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JAKUB JAROŠ

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STATUTORY DECLARATION

I hereby declare that the master's thesis entitled "Development of GT-Power Model of a Gasoline Engine with Low Pressure EGR Suitable for Model-Based Predictive Control of an Air Path" submitted to Czech Technical University in Prague was written by myself under the guidance of prof. Ing. Jan Macek, Dr.techn.Ir. Arief Hariyanto, Francois-Alexandre Lafossas, Kotaro Maeda, Dr. Remi Losero and Dr. Nicola Pompini. I have stated all the resources used to elaborate this thesis.

11th January 2019 in Prague

Jakub Jaroš

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I would like to express my deepest gratitude towards my supervisors, prof. Ing. Jan Macek, DrSc., and Dr.techn.Ir. Arief Hariyanto for their continuous and invaluable guidance provided throughout my research activity.

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ANNOTATION

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ABSTRACT

To develop an engine complying emission limits, reduce fuel consumption and enhance engine power output is a car manufacture's task of the last years. New emission limits make this task even more challenging and to fulfil them requires new approaches. Recirculation of burned gases is promising way how to reduce harmful emissions and improve fuel consumption in case of a spark ignition engine. Introduction of EGR system increase degrees of freedom and it makes engine control more and more complex. To be able to use potential of EGR fully requires to have a precise calibration. Increasing need of precise calibration and higher amount of degrees of freedom increase time demand for Engine management system calibration and validation. If the calibration and validation would be performed on engine test bench only, it would be lengthy and costly.

For this reason, simulation software is used to lower time demand on a test bench for engine management system calibration and validation.

Aim of this work is to create models of spark ignition engine to support Model predictive control (MPC) development. Models are needed in two levels of accuracy. Trend models are part of MPC development where appropriate MPC is selected. Detailed model is needed for MPC validation before implementation of MPC on real engine application.

This work explains which models are needed and what are their role in validation process. It also describes how the models have been created and which data is needed.

CONTENT

1. INTRODUCTION

We live in a society where we do not have to struggle to survive. Nowadays, high-quality food, potable water, good medical care and supporting social system have become a natural thing in developed part of the world. Thanks to the fact, that we live a comfortable life, we can focus more on the impact of our acting. We analyse our everyday activity and we evaluate if our acts have negative effects on quality of our lifes. If so, we try to find the ways how to minimalize them. Automotive industry and vehicle operation have been placed under scrutiny in the last decade because of the greenhouse gases and harmful emissions.

Since the beginning of the 20th century we have been witnesses of a great boom of automotive industry. Highest car ownership per capita was (and still is) in the USA where we can find 831,9 cars per thousand people in 2016. For instance, in the Western Europe were registered 606 cars per thousand people in the same year. The peak in the USA was in 2007, when this number reached 844,5 but the rest of the world, apart from the Western Europe, still has a growing trend of car ownership per capita. [1]

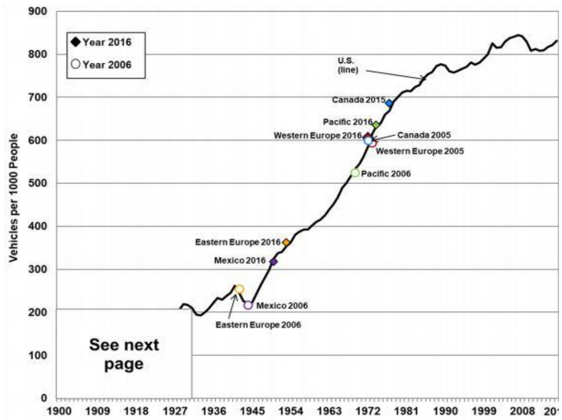

Figure 1: Vehicles per thousand people: U.S. (over time) compared to other countries (in 2006 and 2016) [1]

Country/Region	Vehicles per 1,000 people 2006	2016
Africa	25.2	38.9
Asia, Far East	49.7	105.6
Asia, Middle East	99.8	147.4
Brazil	129.0	209.3
Canada	599.6	686.3
Central & South America	102.4	174.7
China	26.6	141.2
Europe, East	254.4	362.1
Europe, West	593.7	606.0
India	11.6	36.3
Indonesia	31.7	87.2
Mexico	217.6	317.5
Pacific	524.7	634.9
United States	840.7	831.9

Figure 2: Vehicles per thousand people in selected countries/regions, in 2006 and 2016 [1]

It was also in the USA where we could observe first significant impact of car operation on the environment. The results of deteriorating air quality led to the introduction of the first emission limit in the state of California in 1960s. Since then we have been continuously improving efficiency of the cars – reducing the fuel consumption and harmful emissions. In this field have been done a lot but we are facing to new challenges. Even though it takes 100 today's cars to match an average emission of one car made in 1970s [2], we are not able to lower overall emissions. The main reason is the increase of total amount of cars. That is why we need to push fuel consumption and emissions as low as possible.

In 2018, the European Commission announced a CO2 emissions target for the years 2021 and 2030. Current combustion engines need to be improved in terms of efficiency, coupled with electric motor or completely replace by electric motor to meet the criteria.

We need to lower the consumption of fossil fuels to reduce CO2 emission. However, we should keep in mind that even electricity represents certain amount of CO2 emissions and depends strongly on the country where the electricity comes from. Hydrogen is not 'for free' either.

It is already more than twenty years since Toyota started selling the first hybrid car - Prius. [3] It was a first mass production car with hybrid technology and it started the mindset transformation of the society from fossil fuels to clean energy.

Each configuration (combustion engine, combustion engine + electric motor, electric motor, fuel cells) has its benefits and disadvantages, both supporters and opponents. If OEMs want to keep combustion engines 'alive', they will have to make them more attractive and environmentally friendly.

The automotive industry is a highly competitive business. Manufactures are under legislative pressure to meet emission limits, plus they need to deliver a new car model sooner than competitors, in a good quality and for reasonable price. Car manufactures spend a lot of time, effort and money in development of new solutions to keep pace with competitors and fulfil all the criteria at the same time.

For Europe automotive industry is not just an ecological issue but also an economical one. In 2016, in EU28 have been produced 16,5 mil. of new passenger cars and 5,5 mil. of them have been exported, worth over 125 billion euros. In Europe are located 140 car assembly plants and they represent 24 % of worldwide car production. In Europe was 256 million of passenger cars on the roads and each made around 13 000 km/year in average. [4] From the economical point of view, it is extremely important to keep the European car manufactures the key players.

Recirculation of the burned gases in spark ignition engines seems to be one of the promising ways how to achieve a lower fuel consumption and produce less of the harmful emissions. The drawback of this solution is an additional actuator that affects air mass flow into the engine. It would take much more time to create map-based controller in comparison with an engine without EGR architecture. That is the reason for considering an advanced type of control. A model predictive control is one of the methods which sounds promising to reduce the calibration time but requires models suitable for an online simulation and a model predictive control.

2. EUROPEAN LEGISLATION

2.1. History of emission limits in Europe

The first emission limit in Europe has been applied in 1971 and was called '*EHK 15*'. At the beginning this guideline considered only emissions of HC and CO. Limit for NOx emission was introduced in the following years. At the end of the 1980s, EHK 15 had been replaced by a new '*EHK 83*' which took an effect in 1991 in Czechoslovakia. [5]

Today we can hear about another emission regulatory so-called '*EURO*' plus version of regulation. This regulation has been based on EHK 83 and during the years, limits got more strict. We can see in a table below how was EURO changing between EURO 1 and EURO 6. [6, 7, 8] Values are in g/km if not mentioned otherwise and are related to passenger cars. Busses, trucks and vans have different limits.

SI engine

Year	Name	CO	NO_x	$HC + NO_x$	PM	PN [#/km]
July 1992	Euro 1	2,72	-	0,97	-	-
Jan 1996	Euro 2	2,2	-	0,5	-	-
Jan 2000	Euro 3	2,3	0,15	0,2	-	-
Jan 2005	Euro 4	1	0,08	0,1	-	-
Sep 2009	Euro 5	1	0,06	0,1	0,005*	
Sep 2017	Euro 6	1	0,06	0,1	0,005*	$6x10^{11}$**

Figure 3: Emission limits for car with SI engine in Europe

*Direct injection only

** $6x10^{12}$ for the first three effective years

CI engine

Year	Name	CO	NO_x	$HC + NO_x$	PM	PN [#/km]
July 1992	Euro 1	2,72	-	0,97	0,14	-
Jan 1996	Euro 2	1	-	0,7	0,08	-
Jan 2000	Euro 3	0,64	0,5	0,56	0,05	-
Jan 2005	Euro 4	0,5	0,25	0,3	0,025	-
Sep 2009	Euro 5	0,5	0,18	0,23	0,005	$6x10^{11}$
Sep 2017	Euro 6	0,5	0,08	0,17	0,005	$6x10^{11}$

Figure 4: Emission limits for car with CI engine in Europe

As you can notice, level of CO2 emission is not a part of the EURO emission limit. Amount of CO2 emitted by cars was not limited for a long time. In December 1993, European Community decided to follow proposal of the United Nations to maintain emission of CO2 in 2000 at the same level as in 1990. [9] It created additional pressure to increase effort to reduce fuel consumption because the level of emitted

CO2 is related with usage of fossil fuels. Based on that, in 1995 was adopted Community Strategy for reducing CO2 emissions from cars which consist of voluntary commitments from the car industry to cut emissions, improvements in consumer information and the promotion of fuel-efficient cars. In 1998, the European Automobile Manufacturers' Association adopted a commitment to reduce average emissions from new cars sold to 140 g of CO2/km by 2008. [10] Because this voluntarily accepted commitment did not produce adequate results, a new regulation was implemented in 2009 and established a target – average CO2 emission of new passenger cars had to be under 130 g of CO2/km in 2015. This target was met by manufactures without struggling in advance. [11]

Unfortunately, those number are a bit virtual, coming from a laboratory and they are based on reference testing. The reference testing does not consider all driving conditions and driving styles. Due to this fact, the reference testing is changing to be as close to reality as possible. We can see an increasing trend of deviation between the laboratory measurement and on road testing. Straightening emission limits should guarantee lower emission production but recent study shows, that, for example, NOx emissions of on-road testing and laboratory testing are not the same and in case of an on-road testing can be up to sixteen times higher. [12] Another study confirms this trend and shows even worse result.

This difference does not appear only for NOx. With the new mandatory emission target from 2009 introducing a limit of 130 g of CO2/km for 2015, disagreement appeared also for the emission of CO2, as shown in Figure 5.

Authorities are aware of this fact and they try to create a testing protocol which would follow real driving condition to close this gap.

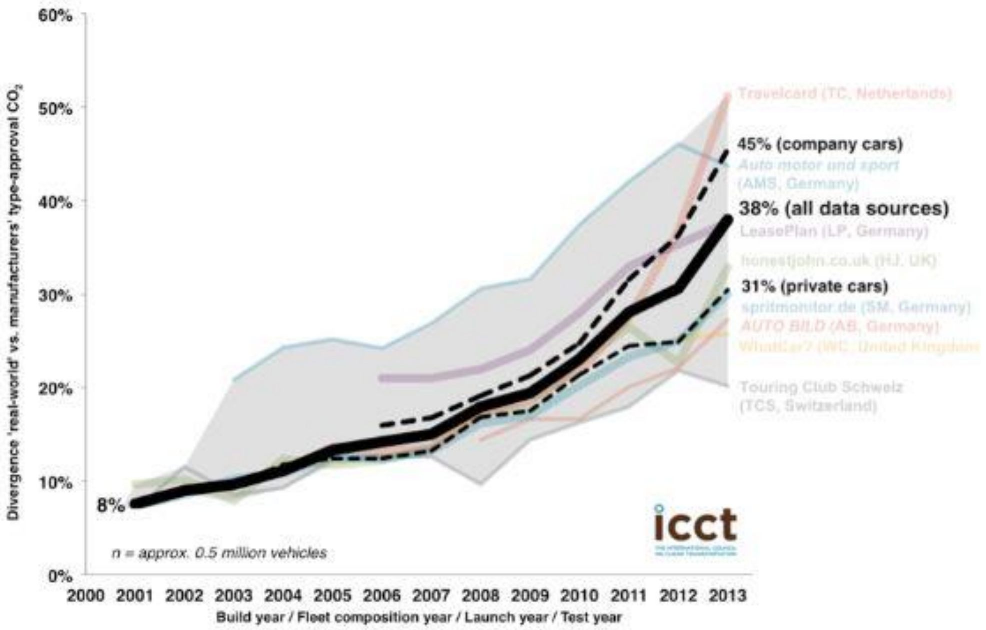

Figure 5: Increasing gap between real-world and official CO2 emissions / fuel consumption for new cars in the EU [44]

2.2. Emission measurement

The laboratory testing follows a predefine protocol which specify how the test should be performed. The testing is done on a chassis dynamometer which functions like a treadmill for vehicle. The emissions are sampled based on constant volume sampling technique. [13, 14] The results are reported in units of g/km except PN, for which we use #/km. The limits for a given regulation are shown in Figure 3 and Figure 4.

2.2.1. UDC + EUDC (MVEG-A)

This emission test cycle was used for approval of EURO 1 and EURO 2 emission limit. It supposed to reflect a driving style in the European cities. The test started with a cold engine but before sampling was 40 seconds warm-up by idling. UDC consists of four ECE cycles during each car gradually accelerates. EUDC represent higher speed and more aggressive driving. [15]

Characteristics	Unit	ECE 15	EUDC	NEDC†
Distance	km	0.9941	6.9549	10.9314
Total time	s	195	400	1180
Idle (standing) time	s	57	39	267
Average speed (incl. stops)	km/h	18.35	62.59	33.35
Average driving speed (excl. stops)	km/h	25.93	69.36	43.10
Maximum speed	km/h	50	120	120
Average acceleration[1]	m/s^2	0.599	0.354	0.506
Maximum acceleration[1]	m/s^2	1.042	0.833	1.042
† Four repetitions of ECE 15 followed by one EUDC [1] Calculated using central difference method				

Figure 6: Characteristics of ECE 15, EUDC and NEDC [15]

2.2.2. NEDC (MVEG-B)

NEDC is used since EURO 3 (2000) and is sometimes called MVEG-B. It was derived from MVEG-A by removing the idling before the start of sampling. It starts just after engine start-up. [15] As we can notice the driving profile is quite smooth and monotonous. It was criticized, even before implementing, in 1998 by Swedish researcher Per Kågeson, who predicted significant

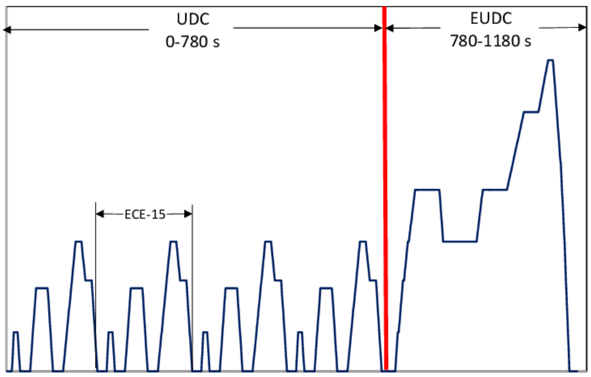

Figure 7: NEDC composition [45]

difference between measurement and reality and raised concern about cheating by software modification. [16]

2.2.3. WLTP

The increasing gap between on-road emission and laboratory emission accelerated the need to have a test procedure that reflects reality more than the current one. WLTP is more dynamic than NEDC and cover bigger area of an engine operation range. The WLTP was developed based on the real driving data gathered from around the world and contains data from city as well as highway. You can find a comparison of NEDC and WLTP in figure below. [17]

	WLTP	NECD
Start temperature	Cold	Cold
Cycle time	30 min	20 min
Stationary time proportion	13 %	25 %
Cycle length	23,25 km	11 km
Speed	Average: 46.5 km/h Maximum: 131 km/h	Average: 34 km/h Maximum: 120 km/h
Drive power	Average: 7.5 kW Maximum: 47 kW	Average: 4 kW Maximum: 34 kW
Influence of optional equipment and air-conditioning	Optional equipment is taken into account for weight, aerodynamics and VES requirements (no-load current). No AC	Not considered at present

Figure 8: A comparison of WLTP and NEDC

Based on the comparison from Figure 7, Figure 8 and Figure 9, we can see, that the new test procedure last about 50 % longer but the cycle length is more than twice longer. Drive power together with average speed increase significantly. The driving resistance is arranged accordingly to the real weight of the vehicle. Speed profile is more dynamic, almost without sections of constant speed.

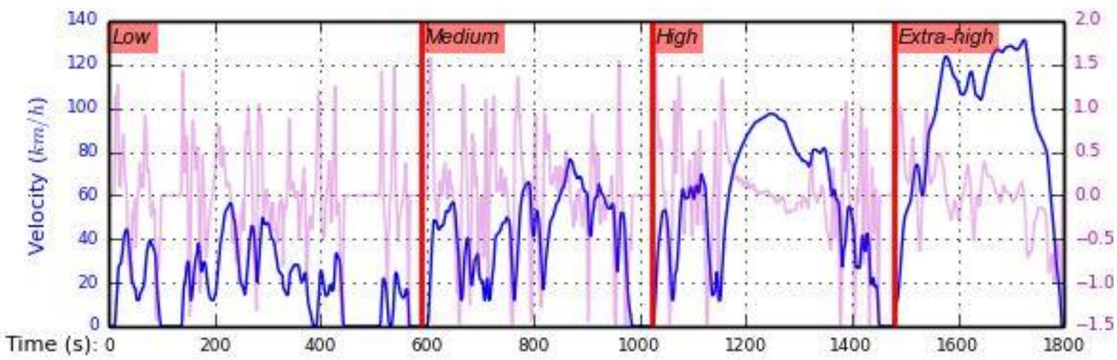

Figure 9: WLTP driving cycle [18]

2.2.4. RDE

The real driving emissions is a testing on the road in real-life conditions. It is an additional test along with the WLTP measurement and it should secure that real emissions are the same as the laboratory emissions. There is no predefined testing protocol, therefore entire engine operation range can be used during a measurement. Combination of the real driving conditions and absence of a predefined testing protocol makes RDE extremely variable. It will create various boundary conditions such as weather, slope, traffic, driving style etc. in the measurement and even if two cycles can be RDE compliant, one vehicle can pass and the other not due to the different severity.

We are using a portable emissions monitoring system (PEMS) to measure the emissions. The test has to last from 90 to 120 minutes and is consist of three phases – urban, rural and highway. Each phase must not be shorter than sixteen kilometres. [19] You can find the other details in Figure 11.

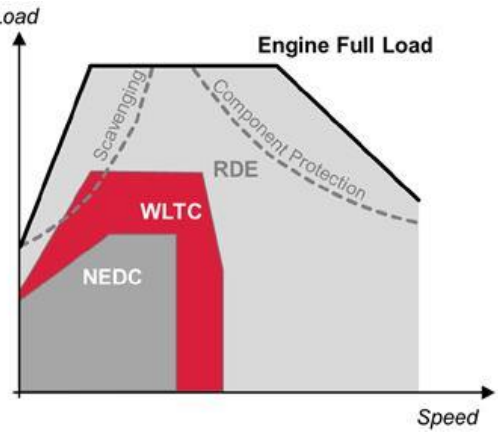

Figure 10: Comparison of emission measurement [21]

THE RDE is going to be introduce in two steps:

Step 1: During testing, a conformity factor 2.1 will apply for NOx emission. That means car can exceed laboratory limit up to 2.1 times during RDE. An error margin factor 0.5 applies also to PN emission. It is in effect of from September 2017 for a new model approval, from September 2019 for all new vehicles. [20, 19]

	Urban	Extra-urban	Highway
Cycle repartition (+/- 10%)	29% < ratio <= 34%	33%	33%
Speed	V < 60 km/h	60 km/h <= V < 90 km/h	V > 90 km/h
Max. speed (+/- 15 km/h for less than 3% of driving time)	-	-	145 km/h
Average speed (stops included)	15 km/h <= Vmoy <= 30 km/h	-	-
Minimum travelled distance	16 km	16 km	16 km
Altitude difference (beginning/end)	100m	100m	100m
Maximum slope	1200m/100km	1200m/100km	1200m/100kn

Figure 11: Criteria of RDE cycle [21]

Step 2: Starting January 2020, all new approvals must meet conformity factor 1.0 but with an error margin of 0,5. As a result of this, the emission of NOx during RDE can be up to 1.5 higher that the laboratory limit. [6]

The engine will operate in problematic areas during RDE measurement, as shown in Figure 10. In low speed/high load is the zone of scavenging. It is used to cool the engine and remove residual gases from combustion chamber to mitigate knock. Scavenging presents a problem from term of CO and HC emission. 'Component protection' in the high speed/load zone is linked with enriching to protect the turbine from too high temperature. The engine is running rich and produces CO and HC emission above the limit. The manufactures will have to find the way how to lower combustion temperature without scavenging, respectively enriching, to be able to pass the RDE emission test. The EGR could be a feasible way how to achieve effective cooling without enriching. This effect is described more in detail in 3.4.

2.3. Current emission limits in Europe and near future

Since September 2017, all new approvals are certified based on WLTP and since 2018 is mandatory for all new car registration to get the EURO certificate. However, the CO2 emission limit is based on NEDC. The emission of CO2 has been reduced significantly in last two decades and should be reduced even more until 2030. In October 2018 was agreed reduction of 15 % until 2025 and 35 % until 2030 compared to year 2021. [22] We can see the progression of CO2 emission target in following figure.[1]

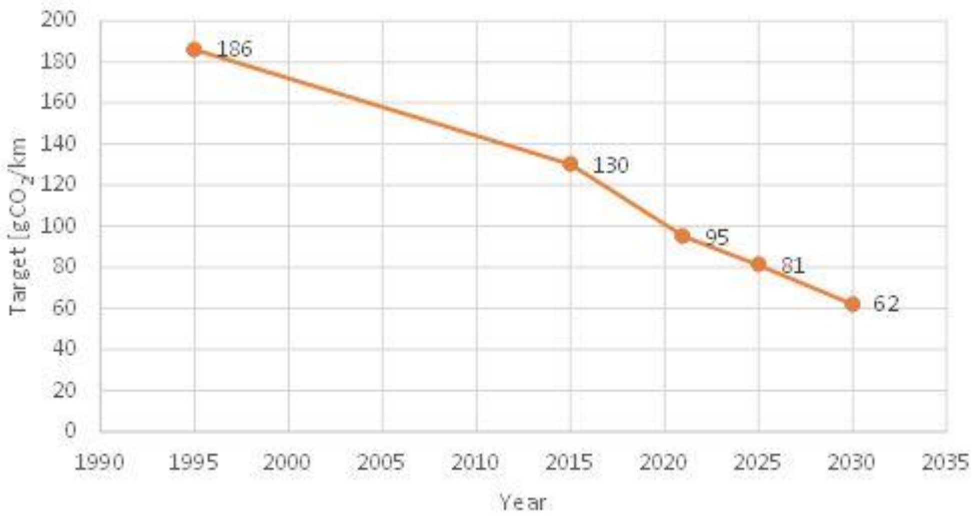

Figure 12: Target of CO2 emission g/km [22] [23] [24]

In 2017, the average new car emissions were 118.5 g of CO2/km [23], therefore we need to reduce CO2 emission about 48 percent in the next 13 years.

The manufactures need to find the new ways of the engine operation and control to have even a chance to reach this target together with EURO standards. It will be necessary to invest a lot of money in R&D to develop and validate the new solutions. One of the possible improvements frequently discussed nowadays could be to introduce EGR to SI engines.

It will be described on following pages on our application which is 2.0 spark ignition turbocharged engine. As already mentioned, EGR in spark ignition engines is one of the meaningful modifications which has a potential to improve engine efficiency. On top of that MPC is new approach of air path control and needs to be evaluated.

[1] Value from 1995 is not CO2 target but average CO2 emission of sold cars.

3. BENEFITS OF EGR IN SI ENGINE

3.1. Introduction

EGR is an acronym for Exhaust Gas Recirculation. The introduction of exhaust gases into the intake pipe leads to a diminution of air fraction at the fixed throttle position.

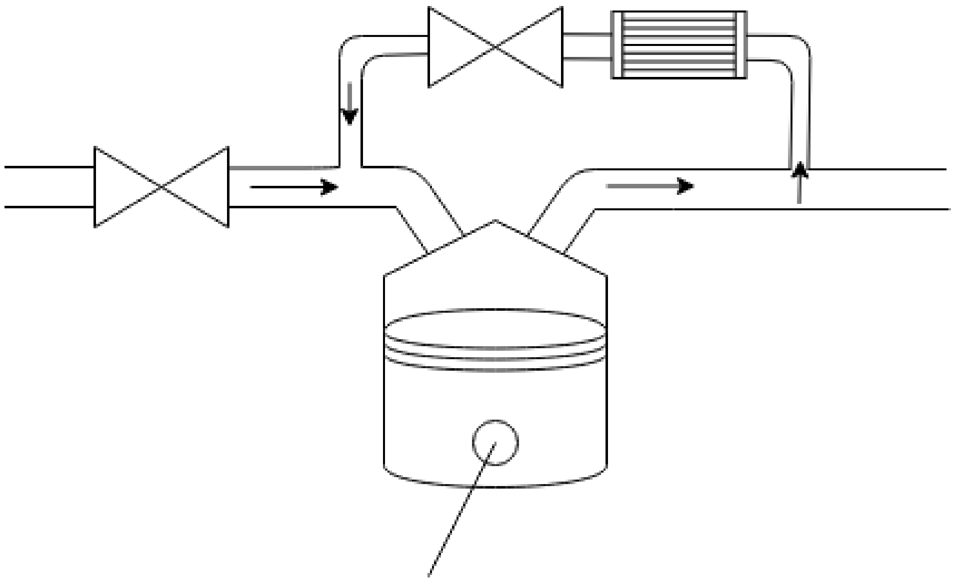

Figure 13: EGR principle

The principle of EGR is to take part of burned mixture downstream a cylinder and re-introduce it back to the intake. The pressure difference between the intake and the exhaust side is a natural driven force for exhaust gas recirculation. Exhaust gas recirculation is not possible if no pressure drop is available.

An EGR valve is used to control amount of EGR in fresh charge. The EGR line is also typically equipped by an EGR cooler. Function of the EGR cooler is to cool the recirculated gases down to lower temperature which increase a potential to reduce the temperature during the combustion. Lower combustion temperature is not the only benefit of EGR.

