A Robust Control Approach on Diesel Engines with Dual-Loop Exhaust Gas Recirculation Systems

THESIS

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ABSTRACT

This thesis presents a control design using dual-loop exhaust gas recirculation (EGR) and variable geometry turbo charging (VGT), installed on a medium-duty, V8 diesel engine, to adjust intake temperature, pressure, and oxygen mass fraction. The dual-loop EGR system is compiled of both high pressure EGR (HPEGR) and low pressure EGR (LPEGR) paths. Turbocharging is achieved through the use of a two-stage system consisting of a fixed geometry, low pressure turbo and a variable geometry, high pressure turbo. This extensive network creates a complex air-path that necessitates the implementation of an advanced feedback control method. Groundwork for the study is a high fidelity GT-Power computational model of the diesel engine equipped with the proposed dual-loop EGR air-path, capable of simulating the one dimensional gas dynamics of the engine. The computational model allows for an identification of system dynamics that, when validated, provides a basis for the development of a multi-input multi-output (MIMO) air-path controller. Attention must be paid to the robust performance of the controller as computational system identification is inherently inaccurate, due to system nonlinearities, variable transport delay, and other unforeseen dynamics not accounted for in simulation.

The focus of the controller is to use the dual-loop EGR system in conjunction with the VGT to establish a high control authority over intake manifold temperature,

pressure, and oxygen mass fraction. Each of these conditions is highly influential on such low emission combustion modes like low temperature combustion, homogeneous charge compression ignition, and pre-mixed charge compression ignition. These combustion modes have high sensitivity to engine intake conditions and high tendency of knock and misfire, which warrant a comparison of the advanced, multivariable feedback control strategy to conventional feedback control using the complex air-path. Strong benefits to using multivariable control are seen through faster response and settling times along with better disturbance rejection capabilities when maintaining desired intake conditions. A feed-forward controller for the complex air-path is also developed and explored for additional improvements in performance when coupled with the multivariable feedback controller. Minimal benefits to the coupling were seen but these could be improved upon through more accurate knowledge of system parameters.

Chapter 1 explores the benefits when operating in advanced combustion modes and potentials for expanding their operating range with the use of the proposed complex air-path system. A linear state-space representation of the air-path system is then identified in Chapter 2 using the GT-Power model which serves as the basis for developing a MIMO feedback controller. The performance of the MIMO feedback controller, and its coupling with a feed-forward controller, are then developed and validated through the GT-Power engine simulation in Chapter 3. Finally, in Chapter 4, a decentralized feedback controller is made which relies on much more basic principles of multivariable control and is compared to the MIMO feedback controller to examine the benefits of using advanced controller development strategies. Concluding remarks and future work are then given in Chapter 5.

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CHAPTER 1: INTRODUCTION

1.1 The Need for Emissions Reduction

Output emissions are the key factor that withholds diesel engines from implementation into meeting the majority of automotive transportation needs in the U.S. This is rooted in the high levels of hazardous contaminants such as nitrous oxides (NO_x) and particulate matter (PM) that are contained within the exhaust gas of diesel engines. Currently the methods used to counteract these dangerous contaminants are aftertreatment systems where growing number of these aftertreatment systems are going to be implemented with the continuing limitations on tailpipe emissions placed by the EPA. Figure 1 shows the EPA regulations that have been placed on heavy-duty diesel engines since 1994. Additionally the reduced emissions standards of the European Union for light-duty diesel vehicles can be seen in Figure 2. The immediate solution is installation of stronger and more complex diesel aftertreatment systems but unfortunately this includes added expenses and may interfere with engine operation. Better methods to controlling engine-out emissions of diesel combustion then become more feasible to offset these costs. This leads to the focus of this work which is to explore how using a complex air-path system can allow for stricter control of intake conditions which can be directly related to methods that dramatically reduce diesel emissions.

Figure 1. EPA heavy-duty diesel emission regulations by year [45]

Figure 2. European emission standards for light-duty diesel vehicles by year [44]

1.2 Diesel Emissions Production

Diesel emissions form as a consequence of using a heterogeneous mixture of air and fuel for combustion. Conventional diesel engines operate on the principle of compression ignition where fuel is injected directly into the combustion chamber and then auto ignites at high pressures. The direct injection of fuel is what is responsible for the heterogeneously mixed air and fuel which forms local areas of both lean and rich mixtures. A mixture is said to be lean when there is not enough fuel present to combust with all the available air. A mixture is rich when there is not enough air to combust with all the available fuel. Therefore an air to fuel ratio is above stoichiometric conditions when the mixture is lean and below stoichiometric conditions when the mixture is rich. When the total mixture is combusted the flame propagates from the point of auto ignition throughout the rest of the combustion chamber. The variance in the air to fuel ratio in the combustion chamber causes a drastic variance in the combustion temperature throughout the cylinder which leads to the formation of unwanted emissions.

Nitrous oxide emissions formation is the result of nitrogen reacting with the oxygen in the combustion chamber. These emissions will form as nitrogen monoxide, NO, or nitrogen dioxide, NO₂ and are normally categorized as NO_x. Formation is fastest at the peak temperatures that exist in the initial stages of the combustion process near 2200 K [2], when pressure is highest and the piston is near the top of the combustion chamber. The NO_x will primarily form in local areas that are near or slightly below stoichiometric conditions, which allows for higher temperatures. While the cylinder temperature remains high enough the NO_x that has formed will begin to disassociate back

into nitrogen and oxygen. This only occurs at a noticeable rate until the predominant part of the expansion stroke is reached and the mean temperature of the combustion chamber drops with pressure. When this happens the disassociation will cease and concentrations of NO_x freeze at their present value. Hastening the cooling of the NO_x molecules is the mixing between the high temperature zones with the rest of combustion chamber. This mixing is only inherent to direct injected diesel engines, also called compression ignition direct injection (CIDI), and is the predominant reason that they produce higher levels of NO_x than their gasoline counterparts [12].

Particulate matter in diesel engines consists mainly of combustion generated carbonaceous material or soot. Formation primarily occurs in the locally rich regions that combust at oxygen starved conditions. In these rich zones the hydrocarbons in the fuel will only be able to partially combust with oxygen. As mixing continues some of the partially combusted hydrocarbons will continue to oxidize as others that are exposed to cooler regions, between 1500 and 2300 K, will condensate into larger graphite like structures or soot [12][10]. A considerable amount of the PM will form near the cooler regions of the cylinder wall and is why injection strategies tend to avoid spraying fuel in close proximity to the cylinder wall. Modern strategies also spray fuel at high pressures to reduce the droplet size giving a reduction in the number of locally rich zones, reducing PM formation [5][10]. Noticeable amounts of PM also form as the heavier lubricating oil that coats the cylinder wall burns during combustion. The formation of PM also places a strong limit on the maximum amount of fuel that can be injected with each engine cycle. It becomes more difficult to reduce the number of locally rich zones in the heterogeneous mixture when more fuel is added. This is the reason that diesel engines will run globally

lean in the sense that the total amount of fresh air and fuel introduced into the engine is above the stoichiometric air to fuel ratio. A summary of the formation between NO_x and PM can be seen in Figure 3 as compared to the local equivalence ratio and burn temperature. The equivalence ratio is the ratio of the actual fuel to air ratio over the stoichiometric fuel to air ratio.

Figure 3. NO_x and PM formation map with respect to equivalence ratio and temperature [1].

1.3 Diesel Operation

Diesel engines also have numerous advantages especially when compared to conventional gasoline engines. The auto ignition of fuel requires a much higher compression ratio than gasoline engines which directly increases the efficiency of the

thermodynamic cycle of reciprocating engines. Spark ignited gasoline engines operate under the thermodynamic principles of the Otto cycle where there is almost a constant volume expansion when the heat energy is released from the fuel. This causes the entirety of the power stroke of gasoline engines to be delivered by an isentropic expansion of the combusted gases. Diesel engines however operate under the diesel cycle which better resembles a constant pressure expansion when the fuel energy is released after auto ignition. This constant pressure expansion contributes to the power stroke of the engine along with the isotropic expansion of combusted gasses, increasing the gross energy released. For these reasons diesel engines produce more torque and better efficiency when compared to gasoline engines of equal displacement.

The use of direct injection can give some flexibility in regards to engine operation and emissions output. The auto ignition of diesel fuel is highly dependent on temperature and pressure [12] which continually increase during the compression stroke. This allows a control strategy to be developed which adjusts injection timing to a corresponding crank angle to produce the best possible performance and emissions characteristics. Newly implemented piezo-driven actuators in modern diesel fuel injectors also offer some improvement on emissions control. The piezo actuator is much more responsive than previously used solenoid actuators allowing multi-stage injections into the combustion chamber. This opens the potential for pilot injections to be introduced into the combustion chamber before the combustion begins to take place. This gives more of the injected fuel adequate time to vaporize leading to a less heterogeneous mixture and reduced particulate matter. Additionally piezo injectors offer a greater performance with variations in injection pressure and spray angle. This allows for added injection

parameters to be made to allow for optimal fuel vaporization to reduce emissions output [4].

1.4 Exhaust Aftertreatment

The aftertreatment method mainly used today to combat particulate emissions is a combination of the diesel oxidation catalyst (DOC) and the diesel particulate filter (DPF). The DOC is usually located just upstream of the DPF where imbedded catalysts are able to eliminate up to 30% of the particulate matter. The DOC is also able to oxidize a portion of the NO into NO₂ which is presumed to be a good oxidation agent for the soot downstream in the DPF [19]. The DPF is contained by a substrate of parallel channels, made from ceramic monolith, that are plugged at alternating ends. The substrate geometry forces the exhaust gas to flow through the monolith wall which filters the particulate. This filtering action is usually effective in reducing PM emissions on the order of 90% although this in turn causes an increase in exhaust backpressure, decreasing fuel efficiency. As the filter will become loaded with PM, backpressure will continue to rise which gives the need to regenerate the DPF through oxidation of the trapped PM.

The key element in DPF regeneration is being able to increase the substrate temperature to allow for the oxidation of soot which is normally around 800 K [12][19]. There are many ways this can be done but regeneration strategies can be categorized as either active or passive. Passive regeneration is when the DPF substrate is able to reach regeneration temperatures under normal driving operation. This can occur when more fuel is injected under high load operation or when DPF loading causes backpressure to reach high enough levels where the additional fuel is required. Active regeneration usually takes place with the addition of excess fuel into the exhaust gas to initiate the

regeneration process, either through in-cylinder post-injection or direct injection into the exhaust gas. The use of fuel to burn trapped PM in the DPF is undesired but can be used to balance the losses experienced by the increased backpressure of a loaded DPF. Obviously to avoid frequent active regeneration minimal engine-out PM levels must be attained.

