Modeling, Analysis, and Open-Loop Control of an Exhaust Heat Recovery System for Automotive Internal Combustion Engines

THESIS

Presented in Partial Fulfillment of the Requirements for the Degree Master of Science in the Graduate School of The Ohio State University

By

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2011

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Abstract

A zero-dimensional fluid and thermodynamic model of an engine, cooling system, and exhaust system was developed in order to simulate the operation of an advanced thermal management system. The model was calibrated with experimental data where available. The thermal management system modeled in this work employed waste heat recovery to reduce engine, coolant, and lubricating fluid warm-up times and fuel consumption following a cold-start. The model was used to develop a control strategy for two valves in the exhaust system which control the flow of exhaust through an exhaust-to-coolant heat exchanger. The objective of the controller was to minimize coolant warm-up time without violating any of the system constraints. A model-based open-loop controller was developed that was able to reduce warm-up time by nearly 35% on the FTP city drive cycle while respecting the limitations of the system.
Amanda, I wouldn’t have done this without you.
Acknowledgments

I’d like to thank my advisor, Prof. Marcello Canova, for all of the guidance and support he provided during my time at the Ohio State University. Thank you to Dr. Giorgio Rizzoni for welcoming me into the Center for Automotive Research from the start of my graduate school experience. I owe a great debt of gratitude to Prof. Timothy Scott for the wealth of knowledge and assistance that he provided during the course of this work. Drs. Fabio Chiara, Lisa Fiorentini, and Shawn Midlam-Mohler all provided invaluable insights and much needed sanity checks along the way. Thank you, all.
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Chapter 1

INTRODUCTION AND SCOPE OF WORK

Automotive fuel efficiency, which has long been of interest to several governmental agencies, has quickly become, over the past decade, one of the top metrics on which consumers compare potential vehicle purchases. The U.S. Department of Energy reports that between the years 2000 and 2011, the number of consumers listing fuel economy as the most important factor when purchasing a vehicle nearly tripled, growing from 11% to 30%. In fact, in their 2011 survey, fuel economy topped the list of buyers’ priorities [1.1]. New propulsion technologies, such as hybrid electric systems and battery electric systems, have been shown to greatly improve vehicle efficiency. There are, however, trade-offs with each, including the additional costs of electric motors and large battery packs as well as range limitations that are not acceptable to many consumers. Manufacturers and researchers have also responded to the call for increased fuel economy with new engine designs and advanced controls that seek to improve efficiency without sacrificing performance or incurring large new costs. Following the law of diminishing returns, however, the efficiency gains in internal combustion engines that engineers can pursue have become increasingly small or narrow in their scope as engine efficiency has improved over the past century. One area of great
inefficiency in current vehicles that has yet to be fully exploited is the operation of the engine immediately following a cold-start.

For the first several minutes of vehicle operation, the temperatures of the powertrain’s lubricating fluids, namely the engine oil and automatic transmission fluid, are well below their optimal operating points. Because oil viscosities are inversely related to temperature, and frictional losses are directly related to viscosity, this slow lubrication system warm-up period leads to greatly increased fuel consumption. Strategies for reducing the warm-up times of these fluids could provide significant benefits to fuel economy after cold-starts.

In addition to their effects on frictional losses, cold engine temperatures also create concerns for combustion efficiency. Among other concerns, cold engine intake runners and combustion chambers can lead to decreased efficiency due to increased heat transfer to the cylinder walls, incomplete combustion due to poor fuel vaporization, and incomplete combustion due to flame quenching [1.2]. Optimizing the coolant system to improve engine warm-up time can also have dramatic effects on vehicle efficiency following a cold-start.

The thrust of this project is to minimize the time that the engine operates with decreased efficiency due to suboptimal fluid temperatures. The proposed method for achieving a reduction in fluid warm-up times is to heat the fluids with energy from the exhaust that would otherwise be lost through the tailpipe. Recovering this energy and distributing it to the necessary fluids, however, requires several additional physical components not found on a typical vehicle. The cooling and exhaust system architectures considered for
this work are shown in Figures 1 and 2, respectively. This proposed coolant system differs from a conventional coolant system in its use of a transmission oil heater, a mechanical/electrical hybrid coolant pump, an electrically-heated thermostat, and additional valves to control coolant routing.

![Proposed Coolant System Architecture](image)

**Figure 1. Proposed Coolant System Architecture**

The exhaust system is nonconventional due to its inclusion of an enlarged exhaust gas recirculation (EGR) cooler, an engine back pressure valve, and an EGR bypass valve.
The engine back pressure valve and EGR bypass valves manage the flow of exhaust gases through the EGR cooler. The EGR cooler recovers heat from the exhaust gases and transfers it into the coolant. The warmed coolant is then able to transfer its heat to the engine oil and automatic transmission fluid through the engine oil cooler and transmission oil heater, respectively.

Two of these new components, the EGR bypass valve and engine back pressure valve, are actuators that introduce additional degrees of freedom to the vehicle’s thermal systems. To maximize the benefits of the exhaust heat recovery system (EHRS), these valves need to be properly controlled. To minimize the time and cost required to design such a control strategy, a computer model has been constructed for the purpose of control development.

In this work, a review of studies concerned with and techniques for improving vehicle efficiency during warm-up is presented. The modeling approaches and physical concepts necessary for developing a model of an engine and its thermal management systems are
discussed. Models for the engine, the exhaust system, and the coolant system under consideration are developed. The calibration of these systems using experimental data is detailed. Lastly, the process for using this model to simulate drive cycles starting from different initial conditions for the purpose of developing a control strategy for the aforementioned valves is reviewed.
1.1 References


The term ‘vehicle thermal management system’ refers to all of the components necessary to maintain the operating temperatures of the vehicle’s subsystems. Managing a vehicle’s thermal systems can be a significant challenge for automotive engineers because it often involves negotiating and maintaining a balance between conflicting interests. In the engine, for instance, the peak gas temperatures are on the order of 2500 $K$ [2.1]. For the engine to efficiently convert this heat into work done on the piston, the heat transfer from the gas to the cylinder walls needs to be minimized. Fundamental heat transfer concepts show that the rate of energy lost to the walls is proportional to the temperature difference between the gas and the wall [2.2]. Therefore, high engine temperatures would reduce this transfer of heat to the cylinder walls, however, several factors preclude an engine from operating this way. Excessively hot combustion chamber temperatures can cause knock and pre-ignition problems. The metals used in engine construction, such as cast iron and aluminum alloys, need to be kept below temperatures of roughly 400°C and 300°C, respectively, to avoid fatigue cracking due to thermal stresses. Cylinder wall temperatures higher than 180°C can cause coking of the engine lubricating oil [2.1].
A colder engine, which might avoid these heat-related issues, will have much greater heat transfer losses to the cold walls and greater frictional losses due to higher oil viscosities. The decreased combustion efficiency due to flame quenching and poor fuel vaporization leads to reduced fuel economy and increased hydrocarbon emissions. The vehicle thermal management system is tasked with maintaining operating conditions that minimize the potential for negative side effects while maximizing vehicle efficiency. For this reason, and countless others, the behavior, design and control of vehicle thermal systems have been rich fields of study for the past century. A vehicle’s thermal systems can be broken into three distinct units: the cooling system, the engine lubrication system, and the transmission lubrication system. Each has its own set of objectives and limitations. This chapter describes each and explains the importance of maintaining the operating conditions that they do. It presents a review of the studies on the topic of warm-up time and the methods that have been attempted to improve the same. The common methods for modeling thermal management systems are also reviewed.

2.1 The Cooling System

As previously mentioned, there are numerous reasons why an engine cannot be left to operate at a temperature dictated by the heat generated during combustion. To function properly, engines require that excess heat from combustion and friction be removed on a continual basis. To this end, engines are fitted with cooling systems that are responsible for absorbing heat in the engine and carrying it elsewhere. In a conventional cooling system, the major elements that perform this job include a belt-driven coolant pump, a
radiator, the coolant flow paths through the engine, a wax thermostat, an engine-driven cooling fan, a cabin heater core and all of the piping and auxiliary components required to route the coolant through the system. Forced by the pump, the coolant flows through a series of passages in the engine block and head to extract heat from the metal by convection. Depending on the temperature of the coolant, the thermostat allows a portion of the coolant to flow through the radiator. Air flow through the radiator removes a some of the heat from the coolant. To supplement the natural air flow due to driving, the fan forces air flow through the radiator and over the surface of the block. From the radiator, the coolant flows back to the coolant pump. Heat is also extracted from the coolant by the cabin heater core to warm the passenger compartment.

Figure 3. Conventional Cooling System, [2.3]
Heywood explains that there are several sources of heat transfer in the engine that, together, determine the full cooling demand on the engine. Of these, the most significant are the “heat transferred to the combustion chamber walls from the gases in the cylinder, heat transferred to the exhaust valve and port in the exhaust process, and a substantial fraction of the friction work.” [2.1] During engine warm-up, much of the heat transferred to the engine goes to raising the temperature of the block and the head. In a fully-warmed engine, however, all of the fuel energy that does not get converted into useful work or exit through the exhaust system needs to be dissipated to keep the engine at a constant operating temperature. A small portion of this energy is converted to heat in the oil by friction in the crankcase, however, the vast majority of this must be carried away by the cooling system.

The portion of the fuel energy that is rejected to the coolant is dependent upon several factors, namely the engine speed and the load on the engine, as shown in Figure 4. Although the percentage of heat rejected to the coolant decreases slightly as speed and load increase, the total power absorbed by the coolant increases nearly proportionally to the engine’s brake power.
The cooling system needs to be sized to handle the cooling demand for all engine operating points. Brake power increases proportionally with both the engine speed and the load on the engine. The flow rate of coolant, when driven by a conventional mechanical coolant pump, increases with the engine speed but not with the engine load. Therefore, the cooling capacity for a given engine speed is designed to cool the engine at full load. This forces the cooling system to use a large radiator and a pump with a high flow rate. The engine rarely operates at full load, though, so the cooling system typically removes more heat from the engine than is necessary, thus, overcooling the engine.

This is especially troubling during cold-starts. The large thermal inertia associated with such a cooling system can greatly impede the engine’s warm-up time and, therefore, its efficiency during the first several minutes of a trip. Pang [2.4] observes that, “the introduction of cold-start legislated, drive-cycle tests has changed the significance of
engine-cooling systems (ECSs), due to their impact on test results. Rapid engine warm-up is critical in attaining low fuel consumption and emission readings in drive-cycle tests.”

2.1.1 Importance of Operating Temperatures During Warm-up

The focus of this work is on vehicle efficiency following a cold-start. The coolant temperature and the time required to warm the coolant have strong influences on engine efficiency. Quantifying what Pang has stated about cold-start drive cycles, Choukroun [2.5] tested a novel coolant routing approach that improved coolant warm-up time to 100°C on a 20 minute drive cycle by 220 seconds, or 40%, compared to a baseline value set by the same vehicle using a conventional coolant routing strategy. This reduction in warm-up time yielded a 2% to 3% improvement in fuel efficiency for the drive cycle.

Figure 5. Coolant Warm-up Time Comparison, [2.5]
Goettler [2.6] was able to reduce the coolant warm-up time of a 1981 Ford Granada by 27.5% by transferring waste heat from the exhaust directly into the coolant. Ambient conditions ranged from 35°F to -15°F. For those ambient conditions, the reductions in fuel consumption over the baseline vehicle during a 7 minute engine stand test were 2.1% and 4.6%, respectively, with all other factors held constant.

Kuze [2.7] investigated the effect on fuel economy of increasing the temperature of the coolant around the intake ports prior to a cold-start for a port injected engine. The author set a baseline fuel consumption value with a conventional engine starting at a uniform 25°C. For the test case, the intake ports were warmed to 40°C by introducing hot coolant before starting the engine. Due largely to improved fuel vaporization, which reduces the need for fuel enrichment, the fuel consumption with a warmed intake port was measured to be 41% lower than the baseline immediately after starting the engine. Hydrocarbon emissions were reduced by nearly 50% relative to the baseline. Although this comparison is an extreme example across a short timescale, it does illustrate the inherent inefficiencies of a cold-start. It is clear from the results of these studies that there is a great opportunity to improve vehicle efficiency and emissions by reducing the warm-up time of the engine and cooling system.

2.1.2 Techniques for Improving Coolant Warm-up Time

In pursuit of these gains in fuel efficiency, researchers have studied several different methods for reducing the amount of time required to bring the coolant to its fully-warmed operating temperature.
Allen [2.8] increased the amount of controls in the coolant system in order to use heat rejected from the engine as efficiently as possible. In the experimental setup on a 2.0 liter SI engine the wax thermostat was replaced with an electrically controlled four-way valve and the belt-driven coolant pump was replaced by an electric pump. In this configuration, the coolant was able to flow through any combination of three routes: the radiator loop, the bypass loop, and the cabin heater core loop. It was also possible to stop coolant flow entirely during the warm-up phase. During a drive cycle, stopping coolant flow entirely until the coolant in the engine was fully-warmed reduced the warm-up time by 190 seconds, fuel consumption by 3% and hydrocarbon emissions by 17%.

Choukroun [2.5] also increased the level of control in the coolant system. He, too, replaced the mechanical pump with an electric, PWM-controlled pump. The thermostat was replaced by a PWM-controlled butterfly valve. In this test case, a 40% reduction in engine warm-up time and a 2% to 3% improvement in fuel economy were accomplished by stopping flow to the engine entirely during the warm-up phase.

These test cases show the potential for savings based on reduced or no coolant flow during warm-up in experimental conditions. This technique has also been implemented in a mass-produced vehicle’s cooling system. BMW has replaced the mechanically-driven water pump with an electrically-driven unit and the passive wax thermostat with an electrically driven valve on its three liter SI engine. Despite its simplicity relative to other systems reviewed in this chapter, BMW’s system does deliver tangible benefits during warm-up. As with the experimental cases, the coolant is not flowed until the engine is fully-warmed.
The benefits of this system are not limited to cold-start performance, though. The electric pump allows BMW to perfectly tailor the coolant flow rate to all operating conditions. Flowing less coolant through the engine during part-load driving allows the engine to operate at a higher, more efficient temperature. Also, opening the radiator valve fully before flowing coolant through that path reduces coolant pumping losses. The net gain from improved warm-up times and increased steady-state operating temperatures is a 1.5% fuel consumption improvement on the EU drive cycle, [2.9]. This result is not as dramatic as the results obtained in lab tests likely because the BMW system is more conservatively to ensure that dangerous operating conditions are not encountered.

To augment the coolant’s warm-up rate, Andrews [2.10] fit a reverse flow exhaust-to-coolant heat exchanger to the test engine’s exhaust ports. Coolant passes over the hot exhaust port runners, reverses direction, and then crosses them again in this system. In so doing, the coolant is able to recapture a substantial portion of the energy that would have otherwise exited the tailpipe. The direct fuel economy improvement due to faster coolant warm-up is not known because the coolant was then used to rapidly warm the engine oil. The combined effect of the increased warm-up rates for the coolant and the oil produced by this system netted a 14% reduction in fuel consumption on the NEDC drive cycle.

Diehl [2.11] studied the effect that placing an exhaust heat recovery system (EHRS) downstream of the catalyst has on passenger comfort levels. The EHRS tested in this paper consisted of a shell-and-tube heat exchanger on one branch of a flow split in the exhaust pipe. A valve was placed on each branch of the split to give control over the flow rate through the exchanger and the back pressure on the engine (Figure 6).
The study focused on the benefit to cabin heater core temperatures rather than fuel economy, but, from an energy standpoint, the system was able to recover 1.7 MJ of waste heat during the first 13 minutes of the NEDC drive cycle. Roughly translated, this amount of energy is enough to heat 5 liters of coolant by 100°C.

Exhaust-to-coolant heat exchangers have been put into production vehicles, as well. The third generation of Toyota’s popular Prius hybrid electric vehicle also places an exhaust-to-coolant heat exchanger downstream of the catalyst. This system allows the vehicle to recover a substantial amount of waste heat, improving overall engine warm-up time and passenger comfort. The exhaust heat recovery system (EHRS), shown in Figure 7, consists of a coolant jacket around the exhaust pipe, a pair of flow diverters in the exhaust stream, and valves that control coolant and exhaust flow.

Figure 6. Exhaust Heat Recovery System, [2.11]
The system has only one actuator which controls both of the valves in the system. When the actuator closes the exhaust valve, forcing gas to flow through the outer portion of the exchanger, the coolant valve is simultaneously opened, and vice versa. To prevent excessive back pressure on the engine caused by use of the heat exchanger, the exhaust valve can open passively, as well, acting as a pressure relief valve. The coolant valve is unaffected by this occurrence, however, and can only be moved by the actuator. This design is used because in the case that the exhaust valve opens due to back pressure there is still a large amount of exhaust flowing through the outer portion of the exchanger. This requires coolant flow through the exchanger in order to prevent the boiling of stagnant coolant. Based on a 23 minute drive cycle begun from a cold-start, Toyota
reports a fuel economy improvement of 9.2% in addition to greatly increased heat availability to the cabin heater core.

On the second generation model of its Prius hybrid, Toyota implemented a creative system called the Toyota Coolant Heat Storage System (CHSS) in the hopes of reducing fuel consumption and hydrocarbon emissions immediately after a cold-start. When the car is turned off after having been fully-warmed, hot coolant is transferred from the engine to a heat-insulated reservoir. Upon the next engine start, the coolant is returned to the engine and focused on the coolant channels around the intake ports. This method has a pronounced effect on hydrocarbon emissions and fuel consumption for the first few minutes of operation.

Kuze [2.7] discusses the development process for this system. A study was conducted to determine the most effective means for reducing hydrocarbon emissions in a cold engine. The three approaches considered were: 1) heating the fuel before it enters the engine, 2) heating the air in the intake manifold, 3) heating the intake ports. After concluding that heating the intake ports provides the greatest benefit, the thermal capacitance of the ports was examined so that the CHSS could be sized appropriately for the task. The heat-insulated reservoir was then designed to be able retain enough heat to warm the ports to 40°C in less than ten seconds upon starting the car. Coolant temperatures of 70°C after 24 hours and 50°C after 72 hours from an initial temperature of 90°C were used as the design targets.

The Prius is uniquely qualified to utilize this type of system. As Kuze explains, “because it uses a hybrid-electric powertrain, there is no need to start the engine with the
ignition switch, so the engine is started automatically after the pre-heating is completed.”

Toyota has found that this system improves fuel economy by 5%-6% on a 10 minute drive cycle.

These experimental setups and practical applications of advanced thermal management systems, although varied in their components and strategies, all demonstrate the benefits of improved warm-up performance through intelligent use of the engine’s heat.

2.2 The Engine Lubrication System

The engine lubrication system is responsible for minimizing frictional losses in the engine by supplying lubricant to all of the parts of the engine where two surfaces move relative to one another. The primary areas of concern are the bearings in the crankshaft and pistons, the contact between the pistons and the cylinder walls, the camshafts, and the valves, as seen in Figure 8. Due to the high operating pressures and tight tolerances of its design, the engine’s sliding surfaces are under considerable loading. Lubrication serves to lower the coefficient of kinetic friction between parts, thus, reducing the amount of power dissipated to friction. It also helps to carry away some of the heat generated by friction in these locations. The engine oil transfers much of its excess heat to the sump which is then expelled to the ambient air by convection. Some systems are also equipped with an engine oil cooler to maintain the proper oil operating conditions.
2.2.1 The Importance of Operating Temperature During Warm-up

The lubricating oil for the engine’s moving parts is similar, in principle, to the coolant in that as it warms, the engine is able to run more efficiently. The modes by which the engine oil’s state affects the operation of the engine are not as numerous as those of the cooling system, but they do have a significant impact on engine efficiency, nonetheless. To a small extent, the temperature of the lubricating oil affects the engine efficiency because the work required to pump the oil through the engine varies inversely with the oil temperature. The primary influence that lubricating oil temperature has on engine efficiency, however, is related to frictional energy losses from the engine’s moving parts.
As oil temperature increases, the viscosity decays exponentially, thus, improving its ability to lubricate. Allen [2.8] shows the effect oil temperature has on viscosity and, in turn, friction in Figure 9. The test performed was to spin an engine at a constant 2200 RPM with an electric motor and measure the power required to do so. The oil temperature was varied while all other parameters were held constant in order to isolate its influence. Allen found that between 110°C and 121°C, the viscosity of a typical 15W40 oil changes from 11.7cSt to 9.4cSt, or 20%. Although these temperatures are both well within the normal operating range for engine oils, the motor friction (power required to spin the engine at a constant speed with no fuel and no net flywheel torque) was reduced by 6% by raising the temperature of the oil 11°C. This disparity grows dramatically when compared to the much cooler oil temperatures of a cold-start.