The EGR has the following benefits in the SI engine application:

- Reduction of pumping losses.
- Reduction of NOx, PM and CO formation.
- Reduction in the knock tendency and enriching.

It results in higher efficiency and it helps developers to get closer to meet the tightenning criteria.

3.2. Reduction of pumping losses

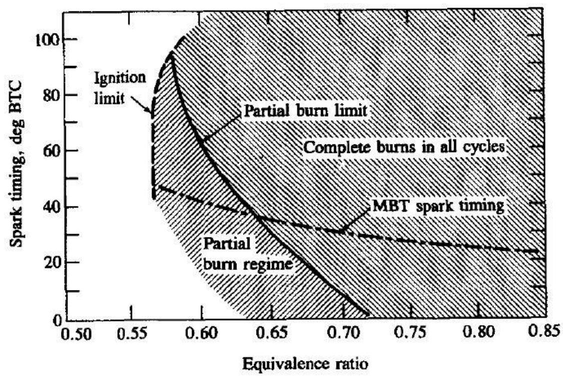

Figure 14: Actual limiting combustion regimes for lean-operating engine. 1200 rev/min, volumetric efficiency ≈ 60 percent, methane fuel, 40 mJ spark energy, 2.5 ms spark duration [24]
**Equivalence ratio = 1/lambda*

Spark-ignition engines operate around lambda 1, where lambda represents the ratio of air/fuel to stoichiometric one. When the lambda is greater than 1, it means we have excess of air and we are running *'lean'*. If opposite, we are running *'rich'*. To maintain lambda ≈ 1 during the different engine load, we have to adjust amount of fuel according to the quantity of air at that operating point. The engine would have higher HC and CO emission and higher fuel consumption when operating rich. On the other hand, too lean mixture would cause misfire or partial burn. [25] We use a throttle valve during partial load to be able to control the air mass flow. The throttle valve creates an additional restriction to the stream of air in the intake which reduces the amount of air. The amount of injected fuel is reduced according to the reduced air in order to maintain lambda equals to one. If we need the engine in full load, the throttle valve is wide open and allows the engine to suck as much air as possible. The throttling during partial load creates vacuum downstream the throttle. In the intake phase, cylinder is moving to BDC but it must overcome the vacuum in the intake manifold and in the combustion chamber. We can say that the vacuum tends to keep the piston in TDC. It is a lost for the engine to overcome this force. The losses during the charge exchange phase are called pumping losses.

EGR allows us to throttle less. As mentioned above, the EGR leads to diminution of the air fraction at the fixed throttle position. We need to open the throttle more to get the same amount of air as originally to reach the same power output with the EGR. The throttle is open more, in the intake manifold is lower vacuum and it results in lower pumping loses. This solution also has its limit. With an EGR rate higher than 25 percent (this limit is not exact and varies with the operating point) [26], the combustion becomes unstable. Slow burns, misfire and partial burn may occur. It leads to worse emissions and harsh behaviour. [25]

EGR brings benefits during a partial load when throttling is needed. There is no pumping loss reduction during a full load.

3.3. Reduction of NOx, PM and CO formation

The effect of NOx reduction is already well-known from CI engines and in case of a SI engine works as well. Reduced combustion temperature helps us to lower NOx, PM and CO emissions.

Effect on CO formation

Reduction of pollutants formation is in a wide spread of engine speed and load. In high load/speed region is CO limited by better air to fuel ratio. In the mixture is lack of air because of an enriching. The enriching is needed to lower exhaust gas temperature to protect the turbine. We can substitute enriching by EGR and be closer to lambda = 1.

CO formation in the remaining engine operation range is related with lower combustion temperature. The CO and CO2 ratio formation is temperature dependant. Higher temperature, higher CO to CO2 formation during combustion. Temperature drops due to the EGR in the most areas of the engine operating range and it lowers CO to CO2 ratio. The amount of produced CO is reduced as a result. [27]

Effect on NOx and PM formation

Formation of NOx is the highest in a slightly lean regime [25, 14] where temperature is high enough and in combustion chamber is excess of air, respectively oxygen. The temperature is not high enough if the mixture is too lean. On the other hand, redundant fuel acts as a heat sink during rich operation which reduces combustion temperature. There is also lack of oxygen to allow higher NOx formation. During higher load the engine burns more fuel and it results in higher temperature in comparison with partial load. NO formation is strongly dependent on temperature. NO represents at least 98 % NOx emission of a SI engine. [25]

If we increase the EGR rate, we increase the heat capacity of the cylinder charge per unit mass of fuel. [25] As a result of the higher EGR rate the temperature drops, the heat transfer coefficient is reduced and the NOx emission is reduced as well. [28] Lower combustion temperature reduces also PM formation. The effect of lower temperature on the PM formation reduction is stronger during low load operation. [29]

It is beneficial to dilute the cylinder charge by the exhaust gases. The exhaust gases are product of a combustion and in case of a SI engine they mostly consist of N2, CO2 and water vapour. Water vapour and CO2 have significantly higher heat capacity in comparison with remaining gases, thus they are able to cool the flame temperature more. [30]

The limit for EGR rate last the same as mentioned in 3.2 and we need to keep in mind that it is a kind of trade-off between NOx reduction and flame speed propagation. [26]

3.4. Reduction in the knock tendency and enriching

These phenomena have been slightly mentioned in 2.2 and is related with Figure 10.

Spark ignition engines are related with knock phenomena. Knocking is autoignition of the mixture in the combustion chamber before the flame-front approaches. The result of knocking is high pressure oscillations and high heat release which can result in engine damage or even engine destruction.

Knock usually occur during high loads, low speed. It makes knock performance limiting factor because it defines maximum temperature and pressure of the end gas as well as it limits compression ratio. [25] We have a several ways how to suppress knocking. We can reduce the temperature and/or pressure during the combustion. We can enrich the mixture to achieve a lower temperature – leads to higher fuel consumption, CO and HC emission. In case we want to reduce a pressure, we can reduce the intake manifold pressure by throttling or by opening the wastegate. It is also possible to retard the ignition timing. All these contra measures lead to lower torque output and that is unwanted. We can significantly reduce knock tendency and improve combustion phasing by introducing a cooled EGR. Lower combustion temperature and better phasing lead to lower exhaust temperatures. Then we can avoid of enriching. [27, 29]

The reason why do we need to limit the exhaust gas temperature is the turbine. High temperature operation can cause damage of the turbine blades. The temperature limit varies with the turbine manufacture, but the latest mass production models have the temperature limit around 1050 °C. [31]

All these benefits are improving BSFC in different part of contribution. Many studies claim 5-10 % fuel savings during NEDC. [26]

3.5. Typical areas of benefits

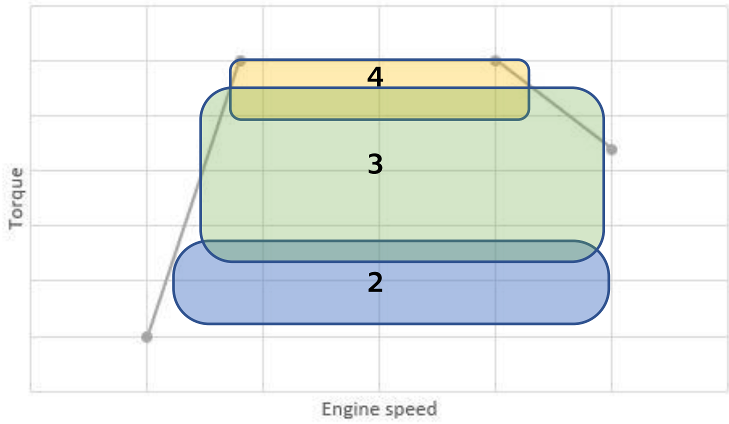

Figure 15: Area of EGR benefits

Figure 15 shows the theoretical dominant area of each benefit presented previously. Each zone is related with the corresponding chapter number:

2 – reduction of pumping losses.

3 – reduction of NOx, PM and CO formation.

4 – reduction in the knock tendency and enriching.

This distribution is significantly simplified. We cannot separate the effect of individual contribution and they are not even uniform. For example, combustion with higher EGR rate is more stable in low speed area than in high speed area. [26]

Some challenges appear if higher EGR is requested in some regions. It will be more discus later in chapter 4.3.

4. EGR line configuration

When designing EGR line we can consider different architectures [30]:

- High-pressure (HP)
- Low-pressure (LP)
- High/low pressure

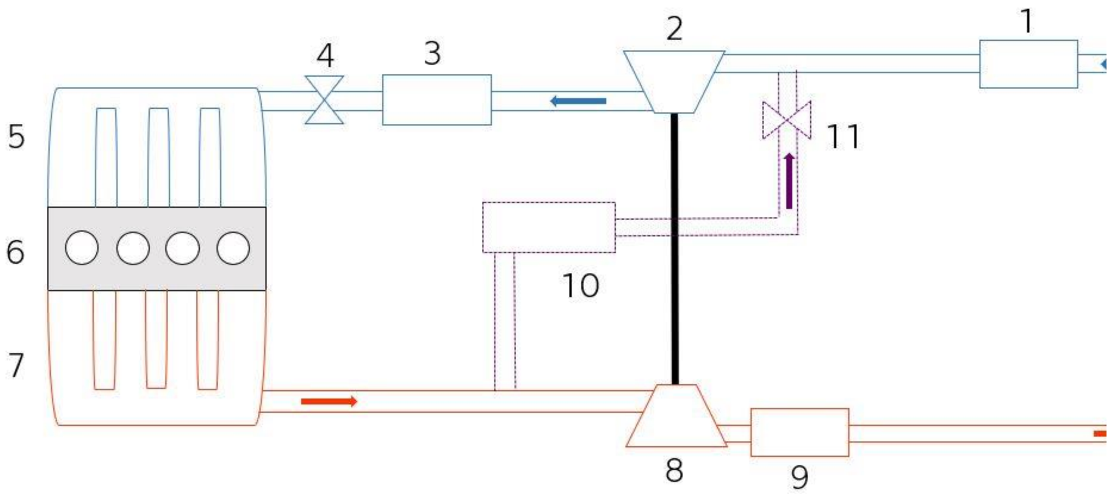

Figure 16: High/low pressure architecture of EGR line

The last-mentioned option is not much preferred in a SI engine and many studies even do not take it into account. The biggest issue is lack of compressor power. As you can see in Figure 16[1], the exhaust gases are extracted before the turbine and introduce back into the intake line before the compressor. The turbine does not provide enough power to compressor to compress intake gases in many points of engine operation. Turbine has a lower gas mass flow than compressor. [30]

Each type of EGR line can be cooled or uncooled. It is favourable to use cooled EGR to further lower the combustion temperature. In case of a LP-EGR, cooled version is more common but hot HP-EGR is more efficient for pumping losses reduction during low load. [27, 29] However, hot HP-EGR becomes useless in higher loads where it is not able to reduce the combustion temperature and can even force knock tendency. The same effect has IGR achieved by VVT. [26]

[1] Legend: 1 – Air filter, 2 – Compressor, 3 – Intercooler, 4 – Throttle, 5 – Intake manifold, 6 – Engine block, 7 – Exhaust manifold, 8 – Turbine, 9 – TWC, 10 – EGR cooler, 11 – EGR valve

4.1. High-pressure

HP-EGR extracts the exhaust gases upstream a turbine and introduces them downstream a compressor. It can be either upstream or downstream an intercooler.

We must assure that EGR cooler is efficient enough to be able to cool the gases down from exhaust temperature to level of the intake temperature in case we introduce exhaust gases downstream the intercooler (Figure 17a). Intake temperature could be considered around 50 °C and constant within operating range.

We need to use a high efficiency EGR cooler with low coolant temperature to be able to achieve such an EGR gas temperature. It makes usage of this architecture more challenging, if we decide not to use it in combination with LP-EGR. [30]

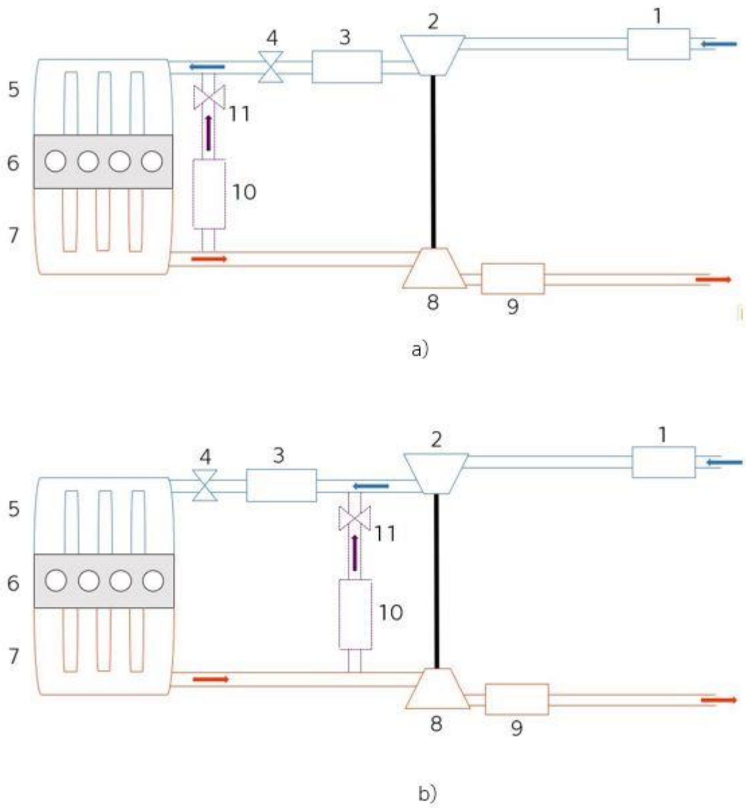

Figure 17: High pressure architecture of EGR line

If we connect recirculated gases upstream the intercooler (Figure 17Chyba! Nenalezen zdroj odkazů.b), we can use intercooler to cool the mixture of fresh air and the recirculated gases to the desired temperature.

The advantage of HP-EGR is faster response in transient thanks to lower volume and higher reduction of pumping losses in comparison of LP-EGR. [32]

The disadvantage is worse mixing of recirculated gases with fresh air and it leads to high cylinder to cylinder variation. [29] It is necessary to take EGR rate into account during compressor design because surge can be a limit, especially in low speed, high load operation. [30]

4.2. Low pressure

In case of a LP-EGR, the separation of exhaust gases is downstream the turbine and they are returned into the intake line upstream the compressor. We can use 'clean' or 'dirty' LP-EGR configuration.

Clean LP-EGR is shown in Figure 18a). Exhaust gases are sourced downstream a three-way catalyst (TWC). The inlet temperature is higher compared to dirty LP-EGR due to the TWC. Small catalyst particles that can damage compressor wheel [30], can be observed, especially at the beginning of the engine operation (first 100 hours). It increases a need of a compressor wheel coating. [32] We can gain the same benefits with lower EGR rate in comparison with dirty LP-EGR. It is because of higher CO_2 concentration. CO_2 has higher heat capacity in comparison with the other gases, as already has been explained in 3.3. Lower amount of the gas is needed to achieve the same temperature reduction. [33]

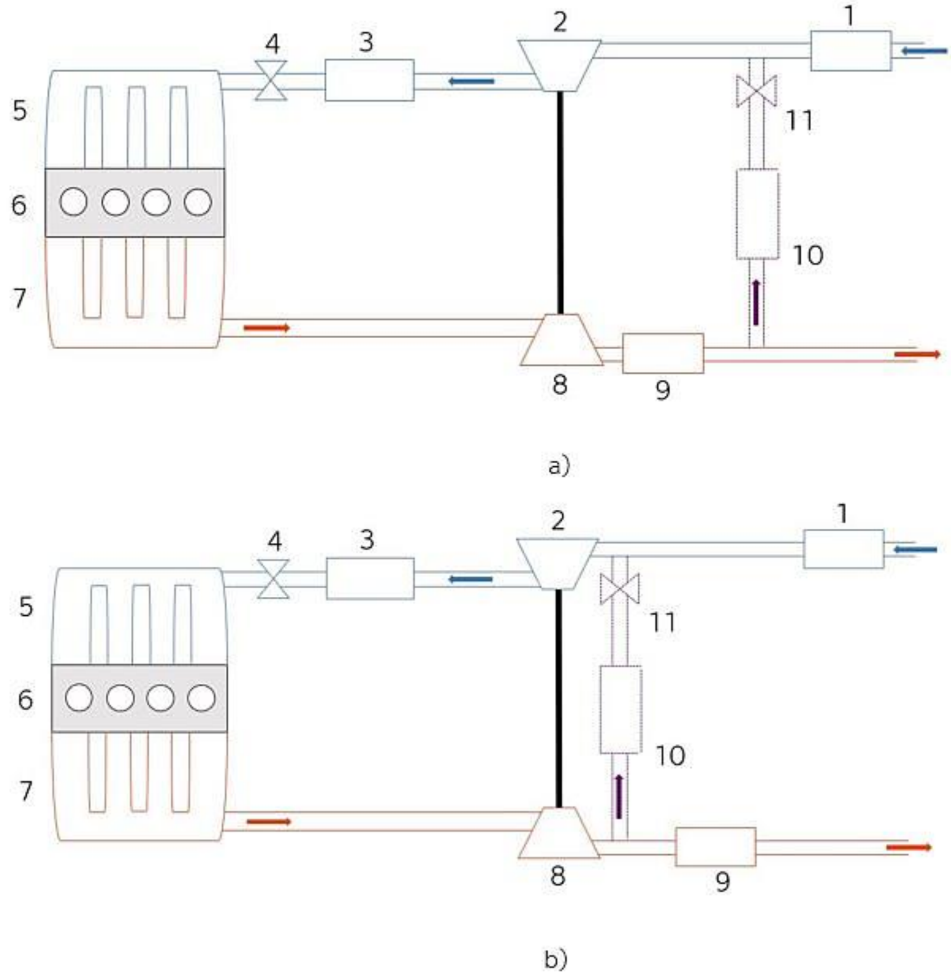

Figure 18: Low pressure architecture of EGR line

Dirty LP-EGR line is connected between the turbine and the TWC, as you can see in Figure 18b). Recirculated gases are raw combustion gases in this case. The benefits of this configuration are faster combustion, improvement of the pressure drop across the turbine, higher pumping losses and HC reduction compare to clean LP-EGR. It enhances the engine peak power. We can also burn CO and HC during recirculation and do not waste them in the TWC. The strongest impact is presence of sticky hydrocarbons upstream the compressor. This is similar situation as if blow-by gases are fed before compressor and requires to be considered for the wheel and the compressor housing design. [32]

Low pressure type of EGR line provides efficient cooling but the pressure difference between the exhaust and the intake line is lower. Hence, we need to select EGR cooler and EGR valve wisely. Only components with good permeability can ensure sufficient flow. [30]

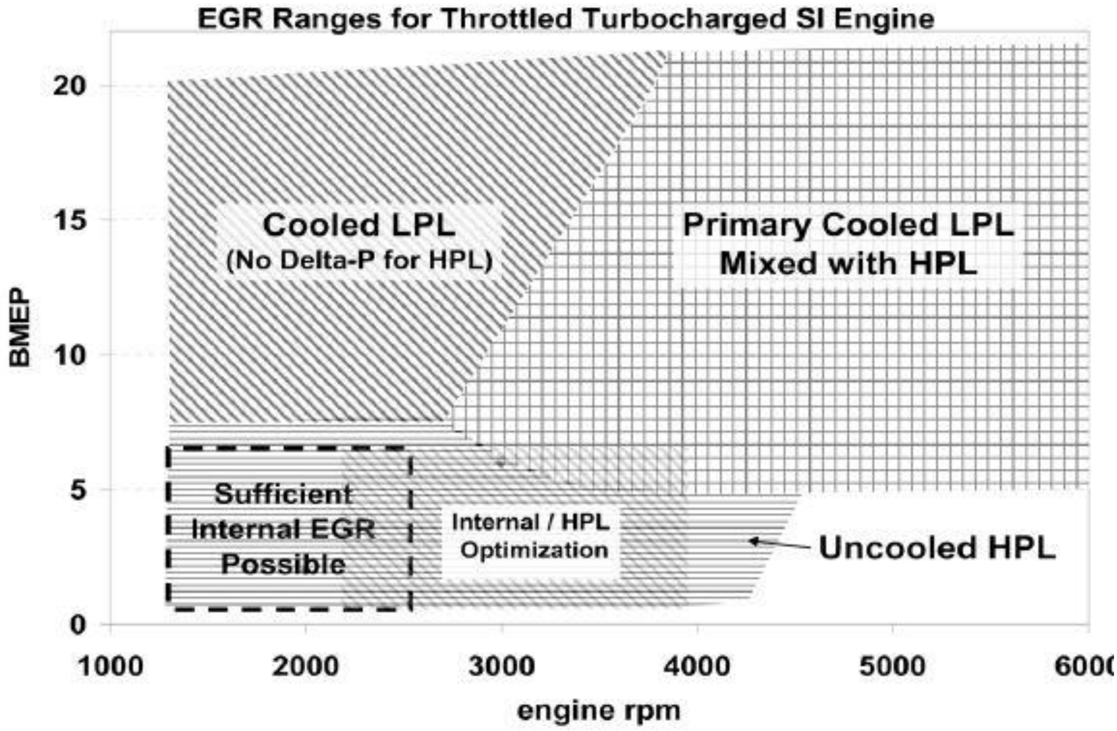

Figure 19: EGR ranges for throttled turbocharged SI engines

The advantage of LP-EGR is better gas mixing and better exhaust gases cooling. It also has better overall performance compared with HP-EGR. [30] The disadvantage is longer response time. Another task is to prevent condensation of water vapour because the water droplets could damage the compressor blades. [32]

To achieve an optimal EGR rate in entire engine operation range, we need to combine different architectures. David B. Roth et al. made the map of possible combination. [32]

4.3. Limitations within operation range

Besides of the intake manifold temperature and the delta pressure limit, there are another limitation of each architecture within operation range.

Area A – low load

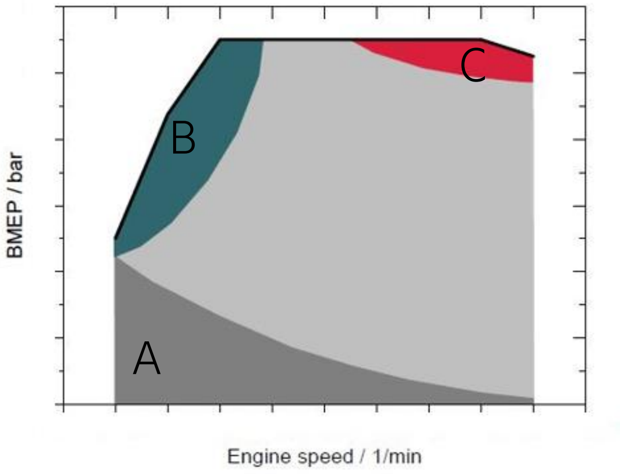

Figure 20: Challenging areas

This area is related with a LP-EGR because of lack of pressure difference. Delta pressure between the exhaust and the intake part is not high enough to allows satisfactory and stable EGR rate during low load/speed operation. This effect can be slightly suppressed by an additional throttle in the intake manifold or a flap in the exhaust line for higher pressure drop. This solution will generate an additional restriction in the intake or exhaust and will be related with additional pumping losses. Therefore, it is necessary to consider whether the gain would be bigger than the cost.

Area B –high load, low speed

The exhaust manifold pressure is lower than the intake manifold pressure in this region. Naturally, for HP-EGR is not possible to recirculate the gases. [30]

There is not have enough boost pressure in case of a LP-EGR to achieve higher EGR rate while keeping high load (depends on boosting device).

It is possible to use a VVT to partially supplement the EGR effect in this region. Scavenging can help us to mitigate the knock as well – it lowers the in-cylinder temperature and removes IGR. Drawbacks of scavenging is higher emission of CO and HC. [26]

Area C – high load, high speed

In this region is high mass flow even without EGR. It needs to be boosted more if it is needed to increase EGR and maintain the same load. It could be a limiting factor because of the maximum turbo speed or turbo efficiency.

All the benefits and limitations have been considered and it has been chosen to use a clean LP-EGR configuration. LP-EGR architecture is discussed in following pages if not mentioned otherwise. This solution provides the better benefits across an engine operation range even though some limits and challenges are present.

5. Challenges related with EGR

5.1. Ideal point of operation

The positive effect of EGR on the combustion process and fuel consumption has been proof by many studies. We can improve the BSFC and emissions but only until the point of reversal. As EGR rate increases, combustion duration increases as well. The combustion becomes unstable and fuel consumption together with emissions of CO rise again. As it has been already said, this point varies with the load and speed but also differs with the engine design. We need to operate an engine as close as possible to this point if we want to use the entire potential of EGR technology. [29]

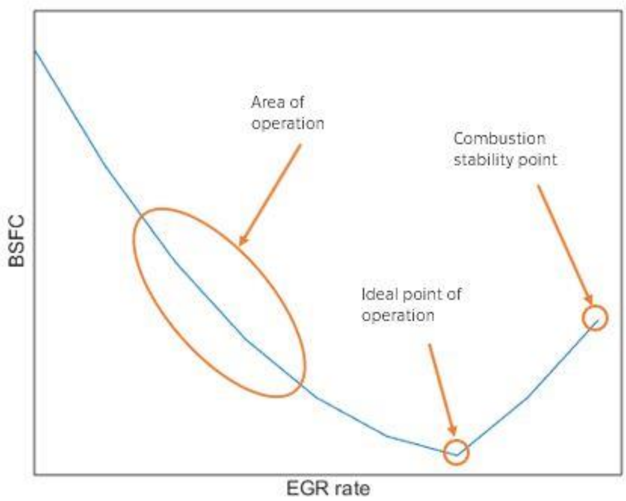

Figure 21: Characteristic points

The optimum and accurate calibration is needed to operate the engine near the ideal point. Accurate estimation and control of airpath system is key for optimal combustion, especially during transient condition. Nowadays, car manufactures do not operate an engine at the ideal point but with the lower EGR rate because of the problems with the estimation. It allows them to assure stable engine operation. Drawback of this solution is the fact that we are not fully using the potential of EGR technology. It is necessary to develop an optimal EGR system architecture, accurate EGR flow estimation and efficient controller to achieve higher benefits. [34, 35, 36]

5.2. Response time

The LP-EGR is associated with higher volume of the airpath compared to the HP-EGR which leads to longer response time. It is because of the longer pipeline and the bigger diameter of pipes. [32] Transport delay between the EGR valve and the cylinder is significant and needs to be considered for controller design to obtain good performance. Our interference in the EGR valve position will have an impact after several engine cycles. It is necessary to maintain suitable combustion conditions for that time, otherwise partial burn or misfire can occur. This is especially relevant for aggressive transient from high load to low load operation.