The most effective method for NO_x after treatment is with the use of selective catalyst reduction (SCR). Urea SCR works with a small injection of ammonia reductant, known as urea, into the exhaust gas which reacts with NO_x on the catalyst to form nitrogen and water vapor. This method proves highly effective and can reduce NO_x by up to 95% when operating at optimal conditions [18]. Potential obstacles exist with the conversion efficiency being highly dependent on the full vaporization of the urea or ammonia leaking into the atmosphere. Also the use of SCR creates a dependency on maintaining urea levels which in turn increases costs for the end user. Lean NO_x traps (LNT) have also shown promising effects on reducing NO_x emissions by storing NO_x through the process of adsorption, where hazardous molecules adhere to platinum on the LNT substrate. Once the LNT storage levels reach full capacity they need to be regenerated in rich exhaust conditions where prominent carbon monoxide (CO) emissions are able to react with the NO_x to form nitrogen and carbon dioxide, relieving the LNT of stored NO_x . Conventional methods of regeneration include those similar to DPF regeneration where additional fuel is allowed to enter the exhaust gas although this is normally needed at a higher frequency than DPFs. NO_x reduction efficiency can be up to 100% in LNTs but this figure is highly dependent on the level of expensive platinum loading on the substrate.

Although methods of aftertreatment for diesel engines are highly effective they come at a high price. The regeneration process needed for DPFs and LNTs is an inefficient use of fuel which sacrifices overall engine operation. Installation of these aftertreatment systems also creates a higher capital investment for end users. LNT efficiency is highly dependent on platinum loading and SCR systems require additional hardware for controlling urea injection and ensuring proper vaporization. Reducing NO_x and PM engine output would not only reduce the capital investment needed for aftertreatment but would also lower the needed regeneration frequency and save fuel.

1.5 Motivation

To utilize the benefits of compression ignition engines it is imperative that new methods of emissions reduction need to go into effect. This can potentially be done with the use of advanced combustion modes that prevent emissions formation inside the combustion chamber. Some examples of these are low temperature combustion (LTC), pre-mixed charge compression ignition (PCCI), and homogeneous charge compression ignition (HCCI) whose operating ranges can be seen in Figure 4. These modes of combustion pose huge benefits in emissions reduction but controlling their stability along desired loads and speeds becomes difficult. The following sections will highlight both the benefits and shortcoming of these methods of combustion along with the effectiveness of how manipulating intake conditions can overcome these shortcomings and extending the combustion stability range.

Figure 4. Combustion mode regimes reflected on NO_x and PM formation map [1].

1.2.1 Exhaust Gas Recirculation

It is not possible to discuss advanced modes of combustion without introducing EGR or exhaust gas recirculation. This is where exhaust gas is recirculated back into the intake manifold to again cycle through the combustion chamber giving the ability to control the in-cylinder oxygen mass fraction. Because diesel engines typically run lean, the fuel when combusted completely will have only reacted with a partial amount of the air within the combustion chamber and is why the exhaust gas is composed of both burned and unburned gases. The burned gas consists of the products of combustion, carbon dioxide and water vapor along with comparatively low concentrations of unwanted emissions. The unburned gas is composed of the remaining fresh air. The

reasoning behind using EGR is that the burned gases are able to reduce the peak temperatures within the combustion chamber. This happens because the burned gases that are recirculated back inside the engine have a higher thermal capacity than fresh air. During combustion, the burned gases are able to reduce the maximum temperature by absorbing more of the heat that is released by combustion. EGR will also delay the auto ignition of the fuel and slow down the combustion process by reducing the oxygen mass fraction, as the burned gases are inert to combustion. This gives more time for the locally rich and lean zones of diesel engines to mix and for fuel to evaporate, reducing emissions. The addition of EGR has negligible impact on the performance of diesel engines as the fuel added will still combust completely. The power output of gasoline engines, which operate at stoichiometric conditions, is negatively impacted by EGR as it directly reduces their volumetric efficiency, or from this perspective, the total amount of oxygen that can be combusted with fuel. This is why diesel engines will generally use higher amounts of EGR than gasoline engines.

1.5.2 Low Temperature Combustion

Low Temperature Combustion works by using extensive amounts of EGR to reduce emissions. EGR is able to reduce emissions both because of its ability to reduce peak temperatures, and by extending the mixing and evaporation time of injected fuel. A reduction in peak temperature regions, normally above 2200 K, directly reduces NO_x formation to near negligible levels. Particulate matter will normally increase with EGR until it reaches the high rates of LTC, on the order of 55%, where there is a sharp decrease in PM formation [2]. This happens because the high level of EGR causes the

predominant number of local temperature regions to cross the prime area for PM formation, 1500 to 2300 K [10] reflected in Figure 3.

Once EGR levels reach such high rates some negative impacts can be seen. Engine emission levels of hydrocarbons (HC), or unburned fuel, will rise along with CO. These two emissions are normally only considered in gasoline engines that burn close to stoichiometric conditions and form when globally rich conditions are reached and combustion is incomplete. The existence of CO and HC in the emissions output gives indication that the combustion process was left uncompleted. Their formation is possible with large amounts of EGR in diesels as there is a reduction in peak temperatures and oxygen mass fraction, leading to a higher number of local areas above stoichiometric conditions. Under conventional diesel operation with high combustion temperatures diesels will burn 99.5% of the fuel injected. When operating in LTC, incomplete combustion can reduce the burn fraction to 95% [23]. This causes an undesired reduction in fuel efficiency as combustion is not completed and fuel is left unburned. Additional reductions in efficiency come from the delay in auto ignition and peak pressures caused by the high levels of EGR. The incomplete combustion of fuel along with the CO and HC production is also what provides the upper load limit to LTC. The lower boundary for LTC is designated by the low combustion temperatures which can cause misfire without significant quantities of fuel for flame propagation [10].

Strong possibilities for reducing the negative impacts of LTC can be seen by controlling both the intake temperature and pressure. By increasing the intake pressure the volumetric efficiency of the engine will rise as a higher charge of air and EGR composition can be inducted into the cylinder. This higher density charge contains more

oxygen that allows for a more complete combustion to occur resulting in reduced HC and PM. For increased intake pressures from naturally aspirated conditions to 1 bar of boost, HC can be reduced by approximately 50% and PM can be reduced by approximately 75%. Increased intake pressure also showed to extend the operating range of LTC into higher loads [10]. The effectiveness of LTC to reduce NO_x and PM can be extended by reducing intake temperatures. Research has shown that a decreased intake temperature of only 10 °C was able to reduce LTC NO_x and PM emissions by 24 and 50% [7]. On the other hand, when considering the lower load limit for LTC, an increased intake temperature would also have the potential to increase peak combustion temperatures to reduce the possibility for misfire. This shows how sensitive LTC can be and gives the need for a strict control strategy to optimized intake temperature, pressure, and EGR rates. CO and HC emissions that are common to LTC also pose little threat to the tailpipe emissions as catalysts that are consistently used in gasoline engines are able to reduce CO and HC to negligible levels.

1.5.3 Homogeneous Charge Compression Ignition

Another mode of advanced combustion that has promising effects on reducing emissions is HCCI. The operation of this combustion mode is possible through similar injection methods used in gasoline engines, where fuel is usually injected inside the intake port. This allows for a homogeneously distributed mixture of fuel to be inducted into the combustion chamber before compression. The only difference between the injection strategies for gasoline and HCCI operation is that the mixture will be predominantly lean, at air fuel ratios similar to conventional diesel operation, at low and medium loads. After compression, the homogenous charge will auto ignite and the fuel will spontaneously combust in a very short duration. This fast combustion keeps the

cylinder temperatures very low in comparison to conventional CIDI where the flame propagates through the combustion chamber during the longer combustion process. These cool combustion temperatures is what gives HCCI operation such low emissions output. The greatest emission reduction of HCCI is seen in NO_x with a 98% decrease over CIDI while PM is reduced by 27% [9][37]. HCCI also offers the efficiencies of CIDI but has the potential to be implemented into gasoline engines where it could save the U.S. as much as half a million barrels of oil per day [38].

The trouble with HCCI is that combustion is purely controlled by chemical kinetics whereas with conventional diesel operation combustion is controlled by the mechanism of direct fuel injection. From a controls perspective this is a big disadvantage since no fuel injection strategy can be implemented to directly control ignition timing or combustion duration. Optimal HCCI auto ignition is normally desired to occur when the piston reaches top dead center (TDC). If ignition occurs too early pressures will reach dangerous levels and the engine will knock. If ignition occurs too late the engine will begin to misfire. This lack of ignition control is what limits HCCI operation to low and medium loads and speeds and what makes maintaining combustion stability challenging. Also as the combustion process takes place at much cooler temperatures it is difficult for the fuel to be fully oxidized. This leads to considerable levels of CO and HC emissions.

EGR proves to be the best method in controlling the kinetics of HCCI combustion. The addition of EGR reduces the oxygen mass fraction, delaying auto ignition, and reducing the rate of combustion. This gives the ability to extend the HCCI operating range into higher loads although increased EGR will give rise to a slight increase in CO and HC emissions [30]. Research has also shown that increasing the

intake pressure will advance the auto ignition of HCCI and can extend the operating range towards both the higher and lower load regimes. It has also been shown that increasing intake pressure will reduce HC emissions in HCCI for the same reason that it does in LTC, the higher mass density allows for more oxygen within the cylinder to complete the combustion process. Unfortunately increased pressure causes CO levels to slightly rise as the higher mass density causes cooler combustion chamber temperatures that slow down the oxidation of CO into carbon dioxide [46]. Increasing the intake temperature during HCCI operation will speed up the chemical kinetics for the auto ignition of fuel and increase the rate that in cylinder pressures rise. This can actually restrict operation at higher loads but is able to extend the abilities of HCCI to operate at higher speeds and lower loads [25]. Higher intake temperatures will also allow for a slight increase in combustion temperatures. This gives for a more complete oxidation of the fuel and reductions in CO and HC formation with only slight gains in NO_x formation [31][26]. Again, CO and HC engine-out emissions do not pose a great threat to tailpipe emissions as catalysts used in gasoline engines are able to reduce CO and HC to negligible levels.

1.5.4 Premixed Charge Compression Ignition

The concept behind PCCI is the early injection of a portion of the fuel into the combustion chamber during the compression stroke with a pilot injection followed by a main injection of fuel to control combustion. The early pilot injection gives a portion of the fuel more time to evaporate and increase the global homogeneity of the mixture when combustion ultimately takes place, guided by the main fuel injection. Studies have shown to allow for the main fuel injection to guide combustion timing there needs to be a

strict control of intake temperature to prevent the auto ignition of the pilot injection [22]. Also by increasing the separation between the pilot and main injections it is possible to achieve a partial HCCI combustion that can be extended into conventional diesel combustion [33]. What makes PCCI attainable in modern diesel engines is the use of piezo injectors where their fast actuation allows for multiple injections per cycle, per cylinder. The variations in PCCI combustion can be as vast as the injection strategies imaginable with multiple injections per cycle and lie anywhere on the spectrum between almost complete HCCI combustion to conventional CIDI. Because of the vast amount of injection strategies specific examples of emissions reduction benefits will not be discussed although one can imagine they will be on the same spectrum between HCCI and CIDI. To improve the emissions reduction of PCCI it is important to extend the mixing time of the pilot injection to allow for more fuel to evaporate and increase homogeneity by retarding the ignition time. This can be done by reducing both intake pressure and temperature of the composition while increasing the EGR of the intake [33][22], showing how intake conditions play an important role in PCCI effectiveness.

