# Maximizing EGR with Heavy-Duty diesel engines

## A concept study

Snijders, H.M.J.

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MSc. Thesis

H.M.J. Snijders
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Advisors: Prof. dr. ir. R.S.G. Baert
ir. R. P. Verbeek

Eindhoven University of Technology
Department of Mechanical Engineering
Division Thermo Fluids Engineering
Section Combustion Technology

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## Abstract

This work is performed as part of a master thesis at the Department of Mechanical Engineering of the Technische Universiteit Eindhoven (TU/e). The concept and assignment were defined by TNO automotive in Delft.

This report describes a concept study of the application of high level Exhaust Gas Recirculation (EGR) on Heavy Duty (HD) compression ignition diesel engines. EGR (Exhaust Gas Recirculation) is an effective way of lowering engine-out NOx emissions by lowering local flame temperature during diesel combustion. Based on previous research at TNO it is determined that in order to reach engine-out NOx emission levels of future (2010) emission standards for HD diesel engines, EGR levels should be high in the range of 30- 70% even at full load (high-EGR).

The purpose of this report is to assess engine-design modifications and their implications, required for achieving high-EGR levels. Furthermore it is aimed at determining which engine configuration is best suited for achieving high-level EGR with future production engines. The US EPA emission standard for 2010 currently has the lowest permitted NOx level and is used as a target level for engine-out NOx emissions and subsequently the required EGR level.

A literature study is performed on the effects of EGR for diesel combustion and on required engine modifications. From this it follows that 'additional EGR' is required in which the AF (air-fuel) ratio is to remain constant. As a consequence the total cylinder charge needs to be increased compared to non-EGR engines. It is calculated that high boost-pressure are required in the range of 5-7 bars, for a 12.9 liter engine with a maximum BMEP of 25bar. For this purpose two-stage turbocharging offers great potential. In order to maintain engine volumetric efficiency and avoid thermal throttling, cooling of the EGR-gas is required. Cooling the EGR-gas can also lower intake charge temperature, which reduces NOx emissions. Cooling of the EGR-gas can be achieved by additional EGR-coolers.

EGR-gas can be routed through the engine in different ways resulting in different engine configurations. Based on literature study the following two configurations are best suited for achieving high levels of EGR:

- External route internally driven high pressure EGR (abbreviated; high-pressure EGR)
- External route internally driven low pressure EGR (abbreviated; low-pressure EGR)

Simulations are performed for both the high-pressure and low-pressure EGR engine configuration with GT-power. GT-power is an engine simulation tool based on one-dimensional gas dynamics. The simulation models are based on the 12.9L DAF MX engine rated at 390kW. The simulation target values are an overall average EGR level of 50% at a minimum air-fuel ratio of 20.3. During simulations the focus is on maximizing EGR. Emissions, both particulate matter (PM) and NOx, are not predicted due to limitations of GT-power on this subject.

From simulation model optimization it comes forward that both the high-pressure and low-pressure configuration require two-stage turbo-charging with the use of a standard VTG and fixed geometry turbo. Charge cooling after each compressor stage is also required. The high-pressure configuration requires turbochargers with a 30% lower mass-flow at equal pressure ratio. The EGR-cooler size needs to be a factor 2 and 5 times the size of a standard DAF EGR-cooler for the high-pressure and low-pressure configuration respectively. The average EGR-cooler wall temperature needs to be low in the range of 313-323K. An exhaust backpressure valve and longer pipe-work is required for the low-pressure configuration compared to the high-pressure configuration.

Based on simulation results it is concluded that the low-pressure engine configuration is slightly better capable of achieving high EGR levels while maintaining a minimum air-fuel ratio compared to the high-pressure configuration. However the high-pressure configuration achieves lower fuel consumption and requires significant smaller packaging. Therefore the high-pressure configuration is concluded more favorable for achieving high-EGR.

