

Institutionen för systemteknik

Department of Electrical Engineering

Examensarbete

Modeling, Simulation and Control of Long and Short Route EGR in SI Engines

Examensarbete utfört i Fordonssystem
vid Tekniska högskolan vid Linköpings universitet
av

Junting Qiu

LiTH-ISY-EX-15/4870-SE

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Linköpings universitet
TEKNISKA HÖGSKOLAN

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Författare Author	Junting Qiu		
Sammanfattning Abstract	<p>Modern engines are faced with increasingly stringent requirements for reduced fuel consumption and lower emissions. A technique which can partly be used to reduce emissions of nitrogen oxides is recirculation of combusted gases (Exhaust Gas Recirculation, EGR). In gasoline engines, it also has the advantage that it can save fuel by reducing pumping losses. To large mixture of EGR in the air to the cylinders will however affect the combustion stability negatively. To investigate EGR rate and dynamics with respect to different actuator inputs, the thesis develops an engine model that includes EGR. The model focus on the air flow in the engine and extends an existing mean value engine model. Two types of EGR-system are investigated. They are short-route EGR which is implemented between intake manifold and exhaust manifold and long-route EGR which is implemented between compressor and turbine. The work provides a simulation study that compares both stationary and transient properties of the two EGR-systems, such as fuel consumption, maximum EGR, and rise time with respect to different actuators.</p>		
Nyckelord Keywords	Mean value engine modeling, EGR, long route, short route, residual gas fraction, PID		

Abstract

Modern engines are faced with increasingly stringent requirements for reduced fuel consumption and lower emissions. A technique which can partly be used to reduce emissions of nitrogen oxides is recirculation of combusted gases (Exhaust Gas Recirculation, EGR). In gasoline engines, it also has the advantage that it can save fuel by reducing pumping losses. To large mixture of EGR in the air to the cylinders will however affect the combustion stability negatively. To investigate EGR rate and dynamics with respect to different actuator inputs, the thesis develops an engine model that includes EGR. The model focus on the air flow in the engine and extends an existing mean value engine model. Two types of EGR-system are investigated. They are short-route EGR which is implemented between intake manifold and exhaust manifold and long-route EGR which is implemented between compressor and turbine. The work provides a simulation study that compares both stationary and transient properties of the two EGR-systems, such as fuel consumption, maximum EGR, and rise time with respect to different actuators.

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Notation

NOTATION

Notation(Unit)	Description
$p_{us}(Pa)$	Pressure upstream
$p_{ds}(Pa)$	Pressure downstream
$p_{im}(Pa)$	Intake manifold pressure
$p_{em}(-)$	Exhaust manifold pressure
$H_r(\frac{Pa^2s^2}{m^2K})$	Flow restriction resistance
$\dot{m}(kg/s)$	Mass flow through restriction
$m_{tot}(kg)$	Total mass
$m_r(kg)$	Residual gas mass
$u_{th}(-)$	Throttle valve control signal
$u_{egr}(-)$	EGR valve control signal
$u_{wg}(-)$	Waste-gate control signal
$T_{im}(K)$	Intake manifold temperature
$T_{us}(K)$	Temperature upstream
$T_{ds}(K)$	Temperature downstream
$\chi_r(-)$	Residual gas fraction

ABBREVIATIONS

Förkortning	Betydelse
MVEM	Mean value engine models library
PID	Proportional, integral, differential (regulator)
EGR	Exhaust gas recirculation
TCSI	Turbocharge gasoline
VEA	Volvo engine architecture
ECU	Engine control unit

1

Introduction

In this chapter, the topic of the thesis is introduced. Based on that information, the problem is formulated followed by limitations, available resources and the method for addressing the problem.

1.1 Background

Modern engines are faced with increasingly stringent requirements for reduced fuel consumption and lower emissions. A technique which can partly be used to reduce emissions of nitrogen oxides is Exhaust Gas Recirculation, EGR. In gasoline engines it also has the advantage that it can save fuel by reducing pumping losses. However, large mixture of EGR in the air to the cylinders affects the combustion stability negatively. Therefore, it is important to have accurate control of the amount of exhaust gas recirculated. EGR is not currently implemented largely on modern gasoline engines, but with more stringent requirements may be necessary.

According to the considerations of EGR foundation, EGR has its advantage of that helps the reduction of emission due to that EGR increases the temperature of the mixture gas and accelerates the mixing of gasoline and air as well as evaporation. It can also improve combustion efficiency and reduce fuel consumption [Olsson et al., 2003] [Yokomura et al., 2003]. Consequently, the implementation of EGR can give an important advantage.

In Automotive Systems a new engine test cell VEA (Volvo Engine Architecture) engine from Volvo Cars new engine family has recently been installed. The plan to equip this for future research with a valve for recirculation of exhaust gases now are carried on. Therefore, it is now interesting to examine how control of different actuators will affect the amount of EGR in the air to the cylinders.

1.2 Problem formulation and goal

The goal of this thesis is to develop an existing simulation environment for the air flow in a gasoline engine. The existing model is a Mean Value Engine Model but handles only air or exhaust gases, not a mixture. The work will extend this model to include EGR and develop appropriate models for EGR valves. A new state which is residual gas fraction is implemented. The residual gas is defined as recirculated gas in this thesis. The work should also investigate how the step response of actuators affects the EGR amount. This will give insight to the possible transient performance of the EGR control. Two different EGR routes, which are long-route EGR and short-route EGR, should be considered. Long route has the EGR-system between turbine and compressor and short route has the EGR-system between intake and exhaust manifold.

- Expand existing MVEM to include EGR as well as residual gas fraction state through all system.
- Develop a model for the EGR valve by modification in compressible restriction block in MVEM_lib and also use multi-port receiver (control volume) instead.
- Examine how the step response of actuators affects the EGR amount.
- Develop appropriate control strategy and ECU to follow set point in the EGR amount.
- Comparison between long and short route EGR.
- Examine the relation between engine speed, engine torque and maximum EGR fraction.

1.3 Limitations

It is assumed that the EGR fraction or Oxygen level are measured or available by an observer. For example, the task of estimating the Oxygen level in the intake will not be considered. The controllers studies in this thesis are limited according to the structure shown in figure 1.1 on page 3. This work is limited to a simulation study, to gain knowledge before a future installation. Therefore measurements will not be available.

1.4 Resources

Since the limitation on measurements, it is important to be able to do simulations and analyze different performance before testing in reality. There are several types of engine model libraries with different design complexities. In this thesis work, Mean Value Engine Models (MVEM) which is favorable for design

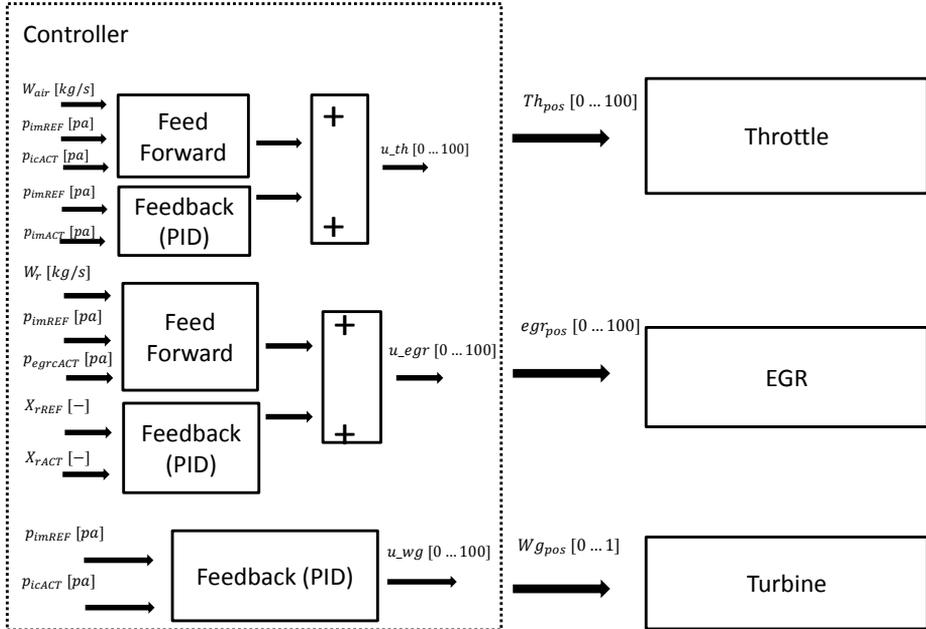


Figure 1.1: An overview of the control system used in this thesis. The throttle valve, EGR valve and wastegate are controlled by a software boost controller implemented in the ECU. Abbreviations in figure: Th_{pos} (throttle valve position), egr_{pos} (EGR valve position), Wg_{pos} (wastegate position), u_{th} (throttle valve control signal), u_{egr} (EGR valve control signal), u_{wg} (wastegate valve control signal), W_{air} (air mass flow), p_{imREF} (intake manifold pressure reference), p_{icACT} (intercooler actual value), p_{imACT} (intake manifold pressure actual value), W_r (residual gas mass flow), $p_{egrcACT}$ (EGR-cooler actual value), X_{rREF} (residual gas fraction reference), X_{rACT} (residual gas fraction actual value)

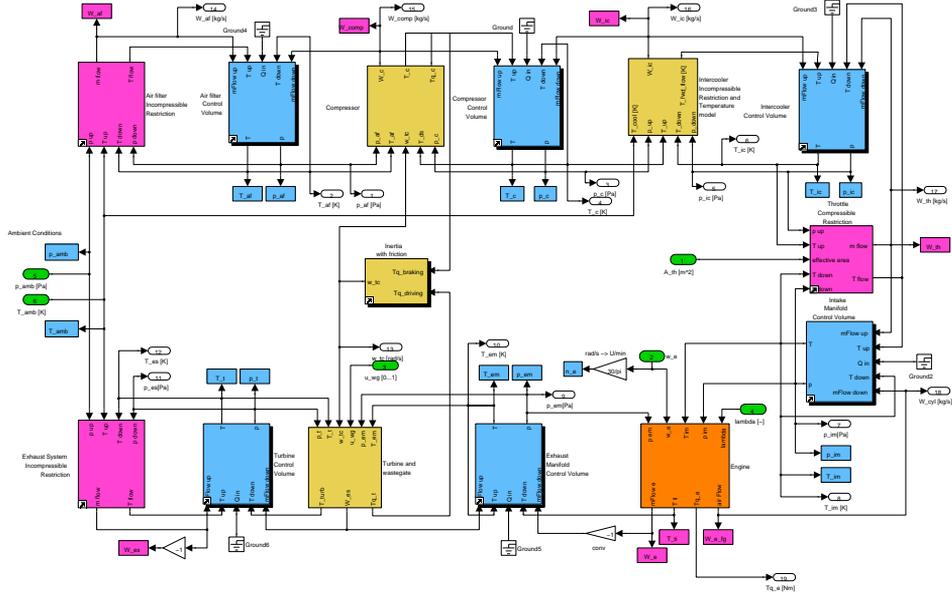


Figure 1.2: Modeling of Turbocharged Engines with MVEM_lib example.

of control and supervision system is mainly used and expanded. MVEM_lib has been designed to be flexible and reusable for both naturally aspirated and turbocharged engines [Eriksson]. All signals are mean value during each cycle. The turbocharged spark ignited engine example which is presented in the figure 1.2 on page 4. For the details of the library components see [Per Andersson], [Eriksson et al., 2002], [Eriksson, 2007] and [Eriksson and Nielsen, 2014]. This engine model is implemented in Matlab/Simulink.

1.5 Method and outline

The method in this thesis is first to expand the existing TCSI engine with residual gas fraction. Then step responses in the control signals will be done to investigate the transient response, in order to tune the controllers in the ECU. When the ECU is done including tuning parameters, step response will be carried on again to determine the performance of the ECU. The strategy utilized is three controllers, the throttle controller with feedforward and feedback, the wastegate controller with feedback and EGR controller with feedforward and feedback. All feedback is implemented as PID-controller with tracking to prevent wind-up. After those above, case studies should be started. The case studies will be range of recirculated gas determination, torque performance based on same fuel consumption and fuel consumption performance with respect to same torque generation. When the Simulink model and plot analysis are done for both long and short

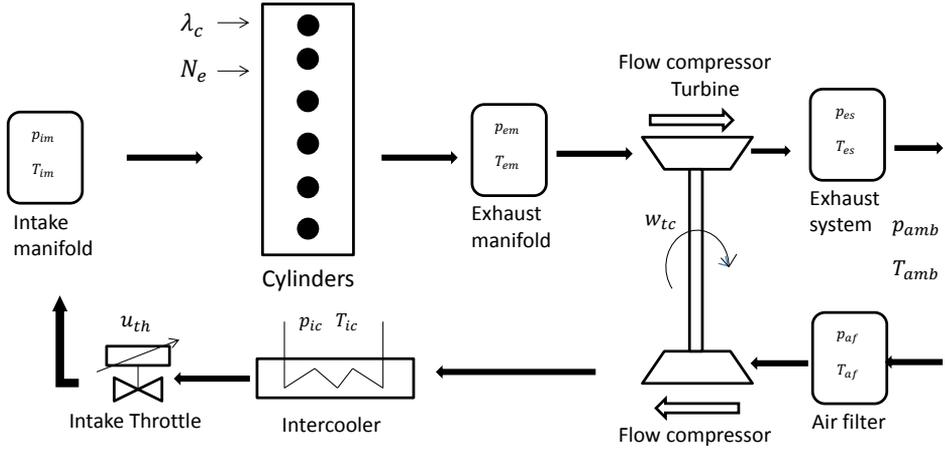


Figure 1.3: Schematic illustration of the existing model.

routes, comparison will be continued and then 2D engine maps which shows the relation between torque, engine speed and residual gas fraction will be done in the end, see figure 4.6.

Chapter 1 describes the background, purpose, limitation, foundation of this thesis. Chapter 2 presents the overview of the EGR model for simulation purpose with step response performance for both short and long route EGR-systems. Chapter 3 demonstrates the behavior of the controllers and discusses the method for operation and parameter tuning. Cases study is done to provide the foundation of EGR. Chapter 4 shows the TCSI engine control result for both long and short routes. 2D and 3D interaction is proved by plotting. During this thesis, many tasks are identified but still need more attention and improvement that is left as future work in chapter 5. Finally, summary and conclusions are presented in chapter 6.

2

EGR model for simulation purpose

In this chapter, the existing MVEM library is expanded. The model is based on turbocharged engines with EGR system including EGR valve, EGR volume control and EGR cooler. Furthermore, the residual gas fraction state is developed. There are two cases for development of EGR which are long route (EGR-system between compressor and turbine) and short route (EGR-system between the intake and exhaust manifolds). The performance will be presented in this chapter.

2.1 Residual gas fraction state

According to the structure which was presented in figure 2.1 on page 8, the dynamic element with two states which are temperature and pressure had been modified with adding one more state which is residual gas fraction χ_r . The variables are determined as follows for intake and exhaust manifold as follows:

$$\dot{m} = f(\dot{m}_i) \quad (2.1)$$

$$\dot{T} = f(\dot{m}_i, T_i) \quad (2.2)$$

$$\dot{\chi}_r = f(\dot{m}_i, \dot{\chi}_{ri}) \quad (2.3)$$

Where $i = 1, 2, 3, \dots, n$ and $n = \text{number of connections}$.

In the receivers with only two connections (e.g. turbine control volume, air filter control volume and intercooler control volume) see figure 2.2a on page 9, i is equal to two, however, in the receivers with three connections (e.g. intake manifold and exhaust manifold) see figure 2.2b on page 9, i is equal to three. Therefore, it is essential to use a multi-port receiver as a control volume. In this thesis, the direction of the mass flow is determined by the sign of the value. Positive value is in and negative value is out. the control volume is marked with

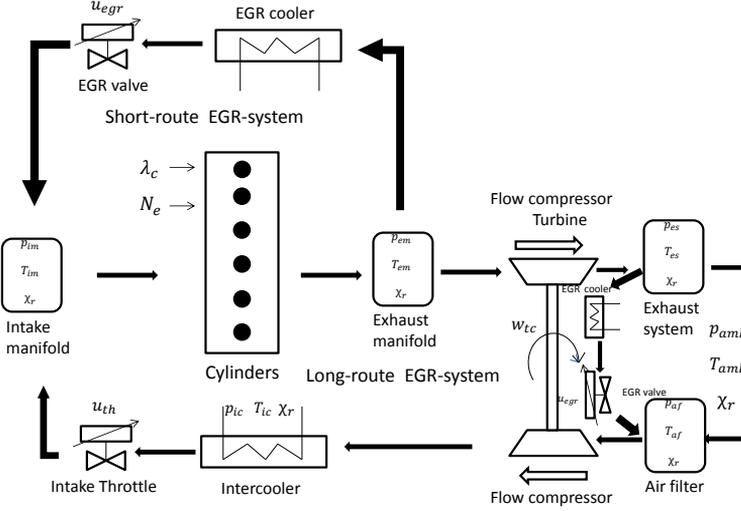


Figure 2.1: Schematic illustration of the modified model with long-route and short-route EGR systems.

dashed lines and the pressure, temperature, mass and new residual gas fraction states are shown in it with the assumption that the volume is constant.

With respect to the mass and energy conservation method, the residual gas fraction balance gives the following time derivative for the residual gas fraction in the receivers:

$$\frac{dm_r}{dt} = \sum_{i=1}^n (\dot{m}_i \chi_{ri}) \quad (2.4)$$

Furthermore, the time derivative can be remarked as follows:

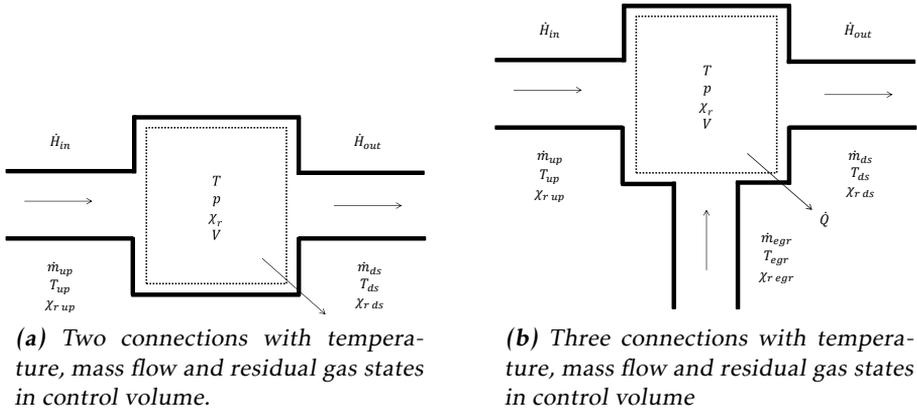
$$\frac{dm_{tot}}{dt} = \sum_{i=1}^n \dot{m}_i \quad (2.5)$$

Where $i = 1, 2, 3, \dots, n$ and n = number of connections. With respect to the residual gas fraction definition literally, the relation can be presented as follow:

$$\chi_r = \frac{m_r}{m_{tot}} = \frac{m_r}{m_{air} + m_r} \quad (2.6)$$

By doing the time derivative to the residual gas fraction vector, the dynamic element of the state can be induced as follows:

$$\frac{d\chi_r}{dt} = \frac{d}{dt} \left(\frac{m_r}{m_{tot}} \right) = \frac{\frac{dm_r}{dt} \times m_{tot} - \frac{dm_{tot}}{dt} \times m_r}{m_{tot}^2} \quad (2.7)$$



(a) Two connections with temperature, mass flow and residual gas states in control volume.