![15W-40 Lubrication Oil](image1)

![Effect of Oil Temperature on Motor Friction 1Liter Diesel Engine](image2)

Figure 9. Engine Oil Viscosity and Effect on Friction, [2.2]

Farrant [2.14] modeled the cold-start behavior of a vehicle to better characterize the effect of oil viscosity on fuel consumption. First, the model was calibrated and validated
using experimental data from a vehicle performing a cold-start and completing the prescribed driving cycle. The model was then used to simulate the same cold-start and drive cycle but with the engine oil temperature held constant. With the oil fixed at 25°C, the model predicted a 30% increase in fuel consumption over the baseline case in which the oil warmed up normally. Fixing the oil at 94°C, the fully-warmed operating temperature for this test, the model predicted a 20% decrease in fuel consumption relative to the baseline.

Shayler [2.15] explains the decrease in frictional forces with decreasing viscosity in Equation 2.1, where \( \dot{P}_f \) is the frictional power loss at fully-warmed conditions, \( \nu \) is the fluid’s kinematic viscosity during warm-up, and \( \nu_{90^\circ C} \) is the same property at fully-warmed conditions.

\[
\dot{P}_{wu} = \left( \frac{\nu}{\nu_{90^\circ C}} \right)^{0.24} \dot{P}_f \tag{2.1}
\]

2.2.2 Techniques for Improving Engine Oil Warm-up Time

A method for improved engine oil warm-up performance that garners much interest is waste heat recovery: the practice of capturing and putting towards a good purpose the heat that would otherwise be lost to the environment through the radiator or tailpipe. Kuze [2.7] estimates that 46% of the chemical energy that enters the combustion chamber is eventually lost in one of these two ways. This wasted heat represents a major opportunity for improving vehicle efficiency.
Andrews [2.10] discusses a form of waste heat recovery intended to reduce engine oil warm-up time. He notes that the warm-up time for engine oil is significantly longer than that of coolant. Due to this difference in warm-up times, in a conventional system the coolant reaches its operating temperature and begins to offload waste heat through the radiator while the oil is still well below its optimal point. For this reason, he employed a coolant-to-oil heat exchanger to transfer heat from the coolant into the oil. The addition of this heat exchanger allowed the coolant to offload its excess heat to the oil, instead of the radiator. Over the course of the NEDC drive cycle, this approach produced an 8% reduction in fuel consumption.

Will [2.16] proposed another form of waste heat recovery to reduce frictional losses. In this work, the author used an exhaust-to-oil heat exchanger to direct exhaust heat into the lubricating oil. In the proposed system, engine oil flows through the exchanger downstream of the oil filter before continuing on its standard route through the engine. The exhaust system features a bypass valve downstream of the catalyst that is able to guide exhaust gas to the heat exchanger or allow it to flow through its normal path.
Figure 10. Exhaust-to-Engine Oil Heat Exchanger System, [2.16]

Using this configuration, the author measured a 7% improvement in fuel economy on the NEDC drive cycle. Based on baseline testing, the difference in fuel consumption between an NEDC cycle driven from a cold-start and one started with a fully-warmed engine is 10%-15%. Therefore, the 7% improvement provided by the exhaust-to-oil heat exchanger represents between 50% to 70% of the potential fuel economy gains to be had by minimizing engine warm-up time.

Samhaber [2.17] used a thermodynamic model developed in AVL BOOST to test the possibility of improving engine oil warm-up time by insulating the oil sump. The goal is to stop the unwanted transfer of heat from the oil to ambient during warm-up. This system is fitted with an engine oil cooler to provide adequate oil cooling once it reaches fully-warmed conditions. After completing the NEDC urban drive cycle from an initial
temperature of 0°C, the oil in the insulated sump was roughly 4°C hotter than the oil in the baseline simulation. This technique caused a decrease in fuel consumption of 0.5%. Though some show more successful results than others, all of these strategies for improving engine oil warm-up time show that there is a possibility to achieve tangible fuel economy benefits through the improved use of engine heat to warm the engine lubrication system.

2. 3 The Transmission Lubrication System

Similar to the engine lubrication system, the transmission lubrication system is responsible for supplying oil to all of the moving pieces in the transmission to reduce frictional losses and carry away the heat generated by friction. In the case of an automatic transmission, the transmission fluid is a working fluid, as well, used to transfer power hydraulically in the torque converter and to manipulate clutches and brake bands on the planetary gear sets to implement gear shifts. The torque multiplication mode of the torque converter leads to large viscous losses in the fluid that cause significant heat rise in the portion of the transmission fluid in the torque converter. Some transmissions are equipped with transmission oil coolers to maintain safe operating temperatures.
2.3.1 The Importance of Operating Temperature During Warm-up

The temperature dependence of the transmission oil system’s behavior is very similar to that of the engine oil system: the faster a fully-warmed operating condition can be achieved the better. Figure 12, which was made using an empirical relationship [2.19] based on component data, shows the relationship between temperature and viscosity in automatic transmission fluid.
To quantify the relationship between transmission fluid temperatures and fuel economy, Farrant [2.14] applied the same simulation procedure described in the previous section to the transmission oil. Using a model based on a conventional thermal management system, the author simulated the ECE15 drive cycle with the transmission oil temperature fixed at 25°C and 94°C. Comparing the results to the case in which the transmission was allowed to warm at a normal rate, the simulations showed changes in fuel consumption of 3% and -4%, respectively.

Semel [2.20] experimentally measured the net effect of rapidly warming the transmission fluid. Using a novel oil routing method, the transmission fluid reached its operating temperature in 40% less time than in the baseline set by a conventional transmission oil system: 15 minutes compared to 25 minutes. This resulted in a fuel economy improvement of 2.2% during a 30 minute drive cycle. As with the engine oil, although

Figure 12. Viscosity of Automatic Transmission Fluid
on a smaller scale, quickly warming the transmission oil has a decidedly positive effect on vehicle efficiency.

2.3.2 Techniques for Improving Transmission Oil Warm-up Time

The novel routing method employed by Semel, mentioned in the previous section, was to create a bypass loop around the transmission oil cooler and control the flow of coolant using a thermostatic valve. Cold fluid exiting the transmission was routed through the bypass loop and directly back into the housing, avoiding the oil-to-air transmission oil cooler. This prevented the unnecessary cooling that occurs in a conventional system when cold fluid is flowed through the heat exchanger during a cold-start. This strategy also allowed the transmission to heat a smaller mass of fluid initially, as some of the fluid was occupying the stagnant oil cooler loop, also reducing warm-up time. Once the unit reached its desired operating temperature the thermostatic valve allowed fluid to begin circulating through the cooler to maintain operating conditions. As previously mentioned, this system netted a 2.2% fuel economy increase during a 30 minute drive cycle.

2.4 Modeling Thermal Management Systems

In the interest of studying the benefits and drawbacks to various thermal management system changes such as the addition of new components, different fluid routings, or novel control strategies, engineers often turn to computer simulations for initial validation of an idea. Simulations are ideal for preliminary design and control development because they
allow researchers the opportunity to change component sizing and functionality, environmental conditions, and a host of other parameters in a fraction of the time and at a fraction of the cost of experimental testing. The thermodynamics of an engine and its thermal management systems can be modeled three-dimensionally, one-dimensionally, or zero-dimensionally. The ideal solution varies depending on the intended purpose of the model and the granularity of detail needed.

For a highly-detailed look at the fluid and heat flows within the engine, as well as the thermal stresses caused by these, the three-dimensional applications of computational fluid dynamics (CFD) and finite element analysis (FEA) can be used. This can be accomplished using software packages with three-dimensional modeling environments, such as AVL FIRE or FORTE. A three-dimensional analysis begins by defining the boundaries between the solid volumes of the structure of the engine and the fluid volumes therein using a three-dimensional solid model of the engine. These volumes are then finely discretized into a finite number of nodes arranged in a mesh pattern. During a simulation, the energy balance and motion (in the fluid volumes) of each node is solved for in an iterative manner. The fluid dynamic analysis is able to account for pressure waves in the fluid, swirl patterns, turbulence at the walls, etc. The meshed nodes interact thermally through convection or conduction depending on the materials involved. The thermal and fluid properties at the boundary conditions are used to solve for the heat transfer characteristics between the solid volumes and the fluid volumes. This type of analysis is ideal for evaluating fluid and heat flow events at a fine level of detail and on a
short timescale, for example, the heat transfer to and from the exhaust port at various
points in the engine’s cycle.
Cipollone [2.21] used three-dimensional analysis to study the heat distribution within the
cylinder wall and the corresponding heat transfers from the combustion gases and to the
coolant. This allowed the author to analyze the temperature differences across the
thickness of the wall and along the length of the cylinder at four different cross-sections
spaced 90° apart along the circumference of the cylinder. Figure 13 shows the results of
this analysis. The author was able to show that the wall temperature increases vertically
in the cylinder and decreases radially (from right to left in the figure). The gas-side
cylinder temperature is nearly independent of angular position in the cylinder, however,
the coolant side temperature is greatly affected, likely due to variations in coolant flow at
different points in the cooling jacket. This method of analysis is extremely
computationally burdensome and is generally practical for modeling individual engine
events rather than entire drive cycles.
Compared to three-dimensional analysis, the one-dimensional analysis of fluid systems is much less computationally expensive. Programs such as Flowmaster, AVL BOOST, GT-COOL and AMESim can be used to study the linear flow characteristics of the fluids within an engine, as well as the thermal interactions between fluids and solids. Fluid flow is treated in a one-dimensional manner, meaning that only flow disturbances in the direction of the flow are considered and cross-sectional, or perpendicular, variations in flow are ignored. This simplifies the analysis by estimating the effect of flow losses at the boundaries rather than solving for them explicitly. These types of programs are able to predict fluid flow characteristics with reasonable accuracy based on pipe geometries.
and properties. They are able to account for flow events on small timescales, such as the pressure waves in the intake manifold due to the opening and closing of the intake valves. Again, the flow dynamics are used to solve for the heat transfer rates based on the change in the Reynolds number (Equation 4.42) for the fluid. Lumped thermal capacitances are the standard method for thermal analysis in these programs. As with the three-dimensional programs, the volumes are discretized into a series of nodes, although the level of discretization is several orders of magnitude less in the one-dimensional analysis tools. Also, this discretization is done manually by the modeler rather than automatically by the program.

To test the effect on engine warm-up rates of artificially increasing the exhaust back pressure on the engine, Samhaber [2.17] implemented a vehicle simulator and a one-dimensional gas dynamics software package together, AVL CRUISE and AVL BOOST. The transient thermal behavior of the engine, the cooling system, and the engine lubrication system were modeled. AVL CRUISE was first used to simulate the longitudinal dynamics of the vehicle on a prescribed drive cycle. This system considered aerodynamic loads, vehicle inertial effects, rolling resistance, etc. to calculate the torque and speed demands on the engine. Based on these values, AVL BOOST calculated the flow of gases (air, combustion, and exhaust) through the engine using one-dimensional flow dynamics. A thermal network of 21 lumped capacitances was constructed in the AVL BOOST modeling environment to represent the mass of the engine. The one-dimensional gas dynamics provided the gas velocities within the cylinder that were used to solve for the heat transfer coefficients between the combustion gases and the cylinder
walls, the head, and the pistons. Flow in the cooling system and the engine lubrication system were modeled zero-dimensionally using the continuity and energy conservation equations.

Figure 14. Fluid and Thermal Networks in AVL-BOOST, [2.17]

The flow rates calculated here were used to solve for the heat transfer rates to the coolant and oil. Energy balance equations were used to solve for the coolant and oil temperatures at various positions in the engine. The friction in the engine was calculated based on the temperature of the engine oil and the experimentally determined friction losses at fully-warmed conditions using an approach similar to the Shayler method described earlier in
the chapter [2.15]. This modeling approach proved successful in modeling thermal transients in a cold engine during a several minute long warm-up period.

Chanfreau [2.22] used the one-dimensional fluid dynamics program FLOWMASTER to simulate the operation of a 42V electric coolant pump. Here, too, the thermal masses of the engine were modeled using a network of lumped capacitances in the software’s modeling environment. The model was used to analyze the power required by the electric pump to provide the proper amount of coolant flow compared to the power consumed by a conventional, belt-driven pump.

Vanderslice [2.23] modeled the cooling system for a 4.5 liter diesel engine in Gamma Technology’s GT-COOL environment. The model uses lumped thermal masses, one-dimensional fluid flow and thermal dynamics, and a lookup table for engine heat rejection. The author first modeled the conventional thermal system and validated it against experimental results. He was then able to simulate the effect on cooling system performance of an electric coolant pump, an electric thermostat valve, and a downsized radiator.

These one-dimensional software packages are an efficient way to test the effects of making modifications or additions to the thermal management system. They are still somewhat computationally burdensome, though, and are not ideal for use in high-volume simulation applications, such as control development. For that purpose, zero-dimensional time averaged models built in numerical solver software environments, such as Matlab/Simulink, are often used.
A zero-dimensional time averaged, or mean-value engine model (MVM), is one in which the variations that occur during an engine cycle are ignored and only the net effects of the cycle are described. These models are typically based on system identification, in which experimental data can be used to describe and predict the operation of the system without full understanding of the processes occurring in the system. In the context of an engine, this means, for example, ignoring the complex gas dynamics involved in a combustion event. On the timescale in which the MVM operates, such events are instantaneous and can be expressed as algebraic equations [2.24]. This allows the torque production, the exhaust temperature, and the heat rejection to the coolant to be estimated based on a few measurable inputs.

Flow calculations in MVM’s are typically based on quadratic relationships with pressure drop in which the flow restrictions in the system are represented by calibrated coefficients. This approach is discussed in more detail in the following chapter. As with the one-dimensional models, thermal analysis is performed using a limited number of thermal capacitances to represent the engine and its fluids. Energy balance equations are used to compute the temperature at each node dynamically.

Cipollone [2.25] describes the use of a zero-dimensional MVM in a 2004 paper. The author applied the conservation of energy equation at each thermal node to determine temperatures in the system. Heat rejections from the engine to the coolant and exhaust gases were handled by steady-state maps. The coolant pressure drop through each device was modeled as being proportional to the square of flow rate and, in the case of valves, inversely proportional to the square of valve position. This made it “possible to consider
the complete cooling circuit fed by the pump as an equivalent quadratic pressure drop deriving from series and parallel connections of the various branches.” In knowing the flow characteristics of each individual component based on experimental data, the author was able to quickly rearrange the system and experiment with different flow routings. Cortona [2.26] developed a mean-value model in which “local differences in coolant temperature or in the coolant flow speed are neglected and only the overall system behavior is considered.” In this model, the engine was represented in three thermal nodes: the cylinder, the coolant, and the block. The conservation of energy equation was applied to each of these nodes. The cycle-averaged heat rejection from the combustion gases to the cylinder wall was mapped as a function of torque, engine speed, and wall temperature. This model was used to evaluate, among other things, the change in heat loss through the cylinder walls that can be achieved with the use of a controlled electric pump.

Trapy [2.27] implemented a zero-dimensional model to analyze the effect on lubricant warm-up time of adding a coolant-to-engine oil heat exchanger. Flow-pressure relationships were used for each of the restrictions in the system. All relationships were solved simultaneously by a non-linear mean least square algorithm. The heat transfer and fluid flow rates in the system were considered quasi-steady. New values for the heat transfer rates were solved for each 30 seconds. Initial values for flow rate were used to find the initial pressure drops through each flow element. These pressure drops allowed the system to then solve for the flow rate in the next time step.
These zero-dimensional models are computationally simple enough to run at real-time or faster on most personal computers using an ordinary differential equation solver [2.24]. They are able to quickly predict the major system outputs based on a limited number of exogenous and control inputs. For this reason, these types of models are ideal for use in controls development.

2.5 Control Approaches for Engine Thermal Management Systems

As shown in this chapter, the control of engine thermal management systems comes in many varieties but a few overarching themes are noticed in most of the published work: minimizing the power consumption of the actuators in the system, improving engine operating conditions through enhanced cooling system control, and minimizing waste heat rejection.

Through the steady progression of technology, electrical devices have become smaller, stronger, smarter, and more affordable. Actively controllable electric devices have the potential to replace, in the near future, a number of passive thermal management system components, such as engine-driven pumps and fans and wax thermostats. The newfound control over the system given by these devices allows engineers to optimize the system’s operation in several ways. Firstly, conventional cooling systems are inefficient in most operating conditions due to flow rates that are unnecessarily high. Secondly, the pressure drop through a partially open thermostat is extremely high and requires the pump to work harder than necessary to achieve the desired coolant flow to the radiator. These problems can be solved by opening the thermostat valve fully whenever flow to the radiator is
required and then controlling the pump to give the flow rate necessary for cooling the engine. This strategy can have a dramatic effect on the cooling system’s energy consumption. Simulations conducted by Vanderslice [2.23] predicted an 87% reduction in power draw on the FTP 74 drive cycle using this control strategy.

Another possibility afforded by an actuated cooling system is that of optimizing the engine’s operating temperature for all driving conditions. During periods of high torque production the engine has much greater cooling demands in order to prevent disadvantageous combustion conditions and thermal stresses on the engine. Under more common part-load conditions, however, the engine can operate more efficiently at higher internal temperatures without experiencing these problems. Reducing coolant flow through the engine allows this to happen. Controlling flow through the engine and radiator, or stopping it altogether, permits the engine to reach its desired operating temperature more quickly following a cold-start by not cooling the engine unnecessarily during the warm-up period. BMW has demonstrated the combined benefit of these techniques with a 1.5% increase in fuel economy on the EU drive cycle [2.9].

Each of the previous control strategies achieves its results by replacing conventional, passive components with their actuated counterparts. Waste heat recovery strategies require additional kinds of hardware to be installed in the thermal management system. These types of systems provide the majority of their benefit during the engine warm-up period. One strategy of waste heat recovery systems is to use the coolant, once hot, to warm the engine and transmission lubricating fluids through coolant-to-oil heat exchangers rather than expelling excess heat through the radiator. Another common
strategy is to use an exhaust-to-coolant heat exchanger to rapidly warm the coolant. Toyota has demonstrated a 9.2% fuel economy improvement during a 23 minute drive cycle begun from a cold-start [2.12] with the latter strategy.

A review of literature describing the study and implementation of advanced thermal management systems has been conducted. Several methods for improving vehicle efficiency have been investigated and tested by academics and industry alike. Methods for modeling these systems have been explored, also. The common theme in all of the documents reviewed is that conventional thermal management systems are not properly optimized for fuel economy during warm-up or fully-warmed conditions and this presents many opportunities for substantial gains in efficiency to be made.
2.6 References


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Chapter 3

BASIC PRINCIPLES OF MODELING AND CONTROL

Accurately simulating an exhaust heat recovery system requires careful integration of several types of system models. Those include: an engine model, an exhaust system model, a cooling system model, and models for each of the heat exchangers in the EHRS. A choice needs to be made about how best to represent each of these systems as there are multiple options for each.

3.1 Cycle-Resolved (Mean-Value) Engine Models

Two types of engine models were considered for this work: a crank angle resolved model (CAR) and a cycle-resolved mean-value model (MVM). The benefits and drawbacks of each were evaluated based on the purpose of the engine model in the context of this work. An additional consideration was given to the type of information available for calibrating the engine model. This analysis led to the selection of a mean-value model to represent the engine in this work.

As the name suggests, CAR engine models calculate fluid and thermal dynamics within the engine as a function of the engine’s position in its cycle. The high frequency nature of this model is ideal when evaluating things that happen on a very small timescale. A couple of examples of such things are the flow dynamic effects in the intake manifold
with the introduction of a resonator or the effect on in-cylinder pressure fluctuations caused by different valve actuation profiles. For the goal of this work, to develop a control strategy for two valves in the EHRS, it is not necessary that the engine model be able to predict these types of values at such high frequencies. The thermal time constants of the components in the EHRS are far too large to be affected by such things. Whereas a CAR model calculates these types of events dynamically, on the timescale in which the MVM operates such events are instantaneous and can be expressed as algebraic equations [3.1]. Only engine values that change over the course of several engine cycles, or time developing values, are modeled dynamically.

