Modeling, Validation and Analysis of an Advanced Thermal Management System for Conventional Automotive Powertrains

THESIS

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Abstract

Reducing vehicle fuel consumption while maintaining same or better performance characteristics has been one of the main focuses of auto car manufacturers. In this sense, OEMs are introducing thermal management system (TMS) in modern vehicles that help attain rapid fluid warm-up during cold-start conditions. This leads to lower fluid viscosities early on in a drive cycle and hence reduced losses in the engine and powertrain components, resulting in lower fuel consumption. Rapid fluid warm-up also helps improve passenger comfort by providing necessary heating or cooling on demand.

Through this work, a model characterizing the low frequency energy and power transfer in the engine and powertrain components is formulated. An advanced TMS consisting of components for waste heat energy recovery is proposed and its model is formulated. The combined set of these models is called the Vehicle Energy Simulator (VES). The model is thoroughly calibrated and validated using experimental data from steady state and transient testing; results are included in detail.

The validated VES is then used to investigate control strategies for valves that are part of the TMS, used to control fluid flow to the various heat exchangers in order to attain rapid warm-up of coolant, engine oil and transmission fluid. It is seen that, the use of advanced TMS, over a conventional thermal management system, results in 3.4% reduction in fuel consumption. The investigation leads to recommendation of a reasonable first generation
for a genetic algorithm optimization to be used to find the “optimal trajectory” for thermal-system-valve actuation during a drive cycle for reducing fuel consumption.
Dedication

I would like to dedicate this work to my Father, Ravindrakumar Agarwal and my Mother Shashi Agarwal. They have been supportive of me in everything I have done in life; the Masters’ degree was no different. I thank my Father for gifting me with the following thought that has helped me through many challenging times, academically and in other aspects of life: “Set your goals and work towards them as if nothing else matters.”

To both my Sisters, who have been nothing less than mother-figures to me, and to both brother-in-laws, who have been examples to me and have guided me through tough times, I extend heartfelt gratitude.

I would also like to take this opportunity to thank my Uncles who have been there for me and my family in trying times. Last, but by no means the least, I would like to thank my friends who have helped me get through the last six years away from home and I hope that they continue to do so.
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Chapter 1: Introduction

The first internal combustion engines were commercial developed in the late 1800s. These engines generated up to six horsepower and had an efficiency of approximately five percent at best [1.1]. Since then, an enormous amount of progress has been made to increase efficiency of engines, leading to high overall vehicle fuel economies and higher torque and power output capabilities. According to a recent technological review by the U.S. Department of Energy (DoE), the fuel economy of vehicles in the last four decades has increased from an average of 13 MPG to more than 22 MPG, as seen in Figure 1.1. This increase has also been accompanied by higher power output and lower 0-60mph times.

Figure 1.1 Light-Duty Vehicles Fuel Economy Trends [1.2]
New advances in engine research are looking at ways to improve engine operation that go beyond the conventional combustion, fuel injection and air path system improvements. Among others, the subject of managing thermal loads by optimal distribution of available heat is gaining attention as a means of achieving more efficient engine operation.

For the purposes of managing thermal loads, technologies broadly classified as Thermal Management Systems have been implemented to improve engine and powertrain operation. Thermal Management Systems, or simply TMS, “aim at balancing needs of multiple vehicle systems that may require heat for operation, require cooling to reject heat, or require operation within specified temperatures for optimal efficiency” [1.3]. TMS are especially important for improving vehicle operations during cold-start conditions, as vehicle fuel economy may be affected by as much as 10% [1.4] in such conditions.

**Section 1.1. Scope**

In a conventional thermal management system, the typical heat exchangers are the radiator, the engine oil cooler, and the transmission oil cooler. The only flow control device present in a conventional TMS, besides the coolant and oil pumps, is the thermostatic valve, which governs the flow of coolant through the radiator. The components of a TMS are designed to maintain the fluids temperatures within specified conditions, as a trade-off between optimal operations and safety.

This work, as part of a large research project, sponsored by Chrysler LLC and supported by the U.S. DoE, aims at developing an advanced Thermal Management System for controlling the thermal loads in a vehicle for achieving rapid warm-up of the coolant,
transmission fluid and engine oil during cold-start conditions. The proposed architecture of the advanced TMS is shown in Figure 1.1. In this architecture, there are additional heat exchangers and flow control valves in the form of an exhaust to coolant heat exchanger (EGRC), valves that flow of exhaust gases and flow of coolant to the EGRC, a transmission oil heater (TOH), and a three-way valve which controls the flow of coolant to the TOH and the cabin heater.

![Figure 1.2 Current Thermal Management System Architecture](image-url)
Through this work, efforts have been made to model the engine mechanical behavior and thermal characteristics that affect the amount of heat rejected to the coolant via the walls of the combustion chamber. In addition, the TMS described above has been characterized starting from physics-based models of the heat exchangers and other components. The model formulation and calibration phase is followed by rigorous validation of the forward-looking model of the vehicle, of which the aforementioned engine and thermal system models are a part. The forward-looking model is called the Vehicle Energy Simulator (VES).

Using the validated VES, a thorough analysis has been conducted to study scenarios where potential effects of engine oil and transmission oil temperature are seen on amount of extraneous heat generated within the engine and transmission due to friction and other mechanisms leading to inefficiencies. Its effect on vehicle fuel economy has also been quantified. A trade-off study has been conducted to observe the effect of various valve position combinations on the rate of coolant, transmission oil and engine oil warm-up via Pareto analysis of simulation results. Viable strategies for actuating valves that control coolant flow have been proposed. Possible design improvements have been recommended to the current thermal management system.

**Section 1.2. Document Layout**

Following the introduction, this document contains four chapters. A brief description of the contents of these chapters is as follows:

- Chapter 2: Here, a general overview of the recent advancements in engine and powertrains technologies is presented. An emphasis is then placed on recent
development of technologies that aim towards engine and powertrain thermal management. Following this, a brief overview of the techniques used to model thermal systems is presented.

- Chapter 3: The formulation, calibration, and validation of component-level models developed as part of this work are explained in detail. The calibration procedure is explained where applicable and relevant plots and figures are shown to support the model development.

- Chapter 4: The process for systematic validation of the vehicle energy simulator is explained in detail in this chapter. Details regarding the transient vehicle tests performed to generate data for validation are provided along with a description of the test conditions, equipment and hardware with which the tests were performed.

- Chapter 5: This chapter delves into the analysis performed using the validated forward-looking vehicle energy simulator. A design of experiments is used to determine the effect of combination of fixed thermal-system-valve positions on transmission efficiency, extraneous heat generation and vehicle fuel economy; fluid warm-up time is taken as a proxy for satisfying these metrics. Conclusions from this analysis are also stated in this chapter.

- Chapter 6: This chapter provides a summary of the conclusions drawn from the work shown in the earlier chapters. This chapter also highlights tasks that may be taken on in the future, followed by some design recommendations provided for other viable designs of the Thermal Management System using the same devices.
Section 1.3. References


Chapter 2: State of the Art

In Figure 2.1, the observed trend in the annual global energy consumption over the past 20 years as well as projections to 2035 are shown. From the figure, it can be noticed that the annual global energy consumption in 2035 is estimated to rise to 770 quadrillion Btu, twice what it was in 1990. Furthermore, the growth in energy consumption is projected to be concentrated primarily in regions that are not part of the Organization for Economic Co-operation and Development (Non-OECD), such as India and China, which are developing at a very rapid pace [2.1].

![Figure 2.1 World Energy Consumption (in quad. Btu): History and projections [2.1]](image-url)
The classification of the total energy consumption based on the source of energy reveals an interesting picture. In Figure 2.2, it is seen that liquid fuels are responsible for meeting the majority of the energy requirements of the world and is projected to follow the same demand pattern in the next 25 years.

![World energy consumption by fuel, 1990-35 (quadrillion Btu) [2.1]](image)

Further investigation sheds light on the sectors that have historically consumed the highest percentage of liquid fuel. As seen in Figure 2.3, the transportation sector alone accounts for more than fifty percent of the liquid fuels consumed globally; this percentage is projected to increase over the course of the next couple of decades.
Figure 2.3 World liquid fuel consumption by sector, 2008-35 (million barrels/day) [2.1]

Since the transportation sector is the highest consumer of liquid fuels in the world, improvements in the overall efficiency of vehicles, would help reducing their consumption rate. In lieu of the current and projected increase in demand for liquid fuels, a great impetus has been provided by government and the private sector to research focused on investigating ways to reduce energy consumption of vehicles [2.1][2.6][2.8][2.65][2.18]. A broad range of technological advancements and innovations have been made in the field vehicular technology and are currently being implemented on modern vehicles. These technologies probe opportunities of reducing fuel consumption. Several more technologies, showing potentials of reducing vehicle fuel consumption are presently being evaluated. The following section gives an overview of some of the research conducted in past decades and topics being pursued at present in the realm of advanced vehicular technologies.
Section 2.1. Overview of Advancements in Engine and Powertrain Technology

Advancements in propulsion technology, those implemented as well as those under development, may be classified into a two basic categories: 1) those that entail modifications of existing hardware leading to improvement via design optimization, and 2) those that entail addition of newly developed hardware (mechanical as well as electrical components). Also, another significant advancement in modern automobiles, in the form of a non-propulsion technology, is the capability to formulate more sophisticated control architectures for controlling complex systems that have a high-level of interdependency. Typically, a combination of all these three approaches is today applied in the design of vehicles.

Modifications to existing engine and powertrain related technological practices implies replacing hardware with more robust and capable components, implementation of more involved control strategies as well as a combination of the two. Modifications made to powertrains at present include improvements to combustion processes [2.36][2.38][2.39][2.43][2.45], start-stop technology [2.4][2.87], reduction of weight using lighter materials[2.59][2.60][2.61], engineering more efficient auxiliary components such as coolant pumps, air-conditioning systems, cooling fans etc., that are more efficient[2.65], and so on.

Some of the advanced engine and powertrain related technologies that have recently been are concepts such as engine downsizing and turbocharging with increased compression ratios [2.67][2.67][2.69], Gasoline Direction-injection (GDI) [2.43][2.44], fully Variable
Valve Actuation Systems [2.70][2.71][2.73], precision cooling and thermal management systems [2.15][2.21][2.19], transmissions with increased number of gears for more drive flexibility [2.78][2.79][2.80], waste heat recovery systems, electrification of auxiliary loads, and various forms of powertrain hybridizations (through mechanical or electrical secondary energy storage systems) [2.83][2.84].

The advancements mentioned above may also be classified based on system-level improvements, via modification or new development, made to one of the following vehicular systems:

- Improvement to engine combustion and air-path systems;
- Improved transmission and driveline technologies;
- Vehicle system optimization (reduction of vehicle mass, use of lighter and stronger materials, engine thermal management, ancillary load reduction);

Each of these categories of improvements are considered in detailed in the following subsections.

2.1.1. Improvement to engine combustion and air-path systems

Improvements to combustion events are generally oriented towards increasing the combustion efficiency and reduce its duration. Fast-burn combustion systems are used to increase the efficiency of SI engines by increasing the fuel burn-rate to attain higher peak pressures and temperatures [2.8]. This is possible by developing more turbulence in the air-fuel mixing process and obtaining a more uniform charge formation. This is the concept, for example, behind Homogenous-Charge Compression Ignition (HCCI)
engines, which has received much attention recently. HCCI combustion involves pre-mixed, homogeneous charge of air, fuel and residual gases that is combusted by auto-ignition [2.110]. The main feature of HCCI combustion is the self-ignition process that results from tight control of the pressure and temperature conditions within the cylinder. HCCI combustion has the potential of lower emissions than in traditional SI and CI engines. Fuel economy benefits of over 16% over a EPA City Drive Cycle with HCCI implementation on a baseline 2.3L direct-injected engine with twin-independent VVT and EGR have been demonstrated [2.13]. Several other studies provide insight on the fuel economy benefits of implementing HCCI technology [2.8][2.35][2.36][2.37][2.38]. As HCCI combustion is based on self-ignition, it leads to problems with engine knocking and durability of the engine. With development of better techniques of Controlled Auto-Ignition (CAI) [2.39][2.40][2.41][2.42], to harness the full potential of HCCI technology by avoiding problems related to pre-ignition and engine knock, HCCI combustion promises to be an integral part of future engines.

A new innovation in engine combustion systems is the introduction of Direct Injection (DI) systems. The recent availability of flexible and reliable common-rail fuel injection systems is giving a push to the development of a new generation of Gasoline Direct Injection (GDI) engines. In [2.43], a GDI system implemented on an engine has been demonstrated to achieve fuel economy benefits up to 10% compared to baseline results from the same engine but without GDI. Other works [2.44][2.45] also provide simulation results as well as results from on-vehicle testing with GDI implemented showing reduced fuel consumption, high torque and better power characteristics. Ando et. al. [2.45]
illustrates the development of two engines: a 1.8L four-cylinder GDI engine and a 3.5L V6 GDI engine with compression ratios of 12.5:1. It is shown here that the introduction of GDI technology in an engine with higher compression ratios and two-stage mixing processes led to 10% higher torque and 20% better fuel economy than the conventional port fuel injected (PFI) engines, while maintaining clean exhaust gas that can meet European step 2 and German D3 emission regulations. The authors of [2.6] provide details about wall-guided and spray-guided gasoline direct injection systems. Such systems help internal mixture formation within the combustion chamber, allowing a remarkable decrease of fuel consumption and pollutant emission reduction along with improvement in performance.

Other combustion-related technologies leading to better engine performance and lower fuel consumption are worth mentioning due to their considerable benefits, in particular, Deceleration Fuel Shut-Off (DFSO) [2.48][2.47], cylinder deactivation [2.46][2.49][2.50][2.51][2.52], Low Temperature Combustion (LTC) [2.53][2.56], swirl control methods [2.54][2.55], and Pre-mixed Charge Compression Ignition [2.57][2.58].
Improved combustion systems typically entail employing modified engine air-path system designs, as combustion systems and engine air-path systems are highly interdependent, as seen in Figure 2.4. For instance, Stan [2.6] provides a broad description of engines that employ sophisticated charge mixture control, adaptation techniques during transient engine operations and increased compression ratio together with tuned intake and exhaust systems and re-circulated exhaust gases (EGR). Such engines are in existence today and offer a 15% reduction in brake specific fuel consumption, with either better or same performance characteristics as compared to the baseline.

Downsized engines are being increasingly deployed in production vehicles. Engine downsizing by 30% is shown to improve fuel economy by 8-10% while improving torque and acceleration performance [2.66], while downsizing by 40% is estimated to improve
fuel economy by more than 20% [2.67]. In [2.67], authors show potential for better fuel-economy in SI-engines through downsizing and super-charging (DSC), especially at part-load conditions. The authors exhibit a fuel economy of 67 mpg (3.5 liters per 100 kilometers) in the European drive cycle MVEG-95 for a light-weight car, with the potential to satisfy ULEV or Euro IV emission limits. As seen in the reports cited above, downsizing is often accompanied by turbocharging, and together they help reduce engine mass and pumping losses while matching or improving the performance characteristics of the baseline engine.

In [2.8], a brief description of the obstacles to downsizing and turbocharging is available with an overview of what is being currently done to overcome these obstacles. This report highlights that downsized and turbocharged engines often require additional equipment to lower charge air temperatures and to dilute further the mixture entering the combustion chamber, thereby allowing leaner engine operation. To this end, charge air coolers are used. Sometimes, reduction of compression ratio is also necessary for port fuel injected engines, which ultimately penalizes fuel economy. To avoid penalizing fuel economy, downsizing and turbocharging is accompanied with a combination of technologies such as dual-cam phasing, and direct injection; these allow utilization of the complete potential of boosted and downsized engine for improving fuel economy of a powertrain by operating in the most efficient regions of the engine map. These technologies work together in synergy to avoid conditions of knock and pre-ignition as well as to maintain performance characteristics of engines, while reducing the amount of fuel consumed. The authors of [2.69] highlight a system level approach followed in the
realm of downsizing and turbocharging to attain high low end torque and specific power capabilities, and low specific fuel consumption over the entire engine map. In the same report, better injection techniques and valve actuation systems have been demonstrated to compound the benefits of downsized and turbocharged engines, providing 15% higher fuel economy by enabling engine operation over the entire engine map at part load and full load conditions.

Honda was the first car manufacturer in the world to implement variable valve actuation system on a vehicle, which can simultaneously switch the timing and lift of the intake and exhaust valves [2.70]. The system, still in production today, is known as Variable Valve Timing and Lift Electronic Control (VTEC). The timings for Intake Valve Closing - IVC (just before the advent of the compression stroke) and Exhaust Valve Opening - EVO (just after the end of the power stroke) greatly affect the torque and power characteristics of an engine [2.2]. Using variable valve actuation systems, the IVC and EVO can be optimized. This is because, depending on when the intake valve closes, the volumetric efficiency and therefore the amount of fresh charge entering the engine changes. Therefore, implementation of variable valve actuation strategies provides more control over the amount of charge entering the combustion chamber, ultimately aiding in reducing the fuel consumption of the engine since the amount of charge that enters the engine is commensurate to the amount of fresh charge that is actually required at that engine operating condition. In addition, by appropriately modulating the times at which the intake and exhaust valves close and open, pumping losses of the engine can be reduced [2.2], leading to the engine operating at higher efficiency conditions. Various
automotive companies have introduced variable valve actuation systems with newer additions to their fleet as successful implementation of VVT/VVA technologies promises an improvement in fuel consumption in the order of 8-9% [2.8]. Variable valve actuation, as mentioned earlier, facilitates engine downsizing, which has a compounding effect on fuel consumption reduction. Several other reports provide details on the benefits of variable valve actuation technology [2.71][2.72][2.73][2.74], and others outlining on the obstacles to global implementation of this technology [2.75][2.76].

Recent technological advancements not specifically related to the engine air-path or engine combustion, are discussed in the following subsection.

2.1.2. Improved transmission and drivetrain technologies

According to [2.8], better and more efficient transmission and drivetrain technologies can help reduce fuel consumption of vehicles in two ways. The first method is to add more gears, so that the engine can be operated at more efficient operating conditions. The second is by lowering the final drive ratio. Out of these two methods, adding gears to the transmission has been shown to have more to offer in terms of fuel economy benefits. This is because, with increased number of gears, the engine can be operated more often at speeds where engine efficiency is higher.. Several models of modern, high-end passenger cars feature 6, 7, and sometimes 8-speed transmissions. Such transmissions are slowly making their way into other segments of the passenger car market. Along with increased number of gears, the efficiency of the transmission itself is increasing, due to improvements in the design of transmission components, optimal sizing techniques, and advanced lightweight materials.
Dual-Clutch Transmissions (DCTs) as well as automatic transmission with higher number of gears can reduce the ratio between engine speed and vehicle speed. Highest flexibility in gear ratios is observed in Continuously Variable Transmissions (CVTs). Theoretically, CVT can achieve infinite number of gear ratios but are limited by the accompanying control system, which selects the operating gear ratio. Studies show that continuous improvement in transmission technologies could potentially reduce overall vehicle fuel consumption by as much as 8% [2.8][2.10][2.11][2.12]. Additional research and trends in new transmission technology are highlighted in [2.77][2.78][2.79] and [2.80].

Vehicles employing hybrid propulsion technologies are gaining impetus from policy makers worldwide. For example, the Partnership for a New Generation of Vehicles
(PNGV) program, between government and the auto industry, aims to double the fuel economy of compact and midsized automobiles through combination of gasoline engines and electric motors in hybrid vehicles and implement regenerative braking systems, among other technologies [2.81]. The benefits of hybridization, depending on the strain of hybrid technology implemented, are, possibility of vehicle operation at optimum efficiency conditions due to an amalgamation of conventional IC engines with hybrid components, and the opportunity to downsize IC engines, resulting in overall higher fuel economy. Hybrid technologies include Plug-in Hybrid Electric Vehicles (PHEVs), Hybrid Electric Vehicles (HEVs), Mechanical Hybrids, and Fuel-cell hybrids. Out of these hybrid technologies, HEVs are most commonplace but mechanical hybrids are also developing as a promising technology. Several research articles and reports from universities and auto manufacturers point towards drivetrain hybridization due to the potentially large fuel economy benefits and reduced emissions [2.82][2.83][2.84][2.85][2.86]. In a study conducted by the National Renewable Energy Laboratory (NREL) [2.14], a parallel hybrid-electric vehicle could lead to 24% better fuel economy than a conventional, engine-powered vehicle. A series hybrid vehicle, could provide 18% better fuel economy than a conventional ICE-based vehicle for the same type of tests. From studies conducted in [2.28], mechanical hybrid technology, flywheel-based systems, as well as hydraulics based systems, have been estimated to improve vehicle fuel economy by 11.5-13% based on tests and simulations conducted using standard drive cycles.
Another important drivetrain related advancement, namely stop-start technology, are receiving tremendous interest from automotive OEMs as a way of reducing fuel consumption by turning off the engine in heavy traffic urban driving conditions. Facts and numbers seen in [2.3][2.4][2.34][2.87] and several other extensive studies and real-world observations are promising and hence, start-stop technology are being implemented increasingly more on production vehicles. The benefits of stop-start systems are that they are can be integrated relatively easily, into production vehicles and provide significant fuel economy improvements with minimal cost without affecting vehicle performance. As part of the work conducted in [2.4], systems have been developed that replace current OEM engine alternators with a starter/alternator driven by a multi-ribbed V belt, which, due to its robustness, enables engine stop-start in stop-and-go traffic. Enhanced power supplies are used in conjunction with this system, based on a standard 12V electrical system, to avoid installation of additional power-electronics hardware like breaks and dual voltage networks. Standard drive cycle tests reveal that using stop-start systems to switch the engine off, when not necessary, could save up to 5.3% fuel in city driving and 4% in highway driving, a combined FE improvement of about 4.8%. Simulations predict up to 10% FE benefits arising from mild-hybridization and introduction of stop-start technology in city driving cycles. On the other hand, stop-start technology may offer only about 5% during highway driving, due to the fewer opportunities for engine stop.
2.1.3. Vehicle system optimization

Optimization of system-level attributes of a vehicle can lead to significant reduction in fuel consumption. Improvements in fuel economy have been observed through vehicle mass reduction [2.59][2.60][2.61][2.62], reduction of ancillary loads through implementation of technologies like solar-reflective glazing, parked car ventilation and solar-reflective opaque surface coatings [2.63], advanced vehicular climate control [2.65], “cool” exterior car colors to reduce air-conditioning energy use [2.64], etc., all due to reduction in parasitic operational losses.

Reduction of vehicle mass by using lightweight frames, polymers, composites, and lightweight alloys is another direction being explored by automotive OEMs. In [2.5], authors mention that the U.S. Department of Energy partnered with automakers to advance research related to reduction of gross vehicle mass, as a part of efforts to achieve higher fuel economy of vehicles in the passenger car segment. Table 2.1 shows the effect of vehicle mass reduction on fuel economy, for this segment. A 10% reduction in vehicle mass, from the reference case, is likely to improve fuel economy by 4.7%. Optimistic considerations involving vehicle mass reductions of 20-30%, result in improved fuel economy predictions in the range of 9-13.3%. These fuel consumption reduction percentages are assuming everything else in the vehicle remains the same as baseline and only mass decreases. In the future, employing mass reduction techniques would become mandatory so that even after the addition of hardware for electric and/or mechanical hybridization, the potential fuel economy benefits of hybridization are not nullified or reduced by the added weight.
According to the National Renewable Energy Laboratory, “while most emphasis is on efficient equipment because suppliers manufacture hardware, the emphasis should begin with reducing the thermal load, such that equipment size is as small as possible” [2.63]. In this sense, using alternatives such as increased vehicle body insulation, advanced glazing with low-emissivity coatings or multiple layers, or parked car ventilation, are suggested approaches to reduce the cooling thermal load on the vehicle cooling system. The same report also shows simulated estimates of increase in fuel consumption for vehicle with four different propulsion technologies when using the air-conditioning system, as seen in Figure 2.6. It is seen that the fuel consumption rises by approximately 2 liters per 100 kilometers, for a medium-sized sedan, over the FTP drive cycle. As a comparison, the baseline fuel consumption for the gasoline and diesel vehicles is 10.2 and 8.6 liters per 100 kilometers, respectively. This means that the A/C system operation resulted in a 20-25% increase in fuel used over the FTP drive cycle during this simulation. These numbers strongly signal towards the importance of reducing thermal loads on the vehicle.
Other potentials of ancillary load reduction include, but are not limited to, optimizations to alternator and weight reduction of front end accessory drive systems [2.88][2.89][2.90], reduction of tire loads through better design [2.91][2.92][2.94], maintaining optimum tire-pressure to reduce rolling resistance [2.93].

With the advent of many engine and powertrain related technologies, it is important to formalize a technique which enables engineers and researchers to blend the best characteristics of these technologies. This will help deliver an end product to the consumer that provides the best in terms of fuel economy without compromising on performance characteristics of an automobile. With advancements such as downsizing, turbo-charging, exhaust gas recirculation, HCCI, hybridization, transmission with increasing number of gears, electrification of components, waste heat recovery etc., it is
challenging, but essential, to fuse these technologies to make them work together seamlessly. Turbocharging by itself can be very complicated to control with the introduction of variable geometry turbines, variable geometry compressors (VGCs), twin-stages and so on. This can be achieved only with the development of robust and sensible control schemes that trigger various actuators in the system for optimum overall fuel economy while still satiating the performance requirements of the user.

A method which has received much attention for improving fuel economy and increasing passenger comfort, engine thermal management and waste heat recovery, will be the focus of the next section. This section describes the methods currently employed and those that are being developed to recuperate some of the exhaust heat energy.
Section 2.2. Thermal Management Systems (TMS)

A large portion of the energy, generated from the engine combustion process, is dissipated through the exhaust gases, as shown in Figure 2.7. Also, approximately another one-third of the fuel energy is removed from the combustion chamber by the coolant, and ultimately dissipated to the environment. The numbers shown in Figure 2.7 are for a mid-size sedan equipped with a 2.5L SI engine.

![Energy Audit for an EPA city cycle](image)

With approximately 65% of the energy produced in the engine being lost via the engine cooling and exhaust circuits, it is important to focus attention on recuperating some of
this waste heat energy. In this sense, powertrain thermal management is an important concept that is gaining increasing support from researchers, auto manufacturers and governmental agencies [2.22][2.25][2.61][2.65]. Thermal Management Systems, or TMS, are systems that focus on “balancing needs of multiple vehicle systems that may require heat for operation, require cooling to reject heat, or require operation within specified temperatures” [2.106]. Some of the strategies that are being implemented in modern vehicles as part of advanced TMS include on-demand cooling via use of electric coolant pumps [2.20][2.21][2.22][2.95], “smart” electronic thermostats [2.15][2.19], compact coolant to oil heat exchangers [2.105][2.108], variable speed radiator fans [2.96][2.97], grille shutters [2.17][2.98], Waste Heat Recovery, for instance through the use of Organic Rankine Cycles (ORCs) [2.99][2.100][2.101], heat storage tanks [2.25]. Through these technologies, the operating temperature of engine and powertrain components is maintained to be higher than in conventional thermal management architectures. As a result, the engine operates at higher thermal efficiency levels [2.109].

Figure 2.8: Viscosity versus temperature (Engine Oil is SAE5W30 and Transmission Fluid is ATF +4)
As can be seen in Figure 2.8, at higher temperatures the viscosity of powertrain fluids such as engine oil and transmission fluid is lower. Therefore, operating at higher temperatures results in an overall reduction of frictional losses in the engine and transmission, with ultimate improvement of vehicle fuel economy [2.104][2.107].

