodynamics Principles
- Atkinson cycle for the in-cylinder dynamics analysis
- Inertial laws

is as described in the preceding chapter and utilized for control design and evaluation.

In this model, late intake valve closing (LIVC) load control strategy is employed, instead of the conventional throttle and constraints on the IVT parameter $'\lambda'$ are essentially depending on the VVT engine operating conditions. The IVT parameter at WOT throttle is considered as a control variable (amount of charge trapped in the cylinder is decided by intake valve closure timing i. e. IVT parameter) and angular speed as an output while evaluating fuel consumption. In the following section the design and development of a control framework for the Atkinson cycle engine with new control input to compute the benefits of the Atkinson cycle engine over conventional SI engine in terms of fuel economy is presented.

4.3 Control Framework Development

With the intention of designing control for the Atkinson cycle engines based on nonlinear EMVEM model through an alternative load handling strategy, controllability of the EMVEM model with its intrinsic control input (IVT parameter) is examined as in the following subsection.

4.3.1 Control-oriented EMVEM Model Analysis and Control Design

To investigate the EMVEM model controllability with an unconventional control input channel, the nonlinear dynamics of the Atkinson cycle engine [2] is represented in state space equations:

$$\dot{x} = f(x) + g(x)u \tag{4.2}$$

where, $x = [Pm, w_e]^T \in \Re^2; u \in \Re; f : \Re^2 \to \Re^2; g : \Re^2 \to \Re^2$; $f(x)$ and $g(x)$ being smooth vector fields and are given as:

$$f(x) = \begin{bmatrix} \Psi_1\chi(p) - 2\Psi_2 P_m \omega_e \alpha(P_m, \omega_e) \\ \\ \frac{V_d}{4\pi J_e}\eta_{atk}\left\{\frac{r_c}{(\gamma-1)(r_e-1)}\right\}\left\{\frac{\zeta r_c^{\gamma-1}}{r_e^{\gamma-1}} - (\gamma-2)\right\}P_m \\ -\frac{1}{J_e}\left(T_{pump} + T_{fric} + T_{load}\right) \end{bmatrix}$$

and

$$g(x) = \begin{bmatrix} \Psi_2 P_m \omega_e \alpha(P_m, \omega_e) \\ \\ \frac{V_d}{4\pi J_e}\eta_{atk}\left\{\frac{r_c}{(\gamma-1)(r_e-1)}\right\}\left\{\frac{\zeta}{r_e^{\gamma-1}} - (\gamma-2)\right\}P_m \end{bmatrix}$$

and u is control input (IVT parameter 'λ'). As the third term in the expression of η_{atk}, during the engine entire operating conditions (within the constraints on IVT parameter i.e. $1 \leq \lambda \leq 1.60$) have utmost value approximately 0.004, for simplicity this term has been ignored for cotrollability analysis.

According to Slotine and Weiping [115], the nonlinear system Eq (4.2), with f(x) and g(x) being smooth vector fields, is controllable if and only if there exists a region $\Omega \in \Re^n$ such that

- the vector fields $\left\{g, ad_f g, ..., ad_f^{n-1} g\right\}$ are linearly independent in $\Omega \in \Re^n$.

The Lie bracket of the f(x) and g(x) is computed as:

$$ad_f g = \begin{bmatrix} \Psi_1\Psi_2\alpha(P_m,\omega_e)\omega_e\left(9\frac{P_m}{P_a}+1\right)\exp\left(9\frac{P_m}{P_a}-9\right) \\ +\Psi_2\alpha(P_m,\omega_e)P_m^2\frac{V_d}{4\pi J_e}\eta_{atk}\left\{\frac{r_c}{(\gamma-1)(r_e-1)}\right\} \\ \left\{\frac{\zeta(r_c^{\gamma-1}-2)}{r_e^{\gamma-1}}-2(\gamma-2)\right\}-\Psi_2\alpha(P_m,\omega_e)P_m \\ \frac{1}{J_e}\left(T_{pump}+T_{fric}+T_{load}\right) \\ \\ -\frac{V_d}{4\pi J_e}\eta_{atk}\left\{\frac{r_c}{(\gamma-1)(r_e-1)}\right\}\left\{\frac{\zeta}{r_e^{\gamma-1}}-(\gamma-2)\right\} \\ \left[\Psi_1\chi(p)+\frac{V_d}{4\pi J_e}\left\{\left(\frac{450}{\pi}-\frac{9\omega_e}{\pi^2}\right)\right\}P_m\right] \\ -\frac{V_d}{4\pi J_e}\eta_{atk}\left\{\frac{r_c}{(\gamma-1)(r_e-1)}\right\}\Psi_2\alpha(P_m,\omega_e)P_m\omega_e \\ \left\{\frac{\zeta(r_c^{\gamma-1}-2)}{r_e^{\gamma-1}}-2(\gamma-2)\right\}-\frac{V_d}{4\pi J_e}\Psi_2\alpha(P_m,\omega_e)P_m\omega_e \end{bmatrix}$$

By using Mathematica, it is found that at the engine entire operating range the controllability matrix

$$rank(Co) = \begin{bmatrix} g & ad_f g \end{bmatrix} = 2$$

has full rank. Therefore, the proposed nonlinear control-oriented EMVEM model of the Atkinson cycle engine is controllable with its intrinsic LIVC control strategy i.e. IVT parameter $'\lambda'$ [115].

4.3.2 Fuel Dynamics System

Moreover, to examine the fuel economy of the Atkinson cycle engine, the fuel dynamics subsystem is utilized [71] and is described as:

$$\ddot{m}_{ff} = \frac{1}{\tau_f}\left(-\dot{m}_{ff} + X_f\dot{m}_{fi}\right) \tag{4.3}$$

$$where,$$

$$\dot{m}_{fv} = (1 - X_f)\dot{m}_{fi}$$

$$\dot{m}_{fo} = \dot{m}_{fv} + \dot{m}_{ff}$$

$$and,$$

$$\dot{m}_{fi} = (2 - \lambda)\Psi_2 P_m \omega_e \alpha(P_m, \omega_e)$$

where,
$\dot{m}_{fv}$ is the fuel vapor mass flow.
$\dot{m}_{ff}$ is the fuel film mass flow.
$\dot{m}_{fi}$ is the injected fuel mass flow.
$\dot{m}_{fo}$ is the fuel flow rate in the combustion chamber.
X_f is injected fuel friction deposited on manifold as fuel film.
τ_f is time constant.

To evaluate the fuel economy of the Atkinson cycle engine through its nonlinear EMVEM model in comparison with conventional SI engine during wide engine load range, both the engines are simulated by using a precisely tuned PID controller having similar specifications. It is to assume that the air to fuel ratio is being controlled/ maintained at stoichiometric ratio by an independent robust controller. The values of the gains computed for PID controller

$$PID = K_P + \frac{K_I}{s} + K_D s \tag{4.4}$$

are given as $K_P = -0.121$, $K_I = -0.0316$ and $K_D = 0.0981$, for the non-minimum nonlinear EMVEM dynamical systems. The schematic representation of the closed loop VVT engine system using the flexible intake valve timing strategy with a PID controller is as shown in Figure 4.1. Whereas, Figure 4.2 demonstrates the functionality of the developed control framework with its intrinsic LIVC load handing strategy. It shows the effectiveness of the control framework in terms of good speed tracking and it is insensitive to load variations as well. Besides, it exhibits the variation of control effort (IVT parameter λ) in accordance with load torque changes as IVT parameter ('λ') is inversely proportional to load torque [2]. For simplicity in this study, a PID controller is used; however any robust controller can be incorporated in this engine control outline as pursued a later chapter. The symbols, their description and values are [10, 71] as depicted in Table 4.1.

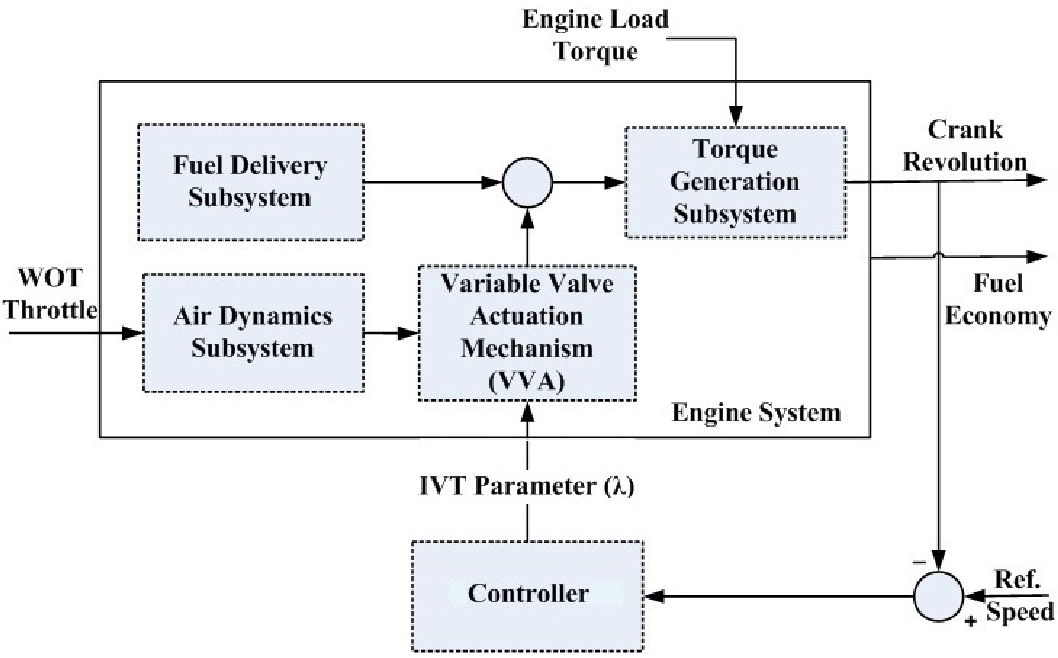

FIGURE 4.1: Schematic representation of closed loop Atkinson cycle VVT engine system [8]

In the following section, with simulation test, the features of the designed controller are investigated, aiming to demonstrate its tracking performance along with its application in evaluation of the VCR Atkinson cycle engine in fuel economy point of view over the conventional Otto cycle engine.

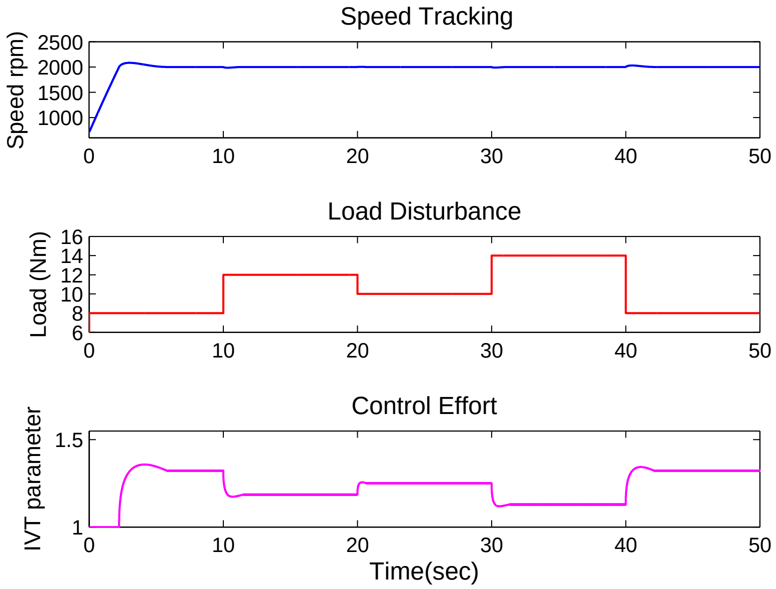

FIGURE 4.2: Demonstration of the functionality of the developed control framework using LIVC load handing strategy

4.4 Control Framework Evaluation in Fuel Economy Perspective

To examine the potential benefits of the VCR Atkinson cycle engine with an unconventional variable LIVC load handling strategy instead of the throttle, a control framework designed for the Atkinson cycle engine is evaluated by using its already developed control-oriented EMVEM model. This research is conducted to assess fuel economy of the VCR Atkinson cycle engine over the conventional Otto cycle engine, at the medium and higher load operating conditions, during the standard NEDC, FUDS and FHDS driving cycles. In this context, both the engine models i.e. control-oriented EMVEM of Atkinson cycle engine as well as the conventional MVEM are simulated by using a precisely tuned PID controller having similar specifications as described before. To cope with the causes of the innovative mechanisms incorporated in the Atkinson cycle VVT engines, on the combustion process ensuing from decline in the engine effective compression ratio as a consequence of delaying of IVC timing. The notion of VCR mechanism is used which can adjust compression ratio to pre knock onset conditions. To

TABLE 4.1: Engine Specifications

Symbol	Description	Value/Units
P_a	Ambient pressure	101325 $Pascal$
T_a	Ambient temperature	298 K
T_m	Manifold temperature	325 K
ϕ_{cl}	Throttle angle at closed position	9.8 deg
V_m	Manifold volume	dm^3
D	Inlet diameter	19 mm
C_D	Throttle discharge coefficient	0.345
γ	Ratio of heat capacities	1.4
r_e	Expansion ratio	11.5
β	Pressure ratio	
R	Specific gas constant	287 $J/Kg.K$
$\alpha(P_m, \omega_e)$	Volumetric efficiency	
η_c	Combustion efficiency	0.9
AFR	Air to fuel ratio	14.7
J_e	Engine inertia	0.20 $kg.m^2$
Q_{LHV}	Heat value of fuel	44 MJ/Kg
C_v	Heat capicity at specific volume	717 J/Km^2
mep	Mean effective pressure	$Pascal$
η_{atk}	Atkinson cycle engine's thermal efficiency	
$\dot{m}_{fv}$	Fuel vapor mass flow	kg/s
$\dot{m}_{ff}$	Fuel film mass flow	kg/s
$\dot{m}_{ff}$	Injected fuel mass flow	kg/s
T_{ind}	Engine indicated torque	N-m
T_{pump}	Engine pumping torque	N-m
T_{fric}	Engine frictional torque	N-m
T_l	Engine load torque	N-m
ω_e	Angular speed	rad/s
N	Angular speed	RPM
X_f	Injected fuel friction deposited on manifold as fuel film	0.22
τ_f	Time constant	0.25 s

model the VCR effect, the combustion chamber's height is changed at each engine load simulation in order to accomplish around the same maximum in-cylinder temperature and pressure.

