Modeling and Control of an Electrically-Heated Catalyst

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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Current model-based design research on automotive catalytic converters mainly fall into three basic categories: either modeling the catalyst as a continuous system based on physics, discretizing the system to reduce modeling complexity, or developing a highly-simplified, mean-value model for control. Continuous models are computationally intensive and therefore not well-suited for implementation into a vehicle model for Hardware in the Loop or control design. Highly-simplified models are calibrated for a particular system without incorporating the governing physical laws into the model, and mean-value models are only able to predict the response for a single lumped element. Although a simplified, mean-value model can be developed to accurately predict system response, it does not lend itself to being extended to broader applications without significant re-calibration efforts. Therefore, a model is needed that can account for the physics of the system so it can be extended to further applications while decreasing computation time to allow the model to be implemented for Hardware in the Loop and vehicle control design.

This research investigates the development of such a model to predict automotive catalytic converter thermal response during warm-up. A one-dimensional, lumped-parameter model of a three-way catalyst was developed in Matlab/Simulink. The catalyst
length was divided into discrete elements. Each discrete element contained states for the temperatures of the gas, substrate, and can wall. Heat transfer mechanisms were modeled from physics-based equations. For each discrete element, these equations modeled the enthalpy of the gas flow axially through the catalyst, convective heat transfer between the gas and substrate, conduction between discrete elements axially along the catalyst for the substrate and for the can, conduction between the substrate and can wall, and convection from the can wall to ambient. Model predictions were validated against experimental results for thermal transients.

The application of this model was analysis for an extended-range electric vehicle application with electrically-heated catalyst (EHC). The model was used to compare the catalyst thermal response with and without the EHC. These results facilitated the development of a control strategy for the EHC, as well as recommendations for improving the overall vehicle control strategy. For further development, this model can also be extended to a two- or three-dimensional application. A two-dimensional catalyst model would be of interest to account for temperature gradients in the radial direction through the catalyst.
DEDICATION

To the one who gave His life for mine.
I would like to thank Dr. Shawn Midlam-Mohler for all of his guidance, his continued support, and his enthusiasm for advanced vehicle technologies and teaching. Thanks to Dr. Giorgio Rizzoni for the unparalleled opportunity to attend Ohio State University and be so involved in the EcoCAR program. Thanks to everyone at the OSU Center for Automotive Research for providing such excellent support of the students and motorsports teams. Thanks to the Advanced Vehicle Technology Competition *EcoCAR: The Next Challenge* program for providing the foundation for such an outstanding hands-on experience and for investing so effectively in the education of students. Thank you to the OSU-Honda Partnership program for the financial support to allow me to complete my graduate program while investing in the OSU EcoCAR program. And finally, I would like to thank the entire Ohio State University EcoCAR team for continually challenging, teaching, and inspiring me.
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CHAPTER 1
INTRODUCTION

1.1 Motivation

Today’s vehicles are subject to increasingly stringent fuel economy and emissions standards set forth by federal regulation. Achieving these standards across the national fleet is possible through numerous means, including vehicle electrification and alternative fuels, all of which require efficient control system design.

Much research and development in the area of vehicle electrification is underway to displace the significant amount of the petroleum-based fuel currently demanded by the transportation industry. As a result of these efforts, many hybrid-electric vehicles (HEVs), plug-in hybrid electric vehicles (PHEVs), and electric vehicles (EVs) are entering the market. Despite significant improvements in energy storage system technology, today’s electrified vehicles are still limited by constraints such as operating temperature, cost, and battery energy density which directly impacts vehicle range. Therefore, the internal combustion engine will still be needed as a major propulsion source for decades to come.

As an interim solution, the versatility of the internal combustion engine is being capitalized on through the use of alternative fuels such as ethanol, biodiesel, and
compressed natural gas, as well as innovative solutions like engine stop/start strategies. Such technological advances in hybrid vehicles and alternative fuels demand effective control systems in order to accomplish the goal of improved fuel economy and emissions.

To promote student interest and knowledge in the area of alternative vehicle technologies, the Advanced Vehicle Technology Competitions (AVTC) series was started in 1987. This program challenges university teams to develop fuel-efficient vehicles with advanced propulsion systems. The most current AVTC competition is EcoCAR: The Next Challenge (EcoCAR), which began in July 2008 and concluded in June 2011. The Ohio State University is a participating team developing a plug-in hybrid electric vehicle.

1.2 EcoCAR: The Next Challenge
Ohio State University (OSU) is one of sixteen North American universities participating in EcoCAR: The Next Challenge (EcoCAR), a vehicle development competition sponsored by the United States Department of Energy (DOE), General Motors (GM), Natural Resources Canada, and many others. This three-year competition challenges student teams to re-engineer a crossover-utility vehicle for improved fuel economy and emissions while maintaining vehicle performance and consumer acceptability. Each team’s task is to design, build, and refine a fully-functioning, prototype vehicle. To accomplish this goal, teams follow a vehicle development process modeled after the practice used by General Motors Corporation. At the end of each year, teams participate in a several day competition event in which vehicles are evaluated over a number of static and dynamic tests. For the EcoCAR competition, 42% of total competition points are...
allocated to emissions and energy consumption with one-fourth of those points designated specifically for tailpipe emissions.

1.2.1 EcoCAR Emissions and Energy Consumption

The Emissions and Energy Consumption Event in the EcoCAR competition is an evaluation that encompasses four equally-weighted areas:

- On-road fuel consumption
- Well-to-wheel petroleum energy usage
- Well-to-wheel greenhouse gas emissions
- Regulated tailpipe emissions (NOx, THC, CO)

Vehicles are tested on-road with a portable emissions measurement system to collect time-resolved emissions measurements. The emissions drive schedule is 100 miles constituting a mix of city and highway driving. Vehicles begin the drive cycle with high voltage battery pack at full state of charge and are allowed to operate in charge depleting and/or charge sustaining modes. The charge depletion range is accounted for using a utility-factor weighting methodology [28].

During Year 3 of EcoCAR, teams were also given the opportunity to perform chassis dynamometer emissions testing at the U.S. Environmental Protection Agency facility in Ann Arbor, Michigan. This testing included time-resolved measurements with a portable emissions measurement system, as well as a certification test sequence using a bag and bench system.
1.3 Ohio State EcoCAR Plug-In Hybrid Electric Vehicle

The vehicle architecture developed by the Ohio State EcoCAR team is a plug-in hybrid electric vehicle, shown in Figure 1. The design features a 1.8-L high compression ratio E85 internal combustion engine (ICE) coupled to an 82-kW front electric machine (FEM) via a unique twin-clutch transmission designed to enable greater operating efficiency through limited parallel operation. The twin-clutch arrangement permits coupling the engine and/or the FEM to the front axle. This transmission design allows the vehicle to operate in a series or parallel hybrid mode. The exhaust aftertreatment system features a close-coupled three-way catalyst with integrated electrically-heated catalyst (EHC) along with secondary air injection (SAIR) system upstream of the EHC for improved cold start and hot restart emissions. A 22-kWh lithium ion battery pack is used for onboard energy storage packaged as a split pack with two modules in the front console area and three modules in the rear of the vehicle. A 103-kW rear electric motor (REM) provides pure electric vehicle capability, and the powertrain allows front and rear axle regenerative braking. In addition, a DC/DC converter and an AC/DC charger are packaged in the rear, allowing the vehicle to charge through a 208-V or 110-V outlet. Selection of this vehicle architecture is further described in [4].
The internal combustion engine incorporated into the Ohio State EcoCAR vehicle was a 2006 Honda R18A3 compressed-natural gas engine. This engine was selected to take advantage of its high compression ratio (12.5:1) with E85 fuel [9]. Thus, the engine hardware was modified for dedicated E85 fuel. This included replacing the fuel system with injectors, rail, lines, pump, and tank that are ethanol-compliant. Other modifications included replacing the original ceramic-monolith catalytic converter with a metal-foil catalytic converter with an integrated electrically-heated catalyst. Selection of this catalyst is discussed in Section 3.2.
The engine control unit (ECU) is a 128-pin Woodward MotoTron ECU-0565-128-0701-C. The engine controller communicates with the vehicle supervisory controller through CAN communication. The supervisory controller initiates engine start-up and shutdown and provides torque requests to the engine controller. The engine controller is responsible for all engine control and engine fault diagnosis, including some auxiliary systems, such as the engine coolant pump, radiator fans, and electrically-heated catalyst system.

1.4 Thesis Overview

This thesis discusses development of the emission system in Ohio State University’s vehicle for the EcoCAR competition with an in-depth analysis and modeling of heat transfer properties of the three-way catalytic converter system and the effects of including an electrically-heated catalyst system. An overview of the chapters contained in this thesis is provided below.
• Chapter 2 – Literature Review
  o Chapter 2 summarizes emissions-reduction methods for stoichiometric spark-ignited engines, particularly work that has been done with electrically-heated catalysts.

• Chapter 3 – Electrically-Heated Catalyst Implementation
  o Chapter 3 details the hardware and software implementation of an electrically-heated catalyst into the Ohio State EcoCAR plug-in hybrid electric vehicle.

• Chapter 4 – Physics-Based Catalytic Converter Thermal Model
  o Chapter 4 discusses the development of a heat transfer model to describe thermal characteristics of the three-way catalytic converter and the impact of adding an electrically-heated catalyst.

• Chapter 5 – Electrically-Heated Catalyst Control
  o Chapter 5 investigates an advanced control strategy for the electrically-heated catalyst using the catalytic converter model as the plant.

• Chapter 6 – Electrically-Heated Catalyst Validation (Experimental)
  o Chapter 6 presents experimental validation results showing performance of the electrically-heated catalyst in the Ohio State EcoCAR vehicle.

• Chapter 7 – Conclusions and Future Work
  o Chapter 7 summarizes the work completed and future applications.
2.1 Introduction

The engine in the Ohio State University’s extended-range electric vehicle developed for the EcoCAR competition is a stoichiometric spark-ignited internal combustion engine running on E85 ethanol fuel. Therefore, the focus of this literature review is emissions technology for stoichiometric spark-ignited engines and for E85 fuel usage.

2.2 Vehicle Tailpipe Emissions

Spark-ignited internal combustion engines combust a mixture of fuel and air to produce torque. Engine-out emissions occur as a result of multiple factors. Burned gas temperatures decrease late in the expansion stroke and during exhaust blowdown, preventing recombination reactions to maintain chemical equilibrium and “freezing” burned gas composition. Emissions are also produced due to deviations from stoichiometric air/fuel ratio, combustion inefficiency, and differences in uniformity between cylinders [17]. The emissions produced through these phenomena include unburned total hydrocarbons (THC), NO\textsubscript{x}, and CO, as well as N\textsubscript{2}, CO\textsubscript{2}, H\textsubscript{2}O, O\textsubscript{2}, and H\textsubscript{2}. 
Vehicle emissions are regulated based on measured exhaust emissions produced at the tailpipe. In the United States for model year 2007 and beyond, fleet averages are required to meet Tier II Bin 5 standards as evaluated on the Federal Test Procedure driving cycle.

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<tr>
<td>NOx (g/mi)</td>
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Engine-out emissions are unable to meet these regulations without exhaust aftertreatment systems, and significant strides in vehicle emissions reduction have been achieved through the use of three-way catalytic converter technology.

### 2.3 Three-Way Catalytic Converter

The three-way catalytic converter (TWC) is a device in the vehicle exhaust system used for reducing tailpipe emissions in spark-ignited engine applications. This is achieved because the TWC catalyzes chemical reactions to oxidize carbon monoxide (CO) and hydrocarbons (HC) and reduce nitrogen oxides (NOₓ) in the exhaust gas, converting these constituents into CO₂, N₂, and H₂O. Many chemical reactions take place in the TWC with a small sub-set of these equations being the most significant. Equations (1) through (7) show the primary reactions that occur in a functioning TWC at operating temperature, where \( C_3H_6 \) represents the alkene and \( C_3H_8 \) represents the alkane components of the fuel [5][33].
\[ CO + H_2O \rightleftharpoons CO_2 + H_2 \]  
\[ 2CO + O_2 \rightarrow 2CO_2 \]  
\[ 2C_3H_6 + 9O_2 \rightarrow 6CO_2 + 6H_2O \]  
\[ C_3H_8 + 5O_2 \rightarrow 3CO_2 + 4H_2O \]  
\[ 2H_2 + O_2 \rightarrow 2H_2O \]  
\[ 2CO + 2NO \rightarrow N_2 + 2CO_2 \]  
\[ C_mH_n + mH_2O \rightarrow mCO + \left(m + \frac{n}{2}\right)H_2 \]  

Figure 3: Close-coupled TWC  
Figure 4: Cross-sectional view of TWC ceramic monolith

The structure of the catalytic converter is a monolithic substrate comprised of many small channels. The surface of the substrate is covered with a washcoat comprised of a blend of silica, alumina, and ceria, with noble metals added to the mixture by suspension. The
noble metals used are platinum (Pt) and palladium (Pd) as oxidation catalysts and rhodium (Rh) as a reduction catalyst. The ceria provides oxygen storage capacity, which is a mechanism for storing surplus oxygen during lean engine transients and releasing oxygen to oxidize CO and HC during rich, oxygen-deficient transients. This oxygen storage allows the catalytic converter to operate over a wider range of air/fuel ratios. Silica and alumina are used for adhering the mixture to the converter substrate and form rough, irregular surfaces which increase the area available for chemical reactions to occur [3][11].

To be actively catalyzing reactions on its surface, the TWC must be above a minimum temperature, called the light-off temperature. Typical light-off temperatures are about 250-300°C [16]. Conventional TWCs are heated by hot exhaust gas flowing through the channels and, once warm enough, also by exothermic catalytic reactions taking place on the surface. Since the catalytic converter is lower than light-off temperature when the engine is started, it takes about 1-2 minutes for the TWC to be sufficiently warmed up [16]. However, it is during this time before reaching the TWC light-off temperature that 50-80% of tailpipe emissions are realized [20]. Therefore, much effort has been invested in reducing this warm-up period.

Once active, the catalytic converter is able to effectively oxidize THC and CO and reduce NOx. Figure 5 shows typical trends in conversion efficiency as a function of air/fuel engine equivalence ratio (EQR_{A/F}), illustrating the trade-offs in conversion efficiency. For rich engine operation (EQR_{A/F} < 1), the TWC is able to effectively catalyze nearly
100% of CO and THC emissions but will allow significant NO\textsubscript{x} output. Conversely, at lean engine operation (EQR\textsubscript{A/F} > 1), NOx conversion efficiency is very high while conversion of CO and THC suffers. Therefore, it is desirable to operate within a window of EQR\textsubscript{A/F} between about 0.97 and 1.03 to achieve effective conversion of all three constituents: CO, THC, and NO\textsubscript{x}.

![Catalyst conversion efficiency vs. open-loop EQR](image)

Figure 5: Example of catalyst conversion efficiency vs. open-loop EQR for gasoline engine operating at 2000 RPM and 50 kPa

To achieve operation within the desirable conversion efficiency window of the catalytic converter, much effort is invested in air/fuel ratio control. The catalytic converter is instrumented with exhaust gas oxygen sensors to measure EQR\textsubscript{A/F} and provide feedback.
to the engine controller. This feedback is used to modify the open-loop fuel command in order to maintain engine operation oscillating closely around stoichiometry.

Conversion efficiency is also affected by catalyst aging. A fresh catalyst initially has a high level of catalytic activity, so most experimental catalytic converters are aged at least 500 miles before collecting results [16]. As the catalytic converter ages, conversion efficiency decreases and this results in greater tailpipe emissions. Therefore, U.S. federal emissions regulations specify emissions standards for vehicle systems at 50,000 miles and 120,000 miles.