Low load operation is much more sensitive to EGR rate than initial high load. Siokos in his work [29] describes this problem more in detail. During tip-out, higher throttling leads to the decrease of intake manifold pressure which increase the internal residual gases. A mixture of fresh air and EGR gases from original state is still in the intake manifold. The combination of EGR and internal residual can create higher total dilutional than is suitable for an efficient combustion with actuator setup for final state. He shows result of a simulation where it takes fourteen engines cycles at 3000 RPM to observe EGR rate decrease.

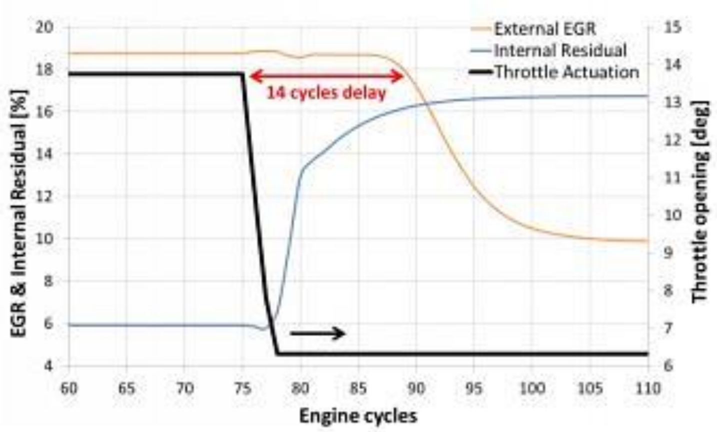

Figure 22: EGR and internal residual response for tip-out at 3000 RPM [29]

Feilong Liu et al. focused on possible solution in their study. [36] Fuel cut-off is a common technique how to adopt to tip-out. It also has a constrains such an emission restriction and possible catalyst damage. During engine re-start it is necessary to run the engine rich because of the catalyst poisoning. It is necessary for the efficient NOx reduction, even though it results in worse BSFC. Another mentioned technique is advance spark timing together with adjustment of additional actuators – such as the EGR valve, throttle, wastegate; or higher valve overlap achieved by VVT to evacuate the EGR gases from the intake manifold faster. These methods have the constrains and some difficulties to be used widely.

5.3. Additional challenges

Also, the ignition system needs to be modified for EGR. The ignition coil energy needs to be greater than that of a conventional engine to secure correct and fast ignition.

Compressor blades can be damaged by water droplets in case of a LP-EGR configuration. This water droplets are condensed water vapour contained in EGR gases. We need to make sure that we are above the condensation conditions or that we can remove the droplets from the mixture before they approach the compressor. It can be achieved, for example, by a venturi, as mentioned by David B. Roth in his paper. [37]

5.4. Engine control

We need to be able to control the wastegate, throttle and EGR valve position for the correct amount of intake gas and its composition. It becomes a complex task due to the coupling effect and various transition delay for each operating point. As a result, we get a multiple-input multiple-output (MIMO) system. It means that each quantity is not a function of only one actuator but multiple actuators. For example, if the EGR valve gets opened, the EGR rate increases. It will lead to dilution of the intake mixture, amount of fresh air will drop and output power in be reduced as a result. We have to do several contra measures to keep the same power output. For example, the throttle needs to be wider opened and boost pressure could be needed to increase which will required to increase compressor speed.

Advanced controller, such a model-based control (MBC), seems to be good approach to handle this.

In my thesis, I will focus on the airpath control of a spark-ignition turbocharged engine with LP-EGR. It means, my models will support control of the air flow, intake manifold pressure and burned gas ratio. To control this, we can use several actuators. One of the most important are EGR valve, throttle and wastegate. Engine itself has limits which need to be also considered. Constrains can be, for example, maximum turbine speed, maximum throttle angle, EGR valve angle or wastegate position.

Unfortunately, this is not the only challenge we need to deal with. Beside of correct air flow and EGR rate, it is necessary to correctly adjust additional actuator – spark timing, valve timing or fuel amount. It is not a part of my thesis but needs to be consider for control of a real plant.

All of this makes control of an engine with LP-EGR very challenging and complex task.

We must be able to measure or estimate EGR rate as precise as possible. Based on the EGR rate we have to adjust spark timing, valve timing, amount of fuel and actuators position to achieve desired air mass flow and engine performance. A driver wants to have a short engine response to his power request; therefore, response time is not only important for the correct control but also for good drivers feeling. All those quantities are in an interaction to each other and cannot be controlled separately but only with respect to others.

It would be possible to use classical control approach which consists of maps and PIDs. However, it would be very expensive and time demand to get well calibrated maps. Even though, time and money would be spent during calibration process on an engine test bench, we cannot be sure about good performance/accuracy.

The model-based control seems to be a good approach. We decided to use a model predictive control (MPC) as one of the possible model-based control candidates.

6. What is model predictive control

The model predictive control is based on the feedback control and uses model to make a prediction of future state of the system or process. It is not possible to make an accurate prediction of future state without an accurate model, hence is necessary to spend time to create and calibrate a model what corresponds to the real plant. Model creation process will be described in my work.

MPC must be able to achieve desired EGR rate and keep the same engine output, adjust variable valve timing, spark advance and actuators positions. It solves optimization task and select the best control action considering constrains (for example maximum compressor speed or valve opening). Air flow with EGR is multiple-input multiple-output (MIMO) system and it means, it has an interaction between its inputs and outputs. This interaction makes usage of traditional controllers, such a PID, more challenging. Instead the MPC can consider multiple inputs and outputs, plus its interaction.

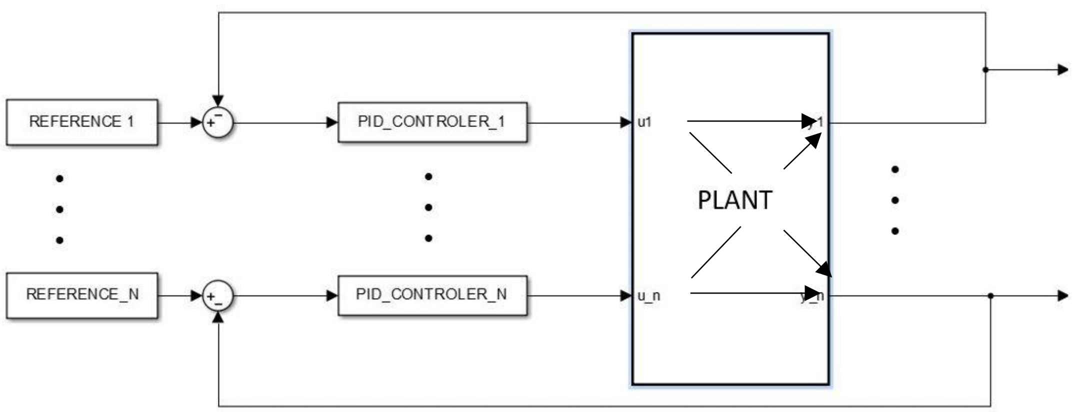

Figure 23: Scheme of Multiple-input multiple-output system

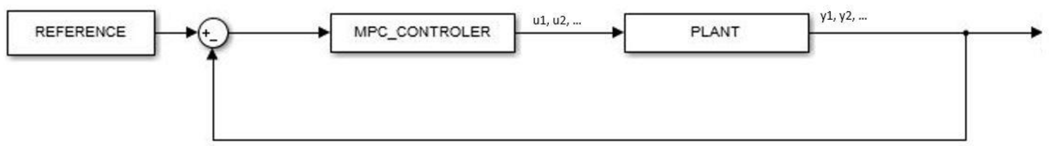

Figure 24: Scheme of system with MPC controller

The MPC needs few parameters to operate: controller sample time, prediction and control horizons, constraints and weights. These parameters change controller behaviour and computational load. The sample time needs to be chosen in balance between computational time and speed of disturbances correction. The prediction horizon is time in the future where MPC tries to predict. It should cover transient time of EGR. Control horizon is a number of time steps for which is controller adjusted. The actuator keeps its last position after this horizon. If the prediction horizon would be too low, we would not get realistic prediction. Increasing control horizon increase computational time and does not need to significantly improve accuracy. The constrains can be soft, which can be violated or hard, which cannot be violated. The outputs are usually considered as a soft constrain and input as a hard one. The weights are basically our preferences. MPC will prioritize based on them.

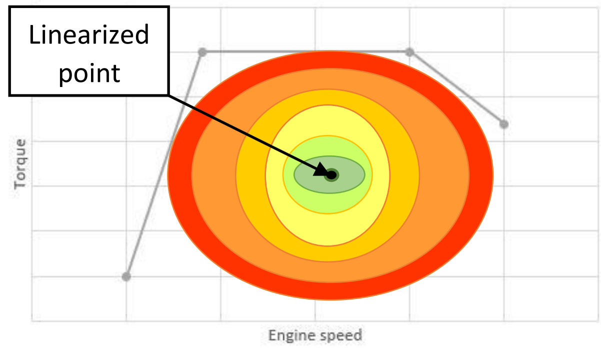

Figure 25: Example of accuracy lost with increasing distance from linearized operating point

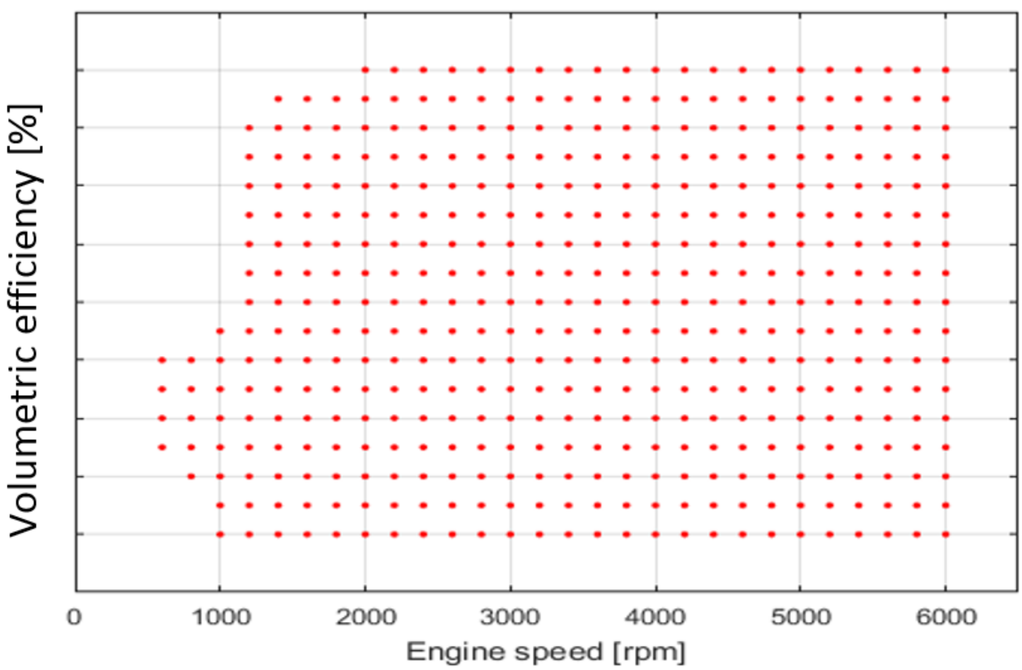

Figure 26: Grid of linearized MPC models

The traditional engine control strategies (e. g. map-based or PID controllers) are use in the cars nowadays because of the fact that MPC requires more powerful and faster ECU than the conventional controllers. As CPU and RAM of ECU are increasing, we can consider using of a MPC in the near future.

MPC computes an optimization online for each time step. Nowadays, it is solvable offline as a problem is inherently nonlinear. It is necessary to make the model lighter to be able to run the simulation online. Computational time can be reduced by a linear MPC as an optimization problem becomes linear. The linear MPC requires much less time than non-linearized without significant loss of accuracy and could be potentially used in today's cars. Regarding the fact that the engine is a strongly nonlinear system, we are not able to describe the engine operating range by one MPC (Linear time-invariant MPC). Thus, we must apply a grid on the operating range and compute a linear model at every point to cover entire engine range. We have to switch between individual linear models during transient operation. [38]

Two examples of different types of possible linear MPC for the airpath application can be: gain-schedule MPC and adaptive MPC.

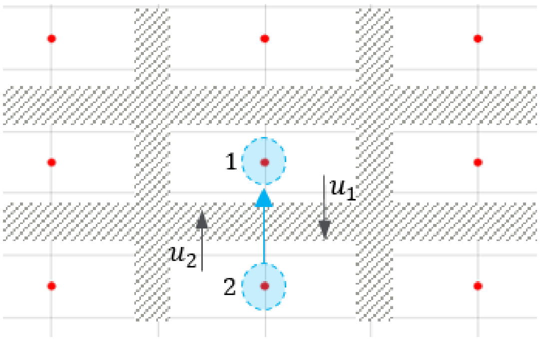

Figure 27: Gain-schedule MPC

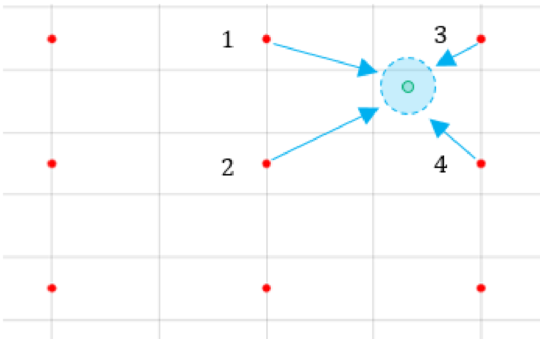

Figure 28: Adaptive MPC

Gain-schedule MPC switches between the unique models and keeps their properties for the entire area around the linearized point.

Adaptive MPC adjusts the MPC parameters (sample time, prediction and control horizons, constraints, and weights) constant based on the distance from surrounding models. It uses weighted average.

GT-Power is able to provide us those linearized models for a MPC control. It is possible to derive them only from a mean value engine model (MVEM). It is the reason why MVEM is needed and needs to be created.

7. Experimental setup and data post-processing

The entire operating engine range has been measured to understand and validate the benefits of EGR for SI engine. Engine measurement has been done externally at an engine test bench.

Engine type	Straight four-cylinder engine
Gear box	Manual test bench gear box (set into 4th gear; gear ration 1:1)
Stroke/bore	86 mm/86 mm = 1.0
Compression ration	10
Power output	175 kW (n = 4 800 to 5 600 RPM)
Maximum torque	350 Nm (n = 1650 to 4 000 RPM)
Injection	Dual, PFI + DI, DI fan type injectors

Table 1: Engine specification

The original engine was 2.0 litre, spark ignition turbocharged mass-production engine with port injection and direct injection system equipped by variable valve timing.

This engine was retrofitted by LP-EGR implemented downstream a three-way catalyst and upstream a compressor, equipped with a mass-production EGR cooler and EGR valve. It is one of a typical clean LP-EGR configuration.

Ambient temperature	25 °C
Ambient pressure	990 mbar
Charge air cooler outlet	XX °C
Oil temperature (at pressure switch)	XX °C
Coolant temperature	90 °C

Table 2: Boundary conditions

The target EGR rate was controlled by EGR valve position. The EGR cooler was supplied with a coolant at constant temperature of XX °C. The EGR cooler was efficient enough to maintain the EGR gas temperature at XX °C +/- 5 degrees across entire operation range. Engine load (BMEP) was kept constant during EGR sweep by adjusting the throttle position. Valve timing was frozen at its baseline values (operation without EGR) during each EGR sweep. Spark timing was manually adjusted to keep the optimal combustion phasing (MFB50 = 8 CAD aTDC) and to keep the same knock intensity to the baseline operation point without EGR. Fuel enrichment was manually adjusted to maintain the turbine inlet temperature at its limit for full load during the EGR sweeps.

Used fuel was RON95 E10 with lower heating value 41.47 MJ/kg.

	Unit	Cylinder: Cooled piezoelectric transducer	Intake: Uncooled piezoelectric transducer	Exhaust: Cooled piezoelectric transducer
Pressure range	bar	0...200	0...5	0...10
Temperature range	°C	-20...350	0...140	0...120

Table 3: Pressure transducers

The sensors placement in airpath was as shown in Figure 29. The temperature and pressure range of pressure transducers can be found in Table 3. Data have been recorded from 1000 RPM to 5000 RPM in term of engine speed and from 2 bar to full load in term of BMEP. Step of engine speed, load and EGR rate had been chosen to cover the entire operating range of the engine.

The data acquisition performed one sample every 0.1 CAD. The instantaneous pressure in the intake manifold, exhaust manifold and cylinders of one operating point are averaged values of 200 cycles.

7.1. Limits of LP-EGR architecture

7.1.1. Test design

Test was planned to cover the engine operating range based on the engine speed and load to evaluate limits of the engine equipped by LP-EGR. Steady-state EGR sweeps have been performed in 18 operating points distributed across the engine operating map. EGR sweep from zero percent to maximum value was planned for every position in the grid. The maximum value had been estimated the same for all operating points.

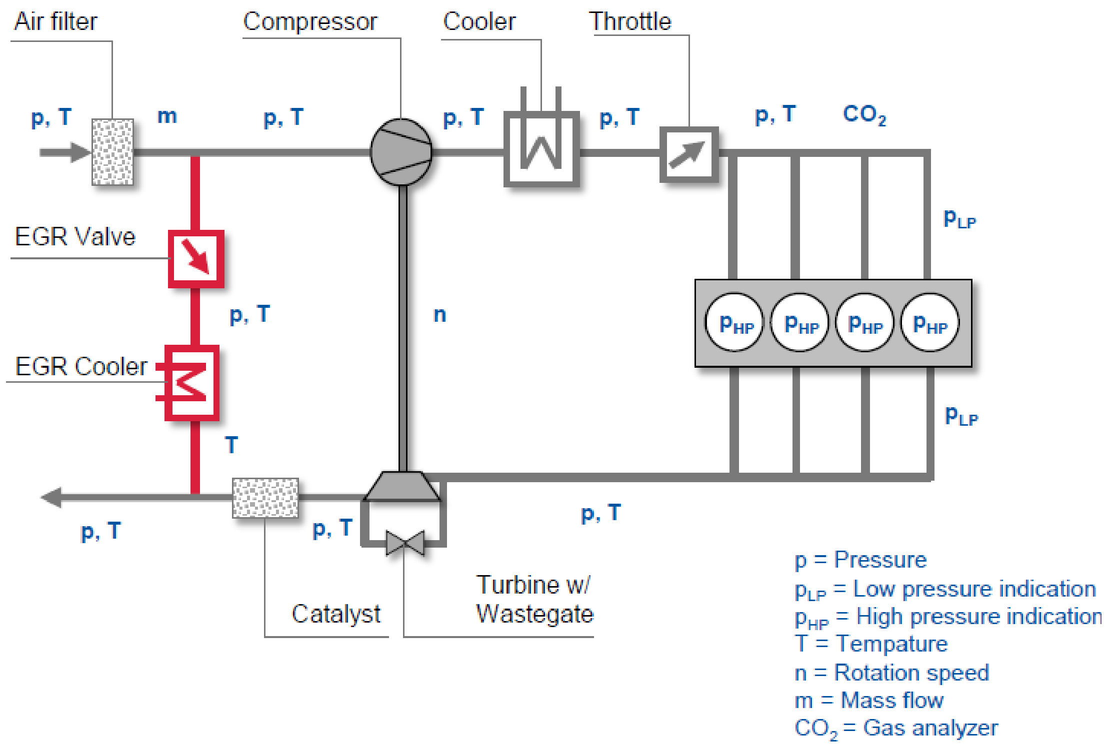

Figure 29: Overview of LP-EGR architecture [39]

Measurement would ended at the last stable EGR rate if it was not possible to reach the desired EGR rate. Because I am not allowed to mention the real values of EGR rate for confidential reasons, I will describe it on the fictive values. Let's say, the EGR sweep had been planned from zero to ten percent with step of two percent for all positions in the grid. If an operating point was not able to run with eight percent of EGR and more, measurement ended up at six percent of EGR.

It resulted in 78 operating points with different values of EGR rate and on top of that, it has been measured 41 reference operating points without EGR.

7.1.2. Confirmation of architecture limits

The delta pressure between the exhaust and intake part was not high enough to allows satisfactory and stable EGR rate during low load/speed operation. The fluctuation of EGR rate was too high and did not stabilize during the measurement. Therefore, average EGR rate for these points does not correspond to reality and these points will not be used to calibrate the engine airpath. It effected 13 operating points. This phoneme was expected based on research and corresponds area A in Figure 20Chyba! Nenalezen zdroj odkazů..

In the literature was mentioned another limit which occurs in high load, low speed area, where the compressor is not able to boost enough to keep the same load and increase EGR rate at the same time. During our measurement we faced to this limitation as well.

Another limit of the LP-EGR architecture is in high load/speed area where the turbocharger is on its limits and, as in previous case, it was not possible to obtain higher EGR rate for the same load. This is related with the zone C in Figure 20: Challenging areas

As a counter measure, the validity range of LP-EGR can be improved by different design of the airpath. As it was mentioned above, a flap in the exhaust path downstream the EGR extraction point could increase the backpressure and hence the delta pressure between the intake and exhaust path. [35] An improvement could be also achieved by an additional throttle in the intake duct upstream the place where the EGR line is connected. It can generate a vacuum by throttling. The result would be higher pressure drop. [28, 36, 40] These pipe modifications are an obstacle in the airpath which needs to be overcome by the engine and we can expect higher pumping losses. It would be necessary to investigate if a potential gain is higher than a cost.

Figure 29 shows placement of CO2 sensor in the intake manifold. Level of CO2 has been used to estimate the EGR rate as shown in Equation (1). A CO2 sensor is usually used as an emission gas analyser and has good accuracy for steady-state conditions. [36] It is not suitable in case of measurement of transient conditions, because this sensor has a longer response time and therefore is not able to measure the transition. It could have a negative effect on the emission formation and combustion control. It is better to use a wide range air fuel (WRAF) sensor to measure instantaneous EGR rate. [36]

7.2. Measured data analysis

The measured data have been postprocessed during and after the measurement campaign. This step is really important to get the results and understand what they represent. It is also desirable to detect measurement problems.

For example, during the postprocessing I noticed that the temperature of EGR gases downstream the EGR cooler was significantly lower than 90 °C, which was the coolant temperature. The most probably it was caused by condensation of water vapour. Condensed water was also present in the EGR line after measurement.

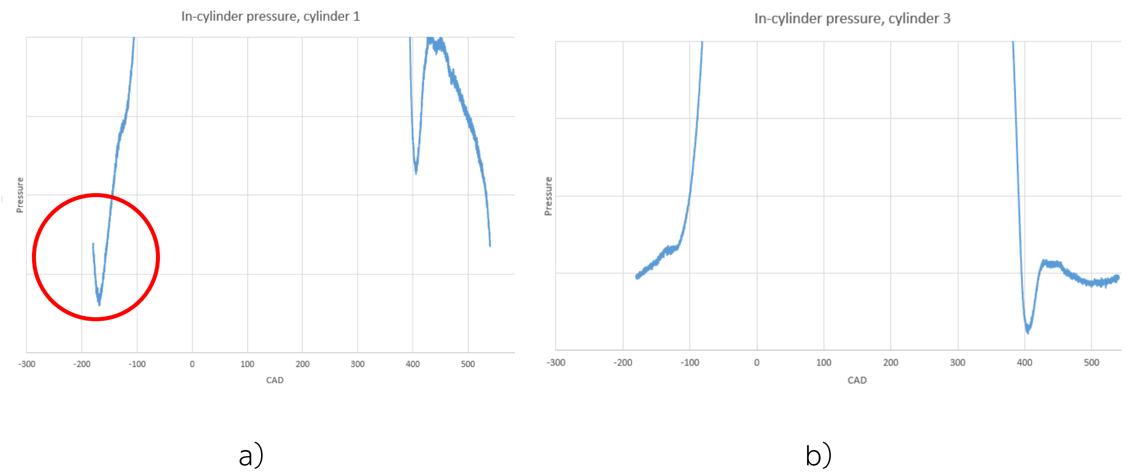

Figure 30: Behaving of in-cylinder pressure of cylinder 1 and cylinder 3

Secondly, the turbine surface temperature in one location was between 35-55 °C for all operating points. After I checked the sensor placement and wiring, I noticed that the wiring is broken and therefore the recorded data is not relevant.

Finally, potentially problematic value was intake manifold pressure, in-cylinder pressure and exhaust manifold pressure. Reliable values of the pressures are important because they are used for the combustion analysis. When I plotted instantaneous intake manifold pressure, in-cylinder pressure and exhaust manifold pressure I saw the pressures are shifted by 180 CAD. After discussion with technician, we found out that data acquisition system was not phased well. This issue has been fixed by re-phasing of measured data.

The in-cylinder pressure and intake manifold pressure have been resynchronized. The in-cylinder pressure needs to be recalculated to absolute value because the pressure transducer of in-cylinder pressure is sensing relative pressure, not absolute one. In-cylinder pressure is considered to be the same as the intake manifold pressure when the intake valve is closed. I plotted pressures (intake, in-cylinder and exhaust) and I noticed an unexpected pressure curve in case of in-cylinder pressure of first cylinder (Figure 30a)). This pressure behaviour has not been explained so far and is assigned to the sensor failure. In Figure 30b) is shown expected in-cylinder pressure trace of cylinder 3.