In order to create an MVM, the modeler must make the assumption that for a given set of inputs, the engine will always return the same output; the cyclic variations in engine operation are ignored and the net effects of each engine cycle are averaged. This assumption allows, for example, a value such as the air/fuel ratio to be given deterministically based on a couple of inputs, eliminating the need to delve into the complexities of wall wetting in the intake manifold or fuel vapor disturbances from the emissions control system.

To create an MVM, first the time developing equations need to be laid out [3.1]. The deterministic algebraic relationships associated with the time developing system can then be added. Moskwa [3.2] suggests breaking the mean-value engine model down into five subsystems: 1) the throttle body, 2) the intake manifold, 3) the fuel injection system, 4) combustion and torque production, and 5) the rotational dynamics. The use of the engine model in this work will differ from Moskwa’s intended use. The torque and speed profile
of the engine will be imposed based on experimental data, therefore, rotational dynamics and torque production models will not be necessary.

Figure 15. Engine Model Block Diagram

The intake manifold can be modeled by a differential equation combining the conservation of mass equation and the ideal gas law to calculate pressure, as described later in this chapter. This requires knowledge of the mass flow rates of air into and out of the manifold. These values can be found algebraically. The throttle body can be modeled with a relationship describing compressible flow across an orifice [3.3], also described later in this chapter. This type of relationship, one that describes an instantaneous value, requires calibration with experimental data. The flow rate into the engine can be found by a relationship involving the volumetric efficiency of the engine, the engine speed, and the displacement of the engine. Each of these values, except the volumetric efficiency (VE), is known explicitly. The VE can be given as a function of known engine variables, such as throttle position, intake manifold pressure, or engine
speed. The rate of fuel injection is another engine variable that can be found algebraically. The air/fuel ratio (AFR) can be determined from a lookup table based on variables such as the throttle position and the engine speed. The AFR and the mass flow rate of air into the cylinder can be used to solve for the mass flow rate of exhaust leaving the engine with a simple physical equation. For the purposes of this work, one last algebraic relationship needs to be made: the exhaust temperature leaving the engine. This can also be considered a function of known engine variables such as the mass flow rate of air into the cylinders and the air/fuel ratio. With the exception of the exhaust mass flow rate which requires no calibration, all of the algebraic relationships described can easily be calibrated using experimental data.

The low computational complexity of the sub-models described here and the low-frequency of their calculations mean that MVMs are typically able to run at real-time or faster, thus, facilitating controls development and the large number of simulations it requires. Given a limited number of inputs, an MVM can provide the EHRS model with all of the values it needs (the exhaust temperature, the exhaust flow rate, and the intake manifold pressure) at a rate that meets or exceeds the frequency of events in the thermal management system. Because in this application the engine is not the focus of the work, the focus being the thermodynamic interactions downstream of the engine, the low calibration effort required for an MVM is greatly preferred to that of a more intensive CAR model. It is the sum of these attributes that makes a mean-value engine model ideally suited to the goals of this work.
3.2 Overview of Models for Heat Exchangers and Thermal Systems Dynamics

The exhaust heat recovery system discussed in this work depends on several heat exchangers to recoup waste heat from the exhaust stream and put it to use in other fluids. Two cross-flow air-to-liquid heat exchangers are employed as the radiator and cabin heater core. In this type of exchanger, the fluid flow paths are perpendicular to one another. Three counterflow exchangers are used, as well. In this type of heat exchanger, the fluid flow paths are parallel but the fluids flow in opposing directions. Two such liquid-to-liquid exchangers are used for the engine oil cooler and the transmission oil heater. Although a parallel flow arrangement is typically used for an EGR cooler, a counterflow gas-to-liquid unit is used in this work.

Each of these heat exchangers has the same basic mode of operation. A hot fluid and a cold fluid are separated by a metal wall. Heat is transferred from the hot fluid to the wall through convection, is conducted through wall, and is then transferred to the cold fluid through convection.

Figure 16. Counterflow Heat Exchanger, [3.4]
These various heat exchangers have different operating conditions that must be considered when deciding how best to model them. Three types of exchanger models are investigated for application in this work: a steady-state log mean temperature difference model (LMTD), a steady-state effectiveness-NTU model, and a transient performance model.

3.2.1 Log Mean Temperature Difference Model
An LMTD model computes an exchanger’s performance based on the overall heat transfer rate for the exchanger, $UA$, multiplied by the log mean difference in temperatures between the two fluids. See, for example, Inropera [3.5].

$$
\dot{Q} = UA \cdot \Delta T_{lm} \quad (3.1)
$$

The overall heat transfer rate is a function of all of the thermal resistances that stand between two fixed temperature surfaces. When applied to a steady-state heat exchanger model, there are three resistances considered due to: convection from the hot fluid to the wall, conduction through the wall, and convection from the wall to the cold fluid. These resistances are a result of the geometry and material properties of the heat exchanger, as well as the flow rates and physical properties of the fluids involved.

$$
UA = \left( R_{conv,h} + R_{cond,wall} + R_{conv,c} \right)^{-1} \quad (3.2)
$$

$$
R_{conv,h} = \frac{1}{h_{conv,h}A_{wall}} \quad (3.3)
$$
The way in which the log mean temperature difference is computed depends on the design of the heat exchanger. For parallel flow units, in which the hot and cold fluids flow in the same direction, the LMTD is calculated using the difference between fluids’ inlet temperatures and the difference between their outlet temperatures.

\[
\Delta T_{lm,parallel} = \frac{(T_{h,in} - T_{c,in}) - (T_{h,out} - T_{c,out})}{\ln \left( \frac{(T_{h,in} - T_{c,in})}{(T_{h,out} - T_{c,out})} \right)}
\]  

(3.6)

For counterflow exchangers, in which the fluids flow in opposite directions, the two temperature terms used are the difference between the hot inlet and the cold outlet and the difference between the hot outlet and the cold inlet.

\[
\Delta T_{lm,counter} = \frac{(T_{h,in} - T_{c,out}) - (T_{h,out} - T_{c,in})}{\ln \left( \frac{(T_{h,in} - T_{c,out})}{(T_{h,out} - T_{c,in})} \right)}
\]  

(3.7)

This approach does not consider the thermal capacitance of the exchanger and is, therefore, best suited to describing steady-state operation. Another limitation of this model is that it is most easily implemented when the inlet and outlet temperatures are known for both fluids and all that remains to be found is the overall heat transfer rate, \(UA\).
3.2.2 Effectiveness-NTU Model

When the fluid outlet temperatures are unknown, the effectiveness-NTU approach is a better option than the LMTD method. This technique requires knowledge of the overall rate of heat transfer which can be found given the inlet temperatures and flow rates. The exchanger’s effectiveness is the ratio of the actual heat transfer to the theoretical maximum heat transfer. The actual heat transfer is the product of either fluid’s mass flow rate, specific heat, and temperature change. The maximum heat transfer is that which would to bring the fluid with the smaller product of flow rate and specific heat, $\dot{C}_{min}$, to the inlet temperature of the other fluid.

$$\varepsilon = \frac{Q}{Q_{max}} = \frac{\dot{C}_h(T_{h,in} - T_{h,out})}{\dot{C}_{min}(T_{h,in} - T_{c,in})} = \frac{\dot{C}_c(T_{c,out} - T_{c,in})}{\dot{C}_{min}(T_{h,in} - T_{c,in})} \quad (3.8)$$

To solve for the fluid outlet temperatures, the effectiveness first needs to be known. As with the LMTD method, the calculation of the effectiveness is dependent on the type of exchanger used. For parallel and counterflow designs, respectively:

$$\varepsilon = \frac{1 - e^{-NTU(1+CR)}}{1 + CR} \quad (3.9)$$

$$\varepsilon = \frac{1 - e^{-NTU(1-CR)}}{1 - CR \cdot e^{-NTU(1-CR)}} \quad (3.10)$$

The capacity ratio, $CR$, is the ratio of $\dot{C}_{min}$ to $\dot{C}_{max}$. The number of transfer units, $NTU$, is the dimensionless ratio between the total heat transfer rate for the exchanger and the minimum capacitance rate, $\dot{C}_{min}$. 

51
As with the LMTD method, the effectiveness-NTU model ignores the thermal inertia of the exchanger. This assumption is acceptable for steady-state or near steady-state operating conditions because the exchanger will change temperature only very slightly if at all, thus absorbing a negligible amount of energy. This type of model is appropriate for modeling the transmission oil heater and the engine oil cooler in this work.

3.2.3 Transient Performance Model

When the operating conditions of a heat exchanger change rapidly, however, the effect of the exchanger’s thermal mass is much more significant and needs to be accounted for. A transient model of a heat exchanger differs from a steady-state model in that the temperature state of the wall separating the two fluids is calculated and affects the heat transfer from one fluid to the other. This works equally as well for steady-state operation as the two steady-state models described above but is uniquely capable of capturing the temperature dynamics within the exchanger.

Rather than using an overall heat transfer rate, the heat transfer process is broken down into two steps: 1) from the hot fluid to the wall, and 2) from the wall to the cold fluid. This delineation is important because each of these transfers is dependent upon the wall temperature. Scott [3.6] describes one such model. The heat transfer between the fluids and the wall is a function of the thermal resistances and temperatures involved.
\[
\dot{Q}_{\text{conv},h} = \frac{T_h - T_{\text{wall}}}{R_{\text{conv},h} + \frac{R_{\text{cond,wall}}}{2}} \tag{3.12}
\]
\[
\dot{Q}_{\text{conv},c} = \frac{T_{\text{wall}} - T_c}{\frac{R_{\text{cond,wall}}}{2} + R_{\text{conv},c}} \tag{3.13}
\]

Here the thermal resistance of the wall is divided by two because the wall’s mass is assumed to be at the center of its thickness. In each step the heat is only transferred halfway through the wall. An energy balance is applied to each of the fluid volumes and the wall to determine their temperatures at each point in the simulation.

\[
\frac{dT_h}{dt} = \frac{\dot{m}_h c_{p_h}(T_{h,\text{in}} - T_h) - \dot{Q}_{\text{conv},h}}{m_h c_{v_h}} \tag{3.14}
\]
\[
\frac{dT_c}{dt} = \frac{\dot{m}_c c_{p_c}(T_{c,\text{in}} - T_c) + \dot{Q}_{\text{conv},c}}{m_c c_{p_c}} \tag{3.15}
\]
\[
\frac{dT_{\text{wall}}}{dt} = \frac{\dot{Q}_{\text{conv},h} - \dot{Q}_{\text{conv},c}}{m_{\text{wall}} c_{v_{\text{wall}}}} \tag{3.16}
\]

Figure 17. Transient Heat Exchanger Diagram
This modeling method’s ability to accurately predict the behavior of a heat exchanger with rapidly varying operating conditions makes it ideal for representing an EGR cooler.

3.3 Principles of Control of Fluid and Thermal Systems

In order to supply the heat exchanger models described above with the information they require, mass flow rates and fluid temperatures, the EHRS’s fluid networks need to be modeled. In the simplest terms, these systems can be represented as thermodynamic volumes, or receivers, separated by flow restrictions.

3.3.1 Receivers

A receiver is a volume with constant dimensions, uniform pressure, and uniform temperature that utilizes simple filling-and-emptying equations to calculate the state of the fluid within it. When applying the conservation of energy equation, Canova [3.7] notes that one can neglect the work term, potential energy, and kinetic energy and consider only the internal energy which yields Equation 3.17.
For incompressible fluids, the flow rates into and out of the volume are equal and the mass contained is, therefore, constant. The mass of compressible fluids in a volume will vary, though, so the conservation of mass must be applied, as well.

\[
\frac{dU(t)}{dt} = \dot{H}_{in}(t) - \dot{H}_{out}(t) - Q_{out}(t) = \dot{m}_{in}(t) c_p T_{in} - \dot{m}_{out}(t) c_p T_{out} - \dot{Q}_{out}(t) \tag{3.17}
\]

The fluid’s temperature can be found based on the change in internal energy and mass in the volume. This calculation assumes an ideal gas with a constant specific heat. The ideal gas law can be applied to solve for the pressure in the volume.

\[
\frac{dT(t)}{dt} = \frac{dU(t)}{dt} \frac{1}{m(t) c_v} \tag{3.19}
\]

\[
p(t) = \frac{m(t) R T(t)}{V} \tag{3.20}
\]

Knowing the pressure in each volume is important because this information is used to compute the flow rate of fluids through the flow restrictions that separate the receivers.

### 3.3.2 Flow Restrictions

Anytime there exists a pressure differential between two receivers connected by a flow channel the fluid in the volume of higher pressure will flow towards the volume of lower pressure. The flow channel will act as a resistance to this flow, creating a pressure drop
between the volumes. The way in which this resistance to flow is modeled depends on the type of restriction and the fluid in question. When constructing a zero-dimensional model, flow control devices are typically modeled as pure steady-state resistances \cite{3.6}.

For incompressible fluids, the mass flow rate through a restriction can be found by applying the Borda-Carnot equation to Bernoulli’s principle to account for the mechanical energy loss in the fluid \cite{3.8}. The cross-sectional area of the flow path is represented by $A_{ref}$. The discharge coefficient, $C_d$, accounts for mechanical energy losses in the flow. For valves, the discharge coefficient is a function of valve position.

$$m(t) = C_d A_{ref} \sqrt{2 \rho (p_1(t) - p_2(t))}$$

or

$$\delta p(t) = C_2 \frac{1}{\rho} \frac{1}{2} m(t)^2$$

For incompressible flow in which the viscosity of the fluid has a non-negligible effect on the mechanical energy losses, the Darcy-Weisbach equation can be used to describe the pressure loss due to flow.

$$\delta p(t) = f \frac{L}{D} \frac{\rho \bar{U}^2}{2}$$

Simplifying and generalizing empirical relationships for the friction factor shown by other researchers, Scott \cite{3.9} proposes an equation with two terms to represent the friction factor.
\[ f = \frac{C_1}{Re} + C_2 \] (3.23)

Combining Equations 3.22 and 3.23 and collecting the geometric constants from each term into the coefficients yields the following expression for pressure loss due to flow.

\[ \delta p(t) = C_1 \frac{\mu}{\rho} \hat{m}(t) + C_2 \frac{1}{\rho} \hat{m}(t)^2 \] (3.24)

The value for each of the coefficients can be fit from experimental data. This equation can also be applied to compressible fluids flowing through pipes and other types of flow restrictions with length. This approach has also been shown to be effective for modeling flow in porous media by Bird, Stewart, and Lightfoot [3.10].

Compressible flow through a valve or orifice requires more considerations, however.

The relationship between two receivers’ pressures and the flow rate between them is dependent upon the heat capacity ratio of the gas. If the ratio between the downstream and upstream pressures exceeds the choked flow ratio then the valve produces sub-critical flow. If the pressure ratio is too low, then the flow becomes choked and a different equation for flow is needed [3.3].

If \[ \frac{p_{\text{down}}(t)}{p_{\text{up}}(t)} \geq \left( \frac{2}{\gamma + 1} \right)^{\frac{1}{\gamma - 1}} \]

then

\[ \dot{m}(t) = \frac{C_d(\alpha)A_{\text{ref}}p_{\text{up}}(t)}{\sqrt{R_{\text{gas}}T_{\text{up}}(t)}} * \left( \frac{p_{\text{down}}(t)}{p_{\text{up}}(t)} \right)^{\frac{1}{\gamma}} \sqrt{\frac{2\gamma}{\gamma - 1} \left[ 1 - \left( \frac{p_{\text{down}}(t)}{p_{\text{up}}(t)} \right)^{\frac{\gamma - 1}{\gamma}} \right]} \] (3.25)
If \( \frac{p_{down}}{p_{up}} < \left( \frac{2}{\gamma+1} \right)^{\frac{\gamma}{\gamma-1}} \)

then

\[
\dot{m}(t) = \frac{C_d(\alpha) A_{ref} p_{up}}{\sqrt{R_{gas} T_{up}}} \sqrt{\gamma} \sqrt{\frac{2}{\gamma+1} \left( \frac{\gamma+1}{\gamma-1} \right)^{\frac{\gamma}{\gamma-1}}} \]

The fundamental energy and flow equations discussed in this chapter are the basic building blocks for modeling a thermodynamic flow network. Adding the heat exchanger models also described gives one all of the tools necessary to construct a zero-dimensional model of an exhaust heat recovery system.
3.4 References


Chapter 4
MODELING AND SIMULATION OF INTERNAL COMBUSTION ENGINE THERMAL SYSTEMS

The previous chapters present an overview of the fundamental equations describing the energy balances and flows within a thermodynamic system, as well as several methods for modeling the performance of heat exchangers. Building on these premises, this chapter explains the process of using these principles and methods together to build a model of an advanced thermal management system. The system to be modeled, which includes an exhaust heat recovery system, will be described. The thermal model will include sub-models for the exhaust flow network, the coolant flow network, the EGR cooler, and all of the necessary thermal devices associated with the coolant flow network. In order to supply this model with the necessary runtime information, an engine model will be developed using the techniques discussed in the previous chapter. This engine model will include a flow sub-model describing the engine breathing process as well as empirical relationships estimating the exhaust properties. The component models within each of these sub-models will be detailed. Some component models will be based purely in fundamental concepts while others will be based on empirical relationships. When available data permits, these empirical
relationships will be calibrated and validated using a mixture of experimental data and simulation results.

4.1 Description of the Exhaust Heat Recovery System Simulator

A thermal management system consists of several components that, together, regulate the temperature of the engine, transmission, and their respective lubricating fluids, as well as supply heat to the cabin heater core, as outlined in Chapter Two. The components traditionally found on such a system include the coolant pump, the coolant flow paths through the block and head, the cabin heater core, the engine oil cooler, the radiator, and the transmission oil cooler. The exhaust system, which consists of the exhaust manifold, the catalytic converter, and the tailpipe, is generally kept separate from this system. The system modeled in this work, however, adds three new components which link the thermal management system to the exhaust system in a unique way. These components include an exhaust-to-coolant heat exchanger and two valves that control the flow of exhaust through this exchanger. The purpose of these devices is to remove heat from the exhaust stream and add it to the coolant stream. For this reason, it is referred to as an exhaust heat recovery system.

The exhaust heat recovery system simulator is a computer model built in Matlab/Simulink intended for the purpose of control development. It was designed to quickly and accurately predict, over a wide range of control inputs and operating conditions, the behavior of the physical components that are responsible for extracting heat from the exhaust gases and routing that energy into the coolant, engine oil, and
transmission fluid. To enhance the simulation speed, the model is comprised largely of zero-dimensional flow models and semi-empirical heat exchanger models that are not computationally burdensome, relative to the more complex one-dimensional and three-dimensional models used for design purposes. These models were calibrated and validated against a mix of experimental data and design simulation results.

![Diagram](image)

Figure 19. Thermal Management System Network

The plant model of the exhaust heat recovery system requires several exogenous inputs (shown in blue) upon which to base its calculations. Those inputs include the exhaust gas mass flow rate and the exhaust gas temperature leaving the exhaust manifold, the intake manifold pressure, the engine speed, and the ambient pressure. The plant is controlled by several actuators (shown in green, inputs outlined in green are out of the scope of this work), each of which requires an input for the model to function. Those control inputs
include the engine back pressure valve position, the EGR bypass valve position, the EGR hi-flow valve position, the EGR coolant valve position, the engine oil cooler coolant valve position, the three-way valve position, the thermostat position, and the water pump speed (if electric overdrive is required). Outputs from the system are shown in red. It is the control of the engine back pressure valve and EGR bypass valve actuators that this project aims to optimize for the task of minimizing the warm-up time of the fluids.