2.3.1 Three-Way Catalytic Converter Design Considerations

Catalytic converter design has been investigated to optimize the choice of converter diameter and length, cell density, foil thickness, substrate material, precious metal loading, etc., to improve the heat transfer properties of the device.

2.3.1.1 Geometric Surface Area

Catalytic converter performance is affected by the amount of surface area available for reactions to occur. Greater surface area allows for more simultaneous reactions, and hence can improve conversion efficiency. Design of the number of channels and their diameter and wall thickness determines the geometric surface area as well as the back pressure of the catalytic converter. The number of channels in the converter is defined as cell density and is specified in units of cells per square inch (cpsi).
2.3.1.2 Heat Transfer Characteristics

Since catalytic converters need to be above light-off temperature to operate effectively, it is desirable for TWCs to have heat transfer properties that enable them to heat quickly. This is affected by substrate material and geometry. Conventional catalytic converter substrates have been made of ceramic monolith; however more recent TWCs are being made with thin metal foil monoliths. Metal foil monoliths can be made with thinner channel walls than ceramic monoliths, which reduces substrate mass and can enable faster heating capability. An additional benefit of thinner walls is reduced back pressure while maintaining similar geometric surface area [15] [20].

2.3.1.3 Mounting Location

Traditionally, catalytic converters were located in an underbody location with a section of exhaust pipe connecting the exhaust manifold to the inlet of the TWC. Today, many TWCs are now packaged in a close-coupled packaging style mounted directly to the engine body to minimize heat loss between the engine and catalytic converter.

2.4 Cold Start

Cold conditions present particular issues for start-up of spark-ignited engines. Traditionally, the term “cold start” is used for ambient temperatures lower than 20-30°C or engine coolant temperatures less than 40°C [16]. However, similar start-up issues are still present even for initial engine start-up at warmer conditions. Therefore, this work uses the term “cold start” to refer to an engine start-up in which the engine had been at equilibrium with ambient temperature before engine start.
Under cold start conditions, the temperature of the catalytic converter is below light-off, resulting in appreciable tailpipe emissions until the catalyst is at an operational temperature. In addition, cold start conditions also present problems with fuel vaporization. Since the engine intake, cylinder walls, pistons, etc., are cold at start-up, injected fuel condenses on these surfaces and resists vaporizing and mixing with the air charge. Vaporization of ethanol fuel is particularly difficult in cold conditions due to the fuel’s low vapor pressure and high flash point. As a result, ethanol-fueled engines tend to experience particularly poor quality engine starts or even no start at all in cold conditions [2]. Most spark-ignited engine controllers perform an initial fuel enrichment phase upon start-up to account for this cold start phenomenon. However, to reduce HC and CO emissions and improve fuel economy, it is desirable to minimize fuel enrichment as much as possible.

Many different methods have been investigated to improve cold start. These methods fall into two categories: (1) engine control and hardware modifications to improve engine startability and prevent cold start engine-out emissions, and (2) aftertreatment to improve cold start catalyst-out emissions.

2.4.1 Startability and Emissions Prevention

Most strategies for improving startability and preventing cold start emissions focus on achieving higher amounts of fuel vapor. Fuel enrichment is a widely used strategy; however, injecting large amounts of fuel can lead to excess HC and CO emissions, as well as dilution of engine oil. Therefore, it can be desirable to seek other options for
improving fuel vaporization. Increasing injector pressure causes slightly better fuel vaporization at the injector nozzle. Colpin et al. shows that increasing injector pressure from 350 kPa to 500 kPa can reduce the number of cycles required to reach engine start by an average of 4 cycles (where engine start is defined as achieving 200 kPa indicated mean effective pressure) [8]. Colpin et al. has also investigated the effect injecting fuel during the intake stroke instead of during the exhaust stroke for better vaporization of fuel into combustion chamber. This change in injection phasing was shown to reduce the number of cycles required to reach engine start by up to 23 cycles. Another strategy is to lower the manifold air pressure during start-up. This can achieve better fuel vaporization because less air charge is mixed with a given amount of vaporized fuel lead, leading to richer air/fuel ratio mixture without requiring as much additional fuel to be injected for the fuel enrichment phase.

These strategies can be altered through engine control software without requiring any modifications to the engine hardware. Additional gains in fuel vaporization have been realized by modifying engine hardware. Colpin et al. also showed startability improvements through modified valve timing using either late intake valve opening or low intake valve lift. This increases the velocity of air through the intake valve, which improves fuel vaporization. Another option is to modify valve timing by using early intake opening or late exhaust valve opening to have positive valve overlap. This causes unburned exhaust gases to remain in combustion chamber and therefore increases the amount of fuel in the combustion chamber. This improves startability, but also leads to
poor combustion stability. These strategies are only feasible if valve timing can be
designed or modified.

Alternatively, much work has been done to develop a means for heating fuel before it is
injected. For this purpose, Kabasin et al. at Delphi Corporation have investigated heated
fuel rails and heated fuel injectors. Heating the entire fuel rail with an internal heater has
undesirable drawbacks including pre-crank warm-up times of up to 60 seconds, uneven
heating, large radiant heat loss, and significant electrical energy consumption. Locally
heating the fuel rail directly above each injector also causes drawbacks including pre-
crank warm-up times of up to 20 seconds, heated fuel rising to the top of the rail away
from the injectors, and first injections consisting of ambient temperature fuel. Instead,
Delphi has developed an ethanol fuel injector with integrated heater to increase fuel
temperature and improve vaporization. The heated fuel injector was shown to achieve 3
second start-up time (crank-to-500 RPM) without pre-crank heating, and 1.8 second start-
up time with 6 seconds of pre-crank heating [19].

Retarding spark timing is a common method for improving warm-up because it increases
engine-out exhaust temperatures. This phenomena is illustrated in Figure 6 and Figure 7.
At the spark timing that achieves maximum brake torque (approximately 30 degrees
spark advance), the maximum amount of work produced by the engine is transferred to
the output shaft. Moving away from maximum brake torque (i.e. less spark advance, more
spark retard), the differential volume within the cylinder decreases and less work is
transferred. Since the same total energy is maintained, more of this energy is transferred
as heat, resulting in higher exhaust temperatures. Higher exhaust temperatures lead to lower emissions because chemical reactions are occurring quickly. Conversely, at lower temperatures, chemical reactions occur more slowly and not as many of the reactants can be converted into products, leading to greater emissions as low temperatures.

Figure 6: Exhaust temperature for spark advance from -10 to +35 degrees from top-dead center (TDC)
Another common method for improving warm-up is operating the engine at higher mass air flows during start-up. Mass air flow directly impacts the enthalpy entering the catalytic converter, so more heated air flowing through the TWC during start-up helps to increase the temperature of the TWC to reach light-off.

Once the catalytic converter has reached operating temperature, air/fuel ratio (AFR) control is used to maintain engine operation close to stoichiometry and prevent emission excursions. AFR control makes use of feedback from oxygen sensors located in the exhaust system, typically before and after the catalytic converter. Many sophisticated methods have been developed for air/fuel ratio control, with one model-based approach being described in [10].
2.4.2 Aftertreatment

Although the emissions prevention methods discussed above have been shown to improve engine-out emissions during cold start, there is still need for aftertreatment methods to further improve the emissions before exiting the vehicle tailpipe. For stoichiometric, spark-ignited engines, exhaust aftertreatment is typically done with secondary air injection and/or advanced catalytic converter technologies.

2.4.2.1 Secondary Air Injection

Secondary air injection is a methodology in which fresh air is injected into the exhaust manifold for the purpose of emissions reduction. Typically, air is injected during the cold-start fuel enrichment phase and provides two benefits. Since exhaust gases are rich during fuel-enrichment, the supplemental oxygen from air injection reacts with unburned hydrocarbons in the exhaust. In addition, reactions in the catalyst are exothermic, which thereby increases the temperature of the exhaust gases and helps to warm up the catalytic converter [31].

In most applications, the secondary air is injected by an air pump added to the vehicle. The air is injected through a one-way valve to prevent engine exhaust from flowing back to the pump. Older secondary air injection systems used passive check valves, but more recent systems have switched to actively-controlled check valves actuated by vacuum pressure, likely due to wave dynamic effects in the exhaust system, causing backflow.
Implementation of a secondary air injection system requires tuning to optimize the control strategy in conjunction with the engine control, because there is a potential for saturating the catalytic converter oxygen storage and this can lead to excess NO\textsubscript{x} emissions. Noise from the air pump can also be a potential issue for vehicle applications.

2.4.2.2 Catalytic Converter Technologies

Different passive and active catalytic converter technologies have been investigated as potential means for reducing cold-start emissions. One passive technology is the hydrocarbon absorber, a device located just upstream of the three-way catalyst used to collect hydrocarbons (HCs) emitted during cold start. Once the main converter has reached light-off, the trapped HCs are released and converted in the TWC. However, most of these devices have been found to lack thermal stability required in the automotive exhaust application. Kiefer et al. show some success in using the HC absorber, but their method required additional heating through a downstream electrically-heated catalyst to ensure that the catalytic converter could achieve high conversion efficiencies during the desorption phase [21].

Another passive technology is the start-up converter which is a small, close-coupled converter located close to or even inside the exhaust manifold and enables very rapid heating of the catalyst. This type of converter can be feasible and relatively low cost but requires the availability of thermally stable washcoats to handle the extreme temperatures of engine-out exhaust gases as well as the local heating generated by exothermic catalytic reactions on the converter surface. Research is also being conducted to develop new
catalysts, such as gold, to achieve high conversion efficiency at low temperatures; however current investigations are still limited by catalyst durability under harsh conditions as well as thermal stability for higher operating temperatures [20].

2.5 Electrically-Heated Catalyst (EHC)
As an alternative, active means of improving cold-start emissions, much research has been done to investigate electrically pre-heating the catalytic converter. The following section describes this technology in greater detail.

2.5.1 EHC Hardware Design
The electrically-heated catalyst (EHC) is an actively-controlled device incorporated into the vehicle exhaust system upstream of the main catalyst for the purpose of improving cold-start emissions. The EHC is made with alternating smooth and corrugated layers of thin, metal foil connected to each other by a brazing process. This foil structure has a resistance of typically 0.05 to 0.35 ohms and is connected to positive and negative electrical terminals for connection to a source of electrical power [6]. The EHC structure is heated through electrical resistive heating and transfers heat to gasses flowing through the EHC. This warmed gas then flows through the main catalyst structure, thereby heating the main catalyst to reach its light-off temperature more quickly than if heated solely by engine-out exhaust gases. Typically, the EHC is integrated into the can of the catalytic converter, as shown in Figure 8.
Since catalytic activity is dependent on the catalytic converter being above light-off, converter temperature is a critical factor for emissions reduction. Therefore, it is important for the EHC to be heated with enough heating energy to sufficiently reach converter light-off. Kubsh and Weber investigated the effect of soak periods on converter efficiency for a 2.5L V-6 underbody catalytic converter through vehicle testing over the Federal Test Procedure (FTP). They varied the length of the hot soak period between Bags 2 and 3 and compared emissions measurements collected during Bag 3 (hot transient cycle). They found that after a 30 minute soak, the catalytic converter temperature had dropped below light-off and an engine restart resulted in 0.191 grams/mi of NMHC, 0.402 g/mi CO, and 0.553 g/mi NO\textsubscript{x} during Bag 3. After a 180 minute soak, the converter had reached ambient temperature, the engine calibration made use of a cold start fuel enrichment strategy, and Bag 3 measurements resulted in 0.449 g/mi NMHC, 1.789 g/mi CO, and 0.593 g/mi NO\textsubscript{x} emissions. Comparable tests with an
EHC show significant improvements, with emissions impact being more beneficial as catalyst temperatures decreased closer to ambient during the soak time [25].

It has been shown that the ideal design is to pair the EHC with a main three-way catalytic converter directly downstream of the EHC. Socha et al. compared various converter configurations with the EHC and main converter in the underbody location versus close-coupled location, as well as having the EHC close-coupled but the main converter in the underbody location. Results from this work show that the most effective strategy is to use the EHC plus main converter in the close-coupled location. Implementing the main converter in an underbody location allows exhaust gases to lose heat before reaching the converter system, hence requiring more electrical energy input to the EHC. Locating the converter near the engine also makes use of heat from the engine after engine crank. In addition, locating the EHC directly upstream of the main converter is desirable because heat from the EHC is transferred directly to the main converter [30]. This study indicates that the function of the EHC is for heating and the TWC main converter performs the catalysis and emissions conversion even during the cold start heating period. Breuer et al. from the EHC manufacturer, Emitec GmbH, discuss design differences for underbody versus close-coupled converter applications [6]. In a close-coupled application, gas temperatures entering the catalytic converter are higher than in an underbody location due to losses along the exhaust pipe. Given these higher gas temperatures enter the close-coupled converter, the catalyst is able to achieve light-off and proper functionality faster than a comparable underbody converter. Therefore, it is advantageous to design a close-
coupled catalytic converter with high cell density and large catalyst diameter (i.e. high catalytic surface area) to enable a greater number of catalytic reactions and reduced tailpipe emissions. Conversely, gas temperatures entering an underbody converter are lower than engine-out gas temperatures and therefore provide less thermal energy into the converter. To minimize the amount of emissions passed through the catalyst before it reaches light-off, it is preferable to select a smaller converter that requires less thermal energy to reach operating temperature. Given these trends, Breuer et al. recommend selecting high catalytic surface (high cell density, large catalyst diameter) for close-coupled applications but optimal heating behavior (low cell density, small catalyst diameter) for underbody applications [6].

In general, it is desirable to reduce the mass of the EHC, because this improves heating rate, reduces back pressure, and reduces electrical energy consumption to self heating. Socha et al. compares a “standard” mass heater (836 grams, 0.16 mm wall thickness) with a lower mass heater (589 grams, 0.11 mm wall thickness), showing that the lower mass heater achieves 995°C using 49 Wh of energy, while the standard mass heater requires 60 Wh to achieve 862°C [30]. In a different work, Kubsh looked at heating the EHC in zones as an alternative to heating the full EHC. He used a 69 mm diameter EHC for all tests and heated different percentages of the EHC cross section. Results of this study showed that for a low-mileage converter only marginal gains in emissions reduction were shown for greater than 44% cross-sectional area heating, although for an
aged catalyst further emissions gains were realized for heating the full 100% cross-sectional area [24]. These results indicate the potential for reduced converter mass.

Holy *et al.* make use of this information by using a “heat cascade” system [18]. This design featured a smaller mass EHC coupled with a support catalyst substrate to act as a light-off converter for fast heating and emissions reduction during cold start. This EHC/support substrate was integrated just upstream of a larger diameter main converter used for increased emissions performance once the engine was fully warmed-up. For a cascade system with small light-off converter (80mm dia., 600 cpsi,) and main converter (110mm dia., 800 cpsi), results show HC conversion efficiency reaching 90% after the first 45 seconds of the FTP and 96% efficiency at 90 seconds. In comparison, for an EHC cascade system (EHC: 80mm dia., 600 cpsi, Support Converter: 80mm dia., 800 cpsi, Main Converter: 110mm dia., 1200 cpsi), 90% HC conversion efficiency was reached in 25 seconds and 98% conversion efficiency being reached after only 50 seconds. Production EHC designs today tend to use a low-mass EHC integrated just upstream of the main converter in the same can package without adding the light-off support substrate.

2.5.2 EHC Emissions System Design

Selection of the EHC and catalytic converter hardware must be done in conjunction with considerations for the entire emissions system, including converter packaging location, air injection, electrical supply, and control strategy.
2.5.2.1 Converter Packaging Location

As discussed in the previous section, it is advantageous to package the EHC and main converter together in either the close-coupled or underbody location. The location of the EHC affects the rest of the system design because it influences the length and routing of electrical cables, which in turn can affect other wire routing and also impacts overall vehicle weight. Location of the EHC also influences battery sizing, since an underbody converter is likely to require more heating power than a close-coupled converter.