7.3. EGR rate estimation

The EGR rate was calculated based on the measurement data in according to Equation (1).

$$x_{EGR} = \left[1 + \frac{\dot{m}_{air}}{\dot{m}_{exh}} \frac{M_{air}}{M_{exh}} \frac{y_{CO_2,exh} - y_{CO_2,IM}\left(1 - y_{H_2O,intake}\right)}{y_{CO_2,IM}\left(1 - y_{H_2O,intake}\right) - 0.0003}\right]^{-1} \quad (1),$$

Where:

$\dot{m}_{exh}$	mass flow rate of humid exhaust gas
$\dot{m}_{air}$	mass flow rate of humid intake air
M_{exh}	molar mass of humid exhaust gas upstream catalyst
M_{air}	molar mass of humid intake air
$y_{CO_2,exh}$	molar fraction of CO2 in humid exhaust gas upstream catalyst
$y_{CO_2,IM}$	molar fraction of CO2 in the dry gas mixture in the intake manifold
$y_{H_2O,intake}$	molar fraction of H2O in humid intake air

Value of 0,0003 refers to CO2 fraction in ambient air.

Due to re-routing of blow-by gases, some CO2 is always present in the intake manifold even though the EGR valve is fully closed and therefore zero EGR should be measured.

8. Models

The models are used in the simulation process. Simulation is a popular method of car manufactures. Its functionalities ranging from a fast concept design to a detailed system or sub-system/component analyses, design optimization and root cause investigation. [41] It allows us to reduce the development time and hence cost by shortening development cycles, calibration on a test bench, reducing numbers of required prototype and is able to detect the problems without needs of prototyping and testing. Simulation allows us to simulate conditions, what would be difficult to achieve during a test bench measurement or it is not easy nor possible to measure. [42]

We decided to use GT-Power software developed by Gamma Technologies, LLC for our application. This commercial software in used by many car manufacturers and suppliers. GT-Power is 0D/1D/3D multi-physics CAE system simulation software which is capable to simulate the physics of fluid flow, thermal, mechanical, electrical, magnetic, chemistry and controls. It has a complete library with templates for given application. [41]

Even though, in Toyota already exist models of original, mass production engine, this engine has not been equipped by EGR line before. It was necessary to adapt the existing model in order to on the one hand consider the LP-EGR architecture and in the other hand to make a model compatible with GT-Power linear tool. This tool was used to obtain linearization of the mean value engine model around a given operating point. It created several linear models. These linear models are used as the prediction models in the MPC controller. I developed following models for this purpose.

EGR line model has been created to allow me to calibrate GT-Power model of a prototype LP-EGR architecture. The model has been calibrated by mean value steady state measurement data.

Mean value engine model of original engine already existed but it was necessary to modified it for EGR application. This model was important for MPC development because the linearized models are derived base on it.

Detailed engine model is important for validation of MCP before an application of MPC with real engine on an engine test bench. The detailed model is more accurate and representative model of our engine.

Two levels of accuracy are needed in my models. The MVEM and detailed engine model with trend accuracy is needed in the MPC development phase. Accurate MVEM and detailed model are needed for MPC validation before the application of MPC on a real plant. Precise models should have the same behaviour as a real plan to give a good background to MPC. MPC is then able to make a good prediction of the future state and select the optimal control strategy. It is very important for validation of MPC on a real plant and even more important for mass production application.

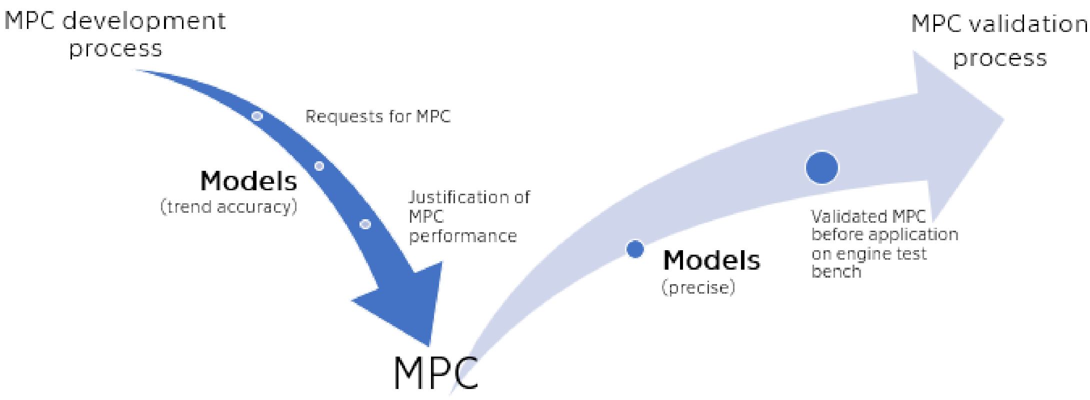

Figure 31: V-shape graph of MPC

I would like to create a naming of pressures and temperatures before I start to talk about a creating and calibrating of models.

By a boost pressure, I call pressure downstream the compressor. It is a pressure between the compressor and the throttle. The pressure downstream the throttle in called an intake manifold pressure. The pressure after the cylinder is called an exhaust (manifold) pressure. I use the same logic for the temperature naming. The sensor placement and naming are shown in Figure 32.

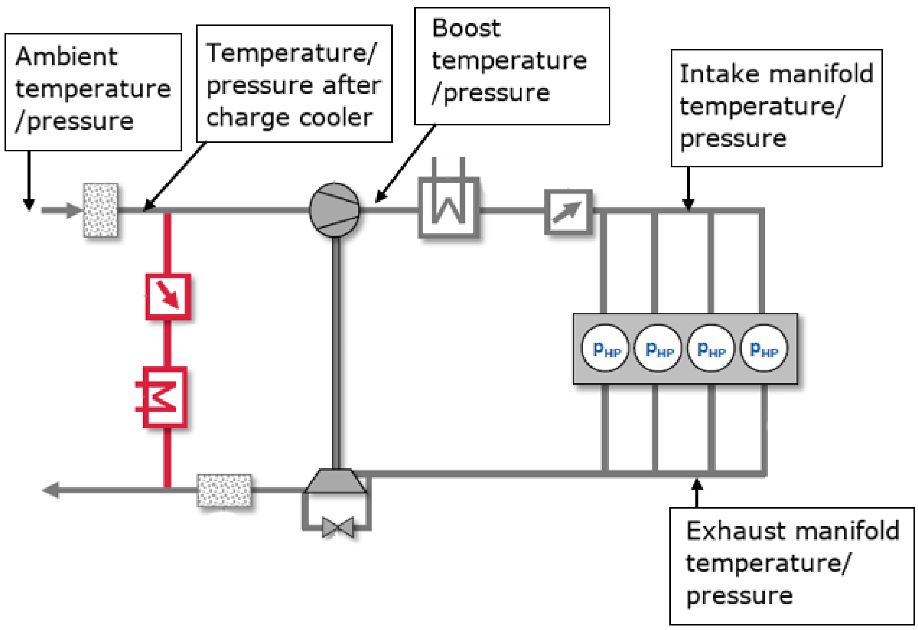

Figure 32: Sensor placement and naming [39]-modified

9. EGR line model

The EGR line model was used for calibration of the LP-EGR part to ensure the correct EGR rate before compressor, correct mass flow and related pressures and temperatures. It is part of a detailed model, but it is much easier to calibrate this separated section. Entire detailed engine model with EGR part would be more difficult to calibrate because it would be difficult to achieve match between measured and simulated properties in locations where pressures and temperatures were measured on the test bench. Then it would be difficult to judge if the deviation between measurement and simulation in the EGR line is because of the model calibration or if it is proportional to the deviation in boundary conditions. By separation of the EGR line I can impose the boundary conditions from the measurement.

The model consists of the already existing intake and exhaust part created by Toyota member. The intake and exhaust line were connected by the EGR line equipped by a valve and a cooler. Which one of the existing models had been selected as an intake and exhaust part will be described in chapter 11.

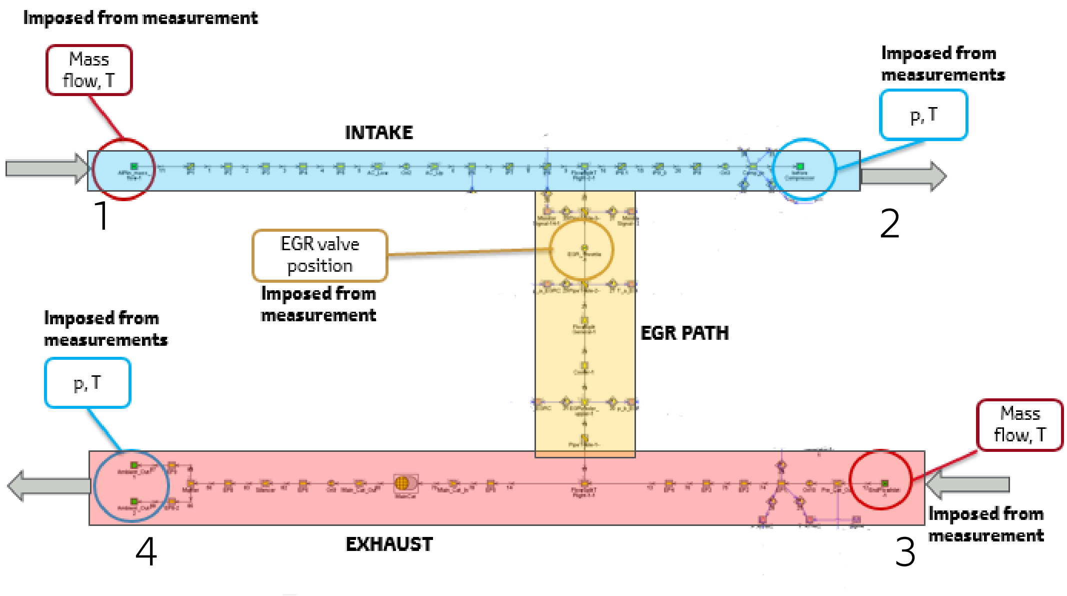

Figure 33: EGR line structure

The structure of the GT-Power model is shown in Figure 33. This model respects the pipeline geometry of the physical prototype. The model has been calibrated with mean value measurement data. In the upper part is an intake side with the air cleaner. In the lower path is an exhaust side compounding the underfloor catalyst and muffler.

I imposed the intake mass flow and intake temperature of a fresh air in the air duct inlet as the boundary conditions in point 1. In point 2, boundary conditions are the temperature and pressure of the intake mixture which were measured before compressor. The mass flow and temperature downstream of the first three-way catalyst are imposed in point 3. The pressure and temperature at the test bench are imposed as an outlet conditions downstream the tail pipes which represents point 4.

The GT-Power manual recommends using the operating points from full load operation to calibrate the flow. In our case these points would be the points with maximum EGR at the high load. After several cycles I decided to use all the operation points from the measurement excluding point from the low load/speed area, where the EGR rate was not stable, a pulsation was present and for some points insufficient delta pressure has existed between the exhaust and intake part. This decision was already mentioned in chapter 7.1

.

As the EGR cooler proofed good performance, I decided to use *'Imposed intercooler outlet temperature'* method in my model. It is a non-predictive approach and it consists in imposing the EGR cooler outlet temperature. It is done by setting the desired outlet temperature as a pipe wall temperature and by increasing the heat transfer multiplier between the fluid and pipe wall. This approach is very effective, but drawback of this method is that the model would not able to interpolate outside the points it was defined on.

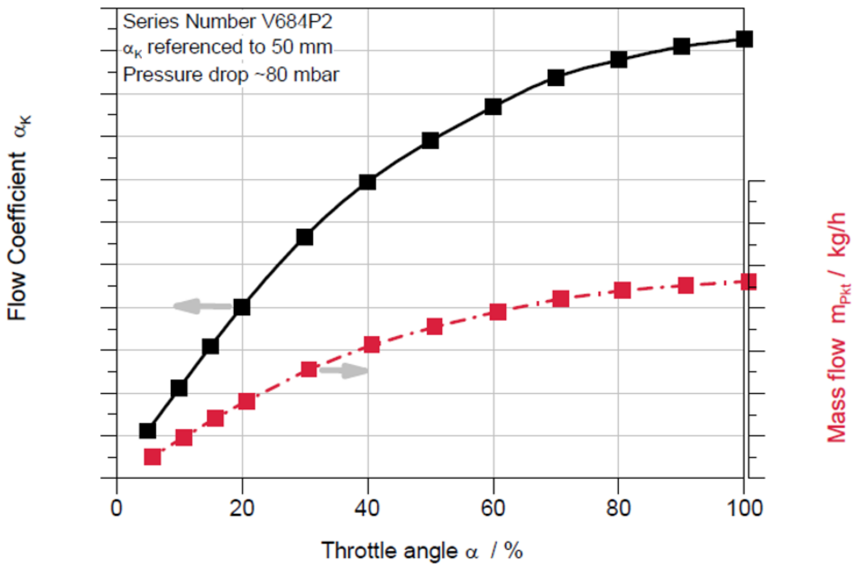

Figure 34: Discharge coefficient of EGR valve [39]

EGR valve has been modelled by *'Throttle'* template. In the throttle is necessary to define a discharge coefficient. The explanation is provided in 9.1. The flow coefficient has been measured as a function of throttle angle with constant differential pressure of approximately 80 mbar and results are plotted in Figure 34.

From the steady-state EGR sweeps operating points is obvious that for the correct EGR rate in the detailed engine model will be necessary to control the EGR valve. Otherwise, I can obtain two different EGR rates for one valve position, as shown in Figure 35. For example, if the EGR valve is around 40 % opening, we can achieve two different values with difference of 25 % of EGR rate range. We can also see that the red points from low load/speed area of engine operation map are farer from expected values. As it has been already told, it is because of instability of EGR rate during the measurement caused by a low delta pressure.

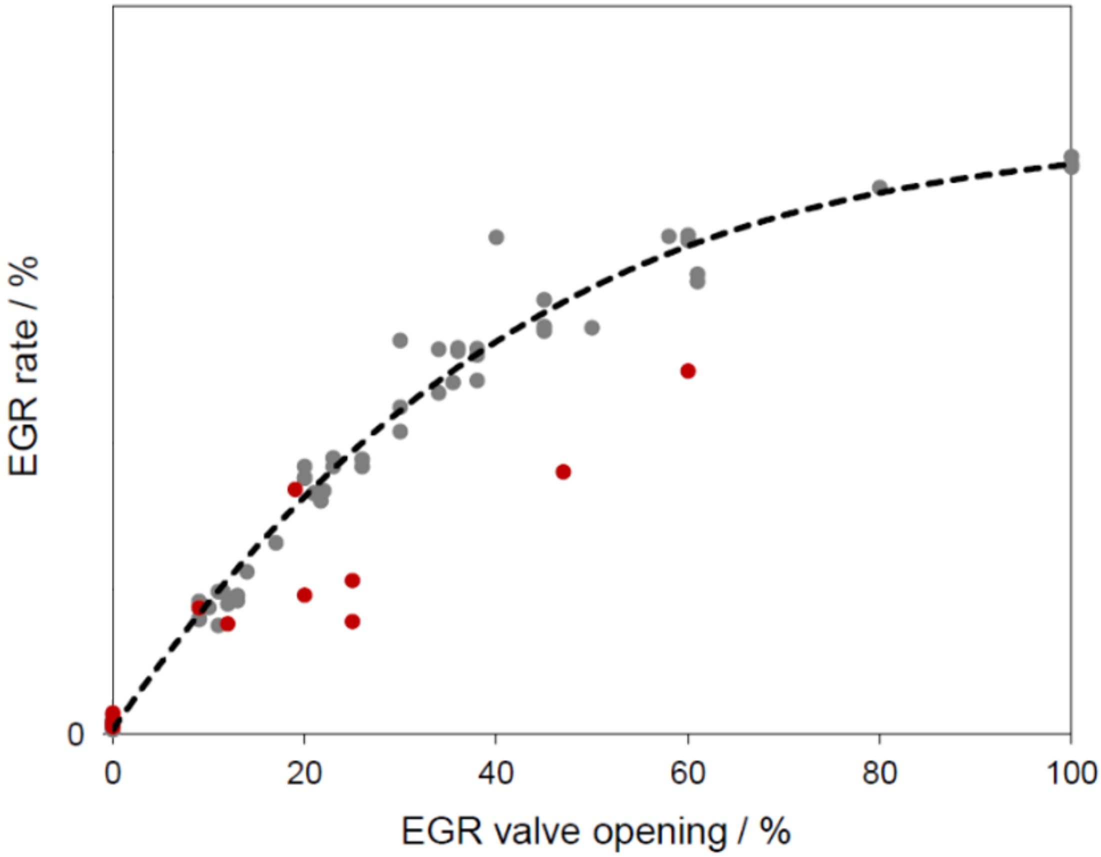

Figure 35: EGR rate vs. valve position [39]

9.1. Correlating an engine model to measured data

It is necessary to tune a model after it has been build. The first step is to check typographical and unit errors in model setup and input values. It usually generates a big deviation or even failure of the simulation. After we make sure that we are using correct input data, we can start to get our model closer to the measured engine. [42]

A calibration is founded on tuning of the parameters. The most relevant parameters in our application are heat transfer coefficient multiplier, friction multiplier and discharge coefficient.

Heat transfer coefficient multiplier scales the calculated heat transfer rate between the fluid and the wall. It results in different heat losses, therefore different temperatures. [42]

Friction multiplier scales the calculated pressure loss due to friction between the fluid and interior wall surface. Higher friction multiplier creates higher pressure losses, lower pressure downstream the part and higher pressure upstream the part. [42]

Discharge coefficient is used for the valves, throttles and orifices. It describes the dependency of mass flow through object as a function of the valve opening or size of the pipe diameter. There are two types of discharge coefficient – forward and backward and they refer with the direction of the linking arrows or the name of a dependency reference object. It could be calculated automatically in case of an orifice. [42]

The calibration has been performed during the open-loop control. It means I was imposing boundary conditions and actuator position from experimental measurement and I observed results of simulation. No value was controlled by controllers.

9.2. Targeted model accuracy

I need to select a deviation what I can consider as the accurate results to be able to evaluate if the calibration has been sufficient and satisfactory. Some KPIs are considered to judge model accuracy. In the GT-Power manual [42] is recommended to calibrate the intake manifold pressure within 2 %. If we consider, that intake part before compressor is near the atmospheric pressure, 2 % error would correspond to 0,02 bar. Because the GT-Power manual does not mention the recommended values for the other properties, such as temperature and mass flow, EGR rate; I had to choose them by myself.

I chose following accuracy:

The deviation can be up to 2 K of temperature, 0,02 bar of pressure, 2 g/s of mass flow and 1 % in absolute value of EGR rate. These values have been applied for the entire LP-EGR line model.

9.3. Results before calibration

The first run was an open loop simulation with the default values of coefficients. It was an initial run without any kind of model calibration and the results are only function of design. The boundary conditions have been imposed as mentioned above. The value of heat transfer coefficient multiplier for the EGR cooler has been chosen default. In Figure 36 we can see, that the temperature before compressor was higher for the most cases and was out of the established limits. The situation had a similar trend in the case of EGR rate and therefore gas mass flow. By the gas

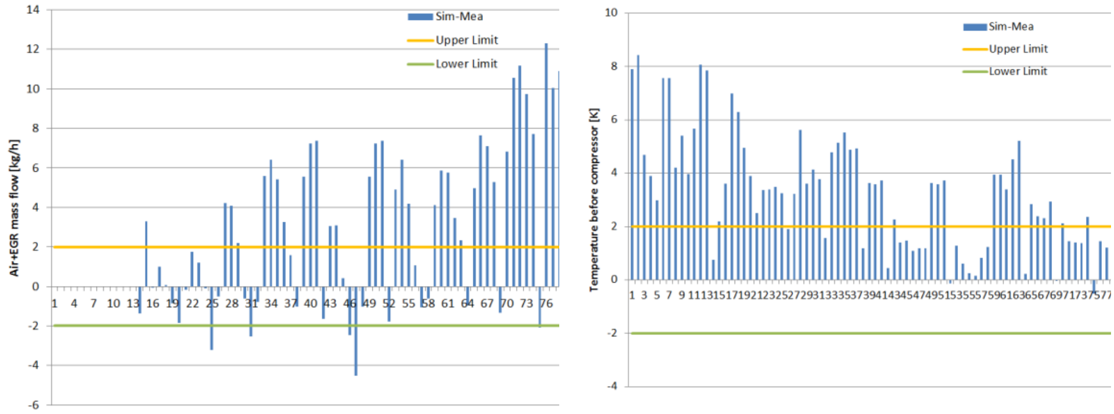

Figure 36: Gas mass flow (on the left) and temperature upstream the compressor (on the right) before calibration

is meant the mixture of fresh air and EGR gases. Either the comparison of simulated temperature and pressure after the EGR cooler was not sufficient. On the other hand, acceptable trend has been obtained. A calibration of the model is needed for better accuracy.

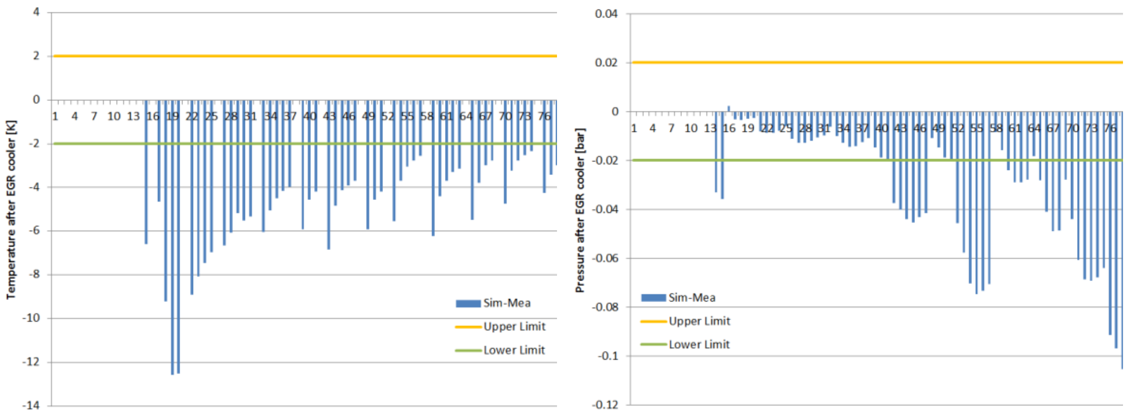

Figure 37: Temperature (on the left) and pressure (on the right) downstream the EGR cooler before the calibration

9.4. Method of calibration

I tried to achieve a match between simulation and measurement by tuning the parameters mentioned in 9.1. As a first step the intake part was tuned during closed EGR valve. Reason, why I used the operating points without the EGR rate at the beginning, was to eliminate an error cause by inaccuracy of the LP-EGR line. It was necessary to slightly adjust the heat transfer coefficient multiplier to lower the temperature.

As a second step I did the same with the exhaust part, still with the cases with closed EGR valve. The pressure at the extraction point of EGR was too low, therefore it was necessary to increase the friction multiplier of underfloor catalyst to increase the upstream pressure. The temperature before the EGR cooler was not so important for me because the EGR cooler will erase the error.

As a last step I added the points with different EGR rate and I connected the intake part and exhaust part by the LP-EGR part. I had to adjust the value of heat transfer coefficient multiplier of the EGR cooler to achieve better correlation between the measured and simulated values. I also had to modify the friction multiplier for EGR cooler.

In the end, I had a satisfying accuracy of the pressures and temperatures but the EGR rate/gas mass flow was not correct and the simulated one was always higher than measured one. It was necessary to use a 'correction multiplier' for the discharge coefficient of the EGR valve. The reason why the simulated EGR mass flow was always higher than measured one could be the fact that the mass flow is a function of pressure power two. I removed pressure pulsation by using the mean values for calibration – the valleys and peaks; and result is higher mass flow through the valve.

9.5. Results after calibration

Here we can compare the improvement of the simulation accuracy. The mass flow of the gas has a weaker performance. It is related with the EGR rate and cannot be much improved without the EGR valve control. It is related with Figure 35. The other properties have been improved to fulfil my target accuracy for most cases. Improvement is evident from Figure 38 and Figure 39, where I present selected results of the calibration.