Apart from the fuel economy benefits of Thermal Management Systems (TMS), they provide the ability to manage the temperature of the engine and other powertrain components, by enabling on-demand operations. On-demand coolant flow enabled by hybrid or electric coolant pumps, varying cooling capacities offered by variable speed radiator fans, smart thermostatic valves, split-cooling, and so on, become extremely important when trying to avoid spark-knock issues. Spark knock is a condition that arises at higher operating temperatures, where the probability of pre-ignition of the charge in the combustion chamber increases considerably, leading to knocking and increased fuel consumption [2.102][2.103]. Engine knock results in the ECU retarding spark timing from MBT conditions, leading to a decrease of the indicated efficiency [2.2]. Engine knocking also causes the oil film on the cylinder walls to be penetrated by the combustion gases, leading to increased heat transfer to the cylinder walls and the formation of hot spots. As more heat is rejected to the coolant, the coolant temperature increases and, due to the thermal lag in the conventional wax thermostat, the thermostatic valve opens to larger values. Eventually, this phenomenon leads to the coolant being brought to lower temperatures than the operating temperature at which the engine is most thermally efficient. The radiator fan operation, when the thermostat is open, causes a substantial drop in coolant temperature. This behavior of the engine is represented as can
be seen in Figure 2.9. On-demand cooling allows for a closer control of the thermostat (requiring an electronically operated valve) and of the radiator fan so that coolant temperature is managed more closely to its optimal set point.

![Figure 2.9 Effects of fluctuation in coolant temperature [2.105]](image)

Some of the strategies used in modern vehicles or currently under development are described below.

**Electronic thermostat**: In conventional engine configurations, a wax thermostat, which is a passive flow control device, starts routing coolant through the radiator when the temperature of the coolant approaches the melting temperature of the wax. The conventional wax thermostat offers reliable operation but starts routing the flow of coolant through the radiator at coolant temperatures as low as 80-85°C [2.19]. This results in the engine operating at temperatures that are only slightly higher than this range. This range of temperatures is not ideal for operation of an ICE as the thermal
efficiency of the engine in this temperature range is not optimum. In [2.20], it is recommended to maintain cylinder bore temperatures at 195°C and steady state engine oil and coolant temperatures at 140°C and 115°C, respectively, for optimum engine operation. To avoid routing the coolant through the radiator at low temperatures and also to have a command over how much coolant flow should be routed through the radiator to maintain optimum engine operating temperatures, smart thermostatic valve are now being used. In [2.19], the authors discuss results of inserting a smart thermostat valve in place of a conventional wax thermostat and compare the coolant warm-up times, from 20 to 87°C, and the deviation from the temperature set-point of 87°C. With the smart thermostat valve, the warm-up time of the coolant for a set engine torque and engine speed profile was 240 seconds, 20 seconds less than the baseline case with a conventional wax thermostat. Also, the deviation in temperature around the desired temperature set-point using the smart thermostat valve was only 0.25°C while it was ±2°C when using the conventional wax thermostat (results from simulation).

After studying an engine cooling strategy that uses only the smart thermostat valve, [2.19] shows tests of an engine cooling strategy that makes use of an electric variable flow pump along with a variable speed radiator cooling fan, and the smart thermostat valve. Through this study, it was seen that the variable flow pump and the variable speed cooling fan enable reduction of the variation in engine temperature, higher overall operating temperatures, as well as reduced power consumption. The power consumed to operate the smart thermostat valve with a mechanical water pump was seen to be much higher than the power required for operating the combination of smart thermostat valve,
the variable flow pump and the variable speed fan. This is because the variable flow pump is shut-off when not required. Also, the variable speed fan is operated to maintain engine coolant operating temperatures only when the coolant flows through the radiator i.e. when the electronic thermostat valve is opened electronically.

**On demand cooling/SMART COOLING:** The use of an electric coolant pump has also been proposed in [2.21] to reduce the parasitic losses off the engine crankshaft, reduce the duration of cold-start phases and to operate engines under optimal thermal conditions. The power consumption of a mechanical pump is constant as well as much higher than that of an electric cooling pump. As seen in Figure 2.10, the power consumption of the electric cooling pump was approximately 15W for 1200 seconds of engine testing, whereas the mechanical cooling pump consumed 75W for the same amount of time.

![Figure 2.10 Power consumption of cooling pump: Electric vs. mechanical [2.21]](image)
The use of electric coolant pumps is a very practical approach that is being largely applied today by manufacturers. For instance, BMW launched the 3-series in 2006, housing the electric cooling pump as a replacement for the conventional mechanical cooling pump, and is now introducing this hardware in other models. The electric cooling pump stops routing coolant around the engine block during warm-up and reduces coolant flow during part load conditions when minimal engine cooling is required. The fuel economy benefits are stated to be in the range of 1.5-2% as a result of reduction of parasitic losses at the crankshaft due to electrification of cooling pump and due to better warm-up characteristics [2.22].

![Electric water (coolant) pump - BMW 3-series](image)

Figure 2.11 Electric water (coolant) pump - BMW 3-series
Active Grille Shutters: A comprehensive study of the effects of engine compartment encapsulation and grille shutter control on engine warm up times and fuel consumption has been conducted in [2.17]. According to the authors, having the grill shutters closed during cold-start conditions and encapsulating the engine results in reduced fuel consumption. The reduction in fuel consumption is a byproduct of higher engine operating temperature due to an encapsulated engine. Also, operating grill shutters appropriately helps in avoiding unnecessary cooling of the engine block following a cold-start at sub-zero or very low temperature ambient conditions. Figure 2.12 shows the fuel consumption reduction estimates from a forward-looking model of the vehicle that was thoroughly validated using data from wind tunnel test conducted in controlled conditions. For high-load driving cycles, such as the Federal Test Procedure Driving Cycle (FTP 75) and Braunschweig (BS) Drive Cycle, the simulations show a reduction in fuel consumption of 1.13% and 0.55% respectively, while during the milder New European Drive Cycle (NEDC) the fuel consumption reduction estimate is 1.92% at an ambient temperature of 20°C. For the case when grille shutter control strategies (as shown in Figure 2.12) are implemented at an ambient temperature of 7°C, the fuel consumption reduction estimates are higher; 2.52%, 1.58% and 2.4% for the FTP, BS and NEDC drive cycles, respectively. The reduction in fuel consumption are higher for lower ambient temperatures is principally due to three factors; i) friction inside the moving mechanical components due to oil viscosity and coolant temperature are reduced by a higher percentage in the case of tests starting from lower temperature and, ii) air density at lower temperatures is higher but when grille shutters are used to stop air from flowing into the
underhood compartment, the air resistance is reduce by a larger percentage, iii) grille shutter closing times. The simulations took into account the contributions of air resistance and reduced heat losses separately. As shown in Figure 2.12, most of the estimates for reduction in fuel consumption are due to reduced air resistance because of closed grill shutters, nevertheless there are also benefits derived from reducing engine heat loss [2.98].

![Figure 2.12 Fuel consumption reduction at two grill shutter settings and two ambient temperatures [2.17]](image)

Use of thermoelectric devices: Thermoelectric devices are devices that produce electricity due to a temperature gradient across the thermoelectric module owing to a phenomenon called the Peltier effect. The Peltier effect cause heating or cooling junctions of two different semiconductors by virtue of flow of electric current across the
boundary of the semiconductors. Conversely, if a temperature gradient is induced across the junctions of two different semiconductors, flow of current will be induced across the boundary layer as a result of this temperature gradient. This is the principle used in thermoelectric generators proposed for use on-board modern vehicles.

![Principle of operation of a thermoelectric device](image)

**Figure 2.13 Principle of operation of a thermoelectric device [2.24]**

In [2.24], the authors propose thermoelectric devices be placed in the exhaust stream of the conventional automobile to recover some of the exhaust heat energy. Tests were conducted for various drive cycles with a thermoelectric pack containing ~800 thermoelectric elements placed appropriately in indirect contact of the exhaust gas stream downstream of the catalytic converter. The authors provide results of how effective these thermoelectric generators were on various drive cycle tests. As seen in Table 2.2, the fuel economy benefits of using a thermoelectric generator are in the order of less than 0.5%,
when using 800 pieces of semiconductor material. Currently, the power density, weight, and cost of such thermoelectric modules are prohibitive factors in large scale use of thermoelectric on modern vehicles. In the future, with the advancement of thermionic technology, using thermoelectric devices could be a potentially beneficial method to power auxiliaries in the vehicle or to provide power for passenger comfort management.

<table>
<thead>
<tr>
<th>Improvement in fuel economy</th>
<th>Reduction of CO2 per km</th>
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<td>LA4</td>
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<tr>
<td>HWY</td>
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<td>SC03</td>
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<tr>
<td>US06</td>
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**Heat storage tanks:** Another example of car manufacturers recognizing the need of thermal management systems was demonstrated by Toyota Motor Corporation with the introduction of the Toyota Coolant Heat Storage System (CHSS) in the Toyota Hybrid System II (THS II), launched in year 2003. This system was developed with a view to facilitate rapid warm-up of the intake ports, cylinder head by flowing hot/warm coolant stored in an insulated tank, over the engine block following a cold-start. This insulated tank is part of the CHSS and enables reduction of engine-out HC emissions during cold-start and warm-up phases. A schematic of this system is shown in Figure 2.14.
During a cold-start, hot/warm coolant stored in this reservoir is routed directly to the cylinder head as shown in Figure 2.15-A. For passenger comfort during a cold-start, hot coolant directly from the cylinder head is routed to the cabin heater (Figure 2.15-B). As the temperature of the coolant reaches steady state operating temperatures after continued engine operation, part of the coolant is routed back to the insulated coolant reservoir where it is stored for use in the next phase of cold condition driving, if need be (Figure 2.15-C). If the engine starts to operate at temperatures that lead to sub-par engine operation, the hot coolant stored in the reservoir is routed back to the cylinder head (Figure 2.15-D). This process is continued in a cycle during the operation of a vehicle. It should be stated, that the operation of the CHSS described here is simplified, and that a
lot of other constraints and operating parameters for CHSS use exist in the CHSS control structure, as are described in [2.25].

The percentage of fuel economy improvement decreases as the time following a cold-start increases, following the law of diminishing returns, as seen in Figure 2.16 [2.25]. But the cumulative reduction in fuel consumption leads to better numbers than the figure

Figure 2.15 Coolant flow configurations in the Toyota CHSS - Adapted from [2.25]
below expresses. Toyota’s CHSS system is estimated to provide fuel economy improvement of about 2-3 % at steady state conditions, as seen below. Considering the fact that these savings are higher initially, the weighted average would result in reduction of fuel consumption of around 4-5% (depending on the duration of the trip).

![Graph showing fuel economy improvement over time](image)

**Figure 2.16 CHSS fuel economy improvement %**

**Transmission oil heaters:** A thermal management system consisting of a transmission oil heater and a cabin heater core, coolant flows to which are controlled using a three way valve is described in [2.15]. The system consists of an ATF heater (transmission oil heater) which is a compact liquid-to-liquid type of heat exchanger. The coolant, which is the other working fluid in this heat exchanger, heats the ATF more rapidly in the initial part of a drive cycle. This method of heating the transmission fluid is faster compared to a conventional system where the transmission fluid is heated only due to the heat...
generated in the transmission and torque converter. The authors state that simulations and experimental tests, considering the EPA city/highway cycle, indicate fuel economy improvements greater than 4.0% when using a thermal management system configuration shown, to heat transmission oil rapidly, over a conventional thermal management system. The fuel economy benefits of faster transmission oil warm-up are also outlined in [2.104] and [2.105].

Section 2.3. Overview of a Thermal Management Scheme

From the systems and components described in the previous section, it is seen that the majority of the heat recuperation technologies being used today entail using the exhaust heat energy to warm-up the coolant more rapidly. The coolant is then used to heat engine oil or the transmission fluid by using compact heat exchangers. With better thermal management, the coolant is able to be routed appropriately to various additional heat exchangers. As stated previously, Thermal Management Systems are employed to rapidly warm-up coolant, engine oil and transmission fluid to reduce viscosities and friction in cold start conditions. This leads to reduction in frictional losses in the engine and transmission and thereby improving fuel economy. According to estimates, fuel consumption can be reduced by up to 10% using advanced thermal management systems to recover heat leaving through the exhaust and coolant [2.16].

The project, of which this work is a part, focuses on Engine Thermal Management (ETM) and Ancillary Load Reduction (ALR). Ancillary Load Reduction part of this project aims towards developing control strategies that enable engine stop-start operation, reduce
parasitic losses through electrification of loads, possibly exploring crankshaft energy harvesting options, et cetera.

The scope of the Engine Thermal Management aspect of this project is to reduce fluid warm-up times during cold-start conditions. For this, an advanced Thermal Management System consisting of additional heat exchangers and valves to control coolant flow to these heat exchangers is used.

Referring to Figure 1.2, the devices that are used as part of the current Thermal Management System (TMS) are as follows,

1. Exhaust-to-coolant heat exchanger i.e. Exhaust Gas Cooler – EGRC
2. Compact coolant-to-transmission fluid heat exchanger i.e. Transmission Oil Heater - TOH
3. A three way valve controlling coolant flow to the TOH; the second branch of the three way valve is used to route flow of coolant to the cabin heater.
4. A valve controlling flow of coolant to the EGRC, and,
5. A valve used to bypass flow of exhaust to the EGRC.

The current work talks specifically about modeling this thermal management system and modeling components that exist in the proposed architecture. During warm-up, coolant can be routed to either the transmission oil heater or the cabin heater, using a three way valve, depending on which strategy provides the fastest warm-up times and best fuel economy. The EGR coolant valve can also be actuated to control coolant flow to the exhaust gas cooler (or coolant heater) and induce pressure drops throughout the coolant
flow circuit, which in turn allow varying the coolant flow rate through the engine oil cooler passively.

In 0, this system is described in detail and mathematical models are formulated to reproduce fluid flows and fluid warm-up characteristics, heat exchanger operations, engine heat rejection and so on. This is to say that, models are developed for describing physical phenomena influencing fluid warm-up characteristics. Before discussing the models developed as part of this work, however, an overview of the various techniques used to mathematically describe engine and thermal system behavior is presented in the following section.

Section 2.4. Approaches to Engine and Thermal System Modeling

Techniques used to model engine and thermal systems vary on the basis of number of dimensions in which phenomena are modeled and also on the basis of the time scale at which a that phenomenon is being modeled.

Some models are formulated to calculate the relevant physical parameters in three dimensions. For example, 3-D computational fluid dynamics (CFD) is used for careful design of engine block and to predict coolant flow around the engine block in order to avoid hot spots, for predicting the mechanical stresses on the main bearings on the crankshaft so that appropriate steps can be taken to avoid mechanical failure, and so on.

Also, with regards to the time-scale categorization, some models are capable of predicting very high frequency dynamics. For example, the formation of pollutants within the combustion chamber is studied by employing models that give the evolution of a flame front in the cylinder during the power stroke of the four-stroke IC engine on a
crank-angle basis or even more discrete than that. Applying chemical kinetics equations capable of describing the time rate of formation of pollutants, the equations are solved at extremely small time steps to calculate the amount of pollutants that form under the influence of various operating conditions of temperature and pressure as well as other important parameters (amount of fuel, spark timing, combustion chamber geometry etc). The hierarchy of models used to compute phenomenon of varying interests within the realm of engine modeling is shown in Figure 2.17.

Figure 2.17 Hierarchy of engine modeling techniques

Models that take anywhere between ~1000-5000 times longer, than the actual process being modeled, to compute the possible outcome are models that solve non-linear partial
differential equations at very small discretization lengths in space and time. These models are predictive in nature i.e. given enough information about the geometrical and other relevant parameters they are capable of predicting the outcome of the process being modeled, within good accuracy.[2.29][2.33].

Software packages like GT-Power, GT-cool, FLOWMASTER, AMESim, etc., also numerically solve non-linear partial differential equations such as continuity and momentum equations as well as some empirical correlations to solve problems at hand. Such models solve for the possible outcome of a process in one dimension only [2.30][2.31][2.32].

For development of models that are faster to solve and can be used to mimic the outcome of a process, much faster than 3-D CFD simulations, the 0-D modeling technique is employed. Faster solution times is a trade-off for accuracy, relative to 3-D CFD or 1-D analysis of a phenomenon, when using -0-D modeling techniques. In the 0-D modeling techniques, there are certain assumptions made which simplify the more complicated processes. 0-D models capture only the bulk effects of a system that behaves on a smaller scale (faster) than one desires to mimic. For example, the intake manifold, in a 0-D model had only one thermodynamic state of pressure and temperature. In reality, there exist tremendous instantaneous pressure variations between the inlet to the intake manifold and the intake runner, which is neglected by a 0-D model. The intake manifold is an example of a 0-D model where several components are lumped together (intake runners, manifold volume, manifold inlet section, bends and curves within the intake manifold, etc.).
0-D models are used to model the engine; here this class of models is known as Mean Value Engine Models, or MVEM. MVEMs are lumped parameter models that capture only low frequency spectrum of input/output behavior of the engine. Substantial information regarding the formulation of Mean Value Engine Models can be found in [2.113]-[2.126] The engine, as an energy-system, is divided into sub-components, thus this modeling technique operates at large discretization lengths and low-pass filtering in the time domain. The time resolution, as mentioned earlier, enables capturing of only low-frequency dynamics of the engine. To account for the unresolved scales (in space and time), calibration parameters such as discharge coefficients, efficiencies, heat transfer coefficients etc., are introduced and are calculated from test data for a given engine configuration. Flow control devices (volumes) and receivers (flow restrictions) are the elementary building blocks of a mean value engine model. Energy and mass conservation equation and other equations such as the ideal gas law are applied to finite control volumes which facilitate in defining the pressure and temperature state of a volume. Quasi-static relations for isentropic, compressible flow are applied to calculate the flow rate into volumes based on the thermodynamic state of a volume upstream and downstream of the receiver (flow restriction).

A type of 0-D modeling approach, called the lumped capacitance analysis method, is used to calculate the temperature variation, when a system undergoes a transient operation. Specifically, in the case of engines, this method is used to describe the temperature evolution of the metal parts of the engine block and head, as well as the temperature of the fluids that flow through the various cooling passages in the engine and
transmission. The lumped capacitance method assumes that the temperature variation within the various parts of the metal under consideration is negligible. This assumption part of the lumped capacitance method helps simplify many heat transfer problems associated with bodies that have complicated geometries. The 0-D modeling approaches mentioned for modeling engine air-path dynamics and engine thermal interactions are briefly described below.

2.4.1. Receivers (Volumes)

![Diagram](image)

Figure 2.18 Mean Value Engine Modeling: Receivers

In the MVEM approach, a receiver is modeled as a thermodynamic control volume to which laws of conservation of energy and conservation of mass are applied. Equation for conservation of mass is used to calculate the time rate of change of mass of gases within this control volume. The law of conservation of energy is used to formulate a first order differential equation for the temperature state within the control volume. The solution of these two equations enables calculation of the pressure within the control volume using the ideal gas law.
Conservation of energy for a control volume

\[
\frac{dE}{dt} = \left[ \dot{m}_{in} \left( h_{in} + \frac{V_{in}^2}{2} + z_{in} \right) - \sum \dot{m}_{out} \left( h_{out} + \frac{V_{out}^2}{2} + z_{out} \right) \right] + Q - W 
\]

Equation 2.1

The form of the law of conservation of energy used in the mean value engine model is simplified with the following assumptions:

- assuming internal energy is not affected by temperature variations
- inlet and outlet flow velocities are the same i.e. no difference in kinetic energy of gases entering and leaving the control volume
- the difference in elevation between inlet and outlet of the volume is negligible
- volume is static i.e. there are no moving parts and no work is done

These assumptions lead to a more simplified form of the energy conservation equation,

\[
m c_p \frac{dT}{dt} = \left[ \dot{m}_{in} h_{in} - \sum \dot{m}_{out} h_{out} + \dot{Q} \right] \]

Equation 2.2

Conservation of mass for a control volume

\[
\frac{dm}{dt} = \frac{d}{dt} \int \rho dV = \sum_{in} \left( \int_A \rho V_n dA \right)_{in} - \sum_{out} \left( \int_A \rho V_n dA \right)_{out} \]

Equation 2.3

The equation defining the law of conservation of mass is simplified with the following assumptions:

- fluid properties of gases is homogenous i.e. there are no local variations
- the product \( \rho V_n \) i.e. mass flux remains constant through a given cross-sectional area
\[
\frac{dm}{dt} = \sum \dot{m}_{in} - \sum \dot{m}_{out} 
\]
Equation 2.4

Ideal Gas law

\[
P = \frac{mRT}{V}
\]
Equation 2.5

Where, \( V \) = volume of the control volume, and \( R \) = gas constant for the gas under consideration.

2.4.2. Flow Control Devices (Flow Restriction)

For gases, the model for “flow restrictions” or “receivers”, in the realm of a MVEM approach, is based on the quasi-static relationships for isentropic, compressible flow through a restriction. The equations for flow of gas are passed on the ratio of upstream to downstream pressures and where this ratio stands with respect to the critical pressure ratio. For choked and sub-critical flow regimes [2.26], the equations for calculating mass flow rate of gases through a flow restriction are as follows,
Sub-critical flow, i.e. when,
\[
P_{\text{down}} \frac{P_{\text{up}}}{P_{\text{up}}} > \left( \frac{2}{\gamma + 1} \right)^{\frac{1}{\gamma - 1}}
\]
\[
\dot{m}_{\text{gas}} = C_d A_{\text{nom}} \frac{P_{\text{down}}}{P_{\text{up}}} \left( \frac{P_{\text{down}}}{P_{\text{up}}} \right)^{\frac{1}{\gamma}} \left\{ \frac{2\gamma}{\gamma - 1} \left[ 1 - \left( \frac{P_{\text{down}}}{P_{\text{up}}} \right)^{\frac{\gamma - 1}{\gamma}} \right] \right\}^{\frac{1}{2}}
\]  
Equation 2.6

Choked flow, i.e. when,
\[
P_{lM} \frac{P_{\text{amb}}}{P_{\text{amb}}} \leq \left( \frac{2}{\gamma + 1} \right)^{\frac{1}{\gamma - 1}}
\]
\[
\dot{m}_{\text{gas}} = C_d A_{\text{nom}} \frac{P_{\text{up}}}{R_{\text{gas}} T_{\text{up}}} \left( \frac{\gamma}{\gamma + 1} \right)^{\frac{\gamma + 1}{\gamma - 1}}
\]  
Equation 2.7

2.4.3. Lumped Thermal Capacitance Method

As mentioned earlier, the lumped capacitance method assumes that the temperature variation between different parts of the body under consideration is negligible. In lieu of this assumption, the use of 1-D heat equation is no longer valid, and the temperature state of the region under consideration is computed by applying an overall energy balance on that control volume defined around that solid [2.27]. The lumped capacitance model simplifies problems related to heat transfer in solids with complicated geometries, in that, it neglects the complicacy by assuming one “lump” as a representative of the entire solid/metal part under consideration. Therefore, only one temperature state exists for the entire body; which is the base assumption of this modeling approach.

In 0, the modeling approaches highlighted here are coupled with other novel modeling approaches found in literature and pertain to modeling of various system and subsystems that are part of thermal management systems. These complex and interconnected network
of models is put together to build a forward looking model of the vehicle energy consumption. This model is then used to shell out estimates of reduction in power and fuel consumption with the use of sensible engine thermal management schemes.
Section 2.5. References


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Chapter 3: Model Description and Development

This chapter describes the Vehicle Energy Simulator and the layout of the system-level models within the simulator used for fuel economy predictions. The chapter details the formulation and development process of various models that collectively amount to a large set of mathematical equations used to characterize the vehicle energy and power transfer dynamics. For the models formulated here, calibration and validation processes based on experimental data are then described and used to verify model implementation and correctness.

Section 3.1. Description of the Vehicle Energy Simulator (VES)

The Vehicle Energy Simulator, or the VES, is a forward-looking vehicle simulation tool capable of predicting low-frequency energy and power transfer dynamics. The VES enables prediction of engine fuel consumption and energy utilization for each of the main subsystems within the vehicle, e.g. the energy utilized by the electrical systems, mechanical systems, thermal and air-conditioning systems. The VES allows for calculation of fuel consumption for a given drive cycle, and is capable of predicting the direct impact of alternative actuator operation strategies on fuel consumption. The VES is formulated as part of a project funded by Chrysler LLC and the U.S. Department of
Energy (DoE), and is developed as a result of a collaborative modeling effort at Center for Automotive Research at The Ohio State University.

The purpose of VES development is to devise mathematical models that describe the energy consumption of engine and powertrain components, as well as model the energy transfer between various components. In this sense, the VES can be viewed as a tool to assist in vehicle energy analysis to identify potentials for reduction of fuel consumption and to eventually help in vehicle-level control system design to realize those opportunities.

Figure 3.1 shows a layout of the components modeled as part of the VES. The modeling approach adopted is based on an initial deconstruction of the vehicle system into a set of subsystems that participate in the energy and power transfer. This enables one to develop stand-alone models characterizing each subsystem, hence following a bottom-up modeling approach. Once models for different subsystems have been developed separately, they are finally coupled with each other to form a model of the entire vehicle system. As seen in Figure 3.1, the main subsystems considered in the VES are the engine air path system and torque production, the transmission, the torque converter, the powertrain thermal system, the electrical system (formed by battery, alternator, and electrical loads), and the air-conditioning system. There are also other intermediate subsystems and effects that need be modeled in order to capture their influence on vehicle energy consumption. For example, the vehicle longitudinal dynamics, the dynamic response of the driver, and the control algorithms governing the engine actuators, the
torque converter lock-up clutch, and the gear shifting scheduling need to be modeled to complete the VES.

Figure 3.1 Breakdown of components in the Vehicle Energy Simulator

For ease of VES formulation and delegation of modeling tasks, the subsystems are grouped into four main modules. The main modules, or systems, that are considered in the VES model development are represented pictorially in Figure 3.2.
The four main systems considered for model development are the mechanical system, electrical system, thermal system and the air-conditioning system. The following classification gives a list of the main components considered in the VES model, based on the energy and power transfer features.

1. Mechanical system
   a. Engine torque production
   b. Torque converter
   c. Transmission
   d. Vehicle longitudinal dynamics model
   e. Miscellaneous mechanical loads
   f. Driver

2. Electrical system
   a. Battery
   b. Alternator
   c. Miscellaneous electrical loads
3. Thermal system
   a. Engine heat rejection
   b. Heat exchangers
   c. Coolant heating and cooling system
   d. Oil heating and cooling system
   e. Transmission heat rejection
   f. Torque converter thermal characteristics
   g. ATF heating and cooling system

4. Air-Conditioning System
   a. Evaporator and condenser heat exchangers
   b. Compressor and mechanical drive
   c. Expansion valve

The VES is a result of a combined modeling effort to develop a comprehensive vehicle simulator for accurate fuel economy evaluation during driving cycles. For this reason, the number of systems taking part in energy interactions in a conventional powertrain is sizeable, and therefore, division of modeling efforts is essential. To this end, the current work describes only a subset of the systems and subsystems listed above. In particular, the focus of the current work is on the formulation, calibration, validation and application of models pertaining to the engine torque production and powertrain thermal management system. Some of the other components mentioned in the list above but not described in this work are part of [3.25].

The main systems for which models have been formulated, calibrated and explained in this work are listed in greater detail as follows.

1. Engine Air-Path Model
2. Engine In-cylinder Processes and Energy Balance Model
   a. Engine Torque Production
   b. Engine Heat Rejection

3. Engine Thermal Dynamics Model

4. Engine Thermal Management Systems Model
   a. Coolant Flow Network
   b. Slow Response Heat Exchangers (Radiator, Cabin heater)
   c. Fast Response Heat Exchangers (Engine Oil Cooler)
   d. Waste Heat Recovery System Model
      i. Exhaust Gas Cooler
      ii. Exhaust Gas Flow Network

5. Transmission Thermal Management Systems Model
   a. Transmission Fluid Flow Network
   b. Slow Response Heat Exchangers (Transmission Oil Cooler)
   c. Fast Response Heat Exchangers (Transmission Oil Heater)

Models for all of the above systems have been formulated and coupled so that the interplay between the different systems may be virtually simulated. Following the model formulation phase, calibration of the models is described, consisting of determining the unknown parameters in the models using known input-output characteristics of given components from experimental data. The calibration datasets are obtained from the manufacturer of respective components, if available, or are recorded from in-house
testing. Only a part of the available data set is used for calibration purposes (about 50-60%). Model validation is then conducted using the remaining part of the manufacturer data for given components.

In the following sections, key components of the models listed above will be described and analyzed. The formulation and calibration of the models is described in Section 3.2. Then a preliminary validation of models is conducted at the component-level using the remaining part of the available data. This last part is explained in Section 3.3.