4.2 Implementation of the Thermal System Model

The exhaust heat recovery system model is divided, based on commonality of function, into four distinct pieces: the exhaust flow network, the coolant flow network, the EGR cooler thermal model, and the coolant thermal models. The exhaust and coolant flow networks deal with the temperature, pressure, and flow dynamics of their respective fluids. They supply the heat exchanger thermal models with inputs necessary to calculate heat transfer rates. The EGR cooler thermal model is a detailed, transient thermal model of the EGR cooler that is able to accurately predict coolant and exhaust gas outlet temperatures despite large fluctuations in inputs. Due to the relative stability of their operating conditions, the coolant-to-oil and coolant-to-air heat exchanger thermal models are less complex, steady-state models that predict coolant and lubricating fluid temperatures to the level of accuracy necessary for control development.
4.2.1 Exhaust Flow Network

The exhaust flow network model is a system of differential equations describing the interactions of four fixed pipe volumes and the six flow restrictions that connect them, as seen in Figure 21. The difference between the pressures in the volumes on either side of a flow restriction equals the pressure loss due to flow through that restriction. A mass flow rate through each restriction is calculated based on the pressure loss (Equations 4.4-4.6, 4.12). Solved in an continuous manner, the flow of exhaust gas into and out of the volumes varies their respective pressures (Equations 4.1-4.3) and leads to new mass flow rates through the restrictions.
The exogenous inputs to this model are the exhaust gas mass flow rate and the exhaust gas temperature leaving the exhaust manifold, the intake manifold pressure, and the ambient pressure. Control inputs for this network are the engine back pressure valve position, the EGR bypass valve position, and the EGR hi-flow valve position. The useful outputs from this model are the exhaust gas temperatures and flow rates to the EGR cooler and to the intake manifold.

The four thermodynamic volumes of the exhaust flow network are nodes within the pipes of the exhaust system that are each considered to have fixed dimensions, uniform pressure, and uniform temperature. These four spaces are the region between the exhaust manifold and the catalytic converter (pre-catalyst volume), the region after the catalytic converter but before the EGR cooler and engine back pressure valve (volume 1), the
region between the EGR cooler and the EGR hi-flow valve (volume 2), and the region downstream of the engine back pressure and EGR bypass valves (volume 3).

Each volume uses three equations to calculate the mass of exhaust gas contained therein and the state variables temperature and pressure (Equations 4.1-4.3). A differential equation based on the sum of the mass flow rates into and out of a volume is used to calculate the mass of the exhaust gas.

\[
\frac{d(m_{\text{vol}})}{dt} = \sum (\dot{m}_{\text{exh,in}}) - \sum (\dot{m}_{\text{exh,out}}) \quad (4.1)
\]

Temperature in each volume is calculated dynamically by an equation that accounts for the flow of energy into and out of that volume, as well as the change in mass of a fixed volume. The model assumes ideal gas with a constant specific heat and no heat loss from the volumes to their surroundings.

\[
\frac{d(T_{\text{exh,vol}})}{dt} = \frac{1}{m_{\text{vol}}c_p} \left[ \sum \dot{m}_{\text{exh,in}}(c_pT_{\text{exh,in}} - c_vT_{\text{exh,vol}}) - \sum \dot{m}_{\text{exh,out}}R_{\text{exh}}T_{\text{exh,vol}} \right] \quad (4.2)
\]

After the temperature and mass have been calculated for each volume, these values are applied to the ideal gas law to calculate the pressure in the volume.

\[
p_{\text{vol}} = \frac{m_{\text{vol}}R_{\text{exh}}T_{\text{exh,vol}}}{V_{\text{vol}}} \quad (4.3)
\]

The flow restrictions of the exhaust flow network determine the pressures of the various thermodynamic volumes by allowing or impeding the flow of exhaust gases into and out
of those volumes. The six flow restrictions are the catalytic converter, the EGR cooler, the engine back pressure valve, the EGR bypass valve, the EGR hi-flow valve, and the exhaust system, or tailpipe.

Each of the three valves is modeled as a flow restriction with a variable discharge coefficient calculated based on its position (Equations 4.4, 4.5). The model first determines whether the ratio between pressures of the downstream and upstream volumes meets the sub-critical flow ratio. The sub-critical flow ratio is dependent upon the ratio of specific heats ($\gamma$). A value for $\gamma$ is found in Equation 4.48. If the pressure ratio is above the critical value, then the sub-critical exhaust mass flow rate is returned. In sub-critical flow, the mass flow rate will increase if either the upstream pressure increases or the downstream pressure decreases. If, however, the pressure ratio falls below the critical value then the valve enters a state of choked flow. During choked flow, the mass flow rate is no longer dependent on the downstream pressure, but it responds only to the upstream pressure, varying with direct proportionality.

\[
\text{If } \frac{p_{\text{vol,down}}}{p_{\text{vol,up}}} \geq \left( \frac{2}{\gamma+1} \right)^{\gamma-1}
\]

then

\[
\dot{m}_{\text{exh,valve}} = C_d(\alpha) \frac{A_{ref} p_{\text{vol,up}}}{\sqrt{R_{\text{exh}}} T_{\text{exh,vol,up}}} \left( \frac{p_{\text{vol,down}}}{p_{\text{vol,up}}} \right)^{\frac{1}{\gamma}} \sqrt{\frac{2\gamma}{\gamma-1}} \left[ 1 - \left( \frac{p_{\text{vol,down}}}{p_{\text{vol,up}}} \right)^{\frac{\gamma-1}{\gamma}} \right]
\]  

(4.4)
If \( \frac{P_{\text{vol.down}}}{P_{\text{vol.up}}} < \left( \frac{2}{\gamma + 1} \right)^{\frac{\gamma}{\gamma - 1}} \)

then

\[
\dot{m}_{\text{exh, valve}} = \frac{C_d(\alpha)A_{\text{ref}} P_{\text{vol.up}}}{\sqrt{R_{\text{exh}} T_{\text{exh, vol.up}}}} \cdot \sqrt{\gamma} \cdot \sqrt{\frac{2}{\gamma + 1}}
\]

The mass flow rate through each valve is proportional to the effective area of the valve: the product of the cross-sectional area of the flow path through the valve, \( A_{\text{ref}} \), and the valve’s discharge coefficient, \( C_d(\alpha) \). The discharge coefficient used for each valve is a function of the normalized valve position. A position of zero corresponds to a completely closed valve, and one to a fully open valve.

Figure 22. Valve Discharge Coefficient

The catalytic converter, unlike the valves in the exhaust system, is a flow restriction with length. Therefore, the modified form of the Darcy-Weisbach equation discussed in
Chapter Three can be used (Equation 4.6). The catalyst does not contain any actuators, so the coefficients used to describe its flow behavior will be constants.

\[
\Delta p_{exh,c} = C_1 \frac{\mu_{exh}}{\rho_{exh}} \frac{\dot{m}_{exh,c}}{\rho_{exh}} + C_2 \frac{1}{\rho_{exh}} \frac{\dot{m}_{exh,c}^2}{\rho_{exh}}
\]  

(4.6)

In addition to being a flow restriction, the catalytic converter also acts a thermal inertia in the exhaust system. Upon startup, the catalyst absorbs much of the heat in the exhaust gas due to the small gas flow channels and large substrate surface area. Once warmed, the thermal inertia of the catalyst has a smoothing effect on the large variations in temperature that the engine produces given rapidly changing speeds and torque conditions. No experimental data for catalyst performance was available when building this model. For this reason, the catalyst model used in this work and described here is a simplified version of the model proposed in [4.1].

This model uses eighteen thermal nodes; the catalyst is broken into six segments longitudinally and each segment contains a gas node, substrate node, and shell node. The exhaust gases transfer heat to the substrate. This, in turn, transfers heat radially to the outer shell and longitudinally to the upstream and downstream substrate nodes. The outer shell conducts heat to the shell nodes upstream and downstream as well as transferring heat to the ambient air through convection and radiation.

To simplify the model for this work, the catalyst was considered to be one segment with three thermal nodes. This removed several state variables and eliminated the longitudinal conduction of heat. This left the model with only three transfers of heat to calculate: gas-
to-substrate, substrate-to-shell, and shell-to-ambient. The heat transfer rate to the substrate is a function of the gas and substrate temperatures, as shown in Equation 4.7.

\[ Q_{\text{sub, exh}} = h_{\text{sub, exh}} * A_{\text{sub}} * (T_{\text{exh}} - T_{\text{sub}}) \]  

The gas temperature can then be calculated based on heat loss, exhaust gas flow, the specific heat of exhaust gases, and the mass of exhaust contained in the catalyst.

\[ \frac{dT_{\text{exh}}}{dt} = \frac{\dot{m}_{\text{exh}} c_{p_{\text{exh}}} (T_{\text{exh,in}} - T_{\text{exh}}) - \dot{Q}_{\text{sub, exh}}}{V_{\text{exh, cat}} P_{\text{exh}} R_{\text{exh}} * T_{\text{exh}} * c_{v_{\text{exh}}}} \]  

The substrate temperature is dependent upon convection from the exhaust gases and conduction to the shell (Equation 4.9).

\[ \frac{dT_{\text{sub}}}{dt} = \frac{\dot{Q}_{\text{sub, exh}} - K_{\text{sub, shell}} (T_{\text{sub}} - T_{\text{shell}})}{m_{\text{sub}} * c_{v_{\text{sub}}}} \]  

The shell temperature loses much of this energy to the ambient air through radiation and convection. The remainder of the energy heats the shell (Equation 4.10).

\[ \frac{dT_{\text{shell}}}{dt} = \frac{K_{\text{sub, shell}} (T_{\text{sub}} - T_{\text{shell}}) - \dot{Q}_{\text{amb, shell}}}{m_{\text{shell}} * c_{v_{\text{shell}}}} \]  

where

\[ \dot{Q}_{\text{amb, shell}} = h_{\text{shell, amb}} A_{\text{shell}} (T_{\text{shell}} - T_{\text{amb}}) \]  

The values given for all of the constants used in the equations describing the catalytic converter are tabulated in Table 1 [4.1].
<table>
<thead>
<tr>
<th>Constant (unit)</th>
<th>Value</th>
<th>Constant (unit)</th>
<th>Value</th>
<th>Constant (unit)</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$h_{\text{sub,exh}} \left( \frac{W}{m^2 \cdot K} \right)$</td>
<td>166</td>
<td>$c_{p_{\text{exh}}} \left( \frac{J}{kg \cdot K} \right)$</td>
<td>1200</td>
<td>$c_{v_{\text{shell}}} \left( \frac{J}{kg \cdot K} \right)$</td>
<td>460</td>
</tr>
<tr>
<td>$A_{\text{sub}} \left( \frac{m^2}{m} \right)$</td>
<td>6.401</td>
<td>$K_{\text{sub,shell}} \left( \frac{W}{K} \right)$</td>
<td>0.152</td>
<td>$h_{\text{shell,amb}} \left( \frac{W}{m^2 \cdot K} \right)$</td>
<td>2.2</td>
</tr>
<tr>
<td>$V_{\text{exh,cat}} \left( \frac{m^3}{m} \right)$</td>
<td>0.0017</td>
<td>$m_{\text{sub}} \left( kg \right)$</td>
<td>0.7689</td>
<td>$A_{\text{shell}} \left( m^2 \right)$</td>
<td>0.0623</td>
</tr>
<tr>
<td>$R_{\text{exh}} \left( \frac{J}{kg \cdot K} \right)$</td>
<td>289</td>
<td>$c_{v_{\text{sub}}} \left( \frac{J}{kg \cdot K} \right)$</td>
<td>460</td>
<td>$m_{\text{shell}} \left( kg \right)$</td>
<td>0.7111</td>
</tr>
</tbody>
</table>

Table 1. Catalyst Model Parameters

The restriction of flow through the tailpipe of the exhaust system is modeled with the Borda-Carnot equation (Equation 3.21). The flow coefficient, which is a calibration parameter, is held constant because there are no actuators in the flow path.

$$\dot{m}_{\text{exh,amb}} = C_1 \sqrt{2 \rho (p_{\text{vol,up}} - p_{\text{vol,down}})}$$  \hspace{1cm} (4.12)

The exhaust gas path through the EGR cooler consists of a set of small tubes running in parallel. This path is also modeled as a fixed flow restriction, as shown in Equation 4.13.

$$\delta p_{\text{exh,EGR}} = C_{1_{\text{exh}}} \frac{\mu_{\text{exh}}}{\rho_{\text{exh}}} \dot{m}_{\text{exh,EGR}} + C_{2_{\text{exh}}} \frac{1}{\rho_{\text{exh}}} \dot{m}_{\text{exh,EGR}}^2$$  \hspace{1cm} (4.13)

This equation can then be rearranged to give the mass flow rate as a function of pressure loss.
The coolant flow network model is a system of pressure loss equations describing the interactions of five coolant flow paths and the coolant pump that supplies them. Each flow path has a fixed resistance in the form of a heat exchanger in series with a variable resistance in the form of a valve. The flow restrictions, in addition to the engine, include the EGR cooler and EGR coolant valve, the engine oil cooler and engine oil coolant valve, the radiator and electrically-heated thermostat valve, and the cabin heater core and transmission oil heater which are separated by a three-way valve (Figure 24).
The only exogenous input necessary for this system is engine speed, which is used to calculate the speed of the pump. The pump can, however, be decoupled from the engine and driven electrically if greater coolant flow is required or less flow is desirable. The other control inputs for this system are the EGR coolant valve position, the engine oil cooler valve position, the three-way valve position, and the thermostat position, which can be controlled by heating the internal wax pellet. The useful outputs generated by the coolant flow network are coolant flow rates through the engine and each of the heat exchangers, as well as the coolant pressure entering the EGR cooler.

Quantifying flow rates in the coolant system is quite a bit simpler than in the exhaust system. Given the compressibility of exhaust gases, the flow rates at two points in the exhaust system are not directly related. The pressure dynamics, which keep flow rates somewhat independent of each other, introduce hysteresis to that system. Due to the
incompressibility of coolant, however, the coolant flow network does not display any meaningful hysteresis. Flow rates therein are, therefore, a direct result of the exogenous and control inputs.

Pump speed and an initial pressure difference across the pump are used to calculate the flow rate exiting the pump. This flow passes through the engine and is then apportioned to the downstream flow paths based on the relative flow restriction of each. Pressure loss through each of the components is calculated given an imposed flow rate. The total pressure loss through the system for a given time step of the simulation becomes the pressure rise across the pump for the next time step. The process is repeated at each time step in this iterative manner.

The pump is modeled using a two-dimensional lookup table. The inputs to the table are the pump speed and the pressure rise across the pump. The table outputs a volumetric flow rate that is converted into a mass flow rate using the density of the coolant at the pump.

\[
\dot{m}_{\text{water, pump}} = \dot{V}_{\text{water, pump}} \cdot \rho_{\text{water}} = f(RPM, \delta p_{\text{pump}}) \cdot \rho_{\text{water}} \quad (4.15)
\]

From the pressure side of the pump, the coolant circulates through the engine’s cooling jacket. Given that the model is based on a V6 engine, flow through the engine is broken down into flow through each of the individual banks. The coolant is fed into each side of the block from the same source and exits to a common pipe. Having shared pressure values at both inlet and outlet, the pressure loss across each side of the engine must, therefore, be the same.
By setting the pressure loss equations equal to one another and noting that the sum of the coolant flow rates through the engine must equal the flow rate from the pump, the respective flow rates can be calculated without prior knowledge of the upstream or downstream pressures.

\[
\delta p_{\text{water,eng}_l} = \delta p_{\text{water,eng}_r} \tag{4.16}
\]

\[
\delta p_{\text{eng}} = \frac{C_{1_{\text{eng}_l}} \mu_{\text{water}}}{\rho_{\text{water}}} m_{\text{eng}_l} + \frac{C_{2_{\text{eng}_l}}}{\rho_{\text{water}}} m_{\text{eng}_l}^2 \tag{4.17}
\]

\[
\delta p_{\text{eng}} = \frac{C_{1_{\text{eng}_r}} \mu_{\text{water}}}{\rho_{\text{water}}} m_{\text{eng}_r} + \frac{C_{2_{\text{eng}_r}}}{\rho_{\text{water}}} m_{\text{eng}_r}^2 \tag{4.18}
\]

where

\[
m_{\text{eng}_r} = m_{\text{pump}} - m_{\text{eng}_l} \tag{4.19}
\]

Substituting the relationship in Equation 4.19 into Equation 4.18 and setting it equal to Equation 4.17 will solve for the flow rate through the left side of the engine based on the known flow rate from the pump. This value can be used to solve for both the flow rate through the right side of the engine and the pressure drop across the engine.

The five flow paths downstream of the engine cooperate in the same way as the two paths through the engine. All begin at the same source, just after the engine, and all end at the same point, just before the pump. It can then be inferred that the pressure loss through each heat exchanger path must be the same.
\[ \delta p_{\text{HEX}} = \delta p_{\text{EGR}} = \delta p_{\text{EOH}} = \delta p_{\text{rad}} = \delta p_{\text{TOH}} = \delta p_{\text{heater}} \tag{4.20} \]

The pressure drop through each of the paths is modeled in terms of the flow through it. These paths contain a mixture of valves and flow restrictions with length in series with each other. The pressure losses associated with each restriction are summed.

The quadratic expression relating flow rate to pressure drop is rearranged to give Equation 4.21. This expression is used for all valves and for any other flow restrictions for which there was not enough data available to calibrate the viscous term of Equation 4.22.

\[ \delta p = \frac{\dot{m}^2}{2 \rho (c_d A_{\text{ref}})^2} \tag{4.21} \]

\[ \delta p = C_1 \frac{\mu}{\rho} \dot{m}(t) + C_2 \frac{1}{\rho} \dot{m}(t)^2 \tag{4.22} \]

\[ \delta p_{\text{EGR}} = \frac{\dot{m}_{\text{EGR}}^2}{(C_d(\alpha)A_{\text{ref, valve}})^2 2 \rho_{\text{water}}} + C_{1_{\text{EGR}}} \frac{\mu_{\text{water}}}{\rho_{\text{water}}} \dot{m}_{\text{EGR}} + C_{2_{\text{EGR}}} \frac{1}{\rho_{\text{water}}} \dot{m}_{\text{EGR}}^2 \tag{4.23} \]

\[ \delta p_{\text{EOH}} = \frac{\dot{m}_{\text{EOH}}^2}{(C_d(\alpha)A_{\text{ref, valve}})^2 2 \rho_{\text{water}}} + \frac{\dot{m}_{\text{EOH}}^2}{A_{\text{eff, EOH}}^2 2 \rho_{\text{water}}} \tag{4.24} \]

\[ \delta p_{\text{rad}} = \frac{\dot{m}_{\text{rad}}^2}{(C_d(\alpha)A_{\text{ref, stat}})^2 2 \rho_{\text{water}}} + \frac{\dot{m}_{\text{rad}}^2}{A_{\text{eff, rad}}^2 2 \rho_{\text{water}}} \tag{4.25} \]

\[ \delta p_{\text{TOH}} = \frac{\dot{m}_{\text{TOH}}^2}{(C_d(\alpha)A_{\text{ref, way}})^2 2 \rho_{\text{water}}} + \frac{\dot{m}_{\text{TOH}}^2}{A_{\text{eff, TOH}}^2 2 \rho_{\text{water}}} \tag{4.26} \]

\[ \delta p_{\text{heater}} = \frac{\dot{m}_{\text{heater}}^2}{(C_d(\alpha)A_{\text{ref, way}})^2 2 \rho_{\text{water}}} + \frac{\dot{m}_{\text{heater}}^2}{A_{\text{eff, heater}}^2 2 \rho_{\text{water}}} \tag{4.27} \]
The flow rate through each of the other paths is solved for in terms of the flow rate through the EGR cooler loop (this is done because the EGR cooler is the only loop that will never have zero flow; the EGR cooler always needs some amount of coolant flow to prevent overheating). For example, the pressure loss Equations 4.23 and 4.24 can be set equal and rearranged to solve for the flow rate through the engine oil heater in terms of the flow rate through the EGR cooler.

\[ \dot{m}_{EOH} = \sqrt{\frac{C_{1EGR} \mu_{water} \dot{m}_{EGR} + \dot{m}_{EGR}^2 \left( C_{2EGR} + \frac{1}{2(C_d(\alpha)A_{ref\,valve,EGR})^2} \right)}{\frac{1}{2} \left( \frac{1}{(C_d(\alpha)A_{ref\,valve,EOH})^2} + \frac{1}{A_{eff\,EOH}^2} \right)}} \] (4.28)

The only variable in the above equation is the flow rate of coolant through the EGR cooler. Similar equations are constructed for each of the heat exchanger flow paths. With all of the flow rates calculated in terms of flow through the EGR cooler, the EGR cooler mass flow rate can then be determined using Equation 4.29.

\[ \dot{m}_{EGR} = \dot{m}_{pump} - \Sigma(\dot{m}_{EOH} + \dot{m}_{TOH} + \dot{m}_{rad} + \dot{m}_{heater}) \] (4.29)

The coolant mass flow rate through the EGR cooler is then used to calculate the other flow rates and the pressure drop through the heat exchanger flow paths. Summing the pressure loss across the engine with the pressure loss across the heat exchangers gives the total pressure rise across the pump.

\[ \delta p_{pump} = \delta p_{eng} + \delta p_{HEX} \] (4.30)
4.2.3 EGR Cooler Thermal Model

The central element of the exhaust heat recovery system is the EGR cooler. The device considered in this work is a shell-and-tube, counter-flow type heat exchanger. It is this component that physically links the exhaust system to the cooling system and is responsible for extracting waste heat from the exhaust gases and transferring it to the coolant. Properly predicting the heat exchanger’s performance is critical to the accuracy of the overall model. One concern when modeling the EGR cooler’s performance is correctly capturing its transient responses to widely varying inputs. The temperature of incoming exhaust gases can change much more rapidly the than large thermal mass of the heat exchanger. This causes the EGR cooler to exhibit hysteresis in the sense that the rate of heat transfer to the coolant at any given instant is not solely dependent on the current exogenous inputs. Instead, it is also affected by the current wall temperature within the exchanger. For this reason, a transient thermal model was used to predict the exchanger’s performance. This type of model accounts for the thermal inertia of the exchanger’s internal tubes as well as the outer shell.
The model requires the following exogenous inputs: exhaust inlet temperature, exhaust mass flow rate, coolant inlet temperature, coolant mass flow rate, coolant inlet pressure, and ambient temperature. The EGR cooler is a passive device and, therefore, does not contain any actuators or take any control inputs. The useful outputs from this model are coolant outlet temperature, exhaust outlet temperature, and the energy ratio for the heat exchanger.