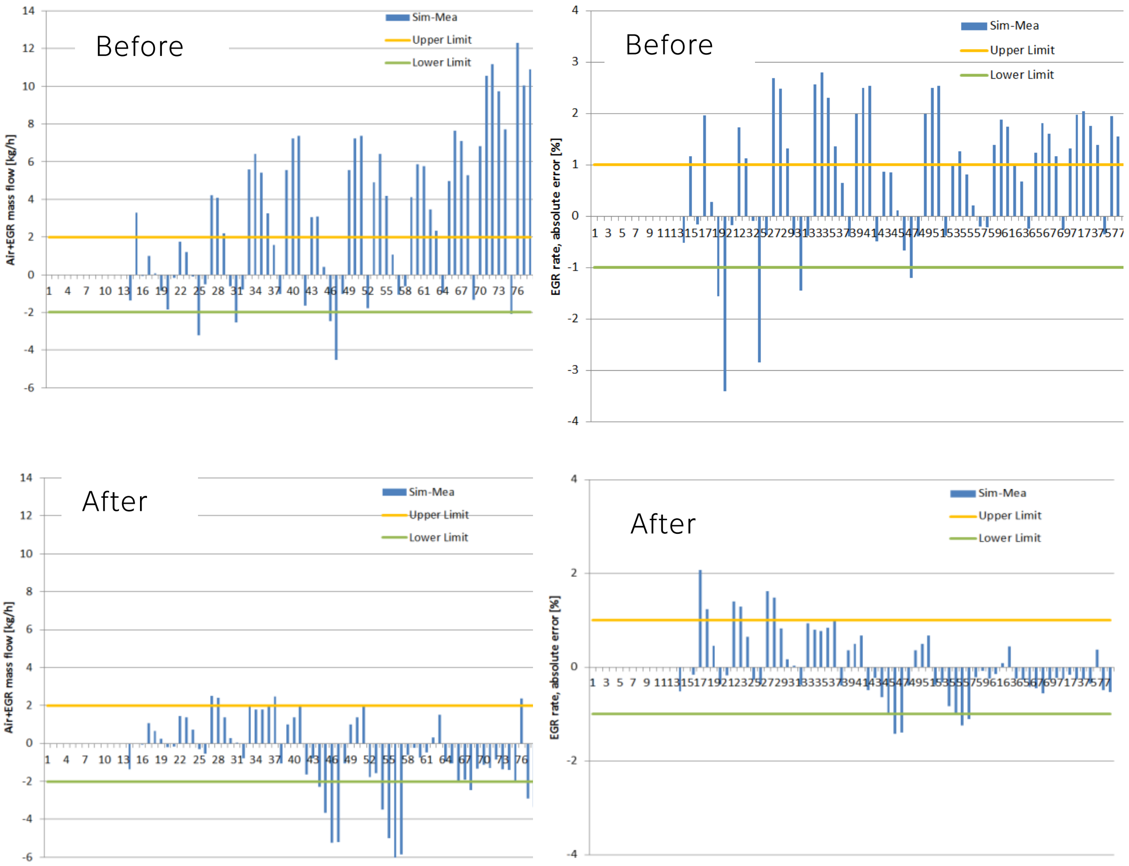

Figure 38: EGR line performance before and after calibration
Gas mass flow on the right, EGR rate on the left

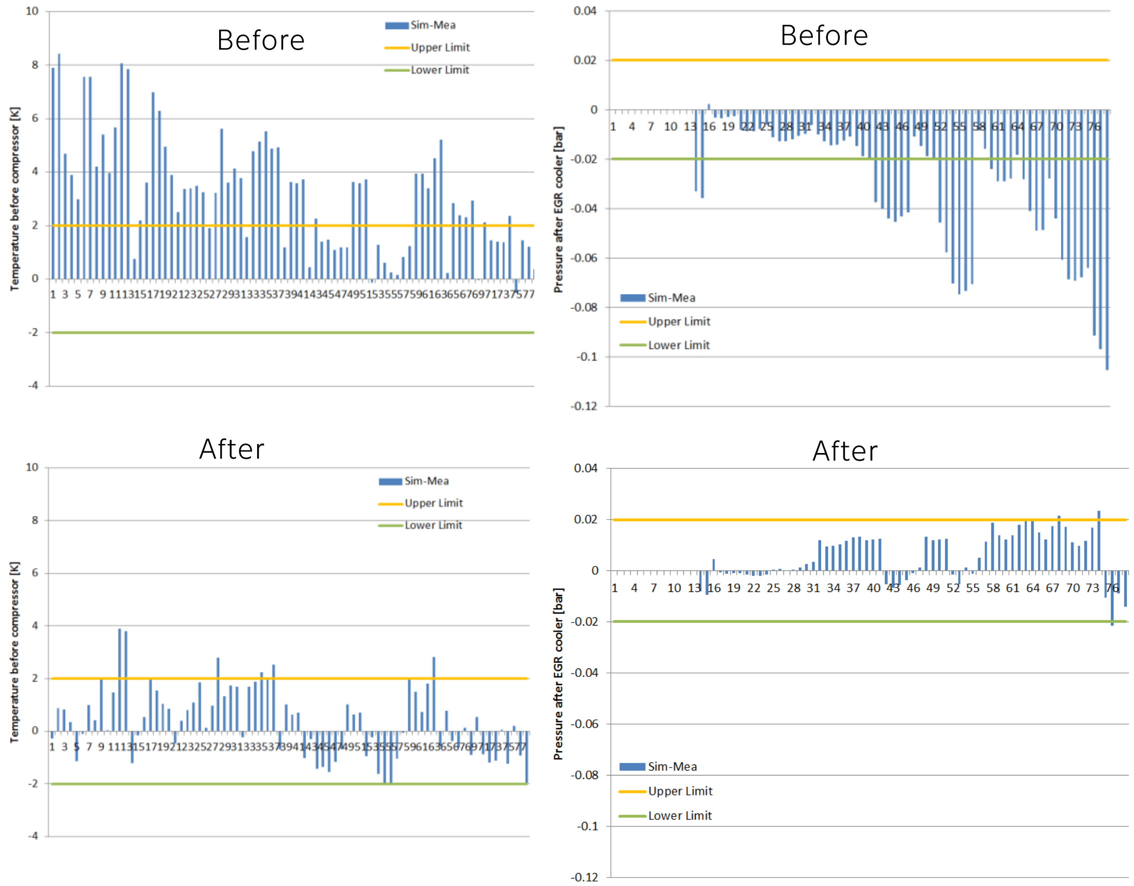

Figure 39: EGR line performance before and after calibration
Temperature upstream the compressor on the left, Pressure downstream the EGR cooler on the right

You can find more numbers related with overall improvement of the EGR line calibration in Table 4. If 78 points are considered, I consider all operating points from EGR sweep measurement (even points without stable EGR rate), 65 points are the points with stable EGR rate and 52 points refers to points with EGR rate higher than zero and stable at the same time. In case of a zero EGR rate, I am not interested in temperature after EGR cooler because this temperature does not have an impact on the mixture temperature before the compressor.

Property	Design		Design + calibration	
Air + EGR massflow	24/65	37%	48/65	74%
EGR rate	26/65	40%	53/65	82%
Temperature before compressor	26/78	33%	71/78	91%
Temperature after EGR cooler	0/52	0%	46/52	88%
Pressure after EGR cooler	31/65	48%	62/65	95%

Table 4: Number of points fulfilling calibration KPIs

10. Mean value engine model

As it was already mentioned in chapter 6, the MVEM will be used to obtain the linearized models needed for MPC control. The linearized models will be derived directly from the MVEM by GT-Power. As described in chapter 8, two levels of accuracy are needed. As the first loop of development, high accuracy is not needed for the MPC design methodology development. However, a precise model is mandatory for the fine calibration of MPC before the engine test bench application.

The mean value model is a simplified engine model using a map-based cylinder model instead of a combustion model. The map-based cylinder uses the maps to characterize the cylinder air flow and the distribution of fuel energy, whence map-based cylinder. The MVEM uses an *EngCylMeanV* object instead of the conventional cylinder object to characterize the engine cylinder performance and for this purpose three quantities must be imposed. One of them is volumetric efficiency of the cylinder which says how much mass flow rate of gas will be imposed to come through the cylinder. The second one is the cylinder feeding efficiency. It describes how much of the fuel energy will be converted into the piston work. Last map is exhaust energy fraction which describes how much of the fuel energy will be used to heat the exhaust gases. [42]

Each of these quantities can be defined as the maps (look-up tables) or they can be calculated externally. A map-based method is usually used for the limited amount of input variables, while an external calculation is mostly used for more complex cases, where map depends on many quantity in the model, for example engine speed, load, intake manifold pressure and temperature, etc. [42]

Simplified engine cylinder model results in faster simulation since the breathing and combustion process is not simulated. An additional contribution to faster simulation has increasing size of the timestep. The maximum timestep for a detailed model is typically limited to maximum of one CAD. The maximum timestep can be increase up to fifteen CAD for a MVEM. The flow within the model is much steadier than in case of a detailed one due to the model nature. There is no pressure pulsation generated by the cylinders, therefore the remainder of the airpath may be simplified as well to further reduce simulation time. Generally, the intake and exhaust system can be lumped into the larger volumes and it is also possible to increase the discretization length. [42] In the GT-Power is a function called *Combine Volume Wizard* which combines variable pipe template to single unit and keeps the identical volume.

10.1. Maps creation

10.1.1. Conventional approach

The usual workflow is to create a detailed model and then derive a MVEM as a simplification of the detailed one. The detailed model is run in DOE functionality to create a variety combination of operating conditions. The data such as engine speed, volumetric, indicated, exhaust and cylinder feeding efficiency are stored. These data are then used to define the correlation between the controlling quantities (engine speed, volumetric efficiency…) and the input variables (efficiencies).

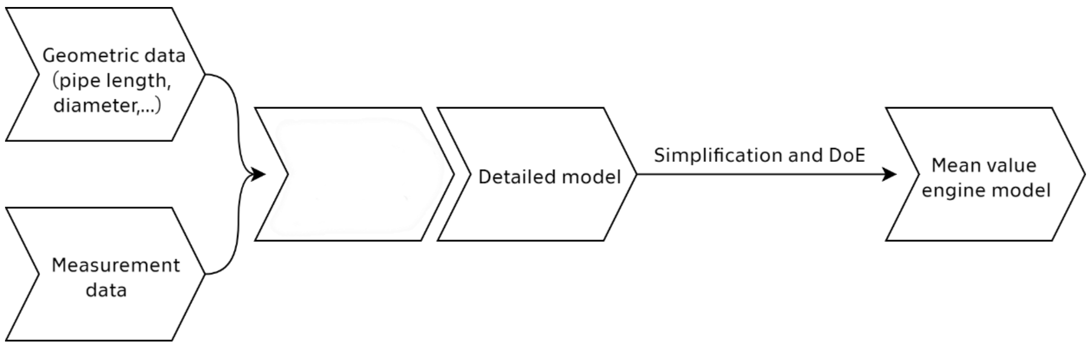

Figure 40: MVEM creation – conventional approach

In TME had been available a MVEM of original mass production engine without the EGR path. This model used maps as a function of engine speed and volumetric efficiency. Volumetric efficiency considers only fresh air, therefore there is no reflection of the EGR rate. It would influence the performance of maps and output values of the maps would be different from the real conditions. It means that engine speed and volumetric efficiency are not enough to integrate EGR effect, hence I had to add additional variable as an input for the maps. I considered two possible variables – the intake manifold pressure and EGR rate. I decided to use the EGR rate as a third input because the combination of engine speed, volumetric efficiency and intake manifold pressure does not have to explicitly corelate with the EGR rate. The intake manifold pressure could be influenced by additional variables such a temperature and it could make up the EGR rate and then the engine performance in the end.

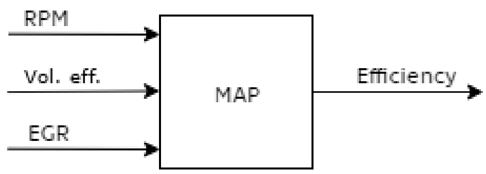

Figure 41: Input of 3D map

GT-Power manual mentions that for a transient simulation can be necessary to include the air to fuel ratio as an additional input for the indicated efficiency map. Reason is that its effect can be included implicitly through other selected variables. [42]

MPC controller will have to be validated during transient operation in last stage of development. For now, the steady state operation is much more dominant because MPC development is in the first development phase, hence I decided not to involve the AFR as one of the inputs.

10.1.2. Map calculation methodology

My next contribution was to update and calibrate MVEM to support the MPC controller development. To modify and calibrate the detailed engine model could be too lengthy and potentially could restrain the MPC development. We decided to use an alternative process to obtain the maps for the MVEM. The model of the original MVEM was available and the detailed model of EGR line had been already created and calibrated by mean values of measured data.

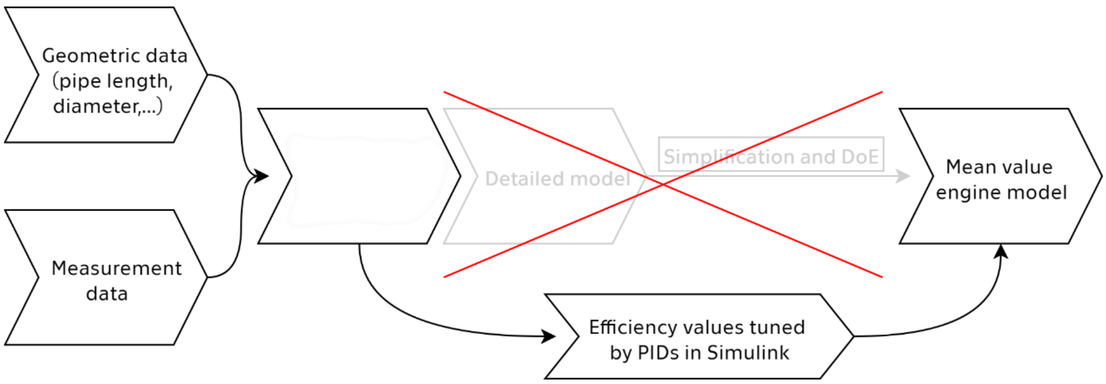

Figure 42: MVEM creation – used method

TME developed a process to create the maps of *Cylinder feeding efficiency*, *Indicated efficiency* and *Exhaust energy fraction* directly from the measurement data without the need to run a DOE on a detailed model. This process uses simplified GT-Power mean value engine model. The tool calculates the map efficiency based on the simplified MVEM. A PID controller is adjusting the efficiencies of the model to get the same engine output as a measurement for all operating points.

The model of the tool contains only the pipes and the cylinders between the intake manifold temperature and pressure sensors and the exhaust manifold temperature and pressure sensor. The MVEM does not used normal cylinder template, but it is replaced by an *EngCylMeanV*. I imposed the boundary conditions such as pressures and temperatures, engine speed and lambda. By a PID controller I controlled the EGR valve to achieve the correct EGR rate in the intake manifold. This model structure is shown in Figure 43.

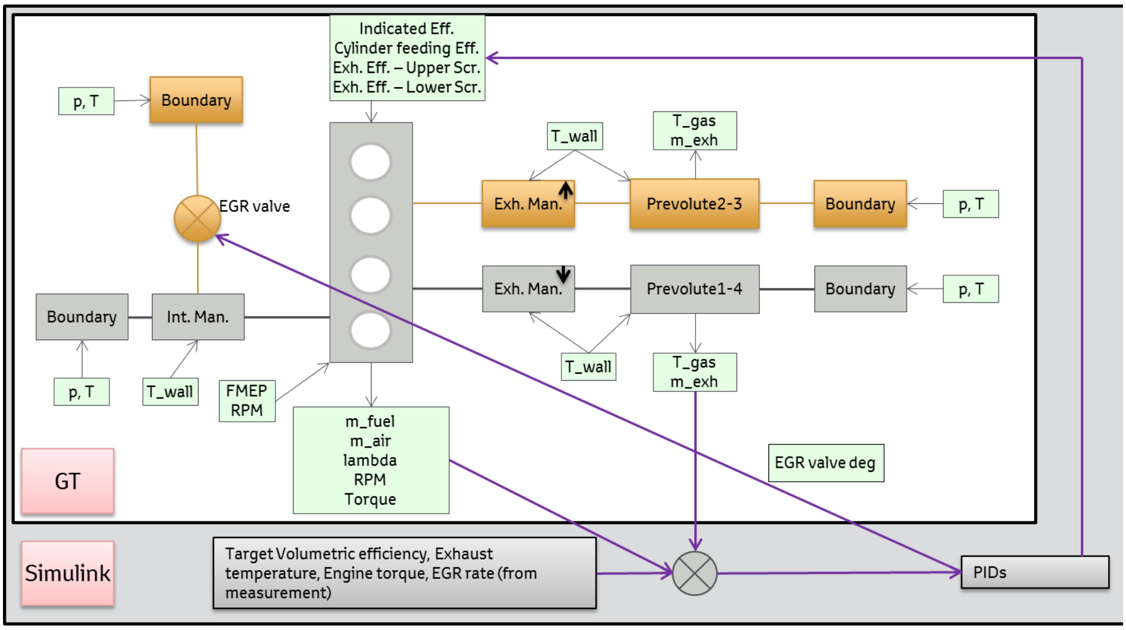

Figure 43: Tool for efficiency maps creation – PID phase

The tool was not prepared for my application and it was necessary to adapt it. The existing tool supported only an engine with single scroll turbine, therefore I had to modify it for an engine equipped by twin-scroll turbine. It was also necessary to involve an EGR part. I also implemented a heat transfer model on the cylinder, exhaust manifold and part of prevolute in a later stage of the MVEM development. In Figure 50 are also presented the results without the heat transfer model.

Procedure to obtain the maps efficiency values is as follows. I sent the torque, engine speed, mass flow of air and fuel, lambda and exhaust gas temperature from the GT-Power to the Simulink. In the Simulink I calculated simulated volumetric efficiency and compared its value with the measured value. I did this also with the value of engine torque, exhaust temperature and EGR rate. Based on the comparison, I tuned the values of our outputs from the three maps by PIDs. These values had been sent back to the GT-Power and imposed to the cylinders. This process had been repeated until I reached the parity with measured data from the engine test bench. When the parity had been reached, I stored the final value of the three efficiencies. The engine speed, volumetric efficiency and EGR rate was stored as well and they supposed to be the same with measurement data. Only the measurement data without EGR rate pulsation (with stable EGR rate) has been used. It created 69 operating points.

Around one hundred points are needed for average 2D map. It means, that we would be under ideal amount of points even in case of 2D map. Our MVEM operates with three inputs and therefore it would need much more points than we have available. Approximately 500 would be needed to have a similar concentration of operating points as a 2D map.

I had to gridded the engine operating range and interpolate the efficiency values between the points to be able create a map. As already indicated above, each map has three inputs and one output. Inputs are the engine speed, volumetric efficiency and EGR rate. The output is a corresponding efficiency. Hence it is clear it was also necessary to create process for 3D map creation. In TME is available tool to create a similar map, but it supports only the maps up to two dimensions. I tried to find the best way how to create 3D map with gridded data. The most adequate way how to create a three-dimensional lookup table with scattered data I have found on the MathWorks support website. On www.mathworks.com had been proposed a function called *'regularizeNd'* what is capable to create a gridded lookup table of scattered data in n-dimensions. [43]

By this function I created two exhaust energy fraction maps, one for the lower scroll and second one for upper scroll; indicated efficiency map and cylinder feeding efficiency map.

I checked the function output in the next step. Because of the lack operating points in the 3D map I was suspicious if the new function would be able to create a reasonable maps surface what would be able to represent the real plant behaviour. I had to validate if the selected approach is correct and if it will be able to provide the sufficient results. I decided to use the following methodology. I grouped the measurement data base on the EGR rate. As EGR rate was not the same for all operating points, I segregated the measured data to four groups based on the level of EGR rate. The EGR rate varies in a single group but it was similar enough to allow me to neglect its effect. These groups can be called iso EGR. I created 2D maps of iso EGR by the existing TME tool – EGR has been fixed, therefore I was able to create a 2D map. This map has an engine speed and volumetric efficiency as an input. Then I created 3D maps from the 69 points by *'regularizeNd'* function. As a next step, I fixed EGR rate in the 3D map and extracted the 'layers' of constant EGR rate which is equal to the average EGR from iso EGR group. It allowed me to plot the efficiency maps as 2D maps and compared them with the corresponding 2D maps created by the TME tool. I plotted these corresponding 2D maps in the same figure together with the operating points and compared them visually. One of the created comparisons plots you can find in Figure 45.

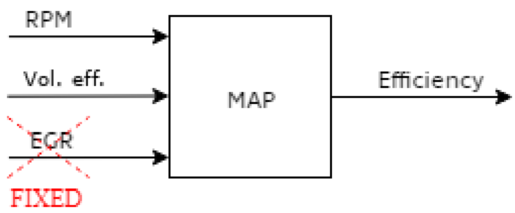

Figure 44: Extraction of a layer from 3D map

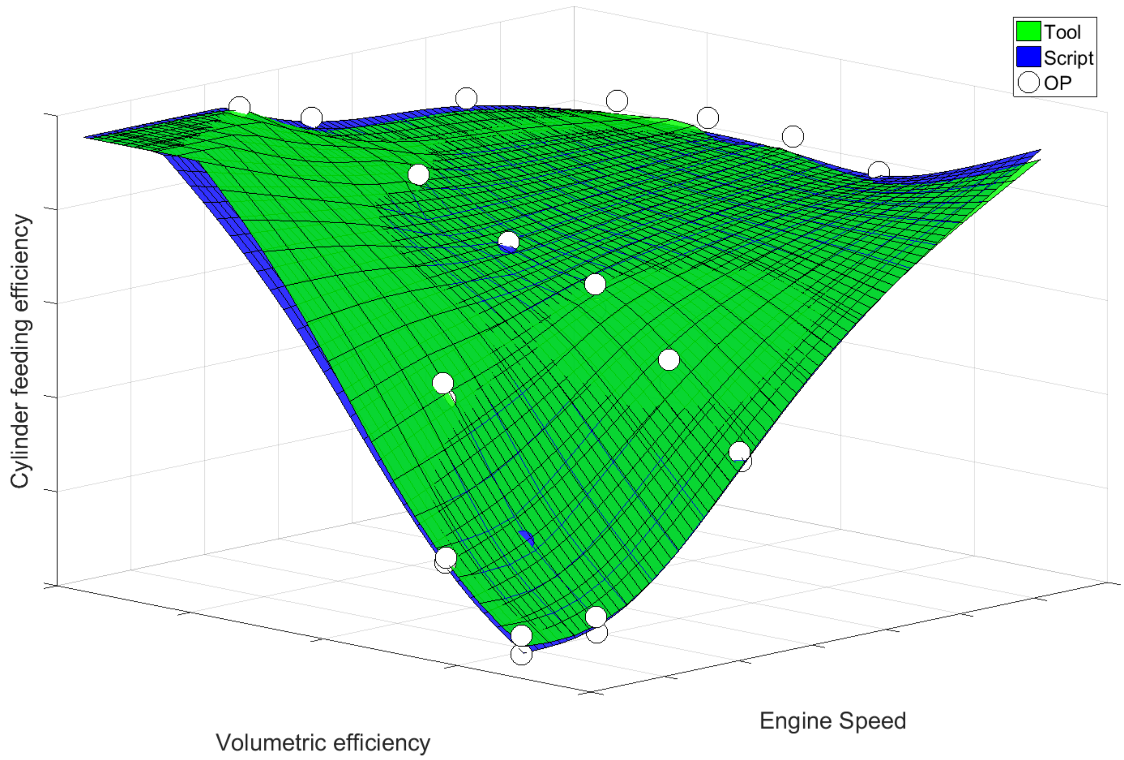

Figure 45: Tool validation - iso EGR

From the comparison is obvious, that 2D maps are not the same but it was not even expected. For confidentiality purpose, I cannot provide EGR rates or what EGR range we used in each group. Let's say, that in a group of the iso EGR were the points with the EGR rate between 3 and 7. I used those points for the green surface. The 'layer' from the 3D map created by the function represents the EGR rate equal to average of 3 and 7 and is represented by the blue surface. The white balls are the operating points in the iso EGR group, so all the points with EGR rate between 3 and 7. PIDs used these points to tune the values of efficiencies.

I cannot expect the same 2D map surface as EGR rate is not the same. Much more important is to see if the function is capable to interpolate between the operating points and to create a map with a realistic behaviour from such a limited amount of points. I was able to judge that the function is able to interpolate in the macro view and it is possible to use the created maps during MPC development.

One more comparison had been evaluated. I took the iso speed and iso volumetric efficiency. It means, I fixed the engine speed and volumetric efficiency instead of the fixed EGR. As a result, I could see dependency of our efficiencies as a function of EGR rate. I plotted all of them in one plot to see their trend. For example, in case of a cylinder feeding efficiency you can notice that with the increasing EGR rate is cylinder feeding efficiency increasing as well. This trend is stronger for the points with lower load where EGR rate reduces throttling and it is related with pumping losses reduction.

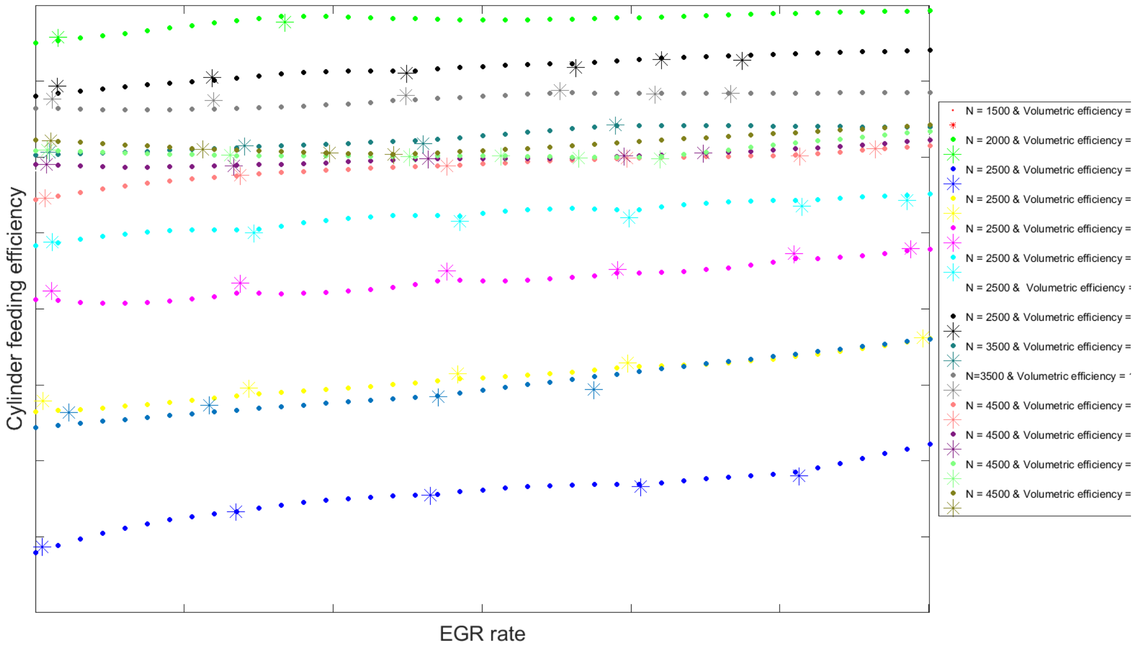

Figure 46: Volumetric map cross section

I also wanted to see the detailed comparison with the TME tool. One example of the map is shown in Figure 47. It is the same cross section as in Figure 46 shows the detailed behaviour of one cross section from Figure 46.