**Section 3.2. Model Formulation and Calibration**

Models have been formulated using approaches that are able to characterize the low-frequency energy and power transfer dynamics of systems, which are most relevant for fuel economy evaluation during driving cycles. Most of the models developed rely on a physically-based approach, namely they are formulated using the laws of mass and energy conservation, in conjunction with constitutive relations for the working fluids (e.g., the ideal gas law). Because the models are simplified in various ways, a number of calibration parameters will be present, requiring identification from performance data of the respective components or systems.
The model of the engine air path is based on the Mean Value Engine Modeling (MVEM) approach [3.1] [3.18] [3.19] [3.21], which characterizes the low frequency dynamics in the engine breathing system, as averaged over multiple engine cycles. Furthermore, the model predicts the effects of the air path variables on the engine torque output and heat rejection to coolant and oil. Figure 3.3, which is a functional representation of the engine model, shows that each of the engine components and the respective models are closely coupled with each other. The engine specifications for this study are shown below in Table 3.1.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Displacement (L)</td>
<td>3.6</td>
</tr>
<tr>
<td>Bore/Stroke (mm)</td>
<td>96/83</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>10.2</td>
</tr>
<tr>
<td>Max. Power (kW @ RPM)</td>
<td>216 @ 6350</td>
</tr>
<tr>
<td>Max. Torque (Nm @ RPM)</td>
<td>353 @ 4400</td>
</tr>
</tbody>
</table>
The model for the throttle valve takes as inputs the ambient pressure, ambient temperature, and the pressure in the intake manifold, downstream of the throttle. Based on these inputs, the model predicts the flow rate of air through the throttle. The intake manifold, which acts as a receiver, is modeled by applying the mass and energy equations to calculate the temperature and pressure as functions of time. The engine air path is modeled by considering a series of interconnected receivers and flow restrictions, as described in Section 2.4. Some of the component models are based on a gray-box modeling approach, which attempts at capturing the low frequency behavior with actual, physics-based equations (conservation laws). This approach has the advantage of requiring a low number of calibration parameters that must be calculated using experimental data for best accuracy [3.20].

The lumped capacitance modeling approach is used, in this work, to model heat exchangers and the engine thermal dynamics. This approach provides a simple method to model the transient heat transfer characteristics of components with complex geometries, such as the engine and heat exchangers. This method allows one to combine masses and thereby their thermal capacities into discrete lumps. Lumping of masses together helps in avoiding the process of taking complex geometries of components into account, which can be time-consuming and not suited for control-oriented models [3.10]. In the current modeling scheme, the mass flow rates and inlet temperatures of the two interacting fluids for a given heat exchanger are known, while the outlet temperatures must be predicted by the model. For this reason, the epsilon-NTU method [2.27] is used here in conjunction with the lumped capacitance analysis method to predict the outlet temperatures, together
with the dynamics related to the heat transfer process, as well as the thermal inertias of the fluids and of the metal masses.

The model for the in-cylinder processes and energy balance predicts the heat rejection from the combustion gases to the walls of the combustion chamber depending on the engine operating conditions as well as predicts the torque output characteristics of the engine. The model for the heat rejection to the walls of the combustion chamber is a detailed model taking into account the heat releases and heat absorptions involved in the formation of pollutants within the combustion chamber. The torque production model takes inputs from the Mean Value Engine Model and predicts the torque output of the engine.

The thermal system model predicts the thermal dynamics of the engine block, the heat exchangers present in the cooling system and a static characterization of the coolant flow through the various heat exchangers. The thermal system model also predicts the flow of exhaust gases through piping downstream of the exhaust manifold, which is necessary for evaluating the catalyst warm-up and possible applications of waste heat recovery.
3.2.1. Engine Air-Path Model

The Mean Value Engine Modeling (MVEM) approach is utilized in this work to model the low frequency dynamics in engine air breathing as well as its effect on engine torque production. Models have been developed for predicting the air flow through the throttle body, volumetric efficiency of the engine at different operating conditions, intake manifold temperature and pressure dynamics, fuel flow dynamics, engine torque production, exhaust port temperature, and pressure and temperature within the exhaust manifold volume. Selected models have then been calibrated using steady-state engine dynamometer data.

3.2.1.1 Throttle

Following a quasi-static modeling approach, the flow of air through the throttle is modeled using the equations for isentropic, compressible flow through a restriction [3.2]. The isentropic flow equations are bifurcated for when the flow is choked or sub-critical, depending on the pressures upstream and downstream of the throttle valve. The flow through the valve is sub-critical when the ratio of downstream pressure to upstream pressure is greater than the critical pressure ratio \( \left( \frac{2}{\gamma + 1} \right)^{\frac{\gamma}{\gamma - 1}} \), where \( \gamma \) is the ratio of specific heats of air. Conversely, the flow through the valve is said to be “choked” when the ratio of downstream pressure to upstream pressure is less than or equal to the critical pressure ratio. The expressions for calculating mass flow rate of air through the throttle plate, at sub-critical and choked flow conditions, are given as follows:

\[
\text{Sub-critical flow:} \quad \frac{p_{IM}}{p_{amb}} > \left( \frac{2}{\gamma + 1} \right)^{\frac{\gamma}{\gamma - 1}} \quad \text{Equation 3.1}
\]
\[
\dot{m}_{thr} = \frac{C_d A_{thr} P_{amb}}{\sqrt{R_{\text{air}} T_{\text{amb}}}} \left( \frac{P_{IM}}{P_{\text{amb}}} \right)^{\frac{1}{\gamma}} \left( \frac{2}{\gamma - 1} \left[ 1 - \left( \frac{P_{IM}}{P_{\text{amb}}} \right)^{\frac{\gamma - 1}{\gamma}} \right] \right)^{\frac{1}{2}}
\]

Choked flow: \( \frac{P_{IM}}{P_{\text{amb}}} \leq \left( \frac{2}{\gamma + 1} \right)^{\frac{\gamma}{\gamma - 1}} \)

Equation 3.2

\[
\dot{m}_{thr} = C_d A_{thr} P_{amb} \sqrt{R_{\text{air}} T_{\text{amb}}} \frac{\gamma}{\gamma - 1} \left( \frac{2}{\gamma + 1} \right)^{\frac{\gamma + 1}{\gamma - 1}}
\]

In Equation 3.1 and Equation 3.2, the value of the discharge coefficient \( C_d \) must be identified experimentally from steady-state engine dynamometer data for a range of operating conditions, varying engine speed and load. The discharge coefficient is generally expressed as \( C_d = f(\alpha_{thr}, N) \). The first step in calibrating the discharge coefficient of the throttle is to invert the relationship in Equation 3.1 and Equation 3.2 to yield \( (C_d) \) as shown below. [3.1][3.26].

Sub-critical flow: \( \frac{P_{IM}}{P_{\text{amb}}} > \left( \frac{2}{\gamma + 1} \right)^{\frac{\gamma}{\gamma - 1}} \)

\[
C_d = \frac{\dot{m}_{thr} \sqrt{R_{\text{air}} T_{\text{amb}}} \left( P_{\text{amb}} \right)^{\frac{1}{\gamma}}}{A_{thr} P_{\text{amb}}} \left( \frac{2}{\gamma - 1} \left[ 1 - \left( \frac{P_{IM}}{P_{\text{amb}}} \right)^{\frac{\gamma - 1}{\gamma}} \right] \right)
\]

Equation 3.3

Choked flow: \( \frac{P_{IM}}{P_{\text{amb}}} \leq \left( \frac{2}{\gamma + 1} \right)^{\frac{\gamma}{\gamma - 1}} \)

\[
C_d = \frac{\dot{m}_{thr}}{A_{thr} P_{\text{amb}}} \sqrt{R_{\text{air}} T_{\text{amb}}} \left( \frac{2}{\gamma + 1} \right)^{\frac{\gamma + 1}{\gamma - 1}}
\]

Equation 3.4

Once the array of discharge coefficients is obtained for available data points from steady state engine tests belonging to the calibration dataset, surface fitting routines in Matlab are used and a surface is created to represent the discharge coefficient as a function of
throttle opening and engine speed. The resulting surface calibrated for the test engine is shown in Figure 3.4. This surface is implemented as a look-up table in the engine model.

![Calibrated surface of throttle discharge coefficient as a function of throttle opening and engine speed]

Figure 3.4 Calibrated surface of throttle discharge coefficient as a function of throttle opening and engine speed

### 3.2.1.1 Volumetric Efficiency

The volumetric efficiency model is adopted from [3.3]. The model entails expressing the volumetric efficiency as a function of engine speed and intake manifold pressure, as given by:

$$
\lambda_v = s_i(N) + \frac{\gamma_i(N)}{P_{IM}}
$$

Equation 3.5
where the terms \( s_i(N) \) and \( y_i(N) \) are calibration parameters, which depend primarily on the engine speed. It is seen that, for this test engine, the dependence of the parameters on the engine speed can be represented by a linear relationship. Therefore:

\[
s_i(N) = s_{i1} + s_{i2} \cdot N
\]

\[
y_i(N) = y_{i1} + y_{i2} \cdot N
\]

Equation 3.6

The model development is initiated by first calculating the volumetric efficiency from engine steady-state experimental data using the speed-density equation [3.1].

\[
\dot{m}_{\text{air,eng}} = \frac{\lambda_v V_d P_{\text{IM}} N}{120 R_{\text{IM}} T_{\text{IM}}}
\]

Equation 3.7

The volumetric efficiency is then calculated based on the mass flow rate of air to the engine, intake manifold pressure, engine speed, and intake manifold temperature by inverting Equation 3.7. The calibration dataset, available from steady-state engine tests containing these engine variables, is used to obtain the array of volumetric efficiencies for the respective engine operating conditions. This array of volumetric efficiency, along with the other engine variables, is then used to calibrate the terms \( s_i(N) \) and \( y_i(N) \) for the volumetric efficiency model. The calibration terms in Equation 3.6 are then computed by applying a simple linear regression:

\[
\beta = (X^T X)^{-1} X^T Y
\]

Equation 3.8

Where,

\[
\beta = [s_i(N), y_i(N)]; \ X = \left[1, \frac{1}{P_{\text{IM}}} \right]; \ Y = [\lambda_v].
\]
The linear fit of calibration parameters $s_i(N)$ and $y_i(N)$ as functions of engine speed are seen in the following figures.

Figure 3.5 Calibration parameter $s_i$ in the VE model

Figure 3.6 Calibration parameter $y_i$ in the VE model
The values of the slopes and intercepts for the curve fit of the model parameters are the values of the terms $s_{t1}, s_{t2}, y_{t1}, y_{t2}$, tabulated in Table 3.2.

<table>
<thead>
<tr>
<th>Calibration parameter</th>
<th>Units</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$s_{t1}$</td>
<td>-</td>
<td>0.7963</td>
</tr>
<tr>
<td>$s_{t2}$</td>
<td>1/RPM</td>
<td>$5.8455 \times 10^{-5}$</td>
</tr>
<tr>
<td>$y_{t1}$</td>
<td>kPa</td>
<td>-19.0422</td>
</tr>
<tr>
<td>$y_{t2}$</td>
<td>kPa/RPM</td>
<td>0.0022</td>
</tr>
</tbody>
</table>

Figure 3.6 shows considerable scatter in the fit of the calibration parameter $y_{t1}$. This variation can be explained by the fact that this engine has variable valve actuation capabilities, which the model does not explicitly consider. However, on careful review of Equation 3.5 it is seen that the term containing the calibration parameter $y_{t1}$ is divided by the intake manifold pressure (in kPa) term. This makes the contribution of this term very small compared to the first one and, therefore, the error in the final model is not significant. This is verified by a comparison of the VE calculated by the model against the VE calculated using experimental data, as seen in the Figure 3.7. From this figure it is inferred that the predicted VE is in agreement with what is observed directly from experimental data.
Figure 3.7 Comparison: model versus experimental results

3.2.1.2 Intake Manifold

The intake manifold is modeled using the conservation of mass and energy and the ideal gas law [3.4]. The first law of thermodynamics is used to calculate the temperature of the gases in the intake manifold as shown by Equation 3.9. The mass conservation equation is applied to the volume of the intake manifold and is used to calculate the mass of gases in the intake manifold at any given time as given by Equation 3.10. The mass in the intake manifold and the intake manifold temperature are then used to calculate the pressure in the intake manifold using the ideal gas law.
\[
\frac{dT_{IM}}{dt} = \frac{1}{m_{IM}c_{v_{IM}}} \dot{m}_{th} c_{p_{air}} T_{amb} + \left( c_{v_{IM}} \frac{dm_{IM}}{dt} - \dot{m}_{eng} c_{p_{IM}} \right) T_{IM} \]
\[
- h_{wall} \Delta A_{IM} \left( T_{IM} - T_{w_{IM}} \right) + \dot{m}_{fuel} c_{p_{fuel}} T_{fuel} + \dot{Q}_{LHV} \]

Equation 3.9

\[
\frac{dm_{IM}}{dt} = \dot{m}_{th} + \dot{m}_{fuel} - \dot{m}_{eng} \]

Equation 3.10

\[
P_{IM} = \frac{m_{IM} R_{IM} T_{IM}}{V_{IM}} \]

Equation 3.11

In the above equations, the term \( h_{wall} \) represents the convection heat transfer coefficient from the walls of the intake manifold to the ambient and \( x_{vap} \) is the fraction of fuel that vaporizes in the intake port after injection. The manifold absolute pressure, \( P_{IM} \), is an input to the throttle model, which calculates the mass flow rate of air through the throttle.

The thermodynamic properties of the gas mixture in the intake manifold are a weighted sum of the thermodynamics properties of the constituent gases, i.e., air and fuel, and are calculated using the set of equations marked as Equation 3.12.

\[
c_{p_{IM}} = \frac{\dot{m}_{th} c_{p_{air}} + \dot{m}_{fuel} c_{p_{fuel}}}{\dot{m}_{th} + \dot{m}_{fuel}} \]

Equation 3.12

\[
c_{v_{IM}} = \frac{\dot{m}_{th} c_{v_{air}} + \dot{m}_{fuel} c_{v_{fuel}}}{\dot{m}_{th} + \dot{m}_{fuel}} \]

\[
R_{IM} = c_{p_{IM}} - c_{v_{IM}} \]

The thermodynamic properties of the component gases are calculated using information available in JANAF tables [3.5] and summarized in [3.6]. For air, it is assumed that it is constituted by oxygen (~21%) and nitrogen (~79%) only.
3.2.1.3 Fuel Evaporation Dynamics

A model of the fuel evaporation is considered here to account for the fuel injection and evaporation dynamics in the intake ports of the engine. The dynamics of fuel evaporation on the walls of the intake ports was first modeled by Aquino [3.7]. Aquino’s model assumes that a fraction of the injected fuel is in liquid form and forms a film on the port walls while the rest enters the combustion chamber directly in vapor form. A certain amount of fuel, from the liquid fuel film on the walls, then vaporizes and enters the combustion chamber while exhibiting characteristic dynamics. This model assumes that fuel entering the combustion chamber is entirely in vapor phase and that no fuel is transported into the engine in liquid state. According to Aquino’s model, the fraction of fuel from the film that enters the combustion chamber in vapor form is directly proportional to the mass of the fuel in the film:

\[
\text{Mass of fuel in the film} \\
\frac{d m_f}{dt} = m_{f,L} - m_{f, ev} = X \dot{m}_{fi} - \frac{1}{\tau} m_{fuel} \\
\text{Fraction of injected fuel mass that vaporizes} \\
\dot{m}_{fi, v} = (1 - X) \dot{m}_{fi} \\
\text{Mass of fuel entering the cylinder} \\
\dot{m}_{fuel} = (1 - X) \dot{m}_{fi} + \frac{1}{\tau} m_f
\]
In cases where the lambda ($\lambda$) value (ratio of actual air-fuel ratio to stoichiometric air-fuel ratio for that fuel) is given rather than the mass of fuel injected, then the mass of the fuel injected can be calculated as,

$$\dot{m}_{f/l} = \frac{\dot{m}_{thr}}{\lambda \cdot (AFR)_{st}}$$

### 3.2.1.4 Exhaust Manifold

The exhaust manifold is modeled in a similar fashion to the intake manifold. The first law of thermodynamics when applied to the control volume of the exhaust manifold provides a first order equation for the temperature state of the gases in the exhaust manifold, given by Equation 3.16. The mass conservation equation in the exhaust manifold, as represented by Equation 3.17, gives the mass of exhaust gases in the exhaust manifold at any time. The exhaust manifold pressure is then calculated using the ideal gas law.

$$\frac{dT_{EM}}{dt} = \frac{1}{m_{EM}c_{v,EM}} \left[ \dot{m}_{eng}c_{p,exh}T_{EM} - \dot{m}_{CAT}c_{p,EM}T_{EM} ight]$$

$$- h_{wall}A_{EM}(T_{EM} - T_{wall_{EM}})$$

$$- c_{v,EM}T_{EM} \frac{d(m_{EM})}{dt}$$

Equation 3.16

$$\frac{dm_{EM}}{dt} = \dot{m}_{eng} - \dot{m}_{CAT}$$

Equation 3.17

$$P_{EM} = \frac{m_{EM}R_{EM}T_{EM}}{V_{EM}}$$

Equation 3.18
3.2.2. Engine In-cylinder Processes and Energy Balance Model

This sub-section describes the models of in-cylinder processes such as heat rejection from the combustion gases to the walls of the chamber, engine torque production, and exhaust gas temperature at the exhaust ports. Models of the friction generated during engine operation as well as the net torque output (brake torque) are explained in this section.

3.2.2.1 Engine Heat Rejection

The rate of heat rejection from the combustion gases to the walls of the combustion chamber is calculated using an energy balance equation applied to the control volume around the combustion chamber [3.1]. Assuming steady-state conditions, the cycle-averaged energy balance existing in the combustion chamber can be described as,

\[ \dot{m}_{\text{fuel}} h_{\text{fuel}} + \dot{m}_{\text{air}} h_{\text{air}} = \dot{Q}_{\text{gas}} + (\dot{m}_{\text{fuel}} + \dot{m}_{\text{air}}) h_{\text{exh}} + P_{\text{ind}} + P_{\text{pump}} \]  \hspace{1cm} \text{Equation 3.19}

![Combustion Chamber Diagram](image)

Figure 3.8 Control volume: combustion chamber

Equation 3.19 takes into the account the rate of energy addition to the combustion chamber (air and fuel entering the combustion chamber), the rate of indicated work, the
rate of pumping work as well as the energy contained in the exhaust gases leaving the control volume.

The rate of energy addition into the combustion chamber is calculated by summing the contributions of air and fuel entering the control volume. The mass flow rates of air and fuel entering the combustion chamber are calculated through the volumetric efficiency models and the fuel evaporation dynamics model, respectively. The right-hand side of Equation 3.19 is computed with inputs from the IMEP and PMEP models.

\[
P_{\text{ind}} = \frac{\text{IMEP} \cdot V_d \cdot N}{120} \quad \text{Equation 3.20}
\]

\[
P_{\text{pump}} = \frac{\text{PMEP} \cdot V_d \cdot N}{120} \quad \text{Equation 3.21}
\]

The mass flow rate of exhaust gas is calculated as the sum of the mass flow rates of fuel and air. The rate of heat rejection from the combustion gases to the walls of the combustion chamber is then obtained by difference, knowing all other quantities. The calculation of enthalpies of the fuel, air and the exhaust gas, is based on [3.6]. The routines published in [3.6], are used to calculate the enthalpy of gases by taking into account the molar fractions of constituent gases. The molar fractions of constituent gases are present in the form of a vector, along with molar fractions of 8 other gases. The order followed is \([\text{CO}_2 \ H_2\text{O} \ \text{N}_2\text{O} \ \text{CO} \ H_2 \ O \ OH \ \text{NO}]\).

In this model, it is assumed that air constituted by \(~79\%\) Nitrogen (\(\text{N}_2\)) and \(~21\%\) Oxygen (\(\text{O}_2\)); the molar fractions of all gases other than nitrogen and oxygen are taken to be zero (0) across the row for calculation of thermodynamic properties of air. Based on the temperature of the air in the intake manifold, the enthalpy and specific heat capacity
at constant pressure are calculated and used in the energy balance equation. The calculation of the thermodynamic properties for individual components of air is based on polynomials that are temperature dependent. The polynomials calculating thermodynamic properties of air are different for two temperature ranges (300K-1000K and 1000K-5000K). After the thermodynamic properties of the constituent gases (nitrogen and oxygen) are calculated, the weighted sum of these properties, based on molar fractions of the individual gases, enables calculation of the thermodynamic properties of the entire mixture of air.

For fuel, the thermodynamic properties are calculated based on the type of fuel and the temperature in the fuel rail. In this case, the type of fuel is Gasoline.

For the exhaust gases, unlike air and fuel, it is impossible to give as input the molar fractions of constituent gases as a known quantity. The model proposed in [3.6] enables calculation of molar fraction of burned and unburned species of the constituent gases in the exhaust based on the exhaust pressure, exhaust temperature and the equivalence ratio. The calculation of the molar concentration is then converted to mass fractions by multiplying the molar fractions by the molecular masses of each of the considered species (in the order shown above). The dependence on the exhaust pressure is very weak, but still included in the model for added accuracy. The enthalpy and specific heat capacity of the mixture is then calculated as the mass averaged sum of the individual exhaust gas components. These properties are then used in the energy balance equation to calculate the rate of heat rejection to the combustion chamber walls, by difference. The enthalpy of the exhaust gases is the enthalpy of formation, and not the sensible enthalpy. Therefore,
the enthalpy calculated in this case is a sum of the sensible enthalpy as well as the reference state enthalpy. For a detailed description of the procedure used to calculate the thermodynamic properties of air, fuel and the exhaust gases, refer to [3.6].

As an example, Figure 3.9 shows a graph of $c_p$ of exhaust gases versus exhaust temperature for different equivalence ratios. It is seen that the specific heat capacity of exhaust gas increases as the temperature of the exhaust gases increase. Also, the specific heat capacity of exhaust gas increase as the equivalence ratio increases. Similar trend is seen in the value for $c_v$, as seen in Figure 3.10.

Figure 3.9 $c_p$ of exhaust gases, as a function of temperature and equivalence ratio
3.2.2.2 Engine Torque Production

In the current approach, the engine torque production model is based on a “black-box” approach, namely is given by a set of maps for mean effective pressures (MEP) used to calculate the torque, based on:

\[
Torque = \left( \frac{V_d}{2\pi n_r} \right) MEP
\]

Equation 3.22

The Pumping Mean Effective Pressure (PMEP) and Indicated Mean Effective Pressure (IMEP), obtained from steady state engine dynamometer tests, are represented as functions of engine speed and manifold pressure (MAP) and may be represented by Equation 3.23, Equation 3.24 respectively.
\[ IMEP_g = f(N, MAP) \]  
Equation 3.23

\[ PMEP = f(N, MAP) \]  
Equation 3.24

The IMEP is calculated from in-cylinder pressure data from steady state engine dynamometer tests. The value obtained is the gross indicated mean effective pressure, and does not include the pumping losses during the charge exchange process part of the cycle. For illustrative purposes, the gross IMEP and PMEP for the 3.6L V6 engine are shown in Figure 3.11 and Figure 3.12, respectively.

Figure 3.11 IMEP surface as a function of engine speed and intake manifold pressure
3.2.2.3 Engine Friction and Torque Output Model

The engine Friction Mean Effective Pressure (FMEP) is calculated from motoring tests conducted in an engine dynamometer test cell. Motoring tests are used to characterize the rubbing and accessory friction from coolant and oil pumps, which cannot be detached from this particular engine. The alternator load, however, is not included in this data set recorded during steady state engine dynamometer testing as there is no alternator attached to the engine in engine dynamometer test. In this model, the FMEP is represented as a function of engine speed and Indicated Mean Effective Pressure:

\[ FMEP = f(N, IMEP) \]  

Equation 3.25

Figure 3.12 PMEP surface as a function of engine speed and MAP
The Shayler factor [3.8] is used to scale the steady state FMEP, measured at a constant reference temperature, using coefficients from experimental data recorded during oil-warm-up. As a result of scaling, FMEP is now a function of engine speed, IMEP as well as oil temperature.

\[ FMEP = f(N, IMEP, T_{oil}) \]

The Shayler factor implementation is conducted as,

\[ FMEP (T_{oil}) = FMEP (T_{oil}^*) \cdot \left( \frac{v(T_{oil})}{v(T_{oil}^*)} \right)^{\alpha} \]

\{For \ T_{oil}^* = 100^\circ C, \ \alpha = 0.24\}

FMEP is represented below, for fully warmed-up conditions, as a function of engine speed and IMEP.

![FMEP surface as a function of engine speed and IMEP](image)

Figure 3.13 FMEP surface as a function of engine speed and IMEP
The Brake Mean Effective Pressure is calculated using the output of the three models for IMEP, PMEP and FMEP, as follows:

\[ BMEP = IMEP - PMEP - FMEP \]  \hspace{1cm} \text{Equation 3.27}

Calculation of BMEP enables calculation of brake torque observed at the flywheel.

\[ T_b = \left( \frac{V_d}{2\pi n_r} \right) BMEP \]  \hspace{1cm} \text{Equation 3.28}

3.2.2.4  \hspace{0.5cm} \textit{Exhaust Temperature}

The model for temperature of exhaust gases at the exhaust port is taken from [3.9]. According to this model, temperature of exhaust gases at the exhaust port is a function of engine speed, Indicated Mean Effective Pressure (IMEP) and temperature of coolant out of the engine. The polynomial function expressing exhaust gas temperature at the exhaust port as a function of factors mentioned above is given by Equation 3.29.

\[ T_{exh} = T_{cint-engout} + aN + bN^2 + (c + dN + eN^2)IMEP \]  \hspace{1cm} \text{Equation 3.29}

The coefficients are determined using MATLAB’s surface fitting toolbox, followed by rigorous verification of the results from the model against the available data from engine tests. Mean error and standard deviation of error in exhaust gas temperature prediction are taken as measures of goodness of model. A comparison between model and experimental data points is seen in Figure 3.14 while Figure 3.15 shows a histogram depicting the spread of error in exhaust temperature prediction.
Figure 3.14 Exhaust temperature model: comparison with engine test results

Figure 3.15 Error and std. dev. in prediction of the exhaust temperature
Figure 3.14 and Figure 3.15 show the agreement between the exhaust temperatures predicted by the model and the measurement from steady-state engine tests. The average percentage error in the exhaust temperature prediction is less than 1%, while the standard deviation is approximately equal to 3%.

3.2.3. Engine Thermal Dynamics Model

A model for the thermal dynamics of the engine (including the metal masses, oil passages, coolant flow jackets, coolant, oil) is designed to predict the temperature of the different parts within the engine, the temperature of the coolant exiting the engine cooling jackets and the temperature of the oil in the oil sump. The basis of this mathematical formulation is the first law of thermodynamics i.e. the conservation of energy for a control volume. Based on the first law of thermodynamics the time rate of change of energy contained within the control volume at time \( t \) equals the sum of the net rate at which energy is being transferred in by heat transfer at time \( t \) and the net rate of energy transfer into the control volume accompanying mass flow less the net rate at which energy is being transferred out by work at that instant of time [3.4]. This principle is applied assuming negligible temperature differences within the different parts of the control volume. This assumption leads to the use of the lumped capacitance analysis technique [3.10].
Consider Figure 3.16, a cross-sectional view of the engine, which divides the engine into four lumped thermal masses. Mass 1 ($m_1$) is in direct contact with the combustion gases i.e. “Hot Masses” and consists of the top part of the piston, the walls of the combustion chamber, piston rings, liner, parts housing the spark plugs, exhaust ports and intake & exhaust valves. Mass 2 ($m_2$) consists of reciprocating metals parts and therefore splashing in oil; the undersides of the pistons, connecting rods, crank train, and oil squirters. The oil sump is considered to be mass 3 ($m_3$), exchanging heat with oil on one side and ambient air on the other side. The head, outer coolant jacket i.e. the “Cold
Masses” are accounted for in mass 4 \((m_4)\). There are two more control volumes defined, one each for the coolant and the engine oil, as seen in Figure 3.17.

![Figure 3.17 Control volumes: coolant and engine oil](image)

The sources of heat in this system are the heat released during formation of the combustion gases and heat generated due to friction. Friction heat is generated in the contact between piston rings and walls of the combustion chamber, as well as due to friction generated in other regions of the reciprocating assembly which include the interface between the king pin and the connecting rod, and the surface between the main bearings on the crankshaft and the connecting rod.