Therefore, in the perspective of fuel consumption investigation, both the engine,s part load losses, thermal efficiencies, air mass flow rate, fuel mass flow rate and subsequently fuel consumptions are worked out for the typical NEDC, FUDS and FHDS driving cycles. The vehicle speed profiles for these driving cycles are as depicted in Figure 4.3. The procedure adapted for evaluation of fuel economy is elaborated comprehensively for NEDC driving cycle, however similar approach is opted for the rest of the driving cycles and concluded in view of fuel consumption

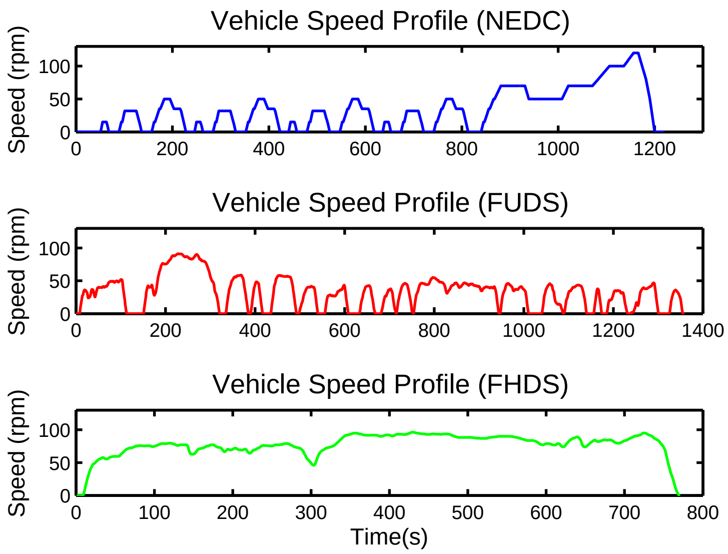

FIGURE 4.3: Vehicle speed profiles for the standard NEDC, FUDS and FHDS driving cycles

improvements . Generally, vehicle speed and the gear selection define the driving cycle. The parameters used to calculate the engine speed demand during driving simulations corresponding to an advanced car specifications having five speed gear ratios are as described in Table 4.2.

TABLE 4.2: Vehicle & VVT engine characteristics

Symbol	**Description**	**Value/Units**
	No. of cylinders	1
B	Engine Bore	70 mm
S	Engine Stoke	55 mm
V_d	Displaced volume	0.21166 dm^3
P	Maximum power	3.5kW @ 3000rpm
	Maximum torque	18N-m @ 2500rpm
r	Tire Radius	0.3 m
g_k	1^{st} gear ratio	3.827
	2^{nd} gear ratio	2.36
	3^{rd} gear ratio	1.685
	4^{th} gear ratio	1.312
	5^{th} gear ratio	0.90
G	Final drive ratio	3.8
	Idle engine speed	700 rpm
	IVO	20 BTDC $[CA]$
	IVC	32 ABDC $[CA]$
	EVO	40 BBDC $[CA]$
	EVC	12 ATDC $[CA]$

The study of fuel consumption at engine level requires engine speed in accordance with the vehicle speed. At every instant of vehicle driving cycle, the engine speed to be considered as a reference is calculated by using the following relation

$$\omega_e = \frac{v_{cyc}.G.g_k}{r} \tag{4.5}$$

where, g_k represents various gear ratios (i.e. 1^{st} to 5^{th}) depending on the vehicle speed, G is the final gear ratio and r is tire radius. The evolution of the engine speed profiles for the three driving cycles, used as a reference has maximum value of 3629 rpm in compliance with the maximum vehicle speed as shown in Figure 4.4. At WOT throttle, LIVC (IVT parameter) load control strategy is employed, whereas exhaust valve timings and intake valve opening have been considered unchanged in this work.

The comprehensive simulation results with engine speed profile as a reference of the closed loop VVT SI engine system using VCR characteristics to cope with the Atkinson cycle VVT engine, performance degrading aspect and various factors having vibrant role in the enhancement of fuel economy at 12Nm load conditions are elaborated here as a case. Finally, compact findings regarding the fuel consumption at different VVT engine operating conditions are presented. Figure 4.5, revealed the tracking capability of the designed PID controller for the VCR Atkinson cycle engine system. The results exhibit pretty good tracking for the NEDC driving cycle.

The flexible LIVC load control strategy has been employed in the EMVEM modeling approach of Atkinson cycle engine instead of the conventional throttle, which has a pivotal role in reduction of SI engine pumping losses (throttling effect) and enhancement in the thermal efficiency [2]. The control signal (IVT parameter λ) evolution throughout the NEDC driving cycle at 12Nm operating load is also

divulge in Figure 4.5. It shows evidence of quite natural intake valve closing timing variations in accordance with the VVT engine operating conditions.

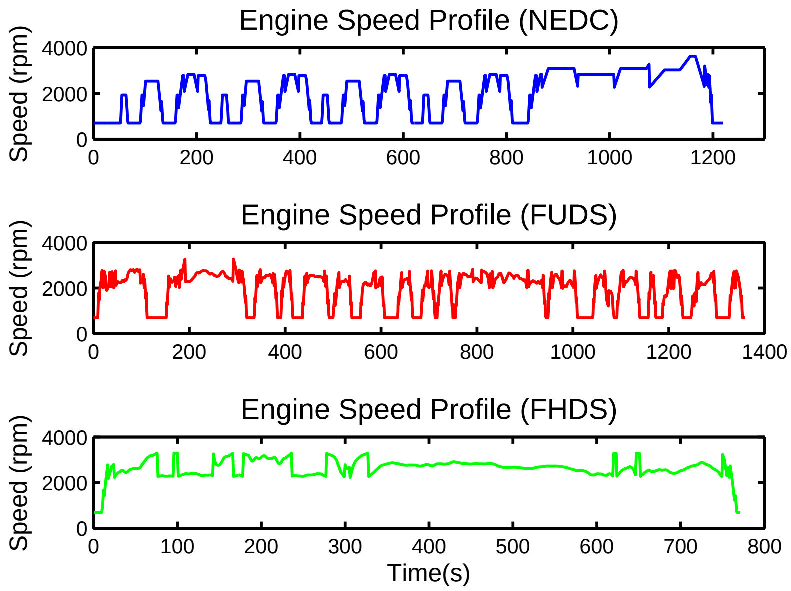

FIGURE 4.4: Engine speed profile in accordance with relative vehicle speeds for the NEDC, FUDS and FHDS driving cycles

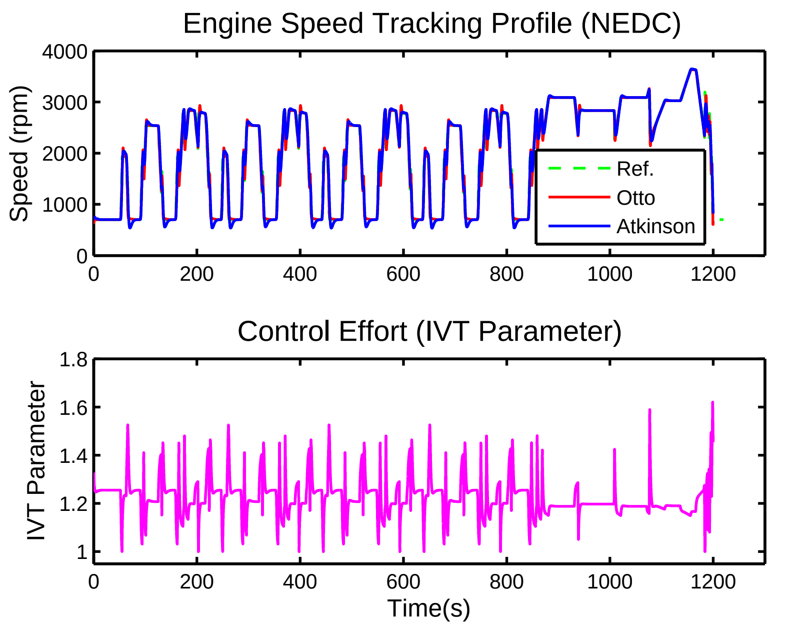

FIGURE 4.5: Engine speed reference tracking and, accordingly, the of evolution of the control effort profile with PID controller at 12Nm load condition during the NEDC driving cycle

A comparative view of the Atkinson cycle engine's pumping losses with that of the conventional Otto cycle during the complete driving cycle at 12Nm load, is illustrated in Figure 4.6. At part load significant decline in the throttling effect is achieved using EMVEM model over the Otto cycle engine model as a result of the LIVC load handling strategy. It can be seen that Atkinson cycle engine part load losses are comparable to WOT losses of the conventional engine. The reduction in pumping losses with LIVC strategy is predominantly the result of back-flow notion of the A/F charge into the intake manifold of the engine or it simultaneously turns into the intake of another combustion cylinder. Thus, suction pressure rests considerably above the atmospheric pressure during the first part of compression stroke, which is a key contributor in lessening of pumping losses.

Figure 4.7 demonstrates the thermal efficiency of the VCR Atkinson cycle engine in association with that of the conventional SI engine throughout the chosen driving cycle. A considerable enhancement in the EMVEM model's thermal efficiency can be seen, as engine load decreases further down the full load (wide open throttle). With the intrinsic LIVC load control approach around 5.63% improvement in the thermal efficiency is achieved at 12Nm load conditions. However, it can be

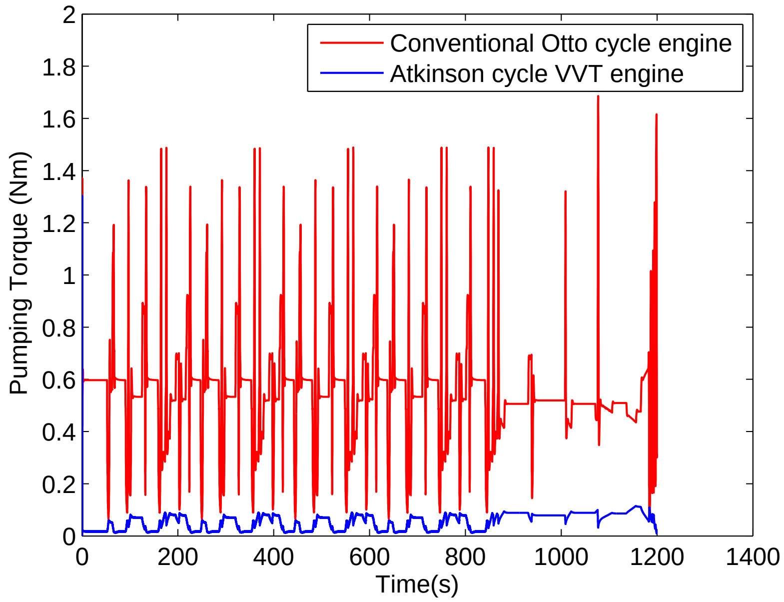

FIGURE 4.6: EMVEM model's pumping losses in comparison with MVEM conventional model during the NEDC driving cycle at 12Nm load conditions

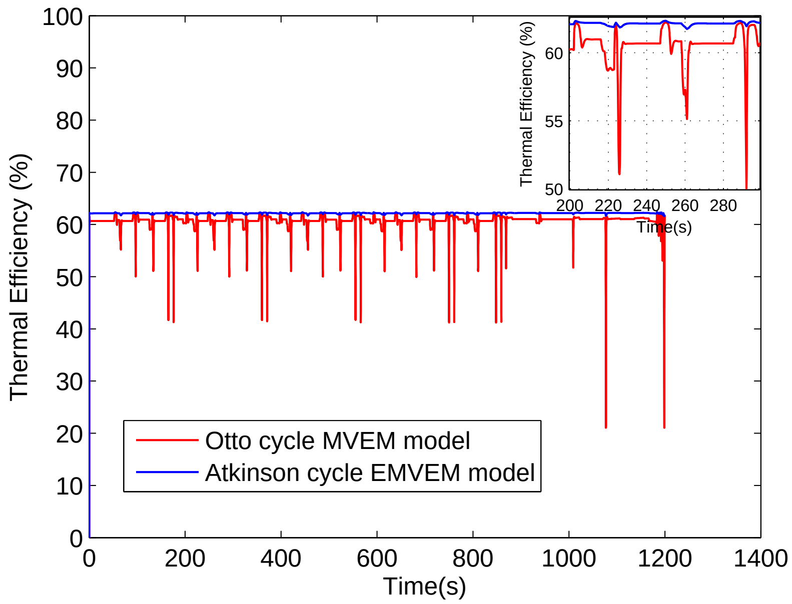

FIGURE 4.7: Thermal efficiency of the EMVEM versus that of the conventional Otto cycle MVEM during the NEDC driving cycle at 12Nm load conditions

augmented about 9.4% if it can be operated for lower loads [2, 46].

The comparison of air mass flow rate and fuel mass flow rate (instantaneous fuel consumption) of VCR Atkinson cycle engine as well as the conventional Otto cycle engine under similar operating conditions during the NEDC are presented in Figures 4.8 and 4.9 correspondingly. Significant reduction in fuel consumption can be seen as a consequence of the enormous decline in the throttling effect (engine pumping losses) and considerable enhancement in the thermal efficiency. By integrating the fuel mass flow rate for the complete NEDC cycle the total fuel consumed during the driving cycle can be obtained and 4.64% improvement in fuel economy is achieved for this particular case.