The last output, energy ratio, is a value that the device’s manufacturer uses to gauge stress on the EGR cooler. Calculated as the quotient of recoverable power in the exhaust stream divided by the power absorption potential of the coolant, the energy ratio provides one of the guidelines for maintaining safe operating conditions in the heat exchanger. This value plays an important role in the exhaust heat recovery system control strategy.

\[
\tau_{energy} = \frac{\text{Recoverable exhaust power}}{\text{Coolant capacity to absorb power}} = \frac{\dot{m}_{exh,EGR}c_{p,exh}(T_{exh,vol}-T_{water,EGR,in})}{\dot{m}_{water,EGR}c_{p,water}(T_{boll}-T_{water,EGR,in})} \tag{4.31}
\]
The recoverable exhaust power is defined as the rate at which energy would need to be removed from the incoming exhaust gas to cool it to the temperature of the incoming coolant. This is the maximum rate at which energy could be transferred to the coolant given a counterflow heat exchanger of infinite length. The coolant’s potential to absorb power is defined as the amount of power required to heat the incoming flow of coolant to its boiling point. Boiling in the coolant is a major concern because the localized overheating of the EGR cooler that can result causes damaging thermal stresses in the component. The manufacturer recommends that this value not exceed 0.45 to ensure proper protection of the device.

The boiling point of the coolant is calculated at each time step as a function of the pressure of the incoming coolant using a derivation of the Clausius-Clapeyron equation.

\[
T_{\text{boil}} = \left( \frac{1}{T_{\text{boil,ref}}} - \frac{R_{\text{univ}} \ln \left( \frac{P_{\text{water,EGR,in}}}{P_{\text{water,ref}}} \right)}{\Delta H_{\text{vap}}} \right)^{-1}
\]  

(4.32)

This equation requires knowledge of the boiling point of the propylene glycol (PG)/water mixture at a reference pressure. In this case, atmospheric pressure was used as the reference point. At this pressure, a mixture of equal masses of PG and water boils at 381K, or 108°C, as seen in Figure 26.
The heat of vaporization for the mixture must also be known. This can be found by calculating a weighted average of the water’s and PG’s heats of vaporization using their molar masses. These heats of vaporization and molar masses are 40,626 J/mol and 76.1 g/mol for PG and 69,546 J/mol and 18 g/mol for water. The weighted average yields a vaporization heat of 46,252 J/mol for a mixture of equal parts PG and water. The resulting boiling point curve is seen in Figure 27.
Following a model proposed by Scott [4.4], the EGR cooler thermal model is comprised of three differential equations that calculate the temperatures of the gas volume, the coolant volume, and the walls separating them. The thermal mass of the tube walls, although not considered in the coolant-to-oil and coolant-to-air exchangers, is considered in this model in order to give a better representation of the heat exchanger’s performance in transient conditions.

In this transient heat exchanger model, the heat transfer between the two fluids is modeled as a two step process rather than a single step. First, heat from the hot exhaust gases is transferred to the cold tube walls. The mass of the tubes is considered to be at a point in the center of their wall’s thickness. From there, the heat is transferred to the coolant.

The temperature of the exhaust in the exchanger is calculated dynamically, based on both the rate of heat transfer from the exhaust to the wall and the flow of fluid through the
volume (Equation 4.33). The coolant volume’s temperature is affected by the rate of heat addition from the tube walls and the flow of fluid, as well (Equation 4.34). The wall temperature is a function of the gas side and coolant side heat transfer rates (Equation 4.35).

\[
\frac{dT_{exh}}{dt} = \frac{1}{m_{exh, EGR} c_{vexh}} \left( \frac{T_{wall} - T_{exh}}{(R_{conv_{exh}} + R_{cond_{wall}})} + \dot{m}_{exh, EGR} c_{pexh} (T_{exh, in} - T_{exh}) \right)
\]  

(4.33)

\[
\frac{dT_{water}}{dt} = \frac{1}{m_{water} c_{p_{water}} + m_{shell} c_{v_{wall}}} \left( \frac{T_{wall} - T_{water}}{(R_{conv_{water}} + R_{cond_{wall}})} + \dot{m}_{water, EGR} c_{p_{water}} (T_{water, in} - T_{water}) \right)
\]  

(4.34)

\[
\frac{dT_{wall}}{dt} = \frac{1}{m_{wall} c_{v_{wall}}} \left( \frac{T_{water} - T_{wall}}{(R_{conv_{water}} + R_{cond_{wall}})} + \frac{T_{exh} - T_{wall}}{(R_{conv_{exh}} + R_{cond_{wall}})} \right)
\]  

(4.35)

The rate of heat transfer between two media is calculated by dividing the difference in temperature between them by the sum of the individual thermal resistances that separate them, \( R_{conv_{exh}}, R_{cond_{wall}}, R_{conv_{water}} \). The rate of change in enthalpy of a volume is divided by its thermal capacitance to determine the rate of temperature change. Note that the thermal capacitance used for calculating the coolant temperature includes the outer shell of the heat exchanger. This is done because the shell temperature will follow the coolant temperature very closely and is a significant heat sink during the coolant warm-up phase.

There are two modes of heat transfer at work in the EGR cooler: 1) convection between the fluids and the surface of the tubes, and 2) conduction through the tube walls.
Resistance to conductive heat transfer is a product of wall thickness ($\delta_{\text{wall}}$), wall area, and thermal conductivity of the material.

$$R_{\text{cond,wall}} = \frac{\delta_{\text{wall}}}{2A_{\text{wall}}K_{\text{wall}}}$$  \hspace{1cm} (4.36)

The wall’s thermal resistance is divided by two because the heat is transferred through the wall in two steps: first, from the exhaust to the mass at the center of the wall thickness, then to coolant. Scott [4.4] expresses the thermal conductivity and specific heat behavior of the tube material, austenitic stainless steel, as functions of temperature in $K$.

$$K_{\text{wall}} = 14 \times \left(\frac{T_{\text{wall}}}{300}\right)^{4.3}$$  \hspace{1cm} (4.37)

$$c_{v_{\text{wall}}} = 450 \times \left(\frac{T_{\text{wall}}}{300}\right)^{4.5}$$  \hspace{1cm} (4.38)

The resistance to convection between a fluid and a wall is defined as the inverse of the coefficient of convective heat transfer multiplied by the surface area across which the transfer is occurring.

$$R_{\text{conv}} = \frac{1}{hA_{\text{wall}}}$$  \hspace{1cm} (4.39)

The convective heat transfer coefficient can be found in a circular tube using the Nusselt number, the fluid’s thermal conductivity, and the diameter of the flow path (Equation 4.40).
For turbulent flow in a circular tube, Incropera [4.5] suggests using the Colburn equation, given in Equation 4.41, to predict the Nusselt number.

\[ Nu = 0.023 \cdot Re^{0.8} \cdot Pr^{\frac{1}{3}} \]  

(4.41)

The Colburn equation is a prediction of the Nusselt number based on empirical observations. Since there exists experimental data against which to calibrate the EGR cooler model, the coefficient of this expression and the exponent applied to the Reynolds number may be modified during the calibration process. For now, they will be replaced with the variables \( C_1 \) and \( \beta \), respectively.

The Reynolds number, \( Re \), is a dimensionless variable describing a fluid’s flow characteristics. According to Incropera [4.5] it is “the ratio of inertia to viscous forces.” For flow in a circular pipe, the expression for the Reynolds number is given by the fluid density \( (\rho) \), dynamic viscosity \( (\mu) \), the diameter of the flow path \( (D) \), and the average fluid velocity \( (\bar{V}) \).

\[ Re = \frac{\rho \cdot D \cdot \bar{V}}{\mu} \]  

(4.42)

Multiplying the average fluid velocity by its density results in the fluid’s mass velocity, \( G \). Substituting this into the equation for the Reynolds number and combining with the
Nusselt number formulae yields the following expression for the coefficient of convective heat transfer:

\[ h = C_1 \frac{K}{D} \left( \frac{D \cdot G}{\mu} \right)^{\beta} \cdot Pr^{\frac{1}{3}} \]  

(4.43)

Scott [4.4] recommends lumping the constants from the previous equation into one term which gives the final form of the expression for the convection coefficient.

\[ h = C_1 \frac{K \cdot G^{\beta} \cdot Pr^{\frac{1}{3}}}{\mu^{\beta}} \]  

(4.44)

Incropera [4.5] defines the Prandtl number as the “ratio of fluid momentum and thermal diffusivities.” It is expressed as the product of specific heat \( (c_p) \) and dynamic viscosity \( (\mu) \) divided by thermal conductivity \( (K) \).

\[ Pr = \frac{c_p \mu}{K} \]  

(4.45)

To determine expressions for the physical properties of typical combustion products, the composition of the gas is determined by assuming complete combustion, a stoichiometric air/fuel ratio, and an average chemical formula for gasoline. With these assumptions, the only gases in the exhaust are CO\(_2\), H\(_2\)O, and N\(_2\) [4.6]. Using a mole fraction averaged approach [4.7], the ideal gas constant for exhaust gas was found to be 289 \( \frac{J}{kg \cdot K} \).

Applying this value to the ideal gas law gives an expression for the density of exhaust. All temperatures are in K and pressures are in Pa unless otherwise noted.
The mass average of each constituent’s specific heat was used to compute the specific heat of the mixture [4.4].

\[ c_{p_{exh}} = \frac{\Sigma (m_i \cdot c_{pi})}{\Sigma m_i} = 976.5 + 0.295 \cdot T \]  

(4.47)

For the purpose of defining a constant value of the heat capacity ratio (\( \gamma \)), a value of 1200 \( \frac{j}{k_{gs} \cdot K} \) was selected for \( c_{p_{exh}} \) using a median temperature for exhaust gas in the system.

The value for \( \gamma \) was then found by dividing \( c_{p_{exh}} \) by the difference between \( c_{p_{exh}} \) and the gas constant for exhaust. This resulted in a heat capacity ratio of 1.317.

\[ \gamma = \frac{c_{p_{exh}}}{c_{p_{exh}} - R_{exh}} = \frac{1200}{1200 - 289} = 1.317 \]  

(4.48)

Thermal conductivity of the mixture was found using an equation proposed by Mason and Saxena [4.8]. This method divides the mole fraction averaged thermal conductivity of the mixture by the mole fraction averaged interaction parameters between each pair of components.

\[ K_{exh} = \sum_{i=1}^{3} \frac{y_i K_i}{\sum_{j=1}^{3} y_i A_{ij}} = -2.548 \cdot 10^{-3} + 9.541 \cdot 10^{-5} \cdot T - 2.95 \cdot 10^{-8} \cdot T^2 \]  

(4.49)

The interaction parameter is a function of each component’s molecular mass (\( M \)) and its dynamic viscosity (\( \mu \)).
The dynamic viscosity of the composition is found by the same method using the same interaction parameters.

\[
A_{ij} = \left[ 1 + \left( \frac{\mu_i}{\mu_j} \right)^{\frac{1}{2}} \left( \frac{M_i}{M_j} \right)^{\frac{1}{2}} \right] \left[ \frac{1}{8 \left( 1 + \frac{M_i}{M_j} \right)^{\frac{1}{2}}} \right]^2
\]  

(4.50)

\[
\mu_{exh} = \left( \sum_{i=1}^{3} y_i \mu_i \right) / \left( \sum_{j=1}^{3} y_i A_{ij} \right) = \frac{1.48 \times 10^{-6} \times T^{\frac{1}{3}}}{T + 173}
\]  

(4.51)

Alshamani [4.9] compiled all of the applicable physical properties for a mixture of 50% water and 50% propylene glycol coolant, by mass.

\[
\rho_{water} = 1050.5 - 0.564 \times [T - 273] - 1.38 \times 10^{-3} \times [T - 273]^2
\]  

(4.52)

\[
c_{p_{water}} = 2477.837 + 3.731 \times T
\]  

(4.53)

\[
K_{water} = 0.3575 + 7.67 \times 10^{-4} \times [T - 273] - 4.08 \times 10^{-6} \times [T - 273]^2
\]  

(4.54)

\[
\log (\mu_{water}) = 3.338 \times 10^{-9} \times [T - 273]^4 - 1.4252 \times 10^{-6} \times [T - 273]^3 + 2.386 \times 10^{-4} \times [T - 273]^2 - 0.026804 \times [T - 273] - 1.744
\]  

(4.55)

4.2.4 Coolant Thermal Models

The coolant thermal model covers the mixing dynamics in the cooling system as well as all heat sources and heat sinks apart from the EGR cooler. This division in the model was made because the dynamics of the coolant system generally have a much larger time constant than those of the EGR cooler. Because thermal transients are not a large
concern in the engine oil cooler and transmission oil heater, the model utilizes simple, steady-state $\varepsilon$-NTU heat exchanger equations [4.5]. The mixing dynamics are handled by differential equations that sum the flow of energy into and out of each volume.

There are two coolant volume temperatures that are computed by the coolant thermal model. The first is directly after the engine. The water temperature in this volume is used to determine heat transfer rates in each of the downstream heat exchanger models: EGR cooler, engine oil cooler, transmission oil cooler, radiator, and cabin heater core.

\[
\frac{dT_{\text{HEX}}}{dt} = \frac{m_{\text{pump}}(T_{\text{eng}} - T_{\text{HEX}})}{V_{\text{HEX}} \rho_{\text{water}}} \tag{4.56}
\]
A similar formula was used for the coolant volume downstream of the five heat exchangers but before the water pump. This is the fluid temperature that is used to compute both the heat transfer within the engine as well as the mass flow rate from the pump.

\[
\frac{dT_{\text{pump}}}{dt} = \frac{1}{V_{\text{pump}}p_{\text{water}}} \left( m_{\text{EGR}}T_{\text{EGR}} + m_{\text{EOC}}T_{\text{EOC}} + m_{\text{TOH}}T_{\text{TOH}} + m_{\text{heater}}T_{\text{heater}} + m_{\text{rad}}T_{\text{rad}} - m_{\text{eng}}T_{\text{pump}} \right)
\]  

After leaving the pump, the first heat exchanger that the coolant encounters is the engine. Heat from combustion is rejected to the coolant flowing through the cooling jackets and cylinder heads. The rate of heat rejection to the coolant for the engine under consideration was known at various engine speeds and loads. This data, however, was obtained from a fully-warmed engine and does not accurately represent the conditions found during an engine’s warm-up period. In a cold engine, the large thermal inertia of the block and head draws much of the energy that would otherwise be carried away by the coolant. To approximate this effect in the model, the heat transfer rates measured for a fully-warmed engine are scaled linearly based on the current coolant temperature \( T_{\text{pump}} \) and the engine’s steady-state operating temperature \( T_{\text{warm}} \).

\[
\dot{Q}_{\text{eng-water}} = f(RPM, \text{Torque}) \cdot \frac{T_{\text{pump}} - 273}{T_{\text{warm}} - 273}
\]

After leaving the engine the coolant can flow through one of five heat exchangers. One of these, the EGR cooler, was described in detail in the previous section. The next two heat exchangers, in terms of importance to the exhaust heat recovery system’s goals, are
the engine oil cooler and the transmission oil heater. The models of these two devices, which differ only in their inputs and calibrated coefficients, employ steady-state ε-NTU models, namely calculating an exchanger’s effectiveness based on fluid inlet temperatures and flow rates and using that to determine the outlet temperatures [4.5].

\[
\varepsilon = \frac{\dot{m}_{\text{water}}c_{p\text{water}}(T_{\text{water,in}} - T_{\text{water,out}})}{\dot{C}_{\text{min}}(T_{\text{water,in}} - T_{\text{water,in}})} = \frac{\dot{m}_{\text{oil}}c_{p\text{oil}}(T_{\text{oil,out}} - T_{\text{oil,in}})}{\dot{C}_{\text{min}}(T_{\text{water,in}} - T_{\text{oil,in}})} \tag{4.59}
\]

The terms \(\dot{C}_{\text{min}}\) and \(\dot{C}_{\text{max}}\) are defined as the lesser and greater, respectively, of the thermal capacitances of the two fluids. The capacitance ratio, \(CR\), is the ratio between \(\dot{C}_{\text{min}}\) and \(\dot{C}_{\text{max}}\).

\[
\dot{C}_{\text{water}} = \dot{m}_{\text{water}}c_{p\text{water}} \quad \text{and} \quad \dot{C}_{\text{oil}} = \dot{m}_{\text{oil}}c_{p\text{oil}} \tag{4.60}
\]

The effectiveness of the exchangers is defined as:

\[
\varepsilon = \frac{1 - \exp\left(-\frac{\{UA\}}{\dot{C}_{\text{min}}}[1 - CR]\right)}{1 + CR} \tag{4.61}
\]

The overall heat transfer rate for the exchanger, \(\{UA\}\), is made up of the convective heat transfer rate between the coolant and the wall, conduction through the wall, and convection from the wall to the oil. The cumulative effect is computed similarly to the net resistance of a set of resistors in parallel.

\[
\{UA\} = \left(\frac{1}{A_{\text{wall}}h_{\text{oil}}} + \frac{\delta_{\text{wall}}}{K_{\text{wall}}A_{\text{wall}}} + \frac{1}{A_{\text{wall}}h_{\text{water}}}\right)^{-1} \tag{4.62}
\]
In the previous equation, $\delta_{\text{wall}}$ is the thickness of the walls within the exchanger, $K_{\text{wall}}$ is the thermal conductivity of the wall material, and $A_{\text{wall}}$ is the total heat exchange surface. $h_{\text{oil}}$ and $h_{\text{water}}$ are the convection coefficients for oil-to-wall and water-to-wall heat transfer, respectively. The convection coefficient is determined based on several physical properties of the fluid and its flow characteristics.

$$h_{\text{fluid}} = \frac{K_{\text{fluid}} Pr_{\text{fluid}}^{1/3} \left( \dot{m}_{\text{fluid}} \right)^{\beta_{\text{fluid}}}}{C_{\text{fluid}} \mu_{\text{fluid}}^2 A_{\text{fluid}}}$$  \hspace{1cm} (4.63)

In the above equation, $K_{\text{fluid}}$ is the thermal conductivity of the fluid, $\dot{m}_{\text{fluid}}$ is the mass flow rate, $A_{\text{fluid}}$ is the cross-sectional area through which the fluid is flowing, $\mu_{\text{fluid}}$ is the dynamic viscosity, $C_{\text{fluid}}$ and $\beta_{\text{fluid}}$ are calibration terms that can adjust the model to match experimental data, and $Pr$ is the fluid’s Prandtl number, a ratio of physical diffusivity to thermal diffusivity. In the below equation, $c_{p_{\text{fluid}}}$ represents the specific heat of the fluid. Based on several sources of supplier data, Scott [4.10] fit equations describing the thermal and physical properties of engine oil and transmission oil.

$$Pr_{\text{fluid}} = \frac{c_{p_{\text{fluid}}} \mu_{\text{fluid}}}{K_{\text{fluid}}}$$  \hspace{1cm} (4.64)

$$K_{EO} = .1721 - 9.46 \times 10^{-5} \times T_{EO}$$  \hspace{1cm} (4.65)

$$\mu_{EO} = \exp \left( -122.69 + \frac{16683}{T_{EO}} + .294 \times T_{EO} - 2.637 \times 10^{-4} \times T_{EO}^2 \right)$$  \hspace{1cm} (4.66)

$$c_{p_{EO}} = 817 + 3.657 \times T_{EO}$$  \hspace{1cm} (4.67)

$$K_{TO} = .1742 - 1.091 \times 10^{-4} \times T_{TO}$$  \hspace{1cm} (4.68)
\[
\mu_{T_0} = \exp \left( -79.48 + \frac{11422}{T_{T_0}} + 0.1699 \times T_{T_0} - 1.421 \times 10^{-4} \times T_{T_0}^2 \right) \tag{4.69}
\]
\[
c_{p_{T_0}} = 838 + 3.77 \times T_{T_0} \tag{4.70}
\]

Rather than implementing these heat exchanger models in Simulink, however, the above equations were used to populate lookup tables that calculate fluid outlet temperatures based on the four inputs: inlet temperature and flow rates of both fluids involved. This simplification shortens computation time and lessens the likelihood of a simulation error occurring.