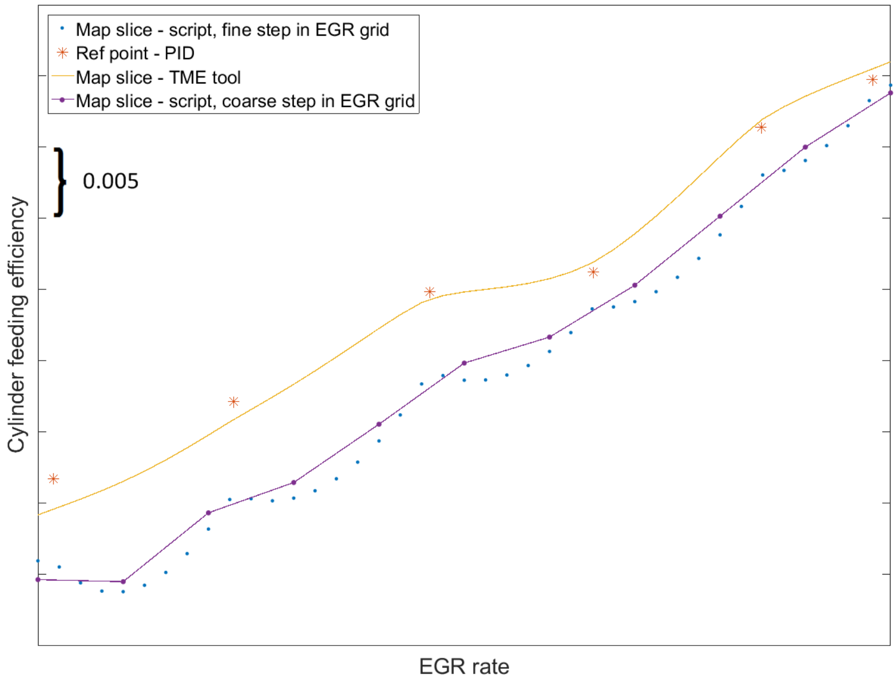

Figure 47: Volumetric efficiency - iso speed, iso volumetric efficiency

A gap between TME tool and the function is present, even though the gap is small. The stars represent the point where measurement was performed and PID calculated the efficiency map value from the measurement. Originally, I created the map with fine EGR grid. I checked the cross section of the map and I noticed unexpected waves in cylinder feeding efficiency profile which does not have a physical explanation. A potential source of this behaviour can be that the curve is attracted by the next layer of the 3D map. I expected with increasing number of measured points this phenomenon will be reduced. I increased the step in EGR grid to supress of the wavy behaviour. In Figure 47 is shown that this behaviour has been supressed without a loss of accuracy.

Based on these results I decided to use regularizeNd function for my application.

Accuracy is limited due to the limited number of the measurement points. I used 69 points to create the 3D maps because no more data had been available. This amount of points would not be satisfactory even for a 2D map. It is not a limit for MPC controller development application because trend accuracy of the model has been reached. However, more precise maps are needed in future for the MPC controller validation. It will require to have more operating points for the more accurate maps. A detailed model calibrated with the available cylinder pressure measurement can be used to generate additional operating points without additional test bench.

10.2. Maps performance validation

The maps performance validation consists of two phases.

Firstly: I want to validate if the PID results are correct. It is done within the tool where I replace PIDs by the newly created maps and I run the simulation in open loop.

Secondly: Maps are imported into the entire mean value engine model with complete air path and turbocharger. In that phase we validate maps performance within entire MVEM

First phase of the maps performance validation is related with the tool where we obtained the values of efficiencies from the PIDs. To validate the values of efficiencies in the maps, we removed the PID blocks in Simulink and replaced them by the look-up tables. They send the efficiency values to the GT-Power based on the engine speed, volumetric efficiency and EGR rate coming from the GT-Power.

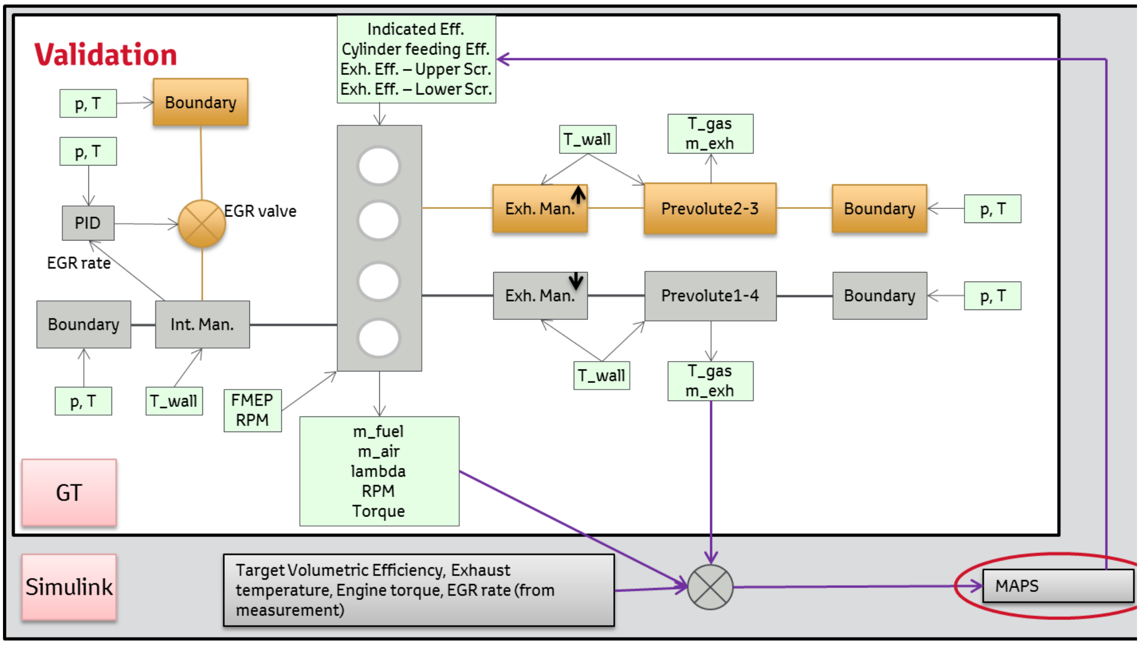

Figure 48: Tool for efficiency maps creation – maps validation phase

I compared simulated values, such as volumetric efficiency, torque, exhaust manifold temperate, with measurement data and almost no deviation is expected. Back-to-back comparison is in Figure 49, where as a reference value is used measurement data and they are compared with simulationFigure 49: Back-to-back comparison results.

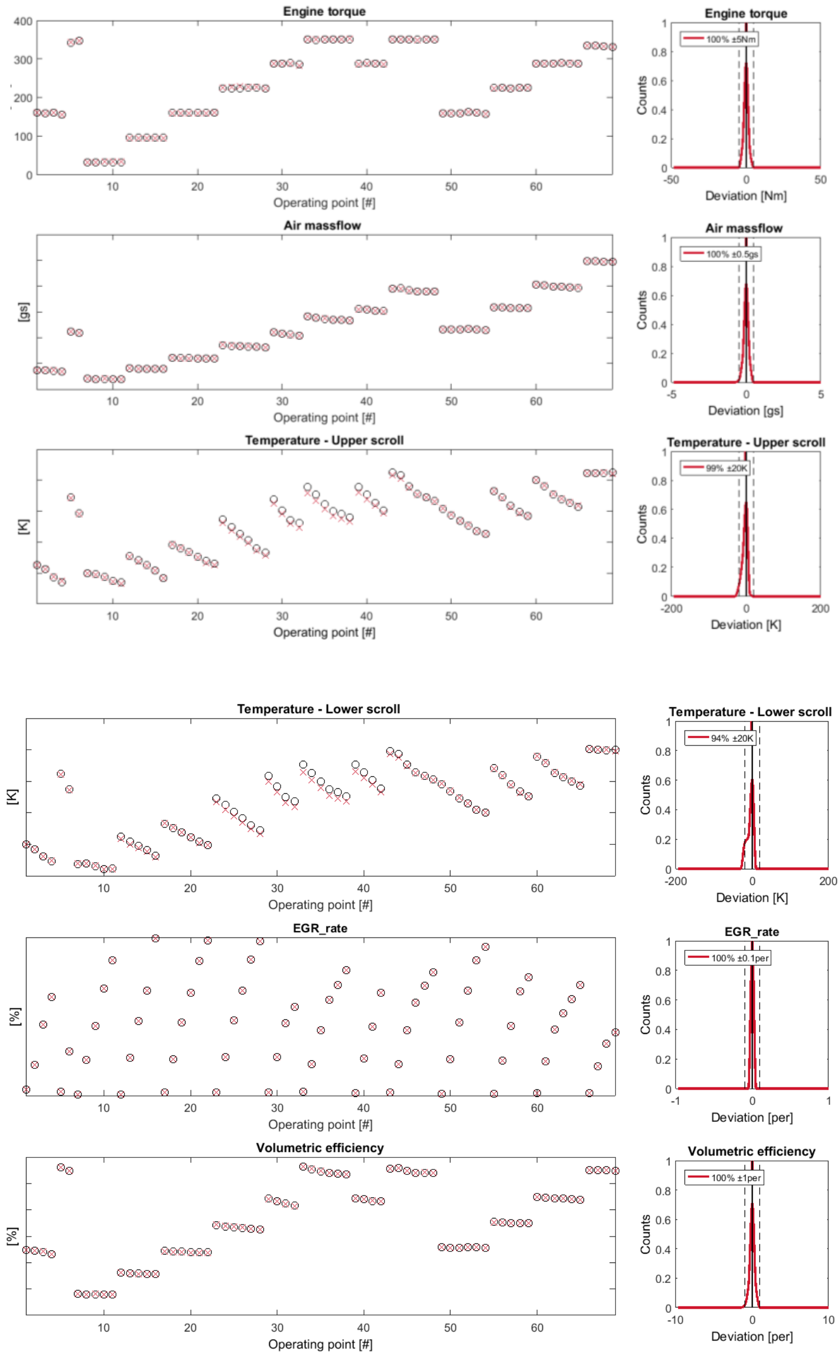

Figure 49: Back-to-back comparison

Equations for error calculation:

$$\textit{Absolute error}: \quad error = simulated - measured \qquad (2)$$

$$\textit{Relative error}: \quad error = \frac{simulated - measured}{measured} \cdot 100\,\% \qquad (3)$$

As you can see, there is a perfect match between the measurement and simulation for the engine torque and volumetric efficiency. The EGR rate is still controlled by the PID. Deviation is present in case of the exhaust manifold temperature. It is related with the exhaust energy fraction maps. I compared data from tool (before the maps are created, just PID tuning results) and from the maps during validation simulation. Difference is present but is small, in order 10^{-3}.

There are two possible explanations why so small deviation created more than 20 K difference in the exhaust temperature. A sensitivity of the exhaust energy fraction can be very high or the value of exhaust energy fraction in the map is not correct. The reason why the value could be different from the optimal value is that we used the convergence criteria or maximum simulation time to stop the simulation. The convergence criteria have been chosen as a 0.05 percent change of the engine torque and air mass flow. If the PIDs had a problem to converge and they oscillated, the simulation would run for its maximum time duration and would be stopped to avoid too long simulation. All the points have been checked for this issue and the oscillating points have been simulated one more time with improved PIDs setup to avoid oscillation.

The convergence criteria have been set only for engine outputs (engine torque and air mass flow) but did not consider change of the engine input parameters. As an engine inputs parameter, I can use tuned efficiency values. It could happen that the engine outputs were steady but the engine inputs (values of efficiencies) was slightly changing.

As a contra measures, I proposed:

- The engine inputs can be also considered as a one of the convergence criteria.
- Additional operating points.

I did not implement such a criterion into the process at that moment trend model has been created. It will be necessary is advanced phase of MPC development when accurate MVEM will be needed.

Once the maps are validated in standalone, we can procced to the second phase. The next step is validation of the maps in the entire mean value engine model which includes the entire air path and turbocharger. Second validation step consist on imposing the boundary conditions (intake pressure and temperature, exhaust pressure and temperature), engine speed, target lambda, FMEP as a map and the other variables. I control the EGR valve, waste gate, throttle, amount of fuel in respect of lambda, boost pressure, intake manifold pressure and lambda. This model also used our maps. The maps outputs are imposed to cylinders.

10.3. MVEM validation

I took the existing mean value engine model of the original engine, introduced the calibrated EGR line and put the new maps in. I used *lookupMD* template and chose *MultiDGridData* as an option how to define our 3D maps.

I was able to achieve the results as shown in Figure 50 with the last version of the maps. In the figure are presented only selected results.

No HT stands for the maps created without heat transfer model and is marked with a triangle. I used a star to mark the point with heat transfer model in the exhaust manifold and cylinder. Circle represents the reference value which is value from the measurement.

EGR rate is targeted and controlled by the PID. As we can see, there is a match for all the points except one, where the deviation is small. It can be issue of PID or EGR rate did not get steady state during the simulation. The plot of EGR valve position confirms the assumption from 9 where we could see that one valve position can have a more EGR rates. It shows the deviation between EGR rate and valve position. It would be difficult to achieve a good EGR rate accuracy by imposing the EGR valve position from measurement.

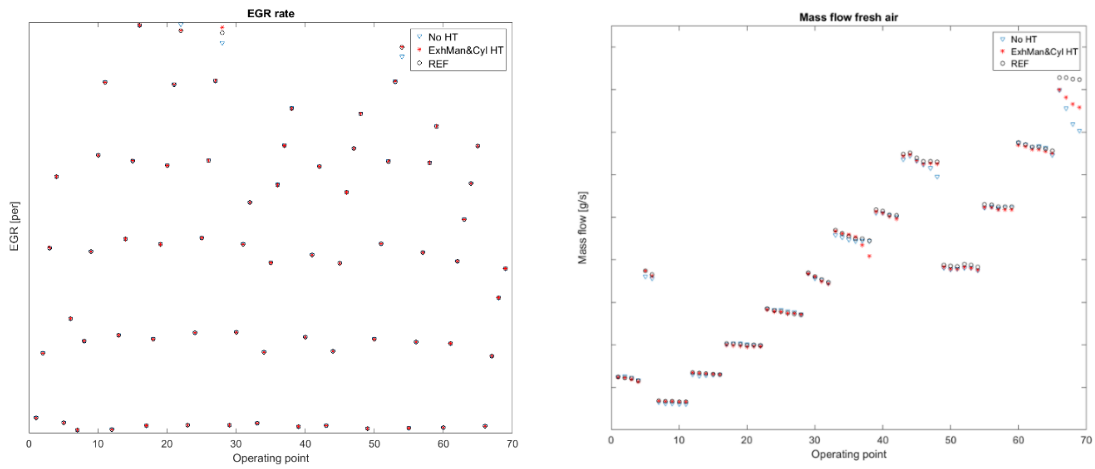

Figure 50: Selected final results of MVEM. EGR rate on the left and mass flow on the right

The boost pressure was targeted and is matched if the compressor was able to provide sufficient boost as it is obvious from the plot. Compressor speed was not controlled nor limited. Compressor speed varies from the reference value because the pressures and temperatures before and after the turbine are not matching perfectly.

The deviation in the exhaust gas temperature and exhaust pressure in both scrolls was much bigger than I expected. I did not expect such a big deviation, especially in the exhaust manifold temperature because back-to-back comparison of the maps is much better.

I investigated why it is so and I noticed that the entire MVEM uses also a thermal connection between the exhaust manifold and prevolute, turbine housing and turbine wheel, which generate an additional heat loss. It has not been simulated during the PID phase in tool when I obtained the efficiency values, due to the fact prevolute, turbine housing and turbine wheel are downstream the exhaust gas measurement point. Therefore, our exhaust energy fraction is too low because the model neglected this additional heat flux from the parts upstream the measurement point to the parts downstream the measurement point and it probably generated the error in the simulation.

I must start with the exhaust temperature and pressure if I want to improve the accuracy of MVEM. I believe that the additional heat losses which are neglected in the tool are causing this error. Originally, I did not implement heat transfer model and the error was even bigger. This claim also supports the fact that the adding of heat transfer model in components presented in the model closed the gap between temperature upstream the turbine more than downstream the turbine.

To update the model of the tool is not so easy because the prevolute, turbine housing and turbine wheel are downstream the measurement point, so I cannot easily connect them. If I would do so, I would lose the correct temperature in the measured point and I would have to calibrate air path properly to achieve a match in the measured point.

I accepted the overall results and I decided to do not investigate more in this step. To improve MVEM accuracy I suggest to:

- Increase amount of operating points as already mentioned above.
- Add engine input parameters as an additional convergence criterion.
- Improved heat transfer model (imposed wall temperature of the prevolutes, added mass of components downstream the measurement point on exhaust side.

The first step in an improvement would be to improve the heat transfer model of the tool. Then I would create the new maps and if the problem would be still present, I would consider the next steps.

For the MPC development it is necessary to keep in our minds that the points where it was not possible to reach the targeted volumetric efficiency or engine torque could be also problematic in the MPC development stage. These points can indicate poor MPC performance due to the unfulfilled targets if we would target performance of the real engine. The model does not reach the performance of real engine especially for high load/speed. We have to target the operating range of model, not the real engine. However, this error would be a problem of the model, not controller design.

11. Detailed engine model

A detailed engine model is a model with the combustion simulation. One-dimensional equations are solved to predict the flow in the intake and exhaust system. The model consists of the air path, modelling of in-cylinder combustion and sub-system such as turbocharger and charge cooler. For now, I decided to focus on steady state operation and do not investigate transient performance of the engine model. This should be considered for further development. Transient performance is one of the MPC controller criteria.

11.1. Selection of initial model

The several models of our mass production engine already existed in Toyota, but they were developed for the different applications. I decided to do a simple comparison to select which detail model will be the best as an initial model for the implementation of LP-EGR. I took the existing models in the condition they were, and I created simulation condition from our measurement data. The model with the most corresponding model will be used as an initial model for my application.

Originally, I had three models. I am going to call them model A, model B and model C.

- Model A – Good details of the air path and turbocharger, charge cooler is simulated, waste gate controls boost pressure, throttle controlled the intake manifold pressure. Calculated wall temperature.
- Model B – Used imposed wall temperature for the entire model. Simplified intake and exhaust runners. Charge cooler is simulated, waste-gate controlled boost pressure, throttle controlled the intake manifold pressure.
- Model C – Was simplified model from the point of view of the air path and turbocharger. The model used imposed wall temperature for the entire model. The waste-gate controlled boost pressure.

I checked how the models are built and controlled even before I created the case setup. I focused mainly on the air path structure, intake and exhaust port modelling, temperature simulation and turbocharger. After initial review I excluded model C. Model C had simplified turbocharger model and air path was much more simplified as well in comparison with models A and B. The aim is to support MPC development and validation process for the air path control, therefore it is important to have a good air path model.

I created a simulation condition for model A and B based on the request of each model and I run simulation. After the results comparison, I decided to use a model A because the gap between the simulation results and the measurement was smaller than in case of a model B.

The LP-EGR path had to be implemented in to the selected model. LP-EGR line was already created and calibrated for MVEM but it needed to be calibrated for the detailed model with the pressure pulsation. I connected the LP-EGR model to the engine model A in respect of the reality. By this, the phase of model building had been finished and I could start the second phase which is to calibrate the model by available measurement data.

11.2. Calibration of detailed engine model

I had a wide range of the measured properties for the calibration. These values allow me to create a finely calibrated model which will be possible to use for the final MPC architecture validation. It will also be used to create the additional operating points for MVEM as already mentioned above.

I went through the calibration in following order:

A. Pressure downstream the compressor
B. Temperature downstream the cooler
C. Intake manifold temperature
D. Intake manifold pressure (if throttle targets mass flow)
E. Volumetric efficiency
F. Combustion
G. Back pressure
H. Exhaust temperature
I. Friction mean effective pressure

At the end of the process I should obtain a model with correct air mass flow and matching brake torque and power. [42]

11.2.1. In-cylinder pressure and combustion calibration

Based on an experience in TME, the cylinder pressure profile calibration is time demanding, even though TME has an automatic tool to make it faster and more user friendly. That was the reason why I started with combustion calibration at the beginning.

Meanwhile I was able to select and modify the detailed model as mentioned in chapter 11.1. Then I started with the calibration as indicated above. The in-cylinder pressure profile calibration is composed from three phases which I call:

- Valve lash and timing DoE – to find the best valve lash and valve timing to minimize IMEP and air mass flow difference.
- Three pressure analysis (TPA) – to analyse initial in-cylinder condition.
- Closed volume model, SI-Turb – to calibrate combustion.

Valve lash and timing DoE

The purpose of DoE, which stands for a design of experiments, is to find the optimal valve lash and valve timing of the intake and exhaust valve in order to minimize air mass flow difference and IMEP difference. The model of the valve lash and timing DoE is the same as the model of TPA which will be explained below. Three operating point are chosen for DoE purpose – low load/speed, mid load/speed and high load/speed. This selection should help us to cover most of the operating range. Ninety combinations of intake valve lash, intake valve timing, exhaust valve lash and exhaust valve timing are used for each point. It generates 270 combinations. More points or combinations could be used but it has been found that additional cost in term of longer simulation time are not worth it because the accuracy improvement is not so significant.

The valve lash variation is between 0 and 0,3 mm and the valve timing variation is spread between +/- 5 CAD. I used an optimization software to select the best combination of valve lash and valve timing to minimize the air mass flow error and IMEP error. The optimization software was trained by the 90 combinations in the three operating points and therefore was able to find such a combination which did not have to be exactly used during the DoE simulation. The selected combination is the same for all operating point. These values are used during TPA.

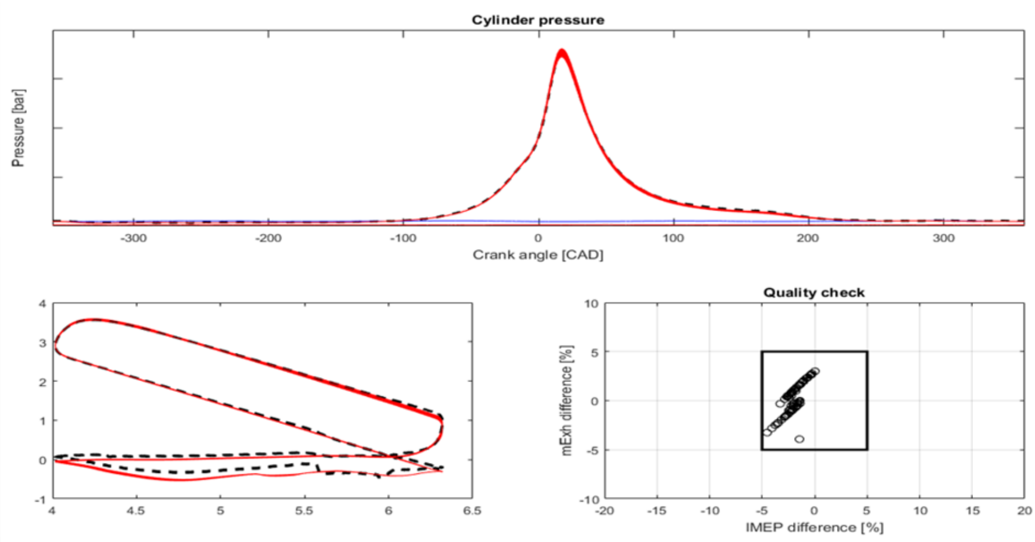

Figure 51: Variation of pressure during Valve lash and timing DoE

TPA

TPA is acronym of three pressure analysis and is used to calibrate the cylinder pressure. The TPA also helps us to determine the values which are impossible or difficult to measure directly. The output of TPA are the properties of initial flow in the cylinder. Trapping ratio, residual fraction or apparent burn rate are calculated during the simulation. Three measured instantaneous pressures are needed as a main input and their measurement needs to be crank angle resolved. One of the pressures is in-cylinder pressure. The other two are intake pressure and exhaust pressure.

These pressures should be measured as close as possible to the cylinder. The runners would be an optimum location for the pressure measurement. In our case was not possible to place pressure transducers to the runners because of the cooling water jackets. Accordingly, the pressure sensors have been placed to the intake manifold and turbine prevolute.

Outputs of TPA

The following properties are an output of three pressure analysis and are unique for each operating point:

- Swirl number
- Tumble number
- Turbulence strength
- Turbulent length scale
- Mean flow strength
- Trapping ratio
- Residual fraction
- Cylinder pressure shift
- Heat transfer multiplier
- End of combustion

These properties are needed for next step which is SI-Turb.

Types of TPA and simulation methodology

In the GT-Power manual [42] are announced two types of analysis – 'TPA steady' and 'TPA multicycle'.

- The purpose of TPA multicycle is to analyse all the cycles to understand the cyclic variation occurring. Instantaneous pressures cannot be averaged for this type of analysis.
- The TPA steady is mentioned as a more common approach of steady-state operating condition and I used it for our application. The pressures are usually averaged or are used pressure traces of one cycle only. As mentioned in 7, as an instantaneous pressure I used an average value of 200 cycles.

The model for TPA

The main advantage of TPA is the prediction of cylinder trapped quantities, including trapping ratio and residual fraction. The accurate flow characteristics of the model are needed to be able to simulate these quantities accurately. For this reason, TPA does not use the entire detailed model. A TPA model consists of one cylinder and its associated ports and valves. As the boundary conditions are imposed measured instantaneous intake and exhaust pressures.

During the adaptation of TPA model is necessary to do couple modifications. First, a special boundary template has to be used for the purpose of TPA. It allows us to enter intake and exhaust manifold pressure profiles as a function of crank angle.

Also fuel injector needs to be replaced. The intake and exhaust runners use variable friction multiplier to dump fluctuations caused by valves. It is achieved by simple control logic.