2.5.2.2 Air Injection

Air injection in the exhaust manifold upstream of an EHC converter system has been shown to provide appreciable reduction in HC and CO with marginal trade-off in NO\textsubscript{x} emissions [25][30]. Including an air pump enables pre-heating the EHC before engine crank, and therefore provides more versatility to optimize the EHC emissions system.

2.5.2.3 Electrical Supply

The vehicle electrical system must be able to source the power demanded by the EHC system in addition to any other required electrical demands from the vehicle. The EHC power demand is dictated by the resistance of the EHC and cables, as well as the air injection pump if included. Design of the electrical system also influences EHC heating time.

Most EHC systems make use of the vehicle’s 12-volt electrical system to provide the energy required for heating. However, the current demand on the system can be upwards of 200-amps. To source these significant power demands, Kubsh \textit{et al.} used an off-
vehicle 12-volt, 1340-cold cranking amp marine battery [25]. Heimrich et al. added a second 12-volt battery in series with the original vehicle battery to achieve a 24-volt power supply [16]. Socha et al. also used a 24-volt battery system [30]. Hanel et al. used the original vehicle 12-volt battery and replaced the stock alternator with a more powerful one [12]. They performed a cyclic analysis comparing the minimum battery voltage during engine cranking for three different batteries (110-Ah, 90-Ah, 70-Ah) over 5,000 engine crank/EHC heating cycles. The 70-Ah battery did not maintain its initial minimum voltage for more than 2,000 cycles, the 90 A-h battery began dropping below its initial minimum voltage after about 3,000 cycles, while the 110-Ah battery was able to maintain its initial minimum voltage through the entire 5,000 cycles. Therefore, electrical supply for the EHC system is a critical design factor that balances trade-offs between EHC heating performance and practical application in a vehicle system.

2.5.2.4 Control Strategy

Much investigation has been done on heating the EHC before crank (pre-heat), after crank (post-heat), or both. For best reduction in cold start emissions, it is desirable to use a pre-heat strategy that transfers heat from the EHC to the front of the catalyst substrate to reach light-off prior to engine crank [16][30]. It has been found that initial exhaust gas flow after engine crank can temporarily cool the catalyst substrate, so there can be benefit to also including post-heating. Cooling can also be caused due to air injection without EHC heating.
It is imperative that the EHC control strategy be designed with consideration for the engine’s cold start fuel enrichment calibration. For example, if a post-heat strategy with air injection is used, the air injection would need to be finished by the time the engine control switches from open-loop to closed-loop control, otherwise the oxygen sensors measurements will be lean and the feedback control will command more fuel than necessary for the desired in-cylinder combustion. Air injection also affects the level of oxygen storage in the catalyst, which impacts the efficiency of the catalytic converter under rich and lean engine excursions.

Kubsh notes that aged catalytic converters were shown to require more heating than low-mileage converters [24]. Therefore, there could also be some benefit to accounting for catalyst aging in the EHC heating strategy.

2.5.3 EHC Vehicle Applications
EHCs have been implemented in production vehicles, mostly in low-volume vehicle platforms. Driven by increasingly stringent European and U.S. emissions regulations, 30,000 to 50,000 Emitec EHC units were integrated into the BMW ALPINA B12, based on a BMW 7 series limousine, from 1991 to 1996 [12][13]. Incorporating an EHC into full-volume production presents a trade-off between emissions reduction versus added weight, cost, and electrical demands. As a result of these trade-offs and the emissions improvements that have been achieved through sophisticated emission control techniques, EHC technology is not prevalent in mass production vehicles today.
2.6 Catalyst Modeling

Catalytic converter models generally fall into one of three main categories: detailed physical models, simplified kinetic models, and highly-simplified models [27][29].

2.6.1 Detailed Physical Models

Detailed physical models are developed based on fundamental principles of chemical kinetics and thermofluidic dynamics occurring in reacting flows. These models describe the heat and mass transfer in one, two, or three dimensions and account for changes in composition of the exhaust gases as they continually react with one another in the catalytic environment. The most detailed models include phenomena such as non-uniform flow distribution throughout the catalytic converter and oxygen storage reactions [27][29].

An example is the detailed model developed by [14]. They modeled a single channel for a three-way catalytic converter that includes a kinetic reaction model and a transport model. The kinetic reaction model encompassed four global reactions, each of which involved multiple steps to total 31 reaction steps, to describe the interaction between the active sites and the reactants in a series of adsorption-desorption and surface reaction steps. For the transport model, a one-dimensional gas model was coupled with a two-dimensional model for the washcoat using heat and mass transfer coefficients. The washcoat was modeled as an annulus of 50 μm thickness. The model was solved with an iterative Newton-Krylov solver implemented on a parallel supercomputer due to the large system of highly non-linear partial differential equations that resulted from the great
number of species present in the reactions. The model predicted light-off performance and showed that the 2-D model predicted a slightly later light-off than a 1-D model that did not include washcoat diffusion.

Detailed physical models such as this require an extensive number of reaction rates and physical parameters, and therefore rely heavily on historical or experimental data. As a result, detailed kinetics models are not appropriate for embedded control applications due to their complexity and significant computation. However, detailed physical models are useful for modeling detailed catalytic performance to enhance the design of catalysts for TWC applications.

2.6.2 Simplified Kinetic Models

Simplified kinetic models are used to represent the reactions between the various gas components and the dynamics of gas storage on the catalyst surface. These models are often spatially discretized and represented by coupled, lumped-parameter differential equations. Therefore, these models require the input gas concentrations of each component to be measured or predicted. Since this is difficult in a practical application, simplified kinetic models are not as widely used in embedded control applications. Usually simplified kinetic models are used as rapid development tools to study the effects of design parameters such as geometry and cell density [27][29].
2.6.3 Highly-Simplified Models

The third class of models is developed for embedded control applications. These simplified models tend to be first-order sub-models that include warm-up characteristics, oxygen storage, and static conversion efficiency curves. Typically, the catalytic converter is represented as a single lumped element. These models consider oxygen storage to be the dominant dynamic process inside the catalyst, with other kinetics occurring over a much shorter and less significant timescale. As a result, simplified control-oriented models are highly empirical and based on large amounts of data collected over widely ranging operating conditions.

An example of a simplified model was developed by [29] to assess cold start behavior. Their model represented the catalytic converter as a single lumped element and included three sub-models. The first sub-model described oxygen storage as a function of catalyst input equivalence ratio. This sub-model evaluated the adsorption and release of oxygen from the catalyst washcoat by assessing the relative amount of oxygen stored on the surface of the catalyst. As the washcoat becomes saturated, the adsorption rate tends to zero and as the oxygen in the washcoat is depleted, the release rate tends to zero. The second sub-model described conversion efficiency as a function of input equivalence ratio and bulk catalyst temperature. These efficiency curves were modeled as a Weibe function and fitted experimentally at the 50% conversion efficiency point. The third sub-model was a thermal dynamics model developed as a lumped thermal model to estimate the temperature in the middle of the catalytic converter based on enthalpy of the exhaust.
gas, heat transfer from the TWC to the surroundings, and heat generation from the exothermic catalytic reactions. The thermal dynamics were described as a function of exhaust gas temperature, catalyst efficiency curves, exhaust mass flow, emissions concentrations, and air/fuel ratio. Model results showed good correlation with experimental data. Notably, results of the model indicated that oxygen storage potential of the catalyst during cold-start is minimal. This is attributed to the fact that engine operation is usually rich during cold-start; as a result, there is little oxygen in the exhaust for the catalyst to store and there are no transient air/fuel ratio excursions to allow the adsorption/desorption mechanism of oxygen storage.

2.7 Additional Emissions Considerations for Hybridized Vehicles

The added complexity of hybrid vehicles in comparison to conventional vehicles presents further considerations for optimizing emissions. This section provides a broad overview of emissions considerations in four categories:

- Cold-start emissions for conventional vehicles
- EHC-assisted cold-start for conventional and HEVs (no pre-heat)
- EHC-assisted cold-start for conventional and HEVs with pre-heat
- Hot re-start for HEVs

Conventional vehicles have limited potential for improving cold-start emissions. Since the engine is started at key-on, engine-out emissions are produced immediately. Therefore, the goal in a conventional vehicle is to reduce the amount of time it takes to
reach the catalytic converter light-off temperature after engine start, and there is little potential for completely eliminating the excess emissions due to starting at temperatures below catalyst light-off.

EHC-assisted cold-start in which EHC heating starts at key-on (no pre-heat) has limited potential for improving cold-start emissions in either a conventional or HEV. Although the EHC would be able to provide some added thermal benefit, this control strategy still does not eliminate the initial flow of engine-out emissions through the catalyst below light-off during initial start-up.

Using the EHC to pre-heat the catalytic converter before engine crank has been shown to be more effective than beginning the heating at engine crank [30]. The pre-heat strategy enables the catalyst to be at operating temperature before any engine-out emissions are produced. However, for a conventional vehicle, pre-heating prior to engine crank prevents torque delivery to the wheels; therefore, a pre-heat time of 10 seconds or more would be undesirable. In addition, implementing an EHC in a conventional vehicle is limited by the capability of the 12V battery to source the energy. Therefore, there is limited benefit from EHCs in conventional vehicle applications. Conversely, there is greater potential for a catalyst pre-heat strategy in hybrid vehicle applications. The greater on-board energy storage capacity of HEVs is more suited to source the electrical energy demands of an EHC. Furthermore, the versatility of a combined engine and electric powertrain could be able to pre-heat before engine crank while still achieve nearly instantaneous torque delivery to the wheels through the electric powertrain.
In terms of warmed-up operation, conventional vehicles are able to achieve nearly 100% conversion efficiency with good air/fuel control. Since conventional vehicles require continuous engine operation, the catalytic converter remains above minimum operating temperature and there is no concern for re-start emissions unless the vehicle is turned off. These emissions performance trends are no longer true for all vehicles given the increased variety of hybrid vehicles entering the market. Many hybrid vehicles have features for engine stop-start strategies, in which the engine is turned off instead of idling and the vehicle is powered electrically. This method reduces fuel consumption, but the by-product is a spike in NO\textsubscript{x} emissions during each re-start event [26]. Furthermore, depending on the length of the stop period, it is possible that the catalyst temperature could drop below light-off, causing further emissions excursions upon re-start. This would especially be an issue for PHEVs that have an appreciable charge depletion range.

This thesis investigates the emissions benefits of using of an electrically-heated catalyst in a plug-in hybrid electric vehicle.
3.1 Introduction

A plug-in hybrid electric vehicle platform is well-suited for an electrically-heated catalyst for three main reasons. First, since PHEVs include substantial on-board electrical energy storage, most commonly through a high voltage battery system, there is opportunity for sourcing the significant amount of current required by the EHC system, which can easily be 100-200A. Second, a PHEV with any amount of charge depletion range provides more flexibility for engine start-up because the electric powertrain is able to provide all the power necessary for driving the vehicle until charge is depleted to the threshold of the energy storage system. If the vehicle does not require power from the engine immediately after key-on, there is opportunity to pre-heat the catalytic converter before the first combustion event occurs. This can significantly improve cold-start emissions. Finally, many HEV and PHEV applications implement engine stop/start control strategies, in which the engine is stopped during periods of idle and started again with an acceleration request. During the period of time that the engine is stopped, if the substrate of the catalytic converter cools to a temperature below the light-off temperature there will be excess tailpipe emissions upon engine re-start until the catalyst again reaches its
operating temperature. In light of these benefits, an electrically-heated catalyst was implemented in the Ohio State University EcoCAR plug-in hybrid electric vehicle.

### 3.2 Catalytic Converter Selection

The stock catalytic converter that came with the original Honda engine was a close-coupled ceramic monolith with a split-bed configuration that included one pre-catalyst universal exhaust gas oxygen (UEGO) sensor and a second UEGO sensor between the first and second catalyst bricks (see Figure 9). This converter used a single exhaust manifold integrated into the flange of the catalyst, allowing it to bolt directly to the engine (see Figure 9).

![Figure 9: Stock engine and original close-coupled catalytic converter](image)

The stock converter was replaced with a new catalytic converter tailored for the OSU EcoCAR vehicle. The converter selected for the OSU EcoCAR vehicle was an Emicat® catalytic converter manufactured by Emitec GmbH. This converter was selected to have a high cell density for greater geometric surface area and metal foil substrate for greater
heat transfer. For improved cold-start performance, the converter was also equipped with an electrically-heated catalyst integrated upstream of the main substrate. Catalytic converter specifications are provided in Table 2. Further discussion of the exhaust aftertreatment hardware and software is discussed in Section 3.

Table 2: Specifications for OSU EcoCAR catalytic converter

<table>
<thead>
<tr>
<th>Manufacturer / Type</th>
<th>Emitec GmbH / Emicat® Serie 6d</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cell Density</td>
<td>400 cpsi</td>
</tr>
<tr>
<td>Substrate Type</td>
<td>Metal Foil</td>
</tr>
<tr>
<td>Platinum Metal Group Loading</td>
<td>155 g/ft³ Pd; 15 g/ft³ Rh</td>
</tr>
<tr>
<td>Mounting Location</td>
<td>Close-Coupled</td>
</tr>
<tr>
<td>Other</td>
<td>Electrically-Heated Catalyst</td>
</tr>
<tr>
<td>Sensors</td>
<td>UEGO Wideband O₂ Sensor: Pre-catalyst</td>
</tr>
<tr>
<td></td>
<td>EGO Switching O₂ Sensor: Post-catalyst</td>
</tr>
<tr>
<td></td>
<td>3 Resistance Temperature Devices (RTDs): Pre-catalyst, Mid-catalyst, Post-catalyst</td>
</tr>
</tbody>
</table>

3.3 System Configuration

Figure 10 shows a schematic of the exhaust aftertreatment system implemented in the Ohio State University EcoCAR PHEV.
The catalytic converter was an integrated EMICAT EHC and three-way catalyst (TWC) converter design developed by Emitec GmbH (see Table 2 for detailed specifications). The EHC element was a 0.08 ohm nominal resistance heater powered by the vehicle 12-V electrical system. The vehicle 12 V electrical system was powered by a 13.8 A-h lithium-ion battery capable of 240 A at a minimum of 10.8 V, supplemented by a 2.2 kW DC-DC converter between the 12 V battery and the high voltage 360 V lithium-ion battery pack. Current to the EHC was high-side driven by a Kilovac contactor used to open and close the circuit. The contactor was controlled by the engine control unit. The circuit was fused with a 120A circuit breaker.
An air injection system was used to provide airflow into the exhaust manifold upstream of the EHC during the EHC heating phase. The air injection system used an air pump controlled by a pulse-width modulated driver circuit to enable variable flow rate. Airflow rate was measured by an in-line orifice meter with a differential pressure sensor. The signal from the differential pressure sensor was sent to the engine control unit and converted into a corresponding mass flow rate determined through calibration. Prior to the exhaust manifold, the airflow routed through an electrically-actuated valve. The valve was normally-closed to prevent exhaust backflow from the exhaust manifold to flow into the air injection lines and to the air pump. Hence, the valve served dual purposes. First, the valve prevented damage to the air pump caused by hot temperatures and/or condensation build-up due to exhaust backflow. Second, the valve prevented undesirable disturbances to the air-fuel ratio closed-loop control that would be caused by uncontrolled net airflow into the catalyst through the air injection system due to wave dynamics in the exhaust system. The effects of these wave dynamics is illustrated in Figure 11. Originally, the valve between the exhaust manifold and the secondary air injection system was a passive check valve. Due to pressure waves in the exhaust, there were times that the pressure in the exhaust manifold was lower than the atmospheric pressure in the secondary air injection system. This pressure drop across the passive check valve allowed airflow through the valve into the exhaust manifold. This was observed experimentally and the passive check valve was replaced with an electrically-actuated valve that could be commanded closed when the secondary air injection system was not active.
Feedback was used to assess the state of the EHC and catalytic converter and determine control actions. The catalytic converter was instrumented with resistance temperature devices (RTD) in three locations: pre-catalyst, mid-catalyst, and post-catalyst. The mid-catalyst thermocouple was located in the substrate at the front of the TWC (see Figure 10). The catalytic converter was also instrumented with a pre-catalyst universal exhaust gas oxygen (UEGO) sensor for measuring wideband equivalence ratio of the exhaust gas and a post-catalyst switching exhaust gas oxygen (EGO) sensor for determining small correction factors to the air-fuel ratio control. Feedback from these sensors were read by the ECU.