1.5.5 Avenues of Advanced Combustion Control

The key to controlling advanced modes of combustion is to achieve desired in-cylinder conditions to control both ignition timing and combustion duration to maintain a stable combustion. This becomes highly challenging at high engine loads where the EGR rate is limited by the larger amount of fuel introduced into the cylinder. To be able to utilize advanced combustion modes and still produce engines that maintain drivability, methods of controlling combustion mode switching can be introduced to be able to achieve stable engine operation at high loads. This strategy revolves on being able to

operate in any of the discussed combustion modes at low to medium loads and switch to conventional CIDI when needed. Minimized operation in CIDI allows for downsizing of expensive diesel aftertreatment systems and lowering regeneration frequency while still maintaining overall emissions reduction with advanced combustion modes. The goal of any strategy for combustion mode switching is to quickly cycle between combustion modes while exhibiting minimal effects on drivability. Conventional control methodologies that use static mapping can pose problems to combustion switching. Misfires, knocking, along with increased NO_x and HC emissions are seen when switching combustion modes without incorporating engine dynamics [3][20]. Smooth combustion mode switching is possible but it is necessary to use more advanced, dynamic controllers in conjunction with additional hardware to achieve a stricter authority over in-cylinder conditions. Examples of successful combustion mode switching can be seen in [39][41]. Research has even suggested using a cycling of combustion modes of LTC and CIDI for LNT regeneration is possible [42][18].

Applications with the right hardware that can perform to meet the proper in-cylinder conditions have the potential to extend the benefits of advanced combustion modes into a larger operating range and further reduce emissions. This is most challenging for HCCI combustion which lacks a direct fuel injection mechanism to control ignition timing and is why HCCI control methods use some of the most notable strategies. Auto ignition of HCCI is purely controlled by chemical kinetics and relies heavily on the in-cylinder oxygen mass fraction and temperature history. Temperature history is mostly affected by two parameters, the initial temperature of inducted charge at intake valve closing (IVC), and the effective compression ratio (ECR), or pressure

change incurred by the inducted gases. Many studies have been able to control the ECR through means variable valve timing (VVT) which allows for the intake and exhaust valves to open and close at the varying crank angles corresponding to different piston locations. An example of this has shown that altering the intake valve closing time changes the initial volume of the isentropic compression of inducted gases, which adjusts in-cylinder pressure and temperature at TDC [35]. This method was successfully used to control combustion timing in an HCCI based simulation. The more direct method of altering the ECR, by adjusting the compression ratio itself, has shown to be even more effective in controlling HCCI combustion timing [11]. VVT can also be used to directly control in-cylinder oxygen mass fraction. This can be done by either an advanced or delayed exhaust valve closing (EVC) to control the amount of internal EGR or exhaust gas that remains in the combustion chamber for the following engine cycle. Advanced EVC will trap exhaust gases in the combustion chamber before the intake stroke begins while delayed EVC extends the overlap between exhaust and intake valves which actually causes exhaust gas to flow from the exhaust manifold into the intake port. This approach to controlling combustion stability with internal EGR has shown successful in [35] and [47]. Both VVT and variable compression offer some of the fastest response times in regard to timing control of HCCI because of the direct access to manipulate in-cylinder oxygen mass fraction and temperature history, and in-cylinder pressure measurements. The simple framework also allows for conventional, decoupled feedback to be used in regards to the multi-input multi-output (MIMO) controllers. Control benefits with direct cylinder access can also be extended to PCCI by maximizing pilot injection quantity and vaporization time. The biggest difficulty with implementing VVT

and variable compression are the high costs of hardware needed. Besides the added expense of these actuators, control systems with direct access to the cylinder require the use of expensive in-cylinder pressure transducers to provide feedback signals for peak pressures and burn rates. In-cylinder pressure transducers also require the use of fast computational data acquisitions systems not standard in most vehicles [40]. The economic hurdles that accompany VVT and variable compression control of advanced combustion modes may be too high for implementation within the near future.

1.5.6 Thesis Objective

The scope of this thesis is on an alternative approach to combustion control which is the use of a complex air-path to adjust desired intake manifold temperature, pressure, and oxygen mass fraction at IVC. Not only could this hasten the implementation of advanced combustion modes but it also stretches the possibilities for combustion mode switching. Air-path control also allows for already widely implemented hardware to be used such as turbocharging, and external EGR. Sensors required only consist of standard pressure transducers, temperature sensors, and UEGO sensors already used in practice. The difficulties with using air-path control lie in the fact that manipulation of intake conditions is not quick enough to where desired cycle by cycle trajectories can be met. An additional obstacle is that the independent control of all three intake conditions is not possible with standard turbochargers and single EGR actuation. This calls for additional complexities to be added to the standard air-path to allow for better manipulation of intake conditions. Examples include multi-stage turbocharging, variable geometry turbocharging (VGT), and dual-loop EGR. Details of how this complex hardware works to control intake conditions lie in the following chapter. In parallel, these complexities

accompany additional non-linearity and cross-coupling between actuators that give the need for advanced MIMO control strategies to be developed.

The following chapters will cover a unique setup of such a complex air-path system and demonstrate the potentials that can be reached when advanced control techniques are applied to its operation. This is done through the use of a GT-Power model of a medium-duty, V8, diesel engine that provides computational modeling of one-dimensional dynamics to better simulate actual engine operation. A linear state-space representation is developed using the GT-Power model which serves as the basis for developing a MIMO feedback controller. The performance of the MIMO feedback controller is then validated through the GT-Power engine simulation and benefits in coupling the MIMO feedback controller with a feed-forward controller are then examined. Finally a decentralized feedback controller is made which relies on much more basic principles of multivariable controller development. It is then compared to the MIMO controller to examine the benefits of advanced controller development strategies.

CHAPTER 2: THE COMPLEX AIR-PATH SYSTEM

2.1 Air-path Description

Figure 5. Complex air-path used for system identification and closed-loop control of intake conditions.

The engine set-up for this work uses a medium-duty, V-8, diesel engine equipped with a two-stage, VGT, and a dual-loop EGR system as shown in Figure 5. This complex air-path system gives the versatility to control intake temperature, pressure, and oxygen mass fraction independently. The two-stage turbo charging system consists of both a low

pressure turbocharger (LPT), which operates as the first compression stage, and a high pressure turbocharger (HPT), that operates as the second compression stage. The high pressure turbo is equipped with variable geometry turbine vanes to control the pressure ratio between the exhaust manifold, section 4, and the volume between the low pressure and high pressure turbines, section 5. The dual loop EGR system is composed of both a high pressure EGR (HPEGR), and low pressure EGR (LPEGR) loop with independent flow control from both HPEGR and LPEGR valves.

2.2 Two-Stage Variable Geometry Turbocharging

The VGT is used to assist in controlling intake manifold pressure, which is done through the use of vanes that are located on the inlet of the high pressure turbine. VGT controls intake pressure by manipulating the vane position to adjust the pressure ratio across the turbine allowing for a given amount of boost. The relationship between the turbine pressure ratio and boost pressure can be derived using the laws for conservation of energy and mass when applied to the turbocharger as seen in Equations (1), (2), and (3).

$$\frac{d}{dt}(N_h^2) = \frac{2}{J_h}(p_{ht} - p_{hc}) \quad (1)$$

$$p_{ht} = W_{45}c_{px}T_4\eta_{ht}\left(1 - \left(\frac{P_4}{P_5}\right)^{\frac{1-\gamma}{\gamma}}\right) \quad (2)$$

$$p_{hc} = W_{21}c_{pi}T_2\frac{1}{\eta_{hc}}\left(\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} - 1\right) \quad (3)$$

Here, J and N represent the impeller inertia and speed, η is the isentropic efficiency, and p represents the power produced across the compressor or turbine of the HPT, denoted by the subscripts ht or hc . T and P are the mean temperature and pressure of their respective section, denoted by the numerical subscripts, W is the mass flow rate between sections, γ is the ratio of specific heats, and c_{pi} and c_{px} are specific enthalpies of the intake and exhaust gas. When a lower intake pressure is desired the vanes open to allow a reduction of the pressure ratio between sections 4 and 5 that reduces the pressure ratio across sections 1 and 2 once the system reaches steady-state. When a higher intake pressure is desired the vanes of the VGT will close to create the opposite effect. A normalized VGT position will be used when discussing the vane position where a vane position of 1 and 0 imply fully open and closed vanes. It is important to note the additional non-linearity that exists in Equations (2) and (3) where the turbocharger efficiencies are also functions of the impeller speed and pressure ratios. Similar equations can be derived for the LPT where

$$\frac{d}{dt}(N_l^2) = \frac{2}{J_l}(p_{lt} - p_{lc}) \quad (4)$$

$$p_{lt} = W_{56}c_{px}T_5\eta_{lt}\left(1 - \left(\frac{P_5}{P_6}\right)^{\frac{1-\gamma}{\gamma}}\right) \quad (5)$$

$$p_{lc} = W_{32}c_{pi}T_3\frac{1}{\eta_{lc}}\left(\left(\frac{P_2}{P_3}\right)^{\frac{\gamma-1}{\gamma}} - 1\right). \quad (6)$$

The two-stage arrangement of the turbochargers poses other advantages in regards to overall turbocharger performance. A key objective in turbocharging is to achieve high

boost pressures by recovering all possible energy from the exhaust gas. In single-stage turbocharging this often includes increasing impeller size to increase the pressure ratio across the turbine and compressor. This leads to a sacrifice in turbocharger dynamic performance, or increased turbo lag, and can be seen in either Equations (1) or (4) where the response of turbo speed is inversely proportional to the impeller inertia. The two-stage system allows for the same high boost pressures to be reached with little change to turbo lag by sequentially compressing incoming air. The sequential compression allows for a smaller pressure ratio across each turbo for reduced impeller inertias, maintaining the dynamic performance when considering the coupled system of Equations (1) through (6). Additionally, a higher overall efficiency can be achieved over single stage turbocharging, where higher compression ratios across turbines and compressors normally accompany a decrease in efficiency [13].

2.3 Dual-Loop Exhaust Gas Recirculation

In a MIMO system with the objective being to regulate three output variables, intake temperature, pressure, and oxygen mass fraction, it is necessary for three actuators to be available otherwise the system would not be controllable. The LPEGR loop added to the medium-duty diesel engine, seen in Figure 5, allows for this controllability. By intercepting the exhaust downstream of the DOC and DPF, where exhaust temperature and pressure are low, the added LPEGR has minimal effects on the temperature and pressure of incoming fresh air. In turn, the LPEGR will have also have minimal effects on turbocharger dynamics. Added to the LPEGR loop is a cooler which further reduces the effect LPEGR has on intake temperature. It is necessary for the LPEGR loop to extract gas downstream of the DOC and DPF to prevent PM from entering through the

turbochargers which can lead to compressor degradation. An exhaust valve is placed downstream of where the LPEGR intercepts the exhaust system to maintain enough upstream LPEGR loop pressure to allow for adequate flow. Similar LPEGR exhaust valve setups have been used to control LPEGR flow in HCCI engines in [48] with the use of observers similar to those formed for air-path control in [32]. For this work the exhaust valve opening is left constant to focus on controlling intake conditions with the use of dual-loop EGR and VGT.