(b) Three connections with temperature, mass flow and residual gas states in control volume

Figure 2.2: The control volume is marked with dashed lines and the pressure, temperature, mass and new residual gas fraction states are shown in it with the assumption that the volume is constant. There are two flows across the boundaries which are upstream and downstream in a) and three flows across the boundaries which are midstream, upstream and downstream. The direction is defined by sign of the vector

Combining expression above as well as $\dot{m}_r = \chi_r \times \dot{m}_{tot}$:

$$\frac{d\chi_r}{dt} = \frac{d}{dx} \left(\frac{\sum_{i=1}^n (m_i \chi_{ri}) \times m_{tot} - \sum_{i=1}^n \dot{m}_i \times m_{tot} \times \chi_r}{m_{tot}^2} \right) = \frac{d}{dt} \left(\frac{\sum_{i=1}^n (m_i \dot{\chi}_{ri}) - \sum_{i=1}^n \dot{m}_i \times \dot{\chi}_{ri}}{m_{tot}} \right) \quad (2.8)$$

2.2 Engine with burned gas mass flow

Since the residual gas fraction is known, so the recirculated gas mass flow and the air mass flow can be determined as follows:

$$\begin{pmatrix} \dot{m}_{air} \\ \dot{m}_r \end{pmatrix} = \begin{pmatrix} 1 - \chi_r \\ \chi_r \end{pmatrix} \times \dot{m}_{tot} \quad (2.9)$$

Based on the same total mass flow, the air mass flow going through the engine block will be reduced due to the amount occupied by the recirculated gas, so that the torque of engine is decreased. In the engine block, an adiabatic mixer is implemented in order to mix the exhaust mass flow with engine out temperature and recirculated gas mass flow with the intake temperature. In the end, the residual gas fraction becomes one since lambda equal to one is assumed in the model.

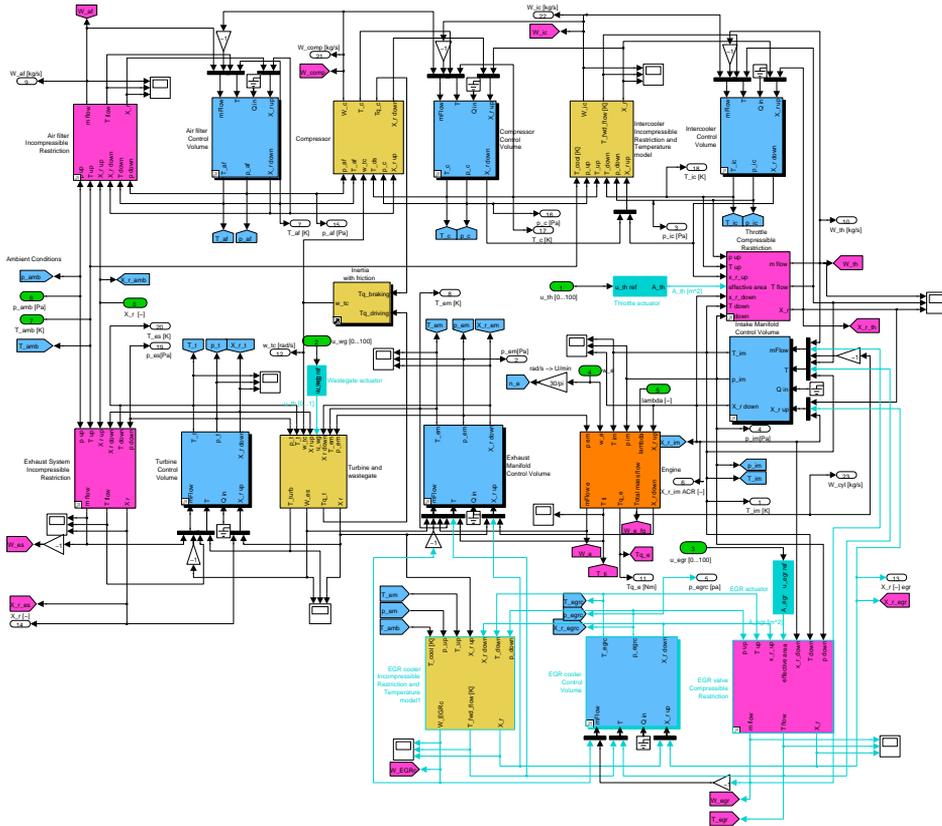


Figure 2.3: Modeling of Turbocharged Engines with short-route EGR implemented in *MVEM_lib*.

2.3 Modeling of EGR in Short route and long route

EGR-system in this thesis consists of three components which are one EGR-cooler, one control volume and one EGR valve. They are selected from *MVEM_lib* and the valve is modeled as a compressible restriction.

The short-route EGR in this work is defined as an EGR-system implemented between intake and exhaust manifolds in the figure 2.3 on page 10.

The long-route EGR in this work is defined as an EGR-system implemented between air filter and turbine control volumes in the figure 2.4 on page 11.

The speculation of the short and long route is that they should both have close performance of controlling the TCSI engine, but long-route EGR will have more time-consuming to achieve the desired value due to more control volume between it comparing with short-route EGR.

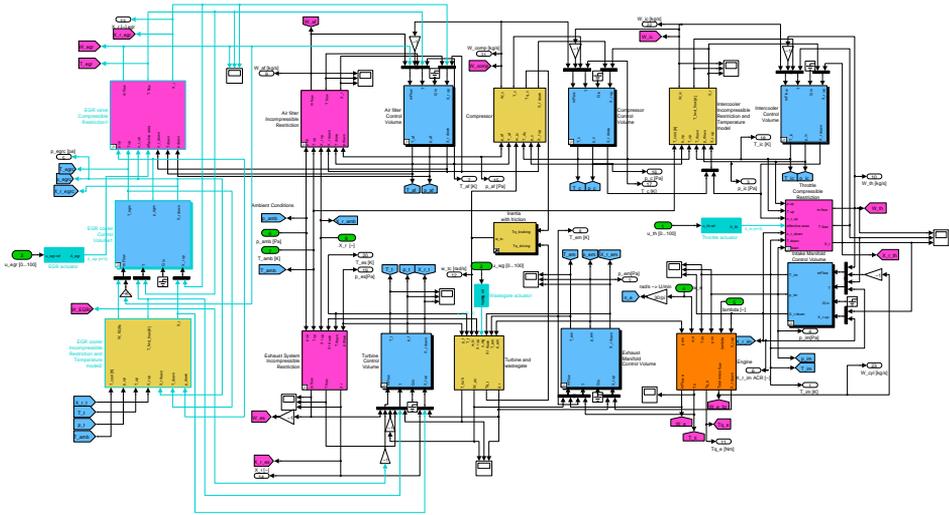


Figure 2.4: Modeling of Turbocharged Engines with long-route EGR implemented in MVEM_lib.

2.4 Step response with MVEM_lib

In this section investigation of the transient performance of the EGR loop using step response are presented.

2.4.1 Method

The method of step response is to operate different set points in the simulation. The nine points used in this section for short-route EGR are: Engine speed is equal to 1000 rpm, 2000 rpm and 4000 rpm according to intake manifold pressure equal to 50 kpa, 100 kpa and 150 kpa in each. For long-route EGR, engine speed is replaced by 2000 rpm, 3000 rpm and 4000 rpm instead. The distribution of the points is shown in figure 2.5 on the page 12.

The method of tuning parameters is to close EGR valve in the beginning and then tune the control signal values to meet the desired intake manifold pressure. Then to do step response for another three cases which are EGR valve 10% open, half open and total open, see table 2.1 on the page 12 and table 2.2 on the page 13. Those tables show the adapted engine with two different EGR-systems have normal performance of throttle valve, EGR valve and wastegate interaction to the intake manifold pressure. The intake manifold pressure increases with throttle control signal increasing, wastegate control signal decreasing when EGR is close. It is better to close wastegate for achieving a lower intake manifold pressure.

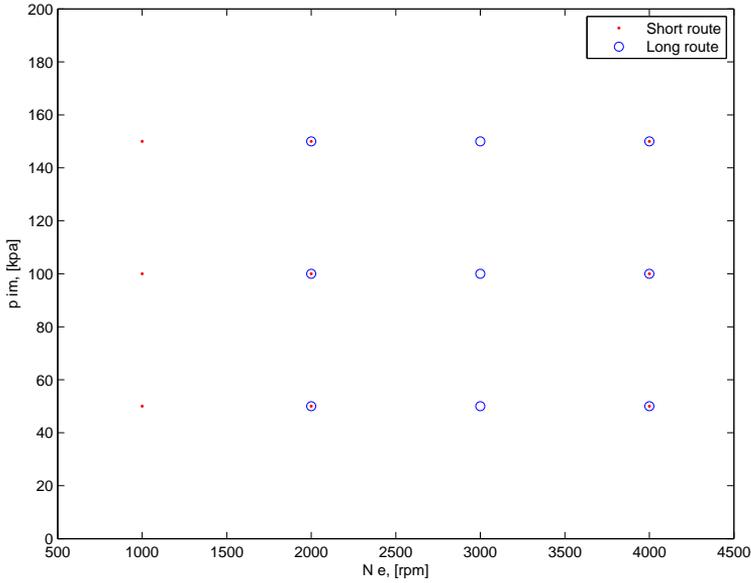


Figure 2.5: Operating points distribution for step response. The difference of choice is due to the order of the process in thesis and the two performances are checked independently. It will not affect the performance since the comparison will be done in same operating points.

n_e	p_imREF	u_th	u_wg	u_egr	p_imACT	correct					p_imACT
1000	1.50E+05	100	0	0	1.06E+05	x	can't reach				
	1.00E+05	10.95	0	0	1.00E+05	OK	x	x	x	x	x
	5.00E+04	0.6325	1	0	5.00E+04	OK	x	x	x	x	x
n_e	p_imREF	u_th	u_wg	u_egr	p_imACT		u_th	u_wg	u_egr	p_imACT	
2000	1.50E+05	100	0.01528	0	1.50E+05	OK	x	x	x	x	x
	1.00E+05	100	1	0	9.96E+04	check	9.99	0	0	1.00E+05	
	5.00E+04	10.18	1	0	8.33E+04	check	2.92	0	0	5.00E+04	
	5.00E+04	10.17	1	0	4.23E+04	check	x	x	x	x	x
n_e	p_imREF	u_th	u_wg	u_egr	p_imACT		u_th	u_wg	u_egr	p_imACT	
4000	1.50E+05	100	0.14735	0	1.50E+05	OK	x	x	x	x	x
	1.00E+05	100	1	0	9.79E+04	check	10.072	0	0	1.00E+05	
	5.00E+04	8.02	1	0	5.00E+04	OK	x	x	x	x	x

Table 2.1: Tuning parameters table for short-route EGR step response. The intake manifold pressure increases with throttle control signal increasing, wastegate control signal decreasing when EGR is close. It is better to close wastegate for achieving a lower intake manifold pressure.

n_e	p_imREF	u_th	u_wg	u_egr	p_imACT	correct
2000	1.50E+05	100	1.7	0	1.50E+05	OK
	1.00E+05	100	84	0	1.00E+05	OK
	5.00E+04	3.19	0	0	5.00E+04	OK
n_e	p_imREF	u_th	u_wg	u_egr	p_imACT	correct
3000	1.50E+05	100	16.53	0	1.50E+05	OK
	1.00E+05	100	79	0	1.00E+05	OK
	5.00E+04	5.06	0	0	5.00E+04	OK
n_e	p_imREF	u_th	u_wg	u_egr	p_imACT	correct
4000	1.50E+05	100	14.3	0	1.50E+05	OK
	1.00E+05	100	77.9	0	1.00E+05	OK
	5.00E+04	6.5	0	0	5.00E+04	OK

Table 2.2: Tuning parameters table for long-route EGR step response. The intake manifold pressure increases with throttle control signal increasing, wastegate control signal decreasing when EGR is close. It is better to close wastegate for achieving a lower intake manifold pressure.

2.4.2 Step response performance

For model identification several step responses in TCSI engine with EGR-system were made. These were done with three cases respectively, EGR valve total open, half open and 10% open based on the first step which is to meet the desired intake manifold with EGR valve close. The control signal values according to those tables shown previously. Two examples are shown in figure 2.6 on the page 14, figure 2.7 on the page 15, figure 2.8 on the page 16 and figure 2.9 on the page 17. The rest can be found in Appendix A. The step performance which is selected as one example based on the operating point, engine speed is equal to 2000 rpm, intake manifold pressure is equal to 100 kpa and EGR is 10% open. See figure 2.10.

Throughout this section these simulation results are treated as dynamic. The step response shows that EGR control signal increases with engine torque decreasing, recirculated gas mass flow increasing and exhaust manifold pressure decreasing. The performance of the step response is normal.

2.5 EGR model summary

The model was developed in the available MVEM. In this section, regarding the step response performance shows reasonable trend in TCSI engine with different situations of EGR. The basic TCSI engine has normal behavior according to step

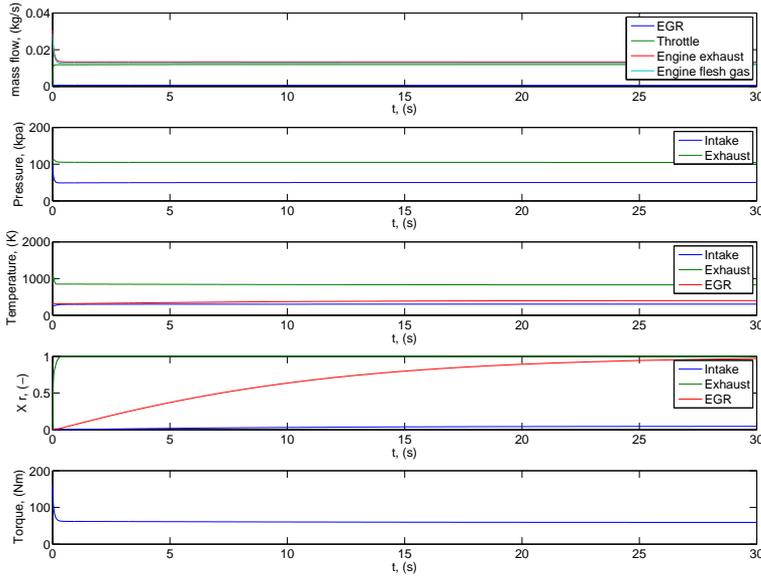
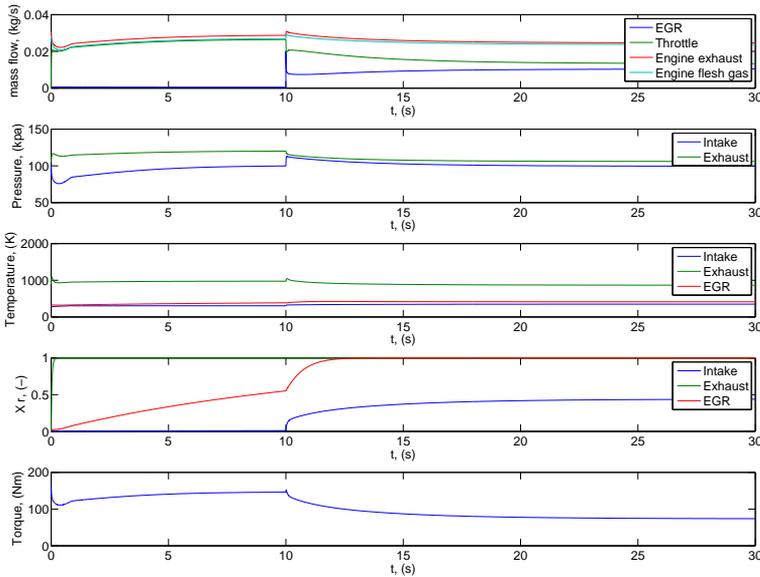
(a) $u_{egr}=0$ for short-route EGR(b) $u_{egr}=10$ for short-route EGR

Figure 2.6: Step response for engine speed equal to 2000 rpm and intake manifold pressure equal to 100 kpa in different egr valve positions for short-route EGR. The χ_r for EGR is in the pipe between intake manifold and exhaust manifold. The EGR valve opens which leads to that the engine torque decreases, recirculated gas mass flow increases and exhaust manifold pressure decreases.

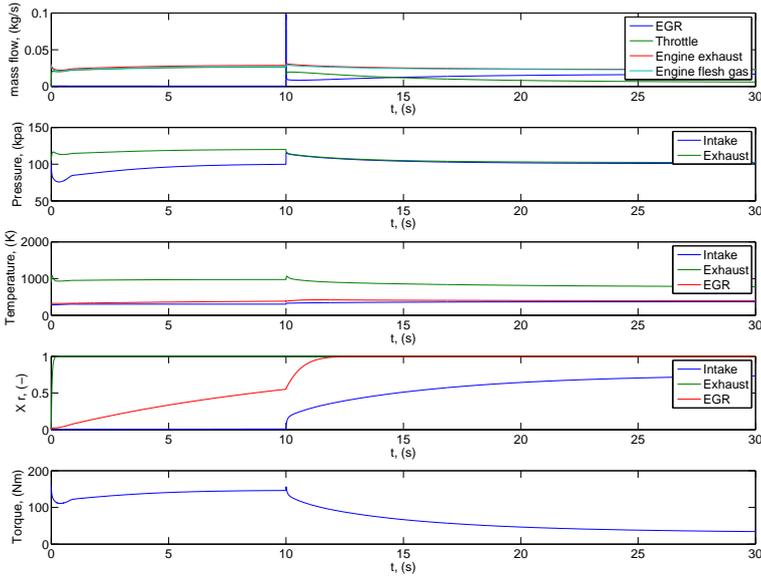
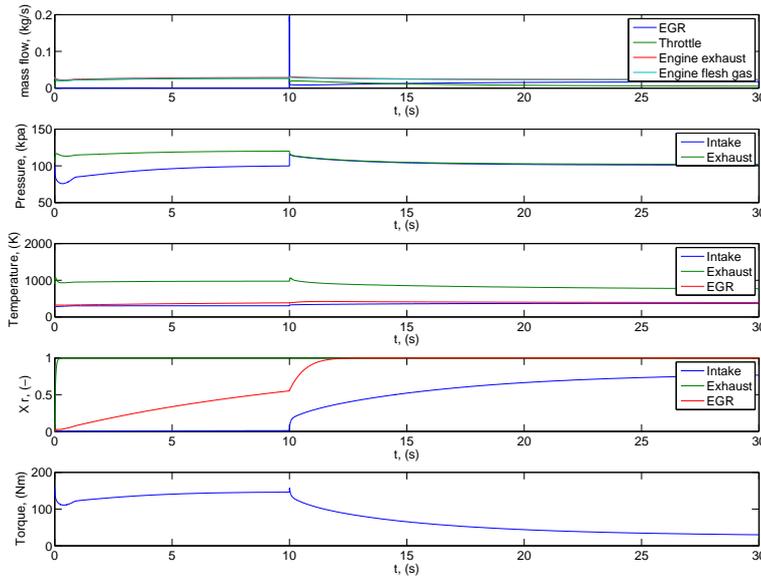
(a) $u_{egr}=50$ for short-route EGR(b) $u_{egr}=100$ for short-route EGR

Figure 2.7: Step response for engine speed equal to 2000 rpm and intake manifold pressure equal to 100 kpa in different egr valve positions for short-route EGR. The χ_r for EGR is in the pipe between intake manifold and exhaust manifold. The EGR valve opens which leads to that the engine torque decreases, recirculated gas mass flow increases and exhaust manifold pressure decreases.