The fuel consumption comparison accomplished with VCR Atkinson cycle engine over the conventional Otto cycle engine at various engine operating conditions, for instance at 8Nm, 10Nm, 12Nm and 14Nm during the standard NEDC driving cycle, is revealed in Figure 4.10. It can be seen that improvement in the fuel economy of the Atkinson cycle engine goes on increasing towards the part load conditions in contrast with the conventional Otto cycle engine. The enhancement in fuel economy at higher load i.e. 14Nm load is 2.02%, whereas augmentation

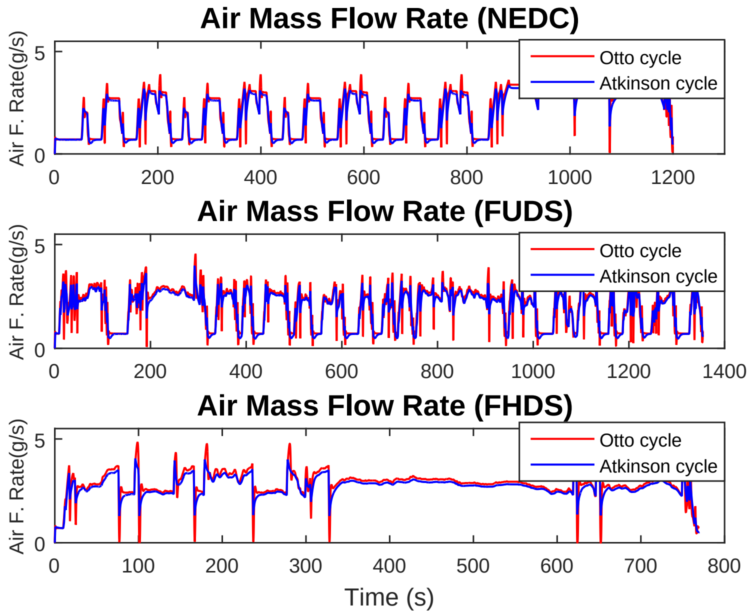

FIGURE 4.8: Air mass flow rate comparison of the Atkinson cycle engine and conventional Otto cycle engine during the NEDC, FUDS and FHDS driving cycles at 12Nm load conditions correspondingly

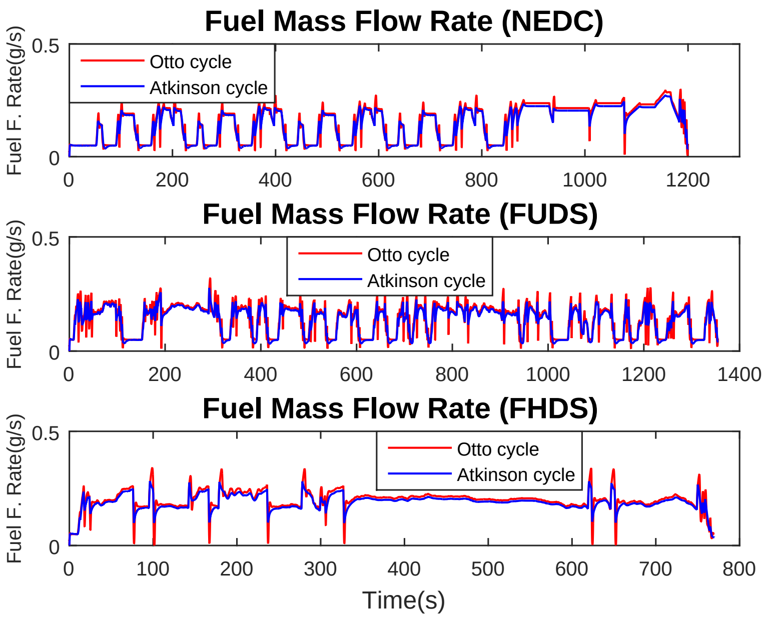

FIGURE 4.9: Fuel mass flow rate comparison of the Atkinson cycle engine and conventional Otto cycle engine during the NEDC, FUDS and FHDS driving cycles at 12Nm load conditions respectively

in fuel economy at part load i.e. 8Nm is around 10.87%. However, on the average around 3.57% enhancement in the thermal efficiency, whereas about 6.22% improvement in the fuel consumption with VCR Atkinson cycle engine over the conventional Otto cycle engine at medium and higher engine operating conditions during the NEDC is attained. Likewise, fuel consumption of the VCR Atkinson cycle engine in comparison with the conventional Otto cycle engine under similar engine operating conditions during the standard FUDS and FHDS driving cycles are divulged in Figures 4.11 and 4.12 accordingly.

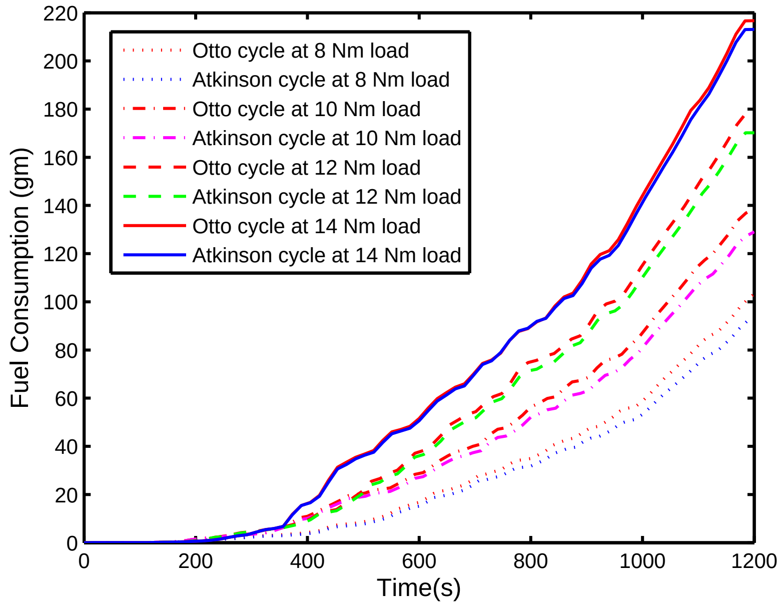

FIGURE 4.10: Fuel Consumption evaluation at various engine operating condition during NEDC

The results comprising of engine operating conditions, thermal efficiency and fuel economy improvements of Atkinson cycle VCR engine over the conventional Otto cycle engine during the standard NEDC, FUDS and FHDS driving cycles are summarized in Table 4.3. The brief simulation outcome exhibits that on the average about 3.87% enhancement in the thermal efficiency and around 7.17% improvement in the fuel consumption with VCR Atkinson cycle engine over the conventional Otto cycle engine at medium and higher load operating conditions during the FUDS are accomplished. The improvement in the thermal efficiency

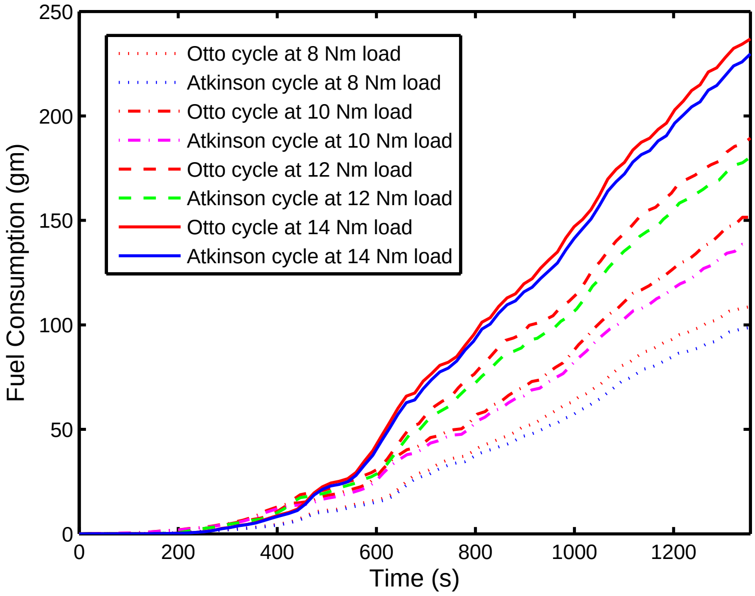

FIGURE 4.11: Fuel Consumption evaluation at various engine operating condition during FUDS

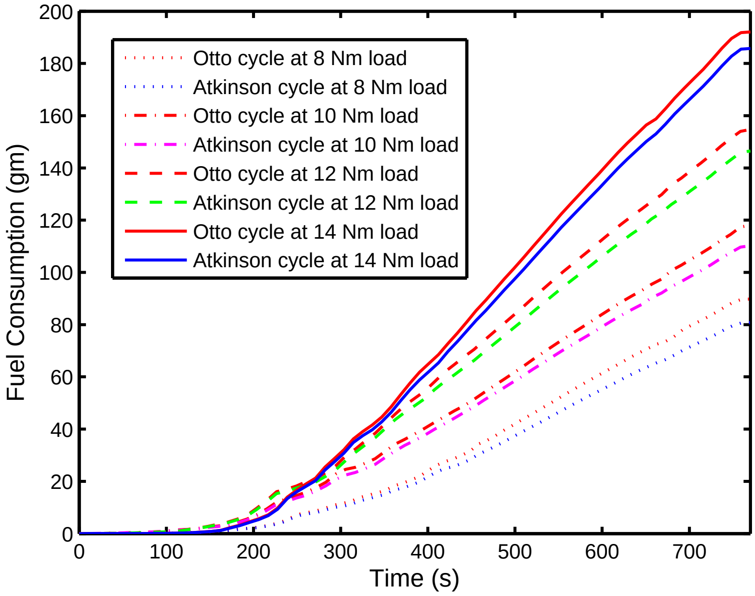

FIGURE 4.12: Fuel Consumption evaluation at various engine operating condition during FHDS

Table 4.3: Fuel Economy Improvements over Otto cycle engine during the New European Driving Cycle (NEDC), Federal Urban Driving Schedule (FUDS) and Federal Urban Driving Schedule (FHDS)

Engine operating conditions	**Comp-ression ratio**	**NEDC**		**FUDS**		**FHDS**	
		Thermal eff. enhanc. (%)	**Fuel economy improv.** (%)	**Thermal eff. enhanc.** (%)	**Fuel economy improv.** (%)	**Thermal eff. enhanc.** (%)	**Fuel economy improv.** (%)
8Nm	14.65	5.63	10.87	7.19	11.30	5.60	10.90
10Nm	14.35	3.89	7.36	3.34	9.31	2.86	7.16
12Nm	13.80	2.52	4.64	3.15	4.99	2.00	4.50
14Nm	13.15	2.24	2.02	1.82	3.13	1.67	2.05
Average improvement		**3.57**	**6.22**	**3.87**	**7.17**	**3.03**	**6.15**

as well as fuel consumption during FUDS is more than that of NEDC and FHDS driving cycles as FUDS driving cycle is inclined more towards the engine part loads as compared to NEDC and FHDS. The VCR Atkinson cycle VVT engine is always more efficient in comparison with the conventional Otto cycle engine over the wide operating range. To avoid the effect of LIVC on ignition delay and engine combustion duration, LIVC can be physically realized for medium and higher loads. Below 40% of the full engine load, the use of throttle may be required as a substitute, without losing the potential benefits of the integrated VVT system [53].

4.5 Conclusion

This research outlines the design and development of a control framework for the Atkinson cycle engine with flexible intake valve load control strategy. The control framework based on physically motivated control-oriented EMVEM model of the Atkinson cycle engine is evaluated for NEDC, FUDS and FHDS driving cycles at medium and higher operating conditions to ensure its better fuel economy in comparison with the conventional SI engine. The part load losses of the Atkinson cycle engine figured out, are comparable to the WOT losses of the conventional SI engine, whereas a considerable boost in the thermal efficiency is accomplished during NEDC, FUDS and FHDS. Both the aforesaid improvements play vital role in reducing the VVT engine's fuel consumption. On the average around **6.22%**, **7.17%** and **6.15%** improvement in fuel economy with VCR Atkinson cycle engine over the conventional Otto cycle engine during NEDC, FUDS and FHDS, is achieved correspondingly. The Atkinson cycle engine is proved significantly efficient than the conventional SI engine, especially for the engine part load operating range.

However, in this work, to deal with challenging speed control problem, a conventional PID controller was employed which endures immensely with limitations such as speed tracking and robustness in the presence of auxiliary loads and uncertain road loads for the various driving cycles. A considerable tracking error is figured out particularly for the FUDS and NEDC driving cycles. Therefore, we plan to evaluate the developed control framework at vehicle level with the robust controller, in presence of an auxiliary loads and an uncertain road loads obtained by standard driving cycles. In the succeeding chapter, a robust higher order sliding mode control (HOSMC) for an Atkinson cycle engine is designed for the pursuance of its fuel economy.

Chapter 5

Higher Order Sliding Mode Control of Atkinson Cycle Engine

Over the last three decades, stringent government conventions aimed at improving fuel economy and reducing emissions, there has been a dramatic evolution in the power-train control system.These regulations call for a paradigm shift in the way the engine speed is controlled. These performance requirements can be potentially met with the introduction of new design parameters by means of innovative mechanical mechanisms i.e. new actuators. The control variables, design parameters in system lingo, provide additional degrees of freedom to optimize the performance of the SI engine during its wide operation range.

5.1 Prevalent Approaches to SI Engine Control

The control of the automotive SI engine equipped with the innovative technologies is currently an active research area [30, 116]. In an automotive application, engine speed control is a challenging classical nonlinear problem. Engine speed performance has significant impact on the vehicle design attributes as fuel economy, emission and comfort i.e. drivability, especially during the traditional vehicle operations [30–32, 110, 111]. Murtaza et al. [8] have developed an alternative control framework for the Atkinson cycle SI engine with unconventional late intake valve closing (LIVC) load control strategy instead of usual throttle, in the perspective of fuel economy evaluation for the standard NEDC, FUDS and FHDS driving cycles. A control framework for the suggested model is proposed. In this work, to deal with speed control problem a conventional PID controller is employed which endures immensely despite limitations such as speed tracking and robustness in the presence of auxiliary loads and uncertain road loads for the various driving cycles. A sizable speed tracking error is figured out particularly for the FUDS and NEDC driving cycles.