As depicted in Figure 3.18, mass 1 receives heat from heat released due to formation of combustion gases and also receives part of the heat generated due to friction. Out of the total heat generated due to friction, only a fraction \(\alpha\) is assigned to mass 1 based on the bifurcation of piston masses between mass 1 and mass 2. Mass 1 exchanges heat with mass 2 via conduction, and with coolant and engine oil by virtue of forced convection.
Equation 3.30 describes the rate of change of temperature of mass 1, taking into account all these different contributions.

\[
\frac{dT_1}{dt} = \frac{1}{m_1 c_1} \left[ Q_{gas} + \alpha Q_{fr} - U_{1\rightarrow oil}(T_1 - T_{oil}) \right. \\
- \left. U_{1\rightarrow clnt}(T_1 - T_{clnt}) - \frac{1}{R_{1\rightarrow 2}} (T_1 - T_2) \right]
\]

Equation 3.30

The only heat input to mass 2 is a fraction $\beta$ of the total heat generated due to friction. Mass 2 exchanges heat with mass 1, as mentioned earlier. The oil interacts thermally with the parts included in mass 2 in two ways; oil is squirted on the underside of the piston induces forced convection and the crank train splashes around in oil during engine operation. All these effects are summed in Figure 3.19.
Equation 3.31 describes the rate of change of temperature of mass 2.

\[
\frac{dT_2}{dt} = \frac{1}{m_2c_2} \left[ \beta \dot{Q}_{fr} + \frac{1}{R_{1 \rightarrow 2}} (T_1 - T_2) - U_{2 \rightarrow oil}(T_2 - T_{oil}) \right]
\]

Equation 3.31

Mass 3 \((m_3)\) is heated by the engine oil which is generally at a higher temperature and also exchanges heat with the ambient by forced convection between the outer surface of the oil sump and the ambient. Equation 3.32 describes temperature dynamics of mass 3.

\[
\frac{dT_3}{dt} = \frac{1}{m_3c_3} \left[ U_{oil \rightarrow 3}(T_{oil} - T_3) - U_{3 \rightarrow amb}(T_3 - T_{amb}) \right]
\]

Equation 3.32
Mass 4 ($m_4$) interacts thermally with the coolant and the air. The air around “mass 4” is at a higher temperature than the ambient air due to the conditions influencing the underhood conditions. Both these terms are seen in Equation 3.33.

\[
\frac{dT_4}{dt} = \frac{1}{m_4 c_4} [U_{cint \rightarrow 4}(T_{cint} - T_4) - U_{4 \rightarrow amb}(T_4 - T_{amb})] \quad \text{Equation 3.33}
\]

The equations for the rate of change of coolant and oil temperature include terms similar to those in Equation 3.30 to Equation 3.33 with the addition of the term taking into account the net rate of energy transfer accompanying mass flow of the coolant and oil into the respective control volumes. The coolant and engine oil temperature dynamics are explained by Equation 3.34 and Equation 3.35, respectively.

\[
\frac{dT_{cint}}{dt} = \frac{1}{m_{cint} c_{cint}} \left[ \dot{m}_{cint} c_{p_{cint}} (T_{cint_{in}} - T_{cint}) + U_{1 \rightarrow cint}(T_1 - T_{cint}) - U_{cint \rightarrow 4}(T_{cint} - T_4) \right] \quad \text{Equation 3.34}
\]
Equation 3.35

\[
\frac{dT_{oil}}{dt} = \frac{1}{m_{oil}c_{oil}} \left[ \dot{m}_{oil}c_{p,oil}(T_{oil,\text{in}} - T_{oil}) + (1 - \alpha - \beta)\dot{Q}_{fr} \\
+ U_{1\rightarrow oil}(T_1 - T_{oil}) + U_{2\rightarrow oil}(T_2 - T_{oil}) \\
- U_{oil\rightarrow 3}(T_{oil} - T_3) \right]
\]

3.2.4. Engine Thermal Management System

An engine thermal management system is devised as part of this project, which consists of a complex network of heat exchangers and fluid flow control valves. This system is designed to distribute the heat contained in the coolant in a way such that the temperatures of engine and powertrain fluids reach their desired set-point as rapidly as possible without violating system constraints.

In Figure 3.22, a layout of the components in the engine thermal management system is shown. The proposed system for managing the thermal characteristics of the engine and related fluids consists mainly of an engine oil cooler, and an exhaust gas cooler; both of which have engine coolant as the second working fluid. The transmission oil heater is shown in this figure as it affects flow of engine coolant but it is discussed in detail in section 3.2.7.

The amount of coolant flowing through the various branches of the cooling circuit is decided by the position of the EGR cooler - coolant side valve and the three way valve, whose positions are controllable. The coolant flow rate through the various branches is also affected by the thermostat opening but the thermostat is passive in its operation.
The amount of exhaust gases routed through the exhaust gas cooler is based on the position of the exhaust bypass valve. A fully closed bypass valve would entail all of the engine exhaust flowing through the EGR cooler. On the other hand, even if the bypass valve is fully open, there is still some flow of exhaust through the EGR cooler.

The flow of engine oil as well as coolant to the engine oil cooler is not controllable. The flow of coolant to the engine oil cooler depends on the pressure drop in the rest of the coolant flow circuit.
The three way valve, which controls the flow of coolant to the transmission fluid heater and the cabin heater, is a complimentary valve. This means that the sum of the opening towards these two heat exchangers is always 100%.

In the next few sections, the model for determining coolant flow rate, the thermal models for the heat exchangers in the engine thermal management system, and the exhaust flow system model will be discussed in detail.

### 3.2.4.1 Coolant Flow Circuit

The coolant flow circuit, part of the current engine thermal management system, is such that the coolant flows through five heat exchangers, plumbing to and from those heat exchangers, the engine block and head, and through the flow control valves in the coolant flow circuit. A general layout of all relevant components in the coolant flow path is shown in Figure 3.23. The flow control devices in this coolant flow circuit are the three way valve, the EGR coolant valve, the thermostat, and the coolant pump. All of these actuators affecting coolant flow are controllable except for the thermostat, which is passive in its operation. The 3-way valve used in the current architecture is a complimentary 3-way valve with one inlet and two outlets, with each outlet routing coolant to a heat exchanger. The combined opening of the valve towards the two heat exchangers is 100%. Therefore, if the fractional opening of the 3-way valve towards one of the flow branches is specified, the default opening towards the second branch is the difference between complete opening and the opening towards the first branch.

All of the coolant is pumped through the engine block and head following which the coolant splits into five different paths. Following the engine block and head, the coolant
flows through the radiator if the thermostat is open. Depending on the position of the coolant valve before the exhaust gas cooler, coolant can flow through the exhaust gas cooler. Coolant flow through the engine oil cooler (EOC) is passive i.e. there is no active control imposed on the flow of coolant though the EOC. Therefore, there is always coolant flow through the Engine Oil Cooler. The coolant split depends on the pressure drops offered by each path which is decided by the position and operation of the flow control devices in the coolant flow circuit. The flow through the transmission oil heater and the cabin heater core is decided by the position of the 3-way valve.

Figure 3.23 Layout of the coolant flow paths in the current test vehicle
The operation of the pump determines coolant flow rate through the pump. The pressure rise across the pump depends on the pump speed and the coolant flow rate but it also depends on the pressure drop across the various heat exchangers and valves in the system through which the coolant flows. The point of intersection of the characteristic pump curves and the system curves is used to determine the pump operating points which are used to match the flow rate of coolant required through the pump. Therefore, an iterative process is required that traverses back and forth between the pump curves and the system requirements. Due to the complex coupling in the non-linear algebraic equations used to describe coolant flow rates in complex flow circuits, solution of these equations in a non-linear ODE solving simulation environment, like Simulink, is not possible. Also, using embedded code in Simulink models (S-functions or embedded Matlab scripts) to solve such a set of equations iteratively is computationally intensive and, therefore, unfeasible for the category of models being developed through this work.

This problem is tackled by introducing the electrical-fluid flow analogy. Under the subtext of the electrical-fluid flow analogy, the heat exchangers or any flow restrictions are represented as resistors, the coolant pump as a voltage source and the flow of coolant (flow rates) through each loop are treated as flow of current. The equivalent circuit of coolant flow, as described by the electrical-fluid flow analogy, is shown in Figure 3.24. For each loop, an equivalent resistance is defined as the sum of the resistance due to the heat exchanger and the resistance due to the flow control devices, if present. The resistance to flow of fluid across the $i^{th}$ branch in the circuit is the ratio of pressure drop to the flow rate through that branch.
Based on circuit analysis methods (Kirchhoff’s voltage and current laws), a system of non-linear algebraic equations are formulated describing coolant flow through each leg of the “flow circuit” for the coolant. The equations are written in matrix form as shown by Equation 3.37. The subscript on a resistance describes the branch for which the equivalent resistance is calculated.

\[ R_i = \frac{\Delta p_i}{q_i}, i = 1 \text{ to } 6 \]

Figure 3.24 Coolant Flow Circuit: based on the electrical-fluid flow analogy
The pressure drop across a heat exchanger is calculated, as a function of flow rate and fluid properties, based on the Hagen-Poiseuille’s pressure drop model explained in [3.2].

The method implies using an equation of the form,

\[ \Delta p_i = C_1 \mu q_i + C_2 \rho q_i^2 \]  

Equation 3.38

The first term on the right hand side of Equation 3.38 accounts for the pressure drop across a component for laminar flow conditions and the second term accounts for pressure drop in turbulent flow regimes. In Equation 3.38, the viscosity term \([\mu = \mu(T)]\) and density term \([\rho = \rho(T)]\) introduce dependence of temperature on pressure drop and thereby on coolant flow rates. The resistances are therefore calculated based on the changing pressure drop and flow rates across the branches through which the coolant flows. Resistance equations for the heat exchangers in the coolant circuit are formulated as shown in from Equation 3.39 through Equation 3.43. These equations also contain the terms which take into account the resistance provided by the valves in addition to the resistance provided by the heat exchanger.

**Exhaust Gas Cooler**

\[ R_1 = \frac{\Delta p_1}{q_1} = C_{1EGR} \mu + C_{2EGR} \rho q_1 + \frac{K_{EGRvalve} \rho q_1}{2A_{EGRvalve}^2} \]  

Equation 3.39

**Radiator**

\[ R_2 = \frac{\Delta p_2}{q_2} = L[C_{1rad} \mu + C_{2rad} \rho q_2] + \frac{\rho q_2}{2(A_{tstat} C_{dtstat})^2} \]  

Equation 3.40
Engine Oil Heater/Cooler (EOC)

\[ R_3 = \frac{\Delta p_3}{q_3} = C_{1EOC} \mu + C_{2EOC} \rho q_3 \]  

Equation 3.41

Transmission Oil Cooler/Heater (TOH)

\[ R_4 = \frac{\Delta p_4}{q_4} = C_{1TOH} \mu + C_{2TOH} \rho q_4 + \frac{K_{3WV_{TOH}} \rho q_4}{2A_{3WV}^2} \]  

Equation 3.42

Cabin Heater Core

\[ R_5 = \frac{\Delta p_5}{q_5} = \frac{\rho q_5}{2(A_{CHC} C_{dCHC})^2} + \frac{K_{3WV_{CHC}} \rho q_5}{2A_{3WV}^2} \]  

Equation 3.43

The values of the coefficients \( C_1 \) and \( C_2 \) calculated for all the heat exchangers in the current architecture (except for the Cabin Heater Core), based on steady state pressure drop data from the heat exchanger manufacturer are listed in Table 3.3

<table>
<thead>
<tr>
<th>Heat Exchanger</th>
<th>Calibration Coefficient</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>EGR Cooler</td>
<td>( C_{1EGR} )</td>
<td>( 2.0813 \times 10^8 )</td>
</tr>
<tr>
<td></td>
<td>( C_{2EGR} )</td>
<td>( 9.0295 \times 10^6 )</td>
</tr>
<tr>
<td></td>
<td>( C_{1rad} )</td>
<td>( 4.0549 \times 10^9 )</td>
</tr>
<tr>
<td></td>
<td>( C_{2rad} )</td>
<td>( 3.5097 \times 10^6 )</td>
</tr>
<tr>
<td>Transmission Oil Heater</td>
<td>( C_{1TOH} )</td>
<td>( 2.2064 \times 10^9 )</td>
</tr>
<tr>
<td></td>
<td>( C_{2TOH} )</td>
<td>( 4.1442 \times 10^7 )</td>
</tr>
<tr>
<td>EOC</td>
<td>( C_{1EOC} )</td>
<td>( 2.2064 \times 10^9 )</td>
</tr>
<tr>
<td></td>
<td>( C_{2EOC} )</td>
<td>( 4.1442 \times 10^7 )</td>
</tr>
</tbody>
</table>

For illustration, the pressure drops across and resistances to flow through the heat exchangers, as a function of flow rate through, are shown in Figure 3.25 to Figure 3.30.
Figure 3.25 Pressure drop across Radiator

Figure 3.26 Resistance to flow - Radiator

Figure 3.27 Pressure drop across EGR Cooler

Figure 3.28 Resistance to flow - EGR Cooler

Figure 3.29 Pressure drop across EOC

Figure 3.30 Resistance to flow - EOC
As mentioned earlier, the three way valve is a complimentary operation valve. Both of the openings of the 3-way valve, through and side, resemble a ball valve. Also, the valve controlling coolant flow to the exhaust gas heater is a ball valve. To this effect, the pressure drop through the ball valve as a function of fraction of valve opening and volumetric flow rate using representative expression from [3.15]. This expression used to model the pressure drop across the through and side branches is given by. Equation 3.44.

\[ \Delta P = \frac{K \rho q^2}{2A^2} \]

Equation 3.44

The flow resistance of ball valves, therefore, also depends on the lumped pressure loss coefficient \((K)\). The lumped pressure loss coefficient is calculated as a function of fraction of valve opening \((\theta)\), as shown in Figure 3.31. The lumped pressure loss coefficient for valves is discussed and explained in detail in [3.14] and [3.15].
For illustration, the pressure drop across the “through” branch of the three way valve as a function of volumetric flow rate, for different valve opening angles, is shown in Figure 3.32. This branch guides flow to the cabin heater in the current architecture.

**Figure 3.32** Three way valve: $\Delta P$ across the through branch

**NOTE:** the x-axis i.e. volumetric flow rate is not the same for all of the subplots in Figure 3.32
3.2.4.2 Slow Response Heat Exchangers

Among the heat exchangers part of the engine thermal management system, the radiator and cabin heater core are liquid-to-air heat exchangers. Unlike the engine oil cooler or the transmission oil heater, which are compact liquid-to-liquid heat exchangers, the thermal inertia of the heat exchanger metal is important in the case of the radiator and cabin heater and it cannot be neglected. The reason for this is that the thermal inertia of the heat exchanger is much larger compared to that of at least one of the working fluids, air in this case, thereby significantly affecting the heat transfer phenomenon. For these reasons, the radiator and the cabin heater) are termed as “slow response” heat exchangers. The working fluids for the radiator and the cabin heater model are the coolant and ambient air.

A lumped parameter modeling technique [3.10] is used to model the slow response heat exchangers by dividing their length into a finite number of lumps, along the direction of coolant flow. Each lump is then divided into three nodes, namely the liquid, the walls of the heat exchanger, and the air side surface [3.11]. First law of thermodynamics is applied to each node. The air side surface is a virtual surface taking into account the effect of fins and their efficiency. Figure 3.33 illustrates the rationale behind the lumped parameters modeling approach described here.
Figure 3.33 Slow Response Heat Exchanger Modeling Principle

The nodal equations representing the \( j^{th} \) node of the liquid, wall and air side surface are given by Equation 3.45, Equation 3.46 and Equation 3.47 respectively.

\[
C_{tiq} \frac{dT_{iq}}{dt} = \frac{n}{A_{face}} \dot{m}_{t iq} c_{piq} \left[ T_{iq_{j-1}} - T_{iq_j} \right] - \frac{C_r \dot{m}_o}{C_3} \left[ T_{iq_j} - T_{w_j} \right]
\]

Equation 3.45
\[ C_w \frac{dT_{wj}}{dt} = \frac{C_f m_0^w}{C_3} \left[ T_{wj} - T_{liq} \right] - \frac{1}{C_2} \left[ T_{wj} - T_{sj} \right] \]  
\text{Equation 3.46}

\[ C_s \frac{dT_{sj}}{dt} = \frac{1}{C_2} \left[ T_{wj} - T_{sj} \right] - \rho_{air} c_{p_{air}} V_{face} P_a \left[ T_{sj} - T_{air_{in}} \right] \]  
\text{Equation 3.47}

In Equation 3.45 to Equation 3.47, \( C_1 \), \( C_2 \), and \( C_3 \) are calibrations parameters, whose calculation is explained later in this section. In Equation 3.47, \( P_a \) represents the effectiveness of the liquid-to-air heat exchanger and is given as follows,

\[ P_a = 1 - \exp \left( - \frac{1}{C_1} \right) \]  
\text{Equation 3.48}

The density of air is given as a function of ambient pressure and temperature,

\[ \rho_{air} = \frac{P_{amb}}{R_{air} T_{amb}} \]  
\text{Equation 3.49}

Where, \( P_{amb} \) and \( T_{amb} \) are atmospheric pressure and atmospheric temperature, respectively. Supporting equations that describe parameters used in Equation 3.45 to Equation 3.48 are,

Specific heat capacity of air [3.4]:

\[ c_{p_{air}} = \alpha + \beta_{coeff} T + \gamma T^2 + \delta T^3 + \epsilon T^4 \]

The coefficient values used in the above equation are organized in Table 3.4.
Table 3.4 Coefficients in equation for calculation of $c_{p_{air}}$

<table>
<thead>
<tr>
<th>Coefficients in equation for $c_{p_{air}}$</th>
<th>Value of coefficient</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\alpha$</td>
<td>3.653</td>
</tr>
<tr>
<td>$\beta_{coeff}$</td>
<td>$-1.337 \times 10^{-3}$</td>
</tr>
<tr>
<td>$\gamma$</td>
<td>$3.294 \times 10^{-6}$</td>
</tr>
<tr>
<td>$\delta$</td>
<td>$-1.913 \times 10^{-9}$</td>
</tr>
<tr>
<td>$\varepsilon$</td>
<td>$0.2763 \times 10^{-12}$</td>
</tr>
</tbody>
</table>

Thermal inertia of the liquid:

$$C_{l_{liq}} = \frac{c_{p_{liq}} \rho_{liq} V_{liq}}{A_{face}}$$

Fluid properties group $C_f$:

$$C_{f_{liq}} = \frac{K_{liq}}{\mu_{liq}^{\beta}} \left[ \frac{\mu_{liq} c_{p_{liq}}}{K_{liq}} \right]^{\frac{1}{3}}$$

Mass flow rate of coolant per unit length:

$$\dot{m}_o = \dot{m}_{liq}/L$$

The calibration procedure used to calculate the calibration parameters of slow response heat exchangers is explained here. The calculation of the parameters $C_1, C_2$ and $C_3$ in the model for the slow response heat exchangers is carried out using steady state data points, however, the model itself is capable of mimicking transient behavior of the liquid-to-air heat transfer characteristics. The heat exchanger performance data is obtained either directly from manufacturer or recorded during in-house testing. The method used to calibrate the three terms above is inferred from [3.11]. The difference between the heat exchanger models characterizing the radiator and the cabin heater is that, due to larger
geometric dimensions of the radiator, the model for the radiator considers 3 lumps whereas the cabin heater is divided into only 2 nodes, due to a smaller width. Manufacturer data available for the slow response heat exchangers includes the temperature and volumetric flow rate of coolant into the heat exchanger, face velocity and temperature of air impinging on the front face of the radiator, the temperature of coolant leaving the heat exchanger and the temperature of air after passing through the radiator fins (measured at multiple locations). Using the P-Ntu method [3.12], the effectiveness of the heat exchanger is calculated for given test conditions at steady state. The effectiveness per pass \( P \) and total effectiveness \( P_b \) are calculated using Equation 3.50 and Equation 3.51, respectively.

\[
P = \frac{T_{\text{air, out}} - T_{\text{air, in}}}{T_{\text{liq, in}} - T_{\text{air, in}}} \quad \text{Equation 3.50}
\]

\[
P_b = 1 - (1 - P)^n \quad \text{Equation 3.51}
\]

Where, \( n \) is the number of passes of the heat exchanger.

The “R” value is then calculated using Equation 3.52. This value is analogous to minimum capacity ratio. In the case of liquid-to-air heat exchangers, the total heat capacity of air is far less than that of the liquid. Therefore,

\[
R = \frac{m_{\text{air}} c_{p, \text{air}}}{m_{\text{liq}} c_{p, \text{liq}}} \quad \text{Equation 3.52}
\]

And

\[
R_b = n \times R
\]

As per the P-NTU method, the number of heat transfer units (NTU) per pass, is then calculated by using Equation 3.53.
NTU_b = -\ln \left( 1 + \frac{\ln(1 - P_b R_b)}{R_b} \right) \quad \text{Equation 3.53}

From the definition of NTU, it is inferred that NTU is the ratio of the overall heat transfer coefficient to the minimum total capacity. Therefore, when this relation is inverted, the overall heat transfer coefficient can be expressed as the product of the NTU per pass and minimum total capacity. This equation enables calculation of the experimental overall heat transfer coefficient for the operating conditions at which the test data is available.

\[ \{UA\} = NTU_b \times C_{\text{min}} \quad \text{Equation 3.54} \]

The functional form of the overall heat transfer coefficient may be represented as being dependent on the sum of three heat transfer coefficients. The three heat transfer coefficients take into account the heat transfer coefficient between the liquid and the metal walls of the heat exchanger by forced convection, the heat transfer between the walls of the heat exchanger by conduction, and the heat transfer between the walls of the heat exchanger and the air. This form of expression for the overall heat transfer coefficient when coupled with Equation 3.54, gives rise to Equation 3.55.

\[
\rho_{\text{air}} c_{\text{p,air}} V_{\text{face}} NTU_b \frac{A_{\text{face}}}{\{UA\}} = \frac{C_1}{\left[\rho_{\text{air}} V_{\text{face}}\right]^\frac{1}{2}} + C_2 + \frac{C_3}{C_{\text{f,liq}} \dot{m}_{\text{liq}}} \quad \text{Equation 3.55}
\]

Utilizing the available heat transfer data from the manufacturer, the coefficients $C_1, C_2$ and $C_3$ are calculated. This calculation is carried out using a least squares curve fitting method for which a routine exists in MATLAB. Only part of the steady state manufacturer data is used for calibration purposes, the remaining part of the data is used
for validation of the formulated model. The values obtained for the calibration parameters for the radiator model, as an example, are shown in Table 3.5.

Table 3.5 Values of calibration parameters for the radiator model

<table>
<thead>
<tr>
<th>Calibration parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_1$</td>
<td>$1.768 \times 10^{-4}$</td>
</tr>
<tr>
<td>$C_2$</td>
<td>$1.732 \times 10^{-5}$</td>
</tr>
<tr>
<td>$C_3$</td>
<td>$3.780 \times 10^{-2}$</td>
</tr>
</tbody>
</table>

Using the calculated values for $C_1$, $C_2$ and $C_3$, the coolant temperatures out of the radiator are calculated for the different test conditions in the manufacturer data.

Figure 3.34 shows a comparison between the radiator-out coolant temperatures from experiments and those obtained from the model. There is good agreement between the temperatures predicted using this model and the experimental values, with an average percentage error in coolant temperature prediction being less than 0.1% and standard deviation in percentage error equal to 0.03%. Similar comparison is seen in Figure 3.35 for the air temperatures after air has passed through the radiator. The mean percentage error and standard deviation in percentage error for air temperatures between model and tests are 0.21% and 0.12% respectively.
Figure 3.34 Radiator model calibration: Coolant temperature comparison

Figure 3.35 Radiator model calibration: Air temperature comparison
3.2.4.3  *Fast Response Heat Exchangers*

The heat exchanger categorized as “fast response heat exchangers” that is part of the existing engine thermal management system, is the engine oil cooler. The engine oil cooler is termed as a fast response heat exchanger because the response of this compact liquid-to-liquid heat exchanger to changes in operating conditions is relatively quick (less than 0.5 second) compared to the time constant associated with other parts of the thermal management system. This means that the time required for the working fluids (coolant and engine oil) to reach equilibrium temperature, following heat transfer between the two via the metal parts of the engine oil cooler, is very small. This is because the size and therefore the thermal inertia of the metal walls of the engine oil cooler is small compared to that of the thermal inertia of the coolant or the engine oil, and hence can be neglected. Therefore, in the case of fast response heat exchangers, only two lumps are considered namely the coolant and the oil. The Transmission Oil Heater (TOH) is also categorized as a fast response heat exchanger, and follows the same modeling principle as the engine oil cooler. In the TOH, however, the working fluids are the transmission fluid and coolant.

The EOC is modeled employing the effectiveness-NTU method [3.10]. This method, traditionally, offers a static approach towards modeling heat exchangers. From knowledge of the overall heat transfer coefficient (UA), the Number of Heat Transfer Units - NTU (Equation 3.56) is calculated. The NTU and the minimum capacity ratio (CR) are then used to calculate the effectiveness ($\varepsilon$) of the heat exchanger at the same operating point (Equation 3.57).

$$NTU = \frac{UA}{C_{\text{min}}}$$  

*Equation 3.56*
In Equation 3.58, \( CR = \frac{m_{clnt}C_{clnt}}{m_{oil}C_{oil}} \) or \( \frac{m_{oil}C_{oil}}{m_{clnt}C_{clnt}} \), whichever ratio is smaller.

An operating point of the heat exchanger refers to a particular value for flow rate of the oil, flow rate of the coolant, inlet temperature of the coolant, and an inlet temperature of the oil. The \( \varepsilon \)-NTU method only requires these parameters as inputs, based on which the model predicts the temperatures of the coolant and the oil at the outlet of the heat exchanger, once the effectiveness is characterized.

The thermal inertias of neither of the fluids are taken into account when this approach is employed, traditionally. Following the \( \varepsilon \)-NTU method as is, the temperatures of the coolant and oil at the outlet of the heat exchanger are given by Equation 3.59 and Equation 3.60, respectively [3.10].

\[
T_{clnt_{out}} = T_{clnt_{in}} + \frac{\varepsilon * C_{min}}{C_{clnt}} (T_{oil_{in}} - T_{clnt_{in}}) \quad \text{Equation 3.59}
\]

\[
T_{oil_{out}} = T_{oil_{in}} - \frac{\varepsilon * C_{min}}{C_{oil}} (T_{oil_{in}} - T_{clnt_{in}}) \quad \text{Equation 3.60}
\]

It is seen that the equations traditionally used to calculate the temperature do not include any dynamics. The static equations of state for coolant and oil temperatures are modified here to include first order behavior in the way the changes in temperatures are captured.
These modifications are reflected in Equation 3.61 and Equation 3.62, for temperature of coolant and oil leaving the heat exchanger, respectively.

\[
\begin{align*}
\dot{m}_{\text{clnt}} c_{\text{clnt}} \frac{dT_{\text{clnt\_out}}}{dt} &= \dot{m}_{\text{clnt}} c_{\text{clnt}} \left[ (T_{\text{clnt\_in}} - T_{\text{clnt\_out}}) + \varepsilon \right] \\
&= \dot{m}_{\text{clnt}} c_{\text{clnt}} \left[ (T_{\text{clnt\_in}} - T_{\text{clnt\_out}}) + CR \right] \\
\dot{m}_{\text{oil}} c_{\text{oil}} \frac{dT_{\text{oil\_out}}}{dt} &= \dot{m}_{\text{oil}} c_{\text{oil}} \left[ (T_{\text{oil\_in}} - T_{\text{oil\_out}}) - \varepsilon \right] \\
&= \dot{m}_{\text{oil}} c_{\text{oil}} \left[ (T_{\text{oil\_in}} - T_{\text{oil\_out}}) - CR \right]
\end{align*}
\]

Equation 3.61

Equation 3.62

These dynamics are introduced to account for the fact that the heat transfer between the coolant and oil is not instantaneous.