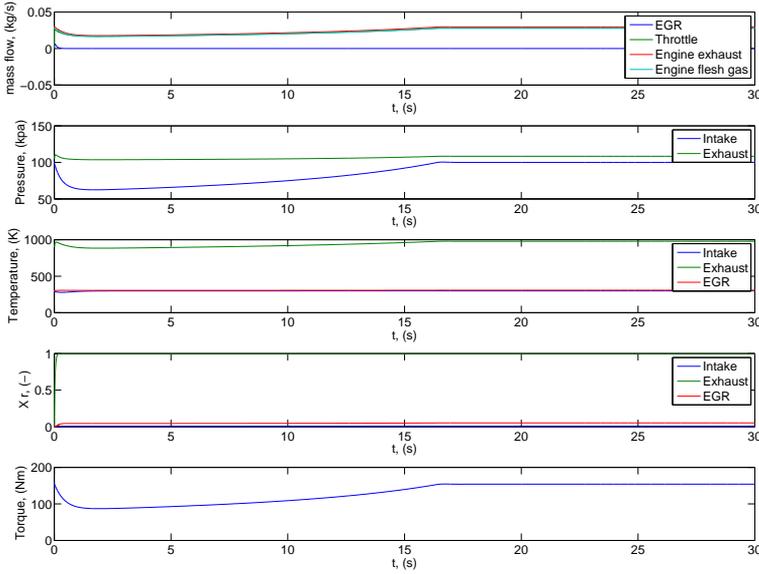
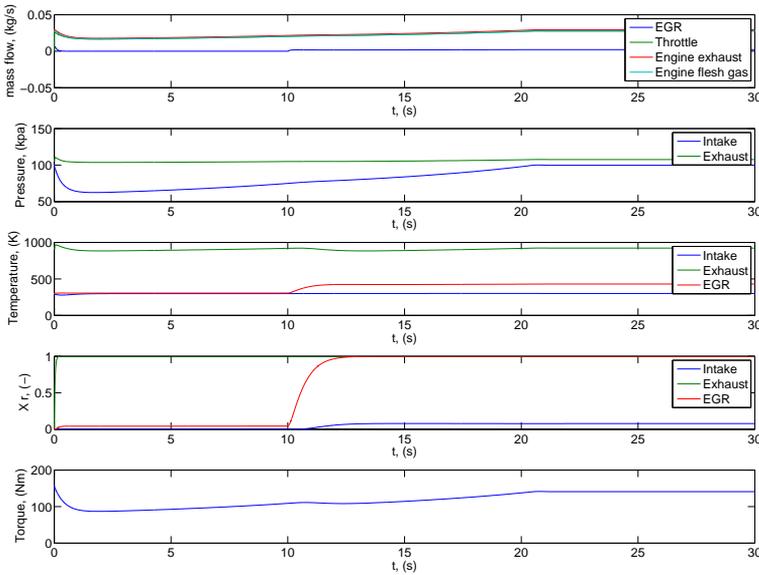
(a) $u_{egr}=0$ for long-route EGR(b) $u_{egr}=10$ for long-route EGR

Figure 2.8: Step response for engine speed equal to 2000 rpm and intake manifold pressure equal to 100 kpa in different egr valve positions for long-route EGR. The χ_r for EGR is in the pipe between intake manifold and exhaust manifold. The EGR valve opens which leads to that the engine torque decreases, recirculated gas mass flow increases and exhaust manifold pressure decreases.

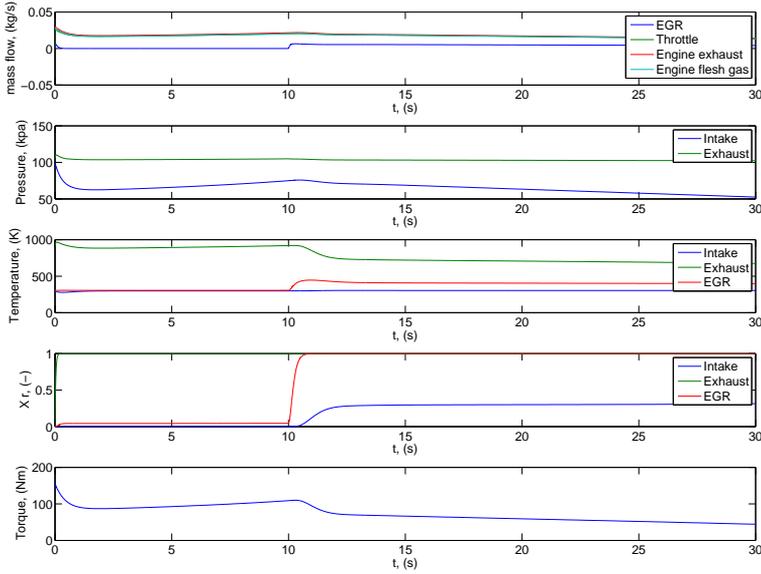
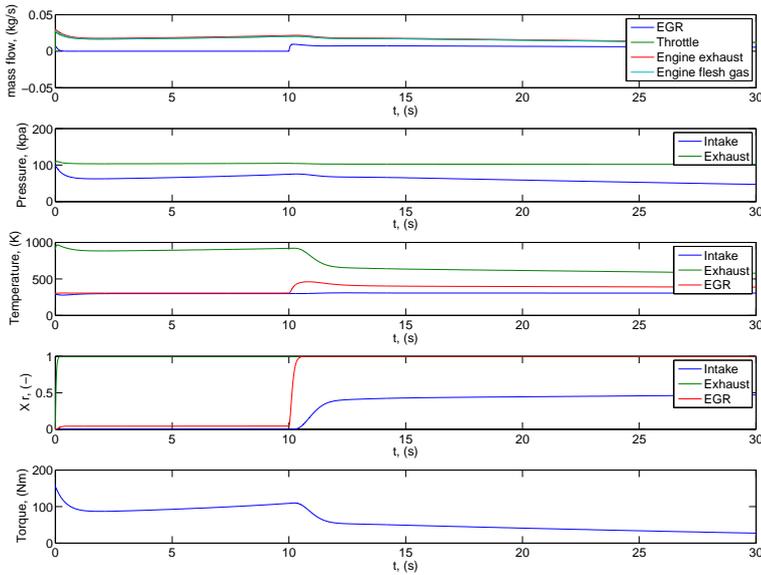
(a) $u_{egr}=50$ for long-route EGR(b) $u_{egr}=100$ for long-route EGR

Figure 2.9: Step response for engine speed equal to 2000 rpm and intake manifold pressure equal to 100 kpa in different egr valve positions for long-route EGR. The χ_r for EGR is in the pipe between intake manifold and exhaust manifold. The EGR valve opens which leads to that the engine torque decreases, recirculated gas mass flow increases and exhaust manifold pressure decreases.

responses of throttle and wastegate as well. Since the model is not tuned to measurement data, a few adjustments need to be done to achieve different cases in real environment. Since no measurements are available, it is hard to validate the quantitative behavior, but the parameter trend and relation between them during the step response can be discussed and compared. There are more step responses in the next section to evaluate behavior of the controllers. Both transient response and how well they follow the reference values.

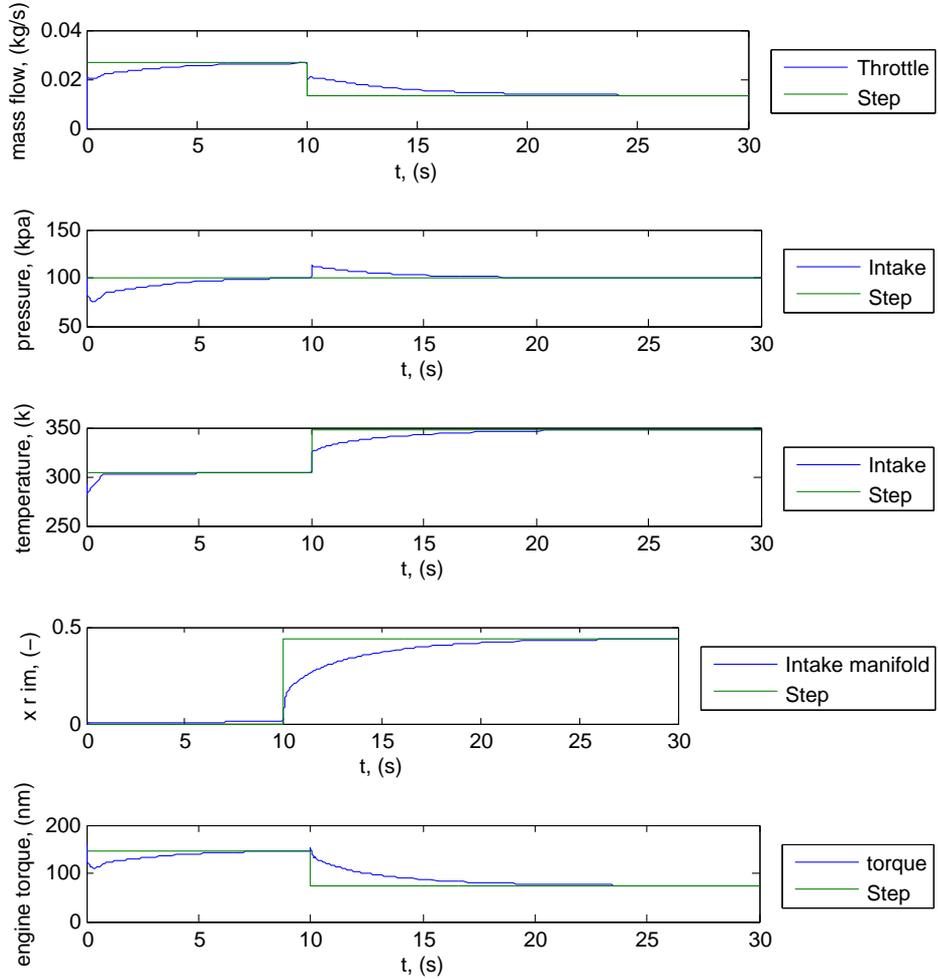


Figure 2.10: Based on step response, a clear step plot is presented comparing with the air mass flow, intake manifold and temperature, recirculated gas fraction in intake manifold and engine torque.

3

Controllers modeling and tuning

In this chapter, PID controllers for throttle, EGR and wastegate are presented. The controllers' structures are described in section 3.1. PID parameter tuning process is presented in the beginning of section 3.2. After those, cases study is made to check the level of the performance in EGR in section 3.3.

3.1 Controller structure

The controllers' standard structures were selected from TSFS09 modeling and control of engines and drivelines project material in Linköping University course, but the controllers are new since the residual gas fraction is implemented and simulated. The feedforward part of the controllers takes reference values of mass flow of the upstream and intake manifold pressure and actual value of the pressure before the controlled valve.

There are two reference value for feedforward part, intake manifold pressure reference and residual gas fraction reference. Intake manifold pressure reference can define the engine flow reference and residual gas fraction reference can define the air mass flow reference and residual gas mass flow reference from the engine flow generation. One difference between long and short routes is the mass flow reference going through throttle feedforward controller part. For short route, the mass flow reference is air mass flow since only air goes into throttle and for long route, the mass flow reference is total engine flow since the long-route EGR position is before throttle.

The feedback part of the controllers is developed by inputting the reference value and actual value of the variables which are required to be controlled. In this thesis for standard version, residual gas fraction reference is inputted into EGR valve controller, intake manifold pressure reference is inputted into throttle valve controller and intercooler pressure is inputted into wastegate controller.

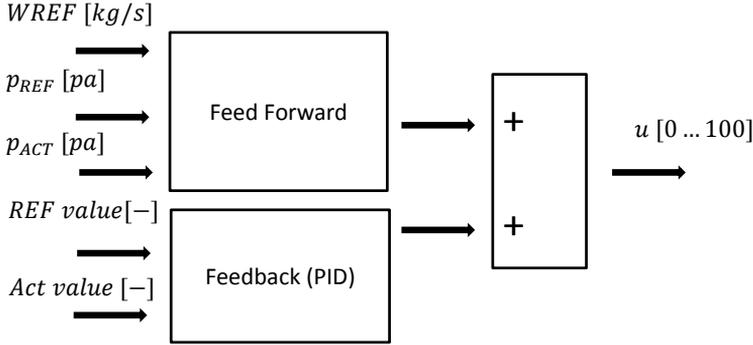


Figure 3.1: Standard controller overview structure in this thesis.

Additionally, another two versions will be presented as well. They are torque controller version and air mass flow controller version for cases study.

$$u(t) = K_p e(t) + K_i \int_0^t e(\tau) d\tau + K_d \frac{de(t)}{dt} \quad (3.1)$$

$$U(s) = (K_p + K_i \frac{1}{s} + K_d s) E(s) \quad (3.2)$$

Based on the description above, the result is static. The aim of developing PID controllers is to minimize the step time during step response simulation and to achieve desired values with less error. It plays a very important role on eliminating oscillations introduction. A standard overview is presented in the figure 3.6 on the page 22. The PID controller is developed in parallel form and the equation 3.1 describes the ideal controller logic as well as the Laplace domain in equation 3.2.

3.2 Tuning method for PID controller

3.2.1 Method

There are various kinds of tuning methods for PID controllers in worldwide literature. Considering the range of the control variables (e.g. engine speed is controlled from 1000 rpm to 6000 rpm) and the difference between long and short route, the method should keep simply and the process is to check the errors in the PID and then to tune the PID progressively. The result is checked by scope in the controllers and then tune the PID to be more accurate. Small efforts before changing types of controllers have to be implemented due to difference reference value. The method of step response is still operation in point form.

3.2.2 PID simulation

In this section, the operation points for reference values are engine speed equal to 2000 rpm, 3000 rpm and 4000 rpm with intake manifold pressure equal to 50 kpa, 100 kpa and 150 kpa, respectively. The reference residual gas fraction is fixed as 0.3 for both long and short routes. One example is shown in this section and rest of them can be found in Appendix B.

3.3 Cases Study

Since the performance of PID shows the good fit to reference value in this adapted model for both long and short route EGR, two cases for investigation can be continued regarding the contribution of EGR (e.g. fuel consumption). The method for the cases study is still to operate in point form and step response is selected to check the performance. In this section, fuel consumption behavior is demonstrated by air induction and torque generation. In this section, short-route EGR is selected as one example and the detail comparison between long and short route is presented in the next section.

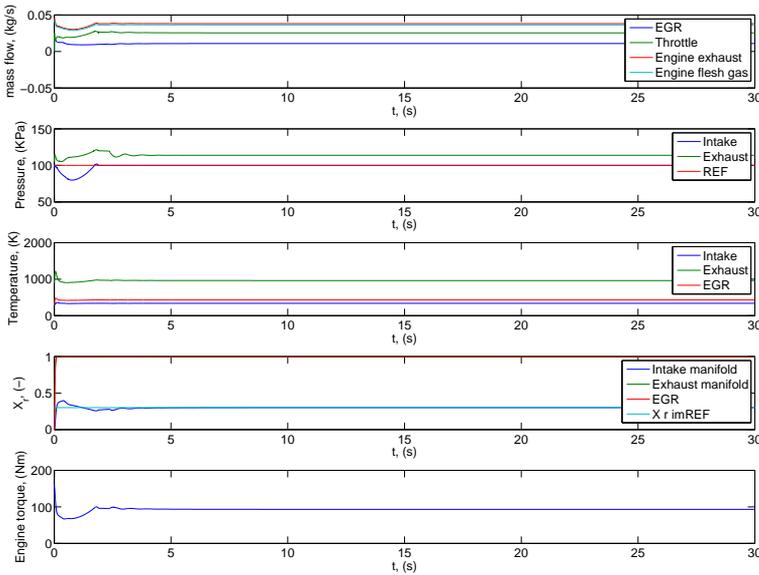
3.3.1 Case 1 Air induction comparison

In this section a simulation for engine speed equal to 3000 rpm with residual gas fraction step response from 0 to 0.3 which is close to the real EGR rate limit [Sinamon and Sellnau, 2008], and intake manifold pressure from 50 kpa to 72.5 kpa is done so that the expected torque generation is not changed. This simulation is based on the step response from EGR fully closed to EGR half open with same percent increased pressure so that the torque will be generated by same air mass flow. The result is shown in figure 3.3 on the page 25.

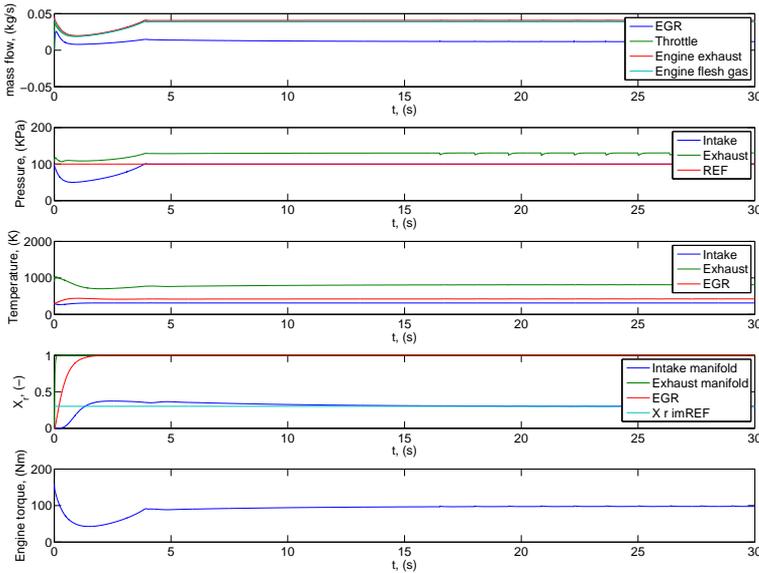
Consequently, regarding to figure 3.3 on the page 25, the result is that error for the torque is 0.13% with oscillation in the step point and the difference of air amount is 0.001 kg/s which is equal to 5% difference. The reason for the oscillation is probably because the PID speed for air mass flow becomes a bit faster than desired. Since it is assumed that the engine works on lambda equal to one in this thesis, air consumption is proportional to fuel consumption which means fuel has been 5% saved by implementing EGR.