## Content:

# 1. Introduction

Increasingly more stringent emission standards for Heavy Duty (HD) diesel engines for road applications, demand lower NOx and PM (Particulate Matter) emissions, as can be seen in figure 1.1. The current most stringent NOx emission level is defined by the US EPA emission standard for 2010 at 0.27 g/kWh. The dilemma generally faced when lowering emissions from diesel engines is the NOx- PM trade-off, which can be seen in figure 1.2. A decrease in NOx emissions has a tendency to increase PM emissions and visa versa at a given engine set-up. Fuel consumption may also suffer from measures aimed at lowering emission.
Exhaust Gas Recirculation (EGR) on HD diesel engines offers the potential for achieving very low NOx emissions levels with relatively small fuel economy and PM emissions penalties. However because of the NOx- PM trade-off the amount of EGR applied, is limited in normal operating HD, CI diesel engines. In practice this means that in order to meet the current European EURO IV emission regulations, a medium level of EGR is applied. Medium-level EGR is maximal up to 30% EGR and 10 to 15% at high engine load. For future emission regulations towards the year 2010 higher levels of EGR are required. These high levels, also referred to as 'high-EGR', are in the range of 30 to 70% even at full load.
Severe engine design modifications are required in order to achieve high-EGR levels and simultaneously minimizing detrimental effects of EGR on diesel combustion and engine performance. Applying EGR on an internal combustion engine can be done in various ways, with various routes for the EGR-gas to travel. This leads to several possible engine-configurations.

This report describes a concept study of the application of high-EGR on Heavy Duty (HD) compression ignition diesel engines. The purpose is to assess engine-design modifications and their implications, required for achieving high-EGR. Furthermore it is aimed at determining which engine configuration is best suited for achieving high-EGR with future production engines. The US EPA emission standard for 2010 is used as a target. The focus however is on maximizing EGR.

In chapter two of this report the effects of EGR on NO formation and on diesel combustion are described based on literature study. In chapter three results from a literature study on the consequences of EGR for the engine design are given. Several engine modifications and engine configurations are given for applying EGR on HD diesel engines. The most relevant engine configurations are described in more detail, and their suitability for applying high levels of EGR are assessed. Also other techniques that may contribute to lowering exhaust emissions in combination with high levels of EGR are given. In chapter four a selection is made of two engine configurations for further research by means of simulation in GT-power. The construction of the simulation models is described. In chapter five the results from simulations are given and analyzed. In chapter six simulation results of both engine configurations are compared. Finally overall conclusions are drawn and recommendations given.

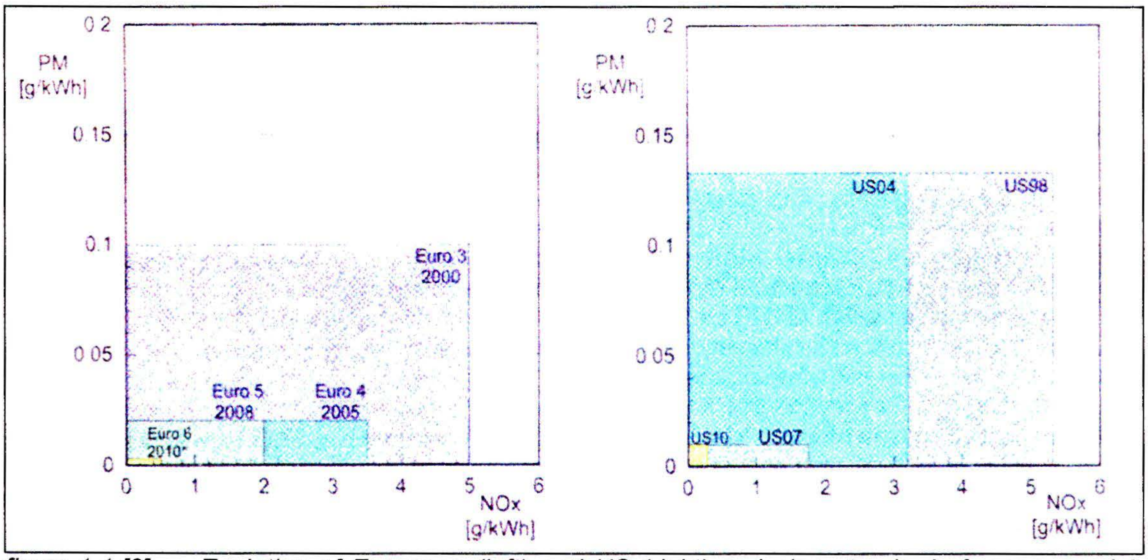



*figure 1.1 [8]* *Evolution of European (left) and US (right) emission standards for commercial vehicles.*