Therefore, an efficient controller need to be designed for this purpose as the prime objective of engine control system is not only to maintain the desired speed in the presence of ever-present load torque disturbances but also exhibits robustness against the modeling inaccuracies, long-term variations in the engine parameters, speed-dependent time delays and changes experienced in the surrounding variables as temperature, besides the aforementioned performances [29, 30, 113, 117]. Sliding mode control approach has shown good robustness properties and has ability to cope nonlinear and uncertain issues [118, 119]. In this connection, a multi-surface adaptive sliding mode control algorithm for a two-state engine model with throttle angle as a control variable, for speed control of SI engine is proposed by Choi and Hedrick [30]. However, engine speed, throttle body airflow, throttle position and manifold pressure and temperature measurements are required for the resulting algorithm. The objective of this research was to develop a robust throttle control strategy for all wire-by-wire applications as autonomous vehicle control etc. The researchers in [120–122] have developed dynamic sliding mode control design schemes using nonlinear engine models with uncertainties for both measurement feedback as well as state feedback. Dynamic sliding mode control design approach has been considered for speed control of four-stroke SI engine, a commercially available simulation-benchmark model was adapted for this work. The study seeks to design a speed controller which exhibits good disturbance rejection properties and robust to uncertainties [112]. A sliding mode controller for idle speed control using a linear model for the design and taking air bypass valve and spark advance as inputs, has been proposed by Bhatti et al. [123]. In this research, to reconstruct the system state for use by the controller an observer was designed. Whereas, [113] has employed 2-sliding super-twisting algorithm for the conventional engine speed control based on the conventional MVEM model of the SI engine. The SMC approach, demonstrates robustness of the closed-loop system against parameter variations, changes in the load torque and initial speed. The work in [29] has also addressed speed control problem of the SI engine by using a proportional feedback controller scheme as well as the stability analysis of the proposed control system by considering its transient performance. Feedback linearization strategy has been employed for the MIMO conventional engine model [124]. The authors in [122, 125] have studied from theoretical view the output feedback stabilization of SISO as well as MIMO nonlinear systems in differential input-output form using dynamic sliding mode. Appropriation of asymptotic stability through such schemes has been advised for the control of differential I-O form, which may not be dynamic feedback linearizable. The investigations also concluded that the nonlinear system does not require to be presented in a regular form, nor it need to have full relative order like the most of the other sliding mode methods to exhibit linearizability. This research aimed at the application of the strategy to automotive engine control. Using mixed sliding mode techniques, complete control of MIMO three state SI engine MVEM model [73] was proposed in [117] and for design purpose represented the model in local generalized controller canonical form (LGCCF). The authors applied conventional sliding mode control by choosing error in A/F ratio as a sliding surface, whereas employed dynamic sliding control for the remaining two stated i.e. speed and manifold pressure.

However, most of the prior studies in the assessment of the over-expanded cycle engine incorporated with the innovative mechanisms regarding fuel economy, have taken into account either the conventional Otto cycle MVEM model or CCEM models of the VVT engines only. Whereas, CCEM has a number of limitations such as, complexity and difficulty in analysis and control design. Development and evaluation of any robust control strategy based on the control-oriented EMVEM model of the Atkinson cycle engine as reported in this chapter is not present in the literature.

The research in [9] seeks to demonstrate sliding mode strategy, for speed control of the Atkinson cycle engine through an alternative breathing (LIVC) control channel, using super-twisting algorithm. The control framework is based on EMVEM model of the Atkinson cycle engine and is evaluated in the view of fuel economy for the standard FUDS and US06 driving cycles. A control-oriented EMVEM model of the Atkinson cycle engine with variable intake valve actuation as a load control channel having relative degree one has been, developed by Murtaza et al. [2] is utilized for this study. The simulation results exhibit the considerable decline in engine part load losses, enhancement in the thermal efficiency and, accordingly, confirm improvement in the fuel economy of the VCR Atkinson cycle engine.

In this chapter, a brief overview of the sliding mode control (SMC) and higher order sliding mode control (HOSMC) as a background to understand the HOSMC design for the Atkinson Cycle Engine are presented. In addition, simulation results and discussions in the viewpoint of the VCR Atkinson cycle engine fuel economy achieved over the conventional Otto cycle engine, are comprehended and finally concluded.

5.2 Sliding Mode Control

Sliding mode control is known as a variant of variable structure system (VSS). Usually, a variety of structures, may be unstable on their own, in different portions of state-space of closed loop system constitute VSS. Sliding mode control has significant advantages over the robust control techniques such as robustness to uncertainties, better convergence rate, parametric variations and input noises. SMC techniques have been efficiently used for control system design with fast convergence rate, insensitive to modeling errors, system parametric variation and external disturbances abanovi and Asif [126], Alwi and Edwards [127], Rizzoni et al. [128], Levant [129], Spurgeon [130] and Shtessel et al. [131]. To ascertain a sliding motion this technique is based on keeping exactly a properly selected constraint (sliding manifold) by means of high frequency control switching. This switching control is used to enforce the system trajectory to move along the desired sliding manifold to ensure switching between the different structures and to achieve the desired system performance.

To establish the desired sliding motion, the behavior of the control system is governed by the designed sliding surface. In SMC, a discontinuous control signal is produced which enforces the system trajectories to repeatedly cross and then re-cross the sliding surface until the deviation from sliding manifold becomes zero and it finally slides along the surface, the associated motion is termed as sliding motion and corresponding system's state is named sliding mode. The two stages; reaching phase as well as sliding phase establish sliding mode control. System trajectories are driven to a stable manifold with the help of appropriate control law in the reaching phase, whereas in sliding phase, the trajectories of the system slide to an equilibrium point, following prescribed constraints. The closed loop system confined to the sliding surface behaves as a reduced order system and becomes insensitive to uncertain parameters of the plant resulting in SMC system robust against external disturbances and parametric variations with high accuracy. In sliding mode, due to the control signals discontinuous behavior, the states of the system switch about the sliding surface rather than lying directly on it [132]. Consequently, highly undesirable for any physical plant or actuators, a high frequency switching phenomenon termed as chattering exhibits, as it can excite un-modeled plant dynamics. Another shortcoming of SMC technique is that it can only be suitable for those control systems having relative degree one or below with respect to the sliding variable. The sliding motion, boundary layer with linked sliding system trajectory along the switching surface, is illustrated in the Figure 5.1 and Figure 5.2 accordingly.

5.3 The Chattering Phenomenon

First order sliding mode controller induces highly undesirable phenomenon so called chattering effect, which can result in early wear out of system components and actuators and thus reducing the reliability and life cycle of real systems. Several approaches have been developed to avoid this chattering effect in the literature such as saturation

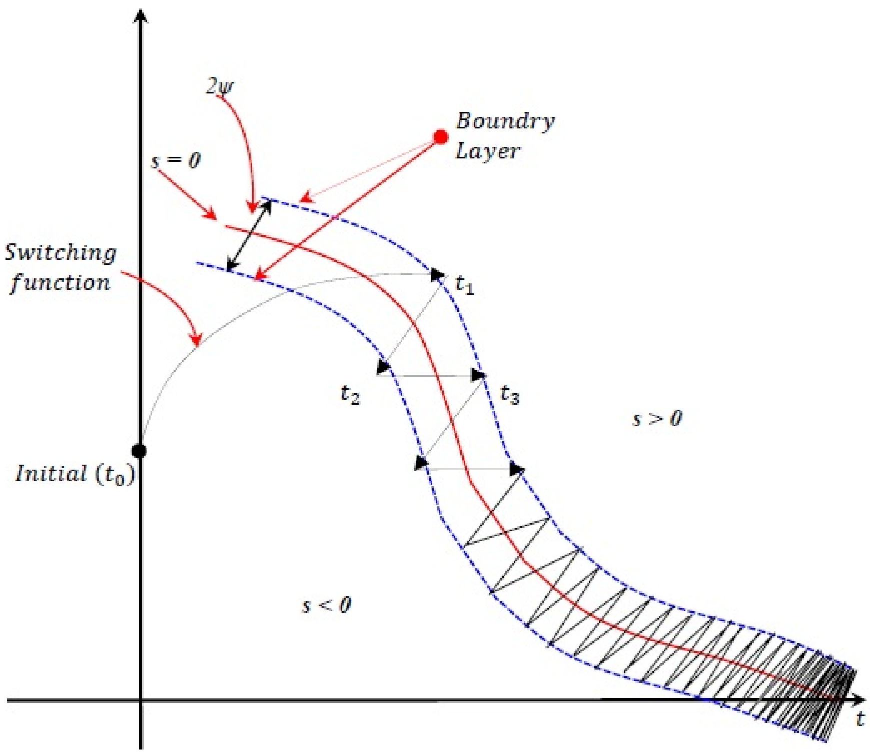

FIGURE 5.1: Switching motion of the system, switching is bang-bang type both inside and outside the boundary layer

approximation [133], higher order sliding mode control [134], dynamic sliding mode control Sira-Ramirez [135], and equivalent control, Utkin [136]. In [137], saturation approximation is illustrated as for instance sign function is replaced with saturation approximation function i.e. by changing the dynamics within small vicinity of discontinuous surface, a smooth transition is proposed near the switching surface. Mathematically, it can be described as:

$$sat(s,\psi) = \begin{cases} sign(s/\psi), & \text{for } |s/\psi| > 1 \\ s/\psi, & \text{for } |s/\psi| \le 1 \end{cases} \tag{5.1}$$

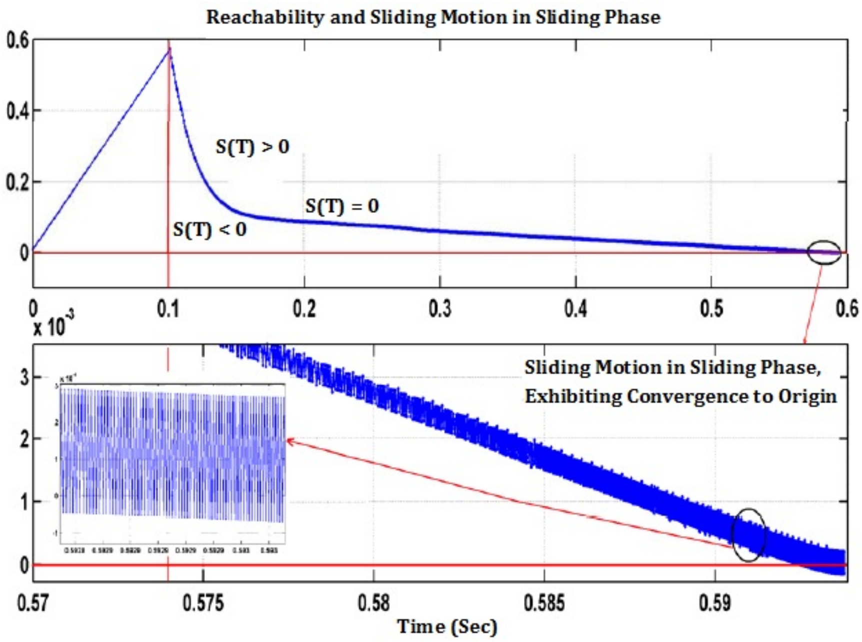

FIGURE 5.2: Illustration of reachability and sliding phase of the sliding mode

The reduction in chattering depends on $\psi \leq 1$, smaller the value of ψ, lesser will be chattering but, at cost of robustness. This technique is commonly used to avoid chattering and have an extensive application to the many practical problems Utkin et al., [136], Iqbal et al., [101].

5.4 Higher Order Sliding Mode Control

The traditional sliding mode control theory exhibits high accuracy and robustness against parametric variations, un-model dynamics, internal and external disturbances. Whereas, this control technique has demerits such as chattering phenomenon associated with high frequency switching of the controlled system, which leads to wear out of actuators as well as system itself. Another constraint of the conventional SMC is that it can only be used for systems with relative degree one. The control researchers have worked out and incorporated new approaches in view of extension in depth and breadth in SMC techniques near a half century. These extension comprise of higher order sliding mode control (HOSM) [127], terminal sliding mode control (TSMC) [138], dynamic sliding mode control (DSMC) [135], dynamic integral sliding mode control (DISMC) Khan et al., [139] and Integral sliding mode control [134]. Such strategies provide better accuracy, besides the chattering removal while retaining the main advantages of SMC.

The key inspiration of higher order sliding mode is to act on higher order derivatives of sliding surface instead of the 1st order derivative in the conventional SMC approach to reach and stay on the switching surface for all future times. This results in chattering attenuation and higher order precision. So higher order sliding mode control provides the constraints for system with relative degree greater than one, additionally removes the chattering effects of standard SMC. The exact tracking of the constraint function is achieved by infinitely high frequency switching of the control signal in conventional and higher sliding modes design. In fact in HOSM design, trajectories of system move on the discontinuous surface of closed loop dynamic system in Filippov sense, Filippov [140]. In 1st order SMC trajectories of system most of the time hit on the sliding manifold more or less vertically as illustrated in Figure 5.3a. whereas, the trajectories of system strike the manifold tangentially, reducing the chance for trajectories to leave the manifold afterward in HOSM as demonstrated in Figure 5.3b,

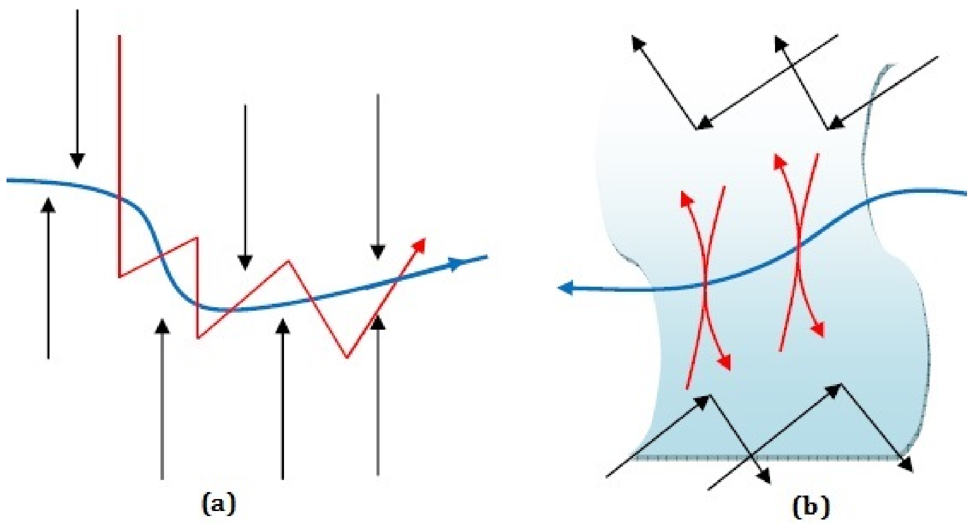

FIGURE 5.3: Trajectories intruding on the sliding manifolds (a) in conventional SMC, (b). in HOSMC

HOSM algorithms particularly the second order sliding mode algorithms [141] such as Real Twisting Algorithm, Super Twisting Algorithm, Drift Algorithm and Hybrid 2-sliding Algorithm may be used to avoid chattering if relative degree of dynamic system is one, But higher order sliding mode is mandatory if the dynamic system have relative degree two or higher, otherwise to estimate the value of derivative of switching manifold an observer will be required.