The process followed for calibration for the fast response heat exchanger model is similar to that for the slow response heat exchangers. Calibrated is required for the three coefficients \( C_1, C_2 \) and \( C_3 \), used to calculate the overall heat transfer coefficient. In this process, the effectiveness of the heat exchanger is first calculated from manufacturer data for different operating conditions. For the engine oil cooler, the “heat transferred” values were provided and therefore the effectiveness of the EOC is calculated using Equation 3.63.

\[
\varepsilon = \frac{\dot{Q}}{C_{\text{min}} \Delta T_{\text{in}}}
\]

Equation 3.63

If the temperatures of the coolant and oil out of the heat exchangers are provided in the manufacturer data from experimental testing rather than the “heat transferred”, then Equation 3.64, the definition of effectiveness, is used to calculate the effectiveness.
Equation 3.64

\[ \varepsilon = \left| \frac{\Delta T_{\text{out}}}{\Delta T_{\text{in}}} \right| \]

Capacity ratio is calculated using Equation 3.58 and then these effectiveness and capacity ratio values are used to calculate the NTU for the test data. This calculation is carried out by inverting Equation 3.57 as follows,

\[ NTU = \frac{1}{1 - CR} \ln \left( \frac{1 - \varepsilon \cdot CR}{1 - \varepsilon} \right) \]  

Equation 3.65

The definition of NTU (ratio of overall heat transfer coefficient to minimum total capacity) is then used to calculate the overall heat transfer coefficient from experiments as shown by Equation 3.66,

\[ NTU = \frac{\{UA\}}{C_{\text{min}}} \Rightarrow \{UA\} = NTU \times C_{\text{min}} \]

Equation 3.66

A semi-physical model for the overall heat transfer coefficient is formulated as proposed by the work described in [3.13]. The overall heat transfer can be expressed by Equation 3.67. This equation gives the overall heat transfer term as an amalgamation of three terms. The three terms account for the heat transfer between the coolant and the heat exchanger walls, the conduction within the heat exchanger walls and the heat transfer between the heat exchangers walls and the oil.

\[ \{UA\} = \frac{1}{Ah_{\text{oil}}} + \frac{1}{K} + \frac{1}{Ah_{\text{cnt}}} \]

Equation 3.67
The proposed model, entails calculation of the forced convective heat transfer coefficient \((h)\) using the definition of Nusselt number and empirical expression for Nusselt number \([3.10]\) as follows,

\[
Nu = \frac{hD_h}{K} = C \cdot Re^\beta \cdot Pr^{\frac{1}{3}} \tag{Equation 3.68}
\]

\[
\Rightarrow h_{oil} = \frac{C_1}{D_h} \left[ KRe^\beta \cdot Pr^{\frac{1}{3}} \right]_{oil} \quad \text{and} \quad h_{clnt} = \frac{C_3}{D_h} \left[ KRe^\beta \cdot Pr^{\frac{1}{3}} \right]_{coolant}
\]

The equation for overall heat transfer coefficient can, therefore, be expressed in terms of the parameters that need to be calibrated as,

\[
\{UA\} = \frac{C_1}{f_{clnt} \cdot m^\beta_{clnt}} + C_2 + \frac{C_3}{f_{oil} \cdot m^\beta_{oil}} \tag{Equation 3.69}
\]

The terms \(f_{clnt}\) and \(f_{oil}\) for coolant and for oil, respectively, are fluid properties groups.

\[
f = K \cdot \frac{1}{\mu^\beta} \tag{Equation 3.70}
\]

Among the calibration terms in the formula for the overall heat transfer coefficient, \(C_2\) is a constant added due to the presence of conduction within the plates of the heat exchanger. This coefficient is usually neglected because its contribution is very small compared to that of convective heat transfer coefficients, as seen in Table 3.6. In this model, however, the term is retained, for some added accuracy. \(C_2\) is given as the ratio between the plate thickness \((d_p)\) and thermal conductivity of the heat exchanger plate material \((K_p)\).

\[
C_2 = \frac{d_p}{K_p} \tag{Equation 3.71}
\]
The other two coefficients, \( C_1 \) and \( C_3 \) are calculated using a linear regression routine. The linear regression needs X and Y variables to be formulated. This is done in the following manner.

Consider the set of equations from Equation 3.72 to Equation 3.77. These are intermediate variables that are calculated as part of the process to obtain the values of the calibration terms \( C_1 \) and \( C_3 \).

\[
C_{f_{clnt}} = \frac{K_{clnt} \left( \frac{C_{p_{clnt}} \mu_{clnt}}{K_{clnt}} \right)^{\frac{1}{3}}}{\mu_{clnt}^{\beta_{clnt}}} \quad \text{Equation 3.72}
\]

\[
C_{foil} = \frac{K_{foil} \left( \frac{C_{p_{foil}} \mu_{foil}}{K_{foil}} \right)^{\frac{1}{3}}}{\mu_{foil}^{\beta_{foil}}} \quad \text{Equation 3.73}
\]

\[
G_{clnt} = \frac{\rho_{clnt} q_{clnt}}{n_{clnt} A_{clnt}} \quad \text{Equation 3.74}
\]

\[
G_{oil} = \frac{\rho_{oil} q_{oil}}{n_{oil} A_{oil}} \quad \text{Equation 3.75}
\]

\[
x_{clnt} = \frac{D_{clnt}^{1-\beta_{clnt}}}{C_{clnt}^{\beta_{clnt}} C_{f_{clnt}}} \quad \text{Equation 3.76}
\]

\[
x_{oil} = \frac{D_{oil}^{1-\beta_{oil}}}{C_{oil}^{\beta_{oil}} C_{foil}} \quad \text{Equation 3.77}
\]

In the set of equations from Equation 3.72 to Equation 3.77, \( n_{clnt} \) and \( n_{oil} \) are the number of plates the coolant and oil flow through, \( A_{clnt} \) and \( A_{oil} \) are the coolant and oil cross-sectional flow area per passage, \( q_{clnt} \) and \( q_{oil} \) are the volumetric flow rates of coolant and oil, \( D_{clnt} \) and \( D_{oil} \) are the hydraulic diameters the coolant and oil flows encounter, respectively.
Let the y-interpolant be represented as $Y_{\text{interp}}$ and x-interpolant as $X_{\text{interp}}$. These are the two variables that are used for linear regression, and are given as follows,

$$Y_{\text{interp}} = \left(\frac{A_{\text{total}}}{UA} - C_2\right)$$  \hspace{1cm} \text{Equation 3.78}

$$X_{\text{interp}} = \frac{x_{\text{cnt}}}{x_{\text{oil}}}$$  \hspace{1cm} \text{Equation 3.79}

The equation for linear interpolation is of the form:

$$Y_{\text{interp}} = C_3 \times X_{\text{interp}} + C_1$$  \hspace{1cm} \text{Equation 3.80}

The slope of the line defined by Equation 3.80 is the value of $C_3$ and the intercept of this line on the y-axis is the value of $C_1$. The values for these calibration parameters for the engine oil cooler are tabulated in Table 3.6.

<table>
<thead>
<tr>
<th>Calibration parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_1$</td>
<td>26.136</td>
</tr>
<tr>
<td>$C_2$</td>
<td>$3.077 \times 10^{-6}$</td>
</tr>
<tr>
<td>$C_3$</td>
<td>1.236</td>
</tr>
</tbody>
</table>

Figure 3.36 shows a comparison between the EOC-out coolant temperatures obtained from the model versus those from experiments. There is good agreement between the two temperatures, with average percentage error in coolant temperature prediction being less than -0.03% and standard deviation in percentage error equal to 0.05%. A similar comparison is seen in Figure 3.37 for the air temperatures after air has passed through the
radiator. The mean percentage error and standard deviation in percentage error for engine oil temperatures between model and tests are 0.11% and 0.17% respectively.

![Figure 3.36 EOC model calibration: Coolant temperature comparison](image-url)
For the purpose of selecting data points for calibration, from the available set of steady state data points from the manufacturer for the EOC, a sensitivity study is conducted. This study is aimed at observing the effect of various combinations of experimental data points, used for model calibration, on the values of the calibration parameters and on the corresponding error metrics for fluid temperature prediction.

As shown in Table 3.7, the values of the calibration parameters $C_1$, $C_2$ and $C_3$ do not vary significantly when different combinations of steady state experimental data points are used for calibration. More importantly, the final output of the model i.e. the EOC-out coolant and engine oil temperature are predicted with the average error being at most 0.05% for coolant temperature prediction and at most 0.2% for engine oil temperature prediction.
Table 3.7 Sensitivity of calibration parameters on calibration data set

<table>
<thead>
<tr>
<th>Data used for calculating values of calibration parameters</th>
<th>Values of Calibration Parameters</th>
<th>Mean error in temperature prediction (%)</th>
<th>Standard deviation in temperature prediction (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>C1, C2 (constant), C3</td>
<td>Coolant, Oil</td>
<td>Coolant, Oil</td>
<td>Coolant, Oil</td>
</tr>
<tr>
<td>All Data Points</td>
<td>26.145 3.077E-6 1.234</td>
<td>-0.05 0.19</td>
<td>0.13 0.47</td>
</tr>
<tr>
<td>First 15 points (60% data)</td>
<td>25.040 3.077E-6 1.272</td>
<td>0.02 0.17</td>
<td>0.14 0.41</td>
</tr>
<tr>
<td>Odd data points (52 % data)</td>
<td>26.136 3.077E-6 1.236</td>
<td>-0.03 0.11</td>
<td>0.05 0.17</td>
</tr>
</tbody>
</table>

### 3.2.4.4 Coolant Mixing

After exiting each of the heat exchangers, the coolant mixes before all of it is pumped around the engine block and head. To model coolant from different branches mixing together before entering the pump, sum of the enthalpies divided by the total capacity of the coolant gives the absolute temperature of the coolant entering the engine, as shown by Equation 3.81.

\[
T_{clnt,engin} = \frac{\Sigma \rho_{clnt,HEX} Q_{clnt,HEX} c_{p,clnt,HEX} T_{clnt,i}}{\Sigma \rho_{clnt,HEX} Q_{clnt,HEX} c_{p,clnt,HEX}} \quad \text{Equation 3.81}
\]

HEX: Heat EXchanger

### 3.2.5 Exhaust Gas Cooler Thermal Model

The exhaust gas cooler is a heat exchanger where the working fluids are the exhaust gas and engine coolant. As the thermal inertia one of the working fluids (exhaust gas) is very low compared to the metal walls of the heat exchanger, the thermal inertia of the metal walls of the exhaust gas cooler cannot be neglected. The exhaust gas cooler is used as a waste heat recovery device. Therefore, if the thermal capacity of the heat exchanger walls
is neglected, the predictions of recovered heat energy would be inaccurate. The lumped capacitance modeling approach followed to characterize the behavior of the exhaust gas cooler. The exhaust gas cooler is discretized into three lumps namely the coolant, the exhaust gas cooler walls and the exhaust gases flowing through the cooler.

![Diagram of EGR cooler](image)

**Figure 3.38 control volume: EGR cooler**

The control volumes along with the heat sinks and sources associated with the exhaust gas cooler, following the lumped parameter model, are shown in Figure 3.38. The energy balance equations for the control volumes representing coolant, wall and exhaust gas are formulated as described by Equation 3.82, Equation 3.83 and Equation 3.84, respectively.

The model mentioned here is proposed in [3.23] and is explained in detail in [3.24].

\[
\frac{dT_{c_{int,EGRC, out}}}{dt} = \frac{1}{m_{c_{int}} c_{p_{c_{int}}} + m_{shell} c_{p_{shell}}} \left[ T_{wall,EGRC, out} - T_{c_{int,EGRC, out}} \left( \frac{1}{R_{conv_{c_{int}}} + \frac{1}{R_{cond_{shell}}}} \right) + m_{c_{int,EGRC}} c_{p_{c_{int}}} \left( T_{c_{int,EGRC, in}} - T_{c_{int,EGRC, out}} \right) \right]
\]

Equation 3.82
The method of calculation of exhaust gas properties in the current work, however, is different; it is based on routines proposed in [3.6] and described earlier in section 3.2.2.1.

**NOTE:** The subscript used here to refer to the exhaust gas cooler is “EGRC”, but in reality the component is used for the purpose of heating the coolant and not to cool the exhaust gas. There is no re-circulation of exhaust gases to the intake manifold in the current system.

The model characterizing the exhaust gas cooler is required to have one more predictive capability. This predictive capability deals with estimation of conditions when the exhaust gas cooler is being used in a manner that may be detrimental to the hardware. A quantity called the energy ratio (or the E-ratio) is used for this purpose. The E-ratio is a ratio of the theoretical maximum amount of energy that can be recovered to the maximum energy the coolant can absorb without boiling. The theoretical maximum energy that may be recovered using the exhaust gas cooler is directly proportional to the difference in inlet temperatures of exhaust gas and the coolant, and the maximum energy that the coolant can absorb before boiling is directly proportional to the difference in...

\[
\frac{dT_{\text{wall,EGRC}}}{dt} = \frac{1}{m_{\text{EGRC,wall}} c_{\text{v,EGRC,wall}}} \left[ \frac{T_{\text{clnt,EGRC,out}} - T_{\text{wall,EGRC}}}{R_{\text{conv,water}} + R_{\text{cond,wall}}} + \frac{T_{\text{EGRC,in,exh}} - T_{\text{wall,EGRC}}}{R_{\text{conv,exh}} + R_{\text{cond,wall}}} \right]
\]

\[
\frac{dT_{\text{exh,EGRC,out}}}{dt} = \frac{1}{m_{\text{exh,EGRC}} c_{\text{v,exh}}} \left[ \frac{T_{\text{wall,EGRC}} - T_{\text{exh,EGRC,out}}}{R_{\text{conv,exh}} + R_{\text{cond,wall}}} + \dot{m}_{\text{exh,EGRC}} c_{\text{p,exh}} (T_{\text{exh,EGRC,in}} - T_{\text{exh,EGRC,out}}) \right]
\]
between the inlet coolant temperature and the coolant boiling point temperature. This expression for E-ratio is given by Equation 3.85. For the purpose of maintaining the health of the exhaust gas cooler and to operate it within durability limits, the energy ratio is capped at approximately 0.4. This conservative limit, as observed from experience, provides a general rule of thumb for the operation of the exhaust gas cooler.

\[
E_{ratio} = \frac{\dot{m}_{exh,EGRC} c_{pexh} (T_{exh,vol,1} - T_{clnt,EGRC,in})}{\dot{m}_{clnt,EGRC} c_{pclnt} (T_{clnt,boiling} - T_{clnt,EGRC,in})}
\]

Equation 3.85

The exhaust gas cooler model is validated using the performance data from the manufacturer. This set of consists of steady state data from the manufacturer where the flow rate of coolant, flow rate of exhaust, inlet temperature of exhaust gases as well as the coolant are varied. These test inputs were varied following a simple design of experiments method. The calibrated exhaust gas cooler model is used to simulate similar conditions as maintained in the experimental tests. A comparison between coolant temperatures at the outlet of the exhaust gas cooler, obtained from experiments versus those predicted by the model, is shown in Figure 3.39. It is seen that the model predicted coolant temperatures are erroneous for high coolant temperatures (a difference of about 3°C). However, coolant temperatures above 115-120°C will not be reached in normal operation in a vehicle.
Figure 3.39 EGRC model: coolant temperature comparison

Figure 3.40 shows a histogram with the percentage error between model-predicted coolant temperatures and experimentally measured coolant temperatures at the outlet of the exhaust gas cooler. The average percentage error in coolant temperature prediction is less than 0.4% and standard deviation in percentage error is 0.7%.
Figure 3.40 EGRC model: error in coolant temperature prediction

A comparison between model-predicted energy ratios versus energy ratio calculated from experimental data is shown in Figure 3.41. This ratio is calculated the same way from experimental and model output parameters. The comparison was carried out to see if the expression used to calculate coolant boiling temperature in this simulation, not available from the manufacturer, is in line with the methodology followed by the component manufacturer. Similar to the coolant temperature comparisons, the energy ratio predictions also start to deviate for energy ratios of greater than 0.6. This condition, however, will never be reached in normal operation of the exhaust gas cooler because the energy ratio limit for the heat exchanger is ~0.4. The manufacturer tests show such high
values because these tests are conducted for durability testing of the exhaust gas cooler, and are mostly destructive in nature.

Figure 3.41 EGRC model: Energy ratio comparison

The mean percentage error and standard deviation in percentage error for e-ratio calculations, compared to those calculated by the manufacturer are -0.2% and 1.7% respectively.
Figure 3.42 EGRC model: error in energy ratio prediction
3.2.6. Exhaust Flow Model

The exhaust system of the V6, 3.6L engine is shown in Figure 3.43. The exhaust gas flows out of the two catalytic converters and are combined into one stream passing through a bellow.

![Exhaust System Diagram](https://example.com/exhaust_system_diagram.png)

Figure 3.43 Exhaust System (Courtesy: Chrysler LLC)

From here, an exhaust gas cooler is housed with a bypass valve in parallel. The exhaust gas cooler is used to recuperate some of the exhaust heat energy, especially during the warm-up phase of the engine. The bypass valve is included in this architecture because once the coolant temperature reaches the desired operating temperature following a cold start, the exhaust gases are routed directly downstream of the exhaust gas cooler. Therefore, to be able to predict the amount of heat recuperated by use of the exhaust gas cooler, an accurate characterization of the mass flow rate of exhaust gases is required. To model the exhaust flow through the piping downstream of the exhaust manifold including
the flow through the exhaust gas cooler and the bypass valve, the system is decomposed into a series of flow restrictions and volumes, as shown in Figure 3.44.

![Diagram of exhaust system decomposition](image)

**Figure 3.44 Decomposition of exhaust system decomposition for modeling purposes**

The modeling approach considers alternating volumes and flow restrictions to maintain causality relations. The exhaust manifold is modeled by using the first law of thermodynamics applied to a control volume (Exhaust Manifold), as explained in 3.2.1.4. The flow through the catalytic converter is modeled considering a pressure drop across the inlet and outlet of the catalytic converter and then calculating the mass flow rate of exhaust gases based on that pressure drop. The volume occupied by the pipes 1 through pipe 14, except for pipe 13, is modeled considering a control volume (Volume 1), and so
on. Following sections give brief descriptions of how each of the volumes and flow restrictions is modeled, starting from the catalytic converter flow restriction model.

### 3.2.6.1 Catalytic converter flow restriction model

The flow of exhaust gases through the catalytic converter is modeled following the work shown in [3.17]. According to the proposed model, pressure drop through the catalytic converter is expressed a quadratic function of the mass flow rate of the exhaust gas through the catalytic converter:

\[
\Delta P_{\text{CAT}} = \frac{K_{1,\text{CAT}} \rho_{\text{exh}}}{\rho_{\text{exh}}} m_{\text{exh,CAT}} + \frac{K_{2,\text{CAT}}}{\rho_{\text{exh}}} m_{\text{exh,CAT}}^2
\]  

Equation 3.86

Equation 3.86 is inverted to obtain the equation for the mass flow rate of the exhaust gases as a function of the upstream and downstream pressures, which, in this case, are the pressures in the exhaust manifold and volume 1, respectively. The equation used to describe the mass flow rate of exhaust gases is given as follows:

\[
m_{\text{exh,CAT}} = \frac{-K_{1,\text{CAT}} \rho_{\text{exh}} + \sqrt{(K_{1,\text{CAT}} \rho_{\text{exh}})^2 + 4 \left(\frac{K_{2,\text{CAT}}}{\rho_{\text{exh}}} \Delta P_{\text{CAT}}\right)}}{2 \cdot \frac{K_{2,\text{CAT}}}{\rho_{\text{exh}}}}
\]  

Equation 3.87

The values of constants $K_{1,\text{CAT}}$ and $K_{2,\text{CAT}}$ in Equation 3.87 are shown in Table 3.8 and are obtained directly from [3.17].

<table>
<thead>
<tr>
<th>Calibration parameters</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$K_{1,\text{CAT}}$</td>
<td>28.4</td>
</tr>
<tr>
<td>$K_{2,\text{CAT}}$</td>
<td>0.41</td>
</tr>
</tbody>
</table>
A graph of mass flow rate of exhaust gases through the catalytic converter versus the pressure drop across the catalytic converter is shown in Figure 3.45 for three different temperatures. The mass flow rate of exhaust gas increases with the pressure difference and is higher for a given pressure drop at lower temperatures.

Figure 3.45 Catalytic converter: $\Delta P$ versus mass flow rate for three temperatures

Figure 3.45 illustrates the importance of scheduling the terms representing exhaust gas properties, in the equation for pressure drop across the catalytic converter, on temperature of the exhaust gases. The temperature dependence of the exhaust gas properties namely
density \((\rho_{exh})\) and dynamic viscosity \((\mu_{exh})\) is modeled using Equation 3.88 and Equation 3.89, respectively:

\[
\rho_{exh} \left( \frac{kg}{m^3} \right) = \frac{349 \times P_{vol,1}}{T_{exh,vol,1}} \quad \text{(Equation 3.88)}
\]

\[
\mu_{exh} = \frac{1.48 \times 10^{-6} \times T_{exh,vol,1}^{3/2}}{T_{exh,vol,1} + 173} \quad \text{(Equation 3.89)}
\]

### 3.2.6.2 Volume 1

The conditions of pressure and temperature in Volume 1 were modeled using a dynamic correlation and a static correlation, respectively. The mass conservation and pressure state equations are given by Equation 3.90 and Equation 3.91 respectively. This approach first assumes that volume 1 is isothermal:

\[
\frac{dM_{vol,1}}{dt} = \dot{m}_{exh,\text{CAT}} - (\dot{m}_{exh,\text{EGRC-bypass}} + \dot{m}_{\text{EGRC}}) \quad \text{(Equation 3.90)}
\]

\[
\frac{dp_{vol,1}}{dt} = \left( \frac{R_{exh} T_{exh,vol,1}}{V_{vol,1}} \right) \frac{dM_{vol,1}}{dt} \quad \text{(Equation 3.91)}
\]

The temperature in volume 1, which is also the temperature at the inlet of the EGRC is modeled following methodologies proposed in [3.28]. First, steady-state energy balance is applied to control volume as:

\[
\dot{m}_{exh,\text{CAT}} c_{p_{exh}} (T_{\text{CAT, out}} - T_{exh,vol,1}) = hA(T_{exh,vol,1} - T_{\text{wall}}) \quad \text{(Equation 3.92)}
\]

The overall heat transfer coefficient is calculated based as follows [3.10][3.29]:

\[
h = C(Re)^{0.8}(Pr)^{1/3} \quad \text{(Equation 3.93)}
\]

where, \(C = f(T_{\text{CAT, out}}, \dot{m}_{exh,\text{CAT}})\)
In the above equation, \( C = f(T_{\text{CATout}}, \dot{m}_{\text{exhCAT}}) \)

\[
\begin{align*}
For, T_{\text{CATout}} > 725^\circ K \\
C &= 0.1534 \cdot \dot{m}_{\text{exhCAT}} + 0.0074 \\
For, 650^\circ K < T_{\text{CATout}} \leq 725^\circ K \\
C &= 0.2134 \cdot \dot{m}_{\text{exhCAT}} + 0.0142 \\
For, T_{\text{CATout}} \leq 650^\circ K \\
C &= -0.6689 \cdot \dot{m}_{\text{exhCAT}} + 0.0663
\end{align*}
\]  Equation 3.96

3.2.6.1 **EGR Cooler Flow Restriction**

The mass flow rate of exhaust gases through the EGR cooler is modeled as a sum of the pressure drop across the EGRC as well as the pressure drop in the lengths of pipe before the EGR cooler as:

\[
\Delta p_{\text{EGRC}} = \frac{K_{\text{1EGRC}} \mu_{\text{exh}}}{\rho_{\text{exh}}} \dot{m}_{\text{EGRC}} + \left( \frac{K_{\text{2EGRC}}}{\rho_{\text{exh}}} + \frac{fL_{\text{pipe}}}{2D_{\text{pipe}}A_{\text{pipe}}^2} \right) \dot{m}_{\text{EGRC}}^2
\]  Equation 3.97

Here, \( L_{\text{pipe}} \) represents the combined length of pipes 12 and 13, \( D_{\text{pipe}} \) and \( A_{\text{pipe}} \) represent the internal diameter and the cross-sectional area of pipe 12 and 13 (refer to Figure 3.43), respectively. The linear term in Equation 3.97 dictates the pressure drop across the EGRC for laminar flow conditions. The term \( \frac{K_{\text{2EGRC}}}{\rho_{\text{exh}}} \) in the quadratic part of Equation 3.97, accounts for the pressure drop across the exhaust gas cooler for turbulent flows. The pressure drop through sections of piping before and after the EGR cooler is calculated based on the friction factor associated with these sections of pipe and is

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directly proportional to the length of the pipe sections. The friction factor \( f \) is expressed by the explicit form of the Colebrook-White equation, which approximates the Moody chart [3.2]:

\[
f = \frac{1}{-1.8 \log_{10} \left( \frac{(s/D)^{1.11}}{3.7} + \frac{6.9}{Re} \right)^2}
\]

Equation 3.98

The mass flow rate of exhaust gases through the EGR cooler is calculated by inverting Equation 3.98:

\[
\dot{m}_{EGRC} = \frac{-\frac{K_{1,EGRC}\mu_{exh}}{\rho_{exh}} + \sqrt{\left(\frac{K_{1,EGRC}\mu_{exh}}{\rho_{exh}}\right)^2 + 4\left(\frac{K_{2,EGRC}}{\rho_{exh}} + \frac{fL_{pipe}}{2D_{pipe}A_{pipe}^2}\right)\Delta P_{EGRC}}}{2\left(\frac{K_{2,EGRC}}{\rho_{exh}} + \frac{fL_{pipe}}{2D_{pipe}A_{pipe}^2}\right)}
\]

Equation 3.99

The density \( (\rho_{exh}) \) and dynamic viscosity \( (\mu_{exh}) \) of exhaust gases is a function of the temperature of the exhaust gases as described by Equation 3.88 and Equation 3.89, respectively. In Equation 3.99, \( \Delta P_{EGRC} \) is calculated based on the pressures in volume 1 and volume 2, as demarcated in Figure 3.44.

To calculate the value of the constants \( K_{1,EGRC} \) and \( K_{2,EGRC} \), which are calibration terms in Equation 3.99, pressure drop vs. flow rate data from the manufacturer is used. The friction factor term accounting for the pressure drop through the pipes is dropped temporarily and, therefore, Equation 3.97 reduces to:

\[
\Delta P_{EGRC} = C_1\dot{m}_{EGRC} + C_2\dot{m}_{EGRC}^2
\]

Equation 3.100

Where, \( C_1 = \frac{K_{1,EGRC}\mu_{exh}}{\rho_{exh}} \) and \( C_2 = \frac{K_{2,EGRC}}{\rho_{exh}} \)
A second order least squares fit is obtained for pressure drop as a function of mass flow rate of gases through the exhaust gas cooler, using manufacturer data. Pressure drop across the exhaust gas cooler as a function of mass flow rate of exhaust gases is shown in Figure 3.46; the pressure drop at zero mass flow rates is force fitted through 0 on the y-axis, as can be inferred from Equation 3.100.

![Figure 3.46 Calculation of constants for the EGR cooler flow restriction model](image)

This procedure gives as outputs the values of $C_1$ and $C_2$. These values are then used to calculate $K_{1,EGRC}$ and $K_{2,EGRC}$. The values calculated for these four constants are shown in
Table 3.9; only $K_{1,EGRC}$ and $K_{2,EGRC}$ are actually used in the EGR cooler flow restriction model.

The calculation of $K_{1,EGRC}$ and $K_{2,EGRC}$ is done by inverting the equations for $C_1$ and $C_2$, because data from the manufacturer was available only for one temperature. It is assumed that the effect of temperature on pressure drop is accounted for as a consequence of having exhaust gas properties scheduled on exhaust gas temperature.