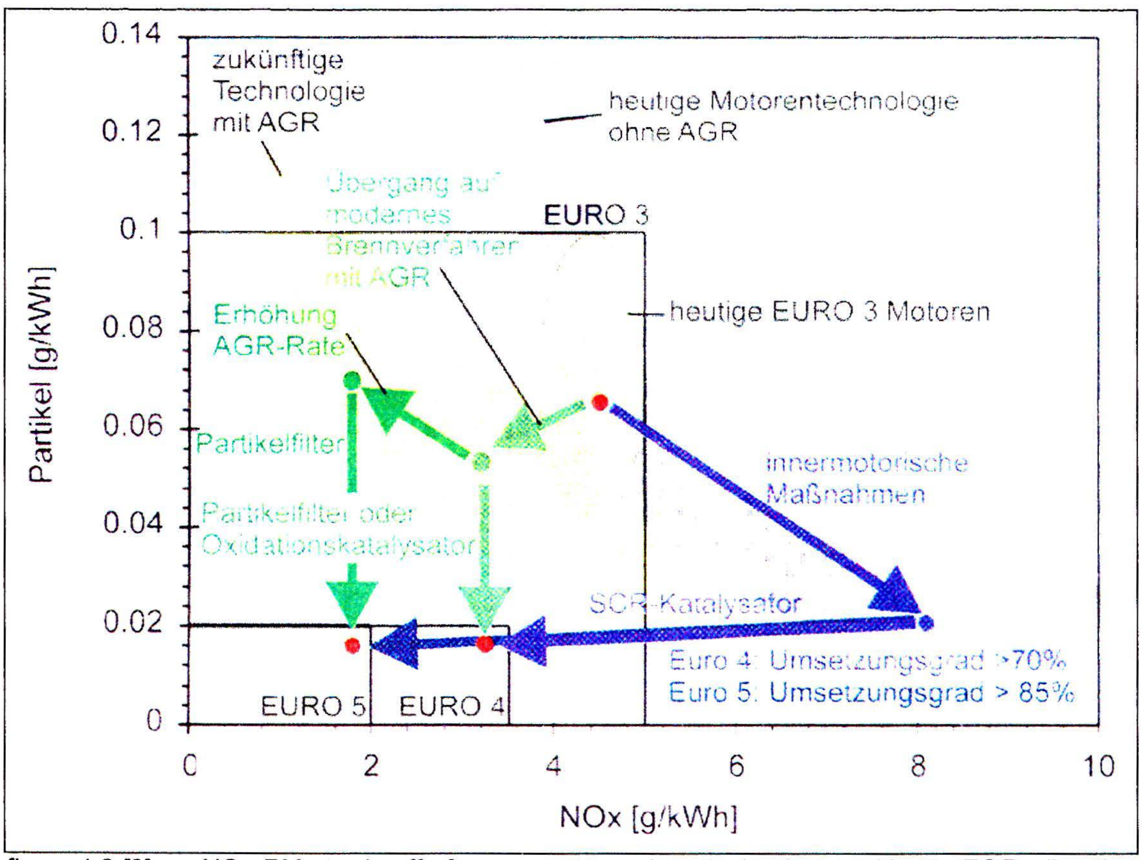



*figure 1.2 [8]* *NOx-PM trade-off for current engine technology without EGR (heutige Motorentechnologie ohne AGR) and future technology with EGR (zukunftige Technologie mit AGR). Heavy Duty diesel engines for commercial vehicles.*

# 2. Effect of EGR on diesel combustion

## 2.1 NO formation and how to reduce it

Exhaust Gas Recirculation has been successfully applied on internal combustion engines both with Otto and Diesel cycle in order to reduce the NOx emission. The formation of NOx in internal combustion engine is often explained by using NO formation theories.

There are two widely accepted chemical pathways that are candidates for NO formation in diesel systems [1]:
1. Thermal or Zeldo'vich mechanism
2. Prompt NO chemistry

A third pathway, conversion of fuel-bound nitrogen to NO, is important only for fuels that contain significant nitrogen within the fuel molecules. Most distillate diesel fuels contain little (<<100 ppm) fuel-bound nitrogen, so this pathway is likely unimportant, and is not considered here [1].