2.4 Mean Value Approximations

Although a mean value model of the air-path in Figure 5 was not created to model engine behavior, the fundamental equations can be used to illustrate the effectiveness and coupled behavior seen with this MIMO system in regards to intake condition dynamics. With the LPEGR loop, a mean value approximation using the conservation of energy and mass, along with the ideal gas law, can be used to derive the dynamic behavior for intake temperature and pressure, and for the oxygen mass fraction of each section of the intake breathing system as shown in Equations (7) through (11).

$$\dot{T}_1 = \frac{RT_1}{P_1 V_1} [W_{21}(\gamma T_{IC} - T_1) + W_{41}(\gamma T_{HPEGR} - T_1) - W_e T_1(\gamma - 1)] \quad (7)$$

$$\dot{P}_1 = \frac{R\gamma}{V_1} [W_{21}T_{IC} + W_{41}T_{HPEGR} - W_e T_1] \quad (8)$$

$$\dot{F}_1 = \frac{RT_1}{P_1 V_1} [W_{21}(F_2 - F_1) + W_{41}(F_4 - F_1)] \quad (9)$$

$$\dot{F}_2 = \frac{RT_2}{P_2 V_2} W_{32}(F_3 - F_2) \quad (10)$$

$$\dot{F}_3 = \frac{RT_3}{P_3V_3} [W_{air}(F_{air} - F_3) + W_{63}(F_{4,\tau} - F_3)]. \quad (11)$$

Here, R is the ideal gas constant and V is the volume of the section of the complex air-path denoted by its subscript. F represents the oxygen mass fraction, T_{IC} and T_{HPEGR} are the temperatures of the gas after exiting both the intercooler and HPEGR valve, W_{air} is the mass flow rate of fresh air measured by the mass air flow (MAF) sensor, and W_e is the mass flow rate from the intake into the engine. The subscript τ in Equation (11) indicates the transport delay from section 4 to section 6. The HPEGR mass flow rate, W_{41} , and the LPEGR mass flow rate, W_{63} , are directly related through the orifice equation to the EGR valve angle given in Equation (12) for subsonic flow, $(p_i/p_j) < [2/(\gamma + 1)]^{\gamma/(\gamma-1)}$, and Equation (14) for choked flow, $(p_i/p_j) \geq [2/(\gamma + 1)]^{\gamma/(\gamma-1)}$, where p_i and p_j are the upstream and downstream pressure.

$$W_{ij} = \frac{C_D(\theta)A_v p_i}{\sqrt{RT_i}} \alpha \left(\frac{P_i}{P_j} \right) \quad (12)$$

$$\alpha \left(\frac{P_i}{P_j} \right) = \sqrt{\frac{2\gamma}{\gamma-1} \left[\left(\frac{P_i}{P_j} \right)^{-2/\gamma} - \left(\frac{P_i}{P_j} \right)^{-(\gamma+1)/\gamma} \right]} \quad (13)$$

$$W_{ij} = \frac{C_D(\theta)A_v p_i}{\sqrt{RT_i}} \sqrt{\gamma \left(\frac{2}{\gamma+1} \right)^{\gamma+1/\gamma-1}} \quad (14)$$

The parameter A_v is the fully open valve cross-sectional area and parameter C_D is the flow discharge coefficient which is dependent on the valve angle, θ . The mass flow rate W_{21} is directly controlled through the normalized VGT position when considering Equations (1) through (3). To make this correlation, it is assumed that W_{45} can be

modeled by the same orifice equation for subsonic flow, (15), where the equivalent area S_{VGT} is a function of the normalized VGT position, ϕ .

$$W_{45} = \frac{S_{VGT}(\phi)P_4}{\sqrt{RT_4}} \alpha \left(\frac{P_4}{P_5} \right) \quad (15)$$

This assumption is valid as the dependency of W_{45} on impeller speed is minimal. Similar assumptions have been used to model turbine mass flow rates of VGTs in [21], [17], and [27]. With the objective of controlling intake temperature, pressure, and oxygen mass fraction in mind, the MIMO system is now controllable with inclusion of a third actuator, the LPEGR.

As with any MIMO system there will be a coupled behavior between actuators and system outputs which is commonly regarded as a major obstacle when designing a control strategy for MIMO systems. Equations (7) through (9), which represent the dynamics of the intake manifold conditions, can be used to give insight into this coupled behavior. It is clear that in Equation (7) there is a strong coupling between the HPEGR and VGT position which both significantly affect intake temperature. Even though W_{4I} is small when compared to W_{2I} , the effect HPEGR mass flow rate has on temperature is magnified by the large difference between T_{HPEGR} and T_I , also considering the intake temperature remains relatively close to that of the intercooler. This same coupled effect can be seen for intake pressure as well in Equation (8). When examining the coupled interaction of the intake oxygen mass fraction it is important to take note of Equations (10) and (11), the dynamic equations for the oxygen mass fractions of sections 2 and 3. For these sections, the mean value approximations are only affected by the LPEGR considering relatively small fluctuations in the mass flow rate of fresh air. This

assumption holds as the difference between F_4 and F_3 will always be much higher than the difference between F_{air} and F_3 . This leads to the intake oxygen mass fraction in Equation (9) where the conclusion can be made to that this state is only predominantly affected by the HPEGR and LPEGR mass flow rates, which can be taken into consideration when developing the MIMO controller.

2.5 GT-Power Complex Air-path Simulation

To develop a MIMO controller it is desirable to first develop a state-space representation of the intake manifold dynamics. This is done with an open-loop, GT-Power simulation that acquires the dynamic response of the intake manifold conditions caused by random excitation of the HPEGR, LPEGR, and VGT. GT-Power is an advanced computational tool used by the automotive industry to simulate engine operation. Namely, what makes GT-Power such a highly regarded utility is its flexibility to be used in a multitude of aspects throughout every facet of automotive design. This can vary from applications in combustion, thermal management, acoustics, and emissions. The versatility of the program comes from its multi-physics platform for constructing models of general systems based on the underlying principals of thermodynamics, fluid flow, combustion, acoustics, etc. These ultimately revolve around the laws for conservation of mass, energy, momentum, and the ideal gas law which are applied to finite volumes of flow [8]. GT-Power's key characteristics that are applied to this work, beyond the conventional scope of mean value modeling, are its ability to simulate variable transport delay along with one-dimensional wave dynamics and heat transfer throughout the engine breathing system.

An important feature of GT-Power that was used in this work is its ability to be coupled with Simulink. This allows for the user to provide input actuation and interpret transient data through Matlab. Here, a masked GT-Power model of an actual medium-duty, V8, equipped with a HPEGR loop and two-stage, VGT, turbocharging system, similar to the one depicted in Figure 5 without the LPEGR loop, was donated by the Ford Motor Company. Included were a multitude of temperature, pressure, oxygen mass fraction, and mass flow rate signals from the different components of the air path from within the black box model that could be used for analysis. The provided model allowed for external actuation of the HPEGR valve, between 0 and 90 degrees, and the normalized VGT position, between 0 and 1. The LPEGR loop, cooler, and valve were added externally to the masked model to complete the complex air-path of Figure 5. The final graphical user interface of the medium-duty, V8, GT-Power engine model can be seen in section A.1 of the appendix.

2.6 Complex Air-path System Identification

The system identification is done through the use of the System Identification Toolbox available in Matlab. The System Identification Toolbox estimates either linear or nonlinear mathematical models of dynamic systems from either measured or simulated data. The GT-Power engine model was coupled with Simulink and supplied random excitation signals for the normalized VGT position, LPEGR, and HPEGR. The dynamic response of the intake temperature, pressure, and oxygen mass fraction were simultaneously read from the GT-Power engine model for evaluation.

Although the GT-Power engine model requires a valve angle command for HPEGR and LPEGR excitation, for system identification purposes it was desired to use

mass flow rate commands for both HPEGR and LPEGR paths. One of the benefits to this approach can be seen in the mean value approximations of Equations (7) through (11) where the intake condition dynamics are directly related to the mass flow rates of the HPEGR and LPEGR. Another reason is that there is a severe non-linearity incorporated in the orifice equation between the valve angle and mass flow rate. This can be seen in Figure 6 which shows the GT-Power simulation of LPEGR mass flow rate as a function of the valve angle.

Figure 6. GT-Power simulation of LPEGR mass flow rate as a function of valve angle

By exciting the GT-Power engine model through a mass flow rate command this non-linearity can be avoided. Accurately commanding the mass flow rate through Simulink was accomplished through inversion of the orifice equations, (12) and (14), to calculate the proper discharge coefficient $C_D(\theta)$. By using steady-state mass flow data from GT-Power it was possible to find a first order function for $C_D^{-1}(\theta)$ to obtain a valve angle command for a calculated discharge coefficient. Additional sensors were needed upstream of the HPEGR and LPEGR valves to measure pressure and temperature. Downstream pressure for the HPEGR is the pressure of intake manifold, an already measured state. Downstream pressure for the LPEGR was assumed constant, at atmospheric pressure. Figure 7 shows the accuracy of the commanded mass flow rates and the actual mass flow rates of the HPEGR and LPEGR paths by inverting the orifice equations. Normally sensors upstream of valves for mass flow rate commanding are not standard equipment in diesel engines but other strategies have been developed to accurately estimate these measured conditions. Examples of such work using observers to estimate conditions upstream of EGR valves can be seen in [17],[40], and [48].

Figure 7. Both commanded and actual mass flow rates of HPEG and LPEG paths in GT-Power

After achieving accurate mass flow rate commanding for the HPEGR and LPEGR, the open-loop simulation of the GT-Power engine model was subjected to excitations of the HPEGR and LPEGR valves using evenly distributed, uniform, random mass flow rate command excitations, W_{41} and W_{63} . Additional uniform excitation was provided to the normalized VGT position, ϕ . The upper limit for the LPEGR mass flow rate excitation signal was set to not let the LPEGR valve exceed 30 degrees, where the LPEGR mass flow rate begins to saturate as seen in Figure 6. The exhaust valve was set to a constant 9 degrees which was found to maintain enough upstream pressure for adequate LPEGR flow at saturation. The HPEGR mass flow rate excitation was limited so that the minimum allowable AFR ratio that diesel engines conventionally operate under was not breached, at an AFR of 21 [12], considering the upper limit on the LPEGR mass flow rate. VGT excitation was chosen in a range that would not extend into the system non-linearity, similar to the work in [25], around a mean value based on provided static mapping of VGT position to engine speed and fueling command. Engine speed and fuel injection quantity can be big disturbances when considering identification of intake system dynamics. For this reason they were kept constant at 2000 RPM and 40 mg injections per cylinder per cycle (mg/cl/cy). Normally with the given speed and fuel injection command the simulated engine output torque is 408.8 Nm, with the exhaust valve fully open. It is important to note that with the given exhaust valve angle used for sufficient LPEGR flow, the mean engine output torque during the open-loop simulation was 390.7 Nm, a reduction of 4.43%. Figure 8 shows the excitation signals used for system identification using commanded HPEGR and LPEGR mass flow rates with normalized VGT position.