Definition 3.1 [142]: *The sliding order r is the number of continuous total derivatives (including the zero one) of the function* $\sigma = \sigma(x,t)$*, whose vanishing defines the equations of the sliding manifold.*

Definition 3.2 [141]: *The sliding set of the rth order associated to the manifold* $\sigma(x,t) = 0$ *is defined by the inequalities*

$$\sigma = \dot{\sigma} = \ddot{\sigma} = ... = \sigma^{r-1} = 0 \tag{5.2}$$

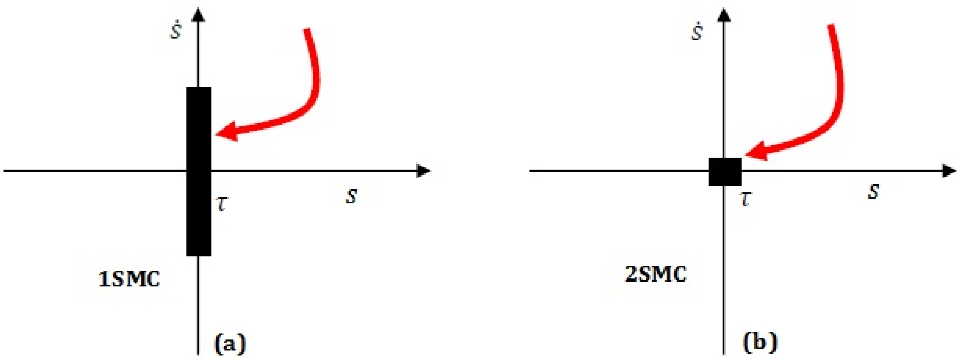

FIGURE 5.4: Demonstration of sliding accuracy in sliding mode (a) conventional sliding mode, (b) 2nd order sliding mode

Definition 3.3 [142]: In a simplified way, the relative degree r means that the control appears explicitly only in the r-th total time derivative of sliding variable σ.

5.4.1 Second Order Sliding Mode Algorithms

To constraint the switching function and its first derivative on a certain manifold in the state space of the closed-loop system is the prime objective of the second order sliding mode (2SMC). Typically, 2SMC are employed to evade chattering effects by zeroing the systems outputs with relative degree one as well as controller design for systems having relative degree 2 with respect to its output constraint function

[142]. In literature, the most often used 2SMC algorithms are described here in brief.

5.4.1.1 Real Twisting Algorithm

Real twisting algorithm is the first recognized second order sliding mode algorithm [101]. The trajectories of system twist around origin of the 2nd order sliding manifold perform infinite rotations and then finally converge to origin within finite time in this algorithm. The algorithm provides good robustness properties along with finite convergence times with reduced chattering effect. Real twisting algorithm can be applied for the dynamical systems having relative degree one and two. However, sampling time sensitivity is the limitation of this algorithm. The control law for the system with relative degree one, can be expressed as [142]:

$$\dot{u} = \begin{cases} -u, \quad \text{if} \ \ |u| > 1 \\ -\mu_m \text{sign}(\sigma), \quad \text{if} \ \ \sigma\dot{\sigma} \leq 0; |u| \leq 1 \\ -\mu_M sign(\sigma), \quad \text{if} \ \ \sigma\dot{\sigma} > 0; |u| \leq 1 \end{cases} \tag{5.3}$$

where,

$\mu_M > \mu_m > 0$

$\mu_M > 4K_M/\sigma_o, \ \mu_m > C_o/K_m$

and

$K_m\mu_M - C_o > K_M\mu_m + C_o$

The availability of the sign of the sliding variable $\dot{\sigma}$ is needed for the computation of $\dot{u}$ at each axis crossing. As $\dot{\sigma}$ is not available for some time so $sign(\dot{\sigma})$ is replaced with $sign(\Delta\sigma)$. With $\Delta\sigma = \sigma(t) - \sigma(t_i - t_{i-1})$ and $t \in [t_i - t_{i-1}]$. The Eq (5.3) can be described as:

$$\dot{u} = \begin{cases} -u, \quad \text{if} \quad |u| > 1 \\ -r_1\text{sign}(\sigma) - r_2\text{sign}(\dot{\sigma}), \quad \text{if} \quad |u| \leq 1 \end{cases} \tag{5.4}$$

where,

$r_1 > r_2 > 0,$

$\mu_M = r_1 + r_2$,

$\mu_m = r_1 - r_2$.

The control law for relative degree 2 can be written as:

$$\dot{u} = \begin{cases} -K_m sign(\sigma), & \text{if } \ |u| > 1 \\ -K_M sign(\sigma), & \text{if } \ |u| \leq 1 \end{cases} \tag{5.5}$$

The illustration of real twisting algorithm phase portrait is shown in Figure 5.5.

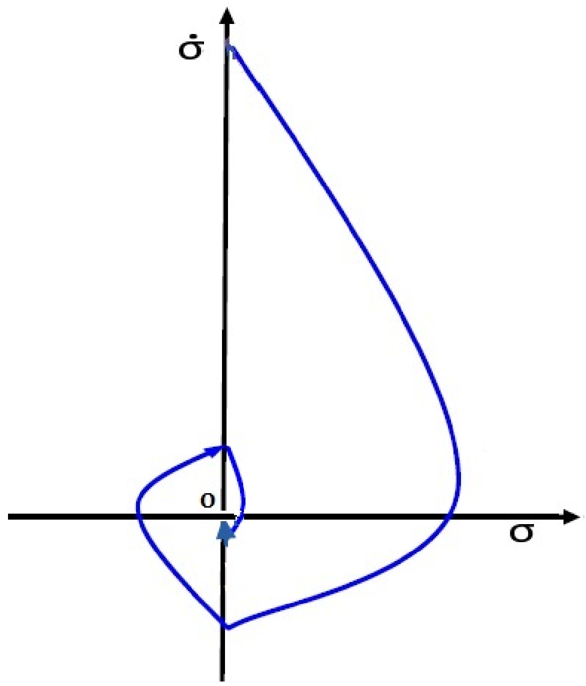

FIGURE 5.5: Phase plane of Real Twisting Algorithm

5.4.1.2 Super Twisting Algorithm

In this algorithm, the trajectories of system also twist around origin of the 2nd order sliding manifold and then finally converge to origin within finite time. The foremost advantage of this algorithm over the rest of 2nd order algorithms is that it does not need sliding manifold derivatives, besides the sampling time sensitivity. Super twisting algorithm can be used for the systems with relative degree one, removes chattering and in-addition, exhibits good robustness properties along with finite convergence times. The control law of super twisting algorithm is primarily linear combination of continuous and discontinuous function of switching variable. It can be described as [142]:

$$u(t) \quad = \quad u_1(t) + u_2(t) \tag{5.6}$$

$$\dot{u_1} = \begin{cases} -u, & \text{if } \ |u| > 1 \\ -\xi sign(\sigma), & \text{if } \ |u| \leq 1 \end{cases} \tag{5.7}$$

$$u_2(t) = \begin{cases} -\mu|\sigma_0|^\rho sign(\sigma), & \text{if } \ |\sigma| > |\sigma_0| \\ -\mu|\sigma|^\rho sign(\sigma), & \text{if } \ |\sigma| \leq |\sigma_0| \end{cases} \tag{5.8}$$

where, $\xi > \mu > 0$, $0 < \rho < 1$, and the condition $|u| = |u_1(t_o) + u_2(T - o)| \leq \xi$ is used to chose the initial values of $u_1(t_o)$. The modified form of this algorithm can also be written as:

$$u \quad = \quad -\mu|\sigma|^\rho sign(\sigma) + u_1 \tag{5.9}$$

The graphical view of the super twisting algorithm is given as in Figure 5.6

In the following Section, higher order sliding mode control is designed for the Atkinson cycle engine by utilizing control-oriented EMVEM model [2] with fuel dynamics [71] as given in Ch. 3 and Ch. 4, for the evaluation of fuel economy of the VCR Atkinson cycle engine in comparison with conventional SI engine. Comprehensive engine model details along with the engine specifications are given in the preceding chapters. In this work, late intake valve closing load control strategy is employed, instead of the conventional throttle. IVT parameter λ at WOT throttle is considered as a control variable and angular speed as an output while evaluating fuel consumption.

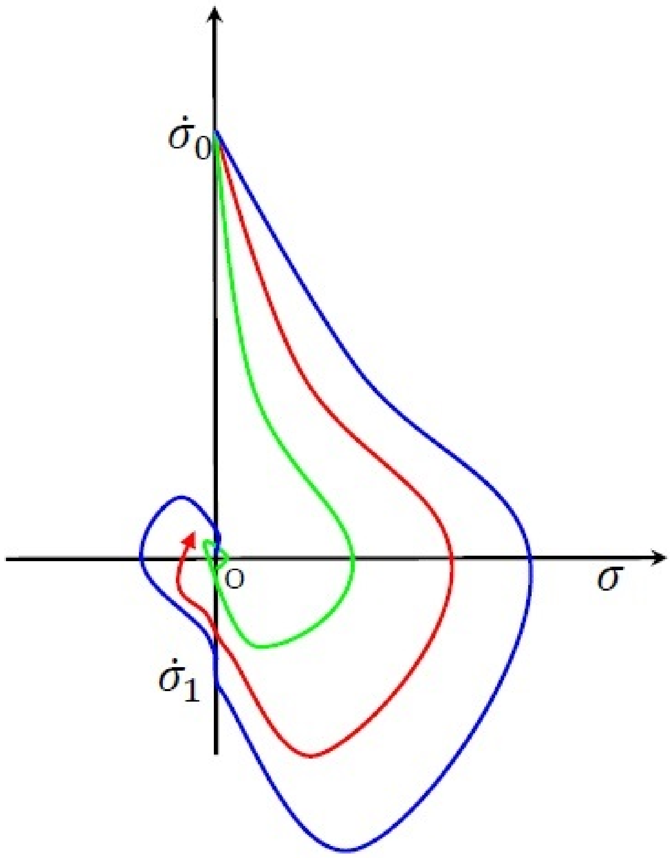

FIGURE 5.6: Phase plane of Super Twisting Algorithm

5.5 HOSM Control Design for the Atkinson Cycle Engine

To design second order sliding mode control for the Atkinson cycle engines based on nonlinear EMVEM model [2, 71] with an alternative breathing control strategy, its design dynamics is described as:

$$\dot{x} = f(x) + g(x, u) \tag{5.10}$$

where, $x = [Pm, w_e, \dot{m}_{ff}]^T \in \Re^3; u \in \Re; f : \Re^3 \to \Re^3; g : \Re^3 \to \Re^3;$ $f(x)$ and $g(x, u)$ being smooth vector fields (see Appendix B1), where u is control input (IVT parameter 'λ'). As the third term in the expression of η_{atk} in the model as given in Eq (3.35), during the engine entire operating conditions (within the constraints on IVT parameter i.e. $1 \leq \lambda \leq 1.60$) have utmost value approximately

0.004, for the computation of the controller design parameters only this term has been ignored for simplicity.

5.5.1 Control Objectives

Establishing the control objective and, accordingly, defining the switching surface is the second stage in the control design practice. In this work, evaluation of the potential advantages of the VCR Atkinson cycle engine with variable LIVC load handling strategy in terms of fuel economy over the conventional Otto cycle SI engine at medium and higher load conditions is the proposed objective. The anticipated aim can be achieved by controlling the engine speed through a controller robust against the disturbances, uncertainties in the model parameters and unmodeled system dynamics, which has pivotal significance in the accomplishment of optimum engine fuel consumption as well as vehicle drivability.

In the variable structure control (VSC) theory framework, the switching surface (control objective) can be described as:

$$\sigma(t,x) = \omega_{eref} - \omega_e \tag{5.11}$$

where ω_{eref} is the engine reference speed and σ is the switching variable must steered to zero.

5.5.2 Controller Design

The sliding surface σ has relative degree one with respective to the control input u (i.e. IVT parameter λ) as u does not appear explicitly in the sliding surface, whereas, it appears in the $\dot{\sigma}$ expression. It is assumed that information about σ is available. Consequently, by differentiating the sliding surface twice; the following relations are derived:

$$\dot{\sigma} = \frac{\partial}{\partial t}\sigma(t,x) + \frac{\partial}{\partial x}\sigma(t,x).\,(f(x) + g(x,u)) \tag{5.12}$$

$$\ddot{\sigma} = \frac{\partial}{\partial t}\dot{\sigma}(t,x) + \frac{\partial}{\partial x}\dot{\sigma}(t,x,u).\left(f(x)+g(x,u)\right) + \frac{\partial}{\partial u}\dot{\sigma}(t,x,u).\dot{u}$$

$$= \varphi(t,x,u) + \xi(t,x,u).\dot{u} \tag{5.13}$$

The above-mentioned expressions details are specified in the Appendix B2.

Afterward, through the EMVEM model analysis, the essential conditions for designing the second order sliding mode control (SOSMC) fulfilled by dynamics Eq (5.10) are demonstrated, which are given as follows.