The thermal NO pathway occurs most quickly at high gas temperatures for which equilibrium chemical thermodynamics favor formation of NO, primarily through dissociation of molecular nitrogen and oxygen. The activation energies of the reactions that form NO through the thermal mechanism are relatively high, so formation rates for NO are only fast enough to be significant in engines at temperatures above about 1900-2000 K. The thermal mechanism is most important for stoichiometric and lean premixed flames, and on the oxidizer side of diffusion flames [1].The prompt NO pathway is initiated by hydrocarbon fragments ( e.g., CHx) that react with atmospheric nitrogen to form intermediate species, such as HCN and NHX [6]. Depending on local conditions, these reactive intermediates are either oxidized to form NO, often termed "prompt NO," or react with NO to re-form molecular nitrogen, as in reburning. Prompt NO chemistry is most active in rich premixed flames and on the fuel side of diffusion flames, where there are zones of both significant production and destruction of the intermediate hydrocarbon species [1].
It is generally assumed that in conventional CI diesel engines, flame temperatures are high enough for the thermal NO mechanism to be the dominant NO formation pathway [1]. "The available body of experimental data largely confirms the first-order dependence of engine-out NOx emissions on adiabatic flame temperature, especially for charge dilution (e.g. EGR) or oxygen enrichment" [1; Musculus]. The thermal NO mechanism is most frequently employed to predict engine-out NOx emissions from diesel engines [1]. Therefore the thermal mechanism will be further explained.

Thermal mechanism

It follows from the thermal NO formation mechanism that NOx formation in internal combustion engines, depends heavily on the temperature of the combustion gases. In [2] the kinetics of NO formation are described, it is generally accepted that in combustion of near stoichiometric fuel-air mixtures the principal reactions governing the formation of NO from molecular nitrogen (and its destruction) are:

$$O + N_2 = NO + N \qquad \text{re; 2.1}$$

$$N + O_2 = NO + O \qquad \text{re; 2.2}$$

$$N + OH = NO + H \qquad \text{re; 2.3}$$

These are also referred to as the Zeldovich mechanism.
The initial NO formation rate may be written as a function of temperature and concentration of oxygen and nitrogen;

$$\frac{d[NO]}{dt} = \frac{6\times10^{16}}{T^{1/2}} \exp\left(\frac{-69.090}{T}\right)[O_2]_e^{1/2}[N_2]_e \qquad \text{for } [NO]/[NO]_e << 1 \qquad \text{eq; 2.4}$$

in which

| | |
|---|---|
| $\frac{d[NO]}{dt}$ | increase of Nitrogen-Oxygen concentration per unit of time $t$ |
| $T$ | temperature |
| $[O_2]_e$ | equilibrium concentration of Oxygen |
| $[N_2]_e$ | equilibrium concentration of Nitrogen |
| $[NO]$ | local concentration of Nitrogen-Oxgen |
| $[NO]_e$ | equilibrium concentration of Nitrogen-Oxgen |

As can be seen from equation 2.4 the temperature in the exponential term makes the NO formation rate depends strongly on temperature. High oxygen concentrations also result in higher NO formation rates.

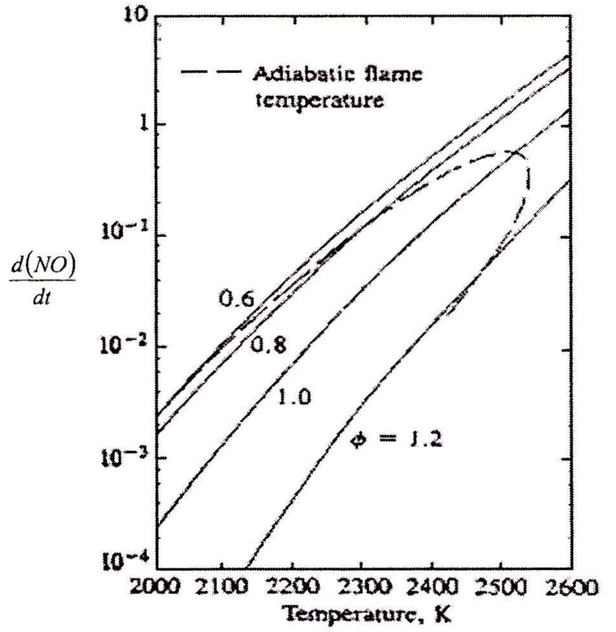



*Figure 2.1: Initial NO formation rate mass fraction per second, as a function of temperature for different equivalence ratios ( Φ) and 15 atm. pressure. Dashed line shows adiabatic flame temperature for kerosene combustion with 700K, 15 atm. air. figure 11-4 [2]*

Prediction of NO formation

Since, according to the thermal NO formation mechanism, the NO formation depends primarily on the temperature, it is generally accepted to make use of a correlation known as 'adiabatic flame temperature' to predict engine out NOx emission.