Figure 8. Excitation signals used for specified HPEGR and LPEGR mass flow rate commands and VGT position

In general, identifying system dynamics consists of several steps needed in both pre-processing data and validation. These primarily consist of properly filtering noise acquired in the system output, removing any offsets or trends that exist within both the input excitation and system response, separating the input excitation and intake manifold response signals into parts for either identification or validation, and finally evaluating uncertainties between the system dynamics and state-space representation [24].

Because GT-Power is able to simulate the one-dimensional wave dynamics throughout the engine breathing system, a large amount of noise was seen for intake manifold conditions as shown in Figure 9 for an arbitrary segment of the dynamic response. It is essential to filter this noise to isolate the dynamic response, caused by the excitation signal, as much as possible to be able to identify the system dynamics with the best accuracy. Analysis of the noise was done with a Fourier transform on the intake response as shown in Figure 10. Through this analysis it can be seen that the primary frequencies that are contributing to the noise are multiples 16 Hz, the intake valve opening frequency of for each cylinder at 2000 RPM. To filter this noise a 9 Hz low-pass filter was applied to the open-loop simulation output of intake conditions. The filtered intake response data can also be seen in Figure 9.

Figure 9. Segment of both filtered and unfiltered open-loop intake manifold dynamic response

Figure 10. Fourier Transform analysis of open-loop intake manifold dynamic response

Prior to identification the excitation signals and intake dynamic responses were subtracted by their mean values so all signals would be oscillating about zero. This was done for Matlab to be able to evaluate the relative change in HPEGR and LPEGR mass flow rates and VGT position to changes in intake conditions with all initial states being zero. Data from the complex air path simulation was then evenly split into two halves. The first half was used to formulate a linear, time-invariant (LTI) state-space representation of system dynamics using the prediction error method in Matlab. Previous research using mean value models for control of dual-loop EGR have shown that each intake manifold condition contains about 4 to 5 states to accurately represent the system dynamics [27][6][40]. This led to model orders of 10 through 20 being specified in Matlab for the prediction error method to identify the system. For each specified model order, the error between the state-space representation and simulation data was evaluated through excitation and comparison with the second half of the data set. The halving of the data into parts for identification and comparison helps to confirm the validity of the state-space model for all possible excitation signals within the simulated operating range. The identified linear model with the least uncertainty can be seen in Figure 11 as compared to the second half of data which was found to have an order of 14. The order of the model makes sense intuitively as there are 6 components in the complex air-path mean value model and two turbochargers. This also coincides with between 4 and 5 states needed to accurately model each of the three intake conditions.

As shown in Figure 11 the identified model most accurately predicts intake manifold temperature with a slight reduction in accurately predicting pressure and oxygen mass fraction. Error caused by the inherent non-linearity of the complex air-path

is most predominantly seen at the limits of the intake conditions where the open-loop response deviates from the state-space model the most. Other open-loop simulations with narrower excitation signals produced a model with less error as the effects of system non-linearity were less pronounced. These narrower models though were not used in developing a MIMO controller as they would not be able to harness the complete operating range of the complex air-path. A further investigation can also localize the errors induced through inaccurate identification of manifold dynamics, with a comparison between the step responses of each individual actuator for all outputs shown in Figure 12. For this evaluation the magnitude of each step input is the maximum allowable excitation used for the open-loop simulation. Figure 12 shows that most of the error in modeling temperature comes from an inaccurate identification of intake temperature dynamics caused by the LPEGR and VGT. Also, the uncertainty in intake pressure dynamics is caused by inaccurate identification of HPEGR and LPEGR and uncertainty in oxygen mass fraction dynamics is caused by inaccurate identification of all three actuators. This evaluation of uncertainty between actuators and intake conditions later becomes imperative when designing a robust MIMO controller.

Figure 11. Identified state-space representation of offset intake manifold dynamics compared to GT-Power

Figure 12. Step response of both GT-Power simulation and identified model

CHAPTER 3: ROBUST MIMO CONTROLLER DEVELOPMENT

3.1 Multivariable Plant Analysis

Development of a multivariable controller will require a key understanding of the difference between response characteristics of MIMO and single-input single-output (SISO) plants. Considering the system of (16) the gain of the SISO plant $H(s)$ is only dependent on frequency ω and is independent of scalar input magnitude $|u(j\omega)|$ which follows Equation (17).

$$y(s) = H(s)u(s) \quad (16)$$

$$\frac{|y(j\omega)|}{|u(j\omega)|} = \frac{|H(j\omega)u(j\omega)|}{|u(j\omega)|} = |H(j\omega)| \quad (17)$$

This is however not the case for MIMO systems where the input is a vector which not only has magnitude but direction as well. Therefore the gain of the MIMO complex air-path plant, $G(s)$, that was identified in the previous chapter, follows the equation

$$\frac{\|y(j\omega)\|_2}{\|u(j\omega)\|_2} = \frac{\|G(j\omega)u(j\omega)\|_2}{\|u(j\omega)\|_2}. \quad (18)$$

Where $\|\cdot\|_2$ denotes the vector norm. For the complex air-path system described in Chapter 2, and the rest of this work, the input will be the vector $u(j\omega) =$

$[W_{41} \quad W_{63} \quad \varphi]^T$ and the output is the vector $y(j\omega) = [T_1 \quad P_1 \quad F_1]^T$. It can be seen

through Equation (18) that the gain of plant $G(j\omega)$ is dependent on both frequency ω and direction of the input vector $u(j\omega)$. To analyze this dependency on input direction the singular value, σ , is used which is equal to the positive square root of the eigenvalue, λ , for the equation

$$\sigma_i(G) = \sqrt{\lambda_i(G^T G)}. \quad (19)$$

Where $G(j\omega)$ is substituted by G for the sake of simplicity. The maximum singular value indicates the maximum gain of the system to an input vector with vector norm equal to one and vice versa for the minimum singular value where

$$\max_{\|u\|_2=1} \frac{\|Gu\|_2}{\|u\|_2} = \max_{\|u\|_2=1} \|Gu\|_2 = \bar{\sigma}(G), \quad (20)$$

$$\min_{\|u\|_2=1} \frac{\|Gu\|_2}{\|u\|_2} = \min_{\|u\|_2=1} \|Gu\|_2 = \underline{\sigma}(G). \quad (21)$$

To perform this analysis on the identified linear model, a normalization on G was done using Equation (22) with the normalization matrices D_u and D_y . These diagonal matrices are composed of the maximum amplitudes of excitation inputs, $\hat{u}$, and intake condition outputs, $\hat{y}$, during the open-loop simulation.

$$\hat{G} = D_y^{-1} G D_u \quad (22)$$

$$D_u = \text{diag}\{\hat{u}\} \quad (23)$$

$$D_y = \text{diag}\{\hat{y}\} \quad (24)$$

The maximum and minimum singular value for the normalized plant, $\hat{G}$, are shown in Figure 13 in the frequency domain. An illustration of the system's dependency on input direction can be seen in Figure 15 which is the steady-state output of $\hat{G}$ with the inputs of Figure 14. The vectors shown in Figure 14 are the normalized inputs, with vector norms equal to one, which correspond to the maximum and minimum singular values given with the vectors in Figure 15. Evaluating the severity of the complex air path system dependency on input direction can be done using the condition number, $\beta = \bar{\sigma}/\underline{\sigma}$, which for the case of the normalized complex air-path is 3.43. This indicates that the gain of the complex air-path may change by up to 3.43 times depending on the direction of the input vector direction. Such a high magnification of system gain could lead to instability and constitutes the need for developing a multivariable feedback controller.

Figure 13. Maximum and minimum singular values of the normalized complex air-path

Figure 14. Inputs corresponding to maximum and minimum singular values, all with vector norms equal to one

Figure 15. Steady-state output of the normalized complex air-path with inputs of Figure 14

3.2 Robust Multivariable Feedback Controller Formulation

The multivariable controller follows the schematic shown in Figure 16 where G' is the simulated GT-Power engine model. The controller performs the same actuator commands as the excitation signals used in identification of the complex air-path dynamics therefore it requires the same inversion of the orifice equations (12) and (14). Low-pass, pre-filters are used to isolate the desired reference conditions to a given frequency range and eliminate high frequency excitation of the controller. The MIMO feedback controller is designed to operate about the mean values of the excitation signals used in identification of the complex air-path. Such mean values exist as the controller's initial conditions and are represented graphically in Figure 16 by offsetting the control

output by the mean values of excitation signals in Figure 8. Doing this allows for the MIMO controller to operate about the linearization point used in system identification and is why the H_∞ controller will only be presumed effective within the operating range set by the excitation signals.

Figure 16. MIMO Feedback control strategy

3.2.1 The MIMO Controller Design

Figure 17. General control figuration for H_∞ MIMO controller formulation

A feedback control strategy can be developed by utilizing the LTI model identified for intake conditions, G . Here, a H_∞ controller formulation is preferable to minimize the effects of cross-coupling between actuators and system dependency on input direction for optimal performance. The goal of the H_∞ control formulation is to find the optimal controller, K , which minimizes the infinity norm from the system inputs to outputs of the generalized plant, P , depicted in Figure 17. The infinity norm is denoted

as $\|\cdot\|_\infty$ and follows the definition of Equation (25). The H_∞ formulation follows Equation (26) where L is the linear fractional transformation of P and K .

$$\|H(s)\|_\infty = \max_{\omega} \bar{\sigma}(H(j\omega)) \quad (25)$$

$$\min_k (\|L(P, K)\|_\infty) = \min_k \left[\max_{\omega} \bar{\sigma}(L(P, K)(j\omega)) \right] \quad (26)$$

In Figure 17 the desired reference signals, r , and the noise, n , of intake temperature, pressure, and oxygen mass fraction are

$$n = [T_n \ P_n \ F_n]^T, \quad (27)$$

$$r = [T_d \ P_d \ F_d]^T. \quad (28)$$

The noise of each signal can be attributed to the higher frequency, one-dimensional wave dynamics simulated in GT-Power that are still present after the filter developed in the section on Complex Air-Path System Identification. It would be undesirable for the controller to react to these high frequency signals. Output q consists of the weighted EGR mass flow rate commands and the VGT position induced by the MIMO controller. The output e consists of the tracking error of intake conditions where

$$q = QW_q[W_{41} \ W_{63} \ \varphi], \quad (29)$$

$$e = EW_e[T_1 - T_d \ P_1 - P_d \ F_1 - F_d]^T. \quad (30)$$

As shown in [36], tuning can be done via weighting functions of inputs and outputs. The frequency dependent weighting function for the observed intake manifold

noise, W_n , is a block diagonal of the transfer function shown in Figure 18. Also in Figure 18 are the weighting functions used for the reference signals, W_r , which are equal to the low-pass filters used in the Pre-Filter block of Figure 16 for each desired intake condition. With Equation (26) in mind, it is intuitive that to stress the minimization of outputs q and e at specified frequencies, their respective weighting functions must be at their highest at these key frequencies. Such representation leads to the weighting functions to be thought of as costs which are to be minimized with controller design. This leads to a low-pass filter design for the weighting function W_e to minimize steady-state error, and a high-pass filter design for the weighting function W_q to minimize controller energy at high frequencies. These weighting functions can be seen in Figure 19 where

$$W_q = \text{diag}\{W_{qHP}, W_{qLP}, W_{qVGT}\}, \quad (31)$$

$$W_e = \text{diag}\{W_{eT}, W_{eP}, W_{eF}\}. \quad (32)$$

Tuning of the controller was performed by adjusting the frequencies at the zero magnitude crossings of the output weighting functions.