3.4 Control Strategy Development

Control of the EHC was performed by the ECU. All ECU software was developed in the Woodward MotoHawk block diagram environment. EHC control was incorporated as an
engine start up sequence dictated by a Stateflow diagram in Simulink. A state machine representation of the modes and transitions is shown in Figure 12.

![State machine diagram](image)

<table>
<thead>
<tr>
<th>Transition</th>
<th>Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>S12</td>
<td>WarmUp_Req==1 &amp;&amp; O2WarmUpComplete==0 &amp;&amp; EHC_Bypass==0</td>
</tr>
<tr>
<td>S21</td>
<td>WarmUp_Req==0</td>
</tr>
<tr>
<td>S23</td>
<td>WarmUp_Req==1 &amp;&amp; O2WarmUpComplete==1 &amp;&amp; (MidCatTemp&gt;LightOffTemp</td>
</tr>
<tr>
<td>S32</td>
<td>MidCatTemp&lt;LightOffTemp</td>
</tr>
<tr>
<td>S14</td>
<td>WarmUp_Req==1 &amp;&amp; O2WarmUpComplete==0 &amp;&amp; EHC_Bypass==1</td>
</tr>
<tr>
<td>S41</td>
<td>WarmUp_Req==0</td>
</tr>
<tr>
<td>S45</td>
<td>WarmUp_Req==1 &amp;&amp; O2WarmUpComplete==1</td>
</tr>
</tbody>
</table>

Figure 12: State machine representation of engine start-up sequence with EHC control

This sequencing is initiated by an engine warm-up request (WarmUp_Req) sent over the CAN network from the supervisory controller. Typically this warm-up request is sent once the vehicle’s high voltage battery state-of-charge reaches the lower threshold for charge depleting mode, indicating that the supervisory controller is anticipating a switch into charge sustaining mode. The engine control unit uses this warm-up request to enter
the warm-up sequence, which can be set to include EHC heating or bypass EHC heating. Both sequences involve a 50-second period for warming up the oxygen sensors; for EHC heating, the oxygen sensor warm-up occurs in parallel with EHC warm-up. Once warm-up is complete, the ECU sends a handshake message back to the supervisory controller indicating that the engine is warmed up and ready to start providing torque.

Diagnostics for the EHC system were performed in three ways. First, air flow across the EHC was required during heating to prevent overheating and damage to the EHC. Air flow through the secondary air injection system was measured by the differential pressure sensor and EHC heating was not allowed unless the airflow was greater than the specified minimum mass air flow. Second, the mid-catalyst RTD measurement was located at the front of the main converter as close to the EHC as possible. This temperature measurement was used to assess functionality of the EHC and to provide feedback for preventing EHC overheating as well as determining when to stop the warm-up phase. Finally, a timer was implemented in the software to keep track of how long the EHC had been heating. If the timer reached the specified time-out value, the warm-up sequence would be forced out of the heating mode and into the warm-up complete mode, which would turn the EHC and air pump off.

3.5 EHC Integration and Testing In-Vehicle

A challenge with integrating an electrically-heated catalyst into a vehicle application was the high power demand of the EHC. With a typical system (low) voltage of 12-14V, the current drawn by the EHC would be between 150-165A. This high current demand was
difficult to source with only the stock 12V lead acid battery in the vehicle. One option was to add a second battery in series to achieve 24V across the terminals of the EHC. However, this would add extra weight, plus extra electrical cable since the low voltage battery was located in the rear of the vehicle far from the EHC. Furthermore, it would require more sophisticated control and/or power circuitry. This added hardware and software was undesirable, especially considering the 24V source would be used for the EHC for only a few minutes at most. Therefore, it was necessary to use only 12V to power the EHC.

Being a plug-in hybrid electric vehicle, the Ohio State EcoCAR vehicle provided a unique opportunity to make use of power from the high voltage battery to supplement the system voltage 12V battery. This transfer of power was done through an on-board DC-DC converter. Initial testing indicated that the majority of current was being supplied through the DC-DC converter due to a decrease in voltage across the lead acid battery during EHC heating.

To minimize this impact, the DC-DC converter’s calibrated system voltage setpoint was increased from 13.7V to 14.0V. Although this did reduce the drop in system voltage, the system voltage still dropped from 14.2V to 12.1V during EHC heating. To further mitigate this sharp plummet in system voltage, the lead acid 12V battery powering the vehicle was replaced with a lithium-ion battery. This hardware modification was desirable because lithium-ion batteries have less voltage drop during discharge than comparable lead acid batteries. For the Ohio State EcoCAR application, maintaining
higher voltage across the 12V battery would require less current to be sourced through the DC-DC. This, in turn, would be less taxing on the entire vehicle electrical system. Figure 13 compares the voltage drop measured in the Ohio State EcoCAR vehicle during EHC heating for the lead acid (PbA) versus the lithium-ion (Li-Ion) 12V battery. With the lead acid battery, the system voltage drops 2.2V, but with the lithium-ion battery, the system voltage only drops 1.6V. In addition, the lead acid battery required a higher DC-DC voltage setpoint and therefore the nominal voltage (voltage during non-EHC heating periods) sat higher at 14.3V, while the nominal voltage with the lithium-ion battery was able to sit lower at 14.1V.

![Graph showing system voltage during EHC heating for PbA versus Li-Ion 12V battery measured in vehicle](image.png)

Figure 13: System voltage during EHC heating for PbA versus Li-Ion 12V battery measured in vehicle

Further results of vehicle testing are presented in Chapter 6.
3.6 Engine Dynamometer Experimental Setup

Two engines were secured by the Ohio State University EcoCAR team and identical E85 modifications were made to both. One engine was instrumented in an engine dynamometer test cell for software development and calibration. The second engine was installed into the EcoCAR vehicle. The engine test cell was a four-quadrant 200-hp DC dynamometer used to control engine speed. The test cell was equipped with a dual sample line Horiba MEXA 7500 emissions analyzer with the capability to measure dry CO, NOx, O2, wet THC and CO2.

The engine was water-cooled with a cooling tower containing a built-in thermostat to regulate coolant temperature, and an external electric water pump was used in place of the stock belt-driven engine water pump. The fuel system used in the engine test cell was a stock fuel tank and fuel pump from the 2008 Chevrolet HHR, compatible with E85 fuel and connected to a custom fuel rail with E85-compliant fuel injectors.

Data acquisition was performed using a combination of ETAS modules and ECU sensor inputs. The ETAS modules were the ETAS ES410 (8 channel analog module), ETAS ES411 (4 channel analog module with sensor supply voltage), and ETAS ES420 (8 channel thermocouple module) to enable BNC and thermocouple connections for measurements such as EHC current and voltage, barometric pressure, humidity, etc. These modules were connected to one another and then connected to INCA through an Ethernet connection. INCA communicates with the ECU through CAN Calibration.
Protocol (CCP) and interfaces with the user through a laptop connected to the CCP network via a PCMCIA card.

INCA was primarily used as an experimental tool, although calibrations can also be modified in INCA. INCA was the user interface used when running engine tests and allowed for the control of all parameters inside the ECU including control parameters such as throttle position and spark timing. INCA was also used to record data. INCA is advantageous because it works easily with the ETAS modules, allowing for seamless acquisition of data.

The remaining sensors were attached to the Woodward 128-pin ECU. The ECU uses Matlab SIMULINK in conjunction with a MotoHawk development library, which allows for the interfacing with the Woodward ECU. MotoTune was used for flashing code to the ECU, as well as indexing and updating calibration files for new software builds.
4.1 Introduction

Most three-way catalytic converter models are either simplified mean-value models or continuous models describing thermal/fluid dynamics and chemical kinetics in varying levels of detail. Mean-value models describe the average behavior of the entire catalytic converter but lose detail of the internal behavior of the converter. Thermal/fluid dynamics and chemical kinetics models are more descriptive, but are complex, require many parameter definitions, and are computationally intensive. For investigating cold start, it is useful to have a lumped-parameter thermal model that describes the heat transfer mechanisms within the TWC without requiring complex solvers and a large number of parameters that need to be defined or calibrated. Therefore, a one-dimensional (1-D) lumped-parameter heat transfer model was developed to investigate the thermal behavior of a TWC.

4.2 System Description

This section discusses the thermal behavior of a three-way catalytic converter. Specifically addressed are differences in axial and radial temperature distributions.
4.2.1 Axial Thermal Behavior

A qualitative depiction of the three-way catalytic converter response to an increase in input gas temperature is shown in Figure 14. During warm-up, the front of the catalytic converter heats up first and subsequently the rest of the catalytic converter warms up. This is, in effect, a thermal wave propagating along the length of the catalytic converter.

Figure 14: Illustration of thermal gradient along catalytic converter axis during warm-up

Figure 15 shows this behavior quantitatively. Internal temperatures were measured at locations along the axis of the catalytic converter and the thermal response to a step increase in inlet gas temperature was recorded. As shown, the front of the catalytic converter ($T_1$) heats first and similar temperature rises are observed subsequently for each adjacent thermocouple location.
The temperature of the catalytic converter is also influenced by heat generation caused by catalyzed exothermic reactions occurring on the surface of the catalytic converter. These reactions occur for engine-out exhaust gas flowing through segments of the catalytic converter that are above light-off temperature. Hence, this heat generation is not a factor during initial cold start warm-up. The phenomena of exothermic reaction heat generation has been observed experimentally. An example of this trend is shown for vehicle data collected over a Highway Fuel Economy Test (HWFET) drive cycle at steady-state operation, showing that the internal catalyst temperature (Mid-cat) is higher than the pre-catalyst temperature (Pre-cat).
4.2.2 Radial Thermal Behavior

The catalytic converter also has a temperature distribution in the radial direction. Figure 17 illustrates the flow distribution of gas entering the catalytic converter, showing that the flow rate is greatest at the center. Since the higher flow rate provides greater enthalpy input and correspondingly higher heat transfer, this flow profile results in higher temperatures at the center of the catalytic converter. An illustration of the temperature gradient is shown on the right side of Figure 17.
This phenomena of higher temperatures at the center of the catalytic converter was observed experimentally. An example is shown by the engine data presented in Figure 18. Temperature was measured by thermocouples placed at the center and mid-radius of the catalytic converter at the same axial length along the substrate. As shown, for a step increase in inlet gas temperature, the temperature at the center of the converter was consistently higher than the temperature measured closer to the edge of the substrate.

Figure 18: Experimental data showing higher temperature measured at center of catalytic converter for a step increase in engine-out exhaust gas temperature

The thermal model developed in this research used a mean-value approach that discretized the catalytic converter in the axial direction only; therefore, no thermal gradients along the radius of the catalytic converter substrate were incorporated. Based on experimental findings that show a temperature gradient in the radial direction, the 1-D
model could be extended to 2-D to include radial heat transfer; however, this was outside the scope of this thesis.

4.3 Model Description

The 1-D lumped parameter model was developed as a physics-based model accounting for enthalpy, convection, and conduction heat transfer mechanisms occurring in the catalytic converter system. The catalytic converter was discretized axially into six disc-shaped elements, as indicated in Figure 19. Three control volumes were contained within each discrete element: gas, converter substrate, and can wall. A temperature state was defined for the mean temperature of each of these control volumes, resulting in a total of 18 temperature states (6 mean gas temperatures, 6 mean substrate temperatures, and 6 mean can temperatures). Gas flow was assumed to be incompressible and follow the conservation of mass equation in which mass of gas flow into the control volume equals the mass of gas flow out of the control volume; therefore, no storage of gas was allowed within a control volume.
The following heat transfer mechanisms were identified to describe the thermal behavior between these discrete elements:

- $\dot{H}_g$: Enthalpy of the exhaust gas entering and exiting the $i^{th}$ control volume
- $\dot{Q}_{\text{conv},g,s}$: Convective heat transfer between the gas and substrate in the $i^{th}$ element
- $\dot{Q}_{\text{cond},s,up}$: Upstream conduction between the $i^{th}$ and $(i-1)^{th}$ substrate elements
• $\dot{Q}_{\text{cond},s,\text{down}}$: Downstream conduction between the $i^{th}$ and $(i+1)^{th}$ substrate elements

• $\dot{Q}_{\text{cond},s,c}$: Conduction between the substrate and can in the $i^{th}$ element

• $\dot{Q}_{\text{cond},c,\text{up}}$: Upstream conduction between the $i^{th}$ and $(i-1)^{th}$ can elements

• $\dot{Q}_{\text{cond},c,\text{down}}$: Downstream conduction between the $i^{th}$ and $(i+1)^{th}$ can elements

• $\dot{Q}_{\text{conv},c,\text{amb}}$: Convective heat transfer between the $i^{th}$ can element and ambient

These heat transfer mechanisms are portrayed graphically in Figure 20. The designation $\frac{dU}{dt}$ is used to describe the change in internal energy of gas, substrate, or can, respectively.

![Figure 20: Heat transfer mechanisms within one discrete element](image)

The model was further extended to incorporate the heat generation and heat transfer of an electrically-heated catalyst integrated into the can of the catalytic converter. The catalytic converter model does not include heat generated by exothermic catalytic reactions because the EHC application uses a pre-heat strategy; however, this could included as an
additional positive heat transfer in which a fraction of the heat generation is attributed to
the substrate and a fraction is attributed to the gas. An air pump provides flow during
EHC heating before engine start; therefore only air, not exhaust, is flowing the converter
during EHC cold-start warm-up. Although calibration data was collected with a firing
engine, the instrumented catalytic converter was coated with neither a washcoat nor
catalyst loading. Furthermore, since catalytic material is generally not active at low
temperatures, the contribution of heat generation due to exothermic reactions was
assumed to be small [29].

4.4 Mathematical Formulation

This section defines the equations used to describe the heat transfer through the catalytic
converter.