3.3.2 Case 2 Torque generation comparison

Case 2 is similar with case 1 but with same air flow instead of torque. The intake manifold pressure is from 50 kpa to 75.7 kpa and the residual gas fraction is the same as case 1. The step response, which is shown in the figure 3.4 on the page 26, shows a good tracking performance to reference. Consequently, the error for the difference of the air amount is less than $1e-7\%$ which can be ignored and the torque generation difference is 3.76 Nm which is equal to 5.8% which is very close to the result in case 1 due to the engine works in lambda equal to one in



(a) Step response for short-route EGR



(b) Step response for long-route EGR

Figure 3.2: PID simulation for important parameters with engine speed equal to 3000 rpm, intake manifold pressure equal to 100 kpa and residual gas fraction equal to 0.3 as reference input value. The closed-loop controlled model shows very good tracking performance.

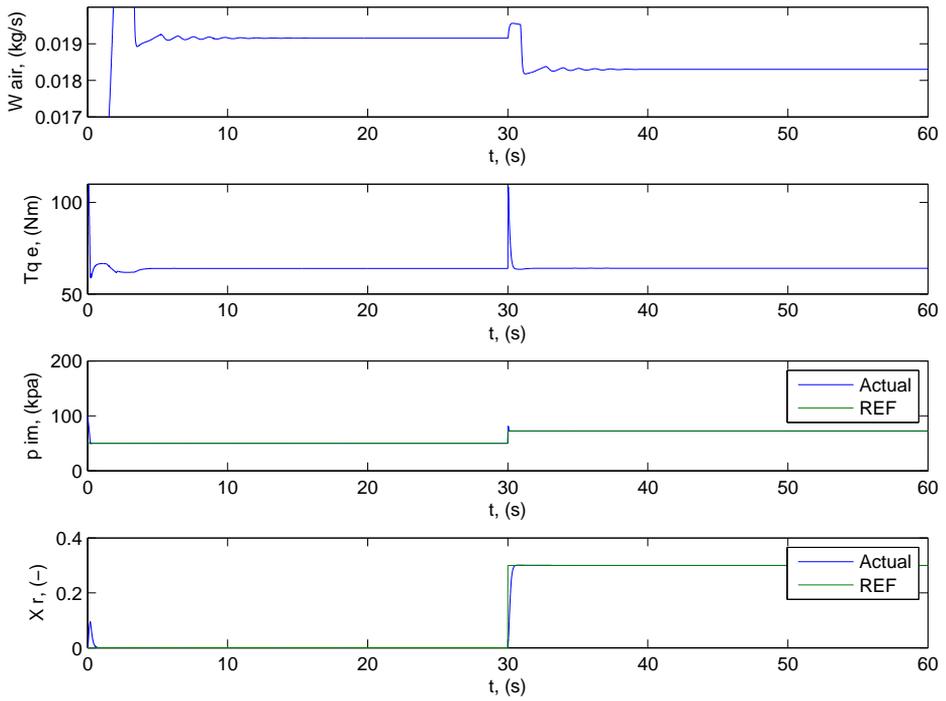


Figure 3.3: Same torque generation with air mass flow difference and step response in case 1.

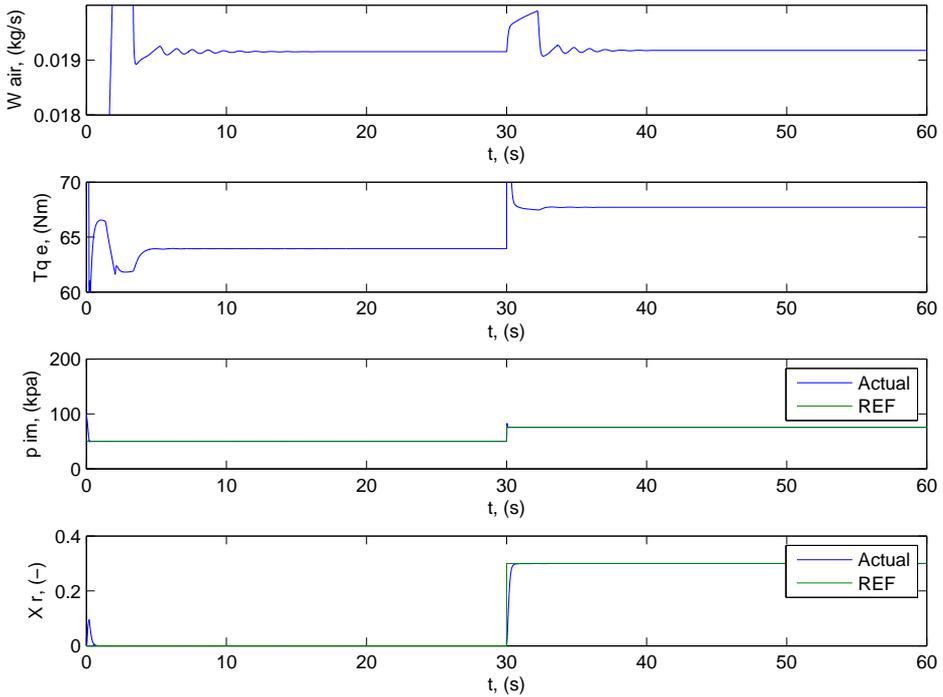


Figure 3.4: same air induction with engine torque generation difference in case 2

this thesis. So 5% more torque propulsion can be generated based on same fuel consumption with EGR implementation.

3.4 PID tuning method and performance summary

The tuning method presented in this chapter is based on multiple step responses by operating in different setting points. The performances of all controllers show good tracking performance to reference values which can be reliable for the next comparison and simulation. The controllers' parameters are uncomplicatedly calculated one at one time and small effort with a single tuning parameter is necessary to provide the PID controller parameters. The performance of comparison and relation plot can be checked in chapter 4.

4

Engine Control results with EGR open and discussion

In this chapter, overall parameters for both long and short routes are demonstrated by comparing simulation performance in different operated setting points with EGR valve opening in specific EGR rate which is presented in section 4.1. Fuel efficiency comparison and 3D parameter interaction performance including maximum residual gas fraction as function of speed and load are presented in section 4.2 and 4.3.

4.1 Comparison between long and short routes EGR

This section compares performance of long and short route EGR and discussion is presented.

4.1.1 Basic parameters comparison

The basic parameters are air mass flow, intake manifold pressure and temperature, residual gas fraction in intake manifold and engine torque. The method is to run stationary operating points, intake manifold pressure equal to 50 kpa, 100 kpa and 150 kpa and engine speed 2000 rpm, 3000 rpm and 4000 rpm, respectively. Residual gas fraction reference is selected as 0.3 which is the most common and effective value, see [Sinnamon and Sellnau, 2008]. In this section the middle point is selected as an example and the simulation result is shown in figure 4.1 on the page 28. Results for the other operating points can be found in Appendix C.

According to this comparison, the reference values, which are the operation points, shows that the actual value follows well enough and the speculation is proved here that long-route EGR takes more time to achieve stationary point due to longer distance with more pipe for long-route EGR. Furthermore, short-route

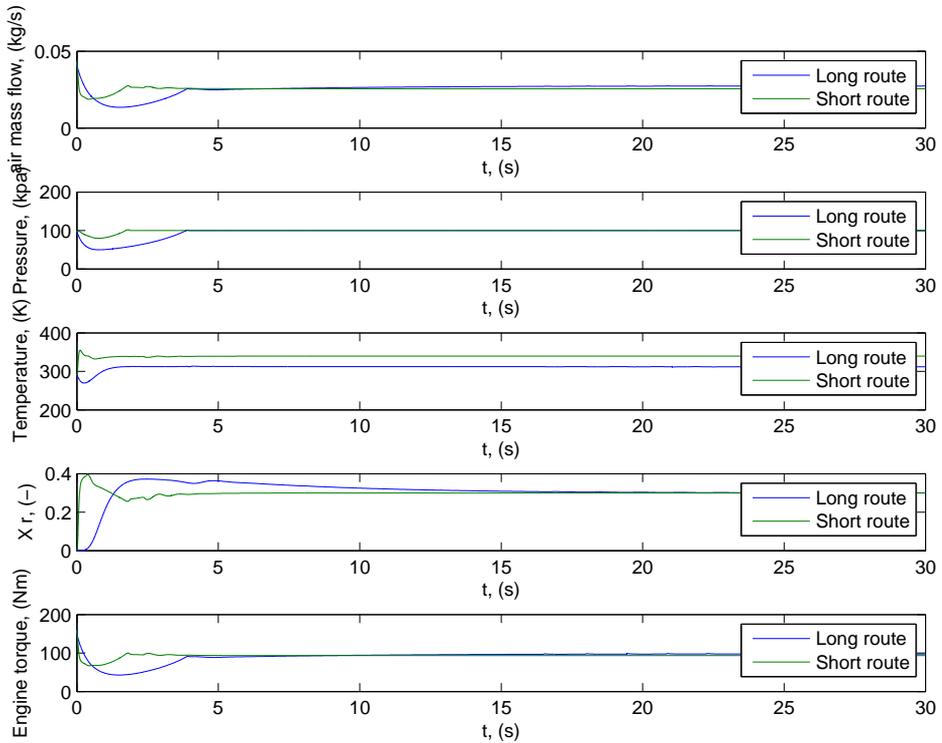


Figure 4.1: Comparison result for long and short routes EGR regarding basic parameters. The operating points are that engine torque is equal to 100 Nm and residual gas fraction is equal to 0.3 as reference values based on the engine speed is equal to 3000 rpm. long-route EGR takes more time to achieve stationary point due to longer distance with more pipe for long-route EGR. Furthermore, short-route EGR has higher intake manifold temperature due to that the long-route EGR has to pass intercooler once with longer distance back to engine.

EGR has higher intake manifold temperature due to that the long-route EGR has to pass intercooler once with longer distance back to engine.

The performance of how the process that converts chemical potential energy contained in a carrier fuel into work needs to be investigated. Therefore, the fuel efficiency comparison between the TCSI engine without EGR, with long-route EGR and with short-route EGR is presented and it is shown in figure 4.2 on the page 30 and the expression is shown in equation 4.1. The operation point is engine torque equal to 80 Nm, engine speed equal to 3000 rpm and residual gas fraction equal to 0.3.

$$\eta_{f,in} = \frac{W_{i,g} - W_{i,f} - W_{i,p}}{m_f q_{LHV}} \quad (4.1)$$

Where $W_{i,g}$ is gross indicated work, $W_{i,f}$ is friction work and $W_{i,p}$ is pumping work.

The comparison of fuel efficiency shows that EGR has the ability to increase the total fuel efficiency. Short-route EGR produces higher efficiency and long-route EGR is following. They are both higher than the normal engine without EGR. Consequently, EGR enable the engine to reduce indicated work loss and to increase the efficiency of the engine.

The last comparison is considered according to the cases study since the torque controller and air mass flow controller have been implemented. In this section, those two cases can be more accurate and the comparison for same torque generation is shown in figure 4.3 on the page 31 based on torque reference equal to 100 Nm, residual gas fraction reference is set from 0 to 0.3 and engine speed 3000 rpm.

The result is that there is 0.014 kg/s fuel saved which is equal to around 4.9% for short route and 0.008 kg/s fuel saved which is equal to 2.8% for long route. The difference between them is 0.006 kg/s, so short-route EGR has better fuel economy.

For same fuel consumption is shown in figure 4.4 on the page 32 based on air mass flow reference equal to 0.025 kg/s and residual gas fraction reference is set from 0 to 0.3. The engine speed is 3000 rpm.

The result is 4.87 Nm propulsion increasing which is equal to around 5.7% difference for short route and 2.6814 Nm power increasing which is equal to 3.1% difference. Accordingly, short-route EGR has better propulsion performance during the comparison.

4.2 2D Engine map for TCSI engine

In this section, engine speed, engine torque and residual gas fraction are presented in 2D engine map, so that it becomes easier to represent how the maximum EGR rate depends on engine speed and torque. Moreover, residual gas fraction can be simulated over the engine's operating range by implementing stationary simulations during the control of engine with operation points on a grid.

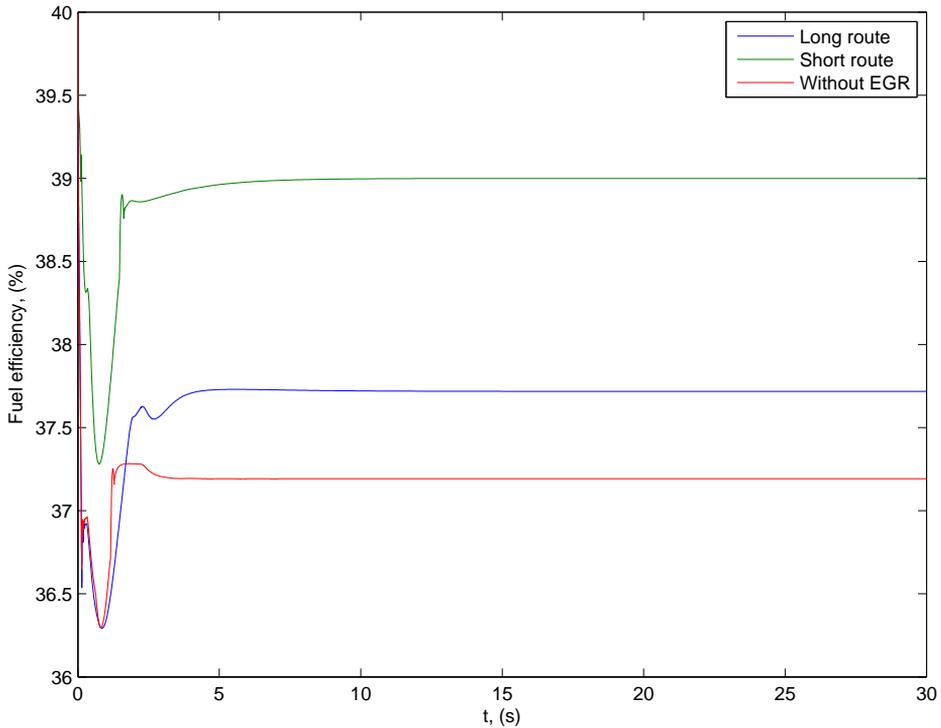


Figure 4.2: Fuel efficiency comparison between normal TCSI engine, short-route EGR TCSI engine and long-route EGR TCSI engine. The net η values with EGR are higher than the value without EGR. The operating points are engine torque equal to 80 Nm, engine speed equal to 3000 rpm and residual gas fraction equal to 0.3.

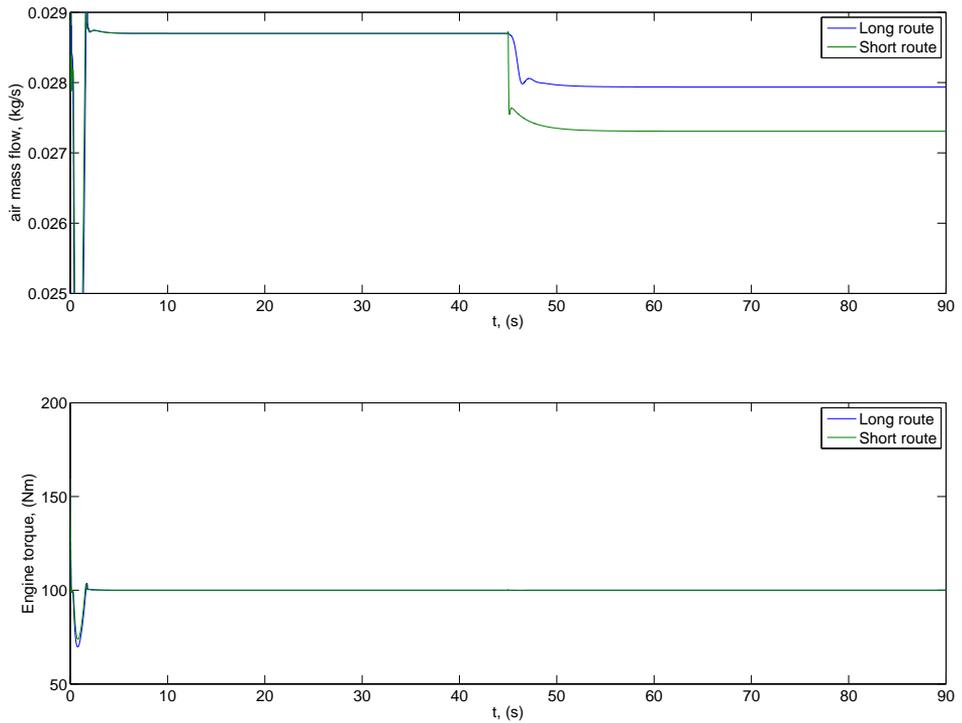


Figure 4.3: Same torque generation with different air mass flow induction based on torque reference equal to 100 Nm, residual gas fraction reference is set from 0 to 0.3 and engine speed 3000 rpm. Since lambda is equal to one so that the difference of propulsion can be check based on same fuel consumption. Short-route EGR has better fuel economy.