- The control signal belong to the set $U = \{u : |u| \leq U_m\}$, with $U_m \in \Re$.
- There exists $u_1 \in (0,1)$ such as for any continuous function u with $|u(t)| > u_1$, there exist $\sigma(t)u(t) > 0$ for $t > t_0$. Therefore, the control effort $u(t) = -sign\left(\sigma(t_0)\right)$, where t_0 is the initial time value, provides hitting the sliding surface $\sigma = 0$ in finite time.
- For the given $\dot{\sigma}$, there exist $+Ve$ constants $\sigma_0, u_0 < 1$, Γ_m, Γ_M, such as if $|\sigma| < \sigma_0$ then

$$0 < \Gamma_m \leq \frac{\partial}{\partial u}\dot{\sigma}(t,x,u) \leq \Gamma_M, u \in U, x \in X. \tag{5.14}$$

  The bounds Γ_m, Γ_M are determined from the detailed analysis and the comprehensive simulation studies of the EMVEM dynamics for medium and higher engine operating conditions for $\sigma_0 = 4.6e^{-5}$ as follows:

$$\Gamma_m = 45.3 \qquad \Gamma_M = 70.6 \tag{5.15}$$

- Furthermore, a $+Ve$ constant $\Phi = 4.197$ has been computed, as within the region $|\sigma| < \sigma_0$, for every time $u \in U, x \in X.$, the following inequality holds:

$$\left|\frac{\partial}{\partial t}\dot{\sigma}(t,x) + \frac{\partial}{\partial x}\dot{\sigma}(t,x,u).\left(f(x)+g(x,u)\right)\right| \leq \Phi \tag{5.16}$$

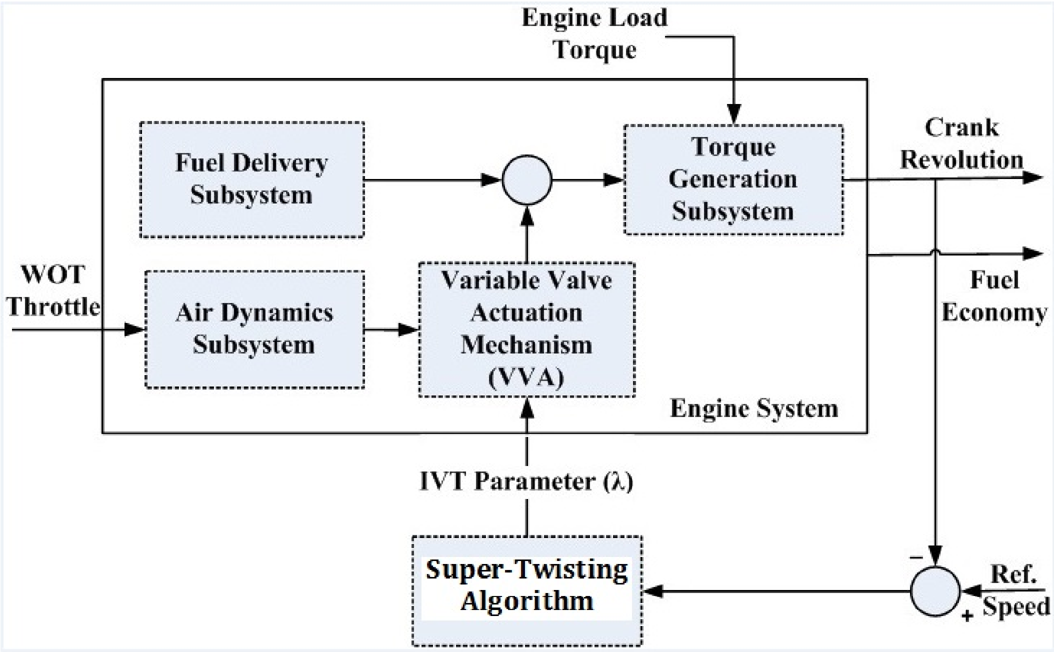

FIGURE 5.7: Schematic diagram of closed loop Atkinson cycle VVT engine system [9]

Thus, with the solutions of the following equivalent differential inclusion by applying SOSMC, the control problem of the system Eq (5.10) with dynamics Eq (5.13) can be solved:

$$\ddot{\sigma} \in [-\Phi, \Phi] + [\Gamma_m, \Gamma_M]\, \dot{u}. \tag{5.17}$$

Finally, the controller parameters can be designed, by using the information of the global bounds of $\varphi(.)$ and $\xi(.)$. A schematic illustration of the developed control framework is appreciated in Figure 5.7.

Accordingly, a control law based on the super-twisting algorithm, the most popular algorithms among the SOSM algorithms is developed [131, 143]. By establishing, the conditions Eq (5.14) and Eq (5.16), the main advantages of this algorithm relies on its robustness to disturbances and parametric uncertainties. As the engine EMVEM model has relative degree one, so the control law can be defined as combination of discontinuous time derivative and continuous function of the sliding surface:

$$u(t) = u_1(t) + u_2(t) \tag{5.18}$$

$$\dot{u}_1(t) = -\xi sign(s) \tag{5.19}$$

$$u_2(t) = \begin{cases} -\mu|\sigma_0|^\rho sign(\sigma), & \text{if } |\sigma| > |\sigma_0| \\ -\mu|\sigma|^\rho sign(\sigma), & \text{if } |\sigma| \leq |\sigma_0| \end{cases} \tag{5.20}$$

where ξ, μ and ρ are controller design parameters. For finite time convergence to the switching manifold the corresponding sufficient conditions are:

$$\xi > \frac{\Phi}{\Gamma_m}$$

$$\mu^2 \geq \frac{4\Phi}{\Gamma_m^2}\frac{\Gamma_M(\xi + \Phi)}{\Gamma_m(\xi - \Phi)}$$

$$0 < \rho \leq 0.5 \tag{5.21}$$

By satisfying the above-mentioned conditions, the controller parameters are designed to ensure the establishment of the SOSMC in finite time. Keeping in view to have low content of high frequency components in the control, the super-twisting controller parameters are selected. The most adequate set of controller parameters chosen as a result of an iterative refining procedure are as follows:

$$\xi = 0.085 \quad \mu = 0.085 \quad \rho = 0.5$$

In the following subsection, fuel economy of the VCR Atkinson cycle engine over conventional SI engine during wide engine load range is examined. It is to assume that the air to fuel ratio is being controlled/ maintained at stoichiometric ratio by an independent robust controller. At WOT throttle, LIVC (IVT parameter) load control strategy is employed, whereas exhaust valve timings and intake valve opening variations have not been considered in this work.

5.6 Evaluation of the Robust Control Framework

To study the potential benefits of the VCR Atkinson cycle engine with its intrinsic variable LIVC load handling strategy instead of the conventional throttle, a second order sliding mode control (SOSMC) framework designed for the Atkinson cycle engine is evaluated by using its already developed control-oriented EMVEM model. This research is conducted to assess fuel economy of the VCR Atkinson cycle engine over the conventional Otto cycle engine, at the medium and higher load operating conditions, during the standard FUDS and US06 driving cycles. In this context, both the engine models i.e. control-oriented EMVEM model of Atkinson cycle engine as well as conventional SI engine MVEM model are simulated by using a precisely tuned SOSM super-twisting controller having similar specifications. The notion of VCR mechanism is used to cope with the causes of the innovative mechanisms incorporated in the Atkinson cycle VVT engines. To model the VCR effect, the combustion chamber's height is changed at each engine load simulation in order to accomplish around the same maximum in-cylinder temperature and pressure. Therefore in the perspective of fuel consumption investigation, both the engine,s part load losses, thermal efficiencies, air mass flow rate, fuel mass flow rate and subsequently fuel consumptions are worked out for the FUDS and US06 driving cycles.

The procedure adapted for evaluation of fuel economy is elaborated comprehensively for FUDS driving cycle. Generally, vehicle speed and the gear selection define the driving cycle. Federal urban driving schedule (FUDS) is chosen to compute the required engine speed at each instant. The parameters used to calculate the engine speed demand during driving simulations corresponding to an advanced car specifications having five speed gear ratios are as described in Table 5.1. The study of fuel consumption at engine level requires engine speed in accordance with the vehicle speed. At every instant of vehicle driving cycle, the engine speed to be considered as a reference is calculated by using the following relation

$$\omega_e = \frac{v_{cyc}.G.g_k}{r} \tag{5.22}$$

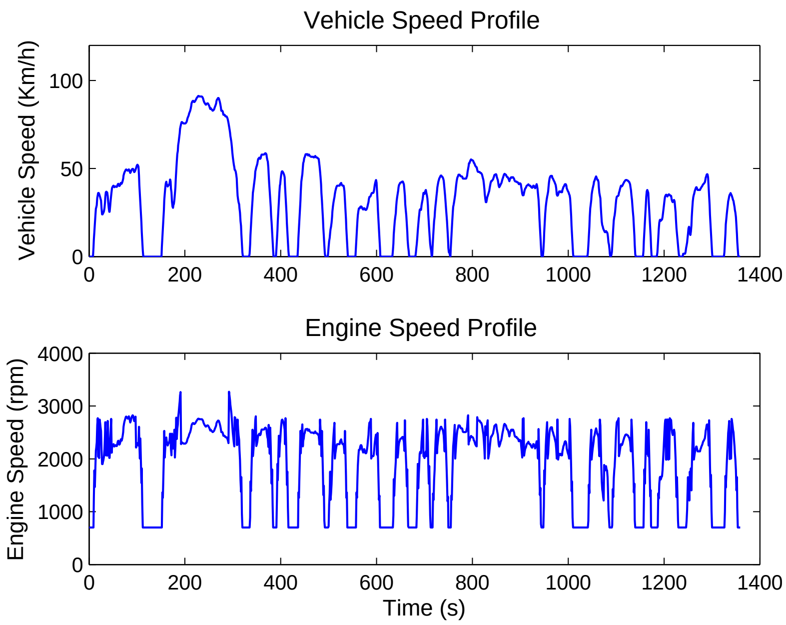

FIGURE 5.8: Vehicle and, accordingly, the engine speed profile for FUDS Driving Cycle

TABLE 5.1: Vehicle & VVT engine characteristics

Symbol	**Description**	**Value/Units**
	No. of cylinders	1
B	Engine Bore	70 mm
S	Engine Stoke	55 mm
V_d	Displaced volume	0.21166 dm^3
P	Maximum power	3.5kW @ 3000rpm
	Maximum torque	18N-m @ 2500rpm
r	Tire Radius	0.3 m
g_k	1^{st} gear ratio	3.827
	2^{nd} gear ratio	2.36
	3^{rd} gear ratio	1.685
	4^{th} gear ratio	1.312
	5^{th} gear ratio	0.90
G	Final drive ratio	3.8
	Idle engine speed	700 rpm
	IVO	20 BTDC $[CA]$
	IVC	32 ABDC $[CA]$
	EVO	40 BBDC $[CA]$
	EVC	12 ATDC $[CA]$

where, g_k represents various gear ratios (i.e. 1^{st} to 5^{th}) depending on the vehicle speed. The vehicle speed and, accordingly, engine speed profiles used as a reference are as shown in Figure 5.8.

The comprehensive simulation results with engine speed profile as a reference of the closed loop the Atkinson cycle VVT engine, performance degrading aspect and various factors having vibrant role in the enhancement of fuel economy at 12Nm load conditions are elaborated here. Finally, compact findings regarding the fuel consumption at different operating conditions are presented. Figure 5.9 reveals speed tracking profile for FUDS using the SOSM controller. The control effort

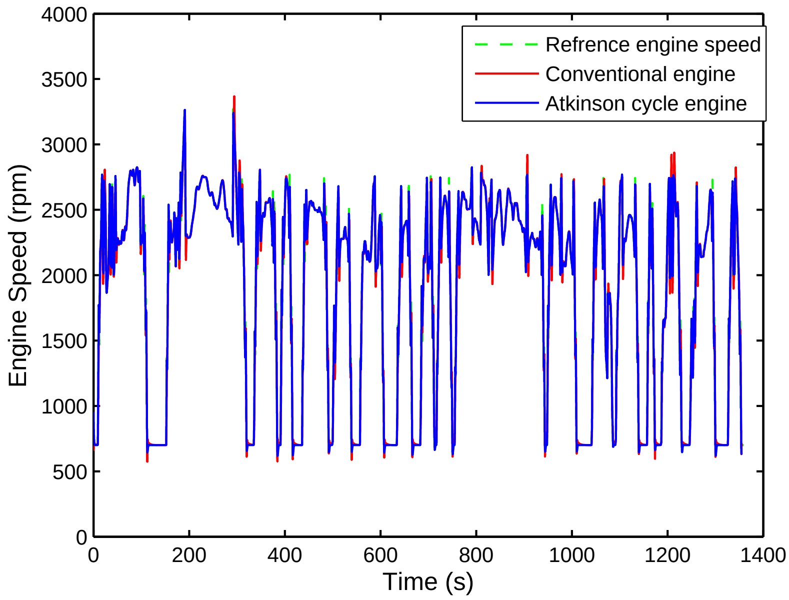

FIGURE 5.9: Speed tracking profile using the super-twisting algorithm at 12Nm load

(IVT parameter λ) and switching surface throughout the driving cycle at 12Nm operating load is as divulge in the Figure 5.10. The intake valve closing timing is primarily VVT engine operating conditions dependent.

A comparative view of the Atkinson cycle engine's pumping losses with those of the conventional Otto cycle during the complete FUDS driving cycle at 12Nm load, is illustrated in the Figure 5.11. It can be seen that Atkinson cycle engine part load losses are comparable to WOT losses of the conventional engine. The reduction in pumping losses with LIVC strategy is predominantly the result of back-flow notion of the A/F charge into the intake manifold of the engine or it simultaneously turns into the intake of another combustion cylinder. Thus, suction pressure rests considerably above the atmospheric pressure during the first part of compression stroke, which is a key contributor in lessening of pumping losses.

Figure 5.11 also demonstrates the thermal efficiency of the VCR Atkinson cycle engine in association with that of the conventional SI engine throughout FUDS.