To complete the oil systems, oil flow rates and temperature dynamics needed to be added. Several assumptions and approximations were made for each of the oil circuits. However, it is not critical to this work that the dynamics of the lubricating fluids be modeled to a high level of accuracy because the oil systems, essentially, act only as disturbances on the exhaust heat recovery system. For the engine oil system, a flow rate coefficient of \(0.1 \frac{LPM}{RPM}\) was multiplied by the engine speed to come up with the volumetric flow rate of engine oil. This was multiplied by the density of the oil to find the mass flow rate.

\[
\dot{m}_{EO} = \frac{0.1 \times RPM}{60 \left( \frac{sec}{min} \right)} \times \frac{\rho_{EO}}{1000 \left( \frac{L}{m^3} \right)} \tag{4.71}
\]
\[
\rho_{EO} = \frac{1}{9.9 \times 10^{-4} + 3.64 \times 10^{-7} \times T_{EO} + 8.46 \times 10^{-10} \times T_{EO}^2} \tag{4.72}
\]
In addition to exchanging heat with the engine oil cooler, the engine oil receives heat
from the engine block. This heat transfer was approximated to be 15% as much as the
transfer from the engine to the coolant. Simple temperature mixing dynamics are used to
calculate the average oil temperature given both the oil temperature exiting the engine oil
cooler and the heat rejection from the engine.

\[
\frac{dT_{EO}}{dt} = \frac{m_{EO}c_{PEO}(T_{EO,EOC,\text{out}} - T_{EO}) + \dot{Q}_{EO,\text{eng}}}{m_{EO}c_{PEO}}
\]  
\(4.73\)

The flow rate of oil in the transmission system was approximated at a constant five liters
per minute based on the recommendation of the supplier. This was multiplied by the
density of the transmission fluid to find the mass flow rate.

\[
m_{TO} = 5 \text{ (LPM)} \ast \frac{\rho_{TO}}{1000 \left(\frac{\text{L}}{\text{m}^3}\right)}
\]  
\(4.74\)

\[
\rho_{TO} = \frac{1}{9.82 \ast 10^{-4} + 4.75 \ast 10^{-7} \ast T_{TO} + 7.59 \ast 10^{-10} \ast T_{TO}^2}
\]  
\(4.75\)

Like the engine oil, the transmission fluid also draws heat from the component it
lubricates. It was estimated that the transmission dissipates 10% of the engine’s brake
power and that of that 10%, 50% is absorbed by the transmission fluid. Again, simple
temperature mixing dynamics are used to calculate the average transmission oil
temperature.

\[
\dot{Q}_{TO,\text{trans}} = \frac{\tau_{\text{brake}} \ast RPM \ast \pi}{30} \ast .05
\]  
\(4.76\)

\[
\frac{dT_{TO}}{dt} = \frac{m_{TO}c_{P_{TO}}(T_{TO,TOH,\text{out}} - T_{TO}) + \dot{Q}_{TO,\text{trans}}}{m_{TO}c_{P_{TO}}}
\]  
\(4.77\)
The radiator does not begin to play a role in the thermal management system until the coolant has reached its fully-warmed operating temperature. Because this work is only concerned with the period before the coolant reaches this condition, the radiator was not modeled in detail. Instead, its behavior was idealized for simplicity. Assuming a constant air speed (as is the case for some dynamometer-based fuel economy tests), a constant ambient temperature, and a relatively constant coolant inlet temperature (102°C to 108°C), a constant coolant outlet temperature of 90°C was estimated. The radiator was, however, given full modeling consideration as a flow restriction in the coolant flow model.

The cabin heater is not utilized during the FTP drive cycles so its thermal effect on the system was not studied or modeled. Like the radiator, the cabin heater core was modeled as a flow restriction in the coolant flow model.

4.2.5 Supplementary Models

The exhaust heat recovery system models described above cannot function by themselves alone. An engine model is required to supply the necessary exogenous inputs. This engine model includes an intake throttle and manifold model, a cylinder breathing model, and an exhaust temperature model.
The throttle model uses the same valve model described in the exhaust flow network section. The pressure ratio between the intake manifold and the atmosphere is used to determine whether or not the throttle is producing choked flow. Based on that calculation, one of two equations is used to compute the flow rate through the throttle. The throttle position was mapped from experimental data into a two-dimensional lookup table. The inputs to that table are engine speed and brake torque.

If \( \frac{p_{IM}}{p_{amb}} \geq \left( \frac{2}{\gamma+1} \right)^{\gamma-1} \)

then

\[
\dot{m}_{air,throttle} = \frac{C_d(\alpha) A_{ref} p_{amb}}{\sqrt{R_{air} T_{amb}}} \left( \frac{p_{IM}}{p_{amb}} \right)^{\gamma} \sqrt{\frac{2\gamma}{\gamma - 1} \left[ 1 - \left( \frac{p_{IM}}{p_{amb}} \right)^{\gamma-1} \right]} 
\]

If \( \frac{p_{IM}}{p_{amb}} < \left( \frac{2}{\gamma+1} \right)^{\gamma-1} \)

then

\[
\dot{m}_{air,throttle} = \frac{C_d(\alpha) A_{ref} p_{amb}}{\sqrt{R_{air} T_{amb}}} \sqrt{\gamma} \sqrt{\frac{2}{\gamma + 1}} \left( \frac{\gamma+1}{\gamma-1} \right)^{\gamma-1} \]

Figure 29. Engine Model
The intake manifold uses a simple differential equation to calculate the mass of gases in the volume. This, coupled with the ambient air temperature, is used to calculate the pressure in the manifold by means of the ideal gas law. A value of $287 \frac{J}{kg \cdot K}$ was used for the specific gas constant of air ($R_{air}$).

$$\frac{dm_{IM}}{dt} = m_{throttle} + m_{EGR} - m_{cyl}$$  \hspace{1cm} (4.80)

$$p_{IM} = \frac{m_{IM} R_{air} T_{amb}}{V_{IM}}$$  \hspace{1cm} (4.81)

The pressure in the manifold is then used to compute the air mass flow rate into the cylinders. This flow rate is directly proportional to the engine’s speed, displacement, and volumetric efficiency. The volumetric efficiency is estimated as a function of intake manifold pressure and engine speed using a relationship developed by Hendricks and Sorenson [4.11]. The terms $s_i(RPM)$ and $y_i(RPM)$ are calibration coefficients for the volumetric efficiency relationship. These terms are functions of engine speed, thus accounting for the speed dependence of the volumetric efficiency.

$$m_{cyl} = \lambda_v \frac{p_{IM} V_d}{R_{air} T_{amb}} \frac{RPM}{120}$$  \hspace{1cm} (4.82)

$$\lambda_v = s_i(RPM) - \frac{y_i(RPM)}{p_{IM}}$$  \hspace{1cm} (4.83)

After the cylinder breathing model calculates the mass flow rate of air into the cylinders, that value is used along with the air/fuel ratio (AFR) to compute the mass flow rate of
exhaust gases out of the cylinder. The AFR is found using a lookup table populated with experimental data.

\[ m_{exh, eng} = m_{cyl} \times (1 + AFR^{-1}) \quad (4.84) \]

The mass flow rate of air into the cylinders and AFR are also used to estimate the exhaust temperature based on a 2\textsuperscript{nd} order approximation.

\[ T_{exh, eng} = a_0 + a_1 m_{cyl} + a_2 m_{cyl}^2 + a_3 AFR + a_4 AFR^2 + a_5 AFR \times m_{cyl} \quad (4.85) \]

4.3 Sub-model Calibration and Validation

The exhaust heat recovery system model has several calculations that are not strictly predictable based on physical constants or that require more information than was given. In these situations the missing information can be inferred or the physical phenomena can be approximated through proper model calibration. Models and calculations that required calibration were:

- the exhaust and coolant flow restriction of the EGR cooler,
- the coolant flow restriction of the engine oil cooler,
- the coolant flow restriction of the transmission oil heater
- the performance of the coolant pump,
- the thermal performance of the EGR cooler,
- the thermal performance of the engine oil cooler,
- the thermal performance of the transmission oil heater,
\begin{itemize}
  \item the throttle position model,
  \item the air flow restriction of the throttle,
  \item the volumetric efficiency model,
  \item the air/fuel ratio model, and
  \item the exhaust temperature model.
\end{itemize}

4.3.1 Description of Calibration Data

In order to properly calibrate a model so that it can accurately represent the performance of the system on which it is based, experimental data is needed for each of the components or systems modeled. Some of the components modeled have been in use for many years and, therefore, experimental data exists for those parts. Much of the exhaust heat recovery system, however, is new and untested, meaning that little experimental data has been collected for those components.

The supplier of the EGR cooler provided simulated test results for the thermal performance and flow characteristics. This data included inlet and outlet temperatures for both fluids, fluid flow rates and fluid pressure drops. The vehicle’s manufacturer was able to provided limited experimental results on the thermal performance and flow restrictions of the engine oil cooler and transmission oil heater. This data contained flow rates and pressure drops, fluid inlet temperatures and heat transfer rates. Documentation on the coolant pump included a few hundred experimental data points for which the pump speed, pressure rise and volumetric flow rate had been recorded.
Information was in much greater supply for the engine than for the thermal management system. Extensive engine testing by the vehicle’s manufacturer provided copious data with which to calibrate and validate each facet of the engine model. The engine modeled here is a Chrysler Pentastar 3.6 liter V-6.

<table>
<thead>
<tr>
<th>Engine Specifications</th>
</tr>
</thead>
<tbody>
<tr>
<td>Layout</td>
</tr>
<tr>
<td>Displacement</td>
</tr>
<tr>
<td>Max Power</td>
</tr>
<tr>
<td>Max Torque</td>
</tr>
<tr>
<td>Fuel</td>
</tr>
<tr>
<td>Injection</td>
</tr>
<tr>
<td>Bore &amp; Stroke</td>
</tr>
<tr>
<td>Compression Ratio</td>
</tr>
</tbody>
</table>

Table 2. Engine Specifications

The available experimental data for the engine came in two distinct varieties. The first was a steady-state test grid that listed, amongst a multitude of other values, throttle position, intake manifold pressure, mass air flow rate, and air/fuel ratio for given engine speed and brake torque values. Figure 30 shows the range of engine speeds and torque values at which the engine was tested.
The second set of information was transient data recorded during an FTP city driving cycle test. The key values from this test were engine speed and brake torque traces for the driving cycle, as well as exhaust temperatures and flow rates at each point in the cycle. The engine speed and brake torque profiles obtained from this data serve as the two exogenous inputs required by the engine model. Although the FTP city driving cycle is 1874 seconds long, this work focuses only on the first 600 seconds because it is primarily concerned with the behavior of vehicle thermal systems during warm-up.
4.3.2 Exhaust Flow Network

The EGR cooler documentation included 96 points for which fluid temperatures, flow rates, inlet pressures, and pressure drops were known. This data was used to solve for the
two flow coefficients used in the model. Exhaust gas temperature and pressure were used to find values for the fluid properties density, $\rho_{exh}$, and dynamic viscosity, $\mu_{exh}$.

As shown in the equation below, by plotting $\frac{\dot{m}_{exh}}{\rho_{exh}}$ against $\frac{\delta p_{exh,EGRp_{exh}}}{\dot{m}_{exh} \mu_{exh}}$ linear regression can be used to solve for the flow coefficients, $C_{1exh}$ and $C_{2exh}$. Using 60% of the available data points resulted in the values $9.319 \times 10^7$ (m$^3$), and $3.273 \times 10^5$ (m$^4$), respectively.

$$\frac{\delta p_{exh,EGRp_{exh}}}{\dot{m}_{exh} \rho_{exh}} = C_{1exh} + C_{2exh} \frac{\dot{m}_{exh}}{\mu_{exh}}$$

(4.86)

![Graph](image)

Figure 33. Exhaust Flow Calibration, EGR Cooler

The remaining data points, those not used for calibration, were used to validate the model. As shown below, the model is able to predict pressure drop values within roughly 10% of the experimentally measured value, as calculated by Equation 4.86. $X$ represents the value of interest and $n$ is the number of data points in the sample.
The data points were used to determine the effective area of the heat exchanger with the below equation. The results were averaged and gave an effective area of \( 7.673 \times 10^{-5} \, m^2 \).

\[
A_{eff} = \frac{\dot{m}_{\text{water}}}{\sqrt{2\rho_{\text{water}}(\delta p)}}
\]  

(4.88)
While there were not enough data points given to validate the effective area, the four individual values that resulted from the above equation were in strong agreement; none differed from the average by more than 4.5%.

Similarly, the documentation for the transmission oil heater included only three points of coolant flow data. This, too, was at a single temperature and, therefore, the same calibration procedure used for the engine oil cooler was followed here. The resulting average effective area is 1.02E-4. The largest difference between any of the individual $A_{eff}$ values and the average was 10.4%.

The coolant pump is modeled as a two-dimensional lookup table based on pump speed and pressure rise across the pump. A large amount of experimental data was available for this model and was used to populate the table, as shown below. Linear interpolation was used to estimate flow values at missing data points.

![Coolant Pump Flow Rate vs. Pump Speed and Pressure Rise](image)

**Figure 35.** Coolant Pump Flow Rate vs. Pump Speed and Pressure Rise
The data for the pump was then used to validate the lookup table’s accuracy. The results of this validation can be seen in Figure 36. Other than a few outliers, the relationship between measured and estimated values is very strong, with an average error of less than 5.5%.

![Figure 36. Coolant Pump Flow Rate Validation](image)

The EGR cooler documentation included 96 points for which coolant temperatures, flow rates, and pressure drops are known. This data was used to solve for the two flow coefficients used in the model. First, the temperature was used to assign a coolant density and dynamic viscosity, $\rho_{water}$ and $\mu_{water}$, to each data point. Plotting $\frac{\dot{m}_{water}}{\mu_{water}}$ against $\frac{\delta p_{water}\rho_{water}}{m_{water}\mu_{water}}$ for the calibration data, along with the corresponding linear trend line, yielded the flow coefficients. $C_{1\text{water}}$ was given by the y-intercept of the resulting trend line and $C_{2\text{water}}$ by the slope. Using 60% of the available data points resulted in the values $2.107\times10^7$ (m$^3$) and $4.376\times10^6$ (m$^4$), respectively.
\[
\frac{\delta p_{\text{water}} \rho_{\text{water}}}{m_{\text{water}} \mu_{\text{water}}} = C_{1_{\text{water}}} + C_{2_{\text{water}}} \frac{m_{\text{water}}}{\mu_{\text{water}}}
\]  \hspace{1cm} (4.89)

Again, the data points not used for calibration were used to validate the model. Coolant pressure drop through the EGR cooler is able to be predicted within 2% of the experimentally measured value, on average.

Figure 37. Coolant Flow Calibration, EGR Cooler
4.3.4 EGR Cooler Thermal Model

Supplier data for the EGR cooler thermal included 96 data points, each consisting of fluid flow rates, inlet and outlet temperatures, and exhaust inlet pressure. Half of these data points were used for the calibration of the heat transfer coefficients used in the model. The other half were reserved for validating the resulting model.

The calibration process for the EGR cooler thermal model involved adjusting the coefficients and exponents, simulating the conditions of each of the calibration data points, and analyzing the results. The two metrics used to determine the goodness of a set of coefficients were, 1) the mean error between the predicted exhaust gas temperature drop from inlet to outlet and the manufacturer supplied values, and 2) the mean error between the predicted coolant temperature rise and the corresponding supplier data.

During the calibration process, as the coefficient values decreased, so did the mean error. Because of the length of the heat exchanger, the large heat transfer surface area, and the
counterflow configuration, the EGR cooler has an extremely high effectiveness. For this reason, the best results were found when the thermal resistance to convection on both the exhaust and gas side was zero and the only remaining resistance was that of conduction through the wall.

With each of the convective resistances at essentially zero (actually slightly above zero, 1E-5, because a value of zero would cause an error in the model), the average error on the coolant side was 5.1%, or 0.7°C. The exhaust side error was 0.9%, or 6.1°C.

Figure 39. EGR Cooler Thermal Model Calibration, Coolant
Testing the model against the remaining steady-state data points proved that the model is very adept at predicting the respective rise and fall of the coolant and exhaust temperatures. The model is able to predict coolant temperature changes to within 4.3%, or 0.6°C, on average. The results are even better on the exhaust side with the average error being just over 1%, or 6.8°C.
4.3.5 Coolant Thermal Models

Documentation for the engine oil cooler contained sixteen points for which the coolant and oil flow rates, inlet temperatures and heat transfer rates were known. Half of these were used to calibrate the heat transfer coefficients.

The engine oil cooler’s effectiveness was calculated at each calibration data point using the known temperature differences and fluid thermal capacities. Linear regression was used to calculate the optimal value of the heat transfer coefficients ($C_{1,\text{oil}}$ and $C_{2,\text{water}}$) for each input value of $\beta_{\text{oil}}$ and $\beta_{\text{water}}$. Those values were then used to predict the exchanger’s effectiveness at each data point. The standard deviation in the error between the experimental effectivenesses and the calculated effectivenesses was used as the metric to find the ideal values of $\beta_{\text{oil}}$ and $\beta_{\text{water}}$.

The best results were obtained with the coefficient and exponent values listed in Table 3.
<table>
<thead>
<tr>
<th>Constant</th>
<th>$C_{oil}$</th>
<th>$C_{water}$</th>
<th>$\beta_{oil}$</th>
<th>$\beta_{water}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Value</td>
<td>1.44</td>
<td>1.77</td>
<td>.685</td>
<td>.520</td>
</tr>
</tbody>
</table>

Table 3. Engine Oil Cooler Thermal Model Calibration Values

With the heat transfer coefficients calculated, the $\epsilon$-NTU model was then used to populate a four-dimensional lookup table. The eight remaining supplier data points were compared to the outputs of this table. On average the table was accurate to within 1.4%.

![Figure 43. Engine Oil Cooler Thermal Model Validation](image)

The same process was used to calibrate and validate the transmission oil heater thermal model. In this case, the documentation contained only ten data points so the calibration and validation each relied on only five points. Nonetheless, the results were good, with the transmission oil heater lookup table predicting heat transfer rates with an average error of only 5.2% using the values in Table 4.
<table>
<thead>
<tr>
<th>Constant</th>
<th>$C_{oil}$</th>
<th>$C_{water}$</th>
<th>$\beta_{oil}$</th>
<th>$\beta_{water}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Value</td>
<td>0.360</td>
<td>0.851</td>
<td>.44</td>
<td>.50</td>
</tr>
</tbody>
</table>

Table 4. Transmission Oil Heater Thermal Model Calibration Values

Figure 44. Transmission Oil Heater Thermal Model Validation

### 4.3.6 Supplementary Models

The engine model was calibrated in logical order, beginning with the throttle and moving towards the exhaust. Steady-state data was used to map the throttle position as a function of engine speed and brake torque, resulting in Figure 45.
Values for air mass flow rate and intake manifold pressure at each of the steady-state operating points were used to compute the throttle’s effective flow area as a function of throttle position and engine speed. The ambient and intake manifold pressures were first compared to determine whether or not the throttle was producing choked flow. Then, depending on the result, the appropriate flow equation was used to find the discharge coefficient of the throttle.