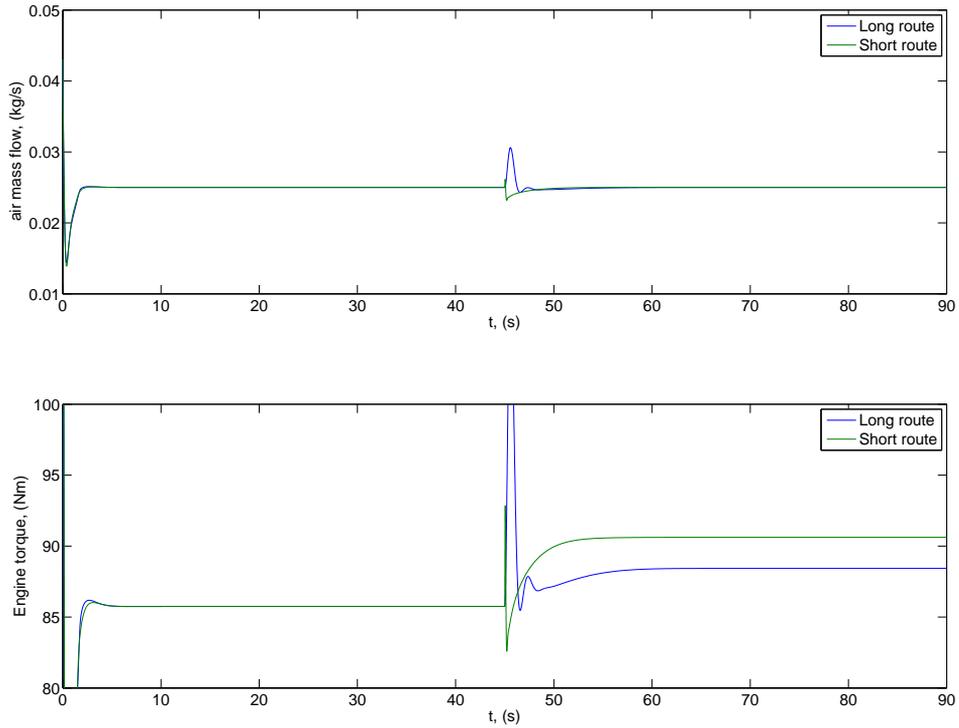


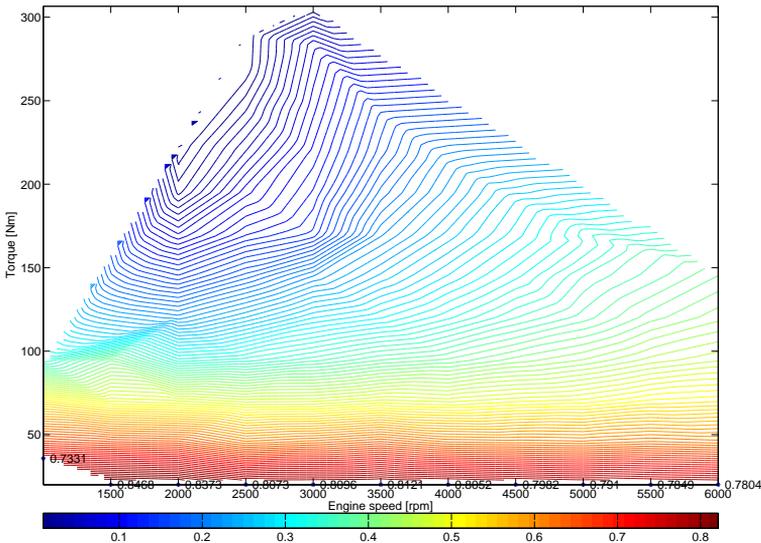
Figure 4.4: Same air mass flow induction with different torque generation based on air mass flow reference equal to 0.025 kg/s and residual gas fraction reference is set from 0 to 0.3. The engine speed is 3000 rpm. Since lambda is equal to one so that the air induction is equal to fuel consumption. Short-route EGR has better torque generation performance.

The method for the data-collection and simulation is to do four cases identification by using torque controller and then to simulate those all steady points with possible local maximum residual gas fraction by tuning the residual gas fraction reference manually. The data which only match the torque reference with 5 Nm tolerance can be gathered. First case is to set throttle total open and set wastegate close. Then EGR rate is controlled by torque PID controller. Second case is to set EGR total open and set wastegate close then to control the throttle valve by torque controller. This case has high amount failure points since EGR rate cannot keep high value all the time. The third case is to set throttle and EGR total open and control the wastegate. Since wastegate is very sensitive so that it is hard to control the engine and the failure points are still happening frequently. The fourth case is to close wastegate to give strong turbo performance and control both EGR and throttle by torque controller. The last case provides most good points. After that a steady testing is done, a tuning process by simulation of possible maximum residual gas fraction is done for those steady points. The results for both short and long routes are shown in figure 4.5 on the page 34 and figure 4.6 on the page 35. The error is controlled to be less than 0.5%.

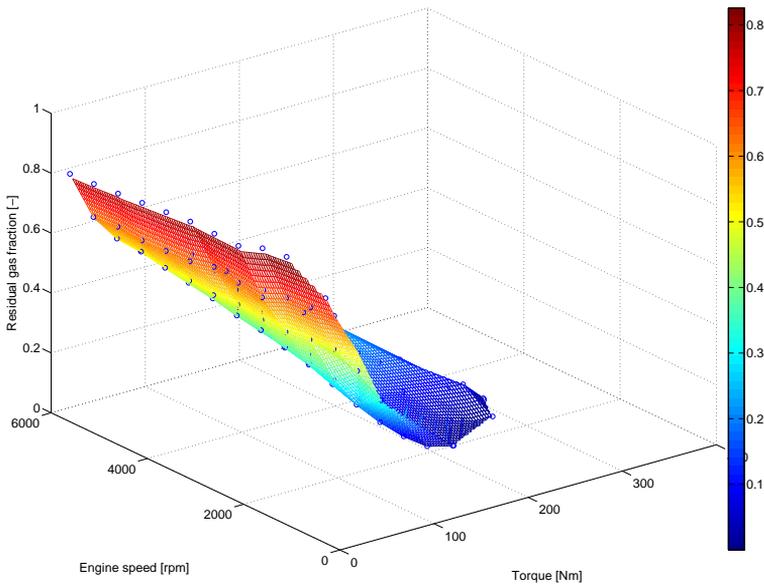
According to the results, The biggest torque value for both routes is around 307 Nm. The relations between engine torque, speed and residual gas fraction for both routes are almost the same with similar controlled range.

The global maximum value for short-route EGR is around 0.846 and 0.5 for long-route EGR. The mean values of local maximum residual gas fraction for short route is 0.8005 and for long-route EGR is 0.4327. The local maximum values for both routes are happened in low torque. Comparing both 2D engine maps, short-route EGR has almost 46% more mean value of local maximum residual gas fraction than long-route EGR.

According to the 3D engine maps with respect to fuel efficiency in the relation with engine speed and engine torque, see figure 4.7 on the page 36, the result is that both routes have common performance that fuel efficiency increases with torque increasing. Therefore, a comparison between the engine without EGR and the engine with long or short route EGR should be considered. So figure 4.8 on the page 37 and figure 4.9 on the page 38 are shown in this section. According to those figures which shows the information of the fuel efficiency difference, positive value means more efficiency for EGR and negative value means more efficiency for the normal engine without EGR. Consequently, residual gas fraction decreases with fuel efficiency decreasing with respect to the comparison between the engine with EGR and the normal engine without EGR. According to figure 4.9, which is a comparison between short route and long route EGR in the same range of engine speed and engine torque, short route EGR has better performance of increasing fuel efficiency based on the level that positive value means more fuel efficiency for short route.

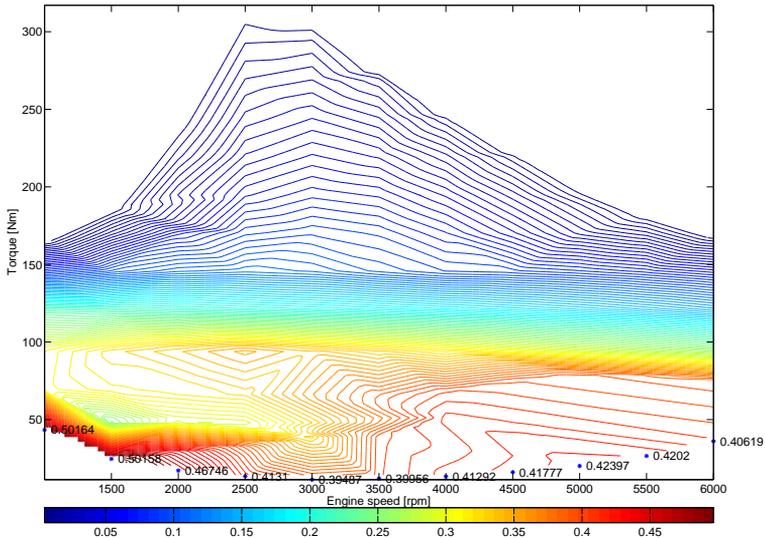


(a) 2D engine map with local maximum residual gas fraction marked for short-route EGR.

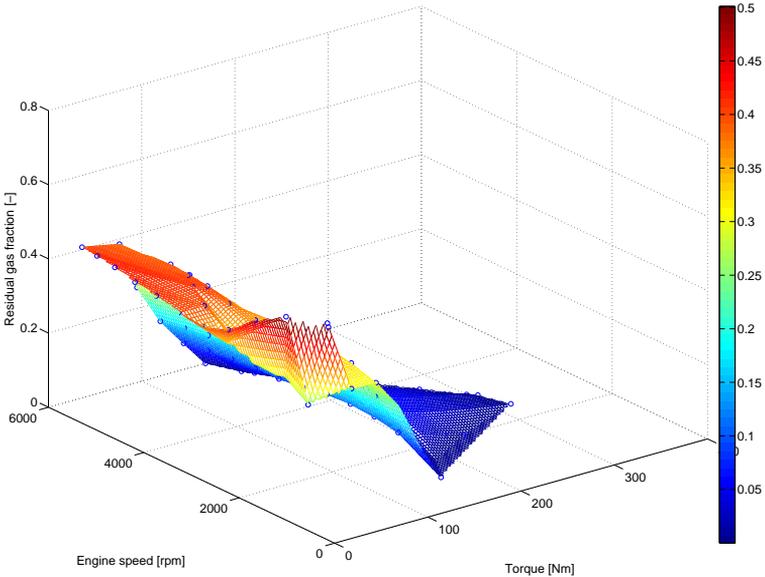


(b) 3D engine map with all residual gas fraction points distribution for short-route EGR.

Figure 4.5: Graphic representations of the parameters stored in 2D and 3D maps for short-route EGR. The three parameters are residual gas fraction, engine torque and engine speed. The local maximum residual gas fraction is marked.

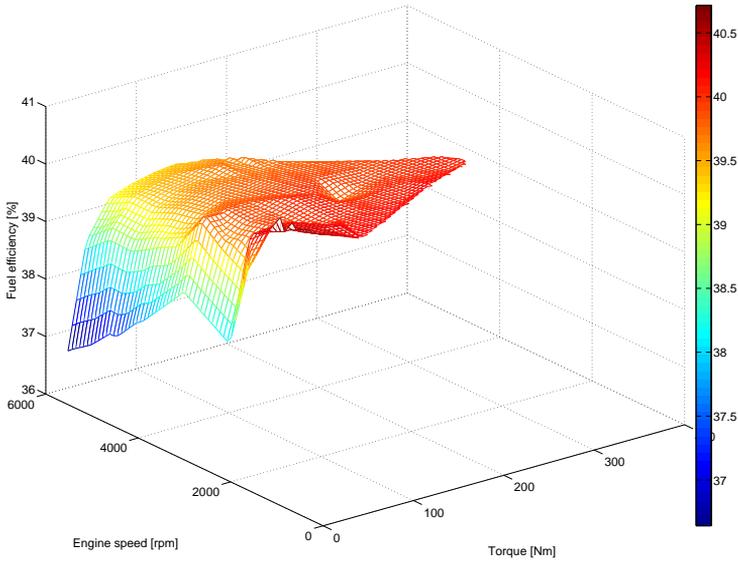


(a) 2D engine map with local maximum residual gas fraction marked for long-route EGR.

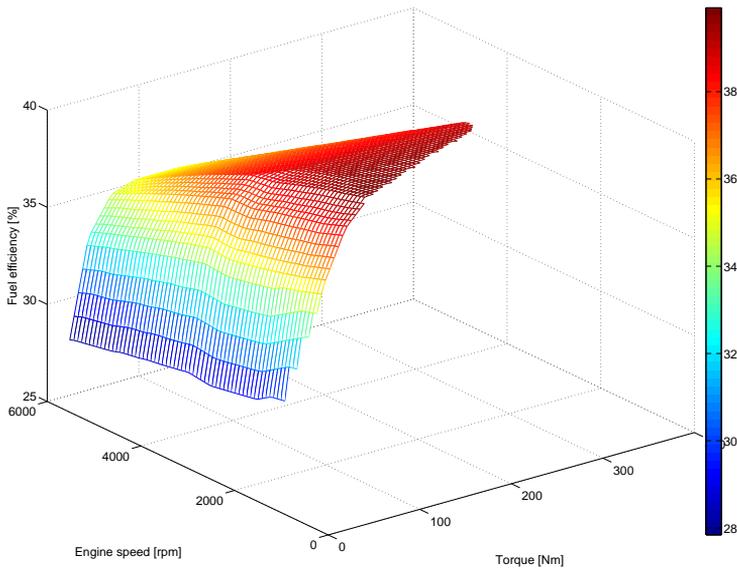


(b) 3D engine map with all residual gas fraction points distribution for long-route EGR.

Figure 4.6: Graphic representations of the parameters stored in 2D and 3D maps for long-route EGR. The three parameters are residual gas fraction, engine torque and engine speed. The local maximum residual gas fraction is marked.

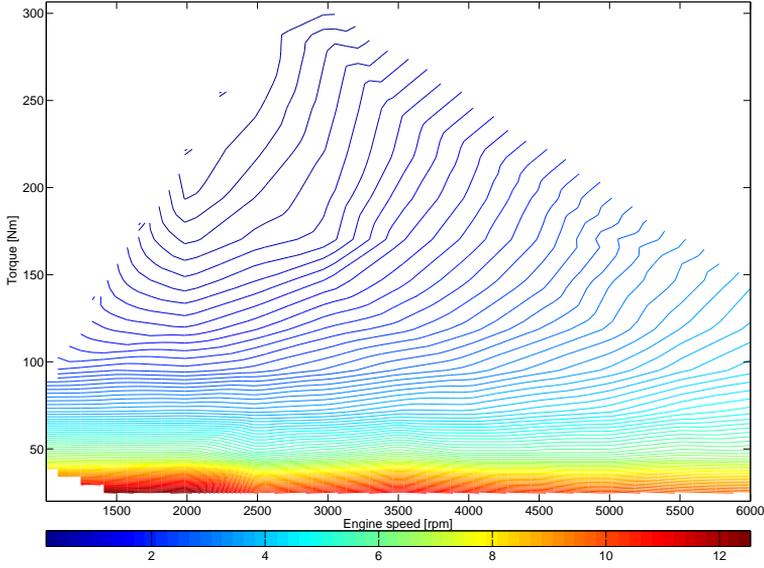


(a) 3D engine map with fuel efficiency for short-route EGR.

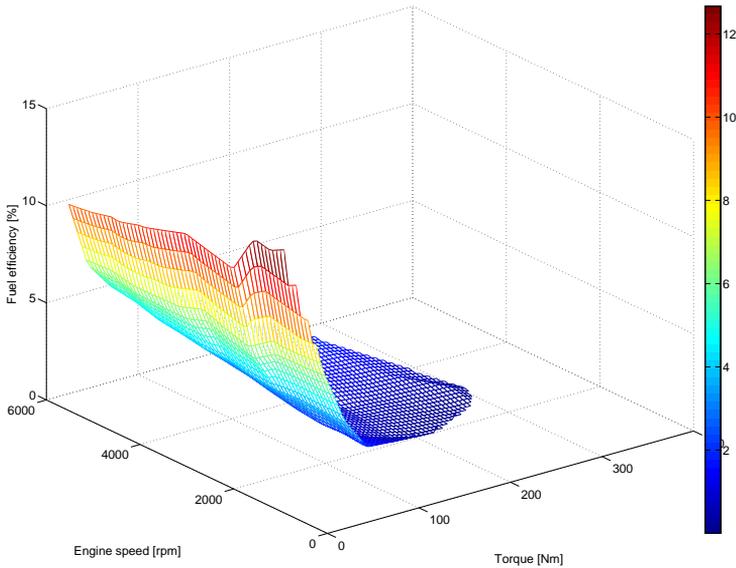


(b) 3D engine map with fuel efficiency for long-route EGR.

Figure 4.7: Graphic representations of the parameters stored in 3D maps for long and short routes EGR. The three parameters are fuel efficiency, engine torque and engine speed.

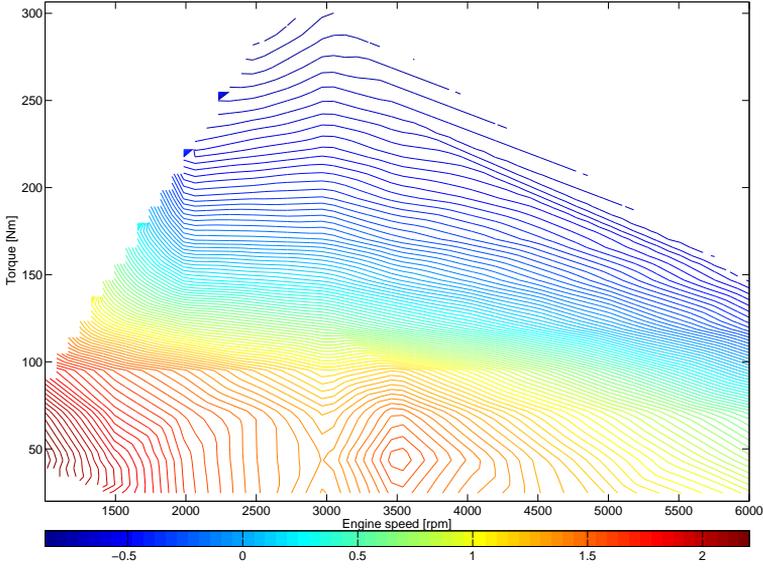


(a) 2D engine map comparison between short route and the engine without EGR.

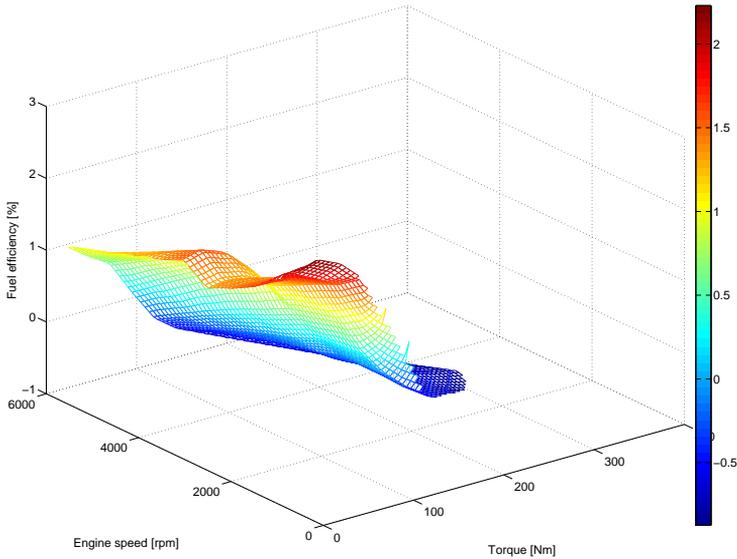


(b) 3D engine map comparison between short route and the engine without EGR.