Table 3.9 Values of constants in the EGRC flow restriction model

<table>
<thead>
<tr>
<th>Constant Name</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_1$</td>
<td>3507.3</td>
</tr>
<tr>
<td>$C_2$</td>
<td>429876</td>
</tr>
<tr>
<td>$K_{1,EGRC}$</td>
<td>29228508</td>
</tr>
<tr>
<td>$K_{2,EGRC}$</td>
<td>150930</td>
</tr>
</tbody>
</table>

3.2.6.2 **EGR Cooler Bypass Valve**

The flow of exhaust gases through the EGR Cooler bypass valve is modeled following a quasi-static modeling approach i.e. isentropic, compressible flow through a restriction. To fully account for conditions influencing the mass flow rate of exhaust gases through the bypass valve, two effects need to be considered. The first is the fact that, for each opening angle, the “discharge coefficient” of the valve depends on the difference between the upstream and downstream pressures. Secondly, for a given pressure drop across the valve, the bypass valve opening significantly influences the flow through both branches; branch containing the bypass valve and the branch housing the exhaust gas cooler. These two effects are taken into account by introducing two coefficients; one which assumes a
fully open bypass valve and takes into account the varying pressure drops across the bypass valve \([K_d(N, \alpha)]\), and second which accounts for fractional openings of the valve \([C_d(\theta)]\). The equations for mass flow rate of exhaust gases through the bypass valve, for sub-critical and critical flow conditions are:

Sub-critical flow:
\[
\frac{P_{vol2}}{P_{vol1}} > \left( \frac{2}{\gamma + 1} \right)^{\frac{\gamma}{\gamma - 1}}
\]

\[\dot{m}_{exh,EGRC-bypass} = C_d(\theta) \cdot K_d(N, \alpha) \frac{A_{nom}}{\sqrt{RT_{vol1}}} \left( \frac{P_{vol,2}}{P_{vol,1}} \right)^{\frac{1}{\gamma}} \left[ \frac{2\gamma}{\gamma - 1} \left[ 1 - \left( \frac{P_{vol,2}}{P_{vol,1}} \right)^{\frac{\gamma - 1}{\gamma}} \right] \right] \]

Equation 3.101

Critical (choked) flow:
\[
\frac{P_{vol2}}{P} \leq \left( \frac{2}{\gamma + 1} \right)^{\frac{\gamma}{\gamma - 1}}
\]

\[\dot{m}_{exh,EGRC-bypass} = C_d(\theta) \cdot K_d(N, \alpha) \frac{A_{nom}}{\sqrt{RT_{vol1}}} \sqrt{\frac{2\gamma}{\gamma + 1}} \]

Equation 3.102

In the above equations, the specific heat ratio \(\gamma\) is calculated for exhaust gases based on the temperature in volume 1. The discharge coefficient \(C_d(\theta)\) in the above equations is assumed to be of the form of a double sigmoid as shown in Figure 3.47.

![Double sigmoid: \(C_d\) as a function of opening fraction](image)

Figure 3.47 Double sigmoid: \(C_d\) as a function of opening fraction
The loss coefficient, however, is calibrated and represented as a function of engine speed and throttle opening. The calibration process entails reverse engineering the loss coefficient for the bypass valve from the available data from steady state engine tests. Under the context of this method, a MATLAB script is developed which, for imposed values of upstream and downstream pressures, calculates mass flow rate of exhaust gas for all the pressure drops.

![Graph showing the loss coefficient of the exhaust cooler bypass valve as a function of engine speed and throttle opening.](image)

**Figure 3.48** Loss coefficient of the exhaust cooler bypass valve for the fully open case

The scripts then finds the value of the loss coefficient such that the mass flow rate calculated using Equation 3.101 or Equation 3.102 is equal to mass flow rate of exhaust gas at that engine operating condition recorded during the steady state engine tests. This
value of the loss coefficient is selected as the optimum loss coefficient by this algorithm, assuming the valve is completely open. The parameter $K_d$ is shown as a function of engine speed and throttle position in Figure 3.48. The adjusted $R^2$ value for the fit shown above is 0.9676 with an RMS error of 0.0063.

3.2.6.3 Volume 2

Volume 2 is modeled as being isothermal i.e. it is assumed that no heat transfer occurs across the boundaries of the control volume that defines volume 2 [3.4]. The model for volume 2 facilitates the calculation of mass flow rates of exhaust across the bypass valve and the EGRC flow restriction by supplying the pressure downstream i.e. pressure in volume 2. Therefore, the model for volume 2, accounts for the pressure variations but not the temperature variations that exist in reality. This is done in order to reduce the computational burden in the simulation environment, as the temperature state need not be accounted for as it is not necessary. The equation describing the conservation of mass of exhaust gases and the pressure in volume 2:

$$\frac{dM_{vol,2}}{dt} = (\dot{m}_{exh,EGRC-bypass} + \dot{m}_{EGRC}) - \dot{m}_{exh,ExhSys}$$  \hspace{1cm} \text{Equation 3.103}

$$p_{vol,2} = \left(\frac{R_{exh} T_{exh,vol,2}}{V_{vol,2}}\right) \frac{dM_{vol,2}}{dt}$$  \hspace{1cm} \text{Equation 3.104}
3.2.6.4  Exhaust System Valve (downstream of the exhaust heat recovery system)

The exit from the tailpipe to the ambient is modeled as a flow restriction using the quasi-static modeling approach i.e. isentropic, compressible flow through a restriction. In the case of the tailpipe, the downstream pressure is the ambient pressure while the upstream pressure is an input from volume 2. As mentioned earlier, it was necessary to include this flow restriction in the scheme to maintain causality relations within the model. The equations used to model the mass flow rate of exhaust gases through the tailpipe, out in to the atmosphere are given by Equation 3.105 and Equation 3.106, for subcritical and choked flow, respectively:

Sub-critical flow: \( \frac{p_{\text{amb}}}{p_{\text{vol},2}} > \left( \frac{2}{\gamma+1} \right)^{\frac{\gamma-1}{\gamma}} \)

\[
\dot{m}_{\text{exh,sys}} = C_d \frac{A_{\text{nom}} p_{\text{vol},2}}{\sqrt{R T_{\text{vol},2}}} \left( \frac{P_{\text{amb}}}{p_{\text{vol},2}} \right)^{\frac{1}{\gamma}} \sqrt{\frac{2 \gamma}{\gamma - 1} \left[ 1 - \left( \frac{P_{\text{amb}}}{p_{\text{vol},2}} \right)^{\frac{\gamma-1}{\gamma}} \right]}
\]

Equation 3.105

Choked flow: \( \frac{p_{\text{vol},2}}{p_{\text{vol},1}} \leq \left( \frac{2}{\gamma_{\text{exh}}+1} \right)^{\frac{\gamma_{\text{exh}}-1}{\gamma_{\text{exh}}}} \)

\[
\dot{m}_{\text{exh,sys}} = C_d \frac{A_{\text{nom}} p_{\text{vol},2}}{\sqrt{R T_{\text{vol},2}}} \sqrt{\frac{2}{\gamma + 1} \left( \frac{\gamma+1}{\gamma-1} \right)^{\frac{\gamma-1}{\gamma}}} \]

Equation 3.106

The procedure employed to calculated the discharge coefficient \( C_d \) as a function of engine speed and throttle position is the same as that for calculating the loss coefficient for the exhaust gas cooler bypass valve; in this case, however, the quantity reverse-engineered is called the discharge coefficient. Once the discharge coefficients are obtained, surface fitting routines in MATLAB are applied to represent the discharge coefficient as a function of engine speed and throttle opening as in Figure 3.49. As is
seen, the values of discharge coefficients are in a range that is lower than the typical range of discharge values for a flow restriction, for example, a valve. The low discharge coefficients is attributed to the existence of plumbing that provides large resistance to the flow of exhaust gases; namely mufflers and the long piping downstream of the EGRC and EGRC bypass loop.

![Discharge Coefficient for Tailpipe to Environment “Flow Restriction” Model](image)

Figure 3.49 Discharge coefficient for tailpipe to environment “flow restriction” model

The adjusted $R^2$ value for the fit is 0.9957, and the RMS error between calculated discharge coefficient and the surface fit to the calculated values is 0.0034.
3.2.7. Transmission Thermal Management System

In Figure 3.50, a layout of the transmission thermal management system being studied as part of this work is shown. This transmission thermal management system consists of a transmission oil heater, in the form of a compact heat exchanger, which aids in rapid warm-up of transmission fluid in the early stages of a drive cycle or in cold-start conditions. This system also consists of a thermostat which routes transmission fluid to an air-cooled transmission oil cooler, which prevents the transmission oil from overheating.

The two heat exchangers and the flow of transmission is modeled and the details are discussed in the following sections.
3.2.7.1 \textit{ATF Flow Circuit}

In the warm-up phase of conventional powertrains, the transmission fluid reaches its operating temperature solely by absorbing the friction heat generated in the gears and torque converter elements. This leads to a slow warm-up of the transmission fluid and results in the transmission working at low efficiency conditions. For achieving a rapid warm-up of the transmission fluid, a transmission oil heater can be utilized and is part of the current transmission thermal management system. This allows for reducing the fluid of the viscosity, resulting in more efficient transmission operations. The transmission fluid flow path is shown in Figure 3.50.

During the initial warm-up phase, the transmission thermostat routes the fluid through the transmission oil heater (TOH). Here, the transmission fluid is heated by the engine coolant. This configuration of transmission fluid flow is seen in Figure 3.51 A.

![Diagram](image)

\textbf{A. Operation during ATF fluid warm-up} \hspace{1cm} \textbf{B. Operation after ATF fluid warm-up}

\textbf{Figure 3.51} Transmission fluid routing diagram during and after warm-up
As the transmission fluid approaches the temperature set-point of the thermostat, the thermostat begins to opens and the fluid flows through the transmission oil cooler (TOC), as seen in Figure 3.51 B. In the current thermal management system layout, the TOC is part of the front end module and occupies roughly 1/4\textsuperscript{th} of the height of the front end module, placed directly in front of the radiator. The lower 3/4\textsuperscript{th} of the front end module is the condenser; the TOC and condenser exist in a combined heat exchanger configuration. Therefore, the transmission fluid flows either through the TOH, or the TOC, or a combination of the two heat exchangers in the phase when the thermostat is only partially open. All the transmission fluid is routed through the transmission internals and torque converter. This way, the transmission oil heated rapidly in the initial stages of driving and then, when the prescribed temperature is reached, the transmission fluid thermostat ensures that the transmission oil temperature is maintained at the desired operating temperature set-point, by routing the transmission fluid appropriately. The flow of transmission fluid to either the TOH, or the TOC, or a combination of the two heat exchangers is modeled as shown in Figure 3.52.

Figure 3.52 Transmission oil flow circuit and mixing models
The implementation of the transmission fluid flow model and the transmission fluid thermostat is done in a simplified way. During the fluid warm-up phase, all the transmission fluid is routed to the transmission oil heater. As the fluid temperature rises and reaches desired operating temperature, the thermostat is commanded to start routing the transmission fluid to the TOC. The fraction of the total transmission fluid routed to the TOC depends on the fractional opening of the thermostat valve, which is a wax-based thermostat. The thermostat fractional opening map, which is a 1-D look-up table in Simulink containing fractional thermostat opening data versus transmission fluid temperature, is coupled with a first order transfer function that accounts for the time constant of melting of the wax element. The fraction of transmission fluid routed to the TOC is directly proportional to the fractional thermostat opening, while the remaining part of the transmission fluid is routed to the TOH. The two flow streams of the transmission fluid eventually mix before entering the transmission and torque converter assemblies. The temperature of the transmission fluid after mixing is given by:

$$T_{TO,trans.in} = \frac{\dot{m}_{TO,TOH}c_{pto}T_{TO,TOH,out} + \dot{m}_{TO,TOC}c_{pto}T_{TO,TOC,out}}{\dot{m}_{TO,TOH}c_{pto} + \dot{m}_{TO,TOC}c_{pto}}$$  

Equation 3.107

3.2.7.2  Slow Response Heat Exchangers

Within the transmission thermal management system, the transmission oil cooler, which is a liquid-to-air heat exchanger, is considered to be a slow response heat exchanger. The TOC is termed as a slow response heat exchanger because the time required by the TOC to respond to transients in input operating conditions is relatively higher than that for a liquid-to-liquid heat exchanger. This means that any changes in the inputs to the TOC,
such as changes in mass flow rates or temperatures of working fluid, take a considerable amount of time to affect the performance of the TOC. This is primarily because of the thermal response characteristics of the metal walls of the TOC that a change in input conditions propagates to influence the output characteristics only after a short time delay. The modeling principle for this type of a heat exchanger is similar to the models for the slow response heat exchanger explained in Section 3.2.4.2 (Engine Thermal Management System: Slow Response Heat Exchangers).

### 3.2.7.3 Fast Response Heat Exchangers

The transmission oil heater is the fast response heat exchanger in the transmission thermal management system. It is termed as a fast response heat exchanger due to its relatively compact size and hence a characteristic response to inputs transients that are more rapid compared to slow response heat exchangers. The technique followed to model the fast response heat exchangers is the same as described in Section 3.2.4.3 (Engine Thermal Management System: Fast Response Heat Exchangers).

### Section 3.3. Component-level Validation

Validation of component models is carried out for components for which manufacturer/test data is available. Only part of the test data, if available and if not used for calibration, is used for validation of the model for a given component. In the current scheme, manufacturer data or test data that may be used for validation purposes is available for the radiator, cabin heater, engine oil cooler, and the exhaust gas cooler. Also, coolant flow rates through some of the heat exchangers at some engine operating points are
known from correspondence with engineers at Chrysler LLC. In the following sections, a comparison between the predicted and experimental values is made. For the models not “validated” in this section, it is assumed that the calibration suffices.

3.3.1. Coolant Flow Network

As mentioned earlier, from prior correspondence with engineers at Chrysler LLC, the flow rate of coolant to the radiator for one operating condition is known. At this operating condition, as described in Table 3.10, the flow rate of coolant to the radiator is known to be 50 LPM (litres/min). As shown in Figure 3.53, the flow rate of coolant through the radiator is obtained to be 50LPM from the model.

Table 3.10 Condition for known coolant flow rate through the radiator

<table>
<thead>
<tr>
<th>Variable</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coolant temperature</td>
<td>100°C</td>
</tr>
<tr>
<td>Engine speed</td>
<td>2000 RPM</td>
</tr>
<tr>
<td>3-way valve position</td>
<td>50/50 towards TOH and CHC</td>
</tr>
<tr>
<td>Thermostat position</td>
<td>Fully open</td>
</tr>
<tr>
<td>EGRC coolant valve</td>
<td>Fully closed</td>
</tr>
</tbody>
</table>
This agreement between the predicted and actual flow rates is taken as verification for the model for the coolant flow. Other conditions and valve configurations could not be tested or verified due to the lack of test data at other operating conditions pertaining to coolant flow rates in the coolant flow circuit.

### 3.3.2. Radiator

The validation of the radiator model is conducted using steady state data points obtained from the manufacturer and not used for calibration purposes. A comparison between the output of the radiator model and experimental data, when given inputs of flow rates and temperatures of coolant and air, is seen in the following figures. The following figures compare the radiator-out coolant temperatures, radiator-out air temperatures, and show
histograms of the error between the outlet temperatures of coolant and air from experimental data and model predictions.

Figure 3.54 Radiator model validation: Coolant temperature comparison
Figure 3.55 Radiator model: Histogram of error in coolant temperature prediction

Figure 3.56 Radiator model validation: Air temperature comparison
Figure 3.54 shows that the agreement between the coolant temperature predicted by the model and that obtained experimentally is very strong. A similar comparison for the air temperature leaving the radiator is made in Figure 3.56, where the predicted radiator-out air temperature matches the experimental values. Figure 3.55 and Figure 3.57 shows % error in prediction of coolant and air temperature out of the radiator. All prediction errors are well within 1.5% of temperatures recorded during experiments for the same test conditions.
3.3.3. Cabin Heater

Since the cabin heater model was not shown in the model calibration section, all the data points are shown in the following figures, but it must be noted that only part of these data points were used for calibration while the others were used for validation purposes. All the data points below were obtained from the manufacturer of the cabin heater and are recorded by varying inlet coolant temperature and air temperature, as well as flow rates for both the working fluids. The recorded outputs are the cabin heater-out coolant temperature and cabin heater-out air temperature. As can be seen, a wide range of operating conditions is tested. Figure 3.58 and Figure 3.59 show a comparison between the predicted and experimental coolant and air temperature out of the cabin heater module. These values agree with each to acceptable accuracy. In Figure 3.60, which shows the % error between the model prediction and the experimental values, it is seen that there are conditions with upwards of 5-8% error between the two values. However, it must be noted that this model, part of the VES, is to be used for simulating scenarios pertaining to the EPA city/highway drive cycle. During this drive cycle, the cabin heating system is not used. The cabin heater core is taken into account to include the pressure drop characteristics of coolant flow through this component, and this thermal model of the cabin heater core is presented here for completeness. Nevertheless, the model presented here, although includes a maximum of 8% error (this error is for conditions with very slow air flow conditions over the cabin heater) may be used to examine patterns in the thermal characteristics of the cabin heater operation.
Figure 3.58 Cabin heater model validation: Coolant temperature comparison

Figure 3.59 Cabin heater model validation: Air temperature comparison
3.3.4. Engine Oil Cooler

The final component for which manufacturer data is available for validation purposes is the engine oil cooler. Figure 3.61 shows a comparison between the EOC-out coolant temperatures obtained from the model and how this compares with the experimental data for the same test cases. Figure 3.62 shows a similar comparison for EOC-out oil temperatures. It is seen that the agreement between model and tests is very strong. This verification process is reinforced on observing Figure 3.63 and Figure 3.64 which show the % error in prediction of EOC-out coolant and oil temperatures. The maximum error in temperature prediction for either of the two working fluids part the EOC is less than
0.5%. The mean % error between measured values for EOC-out coolant and oil temperature and predictions thereof are is -0.1 and 0.1% while the standard deviations in % errors are 0.06% and 0.17%, respectively.

Figure 3.61 EOC model validation: Coolant temperature comparison
Figure 3.62 EOC model validation: Histogram of error in prediction

Figure 3.63 EOC model validation: Oil temperature comparison
Figure 3.64 EOC model validation: Histogram of error in prediction
Section 3.4. References


Chapter 4: Experimental Validation of the Coupled Models

In the previous chapter, the description of calibration as well as validation of component-level models is described. This chapter presents a methodical approach of validation of the VES, which is formed by coupling of the component-level models described in the previous chapter. The various component models described earlier are part of one of many subsystems within the vehicle, namely, the engine air path and combustion, engine thermal management, the transmission thermal management and so on. Some of the subsystem models not described in this work, i.e. the transmission thermal dynamics and mechanical model, the shift scheduling strategy, the Deceleration Fuel Shut-Off (DFSO) model, the Torque-Converter thermal dynamics and mechanical model, the vehicle dynamics model, however have been modeled in [4.1] and are part of the VES.

In order for validating the VES a two step process is followed, as depicted by Figure 4.1. Before the VES is formed by combining all of the component-level models, partial validation of the subsystem models is conducted. Stand-alone models for these subsystems are formed by coupling component-models pertaining to only those respective subsystems.
Figure 4.1 Schematic of steps taken for model validation

For example, to form the engine subsystem model as seen in the figure above, the engine air-path and torque productions models as well as the model for the heat rejection from combustion gases to the walls of the combustion chamber are coupled in one separate subsystem. The thermal management subsystem is formed by combining the models for the heat exchangers, the engine thermal dynamics, and the coolant and oil flow, and so on. In this work, the subsystems for which stand-alone models are first validated before being amalgamated into the VES are:

1. The engine subsystem model, and
2. The thermal management subsystem model.
Stand alone models for these two subsystems are created by coupling the mathematical equations describing each of its individual components. This model, created with Simulink, is then provided with the necessary inputs measured during vehicle driving tests. The process followed to validate the models for these two subsystems is illustrated in the flowchart below.
The outputs from these stand-alone models, obtained by simulating the same conditions as the vehicle drive cycle test, are compared with the corresponding variables obtained from the drive cycle test. If the model outputs and test data agree with each other, then the modeling process is continued to the next step i.e. VES compilation. On the other hand, if there is discrepancy between model output and test data, extensive trouble-shooting is carried out until the two are in agreement with each other, as illustrated in Figure 4.2.

All the data used to validate the stand-alone subsystem models described above comes from one drive cycle test conducted in a chassis dynamometer. This test is performed using a vehicle with a conventional thermal management system (described later in this section).

After the stand alone model outputs agree with the test data, the engine subsystem model and the thermal management system models are coupled in the same model along with [4.1] the models developed in. This assembly, as a whole, constitutes the Vehicle Energy Simulator (VES). The subsystem models developed in [4.1] follow a similar procedure for validation, wherever data regarding the relevant input and output variables is available from drive cycle tests. The coupled version of all stand-alone subsystem models, the VES, is then given as input the desired drive cycle velocity profile. This is the only required input as the VES is a forward looking energy simulator. The outputs from the VES are compared with corresponding variables of interest from test data.

To validate the VES, two different sets of test data are used. The first data set is obtained from drive cycle testing of a vehicle equipped with a conventional thermal management
system. This is the same data set used to validate the stand alone models. The second set of data is obtained from drive cycle testing of a similar vehicle but equipped with an advanced thermal management system. This advanced system consists of a three way valve, an exhaust gas cooler, an exhaust bypass valve and a transmission oil heater.

Following sections provide a brief description of test vehicle and the tests performed using the test vehicle. Also, results from validation of the stand alone models as well as the coupled VES model are discussed in details. Possible sources of modeling errors and disagreement between model output and test data are also explained where necessary.

Section 4.1. Description of Test Vehicle

The test vehicle used to conduct the drive cycle tests is a 2011 Chrysler Town and Country Minivan. Basic vehicle system configuration is seen in Table 4.1.

Table 4.1 System configuration of test vehicle

<table>
<thead>
<tr>
<th>Vehicle Configuration</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vehicle Make and Model</td>
</tr>
<tr>
<td>Chrysler, Town &amp; Country (Minivan)</td>
</tr>
<tr>
<td>Year</td>
</tr>
<tr>
<td>2011</td>
</tr>
<tr>
<td>Edition</td>
</tr>
<tr>
<td>Limited edition</td>
</tr>
<tr>
<td>Engine specifications</td>
</tr>
<tr>
<td>Same as Table 3.1</td>
</tr>
</tbody>
</table>

Section 4.2. Description of Tests Performed

In order to validate models in transient conditions, data from transient vehicle operation is required. The test performed to collect data for model validation purposes consists of a standard drive cycle conducted on a chassis dynamometer. The drive cycle selected for
Transient vehicle testing is the EPA Federal Test Procedure, also called the EPA75 or simply the FTP cycle. This drive cycle consists of the Urban Dynamometer Driving Schedule (UDDS) followed by first 505 seconds of the UDDS cycle \([4.2]\).

A selected drive cycle specifies a fixed vehicle velocity profile to be followed by the driver, within acceptable margin of error. The velocity profile that constitutes the FTP drive cycle is shown in the figure below.

![EPA Federal Test Procedure](image)

**Figure 4.3 Velocity profile: EPA 75 drive cycle**

For the purposes of model validation, data only from the first \(~1370\) seconds of the FTP drive cycle is used. Therefore, the data obtained from the performed drive cycle tests represents the UDDS cycle which includes the cold-start phase and the transient phase that last 505 seconds and 864 seconds, respectively.
The test vehicle response during the FTP cycle is seen in Figure 4.4. This figure shows the vehicle velocity, engine speed, the gear positions, and the throttle openings for the first ~1370 seconds of the FTP drive cycle.
Section 4.3. Validation: Part 1 – Stand Alone Model Validation

As mentioned earlier, models of individual subsystems are validated as a first step towards the VES validation. The benefit of this step is that this partial validation helps conducting a verification of the model implementation and, from careful observation, helps tuning the calibration parameters if necessary. The stand-alone models which are tested as parts of this work, as a prelude to VES validation, are the engine air-path/torque production as well as the engine thermal management system models. The relevant inputs, outputs and intermediate results obtained from validation of stand-alone models are explained in the following sub-sections.

4.3.1. Engine Air Path and Torque Production Model

The input-output structure for the engine air path and torque production model is shown in Figure 4.5. The inputs to the engine model are engine speed, % throttle opening, and temperature of coolant, engine oil and fuel in the fuel rails. The outputs of this model pertain to the torque variables (brake torque, friction etc.), intake manifold pressure, mass flow rate of air and fuel, temperature and pressure in the exhaust manifold and downstream of the catalytic converter, rate of heat rejection to the walls of the combustion chamber, and so on.

For the purpose of validation of this model, the engine speed and throttle opening profiles recorded during the FTP drive cycle test are imposed as inputs. The FTP cycle is performed using a vehicle equipped with the conventional thermal management system.
Figure 4.5 Input-output structure of engine air path and torque production models

The outputs of the engine model for which a comparison is shown here with corresponding test data are: fuel flow rate, brake torque, and cumulative fuel consumption. Comparing model output to test data for variables such as heat rejection to the walls of the combustion chamber is not possible as heat generation/rejection terms are not directly measurable quantities. For comparing the heat generation terms, the IMEP or the indicated power would need to be calculated which would require the installation of in-cylinder pressure sensors; measurements which were not included in the tests mentioned here.
As seen in the figure above, the fuel mass flow rates predicted by the model and that recorded during drive cycle testing are in good agreement with each other. The peaks in fuel flow rate are captured by the model at the same times as they appear in the test data. There are, however, some differences between the engine model and the test data. It shown in Figure 4.6, that there are instances when the fuel flow is zero, but the model does not capture this behavior. This is due to the fact that the DFSO (Deceleration Fuel Shut-Off) strategy, which is part of the controls implemented in the vehicle ECU, is not replicated in the stand alone engine model. The logic followed for shutting fuel flow to the cylinders requires inputs from other models (driver and torque converter models) which are not part of this subassembly of models. The DFSO strategy is, however, highlighted in [4.1], and is part of the VES.
In the figure below, it is seen that the engine brake torque predicted by the model, based on the inputs to this subassembly, closely follows the engine torque output profile recorded during tests. The brake torque predicted during idle is higher than that recorded in the drive cycle tests. This is because, in the chassis dynamometer tests, the alternator adds additional load to the crankshaft thereby reducing the effective brake torque output.

The data used to calibrate the engine torque maps is obtained from steady state engine dynamometer tests where the alternator load is decoupled, as described in Section 3.2.2.2.

![Figure 4.7 Stand-alone engine model validation: Engine Brake Torque comparison](image)

Although not shown in the above figure, this effect is included in the implementation of the VES model, as part of the electrical subsystem model. Another effect not seen in the brake torque profile is the “negative” torque experienced by the vehicle during...
decelerations in the drive cycle test, as the effect of braking is not considered in this subassembly; it is also part of [4.1].

A comparison of the cumulative fuel consumption is shown in Figure 4.8. It is seen that the model and test data agree strongly with each other. The cumulative fuel consumed at the end of 1370 seconds of the drive cycle is 1.045 kg during vehicle testing while the model prediction is 1.049 kg; a difference of less than 0.4%.

![Figure 4.8 Stand-alone engine model validation: Cumulative fuel used comparison](image)

After this step, the engine model, apart from the constant changes to model implementation structure, is considered to be functioning and is held as being validated for its transient response. The next step is to validate the model of the thermal system which, in fact, needs inputs from the stand-alone engine model. This justifies the
rationale behind the order followed in which the models are validated i.e. engine model→thermal system model→VES. This order is followed because there are enough variables being recorded during vehicle drive cycle testing to validate (partially) the engine model, but not the thermal system model.

The validation process is now discussed for the thermal system models which include models of all of the heat exchangers, engine thermal dynamics as well as the coolant flow circuit.

**4.3.2. Thermal System Model**

The input-output structure for the thermal system model is shown in Figure 4.9. The inputs to this subsystem model are obtained from either the drive cycle tests or from the stand-alone engine model. The input from drive cycle tests is the engine rotational speed. The outputs of the engine model which act as inputs to the this thermal system model are: heat rejection rate to the walls of the combustion chamber, heat generation due to friction, temperature at the exit of the catalytic converter and pressure in the post-catalytic converter volume. These are obtained by simulating the stand-alone engine model and saving the trajectory of these variables as a function of time.