Signals for inputs n and r are scaled by matrix diagonals N and R and outputs q and e are scaled by matrix diagonals Q and E as shown in Figure 17 following the suggestions given in [36]. For the noise, the matrix N consisted of the scaling factors equal to the amplitudes of the noise in each intake condition that still existed after being filtered. For the reference signals, the matrix R is equal to D_y , the matrix used in the analysis of the normalized plant in the previous section which represents the intake condition operating range of the complex air-path.

Diagonal matrices Q and E were used to scale the outputs of control energy and tracking error from the generalized plant. Following the same thinking described for developing the weighting functions that represent the cost of controller energy and tracking error, the matrices Q and E are defined with the inversions in Equations (33) and (34). Equation (33) sets the cost scaling matrix Q so the magnitude of QW_q will be equal to D_u^{-1} at the frequency of zero magnitude in Figure 19. Defining Q in such manner gives symmetry to the controller formulation where the controller energy cost is scaled by the operating range of the actuators in the complex air-path and the reference signals are scaled by the intake condition operating range. This symmetry allows for an obscure confinement of the MIMO controller to the operating range in a sense that the weighting functions create an elastic tether on controller actuation. The scaling on the tracking error, E , was set so that at the frequency of zero magnitude, the combined weight would reflect one tenth of the scaled reference signal. For this reason D_y^{-1} is multiplied by 10 in Equation (34) to define the cost scaling matrix E so the magnitude of weighted cost, W_eE , will be equal to ten times R^{-1} at the frequency of zero magnitude in Figure 19.

$$Q = D_u^{-1} \quad (33)$$

$$E = R^{-1} \times 10 \quad (34)$$

Figure 18. Weighting functions used for noise and reference signal inputs

Figure 19. Weighting functions used for controller energy and tracking error outputs

3.2.2 Representation of Uncertainty

In Figure 17, the plant G is the multivariable linear dynamic model of intake conditions identified in the previous chapter which does not represent the actual GT-Power engine simulation, G' , in Figure 16. This simulation is a better representation of actual engine dynamics and can account for system non-linearity and variable transport delay which are neglected by the identified linear model, G , that the MIMO controller design is based on. Differences between the G and G' may cause instability or at least a sacrifice of performance when using a controller that is not robust in regards to these differences. Actual engine control may differ even further because of GT-Power model inaccuracies, engine degradation, variability between engines, changes in atmospheric conditions, etc. In development of a MIMO controller it is important to account for these differences with a representation of model uncertainty. Making the connection between the GT-Power simulation and the identified linear model can be seen in Equation (35) and Figure 17 through additive uncertainty. W_u is a transfer function matrix which weights the upper bounds of uncertainty between the actuators of the LTI state-space model and the simulated complex air-path response in the frequency domain. Δ represents all possible perturbations satisfying the given constraint of Equation (36).

$$G'(s) = G(s) + W_u(s)\Delta(s) \quad (35)$$

$$\begin{aligned} \Delta(s) &= \text{diag}\{\delta_1(s), \delta_2(s), \delta_3(s)\}; \\ \|\Delta(j\omega)\|_\infty &\leq 1 \quad \forall \omega \end{aligned} \quad (36)$$

Defining the upper bounds of uncertainty, W_u , for the MIMO complex air-path began with a comparison between the open-loop, GT-Power simulation and the identified linear model G . Figure 11 shows that most of the uncertainty between G' and G is in the response characteristics of intake pressure and oxygen mass fraction. Evaluating the contribution of each actuator to this uncertainty can be done with the use of Figure 12 which is a comparison between the step responses of each individual actuator to the response of intake condition. For intake pressure, the main contributors to uncertainty are the EGR flow rate commands, W_{41} and W_{63} . For intake oxygen mass fraction, all three controller actuators contribute to a noticeable uncertainty. For intake temperature, LPEGR and VGT are the main contributors but because their uncertainty is small in comparison to the full operating range of intake temperature, no uncertainty is modeled with respect to this intake condition. This leads to the complete matrix for W_u seen in Equation (58) of the appendix. The frequency response of each actuator that contributed to the uncertainty of intake conditions between G' and G was evaluated and the difference between the two was defined as the upper bounds of the uncertainty for each component of W_u in Equation (37). The subscript m represents row index of the intake response and the subscript n represents the column index for the controller input. A complete representation of W_u can be seen in section A.2 of the appendix.

$$|W_{u_{mn}}(j\omega)| = |G'_{mn}(j\omega) - G_{mn}(j\omega)| \quad (37)$$

3.2.3 Robust Stability Conditions and Controller Development

As defined in [36] for a system of robust stability, the system must be stable for all perturbed plants about the nominal model up to the worst-case model uncertainty.

Evaluating the conditions for robust stability for a given controller K can be done through rearranging the system L to define the transfer function M-structure given in [15] where

$$[L(P, K)] \begin{bmatrix} n \\ r \\ \Delta y \end{bmatrix} = \begin{bmatrix} q \\ e \\ u \end{bmatrix}, \quad (38)$$

$$M(j\omega) = W_u(j\omega)L_{33}(P(j\omega), K(j\omega)). \quad (39)$$

This represents the transfer function from the perturbations of the intake conditions, Δy , to the bounded uncertainty of controller inputs, $[W_u]u$, in Figure 17. As stated in [36] and [16], for the complete system to satisfy robust stability the conditions of Equation (40) must be met, which is also equivalent to Equation (41).

$$\det(I - M(j\omega)\Delta(j\omega)) \neq 0; \quad \|\Delta(j\omega)\|_\infty \leq 1, \quad \forall \omega \quad (40)$$

$$\|M(j\omega)\|_\infty < 1; \quad \|\Delta(j\omega)\|_\infty \leq 1, \quad \forall \omega \quad (41)$$

The fact that the perturbations form the diagonal structure of Equation (36) allows for a less conservative constraint on robust stability by taking advantage of the fact that robust stability must be independent of scaling. Proof of structured uncertainty's independence to scaling was first shown in [15]. By scaling the inputs and outputs of M and Δ by the transfer function matrix D , which has the same diagonal structure as Δ , robust stability can still be maintained if $\|DMD^{-1}\|_\infty < 1$ is satisfied where the scaled M-structure now follows $M = DMD^{-1}$ and $\Delta = D\Delta D^{-1}$. To find the least conservative constraint on robust stability it is then possible to follow Equation (42) as suggested by

[36]. This is regarded as the least conservative constraint as we assume all perturbations from the nominal model G are bounded by the components of W_u .

$$\min_D \|D(j\omega)M(j\omega)D(j\omega)^{-1}\|_\infty < 1; \quad \forall \omega \quad (42)$$

The transfer function matrix D gives the additional degree of freedom required for synthesis of a robust MIMO controller. By using the DK iteration method, that was first introduced in [14], a robust controller was developed for the complex air-path through iterations of Equations (26) and (42) by finding the optimal controller K for a constant scaling matrix D , then an optimal matrix D for given controller K , effectively a process of Equation (43).

$$\min_K (\min_D \|DL(P, K)D^{-1}\|_\infty). \quad (43)$$

There are slight disadvantages for the process of Equation (43) where finding the optimal controller may converge to a local optimum as each iteration is dependent on finding the optimal solutions using K or D . The DK iteration can also lead to a computationally intensive controller as it will be at least the order of the plant G , plus the order of the weights, plus the order of the scaling matrix D .

3.2.4 Robust Feedback Controller Validation and Analysis

Validation of the robust feedback controller can be seen in Figure 20 when using the GT-Power model of the medium-duty, V-8, diesel engine, equipped with the complex air-path seen in Figure 5, to simulate the plant. Figure 21 shows both the controller actuation signals and the fuel injection command during the simulation. The simulation was performed only using the sensors in Figure 5 for feedback of intake manifold

conditions and commanding HPEGR and LPEGR mass flow rates. Specified intake conditions in Figure 20 were set to not only test the controller's performance at the boundaries of the complex air-path's operating range but also to showcase the flexibility of intake conditions that can be achieved through the use of dual-loop EGR. Additionally the feedback controller's ability to reject disturbances caused by a change in the fuel injection command is shown at the end of the simulation, where torque was increased from 393.7 to 451.9 Nm and then reduced to 329.5 Nm. During the simulation engine speed was maintained at 2000 RPM and the exhaust valve was maintained at a constant 9 degrees. Flexibility of the dual-loop EGR system can be seen in Figure 20 where intake temperature can be controlled independently of the oxygen mass fraction. This has the potential to widen the operating range of the discussed advanced combustion modes where stability is heavily dependent on oxygen mass fraction and extremely sensitive to temperature. Evaluation of controller stability to the modeled uncertainty of W_u can be performed by including a robust stability margin parameter, k_m , in Equation (40) as introduced in [34]. For the controller developed in Equation (43) the robust stability margin was found to be 2.73 where

$$\min\{k_m | \det(I - k_m M \Delta(j\omega)) = 0; \|\Delta(j\omega)\|_\infty \leq 1, \forall \omega\}. \quad (44)$$

Showing the tracking performance of the controller, the maximum singular value of the closed-loop system transfer function, of Figure 17, from the desired intake condition, r , to the tracking error, e , can be seen in Figure 22. Here the maximum singular value is minimal for the duration of the low frequency bandwidth, which

indicates good steady-state performance, and rises at higher frequencies where tracking performance degrades.

Figure 20. Robust feedback controller validation

Figure 21. Robust feedback controller signals and fuel injection command during validation

Figure 22. Maximum singular value of the closed-loop system from the reference signal to the tracking error

3.3 Coupled Feed-Forward and MIMO Feedback Control

Although the MIMO feedback scheme of Figure 16 performs quite well in controlling intake conditions it is not without its limitations. For example the performance of the MIMO controller is restricted by the linearized operating range of G around the initial conditions of the controller. It is not guaranteed that performance can be maintained outside this operating range due to increased non-linearity and unidentified dynamics. Additionally the MIMO controller is not designed for disturbance rejection caused by for example a change in fueling or engine speed. The added feed-forward controller, as seen in Figure 23, can be used to both mitigate disturbances and designate

actuator set points to reduce the effects of non-linearity. The feed-forward controller is based on the inverted, steady-state, mean value approximations of Equations (1) through (9). Similar feed-forward and MIMO feedback air-path control schemes have been effectively used in [29] and [43]. MIMO feedback control will still be necessary to maintain both steady-state and dynamic performance due to errors naturally inherent in the mean value approximations. Controller performance and stability though may not be guaranteed as the MIMO controller was designed principally on the linear state-space representation, G , identified from the operating range of the open-loop system excitation, which does not include the added dynamics of the controller. If necessary, a more extensive validation may be required for the control scheme of Figure 23.