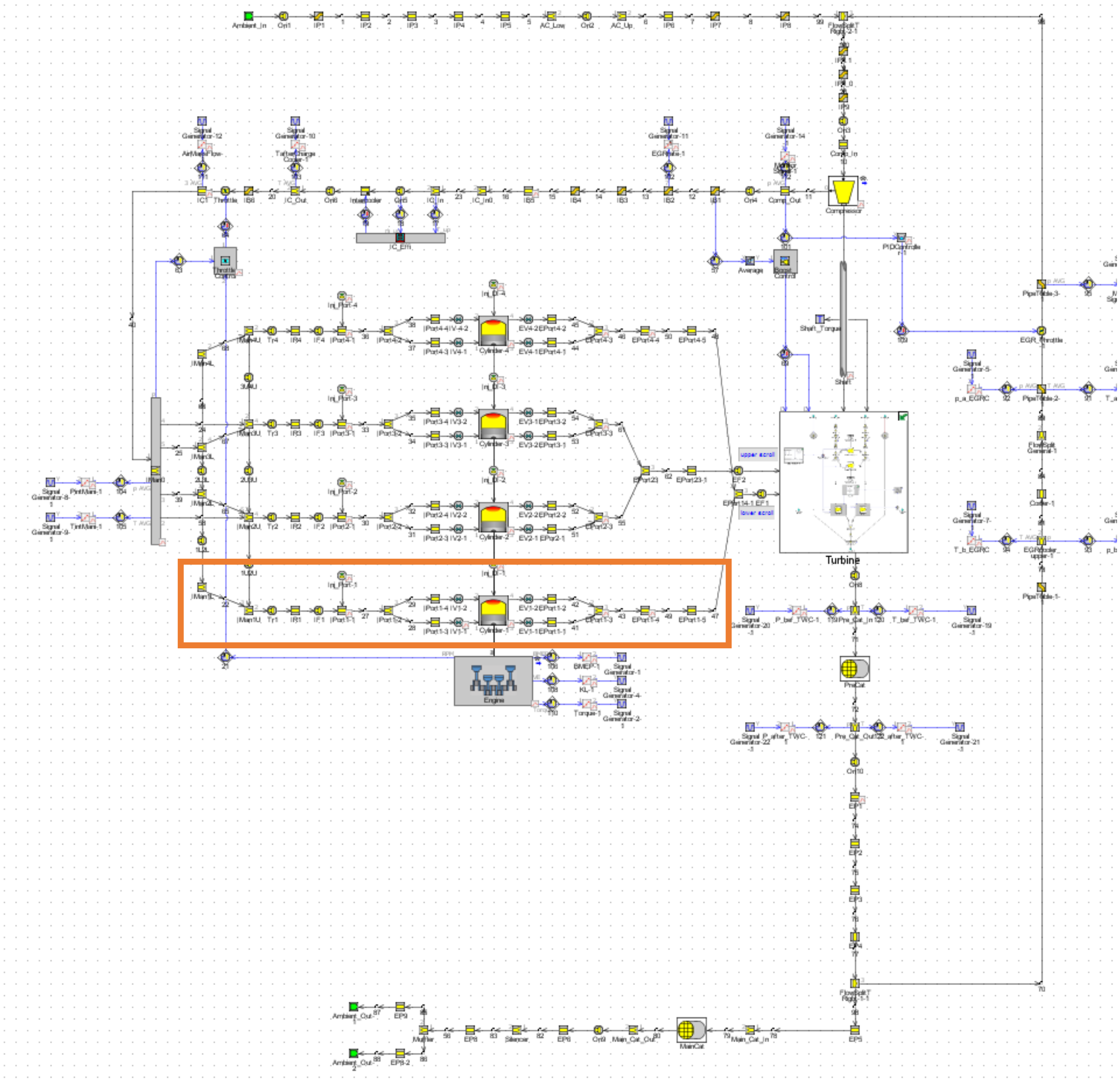

Figure 52: Overview of detailed model with approximately selection of TPA model

An additional change is reduction of discretization length and maximum time step in simulation.

The following data are required on top of the three instantaneous pressures: [42]

- Cycle average intake manifold temperature (measured at the same location as instantaneous pressure).
- Cycle average exhaust pressure.
- Cycle average air mass flow and EGR rate.
- Fuel injection data, such as an injection timing, injected mass, injection profile, AFR.
- Spark timing.
- Geometry data (engine cylinder, ports and valves).

The distance of boundary template from cylinder template has to match with pressure sensor location from cylinder in real engine. This is important for an acoustic. Discretization lengths are reduced for TPA application about half of normal values.

In TME have been developed a tool to make this process automatic. Automatization of TPA leads to significant reduction of calibration time and supress potential human mistake.

TPA tool available in TME is able to run all the operating points in one simulation. The GT-Power is coupled with Simulink for this purpose. The tool has a following simulation methodology which has been automatized:

- First initial cycle with dummy burn rate.
- Second cycle calibrates exhaust pressure shift.
- End of the combustion is calculated during the third cycle.
- Heat release rate is calibrated in last cycle.

The exhaust pressure shift has an impact on volumetric efficiency and in-cylinder pressure. If the 'Appearance heat release rate' becomes zero combustion is considered as a terminated. We add additional 5 CAD because of a noise to be sure that combustion have been really terminated. Its effect is in the tuning of in-cylinder pressure and exhaust pressure.

The tool uses the predefined grid of heat release rate multiplier which is used for the calibration. The heat release rate multiplier with lowest error of IMEP is selected after all. The model used the fixed component temperatures even though the temperature varies mostly with load. The heat release rate should be able to compensate this, and it varies from one to two. The GT-Power manual does not recommend using higher heat release rate than two if there is no extra reason for it.

We decided to start with combustion calibration and it will have some drawback. The using of generalized TPA model could have a mismatch in boundary conditions in comparison with detailed model. It can lead to different intake runner temperature, different mass flow or speed of flow. It will have an effect on the initial conditions in the cylinder and it could result in a different power output. I need to keep this on mind during validation of calibration.

Signal selection

Originally, I wanted to use instantaneous pressures related with the first cylinder. Because of the problems with in-cylinder pressure measurement of cylinder one shown in Figure 30, I was not able to use it. It was necessary to use another in-cylinder pressure. After a quick comparison of in-cylinder pressure curves, I decided to use the in-cylinder pressure of cylinder three.

I checked the simulation results of the original detailed model and I did not see any big deviation in the intake manifold pressure traces. I do not expect the change of intake manifold pressure to be a source of error during TPA if the simulation provided us realistic results. I also discussed the results with TME technicians and I have been told that the intake manifold pressure or the intake runner pressure does not vary much and can be considered as an interchangeable.

The exhaust pressure in upper scroll was available and was used as an exhaust pressure of cylinder three.

Final selection of used signal for TPA application

Because of the problems mentioned above, I used following selection of the pressures for three pressure analysis:

- Intake manifold pressure measured near the runner of cylinder one
- In-cylinder pressure of cylinder three
- Exhaust manifold pressure in the upper scroll.

Outputs of TPA

Following properties are an output of three pressure analysis and are unique for each operating point:

- Swirl number
- Tumble number
- Turbulence strength
- Turbulent length scale
- Mean flow strength
- Trapping ratio
- Residual fraction

- Cylinder pressure shift
- Heat transfer multiplier
- End of combustion

These properties are needed for next step which is SI-Turb.

Simulation results allow as to start with SI-Turb. I am not going to show the numbers because each operating point has a unique result. TPA calculated MFB2, MFB50, MFB10-90 which can be compared with results of SI-Turb.

SI-Turb

The SI-Turb is one of the predictive combustion models and represent Spark-ignition Turbulent flame model. This model is capable to predict the burn rate for homogeneous charge, spark ignition engines based on cylinder's geometry, spark location and timing, air motion and fuel properties. [42]

GT-Power model of SI-Turb

The closed volume model is used for the faster simulation. It means there is no gas exchange phase. It makes the model simple as it is shown in Figure 53. The model consists only from single cylinder object and engine crank train.

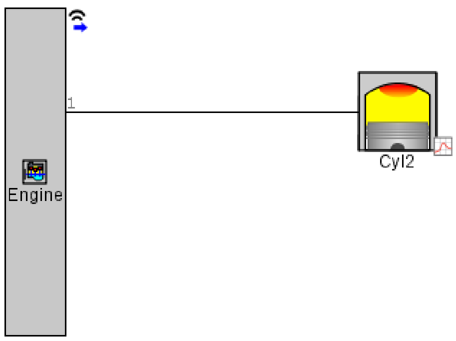

Figure 53: SI-Turb model

Required measurement data and calibration process

At least 25 operating points from entire operating range are needed for calibration. The points should spread over engine speed, load, valve timing and internal or external EGR. [42]

Following data needs to be available at each operating point:

- Output data from TPA
- Dynamic in-cylinder pressure
- Amount of trapped fuel
- Fuel injection profile
- Spark timing
- Exhaust valve opening

Calibration process of SI-Turb

Purpose of calibration process is to find the single set of model constants that will provide the best possible match for the most operating points. I calibrated combustion for the best burn rate match between measured and calibrated one. During this process I tried to find the single set of four calibration parameters which are listed below, to minimize an error between measures and predicted burn rate averaged over all cases.

Following parameters are used for calibration:

- Dilution effect multiplier
- Turbulent flame speed multiplier
- Taylor length scale multiplier
- Flame kernel growth multiplier

The ranges of the calibration attributes are from 0.5 to 3.0 for all of them. I have used Integrated Design Optimizer (IDO) which is part of GT-Power to solve this optimization problem. I set IDO to find minimize '*Improved Burn Rate RMS Error (Meas vs Pred)*'. The search algorithm was 'Genetic Algorithm' with following setting: [42]

Population size	30
Number of generations	34
Mutation rate	0.5
Mutation rate distribution index	15

Table 5: Setting of Genetic algorithm for IDO

It generates 1025 combinations of four parameters.

Output of SI-Turb

The optimization has been run for all 119 available operating points (78 from the EGR sweep measurement and 41 from the reference measurement). The final values of calibration attributes are shown in Table 6.

Dilution effect multiplier	1.085
Turbulent flame speed multiplier	2.994
Taylor length scale multiplier	1.605
Flame kernel growth multiplier	2.989

Table 6:Results of predictive combustion calibration

Beside of this, I compared MFB2, MFB50, MFB10-90 and maximum pressure error with result of TPA. I am going to present only maximum pressure error and MFB10-90.

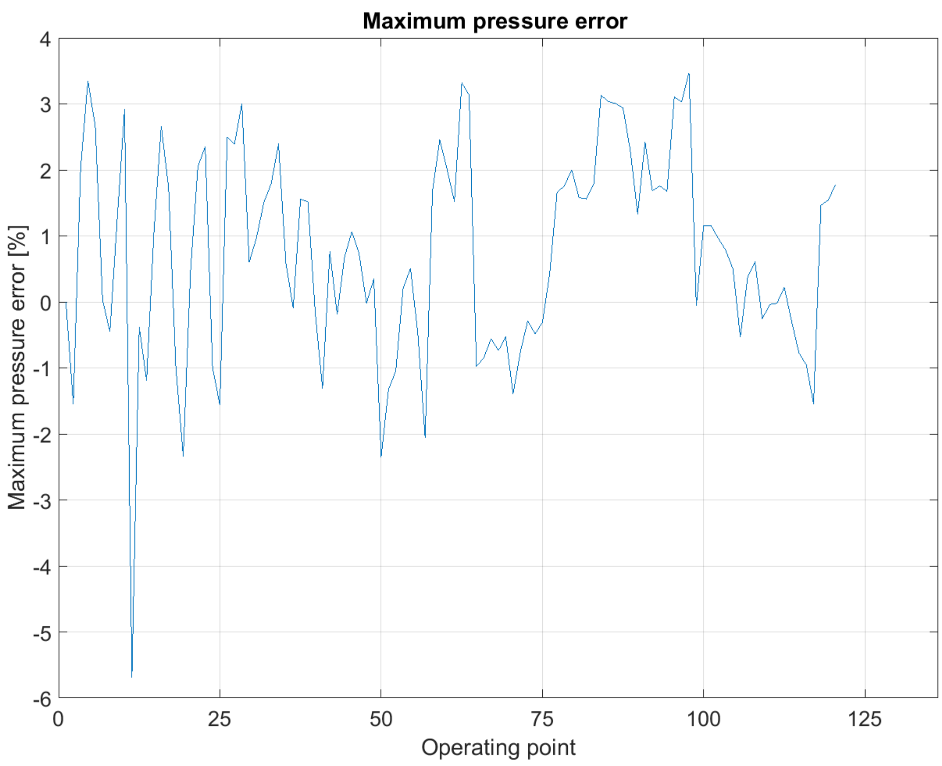

Figure 54:Maximum pressure error, TPA vs SI-Turb

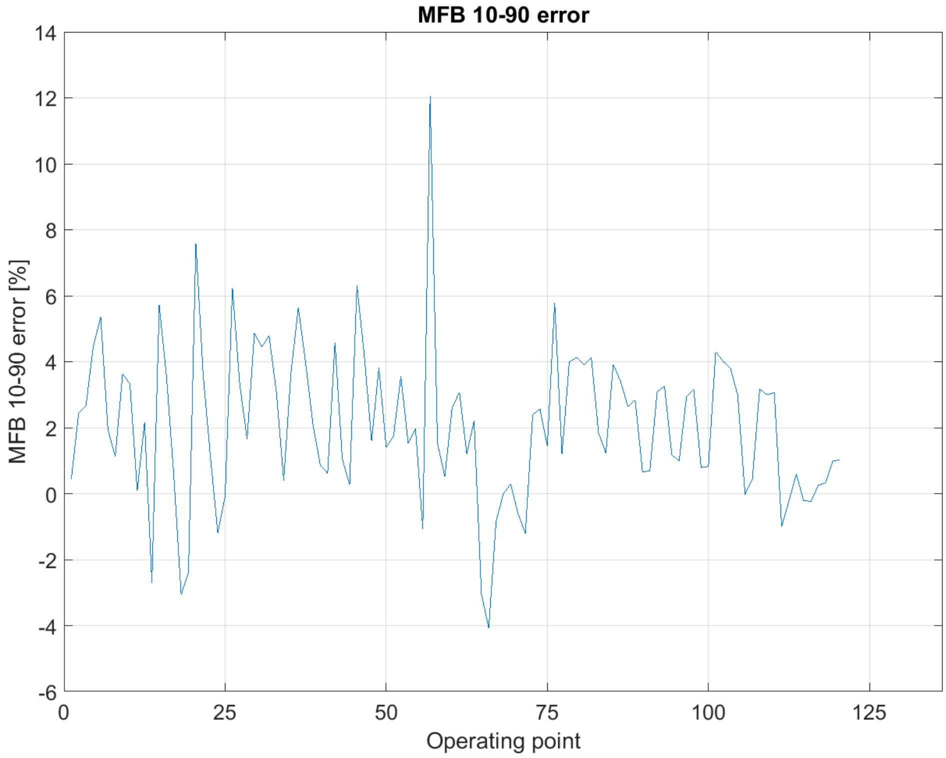

Figure 55: MFB10-90, TPA vs SI-Turb

From the results, we can see that some points have a good accuracy and their error is below 2 %, On the other hand, several points have an error higher than 5 %. Overall, the results are sufficient, but we have to keep in mind the point with poorer performance.

11.2.2. Remaining model calibration

The model itself has the many ways how its accuracy can be improved. It also gives us a lot of possible source of the errors and dead-end roads during the calibration. To be able to say from where a mismatch is coming from it is important to reduce the degrees of freedom of the model. for a detailed model calibration. Otherwise, I would not be able to judge from where a deviation is coming. It is also necessary to procced in small steps to be able to judge the impact of applied change. It could happen that the single changes erase to each other if we would change too many parameters at the same time.

Initial model control

In this paragraph, I would like to mention how the model is controlled, what is imposed and what is a result of the simulation.

The temperature and pressure at the test bench are used as the boundary conditions. EGR rate is controlled by the EGR valve which is controlled by a PID controller. The PID controller is targeting EGR rate from the measurement downstream the compressor. The compressor is controlled by a wastegate opening to match the boost pressure. The intake manifold pressure was controlled by a throttle. Intake charge cooler is simulated as a cooler with prediction capability, while EGR cooler use an unpredictive approach as described in the corresponding chapter.

Engine speed was imposed to the model. Fuel to air ratio is used to define an amount of fuel. Expected final values were selected as the initial conditions to make the simulation converge faster.

Calibration workflow

To be able to calibrate detailed model which consist of initial model A and LP-EGR line I selected following workflow. As a first step, I took the result of model A only (without EGR line). As a second step, I used the entire model (with EGR line) and run the simulation exactly in the same way without any additional changes in setup. Only the points without EGR were selected and EGR valve was closed for this simulation. Base on this result I could judge the impact of EGR line itself on the simulation results. I could procced further because no significant effect was present.

A third step was to select also the operating points with EGR and run the simulation.

I did not observe any kind of unexpected behaviour between the steps. Of course, the results were not the same, they varied, but only slightly. Now I could start to tune the parameters to achieve better match between measurement and simulation.

All operating points are not needed for calibration. I selected 25 operating points, 11 without EGR and 14 with EGR. Their distribution in engine operating range is shown in following figure.

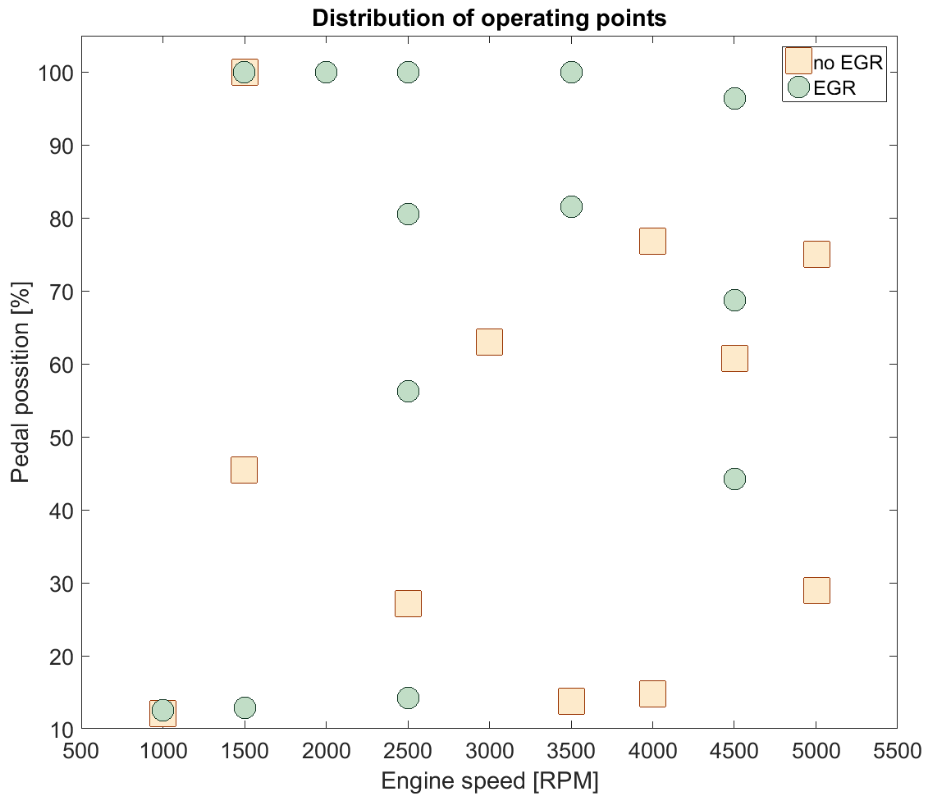

Figure 56: Distribution of operating point used for detailed model calibration and validation

Targeted accuracy

A precise model is needed for MPC validation phase. For MPC validation is necessary to be performed on precise model which should fulfil following deviation criteria:

Torque	3 %
Volumetric efficiency	5 %
Exhaust temperature	20 K (upstream the turbine)
Pressure	2 %

During the calibration process I will try to target mentioned criteria.

Intake part calibration and validation

I will refer to paragraph 11.2 where I listed steps of detailed model calibration and validation. I am going to follow this list now and I will use the corresponding letter.

A) Pressure downstream the compressor

Also called a boost pressure was the first variable what I checked. I did not expect a deviation because it was targeted by turbo. A potential problem could occur if the turbo would not be able to boost enough. As it is shown in the Figure 57, all the points were able to meet the target accuracy of 2 % except two point which are bellow targeted value. I did not do any contra measures to lower the error. I will come back to it after I calibrate pressures and temperatures before and after turbine.

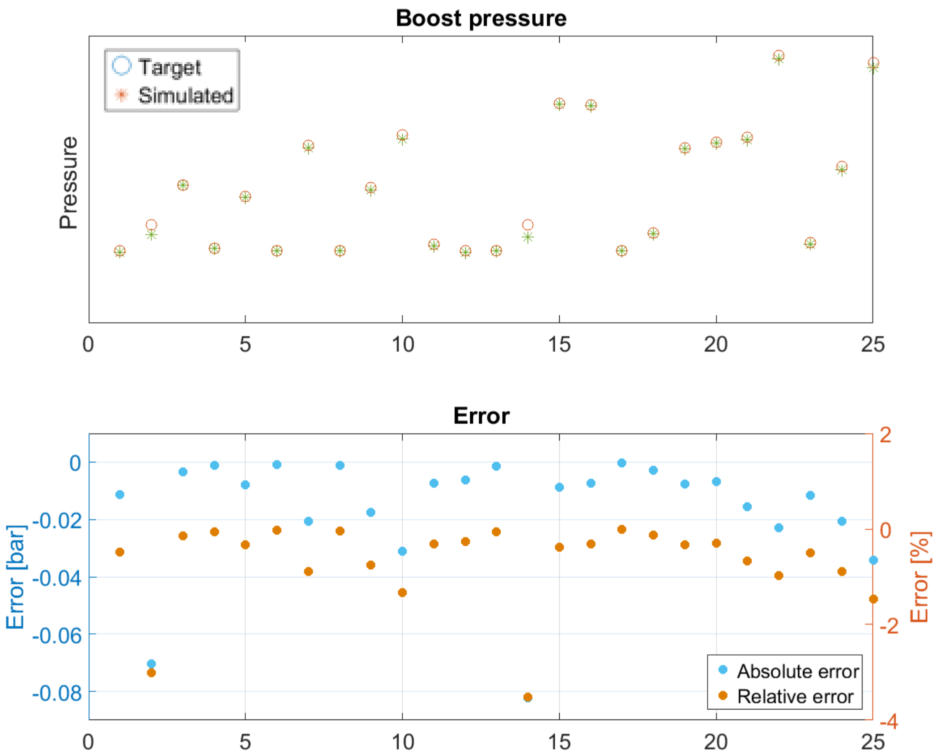

Figure 57: Detailed model calibration and validation: Boost pressure

A-A) Intake manifold pressure

Then I proceed to the intake manifold pressure. The intake manifold pressure is a similar case as the boost pressure because it is controlled by throttle. I did not observe high deviation. The biggest error is around 2 % which is still acceptable deviation. Presented error can be problem of PID or the flow did not get fully steady-state. I did not investigate, because the results are good and further improvement is not needed, at least in the first phase of calibration.

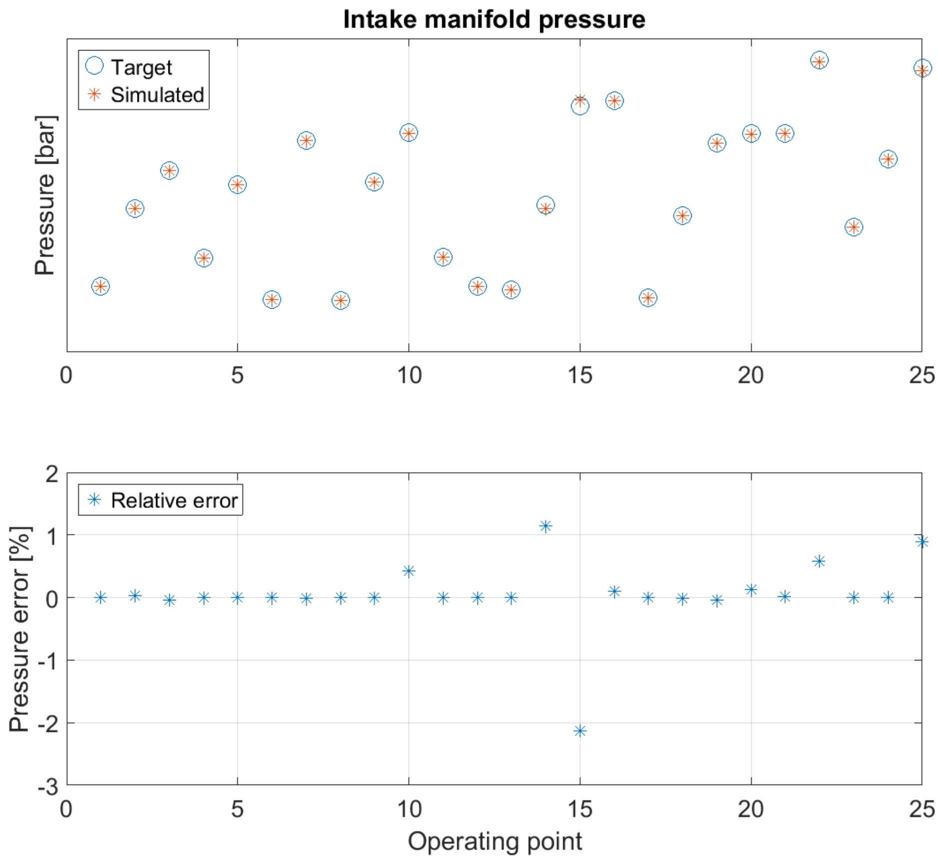

Figure 58: Detailed model calibration and validation: Intake manifold pressure

B) Temperature downstream the cooler

The next step was to check the temperature downstream the charge cooler. A small deviation was present but for the most cases was below 2 K. The deviation was almost constant. I reduced the coolant temperature and slightly increase heat transfer coefficient of the cooler. Temperature error dropped even lower and for the most points is now below 1 K. Therefore, I could consider it as a good result and continue with a following variable.

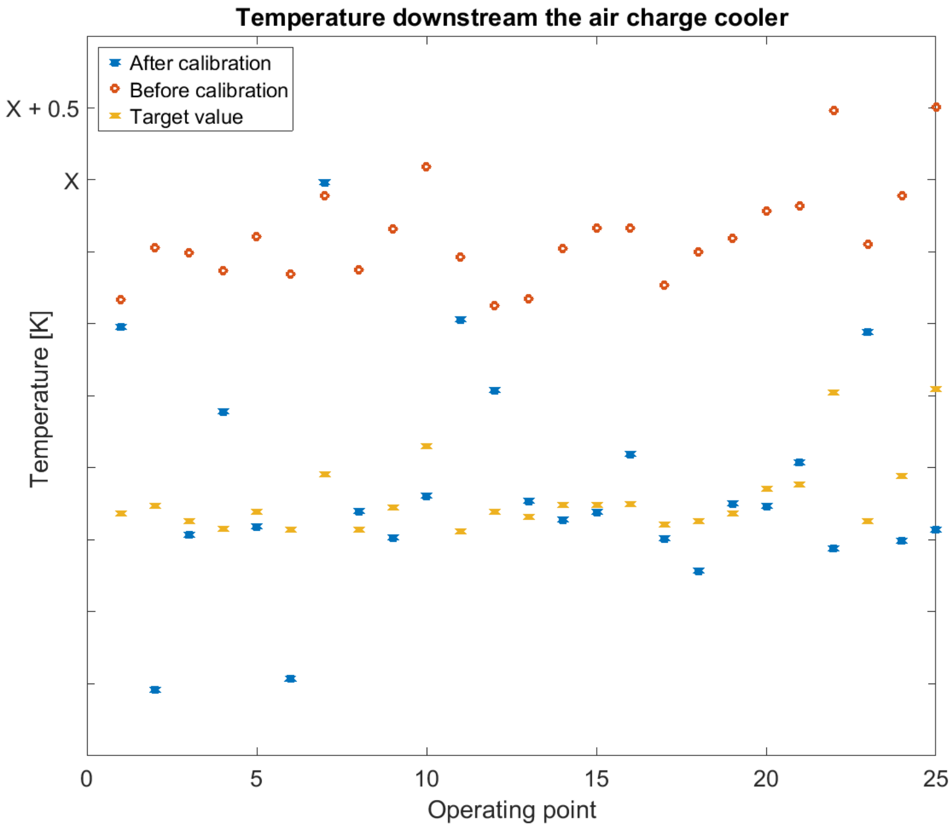

Figure 59: Detailed model calibration and validation: Temperature downstrem the air charge cooler

C) Intake manifold temperature

The following variable was the intake manifold temperature where temperature deviation slightly increases. Temperature of these pipes is calculated. I tried to calibrate the outer conditions of the pipes, but without significant effect. The best results I achieved are presented in Figure 60. For further improvement I would suggest to use a heat transfer multiplier which would be the function of other variable. For example, air mass flow, engine load or speed.