The mass flow rate of the coolant to the engine and heat exchangers as well as mass flow rate of engine oil to the engine are calculated within the thermal system model.
The thermal system model required as inputs the velocity of air impinging the face of the radiator. This is calculated as follows:

\[
V_{face} = \sqrt{\frac{\eta_{fan} \cdot \eta_{elec} \cdot W_{elec}}{\frac{1}{2} \cdot C_d \cdot \rho_{air} \cdot A_{fan}}}
\]

Equation 4.1

where,

\[
W_{elec} = V_{fan,motor} \cdot I_{fan,motor}
\]

is the measured power consumption of the fan,

\[
V_{fan,motor} \quad \text{and} \quad I_{fan,motor}
\]

are the measured fan voltage and current, respectively, \(A_{fan}\) is the nominal area of the fan, \(\eta_{fan}\) is the fan efficiency taken to be equal to 59.3% (per Betz limit [4.3]), and \(\eta_{elec}\) is that fan electrical efficiency taken as 90%.
Validation for the stand alone thermal system model is conducted by comparing the temperatures of the coolant and engine oil from test data obtained from a vehicle equipped with a conventional thermal management system. In this vehicle, not all of the aforementioned heat exchangers that are part of the advanced thermal management system are present. Therefore, the valve positions in the stand alone thermal system model, originally formulated for the advanced thermal management system, are set to specific positions to mimic the conventional thermal management system, as depicted by the figure below.

![Figure 4.10 Conventional and Advanced Thermal Management Systems](image)

Referring to Figure 4.10, in order for the model of the advanced TMS to mimic the conventional TMS, the valve controlling coolant flow to the EGRC heat exchanger is
fully closed and the exhaust bypass valve (not shown in the figure below) is left completely open. Also, the 3-way valve position is such that all the coolant flow is routed to the cabin heater, thereby short-circuiting the coolant flow to the transmission oil heater.

Apart from the differences seen in Figure 4.10, the thermostat used in the baseline case is one which starts opening at 95°C and is fully open at 102°C, as reported in the manufacturer’s data. The thermostat used as part of the advanced thermal management where the thermostat starts opening at 100°C and is completely open by 118°C, as reported in the manufacturer’s data.

The time traces of the heat generation terms provided as inputs to the stand-alone thermal system model from the stand-alone engine model, for validation purposes, are shown in Figure 4.11.
Figure 4.11 Inputs to the thermal system model: Rate of Heat Generation terms

Thermal system model outputs that are compared with the corresponding variables from test data are: the temperature of the coolant out of the engine and the temperature of engine oil in the oil sump. These comparisons are shown in Figure 4.12 and Figure 4.13, respectively. It is seen that the coolant and oil temperature followed closely the trajectories of coolant and engine oil warm-up seen during vehicle drive cycle tests. In the evolution of temperature trace of coolant, however, it is seen that the coolant temperature rises very quickly in the period from ~100 to 275 seconds. Also, the coolant temperature at which the thermostat in the conventional system starts to open is reported...
to be 95°C. But, it is seen in the experimental data for the coolant warm-up that there is a “plateau region” between 290-350 seconds of testing where the temperature of the coolant does not rise. This effect is captured by the model but at a higher temperature. It is inferred that there is fractional opening of the thermostat at a temperature as low as 85-87°C, where this plateau is observed. There is also possibility of leakage flow through the thermostat at an even lower temperature i.e. when the model predicted temperature starts to deviate from the experimental trace for coolant temperature at 65°C, as the model does not account for leakage flows.

Figure 4.12 Stand-alone thermal system model validation: engine-out coolant temperature
For the remaining part of the cycle, the coolant temperature predicted by the model is in close agreement with the experimental trace with the error between the two always being less than 4°C.

It is worth mentioning that the thermostat model formulated in this work is overly simplified, mainly consisting of lookup tables that approximate the steady-state characteristic of the thermostat opening and closing fractions as functions of the coolant temperature, coupled with a first-order transfer function that mimics the opening and closing dynamics. In this sense, the thermostat model could be considerably improved. Furthermore, a more accurate model of the control logic for the radiator fan operation would improve the predictive capability of this model.

Figure 4.13 Stand-alone thermal system model validation: engine oil (oil sump)
The predicted sump engine oil temperature agrees closely with the measured values. For the first ~800 seconds, the deviation between the oil temperature from the model and from the drive cycle test is always less than 2.5°C. The deviation, after this point, increases to 6°C at 1100 seconds but the engine oil temperature is seen to approach the same steady state temperature as observed during the tests.

Section 4.4. Validation: Part 2 - Transient Validation of the VES

In this section, the validation of the entire VES model is explained. In order to conduct the validation of the complete VES, two sets of data are used. The first set is the same as the one used in the previous sections to validate the standalone subsystem models, i.e. from a vehicle equipped with a conventional thermal management system. The second set of data is obtained from a test vehicle equipped with all of the heat exchangers and valves that are part of the advanced thermal management system. The data sets used for model validation and the details of the validation process are described in the following two subsections.

4.4.1. Conventional Thermal Management System Configuration

The first set of transient drive cycle data available for validation of the VES model is from a 2011 Chrysler Town & Country Minivan equipped with the conventional thermal management system. The transient test, as described earlier, is based on an FTP drive cycle test performed on a chassis dynamometer. The vehicle configuration used to conduct this test, along with a list of the heat exchangers present in this system is seen in the table below.
Table 4.2 Configuration of vehicle with conventional TMS

<table>
<thead>
<tr>
<th>Vehicle Make and Model</th>
<th>Chrysler, Town &amp; Country (Minivan)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Year</td>
<td>2011</td>
</tr>
<tr>
<td>Thermal Management System</td>
<td>Conventional</td>
</tr>
</tbody>
</table>

**Heat exchangers and valves present as part of the conventional thermal management system**

<table>
<thead>
<tr>
<th>Component</th>
<th>Status</th>
</tr>
</thead>
<tbody>
<tr>
<td>Radiator</td>
<td>YES</td>
</tr>
<tr>
<td>Engine oil cooler</td>
<td>YES</td>
</tr>
<tr>
<td>Transmission oil cooler</td>
<td>YES</td>
</tr>
<tr>
<td>Cabin heater</td>
<td>YES</td>
</tr>
<tr>
<td><strong>VALVES:</strong> Exhaust gas cooler-coolant side valve</td>
<td>NO</td>
</tr>
<tr>
<td>Exhaust gas bypass valve</td>
<td>NO</td>
</tr>
<tr>
<td>Transmission oil heater, transmission oil thermostatic valve</td>
<td>NO</td>
</tr>
</tbody>
</table>

To model this hardware configuration, the VES was used, however imposing fixed valve positions so as to replicate the behavior of the actual hardware. Table 4.3 shows the positions of the valve in the advanced thermal management system that have been imposed during this validation.

Table 4.3 Valve positions for simulation of conventional TMS

<table>
<thead>
<tr>
<th>Valve Type</th>
<th>Position</th>
</tr>
</thead>
<tbody>
<tr>
<td>3-way valve</td>
<td>100% coolant flow towards the cabin heater i.e. 0 opening towards the TOH</td>
</tr>
<tr>
<td>Exhaust gas cooler - coolant side valve</td>
<td>Fully Closed</td>
</tr>
<tr>
<td>Exhaust gas bypass valve</td>
<td>Fully Open</td>
</tr>
</tbody>
</table>
The ambient pressure and temperature are set to 1 bar and 27°C, respectively, as recorded during the testing. The initial conditions for the temperatures of the masses and fluids modeled in the VES are set to be the same as the ambient temperature, which was recorded after an overnight “soak” of the vehicle in a controlled test cell environment. The input to the VES is the desired velocity profile, which for the following tests, is the standard FTP cycle shown in Figure 4.3.

In Figure 4.14, a comparison between the coolant warm-up trace predicted by the model versus the coolant temperature recorded during the test is shown. The coolant
temperatures predicted by the model agrees closely with that recorded during transient testing of the vehicle. The average error between the predicted coolant temperatures and experimental values is calculated to be 1.6°C and the standard deviation in error is 2.2°C. The time taken by the VES model to reach 90°C is predicted to be 379 seconds whereas during experiments this time is 401 seconds, yielding a difference of 22 seconds, which is considered to be small.

The oil-warm-up curves obtained from the model and experiments are shown in Figure 4.15. The time predicted by the VES for engine oil to warm-up to 80°C is 527 seconds, while the actual warm-up time recorded to reach the same temperature is 573 seconds. The average error between the engine oil temperature prediction and test data is calculated to be 1.9°C with a standard deviation in error of 2.3°C.

The coolant temperature directly affects the temperature of the oil, as they interact via the engine oil cooler. It is seen during the period between 500-1000 seconds that both these temperature are over-predicted.
Figure 4.15 VES Validation (conventional TMS): comparison of engine oil warm-up trajectory (model prediction versus experimental data)

Figure 4.16 shows a similar curve for transmission fluid warm-up. Also in this case, the warm-up trace of the transmission fluid is well captured by the model. The transmission fluid temperature increases very slowly at an approximate rate of 1.7°C/min over the duration of the drive cycle tests. The time predicted for the transmission fluid to reach 50°C is predicted to be 720 seconds by the VES model, whereas the corresponding time for transmission fluid warm-up to the same temperature is recorded to be 719 seconds during transient drive cycle experiments. The average error in prediction of transmission warm temperature is 2.1°C with a standard deviation in error of 1.8°C.
Figure 4.16 VES Validation (conventional TMS): comparison of transmission fluid warm-up trajectory (model prediction versus experimental data)

A summary of the time required by the fluids to warm-up to threshold temperatures as seen in experiment and that predicted by the VES model, the average error in prediction, and standard deviation in error is seen in the Table 4.4.
Table 4.4 Summary of experimental test and model simulation results for vehicle with conventional thermal management system

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Temperature for “warm-up” [°C]</th>
<th>Time till warm-up temperature is reached (experiments) [seconds]</th>
<th>Error in prediction of warm up time [seconds]</th>
<th>RMS Error in temperature prediction [°C]</th>
<th>Avg. error in prediction [°C]</th>
<th>Std. Dev in error [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coolant</td>
<td>90</td>
<td>401</td>
<td>-22</td>
<td>1</td>
<td>1.6</td>
<td>2.2</td>
</tr>
<tr>
<td>Engine oil</td>
<td>80</td>
<td>573</td>
<td>-46</td>
<td>1</td>
<td>1.9</td>
<td>2.3</td>
</tr>
<tr>
<td>ATF</td>
<td>50</td>
<td>719</td>
<td>+1</td>
<td>1</td>
<td>2.1</td>
<td>1.8</td>
</tr>
</tbody>
</table>

4.4.2. Advanced Thermal Management System Configuration

The second set of data available for validation of the VES model is obtained from a test vehicle (test mule) consisting of an advanced thermal management system. For the drive cycle tests using the test vehicle, the thermal management system is manually controlled i.e. there are no automated control strategies implemented on this test vehicle to control the positions of valves in the advanced TMS. The vehicle configuration, heat exchanger module, and the valves that are part of the TMS are listed in Table 4.5.

It should be noted that this was a test vehicle with the same engine specifications as the conventional vehicle. The additions to this test vehicle were the heat exchangers and valves i.e. parts of the advanced thermal management system. The purpose of modifying a baseline vehicle and fitting it with an advanced TMS was to generate data needed for validating models. The objective of developing these models is to aid in model-based control development for the actuators in the TMS.
Table 4.5 Configuration of vehicle with advanced TMS

<table>
<thead>
<tr>
<th>Vehicle Configuration</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vehicle Make and Model</td>
</tr>
<tr>
<td>Year</td>
</tr>
<tr>
<td>Thermal Management System</td>
</tr>
</tbody>
</table>

**Heat exchangers and valves present**

<table>
<thead>
<tr>
<th>Heat exchangers and valves present</th>
<th>YES</th>
</tr>
</thead>
<tbody>
<tr>
<td>Radiator</td>
<td>YES</td>
</tr>
<tr>
<td>Engine oil cooler</td>
<td>YES</td>
</tr>
<tr>
<td>Transmission oil cooler</td>
<td>YES</td>
</tr>
<tr>
<td>Cabin heater</td>
<td>YES</td>
</tr>
<tr>
<td>Exhaust gas cooler, coolant side</td>
<td>YES</td>
</tr>
<tr>
<td>Exhaust gas bypass valve</td>
<td>YES</td>
</tr>
<tr>
<td>Transmission oil heater,</td>
<td>YES</td>
</tr>
<tr>
<td>transmission oil thermostatic</td>
<td>YES</td>
</tr>
<tr>
<td>valve</td>
<td></td>
</tr>
</tbody>
</table>

The simulator used to model this hardware configuration is the one that has been the focus of this work as described here in 0 and partly described in the work shown in [4.1]. For this process, the simulator makes use of all the components that are part of the advanced thermal management system, unlike the baseline where some of the heat exchangers are bypassed during simulation.

Table 4.6 Valve positions while testing vehicle equipped with the valves and heat exchangers part of the advanced TMS and for simulation of advanced TMS

<table>
<thead>
<tr>
<th>Valve</th>
<th>Position</th>
</tr>
</thead>
<tbody>
<tr>
<td>3-way valve</td>
<td>100% coolant flow towards the TOH i.e. 0 opening towards the cabin heater</td>
</tr>
<tr>
<td>Exhaust gas cooler -</td>
<td>Fully Open</td>
</tr>
<tr>
<td>coolant side valve</td>
<td></td>
</tr>
<tr>
<td>Exhaust gas bypass</td>
<td>Fully Closed i.e. all exhaust is routed towards the exhaust gas cooler</td>
</tr>
<tr>
<td>valve</td>
<td></td>
</tr>
</tbody>
</table>

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In order to test a condition when the advanced thermal management system is operational, the scenario with valve positions as shown in Table 4.6 is tested on the vehicle. The VES is then used to simulate the same conditions and results are compared. The thermostat used as part of the advanced thermal management system is designed to maintain the coolant temperature at 105°C at steady state. A higher operating temperature threshold is set due to the benefits on engine operations, as explained in Section 2.2.

Figure 4.17 VES Validation (advanced TMS): comparison of coolant warm-up trajectory (model prediction versus experimental data)

In Figure 4.17, the coolant warm-up trend predicted by the model is compared with what is seen during the drive cycle test. In approximately the first 200 seconds, the agreement
between the coolant warm-up traces from the drive cycle test and the model is very strong. After this period, the coolant temperature starts deviating. This deviation is attributed to inaccuracies in the model of the thermostat which could be improved further.

In the part of the FTP cycle after 600 seconds, the coolant temperatures from both experiments and the model converge to ~105°C. The “cycling” seen in the coolant temperature is not captured by the model as the radiator fan operation logic implemented in the model needs improvement. The time taken during experiments by the coolant to warm-up to 105°C is seen to be 647 seconds, while the model prediction is 636 seconds. The average error in coolant temperature prediction is 3°C while the standard deviation in error is calculated to be equal to 4°C.
In Figure 4.18, a similar comparison of the warm-up trajectories of engine oil from experiments and model is shown. It is seen that even though the high frequency oscillations in oil temperature are not captured by the model, the trend in engine oil warm-up predicted by the model aligns very closely with what is seen in experimental data. The time for the engine oil to warm-up to 100°C is seen to be 788 seconds during experiments. The corresponding warm-up time predicted by the model is 831 seconds; a difference of 43 seconds. The steady state error between model and test data is less than 3°C. The average error in engine oil prediction is calculated to be equal to -2.8°C with a standard deviation of 2.6°C.
The evolution of transmission fluid temperature as predicted by the model for the first part of the FTP cycle is shown in Figure 4.19. The same figure also shows the transmission fluid temperature versus time as seen during FTP drive cycle test. During this test, the advanced thermal management system is operational which makes use of the transmission oil heater. The warm-up of transmission oil is seen to much more rapid as compared to the baseline case seen in Figure 4.16. Also, a relatively higher steady state temperature is attained by the transmission fluid in this case were the transmission oil heater is used.
The high frequency variations in transmission oil temperature are not captured by the model but the general trend of transmission fluid warm-up has been captured. It is seen that the transmission oil temperature is initially under-predicted and then, later in the cycle, is over-predicted. However, the same steady state temperature of 90°C is reached for transmission oil temperature during simulation and actual vehicle testing. The average error in transmission oil temperature prediction is -0.6°C, but due to the high initial under-prediction and over-prediction later in the cycle, the standard deviation is much higher at 4.6°C.

Table 4.7 Summary of experimental test and model simulation results for vehicle with advanced thermal management system

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Temperature for “warm-up” [°C]</th>
<th>Time till warm-up temperature is reached (experiments) [seconds]</th>
<th>Error in prediction of warm up time [seconds]</th>
<th>RMS Error in temperature prediction [°C]</th>
<th>Avg. error [°C]</th>
<th>Std. Dev in error [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coolant</td>
<td>105</td>
<td>647</td>
<td>-11</td>
<td>1</td>
<td>3</td>
<td>4</td>
</tr>
<tr>
<td>Engine oil</td>
<td>100</td>
<td>788</td>
<td>+43</td>
<td>1</td>
<td>-2.8</td>
<td>2.6</td>
</tr>
<tr>
<td>ATF</td>
<td>80</td>
<td>821</td>
<td>-103</td>
<td>1</td>
<td>-0.6</td>
<td>4.6</td>
</tr>
</tbody>
</table>

As mentioned earlier, the validation process outlines potential directions for improving the model accuracy. One of these is a more accurate model of the wax thermostat. Even though the model of the thermostat consisted of different opening and closing curves as well as different time constants for opening and closing, the model was not able to capture the actual dynamics of thermostat opening and closing. The inaccuracies could
have been either due to discrepancy in the data obtained from the manufacturer or due to non-consideration of conditions for which the modeled thermostat opening and closing behavior does not align with the true working of the thermostat. A solution to this would be to model the thermostat using a physics-based model that characterizes the melting of the wax in the thermostat. Also, differences in simulation results for fluid temperatures, especially the coolant temperature during transients, may occur due to the radiator cooling fan operation logic, which was not captured accurately by the model. Due to limited information on the radiator cooling fan operation. Also, the input to the thermal model is the face velocity of the radiator, which, even if the fan operation logic is captured correctly, may not have been calculated precisely using Equation 4.1.

Further investigation into the variation between coolant, engine oil and transmission fluid temperatures shows that, on average, the coolant temperature is over-predicted and the engine oil and transmission fluid temperatures are over predicted. This nature of in the error distribution is seen in Figure 4.20.

The distribution indicates that the coolant temperature is always higher than what is seen in the drive cycle tests while the engine oil and transmission fluid temperatures are lower than what they should be (on average). This deviation could be attributed to the modelling approach i.e. the effectiveness-NTU method followed to characterize the thermal behavior of the engine oil cooler and transmission oil heater. The modeled effectiveness of these two heat exchangers may be lower than the actual effectiveness of the heat exchanger in some operating conditions. The lower value for effectiveness would
result in less heat transfer from the coolant to the transmission fluid and the engine oil via the respective heat exchanger. This would, in turn, result in the temperature of the coolant

Figure 4.20 Distribution of error in fluids temperature prediction
being higher while the temperature of the other two fluids would be lower than the current prediction because the coolant is not able to dissipate enough heat to the two fluids via the compact heat exchanger.
Section 4.5. References


Chapter 5: System Analysis and Optimization

The goal of this chapter is to investigate open-loop control strategies for the thermal system actuators to rapidly warm-up coolant, engine oil and transmission oil during cold-start conditions. Rapid fluid warm-up is important in order to reduce fluid viscosities and, thereby, losses in the engine and powertrain components.

The actuators present in the thermal system under investigation are the exhaust gas bypass valve, the exhaust gas cooler coolant-side valve and the three way valve (refer to Figure 1.2). Possible fractional angular openings of these valves are: Exhaust gas bypass valve $\epsilon [0, 1] \in \mathbb{R}$, Exhaust gas cooler coolant-side valve $\epsilon [0, 1] \in \mathbb{R}$, and Three way valve $\epsilon [0, 1] \in \mathbb{R}$. In each of these cases, $0$ corresponds to fully closed, $1$ corresponds to fully open and the intermediate positions can be defined in a continuous way. The “optimal strategy” of actuation for these three valves is considered to be the one which results in least fuel consumption, highest transmission efficiency and least Friction Mean Effective Pressure (FMEP), for the FTP cycle. However, rapid fluid warm-up is considered to be the proxy for satisfying these requirements.

The only constraint, that needs to be satisfied when trying to achieve these goals, is to maintain the E-ratio for the EGR cooler, defined by Equation 3.85, below $0.4$. Respecting
this limit ensures that the exhaust gas cooler is used in a way that is not detrimental to its health.

The conditions stated here give rise to an optimization problem where the fluid warm-up times, fuel consumption and FMEP have to be minimized and transmission efficiency has to be maximized, while keeping the E-ratio below 0.4.

The larger project, of which this work is a part, entails finding the “optimal” trajectory for thermal system actuators using the Genetic Algorithm (GA) optimization technique. In this work, a preliminary analysis on the system is performed to define the first generation for the GA so that the optimization can be started from reasonable initial conditions. It should be noted that, in complex multi-objective optimization problems, it is highly possible for a numerical algorithm to “get stuck” in a local optimum and provide that as an optimal solution instead of the true global optimum. Since the domain in which the control inputs can vary is large (\([0, 1] \times [0, 1] \times [0, 1] \in \mathbb{R} \times \mathbb{R} \times \mathbb{R}, \text{ or } \mathbb{R}^3\)), providing the GA with reasonable initial conditions will increase the chances of the optimization routine to converge to the global optimum. The analysis conducted to specify reasonable first generations for the GA is shown in sections 5.1.2 and 5.1.3.

The first step of the analysis will be to determine whether the warm-up of one fluid should be prioritized over another or not. This will be done by decoupling the effect of engine oil and transmission oil temperature on, a) Vehicle fuel consumption and, b) the amount of heat generated in the engine and transmission components.

The second step of the analysis will be to study the effect of various combinations of thermal-system-actuator positions on the warm-up time of the three fluids for the FTP
cycle. The effect of fixed actuator positions on fluid warm-up time will be evaluated through this analysis.

5.1.1. Case studies: Engine oil and transmission oil temperature effects

As mentioned earlier, in the first part of the analysis the effect of engine oil and transmission oil warm-up on the vehicle fuel consumption and energy generation in the engine and transmission components will be decoupled. For this purpose, the case studies reported in Table 5.1 are considered.

<table>
<thead>
<tr>
<th>Scenario</th>
<th>Engine Oil Temperature Varied</th>
<th>Transmission Oil Temperature Varied</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case</td>
<td>1 2 3 4 5</td>
<td>1 2 3 4 5</td>
</tr>
<tr>
<td>Engine Oil Temperature [°C]</td>
<td>0 10 30 50 80</td>
<td>Unchanged – typical warm-up</td>
</tr>
<tr>
<td>Transmission Fluid Temperature [°C]</td>
<td>Unchanged – typical warm-up</td>
<td>0 10 30 50 80</td>
</tr>
</tbody>
</table>

In this study, two scenarios are considered: one where the engine oil temperature is artificially kept constant and the other where the transmission oil temperature is artificially kept constant. For each of the two scenarios, five different temperatures are considered. Therefore, in each case, the fluid temperature is kept constant throughout the entire simulation while the other fluid temperature evolution is left unchanged. The ambient pressure and temperature is set as 1 bar and 25°C, respectively.
For these simulations, the use of the thermal management system is disabled for all the cases considered here for both scenarios. Therefore, the position of the three way valve is such that all coolant is flowed to the cabin heater core, the valve routing coolant to the EGRC is fully closed and the exhaust bypass valve is fully open so no exhaust gases are flowed to EGRC.

These scenarios are simulated **only** for the first 600 seconds of the FTP cycle. This is because, in 600 seconds the temperatures of the two fluids reach near-steady state values as seen in preliminary simulations.

The total heat generated in the engine due to friction, and transmission and torque converter are shown in Figure 5.1 and Figure 5.2, respectively.

![Figure 5.1 Frictional heat generation within the engine](image-url)
**Observations:**

1. From Figure 5.1 and Figure 5.2, it is seen that the amount of heat generated within the engine and the transmission components decreases with increase in temperature of the respective lubricating fluids.

2. The total heat generated in the engine due to friction, at a constant engine oil temperature of 0°C, is 16.7 MJ, during the first 600 seconds of the FTP cycle considered in these simulations. When the engine oil temperature is increased to a constant of 80°C, all other conditions remaining the same, this reduces to only 6.8 MJ.

3. For the cases where the transmission fluid temperature is changed from 0°C to 80°C, the amount of heat generated in the transmission and torque converter reduces from 3.55 MJ to 1.7 MJ (first 600 seconds of the FTP cycle).
4. This study demonstrates the need to have engine oil and transmission fluid temperature as high as possible, within specified allowable limits, to reduce the amount of extraneous heat generation in the engine and transmission components. In fact, it can be seen in Figure 5.1 that the amount of frictional heat generated in the engine due to friction when engine oil temperature is held constant at 65°C, is 50% less than what it is when the engine oil temperature is held at 10°C. Similarly, in Figure 5.2, approximately 50% decrease in extraneous heat generation in the transmission components is seen at an operating temperature of 65°C versus 0°C.

5. The amount of heat generated in the transmission and torque converter is less than what it is in the engine due to friction for the same lubricant fluid temperature, in every case.

6. The amount of heat generated in the engine and transmission is considered to be directly correlated with inefficiencies within those respective components. Inefficiencies, leading to internal heat generation, cause destruction of Exergy i.e. reducing the ability to convert heat energy into work, thereby directly penalizing fuel economy [5.4]

7. Support for this conclusion is seen in Figure 5.3 and Figure 5.4 which show the normalized fuel consumption for all cases in both scenarios. It is seen that the penalty in fuel consumption is approximately 11.2 % when the engine oil temperature is held constant at 0°C relative to when it is held at 80°C. The corresponding penalty in fuel consumption when transmission oil temperature is held constant at 0°C, relative the reference at 80°C, is seen to be 17.9%
8. The reduction in the amount of fuel consumed increases following the law of marginal returns when moving from lower to higher oil temperatures i.e. from 65°C to 10°C the fuel consumption penalty is 7.3% but from 80°C to 10°C it is 8.9%

![Normalized fuel consumption for different engine oil temperatures](image)

Figure 5.3 Normalized fuel consumption for different engine oil temperatures

9. As it is seen that the magnitude of heat generated in the engine is higher than what it is in the transmission and torque converter, it helps to explain the reason why the engine oil temperature, in general, rises more rapidly that transmission oil temperature
10. It is seen that the fuel economy penalty when the transmission fluid temperature is held at 0°C is higher than when the engine oil temperature is held at the same temperature. This result may be counter intuitive based on the magnitude of heat generation values for the respective cases seen in Figure 5.1 and Figure 5.2 i.e. it would be expected that if the amount of friction heat generated in the engine components is higher than what it is in the transmission components, the fuel economy penalty would also be higher due to that, which is not true based on the case studies considered here.

**REMARK 1:** The above analysis suggests that, attaining higher transmission fluid temperatures rapidly is more important than attaining higher engine oil temperature when aiming at reducing vehicle fuel consumption.
5.1.2. Case studies: Effect of different open-loop actuation strategies on fluid warm-up

A design of experiment is formulated to see the effect of the three thermal-system-actuator positions on the warm-up time of the coolant, engine oil and transmission fluid. After some preliminary simulations, it is seen that, in order to reduce the combinations of valves positions that need to be simulated, the Exhaust Gas Bypass Valve (EBPV here onwards) can be left fully closed for all scenarios (except baseline). In fact, the only purpose of this valve is to ensure that the E-ratio is not violated. E-ratio depends on the temperature of the coolant and exhaust gases entering the EGRC, and on the mass flow rates of coolant and exhaust. Out of these variables, the only control possible is on the mass flow rates of both the working fluids. By having the EBPV fully closed, maximum heat exchange is achieved via the EGRC which results in faster fluid warm-up, while the E-ratio constraint can be respected by ensuring that there is always enough coolant flow through the EGR cooler.