Figure 4.8: Graphic representations comparison of the parameters stored in 2D and 3D maps for short routes EGR and the engine without EGR. The three parameters are fuel efficiency difference, engine torque and engine speed. Positive value means more efficiency for short route system and negative value means more efficiency for the engine without EGR system.



(a) 2D engine map comparison between long route and the engine without EGR.



(b) 3D engine map comparison between long route and the engine without EGR.

Figure 4.9: Graphic representations comparison of the parameters stored in 2D and 3D maps for short routes EGR and the engine without EGR. The three parameters are fuel efficiency difference, engine torque and engine speed. Positive value means more efficiency for long route and negative value means more efficiency for the engine without EGR system.

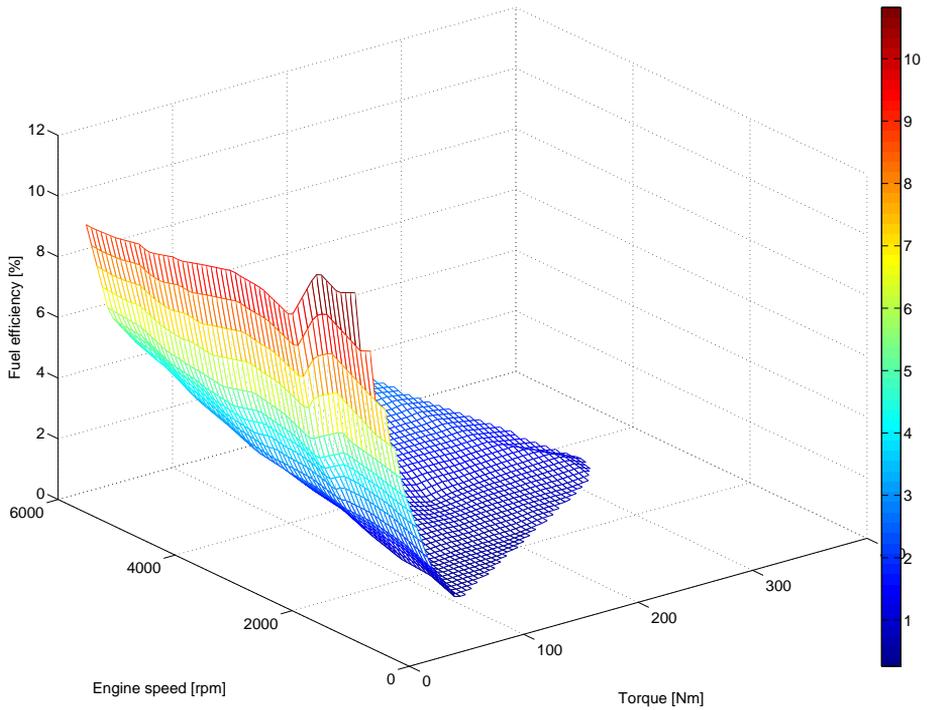


Figure 4.10: Graphic representations comparison of the parameters stored in 3D maps for short routes EGR and long route EGR. The three parameters are fuel efficiency difference, engine torque and engine speed. Positive value means more efficiency for short route and negative value means more efficiency for long route EGR system.

5

Summary and Conclusions

This thesis presents a mean value engine model by using a TCSI engine as an example and the work consists of two parts in Matlab/Simulink and four types of simulations. The two parts are one TCSI engine with short and long routes EGR separately and one engine control unit with three main controllers which are throttle valve controller with feedforward and feedback parts, wastegate boost controller with only feedback PID controller and EGR valve with residual gas fraction in intake manifold reference controller. They are utilized in different scenarios.

This thesis presented the basic model with different step response behaviors. It has shown the result and provide more detailed cases study and comparison. 2D and 3D engine maps are determined in an integral simulation in the end including residual gas fraction with local maximum points marked, engine speed from 1000 rpm to 6000 rpm and available engine torque.

The EGR model for both routes show good agreement with reference value, but it needs more validation against measurement data. The comparison between both routes shows similar control values with different local maximum residual gas fraction, but overall short-route EGR has better performance not only in fuel economy but also torque generation. In the efficiency aspect, EGR increases fuel efficiency comparing with the normal engine without EGR. Among them, short route performs better than long route again.

Future work

There are some issues which could be interesting for future work in this thesis work. They are presented in this section.

Lambda limitation: Now the engine works only in the condition when lambda is equal to one. High lambda simulation will be interesting to be implemented with development of Oxygen fraction state or other air component vector. Investigating how EGR rate performs in different situations.

More advanced controllers: A more advanced controller can be developed to have good dynamic feedback instead of static feedback. An advanced tuning method which can be automatic as a interesting topic for investigation.

Reference values generation: Oscillations lead to a result of the reference value oscillation when step response performs in different pedal positions. A more clear and effective investigation can be done to identify how sensitive analysis can help for improvement of control system.

Delay time implement and advanced engine step response: According to the performance of engine step response in long route EGR, there should be a time delay since it needs more time for the mass flow through the whole pipe for exhaust gas recirculated in reality. More measurement data need to be compared in order to have more accurate contribution.

Driveline implementation: To implement this thesis into a basic driveline with clutch, Vehicle and drive model can be interesting for next investigation. Moreover, a lot of drive cycle can be compared and the fuel consumption can be determined as well to prove the EGR function. EGR rate condition needs to be considered to give more suitable strategy to control the engine.

Appendix

A

EGR Step responses

In this Appendix A, the rest step response figures are presented here according to the operation points shown in the chapter 2

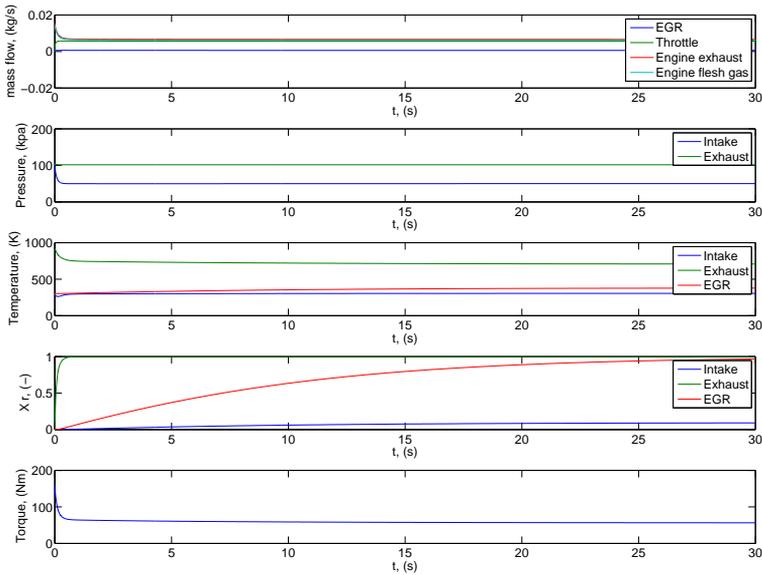
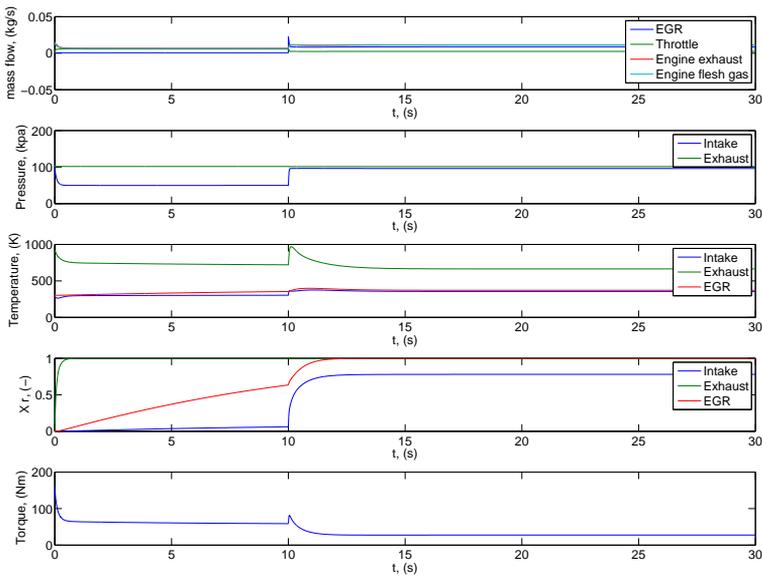
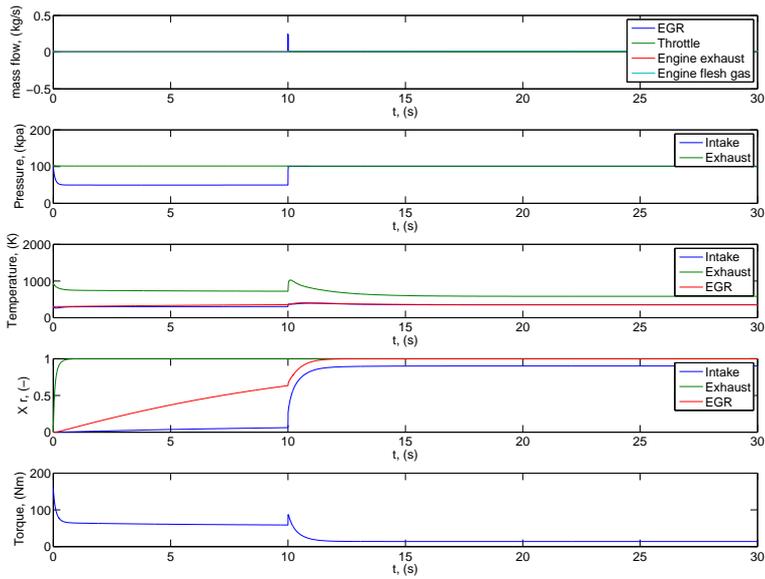
(a) $u_{egr}=0$ for short-route EGR(b) $u_{egr}=10$ for short-route EGR

Figure A.1: step response for engine speed equal to 1000 rpm and intake manifold pressure equal to 50 kpa in different egr valve positions for short-route EGR. The EGR valve opens which leads to that the engine torque decreases, residual gas mass flow increases and exhaust manifold pressure decreases.



(a) $u_{egr}=100$ for short-route EGR

Figure A.2: step response for engine speed equal to 1000 rpm and intake manifold pressure equal to 50 kpa in different egr valve positions for short-route EGR. The EGR valve opens which leads to that the engine torque decreases, residual gas mass flow increases and exhaust manifold pressure decreases.

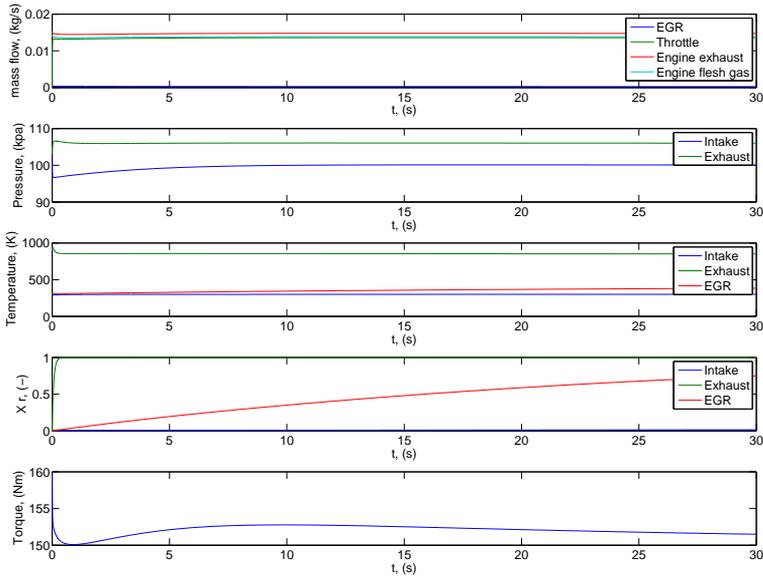
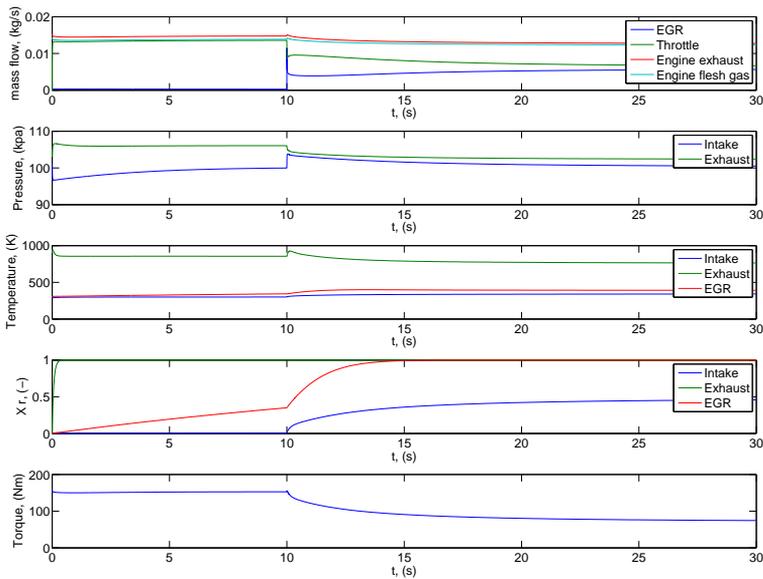
(a) $u_{egr}=0$ for short-route EGR(b) $u_{egr}=10$ for short-route EGR

Figure A.3: step response for engine speed equal to 1000 rpm and intake manifold pressure equal to 100 kpa in different egr valve positions for short-route EGR. The EGR valve opens which leads to that the engine torque decreases, residual gas mass flow increases and exhaust manifold pressure decreases.

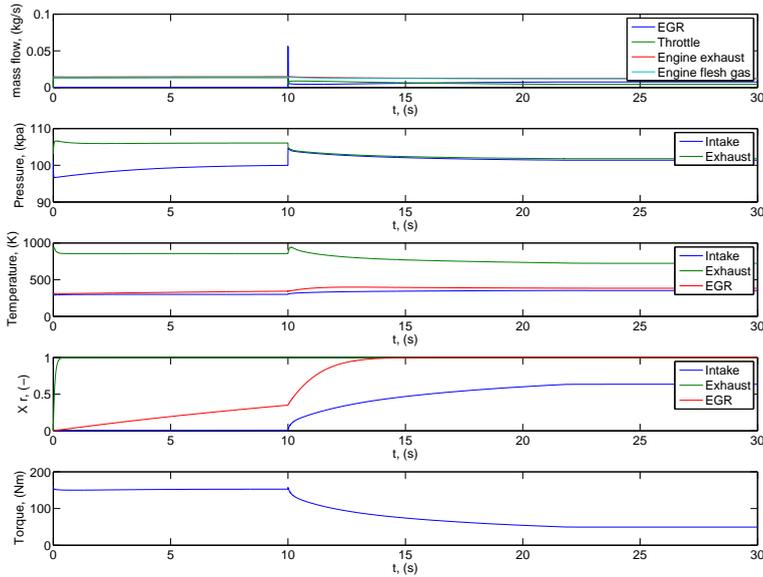
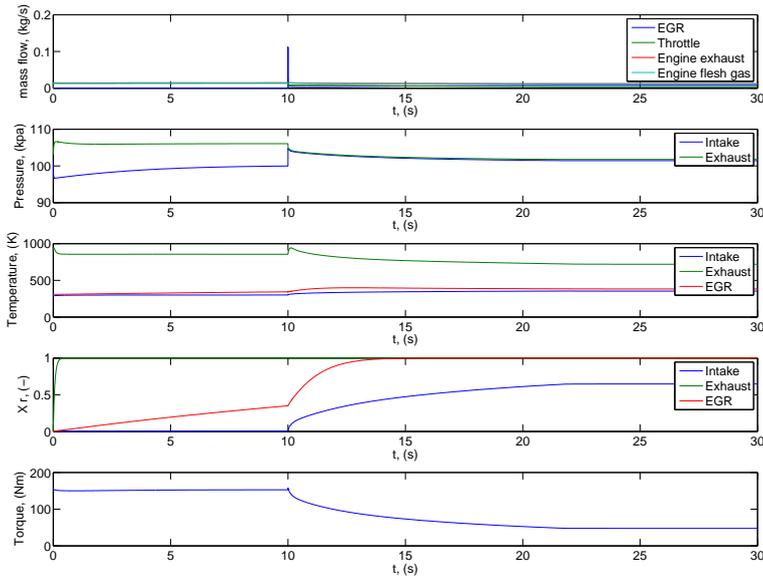
(a) $u_{egr}=50$ for short-route EGR(b) $u_{egr}=100$ for short-route EGR

Figure A.4: step response for engine speed equal to 1000 rpm and intake manifold pressure equal to 100 kpa in different egr valve positions for short-route EGR. The EGR valve opens which leads to that the engine torque decreases, residual gas mass flow increases and exhaust manifold pressure decreases.

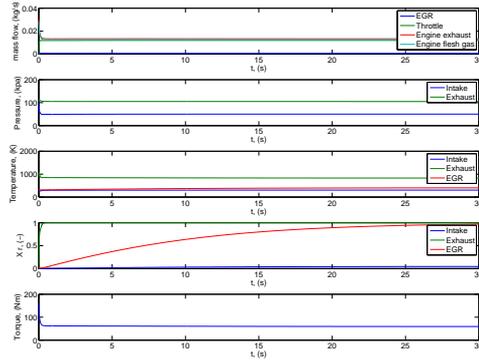
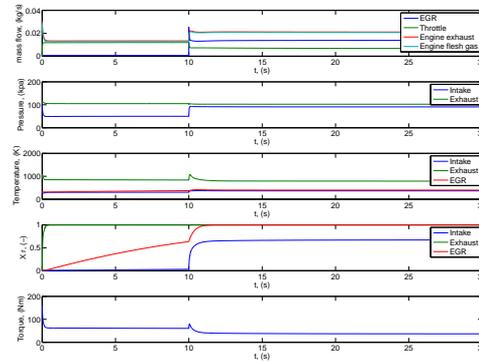
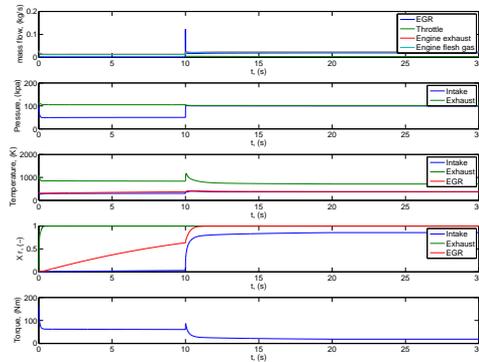
(a) $u_{egr}=0$ for short-route EGR(b) $u_{egr}=10$ for short-route EGR(c) $u_{egr}=50$ for short-route EGR

Figure A.5: step response for engine speed equal to 2000 rpm and intake manifold pressure equal to 50 kpa in different egr valve positions for short-route EGR. The EGR valve opens which leads to that the engine torque decreases, residual gas mass flow increases and exhaust manifold pressure decreases.