4.4.1 Exhaust Gas Heat Transfer

The total heat transfer equation for the exhaust gas control volume in the \(i^{th}\) element is
shown in Equation (8) to describe the change in internal energy of the gas, \(\frac{dU_g}{dt}\). Heat is
added by the enthalpy of the incoming exhaust gas, \(\dot{H}_{g,in}\), and heat is removed through
the enthalpy of the exiting exhaust gas, \(\dot{H}_{g,out}\), and convection from the gas flowing over
the surface area of the substrate, \(\dot{Q}_{\text{conv},g,s}\).

\[
\frac{dU_g}{dt} = (\dot{H}_{g,in} - \dot{H}_{g,out}) - \dot{Q}_{\text{conv},g,s} \tag{8}
\]
As shown in Equation (9), the change in internal energy is a function of the mass of the gas within the control volume, \( m_g \), the specific heat of the gas, \( c_{p,g} \), and the change in mean gas temperature, \( \frac{dT_g}{dt} \), within the discretized element.

\[
\frac{dU_g}{dt} = m_g c_{p,g} \frac{dT_g}{dt} \tag{9}
\]

Equations (10) and (11) describe the enthalpy of incoming and exiting gas, respectively, as a function of gas mass flow rate, \( \dot{m}_g c_{p,g} \), specific heat of the gas, \( c_{p,g} \), and the temperature of the gas entering or exiting the control volume.

\[
\dot{H}_{g,in} = \dot{m}_g c_{p,g} T_{g,in} \tag{10}
\]

\[
\dot{H}_{g,out} = \dot{m}_g c_{p,g} T_{g,out} \tag{11}
\]

The convective heat transfer between the gas and substrate is described by Equation (12), where \( h_{gs} \) is the heat transfer coefficient between the gas and substrate, \( A_{gs} \) is the geometric surface area of the substrate, and \( T_g \) and \( T_s \) are the temperatures of the gas and substrate, respectively, within the \( i^{th} \) element.

\[
\dot{Q}_{conv,g,s} = h_{gs} A_{gs} (T_g - T_s) \tag{12}
\]

### 4.4.2 Substrate Heat Transfer

The total heat transfer equation for the substrate control volume in the \( i^{th} \) element is shown in Equation (13) to describe the change in internal energy of the substrate, \( \frac{dU_s}{dt} \).
Heat is added by convective heat transfer from the gas to the substrate, \( \dot{Q}_{\text{conv,g,s}} \), and conduction from the adjacent upstream substrate element, \( \dot{Q}_{\text{cond,s,up}} \). Heat is removed by conduction to the adjacent downstream substrate element, \( \dot{Q}_{\text{cond,s,down}} \) and conduction from the substrate to the can, \( \dot{Q}_{\text{cond,s,c}} \).

\[
\frac{dU_s}{dt} = \dot{Q}_{\text{conv,g,s}} + \dot{Q}_{\text{cond,s,up}} - \dot{Q}_{\text{cond,s,down}} - \dot{Q}_{\text{cond,s,c}} \tag{13}
\]

The change in internal energy of the substrate is defined in Equation (14) as a function of the substrate mass within the control volume, \( m_s \), the specific heat of the substrate material, \( c_{p,s} \), and the change in mean substrate temperature within the control volume, \( \frac{dT_s}{dt} \).

\[
\frac{dU_s}{dt} = m_sc_{p,s} \frac{dT_s}{dt} \tag{14}
\]

Conduction between substrate elements is described by Equation (15) for upstream conduction and Equation (16) for downstream conduction. This conduction is dependent on the conductive area of the substrate, \( A_s \), the conductive heat transfer coefficient, \( k_s \), the length between temperature states along the axis of conduction, \( L_s \), and the mean substrate temperatures of the \( (i-1)^{\text{th}} \), \( i^{\text{th}} \), and \( (i+1)^{\text{th}} \) elements, \( T_{s,\text{in}}, T_s, \) and \( T_{s,\text{out}} \), respectively. Upstream conductive heat transfer is included for all elements except the first element and downstream conductive heat transfer is included for all elements except the last element.
\[
\dot{Q}_{\text{cond},s,up} = \frac{A_s k_s}{L_s} (T_{s,in} - T_s)
\]

\[15\]

\[
\dot{Q}_{\text{cond},s,down} = \frac{A_s k_s}{L_s} (T_s - T_{s,\text{out}})
\]

\[16\]

The conduction between the substrate and the can is represented as cylinder conduction described by Equation (17). Due to the nature of the substrate design with channels of gas flowing through the material, the radial conduction through the substrate to the can does not have a readily defined conduction geometry. Therefore, the factor \(K_{sc}\) is used as a lumped heat transfer coefficient to describe the conductive term.

\[
\dot{Q}_{\text{cond},s,c} = \frac{1}{\left( \frac{\ln(r_2/r_1)}{2\pi kL} \right)} (T_s - T_c) = K_{sc}(T_s - T_c)
\]

\[17\]

### 4.4.3 Can Wall Heat Transfer

The total heat transfer equation for the substrate control volume in the \(i^{th}\) element is shown in Equation (18) to describe the change in internal energy of the can, \(\frac{dU_c}{dt}\). Heat is added by conductive heat transfer from the substrate to the, \(\dot{Q}_{\text{cond},s,c}\), and conduction from the adjacent upstream can element, \(\dot{Q}_{\text{cond},c,up}\). Heat is removed by conduction to the adjacent downstream substrate element, \(\dot{Q}_{\text{cond},c,down}\) and conduction from the can to ambient, \(\dot{Q}_{c,\text{amb}}\).
The change in internal energy of the can is defined in Equation (19) as a function of the can mass within the control volume, $m_c$, the specific heat of the can material, $c_{p,c}$, and the change in mean can temperature within the control volume, $\frac{dT_c}{dt}$.

\[
\frac{dU_c}{dt} = m_c c_{p,c} \frac{dT_c}{dt} \tag{19}
\]

Conduction between can elements is described by Equation (20) for upstream conduction and Equation (21) for downstream conduction. This conduction is dependent on the conductive area of the can, $A_c$, the conductive heat transfer coefficient, $k_c$, the length between temperature states along the axis of conduction, $L_c$, and the mean can temperatures of the $(i-1)^{th}$, $i^{th}$, and $(i+1)^{th}$ elements, $T_{c,in}$, $T_c$, and $T_{c,out}$, respectively.

\[
\dot{Q}_{\text{cond},c,up} = \frac{A_c k_c}{L_c} (T_{c,in} - T_c) \tag{20}
\]

\[
\dot{Q}_{\text{cond},c,down} = \frac{A_c k_c}{L_c} (T_c - T_{c,out}) \tag{21}
\]

The convection between the can and ambient is described by Equation (22). This convective heat transfer is a function of the heat transfer coefficient, $h_{c,amb}$, the convective area between the can and ambient, $A_{c,amb}$, and the difference between the mean can temperature of the $i^{th}$ element, $T_c$, and the ambient temperature, $T_{amb}$.

\[
\dot{Q}_{\text{cond},c,amb} = h_{c,amb} A_{c,amb} \left( T_c - T_{amb} \right) \tag{22}
\]
\[ \dot{Q}_{c,amb} = h_{c,amb}A_{c,amb}(T_c - T_{amb}) \]  

(22)

4.4.4 Transport Delay

By the nature of a lumped parameter model, this model defined a mean value of temperature within each discrete element at a given time and described heat transfer directly from one (mean) state to the next (mean) state. However, in the real system, heat is transferred along a continuum, meaning heat is transferred to adjacent molecules, then transferred to the next adjacent molecules, and so on. Since the lumped parameter model models the average temperature of all molecules within the control volume, it is necessary to account for the transport of heat transfer along the axial length of the catalytic converter. This was modeled as a transport delay between states of adjacent elements. The gas temperature of the \( i^{\text{th}} \) element was subjected to a transport delay before being received as an input to the \((i+1)^{\text{th}}\) element, and so on for the substrate and can temperatures, respectively. The amount of transport delay was calibrated separately for gas \( (t_{del,g}) \), substrate \( (t_{del,s}) \), and can \( (t_{del,c}) \) delays. These values were assumed to be constant for all six discrete elements.

4.4.5 EHC Heat Generation

The electrically-heated catalyst was represented as an element directly upstream of the first catalytic converter element. This element is similar to the main converter elements in that it includes heat transfer due to enthalpy, convection from gas to substrate, conduction from substrate to can, upstream and downstream conduction of the can, and convection...
from can to ambient, but does not include upstream or downstream conduction of the substrate because it is not connected in either direction. An additional heat generation term was included in the substrate heat transfer to represent the resistive heating behavior of the EHC. Electrical heat generation was modeled with Joule heating as shown in Equation (23), similar to the EHC heat generation model described by [30], where \( P_{elec} \) is the power through the EHC, \( V \) is the voltage across the EHC, \( I \) is the current through the EHC, \( m_{s,EHC} \) is the mass of the EHC substrate, \( c_{p,EHC} \) is the specific heat of the EHC, and \( T_{EHC} \) is the mean temperature of the EHC.

\[
P_{elec} = VI = m_{s,EHC}c_{p,EHC} \frac{dT_{EHC}}{dt}
\]  

(23)

4.4.6 Geometric and Material Parameters

This section specifies the geometric and material properties used to model the Emitec corrugated metal foil EHC/main converter system. A cross-sectional diagram of the EHC/main converter is shown in Figure 21, indicating the diameter and length dimensions specified in Table 3.
The geometry of the Emitec catalytic converter is a corrugated metal foil substrate, as shown in Figure 22.

Table 3: Geometric parameter values for Emitec EHC/main converter

<table>
<thead>
<tr>
<th>Description</th>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length of EHC</td>
<td>$L_{EHC}$</td>
<td>0.010</td>
<td>m</td>
</tr>
<tr>
<td>Length of main converter substrate</td>
<td>$L_{cat}$</td>
<td>0.164</td>
<td>m</td>
</tr>
<tr>
<td>Length of one discrete element</td>
<td>$L_i$</td>
<td>0.0273</td>
<td>m</td>
</tr>
<tr>
<td>Inner diameter of converter can</td>
<td>$D_{cat}$</td>
<td>0.118</td>
<td>m</td>
</tr>
<tr>
<td>Thickness of can</td>
<td>$T_{can}$</td>
<td>0.0015</td>
<td>m</td>
</tr>
<tr>
<td>Thickness of metal foils in substrate</td>
<td>$T_{foil}$</td>
<td>0.00005</td>
<td>m</td>
</tr>
<tr>
<td>Thickness of metal foils in EHC</td>
<td>$T_{foil,EHC}$</td>
<td>0.00004</td>
<td>m</td>
</tr>
<tr>
<td>Cell density of substrate and of EHC</td>
<td>CPSI</td>
<td>400</td>
<td>cpsi</td>
</tr>
<tr>
<td>Number of cells in substrate</td>
<td>$N_{cells}$</td>
<td>6780</td>
<td>cells</td>
</tr>
</tbody>
</table>

Figure 21: Cross-sectional diagram of EHC/main converter
This structure was modeled as a cosine function, as shown in Figure 23, in which one complete period of the cosine function is contained in an $L_{cell} \times L_{cell}$ area.

To determine the value of $L_{cell}$, the converter substrate diameter was used to calculate the circular cross-sectional area of the converter, and this area was multiplied by the substrate
cell density to determine the number of cells in the substrate. The length of one cell was calculated using Equation (24), accounting for the fact that one $L_{cell} \times L_{cell}$ area includes two cell channels with this model.

\[
L_{cell} = \sqrt{A_{cell}} = \sqrt{\frac{2}{CPST}}
\]  

(24)

The area of the foil was calculated as the solid shaded area in Figure 23 by calculating the integral of the upper cosine function minus the integral of the lower cosine and adding the area of the length of one cell multiplied by the foil thickness. This calculation accounts for the foil area attributed to two cell channels; therefore, this calculation was then divided by two to achieve the value of $A_{foil}$ for one cell.

\[
A_{foil} = \frac{1}{2} \left[ \int_0^{L_{cell}} \left( \frac{L_{cell} - Th_{foil}}{2} \cos \left( \frac{2\pi x}{L_{cell}} \right) \right) dx - \int_0^{L_{cell}} \left( \frac{L_{cell} - Th_{foil}}{2} \cos \left( \frac{2\pi x}{L_{cell}} \right) + Th_{foil} \right) dx + L_{cell}Th_{foil} \right]
\]  

(25)

The value of $A_{foil}$ was then multiplied by the total number of cells in the substrate, Equation (26), to determine the cross-sectional area of substrate through which upstream and downstream substrate conduction can occur.
\[ A_s = A_{foil} N_{cells} \]  \hspace{1cm} (26)

The cross-sectional area of one channel, \( A_{\text{channel}} \), is indicated by the hashed lines in Figure 23. This area was calculated using Equation (27).

\[ A_{\text{channel}} = \frac{1}{2} L_{\text{cell}}^2 - A_{foil} \]  \hspace{1cm} (27)

The total volume of gas contained in one discrete control volume is described by Equation (30).

\[ V_g = A_{\text{channel}} L_i N_{\text{cells}} \]  \hspace{1cm} (28)

The geometric surface area, \( A_{gs} \), for one discrete element was calculated to determine the area upon which convective heat transfer from the gas to the substrate occurs. This area is calculated as shown in Figure 24. The length of the gas-substrate interface surface was calculated by approximating the length of the cosine function and adding the length of \( L_{\text{cell}} \). This total length was then multiplied by the length of the discrete element, \( L_i \), and the total number of cells to determine the total surface area, \( A_{gs} \), for one discrete element.
The cross-sectional area of the can, \( A_c \), was simply calculated based on the thickness of the can wall specified by the catalytic converter manufacturer. The area of interface between the can and ambient for one discrete element was calculated based on the circumference of the outer diameter of the can multiplied by \( L_i \).

The EHC substrate was a metal foil geometry similar to the main converter, and therefore, the same area calculation methodology was used. EHC calculations accounted for the smaller foil thickness and shorter axial length, resulting in parameter values for \( A_{gs,EHC} \), \( A_{s,EHC} \), and \( A_{c,amb,EHC} \).

The area parameters and calculated values discussed above are summarized in Table 4.
Table 4: Calculated values of areas for heat transfer

<table>
<thead>
<tr>
<th>Description</th>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cross-sectional area of metal foil substrate for one cell</td>
<td>$A_{foil}$</td>
<td>8.98e-08</td>
<td>m²</td>
</tr>
<tr>
<td>Cross-sectional area of substrate</td>
<td>$A_s$</td>
<td>6.09e-4</td>
<td>m²</td>
</tr>
<tr>
<td>Cross-sectional area of gas flow for one cell channel</td>
<td>$A_{channel}$</td>
<td>1.52e-06</td>
<td>m²</td>
</tr>
<tr>
<td>Volume of gas contained in one discrete control volume</td>
<td>$V_g$</td>
<td>2.82e-4</td>
<td>m³</td>
</tr>
<tr>
<td>Geometric surface area of gas-substrate interface for one discrete element</td>
<td>$A_{gs}$</td>
<td>1.07</td>
<td>m²</td>
</tr>
<tr>
<td>Cross-sectional area of can</td>
<td>$A_c$</td>
<td>5.63e-4</td>
<td>m²</td>
</tr>
<tr>
<td>Geometric surface area of can-ambient interface for one discrete element</td>
<td>$A_{c,amb}$</td>
<td>0.0104</td>
<td>m²</td>
</tr>
<tr>
<td>Geometric surface area of gas-EHC substrate interface for one discrete element</td>
<td>$A_{gs,EHC}$</td>
<td>0.3903</td>
<td>m²</td>
</tr>
<tr>
<td>Cross-sectional area of EHC substrate</td>
<td>$A_{s,EHC}$</td>
<td>4.87e-4</td>
<td>m²</td>
</tr>
<tr>
<td>Cross-sectional area of EHC can</td>
<td>$A_{c,amb,EHC}$</td>
<td>0.0038</td>
<td>m²</td>
</tr>
</tbody>
</table>

The mass of gas within one discrete control volume was calculated using the ideal gas law as shown in Equation (29) and updated dynamically based on gas temperature. Values for $P_g$ and $M_g$ for exhaust gas were specified for equilibrium burned gas at stoichiometry based on Figure 4-14 in [17]. Values for air were defined for ideal gas properties from Table A.5 in [32].
The mass of substrate within one discrete element was calculated with Equation (30), where the density of the substrate, $\rho_s$, was specified for EN grade 1.4509 stainless steel [1].

$$m_s = \rho_s A_s L_i$$  \hspace{1cm} (30)  

The mass of can within one discrete element was calculated with Equation (31), where the density of the can material is again the density of EN grade 1.4509 stainless steel.