Figure 23. Coupled feed-forward and MIMO feedback control scheme

In a multivariable system there are many possibilities available when deriving control laws for feed-forward control because of the cross-coupled effect between actuators. Deciding on the best possible combination for which actuator to control each intake condition was done using the steady-state gains of the complex air-path, shown in

the step responses of Figure 12. To maximize the operating range the highest gain of each actuator for each output was used to delegate the intake condition that the feed-forward control law would be based upon. For a maximum temperature range, the feed-forward temperature control would use the HPEGR mass flow rate, which clearly has the highest temperature gain. This leaves the feed-forward control of pressure to be performed using the VGT and the LPEGR to control oxygen mass fraction.

The feed-forward controller also requires the use of additional sensors to be added which can be seen in Figure 24 of the complex air-path used to simulate the coupled feed-forward and MIMO feedback control scheme. In this example, temperature and pressure signals used for accurately commanding the mass flow rate of the HPEGR now come from the exhaust manifold, section 4, instead of just upstream of the HPEGR valve. This allows for fewer sensors to be used in the coupled control scheme of Figure 23, a considerable benefit when considering mass production. It will later be seen that this consolidation induces a slight error for the feed-forward controller to accurately command W_{41} but the coupled feedback control is able to reduce the affects of this error on intake conditions.

Figure 24. Complex air-path with sensors used for coupled feed-forward and MIMO feedback control

3.3.1 Feed-Forward EGR Control

EGR feed-forward controllers of intake temperature and oxygen mass fraction were developed based on inverted, steady-state representations of the mean value approximations of Equations (7) and (9), respectively. This leads to the HPEGR control law of Equation (45) for the feed-forward intake temperature control and requires the use of the MAF sensor and the exhaust manifold temperature sensor. The feed-forward control law for the oxygen mass fraction using the LPEGR follows Equation (46) where an additional sensor for F_4 was needed and a rough estimate of the transport delay was required. The formulation of Equations (45) and (46) were done under the steady-state assumption of Equation (47) that ignores filling dynamics of the complex air-path.

$$W_{41} = (W_{air} + W_{63}) \frac{T_d - T_{IC}}{T_4 - T_d}. \quad (45)$$

$$W_{63} = \frac{W_{air}(F_{air} - F_d) + W_{41}(F_4 - F_d)}{F_d - F_{4,\tau}} \quad (46)$$

$$W_e = W_{41} + W_{21} = W_{41} + W_{63} + W_{air} \quad (47)$$

3.3.2 Feed-Forward VGT Control

Feed-forward control of the VGT follows the steady-state inversion of Equations (1), (2), and (3) where a desired intake pressure, P_d , must correspond to a desired pressure of section 4, $\tilde{P}_4$, through Equation (48). $\tilde{P}_4$ can then be used to set the VGT position using Equation (49), which assumes turbine mass flow rate can be accurately modeled as an orifice equation, and the inversion $S_{VGT}^{-1}(\varphi)$. Finding the inversion $S_{VGT}^{-1}(\varphi)$ from the masked, GT-Power model was made possible by using a first order function to fit steady-state mass flow rate data from GT-Power.

$$\frac{\tilde{P}_4}{\tilde{P}_5} = \left\{ 1 - \frac{c_{pi}\hat{T}_2}{c_{px}T_3\eta_{ht}\eta_{hc}} \left[\left(\frac{P_d}{\tilde{P}_2} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] \right\}^{\frac{\gamma-1}{\gamma}} \quad (48)$$

$$S_{VGT}(\varphi) = \left\{ \frac{(W_a + W_{63})c_{pi}\hat{T}_2\sqrt{R}}{c_{px}\eta_{ht}\eta_{hc}\tilde{P}_4\sqrt{T_3}\alpha\left(\frac{\tilde{P}_4}{\tilde{P}_5}\right)} \frac{\left[\left(\frac{P_d}{\tilde{P}_2} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]}{\left(\left(\frac{\tilde{P}_4}{\tilde{P}_5} \right)^{\frac{1-\gamma}{\gamma}} - 1 \right)} \right\}^{\frac{\gamma-1}{\gamma}} \quad (49)$$

Because turbo maps were not available, steady-state data from the masked GT-Power engine model was used to estimate the turbine and compressor efficiencies using the mean value approximations of Equations (1) through (6). Efficiency maps were

created as a function of VGT position, HPEGR valve position, and fuel injection command and used in conjunction with the feed-forward controller. Equations (48) and (49) require estimates of the states $\hat{P}_2$, $\hat{P}_5$, and $\hat{T}_2$ which are done in the same fashion as shown for the feed-forward controller of [29]. Here an estimate of $\hat{T}_5$ was attained using a first order approximation based on the temperature of the exhaust manifold, T_4 . The pressure of section 5 can be estimated using Equation (50), which is also based on the assumption that the mass flow rate through a turbine can be modeled from the orifice equation, Equation (15), and the steady-state assumption of Equation (47). This estimation also requires S_{LPT} , the equivalent area of the low pressure turbine, and the known mass flow rate of fuel, W_f . The difficulty with this estimation is that the orifice equation is not invertible with respect to upstream pressure so a constant estimate was made for $\bar{\alpha} \left(\frac{\hat{P}_5}{P_{dlpt}} \right)$ where P_{dlpt} is the measured pressure downstream of the low pressure turbine. The accuracy of this approximation can be seen in Equation (13) where the ratio of specific heats for burned exhaust gas allows for α to remain relatively constant with a changing in pressure ratio because of the numerical proximity in the pressure ratio exponents.

$$\hat{P}_5 = \frac{(W_{air} + W_{63} + W_f) \sqrt{R \hat{T}_5}}{S_{LPT} \bar{\alpha} \left(\frac{\hat{P}_5}{P_{dlpt}} \right)} \quad (50)$$

$$\frac{\hat{P}_2}{P_3} = \left[\frac{\hat{T}_5 \eta_{lt} \eta_{lc} c_{px}}{T_3 c_{pi}} \left(1 - \left(\frac{\hat{P}_5}{P_{dlpt}} \right)^{\frac{1-\gamma}{\gamma}} \right) + 1 \right]^{\frac{\gamma}{\gamma-1}} \quad (51)$$

$$\frac{\hat{T}_2}{T_3} = \left(\frac{\hat{P}_2}{P_3} \right)^{\frac{\gamma-1}{\gamma}} \quad (52)$$

Estimating the pressure in section 2 was done using Equation (51) by inverting Equations (4) through (6) at steady-state. The pressure and temperature of section 3 were assumed constant at atmospheric conditions. The temperature of section 2 was estimated using Equation (52) which assumes isentropic compression through the LPT compressor.

3.3.3 Feed-Forward Controller Performance

Figure 25 shows the complex air-path simulation when only using the feed-forward controller. Engine speed was kept constant at 2000 RPM. The control signals during the simulation can be seen in Figure 26 along with the fuel injection command. The inaccuracy of feed-forward control for intake temperature, specifically at high temperatures, is mostly attributed to the use of the exhaust manifold gas temperature signal, T_4 , in Equation (45) instead of the temperature of the gas entering the intake just behind the HPEGR valve, where considerable thermal losses due to external heat transfer have taken place. Additional lack in steady-state performance is due to the error this induces in accurately commanding the HPEGR mass flow rate as seen in Figure 26. Feed-forward control of intake oxygen mass fraction is only slightly hindered by the inaccuracy in commanding the HPEGR mass flow rate and LPEGR valve saturation. Although the consolidation of the temperature and pressure sensors in the exhaust manifold induces errors in feed-forward control, it makes the use of the coupled controller more economical and the possibility of predicting the temperature of the EGR gas just behind the HPEGR valve using empirical heat transfer parameters is feasible as

shown in [41] and [27]. Also the feedback controller that will be coupled with the feed-forward will be expected to remove these errors and improve performance.

The severe lack of feed-forward controller performance for intake pressure can be attributed to the estimates of $\hat{P}_2$, $\hat{P}_5$, and $\hat{T}_2$ needed for defining the VGT position. These estimates can be seen in Figure 27 for the complex air-path simulation. Obviously the offset error of the estimates is a significant contributor to the steady-state feed-forward pressure error in Figure 25. This was shown in using measured values of P_2 , P_5 , and T_2 , which reduced steady-state, feed-forward pressure control error by half of what is seen in Figure 25. Inaccurate mapping of turbo efficiencies from inside the masked model may also be present in the feed-forward controller due to errors inherent to reverse engineering these values based on mean value approximations. One could assume that feed-forward pressure control would improve significantly if more accurate information regarding the turbocharger efficiencies was available.

Figure 25. Feed-Forward control of the complex air-path

Figure 26. Feed-forward control signals, actual EGR mass flow rates, and fuel injection command

Figure 27. Estimation of states needed for feed-forward VGT control

3.3.4 Coupled Feed-Forward and MIMO Feedback Validation and Analysis

The simulation of the coupled controller using both feed-forward and MIMO feedback control can be seen in Figure 28. The MIMO controller used is the same as the one used in the simulation of Figure 20 with just feedback control. There is a clear improvement in the dynamic performance of the coupled controller over the use of only the MIMO feedback controller. Both intake pressure and oxygen mass fraction have quicker responses and are able to reach steady-state conditions much faster. The faster response can directly be attributed to the use of the feed-forward controller that acts without the delay normally associated with feedback signals. Dynamic performance of the coupled controller to reach desired intake temperatures remained relatively unchanged from the feedback controller which already showed good performance regarding this state. Intake pressure disturbance rejection also improved with only a small sacrifice in disturbance rejection capabilities in the oxygen mass fraction.

Figure 28. Coupled controller validation with feed-forward and MIMO feedback control

Figure 29. Coupled controller signals, actual EGR mass flow rates, and fuel injection command

CHAPTER 4: DECENTRALIZED CONTROL

4.1 Design Approach

The previous chapter showed the advanced approach for MIMO control where an formulation is performed using complete dynamics of the complete multivariable complex air-path. To make an appropriate comparison between the MIMO approach and more basic methods a decentralized controller is designed using the control scheme of Figure 30. Decentralized control of a multivariable system is essentially the assignment of each actuator to control a single system output, or equivalently SISO proportional-integral (PI) controllers working in parallel. The key feature of decentralized control is that controller design still incorporates an analysis of the multivariable system for tuning and actuator selection which mitigates design effort and can optimize controller performance. This also allows for a level comparison of the feedback controllers without committing lack of decentralized controller performance to inappropriate variables and complexities in PI tuning.