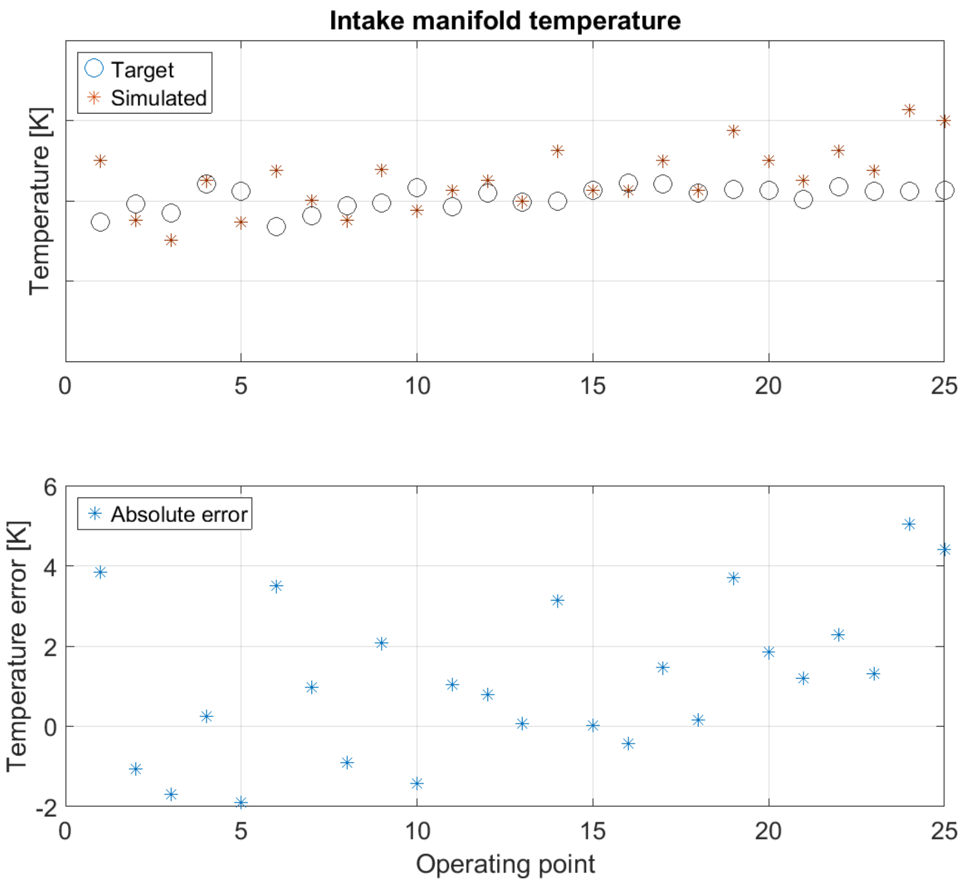

Figure 60: Detailed model calibration and validation: Intake manifold temperature 1

Now I can say that I have an acceptable temperatures and pressures upstream the intake manifold if the boost and intake pressure are controlled. I changed control logic of the throttle and instead of the intake pressure I controlled mass flow of a fresh air. Then I did the same for new control logic to calibrate delta pressure downstream the throttle. For the intake manifold pressure calibration, it is more relevant to use the operating points with higher air mass flow. Such points generate higher pressure deviation and if the pressure will match for them, the operating points with the lower mass flow should not have a bigger error. It is important to have the correct pressure losses because it influences the volumetric efficiency.

D) Intake manifold pressure (when throttle targets mass flow)

Operating point 15 and 16 have a bigger gap between measured and simulated value. These operating points are showing poorer accuracy for the most checked values.

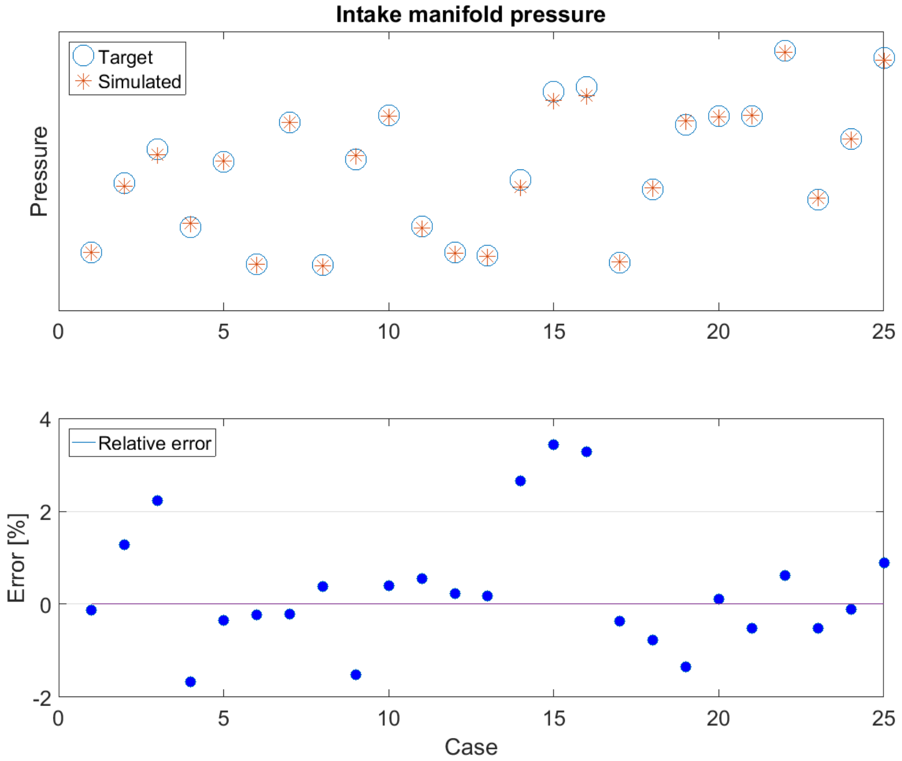

Figure 61: Detailed model calibration and validation: Intake manifold temperature 2

After the validation was done, I came back to previous control logic of the throttle and I kept it for the rest of the calibration.

E) Volumetric efficiency

In the following step, I had to calibrate the intake runners where no direct comparison was available, since no pressure nether temperature was measured. Pressure losses are already involved in discharge coefficient from the flow test bench. Therefore, we can impose the zero friction losses. Instead the temperature can be changed significantly because the intake runners are heated up by the cylinders. A consequence of the temperature will be dominant. The temperature influences the density and it has an impact on the volumetric efficiency which is calculated based on the measured data. The GT-power manual mentions that volumetric efficiency is proportional to average manifold temperature. It has been mentioned that 1 % change in the temperature will change the volumetric efficiency approximately by 1 %. The Intake runners use the calculated wall temperature. I used the heat transfer multiplier to increase or lower the heat transfer between the pipe wall and inner gas.

It is really important to have a correct volumetric efficiency because it is the most influential engine performance datum that affect engine output such a brake torque, power and BMEP.

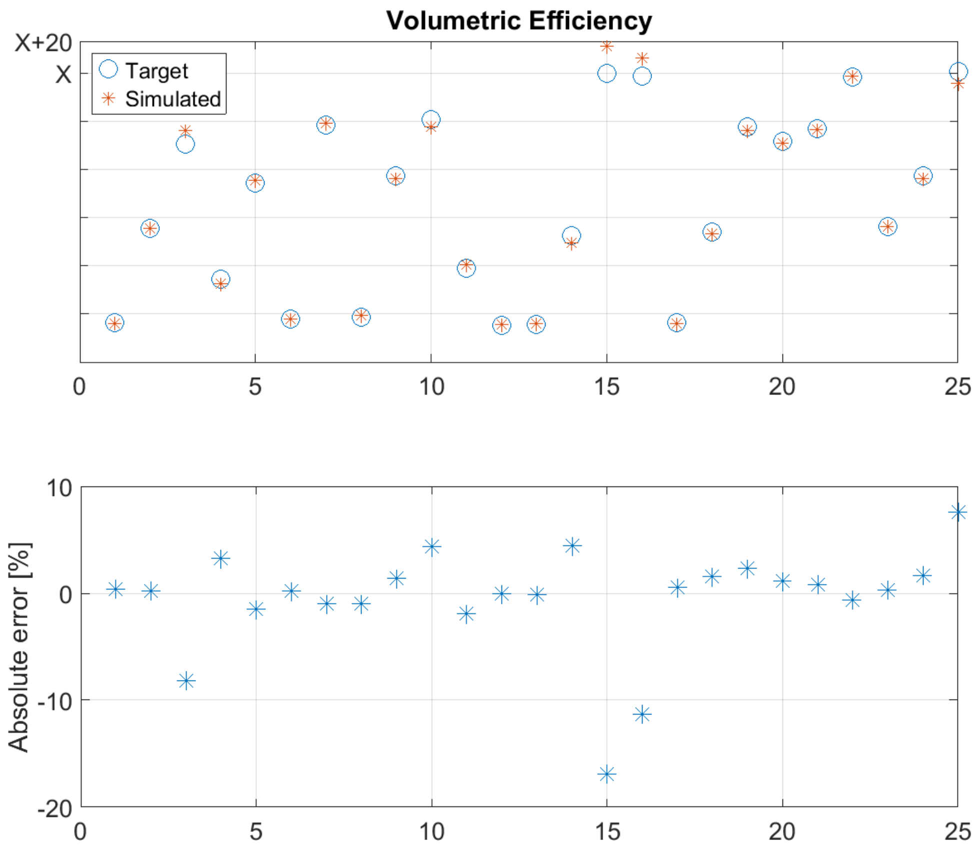

Figure 62: Detailed model calibration and validation: Volumetric efficiency

F) Combustion calibration

At this moment it was the correct time for in-cylinder pressure and combustion calibration. I already obtained the results and now I can impose them. After I imposed the parameters from SI-Turb I checked the combustion results. I focus on BMEP. From the comparison we can see, BMEP and torque are most of the time underestimated. It means that combustion is not well calibrated, we have a lack of air or losses are too high. Volumetric efficiency does not diverge that much to be able to cause this deviation by itself.

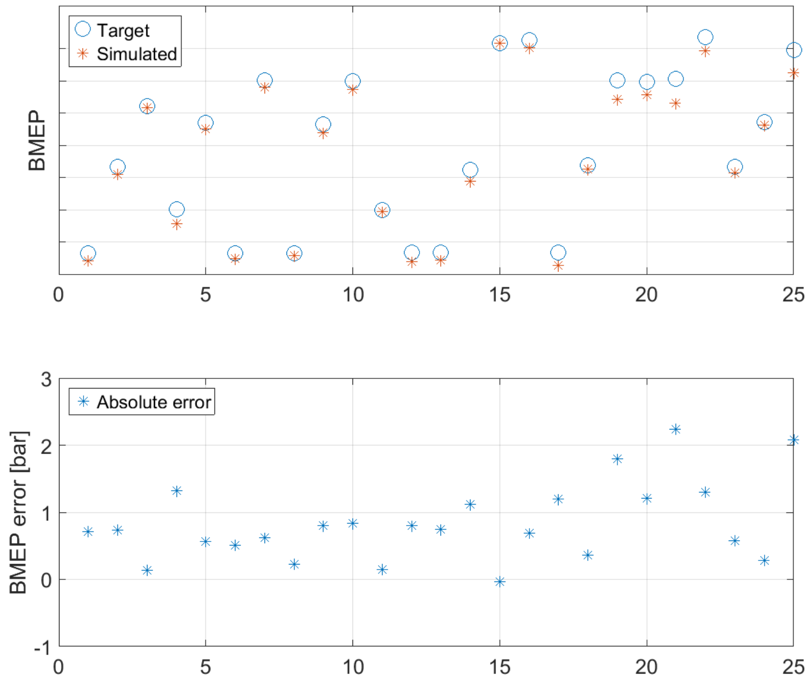

Figure 63: Detailed model calibration and validation: BMEP

G) Back pressure

In-cylinder pressure and combustion calibration were followed by the backpressure calibration. It is tuned by pressure losses. Generally, pressure loss is too high downstream of the location or too low upstream of the location if the simulated pressure is too high. It works transverse in case that simulated pressure is too low. It was necessary to reduced friction multiplier in three-way catalyst to achieve the better results. Even though I was able to reduce the deviation I was not able to achieve desirable accuracy. My suggestion is to start with volumetric efficiency and that to continue with the pressure and temperature in the exhaust manifold.

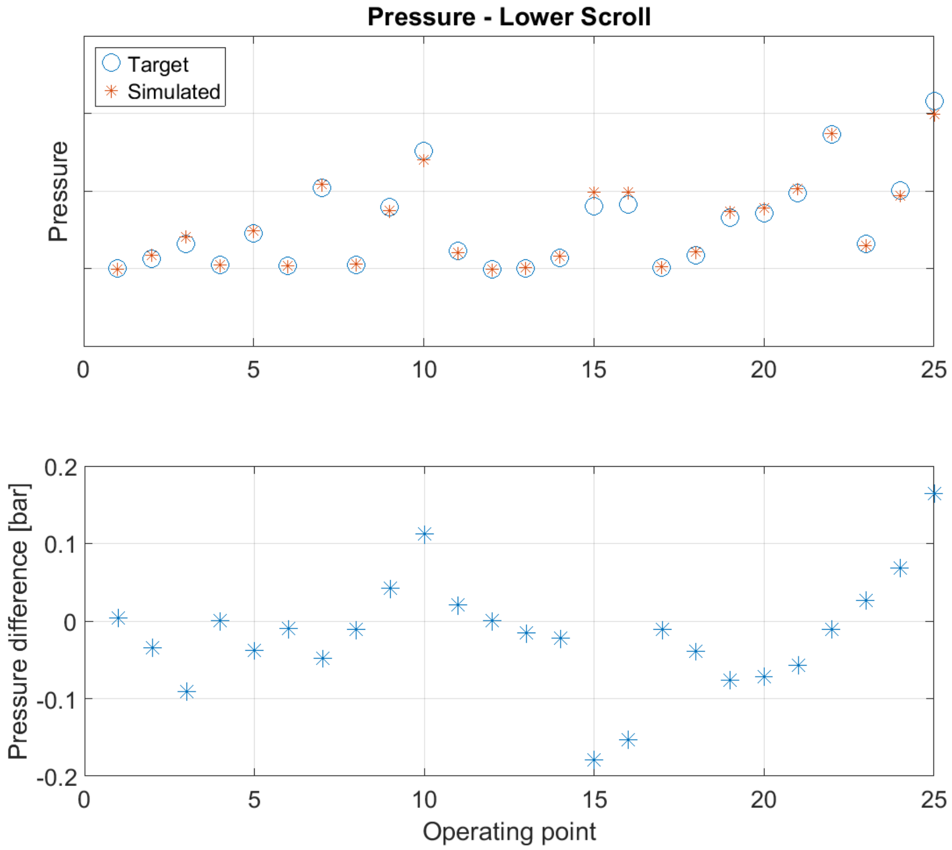

Figure 64: Detailed model calibration and validation: Exhaust pressure - lower scroll

I calibrated the exhaust temperature in the next step. It was necessary to adapt heat transfer multiplier to achieve the reasonable results. I am going to present results of lower scroll. Upper scroll has a similar performance and also the way of calibration was similar.

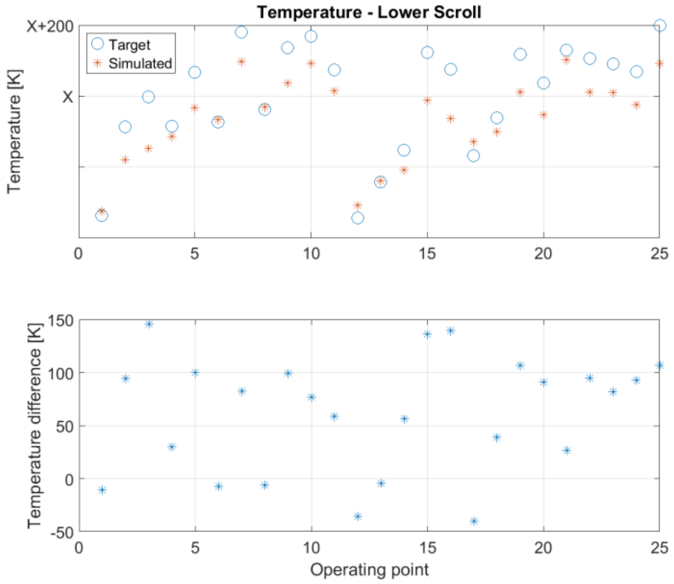

Figure 65: Detailed model calibration and validation: Temperature Lower scroll 1

As we can see, the temperature difference was up to 150 K. I plotted the same results as a function of load and I obtained following trend.

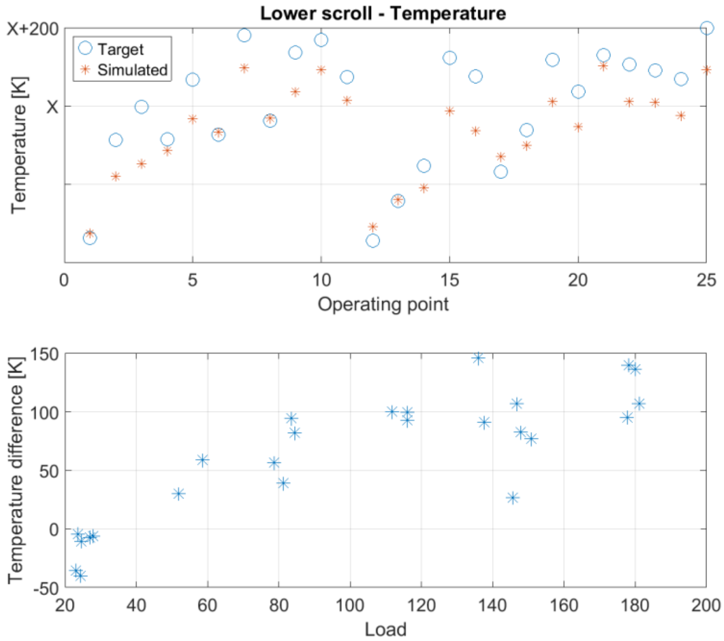

Figure 66: Detailed model calibration and validation: Temperature lower scroll 2

We can notice that the temperature error is load dependent. The operating points with lower load are overestimated and points with higher load underestimated. The heat transfer multiplier for exhaust ports is a function of engine speed and engine load. Therefore, I know that for higher loads I need a lower heat transfer to keep the exhaust gases warm. I changed the HTM map and get a following results.

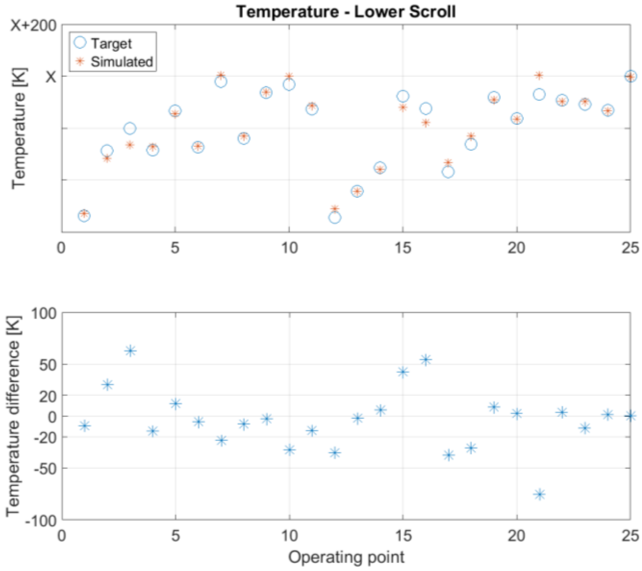

Figure 67: Detailed model calibration and validation: Temperature lower scroll 3

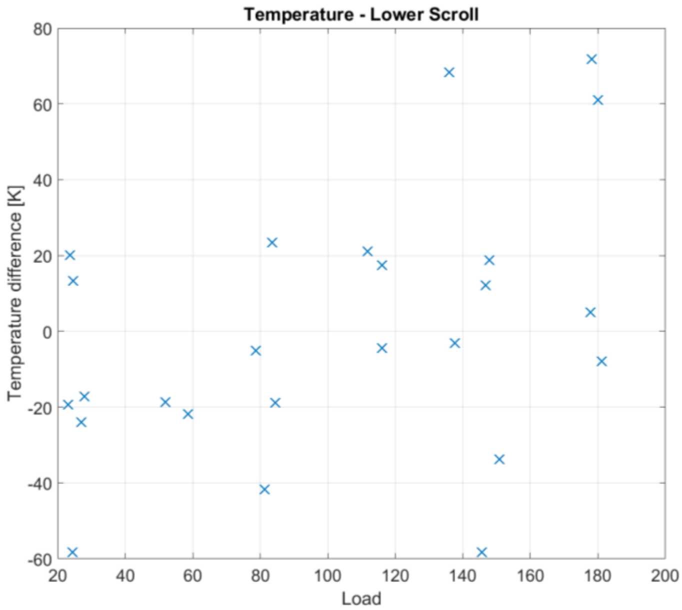

Figure 68: Detailed model calibration and validation: Temperature lower scroll 4

We can notice a significant improvement, even though further calibration would be needed. I also plotted the graph of the temperature error versus load. The trend is not observable anymore. It would be worth it to try dependency on the engine speed, because the heat transfer multiplier map uses it as a second input.

12. Conclusion

In this thesis are presented three model used during the MPC development project.

- EGR-line model
- Mean value engine model
- Detailed model

The EGR line model was used only as an intermediate step and it was able to achieve good accuracy. Further improvement would not have an impact on accuracy of MVEM and detailed model.

The EGR-line was introduced into the MVEM and existing tool was upgraded for multidimensional map creation and validation. After I was able to use 3D maps I created MVEM.

The MVEM is needed in two levels of accuracy to fully support the MPC project. I was able to achieve a trend model accuracy which is fully capable to support MPC methodology study and development. It is necessary to delivered precise MVEM which will provide us precise linearized models for real application of MPC.

As a last model presented in this thesis is detailed model. Selecting detailed model was connected with the EGR-line in the first stage of the project. Then I started with calibration of detailed model to be able to support MPC project in advanced stages. We can consider precise calibration of intake part, where we witnesses only small deviation in term of pressure and temperature. Also, volumetric efficiency was sufficient overall. Problematic was an engine power, where we need to achieved better match between simulated and measured torque and BMEP. Otherwise, model cannot be considered as a precise one.

The detailed model is needed in a precise version only. It is used to give a final validation of MPC before usage on the real engine.

This work was finish during the calibration process of detailed model and further improvement of its accuracy is needed.

13. Next steps

As a next step, accuracy improvement MVEM will be needed to obtain precise models for calibration phase of MPC controller. The precise detailed model will have to be able to allow us to validate accuracy and performance of MPC controller.

In case of mean value engine model, accuracy must be improved. Exhaust efficiency map is apparently facing an issue. This problem is probably related with bad turbine scroll temperature. It can be caused by bad surface temperature and inaccurate heat transfer model. Turbine surface temperature can be imposed based on the measurement data and heat transfer model can be developed more into the detail. If the problem would remain, I would suggest to do sensitivity study of exhaust efficiency.

Calibration of detailed model is on-going.

My suggestion is to re-do the combustion calibration with actual intake and exhaust port conditions. It could improve the combustion calibration and close the gap. It is also needed to check the performance of all operating points and identify weaker areas.

LIST OF USED SYMBOLS

1D	One dimensional
2D	Two dimensional
3D	Three dimensional
AFR	Air to fuel ratio
aTDC	After top dead centre
BDC	Bottom dead centre
BSFC	Break specific fuel consumption
CAD	Crank angle degree
CAE	Computer aided engineering
CI	Compression ignition
CO	Carbon monoxide
CO2	Carbon dioxide
CPU	Central processing unit
DoE	Design of experiment
ECU	Engine computation unit
EGR	Exhaust gas recirculation
EU28	States of European Union
EUDC	Extra urban driving cycle
IGR	Internal gas residual
IVC	Intake valve closure
g/km	Gram per kilometre
GT	Gamma technologies
HC	Hydrocarbon
HP-EGR	High pressure exhaust gas recirculation
km	Kilometre
KPI	Key performance indicator
kph	Kilometre per hour
LP-EGR	Low pressure exhaust gas recirculation
MFB	Mass fraction burned
MBD	Model based design

MFB50	Crank angle for 50 % mass fraction burned
MIMO	Multiple-input multiple-output
mg/km	Milligram per kilometre
MPC	Model predictive control
MVEG-A	Motor vehicle emissions group A
MVEG-B	Motor vehicle emissions group B
MVEM	Mean value engine model
N2	Nitrogen
NEDC	New European driving cycle
NO	Nitric oxide
NO2	Nitrogen dioxide
NOx	Oxides of nitrogen
PEMS	Portable emissions monitoring system
PM	Particulate matter
PN	Particulate number
RAM	Random access memory
RDE	Real driving cycle
RPM	Rotations per minute
SI	Spark ignition
TDC	Top dead centre
TME	Toyota motor Europe
TPA	Three pressure analysis
TWC	Three-way catalyst
UDC	Urban driving cycle
VVT	Variable valve timing
WLTC	Worldwide harmonized Light vehicles Test Cycle
WLTP	Worldwide harmonized Light vehicles Test Procedure
WRAF	Wide range air fuel

LIST OF FIGURES

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