A considerable enhancement in the EMVEM model's thermal efficiency can be seen, as engine load decreases further down the full load (WOT). With LIVC load control approach around 3.52% improvement in the thermal efficiency is achieved at 12Nm load conditions within the constraints of IVT parameter. However, it can be augmented about 9.4% if it can be operated for lower loads [2].

The comparison of fuel mass flow rate (instantaneous fuel consumption) of VCR Atkinson cycle engine as well as the conventional Otto cycle engine under similar operating conditions during the FUDS and US06 driving cycles is presented in the Figure 5.12. By integrating the fuel mass flow rate of the complete FUDS drive cycle the total fuel consumed during the driving cycle can be obtained and 6.53% improvement in fuel economy is achieved for this particular case.

The fuel consumption comparison accomplished with VCR Atkinson cycle engine over the conventional Otto cycle engine at various engine operating conditions, for instance at 8Nm, 10Nm, 12Nm and 14Nm during the standard FUDS driving cycle in the Figure 5.13. It can be seen that improvement in the fuel economy of the Atkinson cycle engine goes on increasing towards the part load conditions in contrast with the conventional Otto cycle engine. The enhancement in fuel economy at higher load i.e. 14Nm load is 4.57%, whereas augmentation in fuel economy at part load i.e. 8Nm is around 11.50%. However, on the average around 4.32% enhancement in the thermal efficiency, while about 7.80% improvement in the fuel consumption with VCR Atkinson cycle engine over the conventional Otto cycle engine at medium and higher engine operating conditions during the FUDS driving cycle is attained. Likewise, fuel consumption of the VCR Atkinson cycle engine in comparison with the conventional engine under similar engine operating conditions during the US06 driving cycle is evaluated and revealed as in the Figure 5.14.

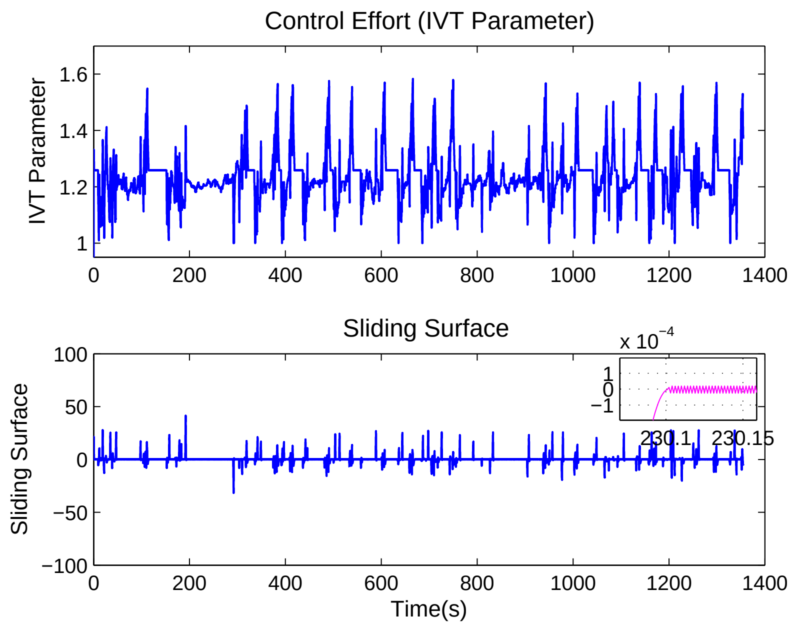

FIGURE 5.10: Control effort (IVT parameter) and switching surface during FUDS at 12Nm load condition

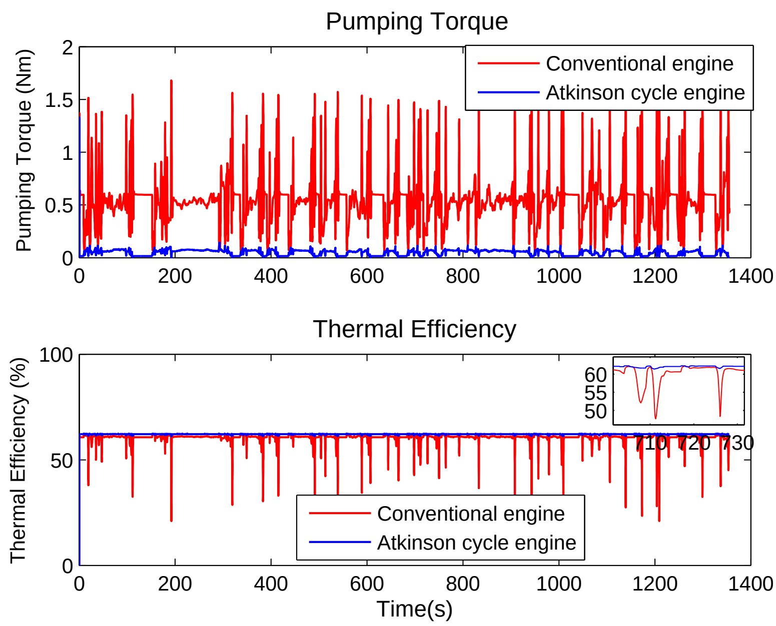

FIGURE 5.11: Pumping losses and thermal efficiency comparison during the FUDS driving cycle at 12Nm load conditions

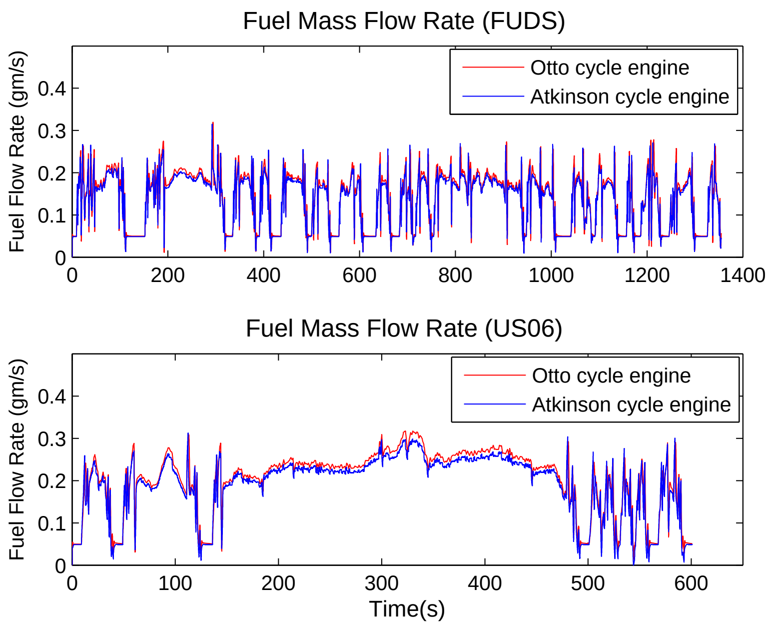

FIGURE 5.12: Fuel mass flow rate comparison during the FUDS and US06 driving cycles at 12Nm load

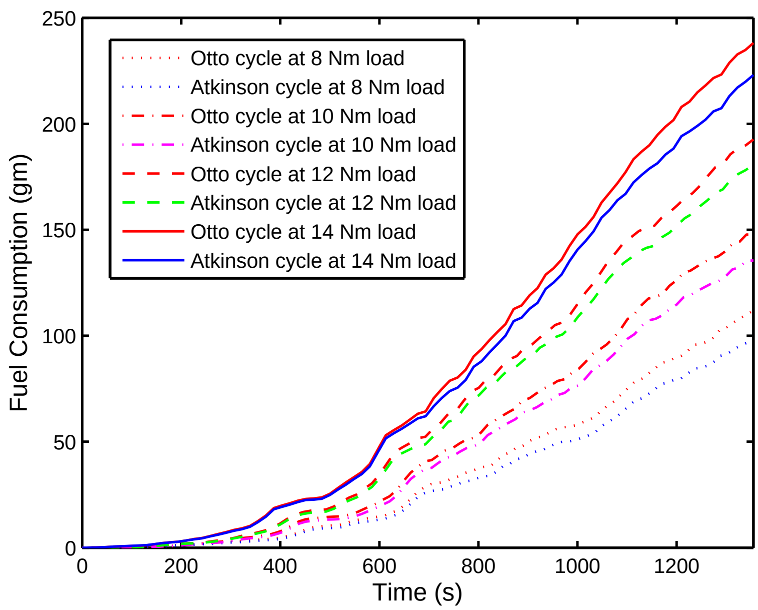

FIGURE 5.13: Fuel consumption for various engine operating condition during the FUDS driving cycle

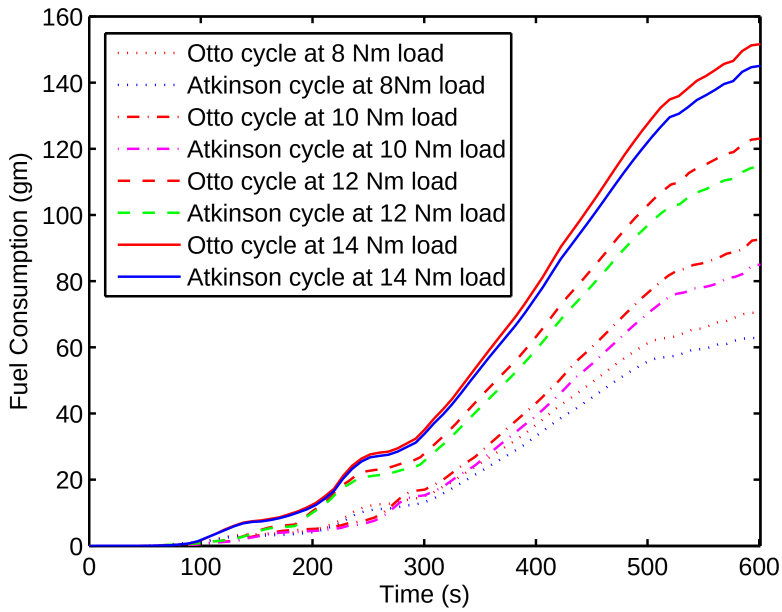

FIGURE 5.14: Fuel Consumption for various engine operating condition during the SU06 driving cycle

TABLE 5.2: Fuel Economy Improvements over Otto cycle engine during the Federal Urban Driving Schedule (FUDS) and US06 Driving Cycles

Engine operating conditions	**Compre-ssion ratio**	**FUDS**		**US06**	
		Thermal efficiency enhanc. (%)	**Fuel economy improv.** (%)	**Thermal efficiency enhanc.** (%)	**Fuel economy improv.** (%)
8Nm	14.65	7.20	11.50	4.59	10.23
10Nm	14.35	4.67	8.62	3.56	8.46
12Nm	13.80	3.52	6.53	2.89	6.19
14Nm	13.15	1.88	4.57	1.66	4.46
Average improvement		**4.32**	**7.80**	**3.18**	**7.33**

The results comprising of engine operating conditions, thermal efficiency and fuel economy improvements of Atkinson cycle VCR engine over the conventional Otto cycle engine during the standard FUDS and SU06 driving cycles, are summarized in Table 5.2. The simulation outcome reveals that on the average about 4.32% enhancement in the thermal efficiency and around 7.80% improvement in the fuel consumption with VCR Atkinson cycle engine over the conventional engine at medium and higher load operating conditions during the FUDS driving cycle are accomplished. The improvement in the thermal efficiency as well as fuel consumption for the FUDS driving cycle is more than that of US06 driving cycle as FUDS driving cycle is inclined more towards the engine part loads as compared to US06 driving cycle. The VCR Atkinson cycle VVT engine is always more efficient in comparison with the conventional Otto cycle engine over the wide operating range. To avoid the effect of LIVC on ignition delay and engine combustion duration, LIVC can be physically realized for medium and higher loads. Below 40% of the full engine load, the use of throttle may be required as a substitute, without losing the potential benefits of the integrated VVT system [53].

5.7 Conclusion

This research outlines the design and development of a sliding mode control framework for the Atkinson cycle engine with flexible intake valve load control strategy instead of the conventional throttle. The control structure based on physically motivated control-oriented EMVEM model of the VCR Atkinson cycle engine is evaluated for the FUDS and US06 driving cycles at medium and higher operating conditions to ensure its better fuel economy in comparison with the conventional SI engine. A SOSM strategy based on super-twisting algorithm is used to solve the VVT engine speed control problem avoiding chattering and simulation results exhibit the suitability of SOSM algorithm to control Atkinson cycle engine with an alternative input channel. The part load losses of the Atkinson cycle engine figured out, are comparable to the WOT losses of the conventional SI engine, whereas a considerable boost in the thermal efficiency is accomplished during the FUDS and US06 driving cycles. Both the aforesaid improvements play vital role in reducing the VVT engine's fuel consumption.

On the average around 7.80% and 7.33% improvement in fuel economy with VCR Atkinson cycle engine over the conventional SI engine during the FUDS and US06 driving cycles, is achieved correspondingly. Besides, robust HOSMC i.e. super twisting control has significantly improved the engine speed tracking, the main shortcoming of PID controller in spite of slight improvement in fuel economy.

Chapter 6

Conclusion and Future Directions

In this thesis, an alternative configuration for use in part load conditions is proposed for a spark ignition engine. It offers the same engine performance as that of the conventional SI Otto cycle engine at full load whereas, better fuel consumption at part load condition. This configuration is based on two principles: 1) Over-expansion: The expansion stroke is longer than the effective compression stroke. The former stroke remains constant during all the working conditions. However, compression stroke depends on the intake valve closing timing. With the use of dissimilar intake valve closing timings the effective air fuel mixture amount trapped in the combustion cylinder is controlled. With flexible late intake valve closing (LIVC) i.e. after BDC, the air fuel mixture back flow phenomenon takes place, a certain part of the inducted air fuel charge flows back into the intake manifold and become intake of the succeeding engine cycle. The higher the difference of IVC, lower will be the amount of air fuel mass trapped in the combustion cylinder, it can be termed as less load applied to the engine.

2) Variable compression ratio: With the use of different IVC timings, the effective compression ratio reduces accordingly depending on the operating conditions. Consequently the in-cylinder pressure and temperature of the combustion chamber during combustion declines, thus decreasing the work per cycle. This inadequacy is dealt with by using the notion of variable compression ratio (VCR) mechanism,

which can adjust compression ratio to pre knock onset conditions and so the efficiency of SI engine can be enhanced significantly at part loads. To model the VCR effect, the combustion chambers height is changed at each engine load simulation in order to accomplish around the same maximum in-cylinder temperature and pressure.