Figure 30. Decentralized control scheme of the complex air-path

Actuator delegation of the decentralized controller was done in the same fashion as the feed-forward controller of section 3.3, where to expand the controller's operating range the highest steady-state gain of each intake condition assigned each actuator. For feedback control of MIMO systems this has a similarity to assigning actuators using the relative gain array (RGA). As given in [36], the components of this array are the ratio between the open-loop gain of the actuator to the gain of the actuator if all other loops are closed and under perfect control. This gives indication to how the gain of an actuator will be affected by the closed-loop control of the other actuators. Determining the RGA for the controller input vector of $u(j\omega) = [W_{41} \quad W_{63} \quad \varphi]^T$ and the output vector of $y(j\omega) = [T_1 \quad P_1 \quad F_1]^T$ follows Equation (53) where $\times$ denotes an element by element multiplication. For the complex air-path, the RGA at steady-state can be seen in Equation (54). The components of the RGA close to one indicate that the actuator assignment of the decentralized controller, PI , should follow Equation (55), where the HPEGR controls intake temperature, the LPEGR controls oxygen mass fraction, and the VGT controls pressure.

$$RGA(G(j\omega)) = G(j\omega) \times [G(j\omega)^{-1}]^T \quad (53)$$

$$RGA(G(0)) = \begin{bmatrix} 1.041 & -0.043 & 0.002 \\ 0.009 & 0.030 & 0.962 \\ -0.0494 & 1.013 & 0.0367 \end{bmatrix} \quad (54)$$

$$PI(s) = \begin{bmatrix} k_1 + \frac{\beta_1}{s} & 0 & 0 \\ 0 & 0 & k_3 + \frac{\beta_3}{s} \\ 0 & k_2 + \frac{\beta_2}{s} & 0 \end{bmatrix} \quad (55)$$

4.2 Decentralized Tuning

For the three controllers of the MIMO complex air-path, tuning can become quite complicated and tedious without a systematic design approach. Therefore in decentralized control it is often common practice to use the steady-state gain matrix of G directly in designing the PI controller parameters. In this case the normalized plant, $\hat{G}$, of complex air-path was used to set PI parameters which invokes additional clarity when tuning by overcoming ambiguities due to signal units. This is why the tracking error and controller signals of Figure 30 are respectively scaled by the diagonal matrices D_y^{-1} and D_u .

Examining the steady-state normalized gain of $\hat{G}$ in Equation (56) shows that the HPEGR mass flow rate has the strongest coupled interaction for all intake manifold conditions. VGT and LPEGR mass flow rate have little effect on intake temperature, obviously because of both the intercooler and LPEGR cooler. As expected the VGT has the prominent affect on pressure. Concern over the coupled interaction comes into play

with the intake oxygen mass fraction where the HPEGR mass flow rate actually poses a higher, normalized, steady-state gain than the LPEGR which is going to control this state. Of course this is only relevant when considering the normalized plant. The steady-state gains of the EGR mass flow rates are equal in the identified model, G , also shown in Equations (9) through (11), but considering that the LPEGR operates near its saturation points, it is necessary to design the controller around the normalized plant that is scaled by the matrix of the open-loop excitation amplitudes, D_u .

$$\hat{G}(0) = \begin{bmatrix} 1.568 & 0.045 & .0638 \\ 0.297 & 0.177 & -1.171 \\ -1.023 & -0.7219 & -0.183 \end{bmatrix} \quad (56)$$

By following the decentralized design procedure given in [36], the LPEGR controller must be faster than the HPEGR controller because of the strong coupled interaction between the two actuators. This gives the constraint of Equation (57) for the components of $\hat{G}(0)$ which was followed in controller design. Because the constraint is based on the HPEGR controller the parameters, k_I and β_1 were designed first in an effort to match or outperform the MIMO feedback controller in the previous chapter. With the set HPEGR parameters, the LPEGR controller was then designed based on the constraint of Equation (57). The VGT PI parameters were then designed in the same effort to outperform the MIMO feedback controller.

$$k_1 + \frac{\beta_1}{s} \leq \frac{\hat{G}_{31}}{\hat{G}_{32}} \left[k_2 + \frac{\beta_2}{s} \right] \quad (57)$$

4.3 Decentralized Controller Validation and Comparison

The validation of the decentralized controller design can be seen in Figure 31 as compared to the MIMO feedback controller designed in section 3.2. Shown here is the best performing HPEGR decentralized controller of intake temperature that was attained with a thorough tuning of k_I and β_1 . It was found that any additional proportional or integral gain on this component of the decentralized controller would cause instability. Figure 31 shows that the robust MIMO feedback control has a faster response and settling time in regards to intake temperature. Only a subtle difference can be seen when comparing the ability of the controllers to maintain desired intake pressure. This would imply that feedback control of intake pressure with the VGT can possibly be withdrawn from the MIMO control scheme to work in parallel with a MIMO EGR controller with negligible changes in performance. This is supported through examination of $\hat{G}(0)$ in Equation (56) which shows little consideration is needed when designing a decentralized VGT controller. Only a slight advantage to using the VGT in the MIMO control scheme can be seen in the comparison of actuator signals in Figure 32 where the MIMO VGT output has better noise attenuation qualities.

The MIMO controller also outperforms the decentralized controller in maintaining the desired oxygen mass fraction with a reduced settling time and better disturbance rejection. An interesting comparison of the LPEGR mass flow rate commands between the two controllers can be seen in Figure 32 at the time of 4 and 10 seconds. At these two points the LPEGR signal of the two different controllers move in opposite directions for the same simulation. This can be attributed to the H_∞ formulation which incorporates the identified dynamics of the complex air-path in controller design

and is possibly the reason for the faster settling time of the MIMO controller after these points. Stability of the decentralized controller also becomes questionable between the times of 7 and 10 seconds where the intake oxygen mass fraction continually oscillates. This occurs because of the previously discussed directional dependency of the complex air-path to the controller signal. During this time period the HPEGR mass flow rate is near its maximum and LPEGR flow is near its minimum, very close to the controller input direction associated with the maximum singular value in Figure 14. This would imply that the increased system gain at this direction causes the decentralized controller to reach its stability margin and highlights a major advantage when using MIMO control.

Figure 31. Decentralized controller validation with comparison to robust MIMO feedback control

Figure 32. Decentralized controller signals during validation with comparison to robust MIMO feedback control

CHAPTER 5: FUTURE WORK AND CONCLUDING REMARKS

5.1 Future Work

In the near future an experimental validation of the MIMO control scheme would further support the multivariable development strategy. Also, without a complete experimental characterization of the operating range of advanced combustion modes seen through strict control of intake conditions, only qualitative benefits can be addressed. Additionally the GT-Power model used in both system identification and controller validation only operates in conventional CIDI. It would be interesting to see how operating in any of the discussed advanced combustion modes would affect the dynamics of the complex air-path and controller performance, which may further support the use of advanced MIMO control techniques. A key factor would be how drastic the temperature reduction is in the exhaust manifold due to advanced combustion mode operation.

Slight performance gains were seen with the coupling of a feed-forward controller with the robust MIMO feedback controller. It is possible that if inaccuracies in the feed-forward controller were reduced with more accurate estimations of heat transfer and turbocharger efficiency parameters, a higher performance gain could be seen with the coupling of the feed-forward controller with the MIMO feedback. Through the development of the decentralized controller and with its comparison to the robust MIMO

controller it was observed that the VGT control has subtle benefits when integrated into the multivariable control strategy. It may be possible to design a multivariable controller around just the intake temperature and oxygen mass fraction outputs using only the dual-loop EGR. Development of such a controller would include a system identification using only two inputs and outputs which would result in a more accurate identification of the complex air-path system dynamics and could potentially lead to a better performing MIMO EGR controller. A slight disadvantage to this approach would be that the VGT would act as an additional disturbance to the MIMO EGR controller.

Better design of hardware to improve the complex air-path response characteristics would also allow for faster response times of the closed-loop system. In this work a GT-Power engine model was used of an actual medium-duty diesel engine that operates in the standard CIDI mode. Design of air-path components with smaller volumes, to hasten filling transients, and reduced turbo impeller inertias would better underline the goal of minimizing the response times of intake conditions crucial to controlling advanced combustion. Another interesting approach to the hardware design of a complex air-path would be to use two HPEGR loops, one with an EGR cooler and one without, which would allow for a faster control of the oxygen mass fraction. One major challenge involved with this idea is controlling the HPEGR mass flow rate of two loops may be difficult as both HPEGR valves would have the same upstream and downstream pressures.

5.2 Concluding Remarks

The sensitivity of the discussed advanced combustion modes makes even the slightest gains in response and settling times crucial to emissions output, stability, and

engine performance. This is why such advanced control techniques may need to be implemented for advanced combustion modes operating with a complex air-path.

Controller validations were performed using a GT-Power model which simulates one-dimensional wave propagations, variable transport delay, heat transfer, and many other characteristics similar to an actual diesel engine equipped with the complex air-path. The use of the H_∞ controller formulation incorporated both the identified dynamics and the actuator directional dependency to develop a MIMO controller which outperformed the basic decentralized control approach. The foundation for the controller formulation was an identified linear model of the complex-air path consisting of HPEGR, LPEGR, and VGT actuators. The DK-iteration method allowed for a robust MIMO controller to be developed around dynamic uncertainty between the actual GT-Power model and the LTI representation, mostly caused by non-linearity, unidentified dynamics, and variable transport delay. This development method could be used to extend MIMO controller formulation around unidentified dynamics of actual diesel engine. The use of an advanced multivariable control strategy showed to develop a better performing controller for the complex air-path of a medium-duty, V8, diesel engine as seen in the comparison between the robust MIMO and decentralized feedback controllers. This opens the possibilities for the implementation advanced combustion modes using the indirect combustion control approach of maintaining a high control authority over intake conditions and would only require the use of basic hardware like dual-loop EGR and two-stage VGT seen in the complex air-path.

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APPENDIX

A.1 Graphical User Interface of GT-Power Engine Model

A.2 Modeling Uncertainty

The weighted transfer function matrix, W_u which represents the upper bounds of uncertainty is modeled in Equation (58). The components of the matrix are the weighted transfer functions of Figure 33 through Figure 37.

$$W_u = \begin{bmatrix} 0 & 0 & 0 \\ W_{HPEGR-P} & W_{LPEGR-P} & 0 \\ W_{HPEGR-F} & W_{LPEGR-F} & W_{VGT-F} \end{bmatrix} \quad (58)$$

Figure 33. Weighting of uncertainty between HPEGR actuation and intake pressure

Figure 34. Weighting of uncertainty between LPEGR actuation and intake pressure

Figure 35. Weighting of uncertainty between HPEGR actuation and intake oxygen mass fraction

Figure 36. Weighting of uncertainty between LPEGR actuation and intake oxygen mass fraction

Figure 37. Weighting of uncertainty between VGT actuation and intake oxygen mass fraction