There is evidence however, that the 'adiabatic flame temperature correlation' to predict engine out NOx (thermal mechanism) has its failures under certain circumstances. According to [1; Musculus]: "it is unlikely that any single parameter (adiabatic flame temperature) could explain all trends in engine NOx emissions".

Correlation has been found between premixed burn and engine NOx emission. "It is generally accepted that exhaust NOx emissions of diesel engines increase with the degree of premixed

burning" [1; Musculus]. A larger premixed combustion phase can result in higher NOx emission even under constant adiabatic flame temperature, under certain circumstances.
However it has not been proven that a causal link exits between premixed burn and NO formation in diesel engines [1].
It is expected that next to the 'adiabatic flame temperature correlation' to predict engine out NOx, a 'premixed burn correlation' with NO formation in diesel engines exists. Meaning that the adiabatic flame temperature can not be regarded as the only parameter in explaining engine NOx emission trends as premixed burn duration may also be of influence.

Reduction of NOx
In order to lower NOx emission of a HD diesel engine, the NO formation rate should be reduced. From the thermal NO formation mechanism it follows that NO formation depends primarily on the temperature, the lower the temperature, the lower NO formation rate. In diesel combustion reducing (local) flame temperatures reduces the NO formation rate and thus NOx emissions.

In figure 2.2 [8] the exponential relationship between NOx formation and flame-temperature is illustrated. According to [4] a reduction in flame-temperature by 20 K could reduce the NOx concentration in the exhaust by as much as 20%.

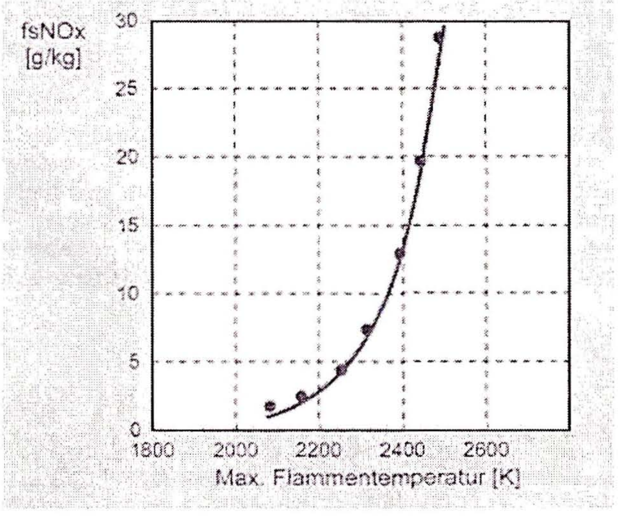



*figure 2.2 [8]: NOx formation as function of flame-temperature (Flammentemperatur)*

Reducing flame-temperatures in HD diesel engines can be achieved by applying EGR or cool combustion concepts such as Homogonous Charge Compression Ignition and rich diesel combustion ($\lambda \leq 1$).
The use of EGR can be effective in lowering the NO formation in a diesel engine via the thermal mechanism, meaning lowering the local flame temperature (*T*) and lowering the oxygen and nitrogen concentration (eq2.4). However lowering the local flame temperature makes EGR most effective in reducing NO formation.

## 2.2 Effect of EGR on flame temperature

Definition of EGR

EGR can be used on internal combustion engines by replacing some of the fresh inlet charge by exhaust gasses; 'replacement-EGR' or by adding the exhaust gas to the fresh inlet charge; 'additional-EGR', keeping the air-fuel (A/F) ratio constant. The percentage of EGR is defined as [2]:

$$EGR(\%) = \frac{mass\ of\ exhaust\ recycled}{mass\ of\ total\ intake\ charge} \times 100\%$$

The most relevant components of the EGR (exhaust gas) for combustion are $CO_2$ and $H_2O$.