$$m_c = \rho_c A_c L_i$$  \hspace{1cm} (31)  

For the EHC, substrate mass was calculated with Equation (32), where $\rho_{EHC}$ is defined for EN grade 1.4509 stainless steel, the same material as the metal substrate, and $A_{s,EHC}$ is the cross-sectional area of the EHC substrate.

$$m_{s,EHC} = \rho_{EHC} A_{s,EHC} L_{EHC}$$  \hspace{1cm} (32)  

The mass of the EHC can was calculated with Equation (32), where $\rho_{EHC}$ is defined for EN grade 1.4509 stainless steel, the same material as the metal substrate, and $A_{c,EHC}$ is the cross-sectional area of the EHC can.

$$m_{c,EHC} = \rho_{EHC} A_{c,EHC} L_{EHC}$$  \hspace{1cm} (33)  

The mass parameters and calculated values are summarized in Table 5.
Table 5: Mass parameter values

<table>
<thead>
<tr>
<th>Description</th>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass of gas within one discrete control volume</td>
<td>$m_{g,i}$</td>
<td>Calculated dynamically</td>
<td>kg</td>
</tr>
<tr>
<td>Pressure at which molar mass and specific heat of gas were defined</td>
<td>$P_g$</td>
<td>Exhaust: 111325 Air: 100000</td>
<td>Pa</td>
</tr>
<tr>
<td>Molar mass of gas</td>
<td>$M_g$</td>
<td>Exhaust: 0.028 Air: 0.029</td>
<td>kg/mol</td>
</tr>
<tr>
<td>Universal gas constant</td>
<td>$R$</td>
<td>8.314</td>
<td>J/mol-K</td>
</tr>
<tr>
<td>Density of substrate</td>
<td>$\rho_s$</td>
<td>7700</td>
<td>kg/m³</td>
</tr>
<tr>
<td>Mass of substrate in one discrete element</td>
<td>$m_s$</td>
<td>0.1281</td>
<td>kg</td>
</tr>
<tr>
<td>Density of can</td>
<td>$\rho_c$</td>
<td>770</td>
<td>kg/m³</td>
</tr>
<tr>
<td>Mass of can in one discrete element</td>
<td>$m_c$</td>
<td>0.1185</td>
<td>kg</td>
</tr>
<tr>
<td>Density of EHC</td>
<td>$\rho_{EHC}$</td>
<td>7700</td>
<td>kg/m³</td>
</tr>
<tr>
<td>Mass of EHC substrate</td>
<td>$m_{s,EHC}$</td>
<td>0.0375</td>
<td>kg</td>
</tr>
</tbody>
</table>

The specific heat, $c_{p,g}$, for exhaust gas was assumed to be 1400 J/kg-K for equilibrium burned gas at stoichiometry based on Figure 4-15 in [17]. For air, this value was specified as 1004 J/kg-K for air based on Table A.5 from [32]. The specific heat capacity for the substrate, can, and EHC were all assumed to be the specific heat capacity for EN grade 1.4509 stainless steel. These specific heat parameters are summarized in Table 6.
Table 6: Specific heat capacity values

<table>
<thead>
<tr>
<th>Description</th>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Specific heat capacity of gas</td>
<td>$c_{p,g}$</td>
<td>Exhaust: 1400</td>
<td>J/kg-K</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Air: 1004</td>
<td></td>
</tr>
<tr>
<td>Specific heat capacity of substrate</td>
<td>$c_{p,s}$</td>
<td>460</td>
<td>J/kg-K</td>
</tr>
<tr>
<td>Specific heat capacity of can</td>
<td>$c_{p,c}$</td>
<td>460</td>
<td>J/kg-K</td>
</tr>
<tr>
<td>Specific heat capacity of EHC</td>
<td>$c_{p,EHC}$</td>
<td>460</td>
<td>J/kg-K</td>
</tr>
</tbody>
</table>

The heat transfer coefficients determined for the thermal model are presented in Table 7, and the transport delay values are shown in Table 8. Further discussion of the calibration method used to determine these values is discussed in Section 4.6.

Table 7: Heat transfer coefficients

<table>
<thead>
<tr>
<th>Description</th>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Conductive heat transfer coefficient of substrate</td>
<td>$k_s$</td>
<td>26.197</td>
<td>W/m-K</td>
</tr>
<tr>
<td>Conductive heat transfer coefficient of can</td>
<td>$k_c$</td>
<td>26.165</td>
<td>W/m-K</td>
</tr>
<tr>
<td>Convective heat transfer coefficient between gas and substrate</td>
<td>$h_{gs}$</td>
<td>166.006</td>
<td>W/m²-K</td>
</tr>
<tr>
<td>Lumped conductive heat transfer coefficient between substrate and can</td>
<td>$K_{sc}$</td>
<td>0.152</td>
<td>W/K</td>
</tr>
<tr>
<td>Convective heat transfer coefficient between can and ambient</td>
<td>$h_{c,amb}$</td>
<td>2.173</td>
<td>W/m²-K</td>
</tr>
</tbody>
</table>
Table 8: Transport delay values

<table>
<thead>
<tr>
<th>Description</th>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Transport delay between elements for gas temperature</td>
<td>$t_{del,g}$</td>
<td>7.252</td>
<td>seconds</td>
</tr>
<tr>
<td>Transport delay between elements for substrate temperature</td>
<td>$t_{del,s}$</td>
<td>0.965</td>
<td>seconds</td>
</tr>
<tr>
<td>Transport delay between elements for can temperature</td>
<td>$t_{del,c}$</td>
<td>0.955</td>
<td>seconds</td>
</tr>
</tbody>
</table>

4.5 Simulation Development

The catalytic converter heat transfer model discussed above was implemented in the Matlab/Simulink simulation environment as a variable step simulation using the ode23tb (stiff/TR-BDF2) solver. A stiff solver was selected due to the different time scales of heat transfer mechanisms within the system. The time constant of the gas temperature is small because the heat is transferred to the gas rapidly during transients, but the time constant of the can temperature is large because the can responds slowly to transients in heat transfer. These different time scales make the system of differential equations stiff.

The simulator was developed in two stages. The first stage was to create and calibrate the simulation of the main converter with six discrete converter elements. Each discrete element contained states for gas, substrate, and can temperatures. The main converter simulation required input vectors of pre-catalyst gas temperature, exhaust mass flow rate, and ambient temperature, and specified constant initial conditions for the substrate and can temperatures. The simulator outputs included the calculated gas temperature within
each discrete element, the calculated substrate temperature within each discrete element, and the calculated can temperature within each discrete element. This main converter simulation incorporated a transport delay block to account for the delay in temperature rise between elements caused by the physical distance between center points in each element.

The second stage of simulation development was to add the electrically-heated catalyst to the simulator. The EHC block consumes an input voltage provided to the EHC and calculates the heat generated by resistive heating in the EHC. This heat generation is incorporated into the substrate heat transfer block. The simulator calculates the transfer of heat from the EHC substrate to the gas passing through the channels and to the can wall and ambient, and then passes the resulting gas and can temperatures to the first element of the main converter. Since there is a physical gap between the EHC substrate and the main converter substrate, no connection was made between these signals. Images of the main converter and EHC Simulink diagrams are shown in Figure 25 and Figure 26.

![Main Converter Model](image)

Figure 25: Main Converter Model
4.6 Simulation Calibration

The main converter simulation was calibrated based on experimental data collected from an uncoated Emitec metal foil substrate catalytic converter without EHC. This converter was instrumented with 1/32 inch diameter, type K thermocouples positioned at the locations specified in Figure 27. Six internal thermocouples ($T_1 - T_6$) were positioned along the length of the substrate at a radial distance 42mm from the center axis of the converter. One thermocouple was positioned along the center axis of the converter and one thermocouple was brazed onto the exterior surface of the can wall; both of these thermocouples were placed in the plane of the $T_3$ thermocouple. Pre-catalyst and post-catalyst temperatures were measured with 1/8 inch diameter, type K thermocouples.
Figure 27: Thermocouple locations in metal foil substrate

Figure 28: TWC instrumented with thermocouples
Data for calibration was collected by installing the instrumented catalytic converter on the OSU EcoCAR 1.8L E85 engine in the engine dynamometer test cell environment. Engine data was collected at constant speed and load starting at steady state operation at maximum brake torque (MBT) and then retarding spark timing 10 degrees to generate a step increase in exhaust gas temperature. Tests were conducted at 1500 RPM and 3000 RPM for 40 kPa, 60 kPa, and 80 kPa manifold air pressure conditions.

A system identification approach was used to identify the parameters for calibration. The variables affecting the heat transfer between the gas, substrate, and can temperature states are indicated in Figure 29. Length, area, and specific heat capacity parameters are known based on the system material and geometry, and therefore, these parameters were considered fixed. These known parameters are shown in black text. The heat transfer coefficients and transport delays \((h_{gs}, h_{c,amb}, K_{sc}, k_s, k_c, t_{del,g}, t_{del,s}, t_{del,c})\) are unknown parameters that needed to be determined empirically and therefore required calibration. These calibrated parameters are shown in red text. Since the mass flow rate of gas was calculated dynamically in the simulation, it is not considered to be a fixed or calibrated parameter.
Values for the calibrated parameters were determined by fixing the known parameters and then implementing a calibration methodology consisting of two phases. The first phase was to select initial values for the calibration parameters using physics-based estimates and perform a preliminary optimization by hand. The initial value for $h_{gs}$ was selected as 166 W/m$^2$-K based on trends in heat transfer coefficients of different sized EHCs created by the manufacturer, Emitec, as reported in [6]. Initial values for $k_s$ and $k_c$ were selected to be 25 W/m-K based on material properties of EN grade 1.4509 stainless steel [1]. Initial values for $K_{sc}$, $h_{c,amb}$, and the transport delays for the gas, substrate, and can were selected by running the simulation model with various initial guesses and
comparing simulated internal substrate and can temperatures with measured results for two sequential exhaust gas temperature step increases.

The internal thermocouples were located at specific spatial locations in the catalyst substrate. Although the experimental data collected by the six internal thermocouples showed the expected trends in warm-up behavior, these measurements indicated the temperature of one point within the substrate, whereas the temperature states in the model described the mean value temperature over the entire control volume. As a result, simulated and measured temperature responses were similar, but not the same. One method for more closely aligning measured and simulated data would be to include more thermocouples at different radial and axial locations within each control volume to determine a more accurate average temperature; however, this would also increase data acquisition demands, cost, instrumentation complexity, and post-processing effort. Given this measurement limitation, correlation between measured and simulated data was assessed by comparing the final substrate temperatures ($T_6$) and the can temperatures.

Engine data collected at 1500 RPM and 40 kPa was selected for calibration because exhaust flow rate at this condition was closest to the air flow rates of the secondary air injection system implemented in the EcoCAR vehicle. Two data sets were collected on different days and were used for calibration. Table 9 presents the values selected after this initial calibration.
Table 9: Parameter values selected for initial calibration (Phase 1)

<table>
<thead>
<tr>
<th>$h_{gs}$ W/m²-K</th>
<th>$h_{camb}$ W/m²-K</th>
<th>$K_{sc}$ W/K</th>
<th>$k_s$ W/m-K</th>
<th>$k_c$ W/m-K</th>
<th>$t_{del,g}$ sim steps</th>
<th>$t_{del,s}$ sim steps</th>
<th>$t_{del,c}$ sim steps</th>
</tr>
</thead>
<tbody>
<tr>
<td>166</td>
<td>2.2</td>
<td>0.16</td>
<td>25</td>
<td>25</td>
<td>6</td>
<td>0.1</td>
<td>0.1</td>
</tr>
</tbody>
</table>

Figure 30 shows simulation results using these parameter values. The measured pre-catalyst temperature was fed into the model as the input gas temperature and resulting simulated substrate temperature states are compared with measured internal substrate temperatures $T_1 - T_6$, as well as simulated and measured can temperature. Simulated results are shown in red and measured results are shown in blue. As shown, each section of substrate increases in temperature sequentially starting with the first element, then the second element, and so forth. These trends are shown in the model, along with good correlation between the simulated and measured can temperatures.
To further refine the calibration, the second phase of calibration was to optimize the calibration parameters using the Matlab Optimization Toolbox. Although the heat transfer equations modeling the system are linear, non-linear system optimization techniques were used due to the transport delays in the model as well as to allow parameters to vary as a function of other inputs or states. The objective function was defined as the root mean square (RMS) error between the simulated and measured values of substrate temperature $T_s,6$ plus the RMS error between the simulated and measured values of the can temperature, as shown in Equation (34).

Figure 30: Simulation results for initial calibration (Phase 1) compared with measured results for step increase in inlet gas temperature at 1500 RPM and 40 kPa
\[ F_{obj} = \alpha \sqrt{\frac{\sum_{j=1}^{n}(T_{s,6,sim} - T_{s,6,meas})^2}{n}} + \beta \sqrt{\frac{\sum_{j=1}^{n}(T_{c,3,sim} - T_{can,meas})^2}{n}} \]  

(34)

The index, \( j \), indicates each simulation step, and \( n \) represents the number of simulation steps. The weighting factors \( \alpha \) and \( \beta \) were defined as 0.90 and 0.10, respectively, to show preference to matching the sixth substrate temperature more heavily than matching the can temperatures. This objective function was evaluated over the period of step increase in temperature, but not the initial settling periods.

The \textit{fmincon} constrained non-linear minimization solver was selected to perform a gradient-based optimization on the objective function. Iterations of the optimization are provided in Figure 31 showing an improvement over the initial values.

![Figure 31: Iterative objective function values over fmincon optimization](image-url)
Resulting parameter values after the optimization are given in Table 10 and resulting simulation results are presented in Figure 32.

Table 10: Parameter values selected for initial calibration (Phase 2)

<table>
<thead>
<tr>
<th>$h_{gs}$ W/m²-K</th>
<th>$h_{camb}$ W/m²-K</th>
<th>$K_{sc}$ W/K</th>
<th>$k_s$ W/m-K</th>
<th>$k_c$ W/m-K</th>
<th>$t_{del,g}$ sim steps</th>
<th>$t_{del,s}$ sim steps</th>
<th>$t_{del,c}$ sim steps</th>
</tr>
</thead>
<tbody>
<tr>
<td>166.006</td>
<td>2.173</td>
<td>0.152</td>
<td>26.197</td>
<td>26.165</td>
<td>7.252</td>
<td>0.965</td>
<td>0.955</td>
</tr>
</tbody>
</table>

Figure 32: Simulation results for optimized calibration (Phase 2) compared with measured results for step increase in inlet gas temperature at 1500 RPM and 40 kPa.

Figure 33 shows a closer view of the second step increase in temperature. As shown, the simulated results show similar rise time and reach approximately the same steady state value after the temperature increase. Although only the sixth temperature state was
incorporated into the objective function for optimization, results show good correlation for all internal temperature states.

![Graph showing measured and simulated substrate temperatures over time.](image)

**Figure 33:** Closer view of simulation results for optimized calibration (Phase 2)

Once the heat transfer and transport delay values were determined, the main converter model was integrated with the EHC model. Discussion of model results and validation is provided in the next section.

### 4.7 Experimental Validation

Validation of the EHC thermal model was done using data collected on the lab bench setup with the secondary air injection system similar to the one incorporated into the EcoCAR vehicle. This lab bench setup is described in the schematic in Figure 34. Electrical power to the air pump was provided by a variable power supply operated at a
range of 0-12 V. Airflow was measured through the orifice plate and differential pressure setup identical to that used in the vehicle application. The valve in the bench setup was a passive check valve. The exhaust manifold port of the catalytic converter was capped with a closed plate. The catalytic converter was an Emitec metal foil catalyst with EHC. The main converter had the same cell density and foil thickness as the uncoated catalyst that was used for heat transfer coefficient calibration. Power to the EHC was provided by a second power supply capable of sourcing 165 A at 12 V. Current to the EHC was measured with a 1 mV/A current clamp. Data acquisition was done with a 16 input, 16-bit National Instruments NI USB-6211 data logger interfaced to a laptop with a LabView .vi used to record air pump voltage, differential pressure sensor voltage, RTD temperature measurements, and EHC voltage and current.