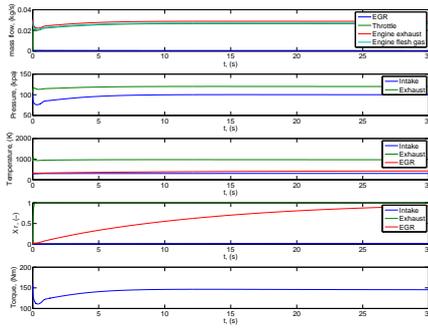
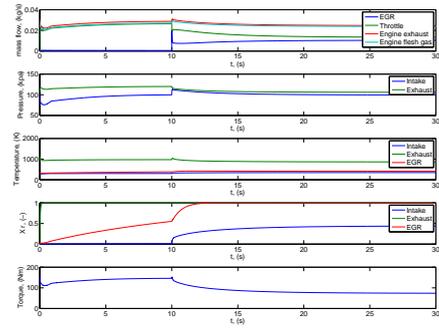
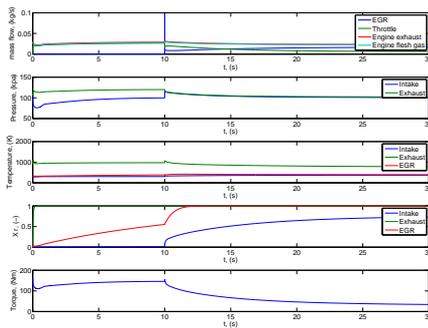
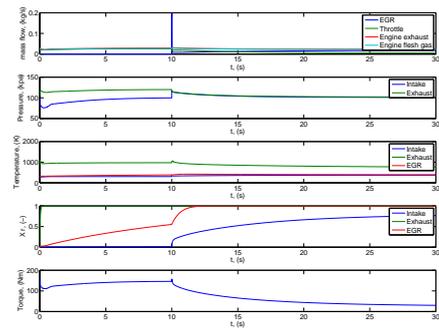
(a) $u_{egr}=0$ for short-route EGR(b) $u_{egr}=10$ for short-route EGR(c) $u_{egr}=50$ for short-route EGR(d) $u_{egr}=100$ for short-route EGR

Figure A.6: step response for engine speed equal to 2000 rpm and intake manifold pressure equal to 100 kpa in different egr valve positions for short-route EGR. The EGR valve opens which leads to that the engine torque decreases, residual gas mass flow increases and exhaust manifold pressure decreases.

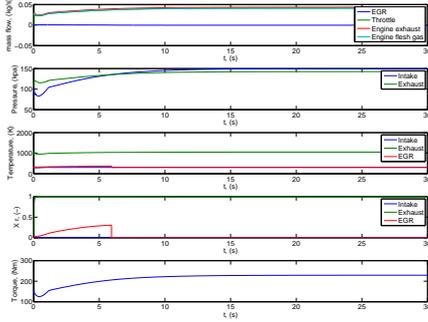
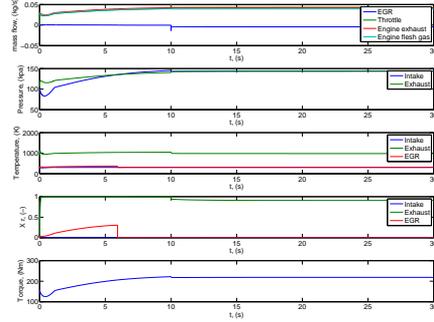
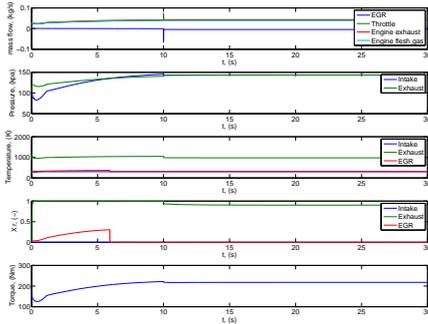
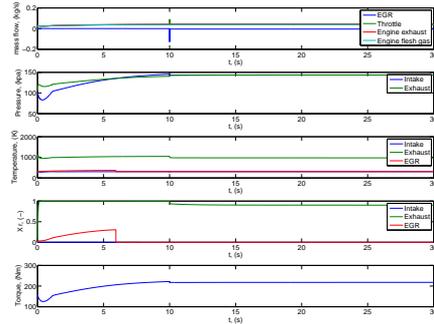
(a) $u_{egr}=0$ for short-route EGR(b) $u_{egr}=10$ for short-route EGR(c) $u_{egr}=50$ for short-route EGR(d) $u_{egr}=100$ for short-route EGR

Figure A.7: step response for engine speed equal to 2000 rpm and intake manifold pressure equal to 150 kpa in different egr valve positions for short-route EGR. The EGR valve opens which leads to that the engine torque decreases, residual gas mass flow increases and exhaust manifold pressure decreases.

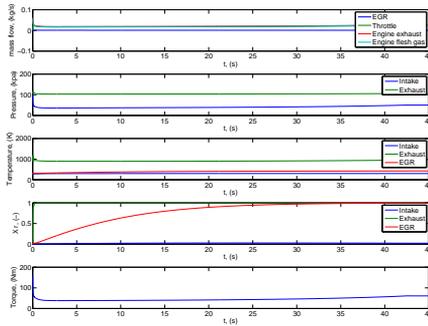
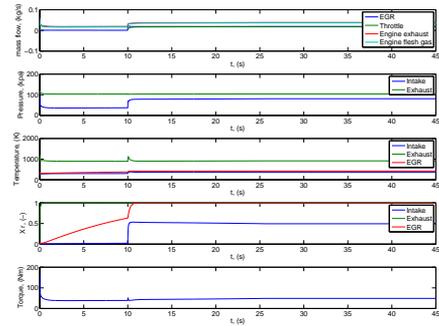
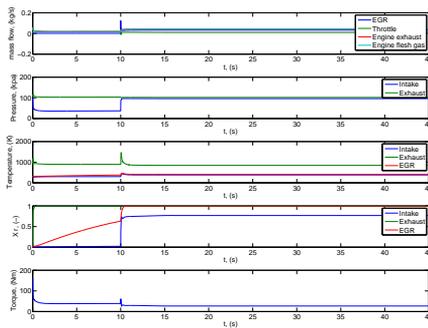
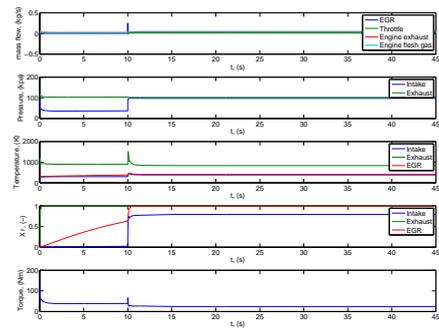
(a) $u_{egr}=0$ for short-route EGR(b) $u_{egr}=10$ for short-route EGR(c) $u_{egr}=50$ for short-route EGR(d) $u_{egr}=100$ for short-route EGR

Figure A.8: step response for engine speed equal to 4000 rpm and intake manifold pressure equal to 50 kpa in different egr valve positions for short-route EGR. The EGR valve opens which leads to that the engine torque decreases, residual gas mass flow increases and exhaust manifold pressure decreases.

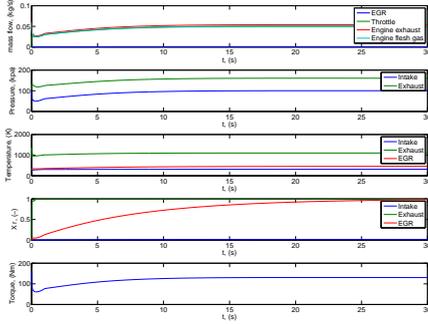
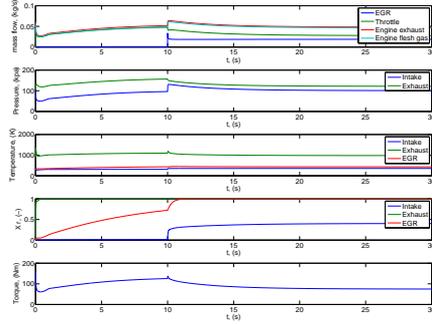
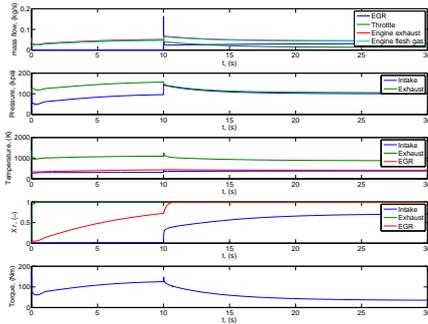
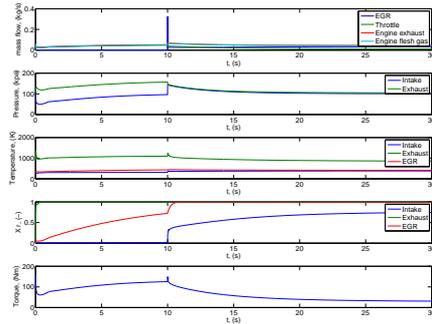
(a) $u_{egr}=0$ for short-route EGR(b) $u_{egr}=10$ for short-route EGR(c) $u_{egr}=50$ for short-route EGR(d) $u_{egr}=100$ for short-route EGR

Figure A.9: step response for engine speed equal to 4000 rpm and intake manifold pressure equal to 100 kpa in different egr valve positions for short-route EGR. The EGR valve opens which leads to that the engine torque decreases, residual gas mass flow increases and exhaust manifold pressure decreases.

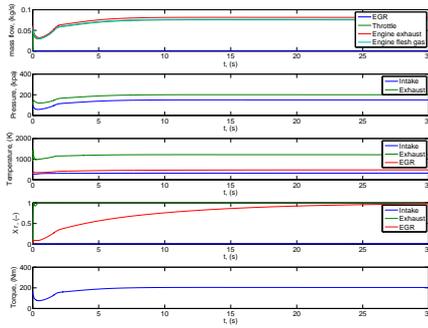
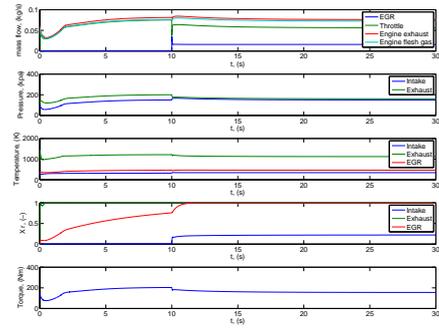
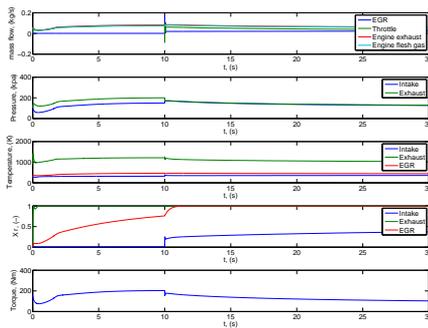
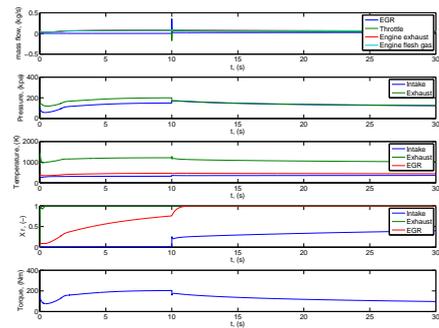
(a) $u_{egr}=0$ for short-route EGR(b) $u_{egr}=10$ for short-route EGR(c) $u_{egr}=50$ for short-route EGR(d) $u_{egr}=100$ for short-route EGR

Figure A.10: step response for engine speed equal to 4000 rpm and intake manifold pressure equal to 150 kpa in different egr valve positions for short-route EGR. The EGR valve opens which leads to that the engine torque decreases, residual gas mass flow increases and exhaust manifold pressure decreases.

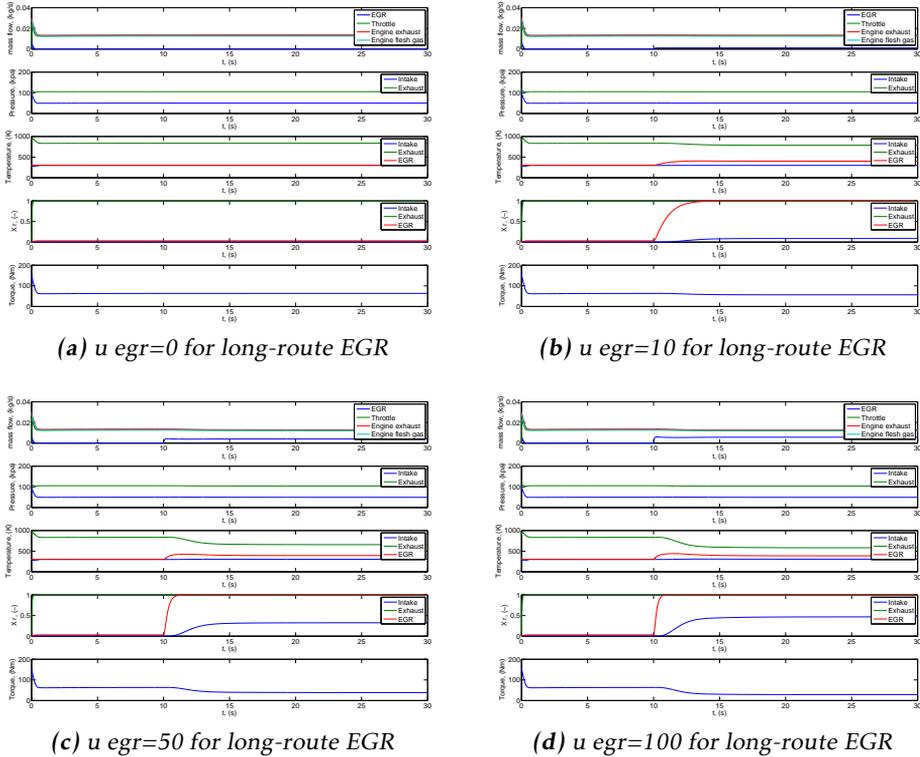


Figure A.11: step response for engine speed equal to 2000 rpm and intake manifold pressure equal to 50 kpa in different egr valve positions for long-route EGR. The EGR valve opens which leads to that the engine torque decreases, residual gas mass flow increases and exhaust manifold pressure decreases.

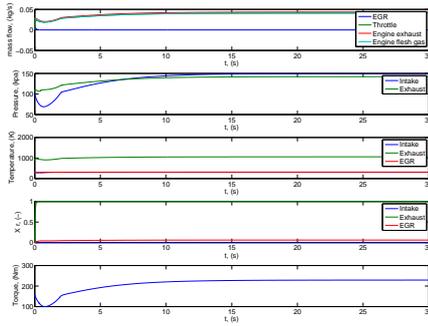
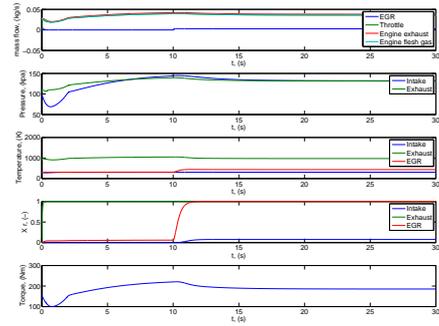
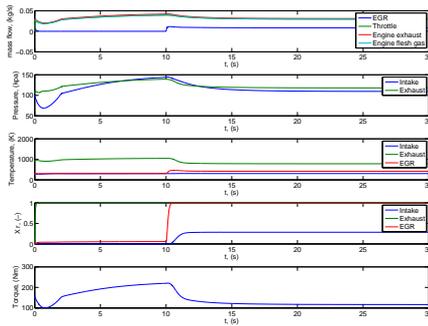
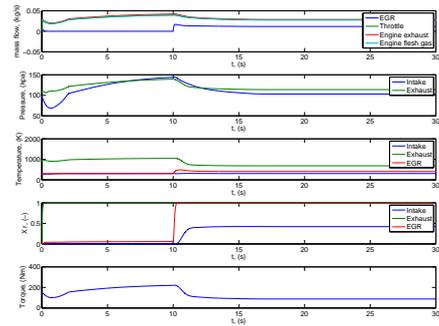
(a) $u_{egr}=0$ for long-route EGR(b) $u_{egr}=10$ for long-route EGR(c) $u_{egr}=50$ for long-route EGR(d) $u_{egr}=100$ for long-route EGR

Figure A.12: step response for engine speed equal to 2000 rpm and intake manifold pressure equal to 150 kpa in different egr valve positions for long-route EGR. The EGR valve opens which leads to that the engine torque decreases, residual gas mass flow increases and exhaust manifold pressure decreases.

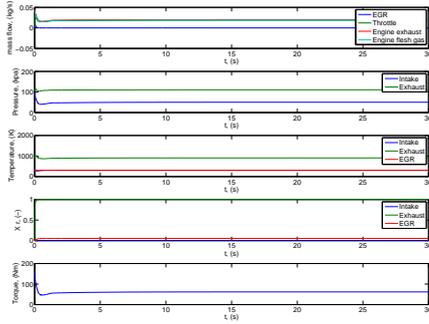
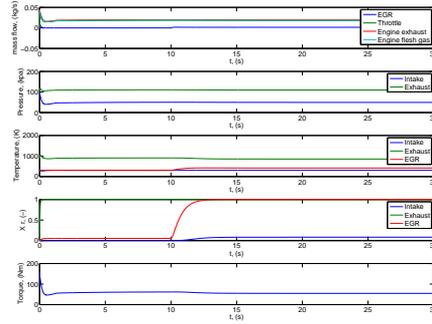
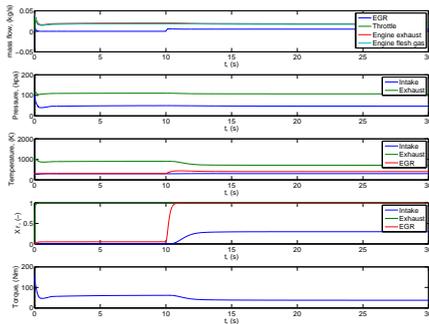
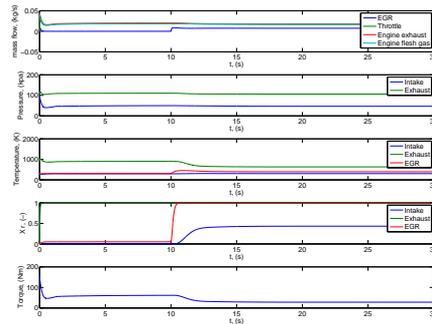
(a) $u_{egr}=0$ for long-route EGR(b) $u_{egr}=10$ for long-route EGR(c) $u_{egr}=50$ for long-route EGR(d) $u_{egr}=100$ for long-route EGR

Figure A.13: step response for engine speed equal to 3000 rpm and intake manifold pressure equal to 50 kpa in different egr valve positions for long-route EGR. The EGR valve opens which leads to that the engine torque decreases, residual gas mass flow increases and exhaust manifold pressure decreases.