EGR & flame temperature

Lowering the local flame temperature makes EGR most effective in reducing NO formation. The lower flame temperature is a result of three effects of EGR in diesel combustion:

1. Dilution effect ; reduction in probability that fuel and oxygen molecules meet
2. Thermal effect ; increase in heat capacity of the working fluid
3. Chemical effect ; dissociation of added (EGR) species of the working fluid

In explaining how these three effects can lower local flame temperature use will be made of a simple thermodynamic combustion equation. This simple combustion equation is based on the first law of thermodynamics, under the assumption that combustion takes place under stoichiometric conditions and that specific heat values of the gasses before and after combustion are about equal. For simplicity, no work is delivered by the system.

$$m_{ch} \cdot Cp_{ch} \cdot \Delta T = m_f \cdot Hu \qquad \text{eq; 2.5}$$

With;

$$\Delta T = (Tb - Tu) \qquad \text{eq; 2.6}$$

Further;

$m_{ch}$ = total mass of the charge participating in the combustion
$Cp_{ch}$ = specific heat of the charge
$\Delta T$ = difference in charge temperature between after- and before combustion (Tb-Tu)
$m_f$ = mass of fuel
$Hu$ = 'Heat of combustion' of the fuel [kJ/kg]

$Tb$ = temperature of the charge after combustion
$Tu$ = temperature of the charge before combustion

The value $m_{ch}$ in eq. 2.5 increases when applying EGR. Applying EGR causes the local oxygen concentration to decrease. As combustion takes place in regions where the air-fuel ratio is at stoichiometric proportions, a lower local oxygen concentration will cause the fuel in a DI diesel engine to be forced to diffuse over a wider area before sufficient oxygen is encountered for stoichiometric mixture to be formed. As a consequence a higher mass of charge, $m_{ch}$, needs to participate in the combustion. This is as a consequence of the *dilution effect* of EGR.

The value $Cp_{ch}$ in eq. 2.5 increases when applying EGR. As EGR gas contains relative large concentrations of $CO_2$ and $H_2O$ with relative high specific heat values compared to oxygen, the total specific heat value, $Cp_{ch}$, of the charge also increases. This is as a consequence of the *thermal effect* of EGR.

As a consequence of the increase in $Cp_{ch}$ the differential temperature over the adiabatic compression by the engine's compression stroke decreases. Leading to a lower temperature *Tu* before combustion.

Applying eq. 2.5; Assume a constant mass of fuel $m_f$ is injected with a constant 'heat of combustion' *Hu*. When applying EGR the terms $m_{ch}$ and $Cp_{ch}$ increase, this will cause the term $\Delta T$ to decrease.
Applying eq. 2.6; when applying EGR the value *Tu* is decreased as a consequence of lower $Cp_{ch}$ during compression, the value $\Delta T$ has also decreased as follows from applying eq. 2.5. Together this leads to a decrease of the value *Tb*. A decrease in *Tb* equals a decrease of the flame temperature.
In figure 2.3 compression and combustion of a CI engine are schematically represented, with and without the use of EGR, as a function of temperature T and pressure P. The curved lines represents the compression phase followed by a jump in temperature, representing combustion.

Besides the already mentioned *dilution effect* and *thermal effect* of EGR in reducing flame temperature a third effect, the *chemical effect*, exist. The *chemical effect* is the result of the potentially dissociation of the $CO_2$ and $H_2O$, from the EGR (exhaust gas), at high temperatures. The products of this dissociation participate in the combustion process. According to [2, 3] energy is required to dissociate the $CO_2$ and $H_2O$ molecules during the combustion process (particularly $H_2O$ which has a highly endothermic dissociation mechanism). The energy is naturally obtained from the high temperature flame front. Leading also to a decrease of *Tb* and thus lower flame temperature.