Figure 34: Lab bench setup for EHC data collection
The catalytic converter was instrumented with three RTD temperature sensors in pre-catalyst, mid-catalyst, and post-catalyst locations to match the sensor setup that was designed for the vehicle. However, initial testing indicated a lag in transient response of the RTD sensors. To illustrate the sensor lag associated with the RTD sensor, a comparison of the step response of an RTD and 1/32 inch diameter, type K thermocouple to a increase in temperature from 25°C to 190°C is shown in Figure 35. This step response was generated by soaking the sensors at room temperature and then inserting them into a thermocouple calibration setpoint heater. The thermocouple responded and reached steady-state at 190°C nearly instantaneously while the RTD took nearly three times as long to reach steady-state.

Figure 35: RTD temperature sensor shows lag in step response as compared with a 1/32 inch diameter, type K thermocouple
The lag in RTD response is attributed to three main factors. First, the structure of the RTD included an outer metal sheath covering an insulating layer that surrounded the resistive device in the center, as shown in the diagram in Figure 36. The insulating layer provided resistance to heat transfer, which would add delay between the surface temperature of the sheath compared to the temperature sensed at the center of the RTD. Second, the RTD had larger diameter and greater mass than the thermocouple and therefore also had greater thermal inertia to resist changes in temperature. Finally, as illustrated in Figure 36, the RTD was placed into the catalytic converter through a fitting at the surface of the can. Since temperatures at the can were consistently lower than temperatures within the catalytic converter substrate, there would also be some heat transfer from the tip of the RTD along its length out to the fitting at the can surface.

Figure 36: Diagram of RTD structure and placement in catalytic converter substrate
In light of these sensing issues associated with RTD-type sensors, the EHC on the lab bench was also instrumented with a 1/32 inch diameter, type K thermocouple located in the same cross-sectional plane as the mid-catalyst RTD (see Figure 34). The locations of the mid-catalyst RTD and mid-catalyst thermocouple were approximately the same axial length along the catalytic converter as the $T_2$ thermocouple shown in Figure 27.

Inputs to the EHC model were the experimentally-measured EHC current and voltage as well as the voltage measured by the differential pressure sensor, which was then converted to a mass air flow rate through a lookup table containing the sensor calibration. Results of the EHC simulation were compared with measured mid-catalyst temperature measurements. Figure 37 shows results for 2000 W heating power and 4.6 g/s airflow. As shown, the simulated substrate temperature in the second discrete element ($T_{s,2}$) heated slightly faster than the measured thermocouple temperature but both reached the same peak temperature and reached this peak temperature at the same time. The response of the RTD was slower and was consistently lower temperature than the simulated and thermocouple values.
Figure 37: Simulation results for EHC model compared with measured results for 2000W EHC heating power (100% duty cycle) and air flow rate of 4.6 g/s

The 2000 W, 4.6 g/s test was defined as the nominal case and was compared with data collected at different airflow inputs. Figure 38 and Figure 39 compare the system response to 2000 W power input to the EHC for 5.2 g/s and 6.0 g/s airflow, respectively. Both cases show similar trends in data, indicating agreement between the simulated $T_{s,2}$ temperature and the thermocouple measurement but a delay and lower peak temperature measured by the RTD.
Figure 38: Simulation results for EHC model compared with measured results for 2000W EHC heating power (100% duty cycle) and air flow rate of 5.2 g/s

Figure 39: Simulation results for EHC model compared with measured results for 2000W EHC heating power (100% duty cycle) and air flow rate of 6.0 g/s
The 2000 W, 4.6 g/s test in Figure 37 was also compared with data collected at different power inputs. Figure 40 and Figure 41 compare the system response at 4.6 g/s airflow for 1770 W and 1420 W heating power, respectively. Again, both cases show similar trends in data, indicating agreement between the simulated $T_{s,2}$ temperature and the thermocouple measurement but a delay and lower peak temperature measured by the RTD.

Figure 40: Simulation results for EHC model compared with measured results for 1770W EHC heating power (92% duty cycle) and air flow rate of 4.6 g/s
Figure 41: Simulation results for EHC model compared with measured results for 1420W EHC heating power (75% duty cycle) and air flow rate of 4.6 g/s

These results showed validation of the EHC/main converter thermal model using thermocouple measurements and also provided better understanding of the EHC emissions system implemented in the Ohio State EcoCAR vehicle.
5.1 Introduction

The physics-based catalytic converter model discussed in Chapter 4 was further extended to a software in the loop (SIL) simulation to develop and investigate an advanced control strategy for the electrically-heated catalyst system designed for the Ohio State EcoCAR vehicle. The desire for an advanced control strategy had multiple driving factors:

- Safety precautions to protect components
- System constraints and consumer acceptability
- Sensing issues due to slow plant response
- Changes in plant due to temperature

To protect the EHC system hardware, safety precautions needed to be incorporated to ensure that the system operated within the limits specified by the manufacturer. These factors included EHC maximum operating temperature and maximum power, as well as sufficient airflow during EHC heating.

System constraints were also a factor. Given the high current demands of the EHC, the power that could be provided to the EHC was dictated by the capabilities of the 12V
battery and the DC-DC converter. Operating the EHC at full power resulted in a high current draw from the EHC which caused noticeable dimming of the lights on the dashboard display as well as the headlights. In addition to reducing performance of the vehicle electrical system, these undesirable effects would directly impact consumer acceptability. Negative contribution to consumer acceptability would lose points in the EcoCAR competition and also reduce the overall quality of the vehicle.

The EHC and three-way catalytic converter system in the vehicle was instrumented with the pre-catalyst, mid-catalyst, and post-catalyst RTDs only. Given the lag associated with these sensors as well as the slow time constant of the plant, changes to the commanded EHC power would not result in instantaneous change in temperature measured by the mid-catalyst RTD. Furthermore, direct measurement of the EHC substrate temperature was not achievable with the limited sensors in the system. Therefore, there was motivation to estimate the EHC temperature based on measurable temperature and voltage signals.

In addition, the resistance of the EHC varied with changes in temperature, adding non-linearity in the plant. As a means of optimizing EHC performance to achieve catalyst light-off quickly while operating within the limitations of the EHC and vehicle electrical system, an advanced control strategy was developed and implemented in SIL.
5.2 Software in the Loop Investigation of Advanced EHC Control

The advanced EHC control was developed as a negative feedback control, as shown in Figure 42. An overview of the control is discussed here, with more detail of each component provided in the following sections.

![Figure 42: Overview of advanced EHC control](image)

Given the slow time constant of the real system, changes to the commanded power would not result in an instantaneous change in temperature. To account for this, a feedforward and simple plant were used to estimate the system delay. The feedforward term predicted the desired EHC heating power based on a constant airflow and mid-catalyst temperature sensor measurement. The feedforward predicted power demand was then used by a simplified plant model to predict the change in mid-catalyst temperature expected if the amount of feedforward power were to be supplied to the EHC. This predicted mid-catalyst temperature was compared to the actual mid-catalyst temperature read by the sensor to produce an error term. The error was used in a simple proportional-integral feedback controller to adjust the EHC power request. This raw power request was then checked to ensure the power demand did not exceed the capabilities of the vehicle or the
safe operation of the EHC system hardware, and then this saturated power demand was converted into a duty cycle (DC) command. Finally, a state controller was used to compare the mid-catalyst temperature sensor to the desired temperature for light-off and assess when heating was needed.

5.2.1 EHC Feedforward

The EHC feedforward term was implemented using Equation (35) to predict the power needed to raise the estimated EHC temperature, $T_{EHC,est}$, calculated in the simple plant model block to the maximum EHC temperature, $T_{EHC,max}$. The maximum EHC temperature specified by the manufacturer was 1000°C. A safety factor of 100°C was applied, so $T_{EHC,max}$ was defined as 900°C. The specific heat, $c_{p,s}$ was defined for the substrate. This calculation neglects heat loss to the surroundings.

$$P_{ff} = m_{s,EHC}c_{p,s}(T_{EHC,max} - T_{EHC,est})$$ (35)

5.2.2 EHC Simple Plant Model

A control-oriented, simple plant model was developed by reducing the physics-based catalytic converter model developed in Chapter 4. Due to the high rate of heat transfer between the gas and substrate within the catalytic converter, it is reasonable to assume that the temperature of the gas is equal to the temperature of the substrate within one discrete element and eliminate the term for convective heat transfer between the gas and substrate from the model. It further follows that the heat lost from the substrate to can and from the can to ambient can be lumped into a single term to describe the heat lost from
the substrate to ambient based on a new, lumped heat transfer coefficient, $K_{s, amb}$. Since the EHC model assessed light-off by comparing the mid-catalyst RTD sensor measurement with the simulated substrate temperature in the first discrete element ($T_{s,1}$), only the EHC element and first discrete element from the main converter model were necessary for the control-oriented plant model. Based on these simplifying assumptions, the physics-based catalyst model was reduced to Equations (13) and (14) and implemented into the advanced EHC control.

$$m_{s,EHC} c_p, s \frac{dT_{s,EHC}}{dt} = P_{ff} + \dot{m}_g c_{p,g} (T_{g,in} - T_{s,EHC}) - K_{s,amb} (T_{s,EHC} - T_{amb})$$

$$m_{s,1} c_p, s \frac{dT_{s,1}}{dt} = \dot{m}_g c_{p,g} (T_{s,EHC} - T_{s,1}) - K_{s,amb} (T_{s,1} - T_{amb})$$

5.2.3 EHC Feedback

The mid-catalyst temperature, $T_{s,1}$, predicted by the simplified plant model was compared to the measured mid-catalyst temperature sensor signal, and the error between these two temperatures was used for proportional-integral (PI) feedback control. The resulting output is a raw power request.

5.2.4 Duty Cycle Calculation

To convert the raw power request into a duty cycle command, the raw power request signal was limited through a saturation block in which the maximum power limit was set
to the manufacturer-specified limit of the EHC (3000 W). The saturated power request was then converted into a duty cycle command using Equations (38) to calculate the ratio between the saturated power request and the achievable power from the system based on the voltage of the 12V battery and the estimated resistance of the EHC and cabling.

\[
DC = \frac{P_{\text{max,req}}}{\left(\frac{V_{\text{batt}}^2}{R_{\text{est}}}\right)} \cdot 100
\]  

(38)

The resistance of the EHC was calculated with Equation (39) by using a calibrated linear fit to account for the temperature dependence of the resistance across the EHC, plus a static calculation of the resistance of the cabling based on the resistivity of copper, \(\rho\), and the length and cross-sectional area of the cables, \(L\) and \(A\) respectively.

\[
R_{\text{est}} = R_{\text{EHC}}(T_{\text{midcat,meas}}) + \left(\rho \frac{L}{A}\right)_{\text{cables}}
\]

(39)

Finally, this resulting duty cycle request was managed by a simple Stateflow decision based on mid-catalyst temperature to determine whether the control should be on or off. The purpose of including the Stateflow was to simultaneously turn the EHC and air pump on or off and to allow for re-heating if the catalyst temperature dropped 20 degrees below the light-off temperature. The output of the Stateflow was a 1 or 0 multiplier on the EHC and air pump commands. The final output commands were then sent to the combined EHC and Main Converter full plant model used to represent the real system, as described in more detail in the next section.
5.3 SIL Implementation

The advanced EHC control was then implemented into a software in the loop simulation using the physics-based catalytic converter model as the plant to represent the real system. Figure 43 shows the highest level of the SIL implementation.

![Figure 43: Top-level view of software in the loop layout](image)

The SIL simulation required initial conditions to be specified for pre-catalyst temperature, engine inlet air temperature ($IAT$), commanded secondary air injection airflow, catalyst light-off temperature, and nominal voltage of the vehicle low voltage system. To represent the noise that would be seen on sensor measurements in a real system, ±2% noise was factored into the mid-catalyst temperature calculated in the EHC and Main Converter Plant and then returned as feedback to the EHC Control.

5.4 SIL Results

The SIL simulation was run with initial conditions selected close to values that would be seen in the vehicle. Table 11 shows the selected initial conditions.
Table 11: Initial conditions for SIL simulation

<table>
<thead>
<tr>
<th>Condition</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_{precat}$</td>
<td>43 °C</td>
</tr>
<tr>
<td>$IAT$</td>
<td>43 °C</td>
</tr>
<tr>
<td>$T_{lightoff}$</td>
<td>275 °C</td>
</tr>
<tr>
<td>$\dot{m}$</td>
<td>6 g/s</td>
</tr>
<tr>
<td>$V_{batt}$</td>
<td>12.8 V</td>
</tr>
</tbody>
</table>

Results of the SIL simulation presented in Figure 44 showed that the control selected a constant 100% EHC duty cycle until the mid-catalyst temperature reached light-off. In other words, SIL simulation indicated that the EHC should be operated at full power for the Ohio State EcoCAR application.

![Duty Cycle vs Time](image1)

![Airflow vs Time](image2)

![Temperature vs Time](image3)

Figure 44: SIL results for light-off at 275°C (2000 W, 6 g/s)

To see why the control selects 100% duty cycle, Figure 45 compares the feedforward power term with the power that could be achieved by the vehicle system, $\frac{V_{batt}^2}{R_{est}}$. As shown, the EHC is power-limited by the voltage of the vehicle 12V system. Although the
Ohio State EcoCAR PHEV has a 360V, 600A on-board battery system, the power to the EHC is limited by the current and voltage limitations of the 12V electrical system and the 2.2kW DC-DC converter.

These results led to further investigation of the design space of the OSU EcoCAR emissions system. A map of the design space was laid out based on the control inputs to the system, which were airflow rate and EHC power. The map is shown in Figure 46. Within this map, multiple system constraints were identified. The minimum and maximum air pump flow rates were determined experimentally to be 2 g/s and 18 g/s, respectively. The manufacturer of the EHC specified a maximum power limit of 3000 W. However, as shown in SIL simulation and experimentally, the vehicle was capable of sourcing only 2000 W to the EHC. The manufacturer also specified a maximum temperature of 1000°C before danger of melting the EHC. The maximum temperature was used to calculate a corresponding amount of power required to raise the pre-catalyst
temperature to the maximum EHC temperature, $T_{EHC, \text{max}}$, over the airflow domain. This calculation is shown in Equation (40).

$$P = \dot{m} c_p, g (T_{EHC, \text{max}} - T_{\text{precat}})$$  \hspace{1cm} (40)

A safety factor of 100°C was applied to the maximum EHC temperature and this temperature was designated as $T_{EHC, \text{max}, SF}$. Equation (40) was applied using $T_{EHC, \text{max}, SF}$ in place of $T_{EHC, \text{max}}$ and the resulting power curve was also included in Figure 46.

After designating all of these system constraints, power curves were calculated to determine the power required for a given airflow rate to achieve a rise in temperature, $dT$. This calculation is described by Equation (41), where $dT$ is the temperature of air exiting the EHC minus the temperature of air entering the EHC. Power curves were calculated for values of $dT$ ranging from 100°C to 300°C and plotted on Figure 46.

$$P = \dot{m} c_p, g dT$$  \hspace{1cm} (41)

For the Ohio State EcoCAR vehicle, an increase of $dT = 300°C$ was desirable to ensure that the catalytic converter reached light-off before engine start. Given this request and the system constraints, the operating region for the Ohio State EcoCAR emissions system was reduced to the shaded area indicated in Figure 46.
It was hypothesized that operating the emissions system at maximum power with maximum airflow within this operating region, would achieve the fastest heating time to reach light-off. This hypothesis was tested in simulation using the EHC/Main Converter model. Ten power/airflow combinations along the edges of the operating region were selected. These points are indicated graphically in Figure 47 and specified numerically in Table 12. For all simulations, light-off temperature was specified as 275°C and the system voltage was set to 12.8V.
Figure 47: Simulated time and energy consumption to reach light-off temperature at specified airflow and power operating point.