If \( \frac{p_{IM}}{p_{amb}} \geq \left( \frac{2}{\gamma + 1} \right)^{\frac{\gamma}{\gamma - 1}} \)

then

\[ C_d(\alpha) = \frac{\dot{m}_{air,throttle} \sqrt{R_{air} T_{amb}}}{A_{ref} p_{amb}} \left( \frac{p_{IM}}{p_{amb}} \right)^{\frac{1}{\gamma}} \left( \frac{2\gamma}{\gamma - 1} \left[ 1 - \left( \frac{p_{IM}}{p_{amb}} \right)^{\frac{\gamma - 1}{\gamma}} \right] \right)^{-1} \]

(4.90)
If \( \frac{P_{IM}}{P_{amb}} < \left( \frac{2}{y+1} \right)^{\frac{y}{y-1}} \)

then

\[
C_d(\alpha) = \frac{\dot{m}_{air,throttle} \sqrt{R_{air} T_{amb}}}{A_{ref} P_{amb}} \left( \sqrt{y} \left( \sqrt{\frac{2}{y+1}} \right)^{\frac{y}{y-1}} \right)^{-1}
\]

The steady-state experimental data was used to populate a two-dimensional lookup table for the throttle’s equivalent area, the product of its variable discharge coefficient and its fixed reference area. The two inputs to this table are the engine speed and the throttle position. Plotting the table’s values versus its indices gives the following graph. The plot has been normalized based on the throttle’s reference area.

![Figure 46. Throttle Valve Equivalent Area vs. Engine Speed and Throttle Position](image-url)
The table’s accuracy was validate against the experimental data, as well. As seen in Figure 47, the agreement between the predicted and experimental values is extremely strong.

Figure 47. Throttle Effective Area, Validation Results

For the volumetric efficiency (VE) model, as with the throttle area model, the steady-state experimental engine data was used for calibration. The initial step in this calibration process was to determine the VE at each data point. Volumetric efficiency is the ratio of the actual flow rate of air into the cylinders divided by the maximum rate at which the engine can intake air. The maximum intake rate is found using the ideal gas law, the engine displacement, and the engine speed. For every two revolutions, the engine can, at most, intake a volume of air equal to its displacement. The ideal gas law computes the mass of that volume of air. Note that in this equation the engine speed has the unit revolutions per second. The resulting VE profile is also shown below.
The Hendricks-Sorenson equation contains two coefficients, $s_i(N)$ and $y_i(N)$, which are functions of the engine speed. Determining the expressions of these two functions was the next step in calibrating the VE model. After grouping all of the calibration data points into sets based on common engine speeds, the product of the intake manifold pressure and the volumetric efficiency was plotted against the intake manifold pressure, $p_{IM} \cdot \lambda_v$ vs. $p_{IM}$, for a fixed engine speed. The resulting graph had a linear trend, as seen in Figure 49.
The slope of this trend line was the the $y_i$ value at that engine speed. The $s_i$ value was given by the y-intercept. This process was repeated with each of the fixed engine speed data sets. All of the $s_i$ and $y_i$ values were then plotted against their corresponding engine speeds. This gave the trends for $s_i(N)$ and $y_i(N)$. The coefficients obtained from the experimental data were $s_0 = 0.796$, $s_1 = 5.846 \times 10^{-5}$, $y_0 = 19.042$, and $y_1 = -2.152 \times 10^{-3}$. 

Figure 49. Calibration of Coefficients, Volumetric Efficiency Model
The VE model was then validated using the steady-state data points. The expression found for the VE was used to compute the mass flow rate of air into the cylinder at each validation data using Equation 4.81, the given engine speed, and the given intake manifold pressure. When compared to the experimentally measured values for air mass flow, it is evident that the VE model, and, therefore, the cylinder breathing model, has a high level of accuracy.

Figure 50. Volumetric Efficiency Model Calibration
The air/fuel ratio (AFR) is estimated by a two-dimensional lookup table that was populated with steady-state engine data, as shown in Figure 52.

Figure 51. Volumetric Efficiency Model Validation Results

Figure 52. Air/Fuel Ratio vs. Engine Speed and Brake Torque
In order to simplify the model, the combustion effects caused by spark timing, valve timing and fuel enrichment during warm-up were nominally ignored. The exhaust temperature was, instead, correlated to the mass flow rate of air entering the cylinders and the air/fuel ratio. Transient, drive cycle engine data was used to calibrate a polynomial function relating by means of linear regression. An ordinary least squares estimation approach was used to find the appropriate coefficient for each of the terms in the second order fit. The dependent variable in this regression \( y \) is a vector of the exhaust temperature values at each of the experimental data points. The regressor \( X \) is a matrix of consisting of the combinations of independent variables found in the polynomial function for exhaust temperature at each experimental data point. Those independent variables are the \( \dot{m}_{cyl} \) and the AFR. The regression coefficients, given by the vector \( \beta \), are the coefficients used in the polynomial function for exhaust temperature. Equation 4.92 is used to find the regression coefficient values that minimize the sum of the squared errors for the data set. The error is defined as the difference between the experimental value for the exhaust temperature at each data point and the predicted value given the experimental values of the \( \dot{m}_{cyl} \) and the AFR as inputs. Equation 4.95 includes the optimized regression coefficients.

\[
\beta = (X'X)^{-1}X'y
\]  

(4.93)
Figure 53 shows the results of the mean-value engine model validation. The FTP city drive cycle was performed and the model’s predictions were compared to the experimental data. The profiles shown here (the exhaust flow rate and the exhaust temperature) are the inputs that are necessary for the EHRS simulator to run properly. The exhaust temperature is estimated to within 7.7%, on average. The error for the exhaust flow is quite high, roughly 30% for any given sample, but the trends and magnitudes appear to be fairly well in line with the experimental data overall. Note, some of the experimental data is of questionable integrity, exhibiting behavior that does
not make physical sense. Obvious sensor errors were expunged from the data, however, minor errors in the data remain.

The result of developing all of these subsystem models is a tool capable of simulating the operation of a vehicle thermal management system with special attention paid to the exhaust heat recovery system. This model was used to simulate an FTP city cycle starting from 30°C with exhaust gases recirculation (EGR) disabled and also with EGR enabled to show that the model’s results make sense, physically.

![Figure 54. Coolant Warm-up, Model Validation](image)

Although a direct validation of the complete model was not possible with the available data, personal correspondence with the vehicle’s manufacturer confirmed that the warm-up time with no EGR, 348 seconds, appeared to be reasonable and that the warm-up time with EGR, 272 seconds, was similar to preliminary results from their own simulations.
The additional lines on Figure 54, at 100°C and 105°C, are the temperature to which warm-up time is measured and the fully-warmed operating temperature, respectively.

In this chapter a working model on an exhaust heat recovery system was developed. The EHRS model is comprised of four sub-models: the exhaust flow network, the coolant flow network, the EGR cooler thermal model, and the cooling system thermal models. In addition to these models, a mean-value engine model was created to supply the EHRS with the necessary inputs. Individual components of the model were calibrated with experimental data when available. If the amount of data for a component was sufficient, a portion was set aside to serve as unique data against which the model could be validated. The models that were able to be validated with experimental data were typically accurate to within 5% of the experimental results.

In the next chapter, this model will be applied to analyze the potential for improvements in coolant warm-up time. It will then serve as the plant model with which to develop an open-loop control strategy for the EGR bypass and engine back pressure valves to optimize coolant warm-up time.
4.4 References


[4.9] Alshamani, K. “Equations for Physical Properties of Automotive Coolants”; 2003 SAE World Congress; March 2003; Detroit, Michigan; USA.


Chapter 5

SYSTEM ANALYSIS AND CONTROL DEVELOPMENT FOR AN EXHAUST HEAT RECOVERY SYSTEM

A control problem needs to be understood from all angles before an encompassing, cohesive control strategy can be created. This means fully defining the objectives of the control, the constraints of the system in question, and the tools that are available to solve the problem. The system on which the controller will operate needs to be analyzed to understand its dynamics and the full range of the conditions that the controller is able to bring about. Only once all of these things are understood can a control strategy be formed fully.

5.1 Definition of the Control Problem

Before a control strategy can be developed, several questions about its role need to be answered. What are the sensors and actuators in the system to which the control will have access; what information will be available with which to make decisions and how can the controller implement those decisions? What is the overall objective of the controller? What are the system’s boundaries or constraints which the controller will need to ensure are not violated? What are the controller’s own limitations? These can include the ranges and the speeds of the actuators that the controller manipulates. Lastly,
what are the disturbances on the system for which the controller will need to account and compensate?

5.1.1 Sensors and Actuators in the Plant

The plant model on which the controller will be acting is a thermal management system model that includes a vehicle coolant system, an exhaust system, and an exhaust heat recovery system (EHRS). The EHRS links the conventionally separate exhaust and coolant systems. It consists of an exhaust-to-coolant heat exchanger (the EGR cooler), two valves for routing the flow of exhaust gas (the engine back pressure valve and the EGR bypass valve), and a valve that controls the flow of coolant through the EGR cooler (the EGR coolant valve). There are several more actuators that can be controlled in the exhaust system and the coolant system, however, the scope of this control problem is limited to two actuators: the EGR bypass valve and the engine back pressure valve. With these two valves, the controller will be able to manage the flow of exhaust gas through the EGR cooler and the back pressure on the engine.
To aid the controller in making the decisions on how best to position these valves, all of the information monitored by the engine control unit (ECU) will be available. The ECU variables that will be most useful to the exhaust heat recovery system (EHRS) controller are those which affect the flow rates and temperatures of the exhaust and coolant, including the throttle position, the engine speed, the air mass flow rate into the intake manifold (MAF), the intake manifold pressure (MAP), the air/fuel ratio (AFR), and the coolant temperature measured upstream of the EGR cooler and the other heat exchangers.
5.1.2 Controller Objective and Constraints

Using the information available to it, the EHRS controller’s objective will be to minimize the time required to warm the coolant to 100°C, based on the engine manufacturer’s specification. The way in which the controller affects coolant warm-up time is by directing exhaust gas towards or away from the EGR cooler. The EGR cooler recovers heat from the exhaust and transfers it to the coolant. Sending more exhaust flow through the EGR cooler increases the rate at which heat can be transferred to the coolant. Based on this alone, the solution to this problem seems simple; in order to reduce coolant warm-up time, the controller should recover as much heat from the exhaust gases as the effectiveness of the EGR cooler will allow.

There are, however, a few constraints which limit the range of conditions that the controller is able to command. The first constraint concerns the exhaust back pressure on the engine. Altering the flow of exhaust gases can have negative side effects on the engine’s operation. Higher exhaust back pressures on the engine increase the pumping work required to empty the cylinder during the exhaust stroke. This additional work could potentially offset any efficiency gains resulting from quicker coolant warm-up. For this reason, the controller cannot increase the back pressure on the engine above the levels it would have experienced had a conventional exhaust system been used.

The other constraint on the system is due to the EGR cooler’s physical limitations. The manufacturer of the EGR cooler has determined that there are certain operating conditions which will place great stress on the device and contribute to substantially
shortened part life. In particular, the concept of an energy ratio, introduced in Chapter 4, is here used as a metric for gauging the severity of a given operating condition. The energy ratio is the ratio of the recoverable energy entering the device divided by the amount of energy required to bring the coolant stream to its boiling point. Based on empirical data and historical trends, the manufacturer has recommended a maximum threshold of 45% for the energy ratio in order to avoid spot boiling and the localized overheating of the tube walls that it can cause.

The final constraint concerns the actuators that the controller operates on. These valves have some delay in reacting to control inputs; they cannot jump from one position to another instantaneously. For this reason, the desired control output will be filtered by a transfer function with a one second time constant. This will ensure that the actual control output is continuous and smooth, thus allowing the valves to stay in synch with the controller.

5.1.3 Disturbance Considered in the Control Problem

While the controller is doing everything within its authority to accomplish its objective, reducing the coolant warm-up time, and to respect the system’s constraints, there will be disturbances on the system to which the controller will need to respond quickly and appropriately. Most of these disturbances will occur on the coolant side of the system. A change in the position of any of the valves in the coolant flow paths will affect the coolant flow rate and pressure at the EGR cooler. Additionally, heat taken from the coolant or rejected to the coolant by any of the other heat exchangers in the system will
affect the conditions in the EGR cooler. The only real disturbance on the exhaust gas side will be the engine’s requirement for recirculated exhaust gases. Even if the controller close the EGR bypass valve entirely in an effort to stop the flow of exhaust through the EGR cooler, the engine’s demand for EGR will cause some portion of the exhaust stream, up to roughly 30%, to flow through the heat exchanger.

Due to the limited scope of this control problem, the effect of coolant valve positions will not be investigated. The control structure implemented will, though, be made in a modular way such that considerations for coolant flow disturbances can be accounted for. Disturbances due to the other heat sources and sinks in the system do not require any additional considerations in the control strategy because the effect they have on the system is measured directly by the coolant temperature sensor, which is an input to the controller. Because the disturbances act on a controller input, their effects are known explicitly. If the disturbances were acting on the feedback to the controller or its output, then compensating for them would require careful calibration. The controller will be designed to account for the engine’s demand for EGR.

Based on the control problem that has been laid out in this section, the system’s performance will be analyzed to determine the limits of the controller’s ability to manipulate the system.

5.2 Preliminary System Analysis: Simulation of Limit Cases

Before beginning to design a control strategy, it is necessary to define the range of results that the controller is able to effect. To this end, simulations of the FTP city drive cycle
were conducted at an ambient temperature of 30°C with the two exhaust valves in various fixed positions. With one valve completely open, the other valve was held at one of three positions: closed, 50% open, and 100% open. Figure 57 shows the inputs to the EHRS model. Figures 58 and 59 show the results of these cases. Another case was run in which each valve was 50% open.

Figure 57. EHRS Model Simulation Inputs

Figure 58. Coolant Warm-up Rates and Energy Transferred by EGR Cooler
In addition to the warm-up time, the maximum value of each of the constraint variables was recorded. Because the control problem is only concerned with the coolant warm-up period, the final and maximum energy ratios listed are in reference to the period before the coolant reaches 100°C. The results from the tests are shown in Table 5. Red numbers indicate that a constraint was violated.

Figure 59. EGR Cooler Energy Ratios and Engine Back Pressures
According to these simulation results, the exhaust heat recovery system has the potential to improve the coolant warm-up time by 100 seconds, or roughly 37%. This is the difference between the baseline case in which the only the EGR cooler only extracts heat from the gas that must flow to the engine and the extreme case in which all of the exhaust flows through it. The case that produced this result, however, violated the constraint placed on the energy ratio. The greatest improvement in warm-up time produced by a case that did not violate any of the constraints was only 28%.

It can be seen from the results that the back pressure on the engine is actually lowest when all of the exhaust gases flow through the EGR cooler. This valve configuration even results in a lower back pressure than when both flow paths are fully-opened. Although this is counterintuitive, there is a simple explanation for this physical phenomenon. Much of the pressure drop in the exhaust system occurs downstream of the
EGR cooler, after the two flow paths have reunited. The cold gases exiting the EGR cooler have a much higher density than the hot exhaust gases, thus creating much less pressure drop through the long section of tailpipe that runs the length of the vehicle and includes the muffler. As Equation 5.1 shows, pressure loss from flow is inversely proportional to density.

$$\delta p(t) = C_1 \frac{\mu}{\rho} \dot{m}(t) + C_2 \frac{1}{\rho} \dot{m}(t)^2$$  \hspace{1cm} (5.1)

Figure 60 shows the difference between the two extreme cases, 1 and 5, in which one of the valves was shut and the other was open. The case in which warm-up was achieved the quickest without violating the constraints, case 4, is also shown (Figure 61). Each has been limited to the period of time that the coolant is below 100°C (shown as a solid line on the temperature plot). The boiling point of the coolant is also shown.

Figure 60. Open-Loop Analysis, Cases 1 and 5
The same set of tests was run again from a starting point of 0°C to see the effect that ambient temperature of has on system performance. Though the warm-up times were all longer due to the colder starting temperature, the trends of the results hold. Those results are tabulated below.

<table>
<thead>
<tr>
<th>Case</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Backpressure (%)</td>
<td>100</td>
<td>100</td>
<td>100</td>
<td>50</td>
<td>0</td>
<td>50</td>
</tr>
<tr>
<td>Bypass (%)</td>
<td>0</td>
<td>50</td>
<td>100</td>
<td>100</td>
<td>100</td>
<td>50</td>
</tr>
<tr>
<td>Time to 100°C (s)</td>
<td>NA</td>
<td>299</td>
<td>279</td>
<td>240</td>
<td>204</td>
<td>252</td>
</tr>
<tr>
<td>Final E-Ratio</td>
<td>0.02</td>
<td>0.30</td>
<td>0.47</td>
<td>0.56</td>
<td>0.67</td>
<td>0.43</td>
</tr>
<tr>
<td>Max E-Ratio</td>
<td>0.05</td>
<td>0.30</td>
<td>0.47</td>
<td>0.56</td>
<td>0.67</td>
<td>0.43</td>
</tr>
<tr>
<td>Water Temp @ Max (°C)</td>
<td>91.4</td>
<td>100.0</td>
<td>100.0</td>
<td>100.0</td>
<td>100.0</td>
<td>100.0</td>
</tr>
<tr>
<td>Max Back Pressure (bar)</td>
<td>1.16</td>
<td>1.04</td>
<td>1.02</td>
<td>0.94</td>
<td>0.84</td>
<td>0.99</td>
</tr>
</tbody>
</table>

Table 6. Open-Loop System Analysis Results, 0°C Initial Condition, FTP City Cycle
5.3 Development and Optimization of an Open-Loop Control Strategy

Based on the preliminary results, a feed-forward control strategy for the EGR bypass and engine backpressure valves was developed. During the analysis of the open-loop results it was noticed that the case that creates the most back pressure on the engine is the case in which all of the exhaust not needed required by the engine for EGR flows through the conventional exhaust system, avoiding the EGR cooler (case 1). A simulation in which the flow restriction for the engine back pressure valve was removed was used to determine the back pressure on the engine with a conventional exhaust system. The resulting back pressure was found to be only 2% lower than in case 1. Case 1, then, gives a good approximation of the back pressure in a conventional system and can, therefore, be considered the constraint case. Back pressure was lowest when all of the exhaust was flowed through the EGR cooler. The case in which each valve was 50% open resulted in a back pressure roughly halfway between the values for the two extreme cases. These results show that the when the EGR bypass valve is 100% open, the engine back pressure valve does not need to be open. The converse is true, as well, because this is the case that approximates a conventional exhaust system. In order to ensure that the exhaust back pressure constraint is not violated at any point between these two valve configurations, as one valve begins to close the other needs to begin opening at least as quickly as the other is closing. The simplest form of this type of control is one in which one valve position is the complement of the other, meaning that the sum of their normalized positions equals one. For example, if the EGR bypass valve is 75% open, the other must be 25% open. In
addition to guaranteeing that the engine back pressure constraint is not violated, this complementary strategy has the advantage of reducing the complexity of the controller. Instead of having to calculate an ideal value for each valve, the controller needs only to calculate the position of one valve. The position of other will follow naturally.

A control strategy was developed wherein the EGR bypass valve remains completely open as long as the energy ratio stays below a safety threshold value ($ER_{st}$). If the energy ratio exceeds this safety threshold then the controller begins to calculate that valve’s position as a function of the difference between the energy ratio and the final threshold value ($ER_{ft}$). This difference is normalized by the difference between the safety threshold and the final threshold so that it can act as a command to the actuator. The difference between the safety threshold and the final threshold acts as a buffer region that prevents the energy ratio from quickly spiking to unsafe levels. As the energy ratio increases above the safety threshold, the controller begins to close the EGR bypass valve and open the engine back pressure valve, reducing the portion of exhaust gases flowing through the EGR cooler and, thus, the energy ratio. If the energy ratio continues to rise in spite of the controller’s intervention and eventually reaches the final threshold, then the EGR bypass valve is closed completely. Equations 5.2 and 5.3 show the control strategy.

\[
\alpha_{\text{Bypass}} = \left( \frac{ER_{ft} - ER}{ER_{ft} - \min(ER, ER_{st})} \right)^x
\]

\[
\alpha_{\text{Back pressure}} = 1 - \alpha_{\text{Bypass}}
\]
This control function provides three degrees of freedom when tailoring the controller to a specific control problem: 1) the final threshold, 2) the safety threshold, and 3) the shape of the valve position profile between the safety and final thresholds. The value of the final threshold determines the point at which the controller closes the EGR bypass valve entirely in an attempt to reduce the energy ratio. Increasing or decreasing the safety threshold value increases or decreases, respectively, the range of energy ratio values for which the controller allows full exhaust flow through the heat exchanger. The lower the safety threshold is set relative to the final threshold the more conservatively the control strategy will function because the controller will begin to limit the output variable sooner.