During preliminary analysis, the opening for the Exhaust-Gas Cooler Coolant Valve (EGRCC) is varied to find the minimum value for which the E-ratio constraint is not violated. It is found that, with an EGRCC opening of 0.2, the E-ratio threshold is barely reached. Therefore, the open-loop control space chosen for the EGRCC is [0.2, 1]. The three way valve (3WV), like the EGRCC, is a ball valve. No appreciable difference in the flow rate of coolant is seen in the larger angular openings due to lumped pressure loss coefficients being similar at larger valve openings, as seen in Figure 3.31. The three way valve positions are chosen such that various coolant flow rates through the TOH are
represented. Hence, a condition when there is no coolant flow to the TOH (3WV opening of 0%), a condition when there is maximum coolant flow to the TOH (3WV opening of 100%) as well as two intermediate positions are chosen for this analysis. Therefore, for the purposes of limiting the dimension of analysis to a reasonable size while ensuring that necessary and sufficient operating conditions are captured, four openings each are selected for the EGRCC and the 3WV.

The complete DOE consisting of all the combinations of valve positions is shown in Table 5.2. Each of these conditions was simulated for the first 1000 seconds of the FTP cycle.

<table>
<thead>
<tr>
<th>Scenario (S)</th>
<th>0</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
</tr>
</thead>
<tbody>
<tr>
<td>EBPV position</td>
<td>1</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>3WV position</td>
<td>0</td>
<td>1</td>
<td>0.6</td>
<td>0.3</td>
<td>0</td>
<td>1</td>
<td>0.6</td>
<td>0.3</td>
<td>0</td>
</tr>
<tr>
<td>EGRCC position</td>
<td>0</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>0.45</td>
<td>0.45</td>
<td>0.45</td>
<td>0.45</td>
</tr>
</tbody>
</table>

Therefore, the DoE consists of 17 simulations numbered 0 through 16. Scenario 0 is when EBPV is fully open, the EGRCC is fully closed, and the 3WV is such that no coolant flows through the transmission oil heater. This scenario is equivalent to the case where the conventional thermal management system is used but with a thermostat with a higher temperature set-point. This is here onwards referred to as the baseline case.
The warm-up trends for the three fluids are compared based on time taken to reach two different sets of temperature. The evolution of the three fluid temperatures, for all 17 scenarios, is included in Appendix A for reference. These two temperature set-points, named T-set-A and T-set-B, are listed in the table below.

<table>
<thead>
<tr>
<th>Name of set</th>
<th>Coolant [°C]</th>
<th>Engine Oil [°C]</th>
<th>Transmission Oil [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>T-set-A</td>
<td>90</td>
<td>60</td>
<td>50</td>
</tr>
<tr>
<td>T-set-B</td>
<td>105</td>
<td>95</td>
<td>80</td>
</tr>
</tbody>
</table>

The E-ratio for all the scenarios is seen in Figure 5.5. It is seen that for scenario 13, the E-ratio constraint is barely violated. However, this scenario is still considered for all the following analysis as this result is significant as it helps observe the trend in E-ratio and track the best scenario for fluid warm-up that does not result in the E-ratio constraint violation.
The trends in time taken warm-up to T-set-A for the three fluids are seen in the left part of Figure 5.6 for all of the simulated scenarios. The right side of this figure shows the percentage increase in warm-up times for each scenario with respect to the scenario for which the warm-up of a given fluid is minimum.

For conditions when the TOH is not used, the time taken by the transmission oil to reach 50°C is ~600 seconds; 240 seconds more than the best case scenario (scenario 13) and about 200 seconds more than all other scenarios. The time taken by the transmission oil to reach this temperature decreases as the EGRCC opening decreases, for a fixed 3WV position. On the other hand, if the 3WV opening is increased, for a fixed EGRCC position, the time for trans. oil warm-up decreases.

A reduction in warm-up time for the coolant and engine oil is seen for decreasing EGRCC openings at a fixed 3WV position. On the other hand, if the EGRCC position in
held constant, the warm-up time for the coolant and engine oil is improved for lower openings of the 3WV.

Figure 5.6 T-set-A: Actual fluid warm-up times and percentage difference relative to the minimum warm-up time

Figure 5.7 T-set-B: Actual fluid warm-up times and percentage difference relative to the minimum warm-up time

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Figure 5.7 shows a similar trend when comparing the times required for the three fluids to warm-up to T-set-B. However, to reach the higher temperatures defined by T-set-B, the effect of larger EGRCC opening is more detrimental compared to what it is to reach T-set-A. This is to say that, the % increase in warm-up time seen in the scenarios 1-4 for coolant when compared to the “best” warm-up case (in scenarios 13-16) is much large for reaching T-set-B than T-set-A. The corresponding engine oil warm-up is also seen to be slower.

No coolant flow through the TOH, seen in scenarios 4, 8, 12 and 16 results in very slow warm-up of the transmission oil but does not offer a large advantage for engine oil and coolant warm-up. However, when the 3WV is opened to flow coolant through the TOH even by 30%, considerably lesser time is taken by the transmission oil in comparison with 0% 3WV opening, without much penalty on the engine oil and coolant warm-up times.

Scenarios 13, 14 and 15 represent the same 3WV opening percentages as scenarios 9, 10, and 11, respectively, but for a smaller EGRCC opening. For scenarios 13, 14 and 15, it is seen that the E-ratio for the EGRC is in the range of 0.33 to 0.42 (Figure 5.5); scenario 13 violates the E-ratio. The warm-up times in scenario 9, 10 and 11 are comparable to scenario 13, 14 and 15, respectively, but the E-ratio is not violated.

**REMARK 2:** Analysis suggests that, an EGRCC opening between 20-35%, a fully closed EBPV, and 3WV opening between 30-100%, will be ideal during the warm-up phase for rapid warm-up of engine and powertrain fluids while respecting the E-ratio constraint.
The warm-up times for coolant, engine oil and transmission oil to reach $T$-set-A and $T$-set-B shown in Figure 5.6 and Figure 5.7 are reported in the tables below. The scenarios resulting in minimum warm-up times are highlighted.

Table 5.4 Time required to warm up to temperature set-point 1

<table>
<thead>
<tr>
<th>Scenario</th>
<th>Time taken to reach the respective set-point (seconds)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$T_{clnt} = 90^\circ C$</td>
</tr>
<tr>
<td>0</td>
<td>234.4</td>
</tr>
<tr>
<td>1</td>
<td>240.8</td>
</tr>
<tr>
<td>2</td>
<td>241.9</td>
</tr>
<tr>
<td>3</td>
<td>244.5</td>
</tr>
<tr>
<td>4</td>
<td>227.6</td>
</tr>
<tr>
<td>5</td>
<td>230.1</td>
</tr>
<tr>
<td>6</td>
<td>231.1</td>
</tr>
<tr>
<td>7</td>
<td>232.6</td>
</tr>
<tr>
<td>8</td>
<td>215.4</td>
</tr>
<tr>
<td>9</td>
<td>224.6</td>
</tr>
<tr>
<td>10</td>
<td>225.5</td>
</tr>
<tr>
<td>11</td>
<td>227.1</td>
</tr>
<tr>
<td>12</td>
<td>210.3</td>
</tr>
<tr>
<td>13</td>
<td>215.4</td>
</tr>
<tr>
<td>14</td>
<td>216.0</td>
</tr>
<tr>
<td>15</td>
<td>216.5</td>
</tr>
<tr>
<td>16</td>
<td>204.9</td>
</tr>
</tbody>
</table>
In order to quantify the effect of different warm-up times seen in the scenarios simulated, three performance variables, namely the average transmission efficiency, average FMEP, and cumulative fuel consumption are computed for the different scenarios and compared. In Figure 5.8, the percentage difference of average FMEP, average transmission efficiency and cumulative fuel consumption for each scenario is shown with respect to the best values achieved in these scenarios.

Scenario 16 results in the least FMEP but high cumulative fuel consumption and low transmission efficiency, while scenarios 13 and 14 results in highest transmission efficiency and least fuel consumption but higher FMEP. The FMEP in scenarios 13 and 14 is <1% more than the FMEP in scenario 16, which is negligible. It is seen that

<table>
<thead>
<tr>
<th>Scenario</th>
<th>Time taken to reach the respective set-point (seconds)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>( T_{\text{clnt}} = 105^\circ\text{C} )</td>
</tr>
<tr>
<td>0</td>
<td>460.7</td>
</tr>
<tr>
<td>1</td>
<td>383.1</td>
</tr>
<tr>
<td>2</td>
<td>384.6</td>
</tr>
<tr>
<td>3</td>
<td>410.6</td>
</tr>
<tr>
<td>4</td>
<td>287.9</td>
</tr>
<tr>
<td>5</td>
<td>289.9</td>
</tr>
<tr>
<td>6</td>
<td>294.4</td>
</tr>
<tr>
<td>7</td>
<td>303.5</td>
</tr>
<tr>
<td>8</td>
<td>256.6</td>
</tr>
<tr>
<td>9</td>
<td>278.5</td>
</tr>
<tr>
<td>10</td>
<td>280.6</td>
</tr>
<tr>
<td>11</td>
<td>283.3</td>
</tr>
<tr>
<td>12</td>
<td>247.3</td>
</tr>
<tr>
<td>13</td>
<td>259.9</td>
</tr>
<tr>
<td>14</td>
<td>260.8</td>
</tr>
<tr>
<td>15</td>
<td>261.3</td>
</tr>
<tr>
<td>16</td>
<td>244.2</td>
</tr>
</tbody>
</table>

Table 5.5 Time required to warm up to temperature set-point 1
scenarios 9-10-11 and 13-14-15 are similar in all the three attributes/metrics, respectively. These scenarios seem to offer the best compromise between the three attributes considered here.

Also, as noted in REMARK 1 in the first analysis, higher transmission efficiency is more important than FMEP in order to reduce cumulative fuel consumption.

Therefore, based on the analysis conducted hereunto until this point, scenario 13 and 14 appear to represent the set of most desirable valve positions.

In order to better investigate the trade-offs between: a) low FMEP, and b) higher transmission efficiency and low fuel consumption, a Pareto analysis of the results is carried out in the next section.
5.1.3. Interpretation of Results using Pareto Analysis Method

In this section, the trade-off between the different warm-up times for the scenario outlined in Table 5.2 is analyzed using Pareto analysis.

5.1.3.1 What is Pareto Analysis?

Pareto analysis is a simple yet very powerful tool to study the trade-offs in an optimization problem where multiple performance variables are considered. Pareto analysis is useful to graphically represent the trade-off between at least two performance variables in a system. This allows for identifying the Pareto front i.e. the locus of conditions for which when improving one performance variable makes at least one another performance variable worse.

Figure 5.9 Illustration of a Pareto front in the Pareto analysis methodology [5.2]
A point belongs to the Pareto-efficient set if there are no other points that can improve one of the objectives without degrading at least one of the others [5.2]. Through this analysis, it is intended to analyze how each actuator affects the system performance in terms of warm-up time. Since the EBPV is fixed at fully closed, the effect of position of EGRCC and 3WV is considered and trade-offs between the warm-up of coolant, engine oil and transmission oil are analyzed.

5.1.3.2 Objective Function in the Current Analysis

To study the effect of various valve positions and quantify this effect in terms of a quantity that can be compared between different scenarios, warm-up times are normalized. Consider the following equation,

\[ WTR_{\text{fluid},S} = \frac{\min(t_{\text{fluid}})}{t_{\text{fluid},S}} \]

Equation 5.1

\( WTR_{\text{fluid},S} \) is defined as the ratio of the least time required by a fluid to reach a temperature set-point T-set-A or T-set-B \( \left[ \min(t_{\text{fluid}}) \right] \) amongst all scenarios, to the time required by that same fluid to reach the same temperature \( t_{\text{fluid}} \) for one set of thermal-system-valve positions \( S \), where \( S \) is one of the scenarios in Table 5.2. Therefore, for each fluid, the maximum value that \( WTR \) can attain is 1, which represents the set of valve-positions for which the fluid takes the minimum time to warm-up to the prescribed operating temperature.

5.1.3.3 Pareto interpretation

Trade-offs in coolant and engine oil warm-up to T-set-A:
In Figure 5.10, a Pareto chart comparing the WTRs for the coolant and engine oil warm-up to T-set-A is shown. It is observed that the scenarios where coolant warm-up is fast, the engine oil warm-up is fast as well. This is because, when the coolant temperature increases, the engine oil temperature also increases due to additional heat exchange via the engine oil cooler.

![Pareto chart showing WTRs for coolant and engine oil warm-up](image)

Figure 5.10 Pareto: Coolant and engine oil warm-up time trade off (coolant - 90°C, engine oil - 60°C)

Scenario 16 results in the best warm-up time for the coolant and the engine oil. This is when the EGRCC is opened 20% while the flow of coolant to TOH is stopped. Therefore,
majority of the heat that the coolant gains at the exhaust gas cooler is transferred to the engine oil.

In general, faster coolant and engine oil warm-up rates are seen when there is no flow of coolant to the TOH (scenarios 4, 8, 12 and 16). From scenarios 4-8-12-16, the EGRCC opening decreases and the coolant and engine oil warm-up faster, for a fixed 3WV position. This is because, due to progressively lower flow of coolant to the EGRC, the temperature rise across the EGRC is relatively larger in each case and therefore the coolant and engine oil warm-up is also progressively slower.

For a fixed EGRCC position, it is seen that the coolant and engine oil warm-up times get better as the 3WV opening is decreased. This is because, as the 3WV opening decreases, lesser coolant is flowed to the TOH which means that more heat is contained within the coolant and is transferred to only the engine oil, instead of being transferred to the transmission oil and the engine oil.

**Trade-offs in coolant and engine oil warm-up to T-set-B:**

A clearer picture is seen with respect to the trade-offs in warm-up times to T-set-B. Each of the scenarios circled in Figure 5.11 represents one opening of the EGRCC valve and varying 3WV positions, excluding scenarios where the 3WV is fully closed. It can be observed that, for a fixed 3WV opening, the lower the EGRCC opening, the better is the warm-up time for both coolant and engine oil. However, for a fixed EGRCC opening, the trade-off in warm-up of coolant or engine oil is seen for different percentage openings of the three way valve. For lower three way valve openings, the coolant warm-up is slower
but the engine oil warms-up fast while for higher 3WV openings, the coolant warm-up is faster but the engine oil warm-up is slower. The best coolant and engine oil warm-up, however, is observed when the 3WV is completely closed i.e. when there is no flow of coolant to the TOH.

Figure 5.11 Pareto: Coolant and engine oil warm-up time trade off (coolant - 105°C, engine oil - 95°C)
Trade-offs in engine oil and transmission oil warm-up to T-set-A:

In Figure 5.12, the WTRs for the engine oil and transmission oil to reach T-set-A are shown. As seen in this figure, scenarios 4, 8, 12 and 16 result in the slowest transmission oil warm-up as there is no coolant flow to the TOH because the 3WV is completely closed. Scenarios 1, 5, 9 and 13, where the EGRCC valve is open 100%, but the 3WV opening is varied, result in progressively faster transmission oil warm-up times as the 3WV opening towards the TOH is increased. Also, it is seen that the variation in WTR for the engine oil is smaller when the position of the 3WV is varied for a given EGRCC opening. The best WTRs for the engine are observed for when the 3WV is fully closed.
Therefore, it may be concluded that the best scenario for transmission and engine oil warm-up, considering the underlying trade-offs due to distribution of coolant flow amongst the two respective heat exchangers, is scenarios 13. But scenario 14 is more desirable as in this case the E-ratio is not violated.

**Trade-offs in engine oil and transmission oil warm-up to T-set-B:**

In Figure 5.13, a Pareto comparison of WTRs for the engine oil and transmission oil to reach T-set-B is shown. It is seen that as the EGRCC opening is decreased for a fixed 3WV position, the WTR, the function that is to be maximized, increases for both the engine oil and transmission oil, except for 3WV opening of 0%. Amongst all the simulated scenarios, therefore, the Pareto-efficient set consists of scenarios 13, 14, 15 and 16 when considering the trade-offs between engine oil and transmission oil warm-up times.

The highest WTR for the transmission oil is achieved with the set of valve positions in scenario 13 while scenarios 16 yields the best WTR for the engine oil. However, the marginal improvement in engine oil WTR (~7%) is a result of a large penalty (~55%) on the WTR for the transmission oil when moving from scenario 13 to 16.

Since in scenario 13 the E-ratio is violated, the opening of the EGRCC for which best warm-up times for both engine oil and transmission oil can be achieved is inferred to be between 20-35%, for a 3WV opening of 30-100%, the EBPV being fully closed.
Trade-offs in coolant and transmission oil warm-up to T-set-A:

The WTRs for the coolant and transmission oil to reach T-set-A are shown in Figure 5.14. On observing points on the extreme left of this figure (scenarios 4, 8, 12, and 16) it is seen that, the coolant warms up rapidly as the EGRCC opening is decreased but the transmission oil does not benefit from the rapid coolant warm-up when 3WV opening is 0%. However, when coolant is routed through the TOH by opening the 3WV to only 30%, for the case when EGRCC opening is 20%, the transmission oil is seen to warm-up much faster thereby increasing the WTR for the transmission oil from 0.6 to 0.95. The larger is the opening of the 3WV, the better is the transmission oil WTR.

Consider a fixed EGRCC opening, say 35%. For this EGRC opening, the coolant WTR is 0.95 for a 3WV opening of 0% (scenario 8). As the 3WV opening is increased to 30%,
the coolant WTR decreases to 0.88 (scenario 7). But as the 3WV is opened further, the coolant WTR in fact increases to slightly higher values (scenario 6 and 5).

A similar trend is observed when WTRs are compared to reach higher coolant and transmission oil temperature set-point i.e. T-set-B, as seen in Figure 5.15.

![Figure 5.15 Pareto: Coolant and transmission oil warm-up time trade off (Coolant - 90°C, transmission oil - 50°C)](image)

**Trade-offs in coolant and transmission oil warm-up to T-set-B:**

Therefore, from the previous two figures, it can be inferred that there is a trade-off between coolant and transmission oil WTRs for 3WV openings of less than 30%. Also, lower the EGRCC opening, the better is the WTRs for both fluids.
Following is a summary of the Pareto analysis:

- Engine oil warm-up is dependent on the coolant warm-up due to the thermal interactions via the engine oil cooler.

- When comparing the WTRs for the engine oil and transmission oil at the higher temperature set-points (Figure 5.13), there is a trade-off between their warm-up times; the warm-up of one fluid is penalized to improve the warm-up of the other, except for when the 3WV is fully closed. When the WTR for the engine oil and transmission oil at the lower temperature set-points are compared (Figure 5.12), however, the effect on WTR for the engine oil is very small (less than 1%) due to different 3WV position (except for when the 3WV is fully closed) for a fixed
EGRCC position, but the transmission oil warms-up much faster for increasing openings of the 3WV

- The scenarios that results in rapid warm-up of all three fluids, when considering the primary trade-offs, are 13, 14 and 15. This should be supplemented by saying that in scenario 13, the E-ratio constraint for the EGRC is violated, but not by much. Scenarios 9, 10 and 11 are also reasonable first solutions for rapid warm-up, when considering fixed valve positions. Therefore, the best conditions for which the trade-offs for all three fluid warm-up are minimized can be inferred to lie between EGRCC opening of 20-35%, 3WV opening of greater than 30%, and a fully closed EBPV.

5.1.4. **Evaluation of Simulated Scenarios w.r.t. Baseline Scenario**

The purpose of this study is to compare the results from the use of an advanced thermal management system to a baseline case without the use of TMS. In order to see the improvement in warm-up times for the three fluids with respect to the baseline condition i.e. scenario 0, the percentage increase in warm-up times of the three fluids was calculated for each of the scenarios in Table 5.2 taking scenario 0 to be the reference; this is shown in Figure 5.16.

It is seen that for scenarios 1 through 7, the time required by the engine oil to warm-up to 60°C is more than what it is for baseline, therefore, these scenarios represent valve positions that are not good with respect to warm-up times. Also, in scenarios 9, 10 and 11, the warm-up time for engine oil is perceivably the same as that for the baseline case, but the warm-up time for coolant (to reach 90°C) is ~4% faster and the warm-up times for
transmission oil (to reach 50°C) are about 30-40% faster, varying due to different 3WV opening.

Scenarios 13, 14 and 15 are seen to exhibit much faster warm-up times than the baseline for all three fluids. The decrease in warm-up times for coolant, engine oil and transmission oil is 8%, 3% and 35-39%, respectively, in each scenario from 13 through 15.

Figure 5.16 Time required to reach temperature set-points 1 w.r.t. baseline
In Figure 5.17, the percentage differences in time required for the coolant, engine oil and transmission oil to warm up to T-set-B, are shown with respect to the baseline scenario 0. Trends similar to those seen in the previous figure can be observed in this figure also. The percentage improvement in time w.r.t. baseline case is higher in magnitude when comparing the time reaching the temperature set-point 2. For scenarios 9-10-11, the warm-up times for the engine oil are reduced by 1-3%, for the coolant by ~40%, while those for transmission oil are reduced by 48-54%.

For scenarios 13-14-15, the improvement in warm-up time of the engine oil is 4-7%, while the improvement in coolant warm-up times is 43% . The improvement in transmission oil warm-up times is comparable to that for scenarios 9-10-11 i.e. 50-55% faster than the baseline.

In Figure 5.18, it is seen, for the baseline case, the average FMEP over the first 1000 seconds of the FTP cycle is 0.521 bar. Using this value as reference, the average FMEP,
over the same duration, decreases by 1.5% for scenario 16. Scenario 16 results in lowest FMEP and also fastest engine oil warm-up. Since the engine oil warm-up is relatively similar for all of the scenarios, the decrease in FMEP is also similar with it being within -1.8% and +1.5% amongst all scenarios.

Figure 5.18 Comparison of metrics w.r.t. the baseline scenario

The average transmission efficiency is 65.3% for the baseline case when there is no flow to the TOH while the highest average transmission efficiency, approximately 12% higher, is seen for scenario 13, when 3WV routed maximum possible coolant to the TOH. The baseline cumulative fuel consumption is 930 grams while for scenario 13 it is 898 grams; a fuel consumption reduction of 3.4%.
It is seen that, as EGRCC opening is decreased, holding the 3WV position fixed, the FMEP and fuel consumption decrease while the transmission efficiency increases, except for when the 3WV is fully closed. (scenario 9, 10, 11 and scenario 13, 14 and 15).

When the 3WV position changes from being completely closed to being only 30% open (scenario 4 to 3, 8 to 7, 12 to 11 and 16 to 15), for a fixed EGRC position, the average FMEP increases by not more than 2% compared to the baseline. However, the % improvement in transmission efficiency and cumulative fuel consumption is significant.

5.1.5. Concluding Remarks of Analysis

From the first part of the analysis conducted in this chapter, effect of engine oil and transmission oil temperatures on vehicle fuel consumption and extraneous heat generation in the engine and transmission components are investigated. Based on the results, it is concluded that a larger improvement in fuel economy is seen when transmission fluid warm-up is prioritized over engine oil warm-up. The heat generation in the engine due to friction is larger in magnitude than what it is in the transmission but the transmission inefficiencies play a bigger role in fuel consumption of the vehicle. This is because the losses due friction heat generation in the engine are a small percentage of the total work. On the other hand, inefficiency in the transmission components has a direct impact because the inefficiency is in series to the path of energy transfer.

Through the second step of this analysis, first generation for a genetic algorithm optimization is proposed. It is seen that the EGRCC valve opening should be kept as small as possible, without violating the E-ratio constraint. The ideal range of EGRCC operation is seen to be between 20% and 35% opening. The specified range is for the
case when the EBPV is fully closed i.e. all of the engine exhaust flows through the EGRC; this enables maximum heat transfer as well as observance of the E-ratio being below the specified limit of 0.4. The range for the 3WV, seen as reasonable through this analysis, is 30-100% of opening.
Section 5.2. References


Chapter 6: Conclusions and Future Work

Section 6.1. Conclusions

In the first part of this work, motivation for advanced engine and powertrain research was explained. It was concluded that a great deal of research, in this era, is focused on reduction of fuel consumption of vehicle. This direction of research, stimulated by many factors such as more stringent governmental legislations, increasing oil prices, and the need to reduce the rate of depletion of fossil fuels, has led researchers to develop technologies that enable reduction of fuel consumption. One such enabler is the technological advancement that is broadly classified as thermal management systems. The goal of such systems is to rapidly warm-up engine and powertrain fluids in order to improve the engine and powertrain operating efficiencies.

Through this work, a model for such a thermal management system was developed. Supplementary to the thermal system model, an engine model was also developed. The model development process, for the thermal system as well as the engine, was explained starting with model formulation, calibration, and finally model validation. The goal of developing this model was to mimic the low frequency energy and power transfer dynamics within the engine and powertrain as well as to capture thermal interactions in
the heat exchangers, engine and the ambient. The model was aptly named as the Vehicle Energy Simulator (VES).

The validated VES was then used to perform analysis on the thermal system under consideration, to see effects of engine and transmission oil temperature on performance metrics such as vehicle fuel economy, heat generation in the engine and powertrain components, FMEP and transmission efficiency. Fluid-warm-up time was considered to be a proxy for gauging the effect of fluid temperatures on all of these performance metrics.

The first part of the analysis was used to decouple and observe the effect of engine oil and transmission oil temperatures on vehicle fuel consumption. This study revealed that the warm-up of transmission oil should be prioritized over engine oil warm-up because the transmission oil temperature was seen to have a larger bearing on vehicle fuel consumption. This is to say that rapid warm-up of the transmission oil and higher steady state operating temperatures of the transmission oil, within allowable limits, led to lower vehicle fuel consumption than when the priority for warm-up was given to the engine oil.

The second part of system analysis entailed investigating control strategies for flow control devices used to route coolant to the heat exchangers. Trade-offs in fluid warm-up times were studied for different fixed valve positions for three valves; the exhaust gas bypass valve, the exhaust gas cooler coolant-side valve and the three way valve (refer to Figure 1.2). For some combinations of openings of these three valves, it was seen that the use of advanced TMS, over a conventional thermal management system, resulted in a 3.4% reduction in fuel consumption. This investigation led to proposal of a set of
reasonable valve openings to be used as a first generation for a genetic algorithm optimization. The proposed range for the positions of the three valves was: EGRCC opening of 20-35%, EBPV opening of 100% for all situations, and 3WV valve openings of 30-100%. This proposition was intended to help the GA start from reasonable initial conditions such that the changes of the optimization routine “getting stuck” in a local optimum are reduced.

Section 6.2. Future Work

As for future work, following is a list of planned tasks and some tasks that are recommended:

1. Use genetic algorithm optimization to find the “optimal trajectory” for valve actuation during a given drive cycle, using the first generation proposed here as initial conditions for the GA

2. As mentioned earlier, the three way valve is modeled as a combination of two ball valves. Therefore, at 30% angular opening, the effective area of coolant flow to the TOH, and thereby the coolant flow rate to the TOH, is substantial. At smaller angular openings of the 3WV towards the TOH, the pressure drop characteristics in the circuit may allow more coolant flow through the cabin heater, which receives coolant flow through the complimentary opening of the three way valve. Here, the coolant looses less heat when the cabin heater is not used. Hence, there may be enough coolant flow to the TOH to warm-up the transmission oil but also more coolant flow to the EOC at a higher temperature, where the engine oil warm-up could be made faster without increasing the warm-up time for the
transmission fluid. Therefore, it is recommended that 3WV openings of less than 30% also be investigated to see the effect on the aforementioned performance metrics.

3. Develop thermal system actuator control strategy for steady state operating conditions that were not in the scope of this work. For steady state operating conditions, investigate control strategies for the radiator cooling fan and electronic thermostat. These were not considered here as these technologies work in an opposing way to the objective of rapid fluid warm-up.

4. Investigate other design approaches for the TMS:
   a. The existing EGRC is somewhat oversized, as can be inferred from the low coolant flow rate required to respect the E-ratio constraint even when 100% exhaust is flowed to the EGRC. Therefore, it is recommended to investigate the possibility of positioning the EGRC closer to the outlet of the catalytic converter. Here, the EGRC would see higher exhaust temperature at the inlet and this could result in a potentially larger amount of energy being recovered without violating the E-ratio as there is potential to flow more coolant through the EGRC by further opening the EGRCC valve.
   b. Investigate the possibility of using a smaller EGRC which is also positioned closer to the outlet of the catalytic converter.
Appendix A

Following figures show the coolant, engine oil and transmission oil temperature evolution for the 17 scenarios simulated (shown in Table 5.2) for the first 1000 seconds of the FTP cycle.
Coolant temperature for all scenarios
Engine oil temperature for all scenarios
Transmission oil temperature for all scenarios
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Chapter 4


**Chapter 5**