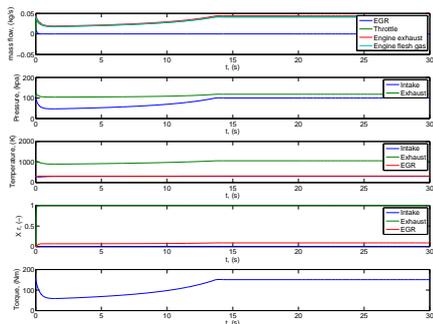
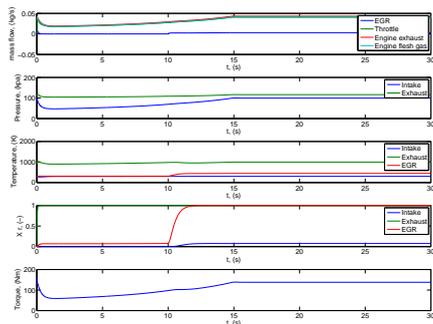
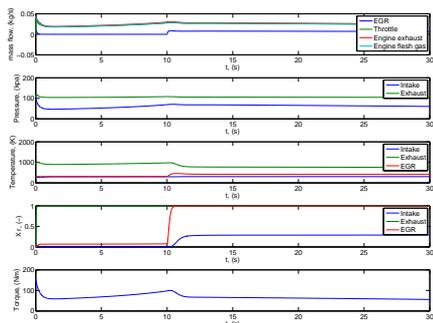
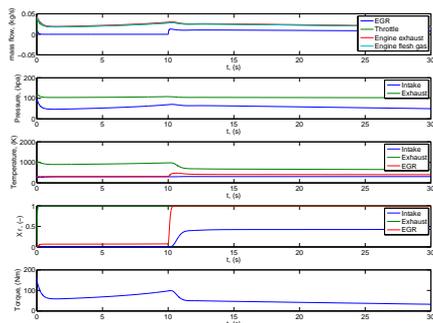
(a) $u_{egr}=0$ for long-route EGR(b) $u_{egr}=10$ for long-route EGR(c) $u_{egr}=50$ for long-route EGR(d) $u_{egr}=100$ for long-route EGR

Figure A.14: step response for engine speed equal to 3000 rpm and intake manifold pressure equal to 100 kpa in different egr valve positions for long-route EGR. The EGR valve opens which leads to that the engine torque decreases, residual gas mass flow increases and exhaust manifold pressure decreases.

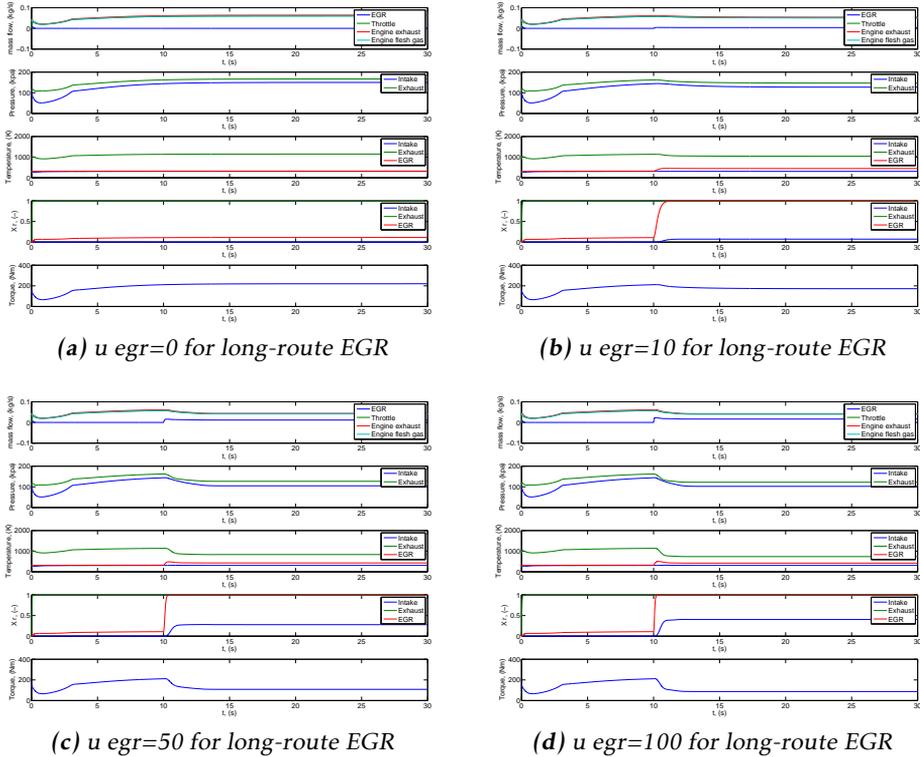


Figure A.15: step response for engine speed equal to 3000 rpm and intake manifold pressure equal to 150 kpa in different egr valve positions for long-route EGR. The EGR valve opens which leads to that the engine torque decreases, residual gas mass flow increases and exhaust manifold pressure decreases.

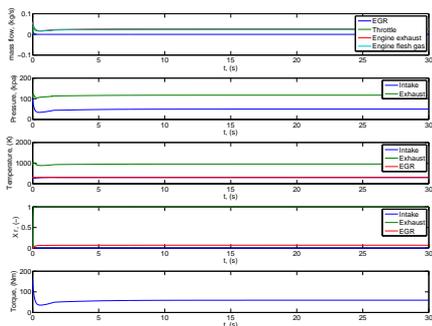
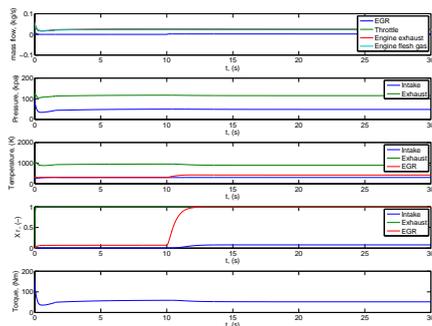
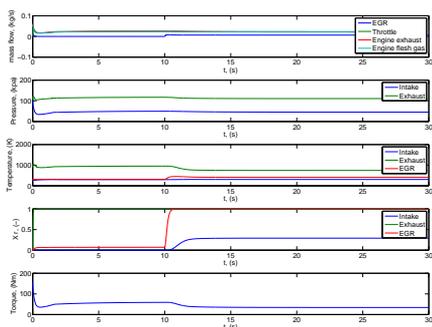
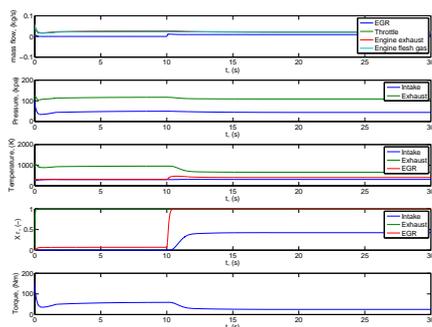
(a) $u_{egr}=0$ for long-route EGR(b) $u_{egr}=10$ for long-route EGR(c) $u_{egr}=50$ for long-route EGR(d) $u_{egr}=100$ for long-route EGR

Figure A.16: step response for engine speed equal to 4000 rpm and intake manifold pressure equal to 50 kpa in different egr valve positions for long-route EGR. The EGR valve opens which leads to that the engine torque decreases, residual gas mass flow increases and exhaust manifold pressure decreases.

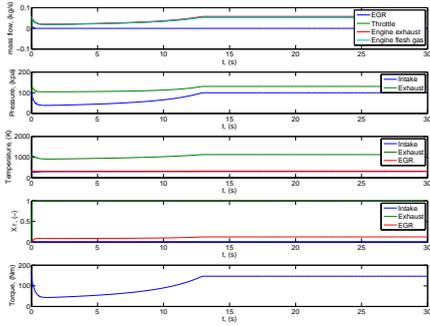
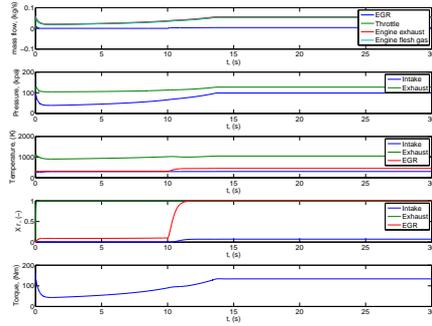
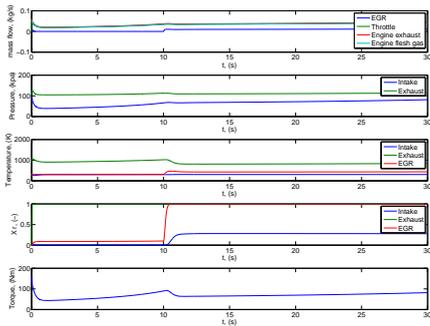
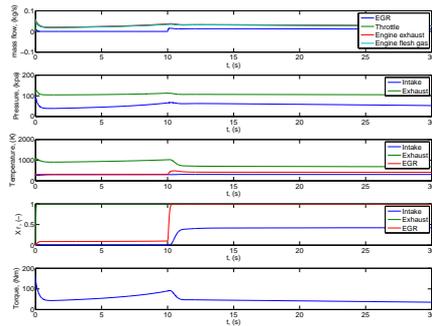
(a) $u_{egr}=0$ for long-route EGR(b) $u_{egr}=10$ for long-route EGR(c) $u_{egr}=50$ for long-route EGR(d) $u_{egr}=100$ for long-route EGR

Figure A.17: step response for engine speed equal to 4000 rpm and intake manifold pressure equal to 100 kpa in different egr valve positions for long-route EGR. The EGR valve opens which leads to that the engine torque decreases, residual gas mass flow increases and exhaust manifold pressure decreases.

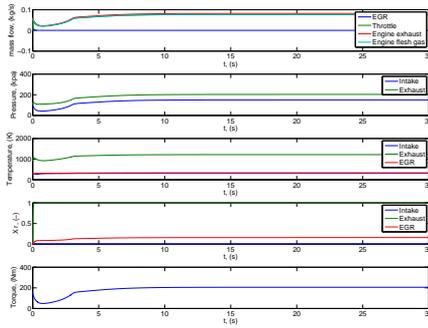
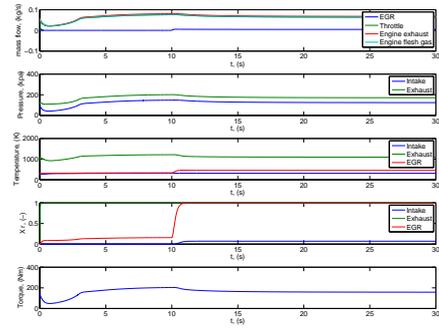
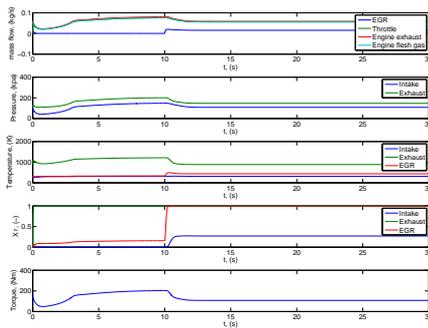
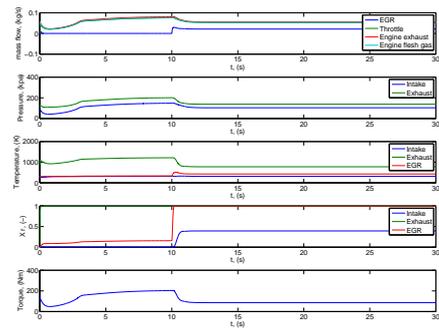
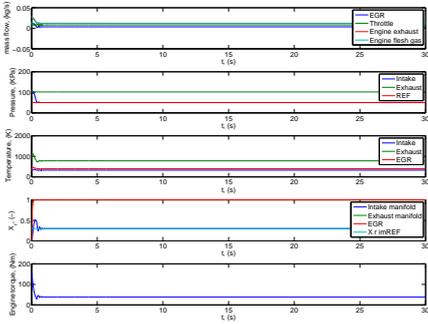
(a) $u_{egr}=0$ for long-route EGR(b) $u_{egr}=10$ for long-route EGR(c) $u_{egr}=50$ for long-route EGR(d) $u_{egr}=100$ for long-route EGR

Figure A.18: step response for engine speed equal to 4000 rpm and intake manifold pressure equal to 150 kpa in different egr valve positions for long-route EGR. The EGR valve opens which leads to that the engine torque decreases, residual gas mass flow increases and exhaust manifold pressure decreases.

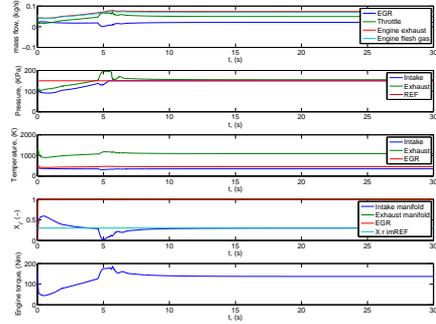
B

Controllers simulation and Comparison result

In this Appendix B, the rest step response figures are presented here for the controllers' performance according to the operation points shown in the chapter 3. the rest of the comparison between long and short routes is presented according to the operation points shown in the chapter 4.

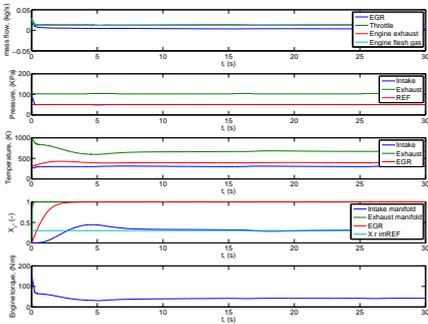


(a) Engine speed = 2000 rpm and intake manifold pressure = 50 kpa

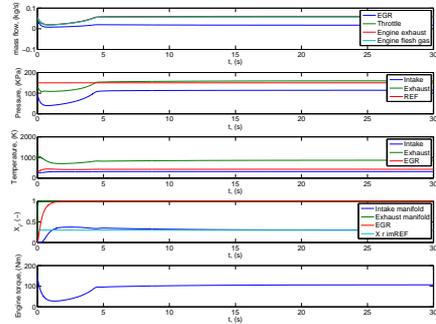


(b) Engine speed = 4000 rpm and intake manifold pressure = 150 kpa

Figure B.1: PID simulation for important parameters with residual gas fraction equal to 0.3 as reference input value. The adapted model shows very good fit to reference value. These figures are simulated based on short-route EGR



(a) Engine speed = 2000 rpm and intake manifold pressure = 50 kpa



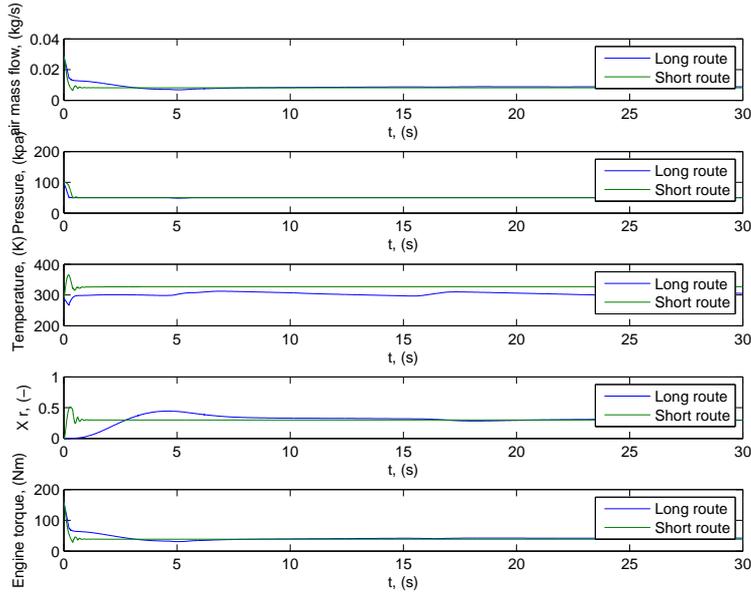
(b) Engine speed = 4000 rpm and intake manifold pressure = 150 kpa

Figure B.2: PID simulation for important parameters with residual gas fraction equal to 0.3 as reference input value. The adapted model shows very good fit to reference value. These figures are simulated based on long-route EGR

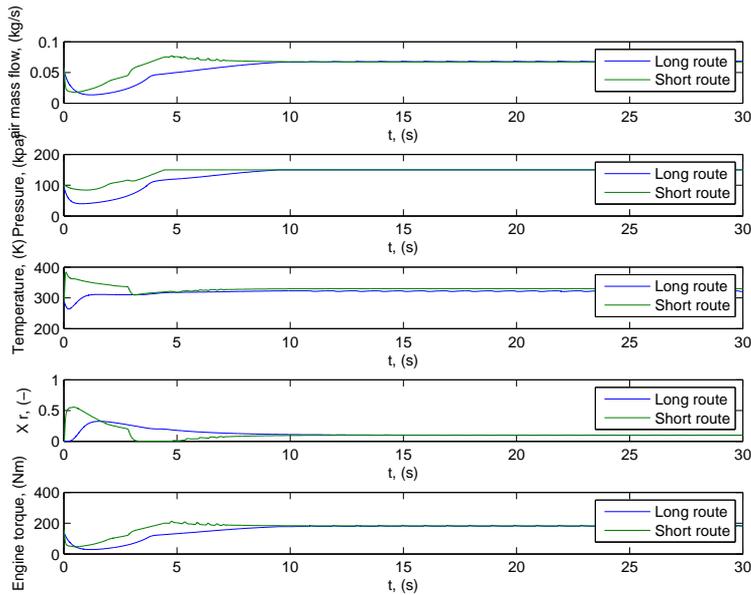
C

Routes Comparison result

In this Appendix C, the rest of the comparison between long and short routes is presented.



(a) Engine speed = 2000 rpm, intake manifold pressure = 50 kpa and residual gas fraction = 0.3



(b) Engine speed = 4000 rpm, intake manifold pressure = 150 kpa and residual gas fraction = 0.1

Figure C.1: step responses for important parameters for routes comparison
The adapted model shows very good fit to reference value.

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