In the case of this control problem, both threshold values were dictated by the EGR cooler’s manufacturer. A final threshold value of 0.45 and a safety threshold value of 0.30 will be used.

Within a predetermined buffer region the controller can still be made to act more or less aggressively based on the shape of the control profile. Any number of control output profiles can be used, including a linear profile, an exponential decay of the output, a quadratic profile, etc. For this work, an exponent \((x)\) was applied to the control function. The controller was tested with three different exponent values (0.5, 1, and 2) to determine the optimal profile. The profile given by each is shown below.
To test both the validity of this control approach and the effectiveness of different exponent values, a drive cycle was simulated in which the controller had full knowledge of the system. This was accomplished by sending the actual energy ratio, as calculated by the model, to the controller. The controller was tested on the FTP city cycle at an ambient temperature of 30°C. Based on the results from these runs, it was determined that none of the control profiles provided a substantial benefit in terms of warm-up time over the others; the difference between the three was only 0.6%. The maximum energy ratio did vary quite a bit, though. The case in which the exponent used was 0.5, the energy ratio reached the 80th percentile of the buffer region. This profile was judged to be too aggressive for this application because the exact energy ratio will not be known and the estimate used will have some degree of error. For the cases in which the exponents were 1 and 2, the corresponding values were 53rd and 40th percentiles, respectively. Because there was little discernible difference in their performance, the linear solution ($x = 1$) was selected.
for its simplicity. The results utilizing the linear control profile are shown below. Again, only the warm-up portion of the drive cycle is shown. The energy ratio safety threshold is shown as a dashed line, the final threshold is shown as a solid line.

![Graphs showing temperature, energy ratio, and EGR angle over time.]

Figure 63. Idealized Control Validation, 30°C Initial Condition, FTP City Cycle

Upon analyzing these results it was determined that this control strategy is an appropriate solution to the problem at hand. The warm-up time was reduced by 35% from the baseline, nearly matching the extreme case in which all of the exhaust is flowed through the EGR cooler but without violating the energy ratio constraint or the back pressure constraint. The controller functioned as it was intended to, allowing the energy ratio to approach the threshold without ever crossing it. In this way it was able to recoup a substantial portion of the energy in the exhaust stream without exposing the EGR cooler to harm. The controller first intervenes in the exhaust flow around 120 seconds into the drive cycle. At this time, the energy ratio exceeds the safety threshold value so the EGR
bypass valve begins to close and the engine back pressure valve begins to open. The controller succeeds in stabilizing the energy ratio below the final threshold until roughly 160 seconds when the energy ratio begins to decrease. At this point the EGR bypass valve is opened fully to recoup more exhaust waste heat. As the coolant temperature nears its operating temperature, the controller begins to shut the EGR bypass valve again. These results were accomplished by an idealized implementation of the control strategy where it is assumed that all values in the system are measurable and the EHRS controller has access to those measurements. In order to turn this into an open-loop controller, a feed-forward estimation of the energy ratio is needed. There are a total of seven unique terms in the equation for the energy ratio: 1) the exhaust flow rate, 2) the exhaust inlet temperature, 3) the specific heat of the exhaust, 4) the coolant inlet temperature, 5) the coolant flow rate, 6) the specific heat of the coolant, and 7) the boiling point of the coolant.

\[ r_{energy} = \frac{\dot{m}_{exh,EGR} \times c_{p,exh} \times (T_{exh,vol,1} - T_{water,EGR,in})}{m_{water,EGR} \times c_{p,water} \times (T_{boil} - T_{water,EGR,in})} \]  

(5.4)

Of these variables, two are known either directly or indirectly and one can be approximated as a constant. The inlet temperature of the coolant is a measured value because the coolant entering the EGR cooler is at the same temperature as the water at the thermostat, where the measurement is taken. The specific heat of the coolant can then be calculated as a function of temperature, just as is done for the EHRS model.
in the model, the specific heat of the exhaust can be approximated at a constant value of

$$c_{p\text{water}} = 2477.837 + 3.731 \times T$$  \hspace{1cm} (5.5)

The remaining values needed to be estimated based on information available to the ECU. As described in the previous chapter, the exhaust temperature and flow rate can be estimated fairly accurately based on the air/fuel ratio and the mass flow rate of air into the cylinders. The mass flow rate of air into the cylinders is not a measured value, however, so the mass flow rate of air through the throttle (MAF) is used in its place. While this is not a perfect substitution, in steady-state conditions the MAF is roughly equal to the flow rate into the cylinders when averaged for an entire engine cycle. Under this assumption, the formula for the exhaust temperature shown in chapter four can be applied.

$$T_{\text{exh,eng}} = 2863 + 1.300 \times 10^4 \times MAF - 3.187 \times 10^4 \times MAF^2 - 375.8 \times AFR$$  
$$+ 15.79 \times AFR^2 - 342.9 \times MAF \times AFR$$  \hspace{1cm} (5.6)

$$m_{\text{exh,eng}} = MAF \times (1 + AFR^{-1})$$  \hspace{1cm} (5.7)

The temperature given in the above equation is, however, the temperature of the exhaust leaving the engine, before it passes through the catalytic converter. Based on simulation results it was determined that the catalyst has a smoothing effect on the exhaust temperature similar to a transfer function with a 56 second time constant. The mass flow rate found in the above equation is also the value for exhaust leaving the engine. For that reason, it was necessary to determine a flow split between the bypass and back pressure
legs of the exhaust system. The variables affecting the flow split are the position of the 
bypass and backpressure valves (which can be considered one variable given the 
complementary nature of their actuation), the engine’s demand for EGR, and the total 
flow of exhaust gases from the engine. Simulations were run in which the engine speed, 
the throttle position, and the two exhaust valve positions were held constant. The fraction 
of exhaust gases flowing through the EGR cooler was recorded for those input values. 
An input value was changed and the new split was recorded. This process was repeated 
until a three-dimensional map had been constructed. The maps for a closed bypass valve 
and half open bypass valve are shown in Figure 64, respectively.

![Figure 64. Exhaust Flow Split to EGR Cooler, Bypass Valve 0% and 50% Open](image)

In the controller, the flow split returned by the table is multiplied by the estimation of the 
flow rate exiting the engine to give the mass flow rate through the EGR cooler. This 
estimation was calibrated using the FTP city cycle. Both the flow rate and the 
temperature estimates were validated using the FTP highway cycle. Figures 65 and 66
show the controller’s model based estimates for the exhaust temperatures and flow rate into the EGR cooler compared with the simulated values. The agreement between the estimated values and the values calculated by the flow and thermal models is extremely strong.

Figure 65. Exhaust Temperature Prediction Validation, 30°C Initial Condition, FTP Highway Cycle (first 600s shown)
The coolant flow rate is affected by the pump speed and the position of each of the valves in the cooling system. Due to the limited scope of this control problem, the effects of valve positions were ignored, assuming the positions to be constant. A relationship was made between the engine speed and the mass flow rate of coolant through the pump. Linear regression was used to fit a quadratic expression for the flow rate to the pump speed based on simulation results. The coolant flow split through the EGR cooler was modeled as a one-dimensional table based on the engine speed for this work, but the controller was designed in such a way that a function predicting the flow split based on the coolant valve positions can be easily added. The product of these estimations was validated on the FTP highway cycle. The results were accurate to within 6.6%, or 0.025 kg/s, on average.
The only remaining variable to account for is the boiling point of the coolant. Because a known equation is used to find the boiling point of the coolant based on pressure, the coolant pressure at the inlet of the exchanger needs to be estimated. The flow restriction through the EGR cooler is constant and the pressure downstream, at the inlet of the pump, is nearly constant. For this reason the pressure at the inlet of the EGR cooler can be approximated as a function of the estimate for the coolant flow rate through the heat exchanger. Since the boiling point is a function of the coolant pressure and the coolant pressure is a function of the mass flow rate of coolant through the exchanger, the boiling point can be expressed as a function of the mass flow rate of coolant through the EGR cooler.
With all of these estimates in place, the true open-loop control strategy was tested. Simulations were run for both the FTP city and highway drive cycles at 30°C. The city cycle was also run at 0°C. The results from the city cycle are compared with values from the baseline cases and the control strategy that used full knowledge of the system.

![Graphs showing temperature, energy ratio, and valve position over time.](image)

Figure 68. Open-Loop Controller Validation, 30°C Initial Condition, FTP City Cycle

\[ T_{water, boil} = 381.5 + 2.377 \times m_{water, EGR Cooler} + 0.6324 \times m_{water, EGR Cooler}^2 \] (5.9)
Figure 69. Open-Loop Controller Validation, 30°C Initial Condition, FTP Highway Cycle

<table>
<thead>
<tr>
<th>Case</th>
<th>1</th>
<th>4</th>
<th>5</th>
<th>7</th>
<th>8</th>
<th>9</th>
</tr>
</thead>
<tbody>
<tr>
<td>Backpressure (%)</td>
<td>100</td>
<td>50</td>
<td>0</td>
<td>Idealized Control</td>
<td>Model Based Control</td>
<td>Model Based Control (0°C)</td>
</tr>
<tr>
<td>Bypass (%)</td>
<td>0</td>
<td>100</td>
<td>100</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Time to 100°C (s)</td>
<td>272</td>
<td>196</td>
<td>172</td>
<td>177</td>
<td>178</td>
<td>207</td>
</tr>
<tr>
<td>Final E-Ratio</td>
<td>0.13</td>
<td>0.40</td>
<td>0.44</td>
<td>0.37</td>
<td>0.34</td>
<td>0.38</td>
</tr>
<tr>
<td>Max E-Ratio</td>
<td>0.13</td>
<td>0.41</td>
<td>0.92</td>
<td>0.38</td>
<td>0.36</td>
<td>0.40</td>
</tr>
<tr>
<td>Water Temp @ Max (°C)</td>
<td>100.0</td>
<td>67.9</td>
<td>84.3</td>
<td>78.9</td>
<td>79.2</td>
<td>43.6</td>
</tr>
<tr>
<td>Max Back Pressure (bar)</td>
<td>0.98</td>
<td>0.80</td>
<td>0.71</td>
<td>0.98</td>
<td>0.98</td>
<td>N/A</td>
</tr>
</tbody>
</table>

Table 7. Comparison of Control Results, 30°C Initial Condition, FTP City Cycle
The model based estimate control strategy (case 8) was able to improve the warm-up time over the baseline value set by the conventional exhaust system (case 1) by 94.2 seconds, or 34.6%, on the FTP city cycle at 30°C. This is only 3.5% slower to warm-up than the extreme case in which all of the exhaust was flowed through the EGR cooler (case 5), but it managed to do so without violating any of the constraints on the system, reducing the maximum energy ratio by 61%. The case run from 0°C on the city cycle represented a 31% improvement over the best static valve test case in which the constraints were not violated.

In this chapter, the objectives and boundaries for a control problem were defined. The plant model was investigated to determine the range of outcomes that the controller could effect. A preliminary control strategy was proposed to minimize the coolant warm-up time without violating the constraints of the system. The full potential of this strategy was tested in an idealized system in which all values were known. The estimates necessary to make the control strategy function in an open-loop system were then calculated and calibrated using a mix of experimental and simulation data. The open-loop controller produced a 34.6% reduction in coolant warm-up time compared to a conventional thermal management system without violating the system constraints.
6.1 Conclusions

This thesis describes a project that was undertaken to develop a control strategy to minimize the coolant warm-up time in an automobile following a cold-start. The resulting controller would need to manipulate two valves on the exhaust side of an exhaust heat recovery system (EHRS) and would need to function in the larger context of an advanced thermal management system. To begin, a review of literature describing the study and implementation of advanced thermal management systems was conducted. This review revealed that several methods for improving vehicle efficiency have been investigated and tested by academics, governmental agencies, automakers, and suppliers. Some of these tests were carried out physically while others were based entirely in simulation. Regardless of the methods employed by these researchers, all came from the same perspective: that conventional thermal management systems are not properly optimized for fuel economy. All found that substantial gains in efficiency could be made through increased control of the cooling system, electrification of components, or through the addition of certain new components to the thermal management system.

Next, a review of the methods for modeling a thermal management system was conducted. Fundamental energy and flow equations describing the behavior of fluid flow
networks were discussed. Various techniques for modeling heat exchangers were
described and the appropriateness of each for representing a specific application was
evaluated.

Those fundamental equations and heat exchanger modeling approaches were then used to
develop a working model of an exhaust heat recovery system. This system included
models for an exhaust flow network, a coolant flow network, an EGR cooler, and all of
the necessary thermal devices in the cooling system. To supply these models with the
necessary runtime information, a mean-value engine model was also created. Individual
components of the system were calibrated with experimental data, when available. If at
all possible, models were also validated against experimental data.

Next, the control problem to be studied was thoroughly defined. This consisted of
describing all of the actuators and sensors available to the controller, the exact goal of the
controller, and the system constraints that the controller would need to respect. The
controller would have access to all of the measurements and commands processed by the
engine control unit (ECU). It would need to implement its strategy through two valves:
the EGR bypass valve and the engine back pressure valve. The purpose of the controller
would be to minimize the time required to warm the cooling system’s coolant to 100°C
without increasing the back pressure on the engine or exceeding a critical performance
constraint on the EGR cooler.

With the calibrated and validated system model, different open-loop control inputs were
tested to determine the potential for improvements in coolant warm-up time. From these
initial tests, it was concluded that the constraint on engine back pressure would not have a
significant effect on the controller’s operation as long as a complementary valve position strategy was used. This meant that the operation of one valve would mirror the other; for example, if one valve opened fully the other would close fully. A control strategy using this principle was proposed with the intent of minimizing the cost function and observing all of the system constraints. To test the effectiveness of this strategy, simulations were run in which the controller had full knowledge of all critical measurements, even those not available to the ECU. After determining that this control strategy functioned as intended, the strategy was implemented in a open-loop manner. To do this, several unmeasured values were estimated based on a mixture of experimental of and simulation results.

The resulting open-loop controller was tested on both the FTP city and highway drive cycles. Starting from an ambient condition of 30°C and running the FTP city cycle, the controller was able negotiate within the constraints on the system to produce a 34.6% reduction in coolant warm-up time compared to a conventional thermal management system on the same vehicle.

6.2 Future Work

Going forward, the model developed in this work will be integrated into a larger, more comprehensive network of models called a vehicle energy simulator. The goal of this vehicle energy simulator will be to develop an overarching energy control strategy for the vehicle. This controller will oversee the operation of not only the exhaust and coolant
systems but also the electrical loads on the vehicle, belt-driven accessories, and HVAC components.

As mentioned previously, due to the limited scope of this control problem, the effects of several actuators in the cooling system were ignored and approximations were made instead. The high level vehicle energy controller will have command over these actuators and will integrate their control with the EHRS control strategy developed here. In the places where considerations were made for them, feed-forward estimations of the effects of these variables will need to be added to the EHRS controller. This will enable it to function over the wide range of operating conditions to which it will be exposed in the vehicle energy simulator.
References

Chapter 1


Chapter 2


[2.21] Cipollone, R. “On the Thermal Fields of I.C.E. Cylinder Liners”; International Congress and Exposition; February 1990; Detroit, Michigan; USA.


Chapter 3


Chapter 4


[4.9] Alshamani, K. “Equations for Physical Properties of Automotive Coolants”; 2003 SAE World Congress; March 2003; Detroit, Michigan; USA.


Appendix A: NOMENCLATURE

\[A\] Area (\text{m}^2)
\[A_{ij}\] Interaction Parameter
\[AFR\] Air/Fuel Ratio
\[C\] Thermal Capacity (J/K)
\[C_{3}, C_{2}, C_{fluid}\] Calibration Coefficients
\[\dot{C}\] Rate of Thermal Capacity (W/K)
\[C_d\] Discharge Coefficient
\[c_p\] Specific Heat at Constant Pressure (J/kg-K)
\[c_v\] Specific Heat at Constant Volume (J/kg-K)
\[D\] Diameter (m)
\[ER\] Energy Ratio
\[f\] Friction Factor
\[G\] Mass Velocity (kg/m^2-s)
\[H\] Enthalpy (J)
\[\dot{H}\] Rate of Change of Enthalpy (W)
\[h\] Coefficient of Convection (W/m^2-K)
\[\Delta H_{vap}\] Latent Heat of Vaporization (J/mol)
\[K\] Thermal Conductivity (W/m-K)
\[L\] Length (m)
\[M\] Molar Mass (g/mol)
\[MAF\] Mass Air Flow Rate (kg/s)
\[m\] Mass (kg)
\[\dot{m}\] Mass Flow Rate (kg/s)
\[N\] Engine Speed (rev/s)
\[NTU\] Number of Transfer Units
\[Nu\] Nusselt Number
\[\dot{P}\] Power (W)
\[p\] Pressure (Pa)
\[Pr\] Prandtl Number
\[\dot{Q}\] Heat Transfer Rate (W)
\[R\] Specific Gas Constant (J/kg-K)
\[R_{conv}, R_{cond}\] Thermal Resistance (K/W)
\[R_{univ}\] Universal Gas Constant (J/mol-K)
\[Re\] Reynolds Number
\[RPM\] Engine Speed (RPM)
\[T\] Temperature (K)
\[t\] Time (second)
\[U\] Internal Energy (J)
\[UA\] Overall Heat Transfer Rate (W/K)
\(V\) Volume (m\(^3\))
\(\dot{V}\) Volumetric Flow Rate (m\(^3\)/s)
\(\bar{V}\) Average Velocity (m/s)
\(y_i\) Mole Fraction (unless otherwise specified)

**Greek Symbols**

\(\alpha\) Angle (Normalized 0-1)
\(\beta\) Calibration Term
\(\gamma\) Ratio of Specific Heats \((c_p/c_v)\)
\(\delta\) Thickness (m)
\(\epsilon\) Effectiveness of Heat Exchanger
\(\varepsilon\) Error
\(\lambda_v\) Volumetric Efficiency
\(\mu\) Dynamic Viscosity (kg/m-s)
\(\nu\) Kinematic Viscosity (m\(^2\)/s)
\(\rho\) Density (kg/m\(^3\))
\(\tau\) Torque (N-m)

**Subscripts**

\(amb\) Ambient
\(Back\ Pressure\) Engine Back Pressure Valve
\(Bypass\) EGR Bypass Valve
\(c\) Cold Side
\(cat\) Catalytic Converter
\(cond\) Conduction
\(conv\) Convection
\(counter\) Counterflow
\(cyl\) Cylinder
\(down\) Downstream
\(EGR\) EGR Cooler
\(EO\) Engine Lubricating Oil
\(EOC\) Engine Oil Cooler
\(eff\) Effective
\(eng\) Engine
\(exh\) Exhaust Gas
\(exp\) Experimental
\(ft\) Final Threshold
\(h\) Hot Side
\(heater\) Cabin Heater Core
\(HEX\) Heat Exchangers
\(f\) Friction
\(IM\) Intake Manifold
\( l \) Left Side
\( lm \) Log Mean
\( oil \) Engine Oil or Transmission Oil
\( parallel \) Parallel Flow
\( pred \) Predicted
\( r \) Right Side
\( rad \) Radiator
\( ref \) Reference
\( shell \) Outer Shell of Device
\( st \) Safety Threshold
\( sub \) Substrate
\( TO \) Transmission Oil
\( TOH \) Transmission Oil Heater
\( trans \) Transmission
\( tstat \) Thermostatic Valve
\( up \) Upstream
\( vap \) Vaporization
\( vol \) Volume
\( wall \) Internal Wall of Heat Exchanger
\( water \) Coolant/Water Mixture
\( wu \) Warm-up