Table 12: Time and energy to reach light-off for different EHC power and airflow combinations

<table>
<thead>
<tr>
<th>Test</th>
<th>Power (W)</th>
<th>EHC Current (A)</th>
<th>Air Flow (g/s)</th>
<th>Time to Light-Off (s)</th>
<th>Energy Consumption (W-h)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2000</td>
<td>154</td>
<td>2.3</td>
<td>27.0</td>
<td>15.0</td>
</tr>
<tr>
<td>2</td>
<td>2000</td>
<td>143</td>
<td>3</td>
<td>25.6</td>
<td>14.2</td>
</tr>
<tr>
<td>3</td>
<td>2000</td>
<td>154</td>
<td>4.3</td>
<td>24.6</td>
<td>13.7</td>
</tr>
<tr>
<td>4</td>
<td>2000</td>
<td>154</td>
<td>5</td>
<td>24.6</td>
<td>13.7</td>
</tr>
<tr>
<td>5</td>
<td>2000</td>
<td>154</td>
<td>6.5</td>
<td>27.7</td>
<td>15.4</td>
</tr>
<tr>
<td>6</td>
<td>1770</td>
<td>136</td>
<td>2.0</td>
<td>29.7</td>
<td>14.6</td>
</tr>
<tr>
<td>7</td>
<td>1250</td>
<td>96</td>
<td>2.0</td>
<td>37.3</td>
<td>12.9</td>
</tr>
<tr>
<td>8</td>
<td>1250</td>
<td>96</td>
<td>3.0</td>
<td>37.6</td>
<td>13.0</td>
</tr>
<tr>
<td>9</td>
<td>1250</td>
<td>96</td>
<td>4.3</td>
<td>47.8</td>
<td>16.6</td>
</tr>
<tr>
<td>10</td>
<td>600</td>
<td>46</td>
<td>2.0</td>
<td>120.0</td>
<td>20.0</td>
</tr>
</tbody>
</table>
Results from this exploration showed that the fastest light-off times occurred with 2000W power and airflow between 4 and 5 g/s. In other words, light-off would be achieved most quickly by using the maximum available power with airflow that is neither the maximum or minimum of the operating region. Figure 48 further investigates the effect of airflow.

![Diagram comparing effects of air flow rate on temperature gradient along length of catalytic converter](image)

Figure 48: Diagram comparing effects of air flow rate on temperature gradient along length of catalytic converter

At low airflows, heating remains mostly localized in the front of the catalytic converter and results in a temperature gradient along the catalytic converter. At high airflows, temperatures are more similar in magnitude along the length of the catalytic converter and less temperature gradient is observed. From simulation and experimental testing, it was found that high airflow limited the temperature increase that could be achieved at a given power. An example of this is illustrated in Figure 49, showing that the substrate temperature in the first discrete element of the main converter reaches a steady-state temperature of 189°C.
These findings contributed to the selection of an on/off control strategy for the Ohio State EcoCAR vehicle with a commanded airflow rate of 5 g/s.

5.5 Further Considerations for Implementing Advanced EHC Control in Vehicle

The advanced EHC control strategy discussed in this section was created in a continuous time simulation environment. If the EHC were not power-limited by the capabilities of the vehicle electrical system, it could be desirable to implement an advanced strategy such as this into a vehicle application. If the advanced EHC control strategy were to be implemented on real controller hardware, the control software would need to be modified to run in discrete time. The main components of the model that this would affect would be the integrators and transport delays used in the model. Integrators would need to be
changed from continuous time $\frac{1}{s}$ integrators to discrete time $\frac{1}{z}$ integrators. Special consideration would need to be taken for the transport delays, because use of transport delays can require significant random access memory (RAM) space on the controller. This would particularly be an issue for the Ohio State EcoCAR vehicle application because the sophistication of the software on the ECU has left little space for additional demands.

In addition to these software and controller considerations, use of the advanced EHC control would require a driver to modulate power to the EHC. The driver circuitry would be required to handle high currents (~150A) and sufficiently dissipate heat from the driver. Development of a reliable driver circuitry presented three main challenges. First, it was difficult to find a driver in a small form-factor that could handle the high current application. Second, interfacing the driver with the ECU through transistor logic to invert and amplify the signal from the ECU was found to affect the integrity of the square wave signal. This affected the performance of the driver because it was not cleanly switching on and off. This exacerbated the third challenge, which was cooling the driver properly. Due to the high current and resulting heat generation at the driver, forced convection would be necessary. In light of the SIL results indicating that on/off control was appropriate for the Ohio State EcoCAR application, combined with the added vehicle weight, packaging, and complexity of using a high current driver for modulating EHC control, the solution was to use the ECU to control a contactor to open or close the 12V circuit to the EHC.
CHAPTER 6
ELECTRICALLY-HEATED CATALYST VALIDATION (EXPERIMENTAL)

6.1 EHC Emissions Results In-Vehicle

Vehicle emissions were tested at the U.S. Environmental Protection Agency National Vehicle and Fuel Emissions Laboratory (EPA NVFEL) in Ann Arbor, Michigan. The Ohio State University EcoCAR vehicle was set up on a four-wheel drive chassis dynamometer and instrumented for emissions sampling with a portable emissions measurement (PEM) system as well as a bag and bench system. Through this setup, both modal and bagged emissions data were recorded. The vehicle was subjected to an overnight ambient soak period and then driven over a Federal Test Procedure (FTP) drive cycle without EHC heating to collect benchmark cold-start emissions data. The vehicle was again subjected to an overnight ambient soak period and then a second FTP cycle was conducted with the EHC heating strategy enabled.

Modal data for FTP cold start without and with EHC heating is compared in Figure 50 and Figure 51. In the figures, t=0 designates engine start and the time axis of the plot begins at the key-on event (t=-50 for Figure 50 and t=-107 for Figure 51). During both drive cycles, at the key-on event the vehicle started in all-electric mode and the supervisory controller requested engine warm-up, thereby initiating the warm-up phase.
For the first FTP test session, the warm-up phase consisted of warming up the oxygen sensors for 50 seconds before allowing engine start. For the second FTP test session, the warm-up phase included simultaneous oxygen sensor warm-up and EHC heating with air pump flow for 107 seconds before allowing engine start.

Figure 50: FTP cold start emissions without EHC heating
As shown by the measured vehicle speed in the first subplot of each figure, driving began 30 seconds after key-on, which was at least 50 seconds before the engine started. The vehicle was able to follow the prescribed FTP drive cycle before engine start because the hybrid powertrain design allowed the vehicle to operate electrically during this period. As shown in the cumulative emissions graph in the third subplot of each figure, no emissions were produced during this all-electric period with a clear increase in emissions upon engine start. Emissions without EHC heating were markedly higher without EHC heating than with EHC heating, which is a result of the increased catalyst temperature shown in the second subplot of each figure. This trend is shown more closely in Figure 52.
For a conventional engine start without EHC heating (left subplots) the entire catalyst remained at ambient temperature (about 30°C) until engine start. Conversely, with EHC pre-heating, the mid-catalyst temperature was already at 350°C by the time the engine started. As shown in the lower subplots, having the front of the catalytic converter heated above light-off before starting the engine significantly reduced THC and NO\textsubscript{x} emissions. These results for the first bag of the FTP cycle tested at EPA NVFEL are shown in tabulated form in Table 15 and as a bar chart in Figure 52 to compare the effect of EHC heating. The data shows that EHC pre-heating reduced Bag 1 cold start THC and NO\textsubscript{x} emissions by 80-90\%.
Table 13: Tabulated FTP Bag 1 (cold start) emissions without and with pre-heating

<table>
<thead>
<tr>
<th></th>
<th>HC-FID (g/mi)</th>
<th>CO (g/mi)</th>
<th>NOx (g/mi)</th>
<th>NMOG (g/mi)</th>
</tr>
</thead>
<tbody>
<tr>
<td>FTP Bag 1 without EHC</td>
<td>0.537</td>
<td>2.055</td>
<td>0.149</td>
<td>0.394</td>
</tr>
<tr>
<td>FTP Bag 1 with EHC</td>
<td>0.123</td>
<td>2.304</td>
<td>0.046</td>
<td>0.111</td>
</tr>
</tbody>
</table>

Figure 53: Bar chart of FTP Bag 1 (cold start) emissions without and with pre-heating

After testing at EPA NVFEL, vehicle emissions were also tested at the Transportation Research Center (TRC) in East Liberty, Ohio. This testing was conducted on a four-wheel chassis dynamometer and bagged emissions data was collected for an FTP drive cycle. Table 14 describes the modifications made to hardware and software between testing at EPA NVFEL and subsequent testing at TRC.
Table 14: Description of vehicle hardware and software differences between emissions testing at EPA NVFEL versus TRC

<table>
<thead>
<tr>
<th>Testing at EPA NVFEL</th>
<th>Testing at TRC</th>
</tr>
</thead>
<tbody>
<tr>
<td>• Passive check valve in emissions system</td>
<td>• Replaced with electrically-actuated valve</td>
</tr>
<tr>
<td>• Lead acid 12V battery</td>
<td>• Lithium-ion 12V battery</td>
</tr>
<tr>
<td>• No cold start fuel enrichment</td>
<td>• No cold start fuel enrichment</td>
</tr>
<tr>
<td>• Initial pre-catalyst and post-catalyst O\textsubscript{2} controller calibration.</td>
<td>• Refined pre-catalyst and post-catalyst O\textsubscript{2} controller calibration.</td>
</tr>
</tbody>
</table>

Table 15 compares the emissions results measured at EPA NVFEL and TRC to the standard for Tier II Bin 5. Emissions were measured over the entire FTP cycle and the results were weighted according to Tier II regulation with weighting factors being 0.43 for the cold start phase (Bag 1), 1.0 for the transient phase (Bag 2), and 0.57 for the hot start phase (Bag 3) [7]. Results are presented as a bar chart in Figure 54.

<table>
<thead>
<tr>
<th></th>
<th>HC-FID (g/mi)</th>
<th>CO (g/mi)</th>
<th>NO\textsubscript{x} (g/mi)</th>
<th>NMOG (g/mi)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tier 2, Bin 5 Standards</td>
<td>---</td>
<td>3.40</td>
<td>0.050</td>
<td>0.075</td>
</tr>
<tr>
<td>Results from EPA NVFEL without EHC</td>
<td>0.503</td>
<td>0.108</td>
<td>0.138</td>
<td>0.501</td>
</tr>
<tr>
<td>Results from EPA NVFEL with EHC</td>
<td>0.028</td>
<td>0.93</td>
<td>0.056</td>
<td>0.094</td>
</tr>
<tr>
<td>Results from TRC with EHC</td>
<td>---</td>
<td>0.49</td>
<td>0.011</td>
<td>0.069</td>
</tr>
</tbody>
</table>
Comparison of weighted FTP test data measured at EPA NVFEL with and without EHC pre-heating showed significant improvement in weighted FTP emissions:

- 14% reduction in CO
- 60% reduction in NO\textsubscript{x}
- 81% reduction in THC

The EPA NVFEL data was used to assess the energy trade-off associated with heating the EHC. Equations (42) was used to calculate the energy consumption, \( EC \), of the Ohio State EcoCAR vehicle. The numerator of this calculation determines the usable energy from the high voltage battery pack based on the energy capacity of the battery pack, \( E_{batt} \), and assuming that 80% of the battery capacity can be charged and discharged.
safely without damage to the battery pack. This calculation also accounts for the efficiency of the DC-DC converter, $\eta_{DCDC}$. Values for these parameters are defined in Table 16.

$$EC\left[\frac{kWh}{miles}\right] = \frac{0.8 E_{batt} \eta_{DCDC}[kWh]}{AER[miles]} \quad (42)$$

Table 16: Definition of parameters used to calculate the energy trade-off for EHC heating

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$E_{batt}$</td>
<td>Energy storage capacity of on-board battery pack</td>
<td>21 kWh</td>
</tr>
<tr>
<td>$\eta_{DCDC}$</td>
<td>Efficiency of DC-DC converter</td>
<td>97%</td>
</tr>
<tr>
<td>$AER$</td>
<td>All-electric range of vehicle</td>
<td>40 miles</td>
</tr>
</tbody>
</table>

The electrical consumption of the vehicle was then used to determine the amount that the all-electric range of the vehicle would be reduced by allocating energy to EHC pre-heating, $AER_{offsetEHC}$. This calculation is shown in Equation (43).

$$AER_{offsetEHC} = \frac{E_{EHC}[kWh]}{EC\left[\frac{kWh}{miles}\right]} \quad (43)$$

Energy consumed by the EHC for pre-heating, $E_{EHC}$, was calculated by integrating the power consumed by the EHC during that period. These calculations showed that 0.05 kWh of energy were consumed by EHC pre-heating. The energy from the battery pack used to source power to the EHC amounted to 0.3% of the total on-board energy storage. As a result, using the EHC pre-heat strategy translated to a 0.1 mile reduction in all-electric range of the vehicle.
7.1 Conclusions

A physics-based approach was used to model the thermal behavior of a catalytic converter as a one-dimensional, lumped parameter model. This model of the main converter was calibrated with experimental engine data to determine empirical heat transfer coefficients. Simulated results demonstrated good correlation with measured data. An electrically-heated catalyst element was then incorporated into the main converter model, and this model was validated by data collected on a lab bench setup.

To account for sensing issues caused by the long time constant of the system as well as for the purpose of component safety, the development of an advanced EHC control was investigated through the use of Software in the Loop techniques. The one-dimensional, lumped parameter EHC model was incorporated as the plant and a simplified form of the model was used as a feedforward term in the controller. Results indicated that a simplified on/off control would be sufficient for the Ohio State EcoCAR PHEV application but that more complex model-based control would be useful in applications able to provide greater heating power to the EHC system. Use of the physics-based model
and the Software in the Loop simulation provided better understanding of the emissions system in the Ohio State EcoCAR PHEV and aided in refining the system calibration.

Vehicle implementation of the EHC emissions system using a pre-heat control strategy showed 80-90% reduction in FTP Bag 1 cold start emissions measured at the U.S. Environmental Protection Agency National Vehicle and Fuel Emissions Laboratory. Use of the EHC pre-heat feature in conjunction with refined air/fuel ratio control significantly contributed to achieving Tier II Bin 5 emissions as measured at the Transportation Research Center.

7.2 Future Work

Use of a physics-based modeling approach enables the catalytic converter thermal model to be extended to other catalytic converter systems with relatively low calibration effort.

The model developed in this thesis will be incorporated into a research project that will develop a thermal model of a vehicle exhaust system. In this project, the main converter model will be used to simulate catalytic converter temperature transients during convention warm-up with engine-out exhaust gases. For this model, it would be appropriate to incorporate heat generation due to exothermic catalytic reactions on the substrate surface once each discrete substrate element reaches light-off.

Further extensions of the catalytic converter thermal model could include incorporating a map or model of catalytic conversion efficiencies into the discrete elements of the thermal model to show trends in catalytic performance as the catalytic converter warms
up during cold start. Another extension of the model would be to expand it to two-dimensions by including radial heat transfer in addition to the axial heat transfer that was modeled in this research.
BIBLIOGRAPHY


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