A novel control-oriented extended mean value engine model (EMVEM) of the Atkinson cycle engine was developed, wherein the characteristics such as flexible valve timing, over-expansion and variable compression ratio has been incorporated. The EMVEM model encompasses an unconventional late intake valve closing (LIVC) engine load control strategy instead of the throttle valve. This has a vital role in modeling the inclusive nonlinear dynamics of the SI engine system and to deal with engine performance degrading aspects such as the throttling effect and thermal efficiency etc. The proposed model was validated with the experimental data of a VVT engine, obtained from literature, to ensure that the proposed model has the capability to capture the dynamics of the Atkinson cycle engine, and engine load can be controlled by LIVC approach, instead of the conventional throttle. The potential benefits of late intake valve closing (LIVC) scheme and copious integrated characteristics were also appreciated in the view of fuel economy. Besides, constraints on the LIVC were investigated as well.

The EMVEM model was used to evaluate the part load performance degrading aspects i.e. pumping losses and thermal efficiency of the Atkinson cycle engine through LIVC load handling methodology and compared with the conventional Otto cycle engine. The part load pumping losses figured out that comparable to that at wide open throttle of the conventional engine and computed considerable enhancement in the thermal efficiency as well. In addition, an alternative robust control framework is developed to evaluate the part load losses, thermal efficiency and subsequently the fuel economy of the VCR Atkinson cycle engine for the various standard NEDC, FUDS, FHDS and US06 driving cycles at medium and higher load operating range. Around 7% fuel economy was accomplished.

6.1 Contributions

After brief discussions on the presented research work, the foremost contributions of the thesis can be described as follows:

- Development of control-oriented extended mean value engine model (EMVEM) of the Atkinson cycle engine
- Analysis of the developed engine model in the frame of controllability for the conventional PID controller as well as for the robust nonlinear higher order sliding model control strategy.
- Design, development and evaluation of a robust control framework for the Atkinson cycle engine in fuel economy perspective.
- Introduction of an alternative control scheme resulting in around 6.7% fuel savings.

6.2 Future Directions

The research work carried out so far permitted some conclusions to be reached as illustrated in the preceding section. However, more work need to done to reach on more comprehensive findings to enhance the SI engine performance. The work in this dissertation can be extended in many directions. Few proposals for future directions are as follows:

- A physics based sub model of the volumetric efficiency of the Atkinson cycle VVT engine for dynamic validation of the developed control-oriented EMVEM model can be used.
- A dynamic compression ratio mechanism can be device in order to maintain the compression ratio which has significance to deal with combustion process inadequacies during the entire engine operating conditions.

- The notion of exhaust valve timing to control exhaust gas recirculation (EGR) a corresponding load control approach to intake valve closure , valve overlapping and valve lift can be incorporated in the EMVEM model. Which have an impact on the engine thermal efficiency improvement, emission reduction (reduction of NOx and unburned hydrocarbons), combustion efficiency improvement as well as decline of pumping work resulting in substantial reduction in engine fuel consumption[24].

- Robust control can also be designed to ascertain the air to fuel ratio at stoichiometric ratio which has pivotal role in reduction of engine emissions.

- The Atkinson cycle model EMVEM model with intrinsic LIVC load control strategy required to be validated with dynamic engine data to affirm its transient response as well.

- The developed control-oriented EMVEM model of the Atkinson cycle engine can be investigated/examined from implementation perspective.

- The parameters estimation of the torque friction model employed in this work can be performed as well in accordance with the VVT engine specifications in order to enhance the accuracy of the developed model.

- The developed control framework can be evaluated at vehicle level with the robust controller in presence of an auxiliary loads and an uncertain road loads obtained by standard driving cycles. Moreover, driving cycle used for evaluation of fuel economy as well as emission decline can be improved by incorporating the more realistic road traffic flow, environmental conditions etc which have significant impact on the the engine/vehicle performance.

- Temperature throughout the intake manifold has been assumed constant. The EMVEM model accuracy can be increased if one can include the intake manifold temperature dynamics [**?**].

- With the late intake valve closure (LIVC) load control strategy, SI engine performance can be assessed during the engine entire operating conditions as in this work, it is appraised for engine medium and higher load range.

- The early intake valve closure (EIVC) load control strategy can also be integrated in the conventional Otto cycle engine model to study the engine performance at lower level of load than that examined in this research. A similar comparison study can be carried out.

- Implementation and evaluation of a Atkinson VCR cycle in a two stroke engine in order to pursue its benefits in frame of fuel economy and emission reduction at part load.

APPENDICES

Appendix-A: Derivation of Thermal Efficiency of the Atkinson Cycle Engine:

From an analysis of the ideal Atkinson cycle as shown in Figure 1.2, the effect of over-expansion on the thermal efficiency can be estimated. The specific heat of an ideal gas working fluid has been assumed constant throughout the cycle.

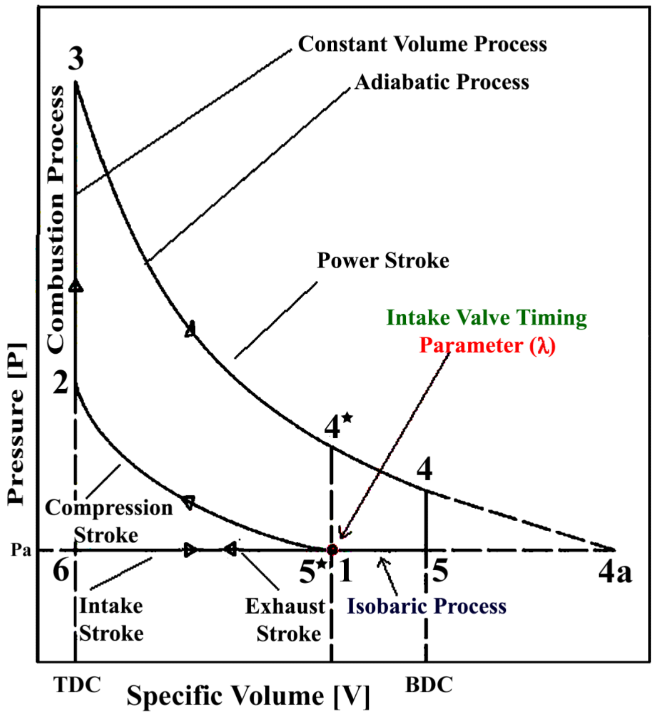

FIGURE 1: Theoretical PV representation of an ideal Atkinson cycle engine [2, 3]

Using the laws of thermodynamics, thermal efficiency of the over-expanded cycle can be described as [3, 42]:

$$\eta_{atk} = 1 - \frac{Q_{4-5} + Q_{5-1}}{Q_{2-3}}$$

where, Q_{2-3} and Q_{4-5} are the heat added and released at constant volume respectively, while Q_{5-1} is the heat released at constant pressure.

Mathematically, these quantities can be expressed in terms of specific heat at constant volume and pressure and corresponding temperature differences as below:

$$\begin{aligned} Q_{2-3} &= C_v(T_3 - T_2) \\ Q_{4-5} &= C_v(T_4 - T_5) \end{aligned}$$

and

$$Q_{5-1} = C_p(T_5 - T_1)$$

where, C_v and C_p are specific heat constants at constant volume and pressure respectively. Thus,

$$\eta_{atk} = 1 - \frac{\left\{\left(\frac{T_4}{T_1} - \frac{T_5}{T_1}\right) + \gamma\left(\frac{T_5}{T_1} - 1\right)\right\}}{\left(\frac{T_3}{T_1} - \frac{T_2}{T_1}\right)}$$

Where $C_p - C_v = \gamma$, furthermore the temperature and pressure values at 1 are considered slandered atmospheric temperature and pressure.
The isentropic relationship for ideal gases for 1-2 and 3-4 are

$$\frac{T_2}{T_1} = r_c^{\gamma-1}$$

and

$$\frac{T_3}{T_4} = r_e^{\gamma-1}$$

Since $\frac{r_e}{r_c} = \lambda$, thus with assumption that the pressure at point 5 remains same as that at point 1 and using perfect gas laws we have

$$\frac{T_5}{T_1} = \lambda$$

The heat added during the isochoric process as a consequence of combustion of air fuel mixture, can be expressed as:

$$Q_{2-3} = m_m \; C_v \; (T_3 - T_2) = m_f \; Q_{LHV} \; \eta_c$$

where, η_c is the combustion efficiency, m_f and m_m are the mass of the fuel and mass of the mixture inside the cylinder respectively, whereas Q_{LHV} is the lower heating value of the fuel.

Eventually,

$$T_3 - T_2 = \frac{Q_{LHV}\eta_c}{(AFR+1)C_v}$$

where AFR is air to fuel ratio.

$$\frac{T_3}{T_2} = 1 + \frac{Q_{LHV}\eta_c}{(AFR+1)C_v T_1 r_c^{\gamma-1}}$$

It can also be expressed as:

$$\frac{T_3}{T_2} = 1 + \frac{\zeta}{r_c^{\gamma-1}}$$

where

$$\zeta = \frac{Q_{LHV}\eta_c}{(AFR+1)C_v T_m}$$

As $T_1 = T_m$, the temperature at point 3 w. r. t the temperature at point 1 can be derived as:

$$\frac{T_3}{T_1} = r_c^{\gamma-1} + \zeta$$

Similarly, the temperature at point 4 can be evaluated w. r. t the temperature at point 1 and has been given as:

$$\frac{T_4}{T_1} = \frac{1}{\lambda^{\gamma-1}} + \frac{\zeta}{r_e^{\gamma-1}}$$

Therefore, by substituting the values, the thermal efficiency of Atkinson cycle VVT engine derived is:

$$\eta_{atk} = 1 - \frac{1}{r_e^{\gamma-1}} - \frac{\{(\gamma-1)\lambda^{\gamma} - \gamma\lambda^{\gamma-1} + 1\}}{\zeta\lambda^{\gamma-1}}$$

This is the mathematical expression for the thermal efficiency of Atkinson cycle VVT engine in comparison with that of the conventional Otto cycle engine.

Appendix-B:

APPENDIX B1: Engine Model State Space Representation

State space equations of the proposed EMVEM Model of Atkinson Cycle Engine and fuel dynamics developed by Hendricks et al. [71] as defined in chapter 5 is given as:

$$\dot{x} = f(x) + g(x, u)$$

where, $x = [Pm, w_e, \dot{m}_{ff}]^T \in \Re^3; u \in \Re; f : \Re^3 \rightarrow \Re^3; g : \Re^3 \rightarrow \Re^3;$ $f(x)$ and $g(x, u)$ being smooth vector fields and are considered as follows:

$$f(x) = \begin{bmatrix} \Psi_1 \chi(p) - 2\Psi_2 P_m \omega_e \alpha(P_m, \omega_e) \\ \\ \frac{V_d}{4\pi J_e} \eta_{atk} \left\{ \frac{r_c}{(\gamma-1)(r_e-1)} \right\} \left\{ \frac{\zeta r_c^{\gamma-1}}{r_e^{\gamma-1}} - (\gamma - 2) \right\} P_m \\ -\frac{1}{J_e} \left(T_{pump} + T_{fric} + T_{load} \right) \\ \\ \frac{1}{\tau_f} \left(-\dot{m}_{ff} - 2X_f \Psi_2 P_m \omega_e \alpha(P_m, \omega_e) \right) \end{bmatrix}$$

and,

$$g(x) = \begin{bmatrix} \Psi_2 P_m \omega_e \alpha(P_m, \omega_e) \\ \\ \frac{V_d}{4\pi J_e} \eta_{atk} \left\{ \frac{r_c}{(\gamma-1)(r_e-1)} \right\} \left\{ \frac{\zeta}{r_e^{\gamma-1}} - (\gamma - 2) \right\} P_m \\ \\ \frac{1}{\tau_f} \left(X_f \Psi_2 P_m \omega_e \alpha(P_m, \omega_e) \right) \end{bmatrix}$$

APPENDIX B2: Controller Bounds Evaluation

Switching Surface and its Derivatives

$$\sigma(t,x) = \omega_{eref} - \omega_e$$

$$\dot{\sigma} = -\dot{\omega}_e$$

$$\dot{\sigma} = -\frac{1}{J_e}\left(T_{ind}(\lambda) - T_{pump} - T_{fric} - T_{load}\right)$$

$$\ddot{\sigma} = \frac{\partial \dot{s}}{\partial P_m}\dot{P}_m + \frac{\partial \dot{s}}{\partial \omega_e}\dot{\omega}_e + \frac{\partial \dot{\sigma}}{\partial \dot{m}_{ff}}\ddot{m}_{ff}$$

where,

$$\frac{\partial \dot{\sigma}}{\partial P_m} = -\frac{1}{J_e}\left\{\frac{V_d \eta_{atk}}{4\pi}\frac{mep}{Pm} + \frac{V_d}{4\pi}\right\}$$

$$\frac{\partial \dot{\sigma}}{\partial \omega_e} = \frac{V_d}{4\pi J_e}\left\{\frac{450}{\pi} + \frac{900}{\pi^2}\omega_e\right\}$$

$$\frac{\partial \dot{\sigma}}{\partial \dot{m}_{ff}} = 0$$

and

$$\varphi(t,x) = \left[\frac{\partial \dot{s}}{\partial P_m}\,\frac{\partial \dot{\sigma}}{\partial \omega_e}\,\frac{\partial \dot{s}}{\partial \dot{m}_{ff}}\right].\left(f(x) + g(x,u)\right)$$

$$\xi(t,x) = \frac{\partial \dot{s}}{\partial u}$$

$$\xi(t,x) = \frac{V_d \eta_{atk}}{4\pi J_e}\left\{\frac{r_c}{(\gamma-1)(r_e-1)}\right\}\left\{\frac{\zeta}{r_e^{\gamma-1}} - (\gamma-2)\right\}P_m$$

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