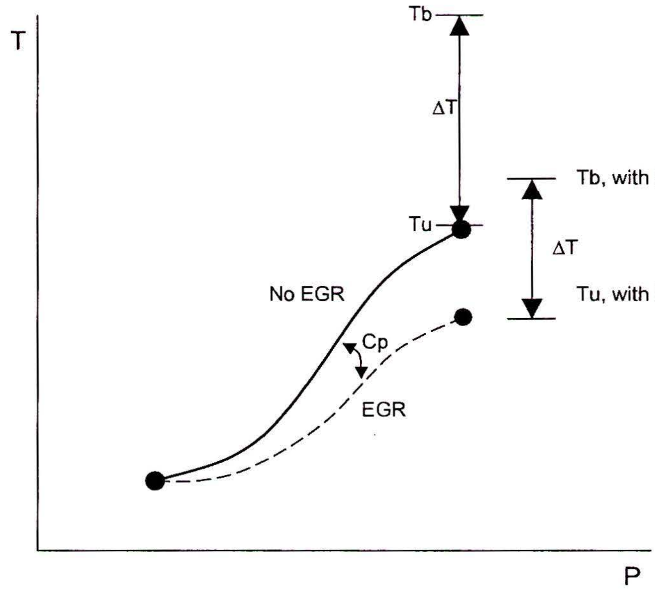



*Figure 2.3: Illustration of simple compression and combustion.*
*Vertical axis; temperature T*
*Horizontal axis; pressure P*

NOx emissions

According to [4] the dilution effect of EGR is the major effect in NOx reduction by means of lowering flame temperature, the thermal effect is of secondary importance. According to [3] the chemical effect of EGR is non-negligible but considerably less influential than the thermal effect. According to [3] the dilution effect also has a very small effect on lowering NOx emissions by means of lower oxygen availability, when applying EGR. This is explained by eq.2.4 in which a lower $O_2$ level leads to a lower NO formation rate.

The degree of premixed burn can have an influence on the NOx emission, however opinions vary on this subject:

"It is generally accepted that exhaust NOx emissions of diesel engines increase with the degree of premixed burning" [1; Musculus]. And in contradiction: "for typical diesel conditions virtually all of the premixed combustion is fuel rich, in the range of an equivalence ratio of 4. This includes both the initial premixed flame after auto-ignition and the hypothesized standing premixed flame during the mixing controlled burn. These conditions are not conducive to NO production either by "thermal" or "prompt" mechanisms. Little oxygen is present and adiabatic flame temperatures (~1600 K) are far below those required for significant thermal NO production. For prompt NO, calculations and experiments show little NO produced at equivalence ratios above 1.8" [58; Dec].

The A/F ratio can be of influence on the NOx emission. "Additional-EGR" keeps A/F ratio constant while "replacement-EGR" leads to a lower A/F ratio when compared to engine operation without the use of EGR. In case of replacement-EGR the $O_2$ concentration will be less than with additional-EGR. A lower $O_2$ concentration will result in an increased dilution effect, increasing the mass of charge participating in the combustion ($m_{ch}$). As can be seen in equation 2.7 below, a higher $m_{ch}$ results in a lower temperature Tb. In equation 2.8 it can be seen that a lower temperature *Tb* and lower oxygen concentration [$O_2$] result in a lower NO formation rate. As a consequence the achievable NOx reduction is less with the use of additional EGR and constant A/F ratio compared to the use of replacement-EGR and lower A/F ratio [6, 7].

The charge temperature prior to combustion proves to have an influence on the NOx reduction, a higher inlet charge temperature leading to higher NOx formation. This because the higher gas temperature prior to combustion (*Tu*) will lead to higher temperature *Tb*. Furthermore it is possible that due to the increased inlet charge temperature the combustion will take place closer to TDC of the engine and thus increasing temperature *Tb*, this of course also depends on injection timing [4]. (".the shortening of the ignition delay caused combustion to take place closer to TDC, resulting in increased gas temperature and higher NO formation rate" [4; Ladommatos])

Equation 2.7 derived from equation 2.5 and 2.6. Tb: temperature of charge after combustion.

$$Tb = Tu + \frac{m_f \cdot Hu}{m_{ch} \cdot Cp_{ch}} \qquad \text{eq 2.7}$$

Equation 2.8 is based on equation 2.4 and shows the dependence of NO formation on temperature (Tb) and equilibrium oxygen and nitrogen concentration ([$O_2$] & [$N_2$]).

$$\frac{d[NO]}{dt} = f(Tb;[O_2]_e;[N_2]_e) \qquad \text{eq. 2.8}$$

## 2.3 Other effects of EGR

Combustion

- *Ignition delay*

Applying EGR on diesel engine may have an influence on the ignition delay. Applying EGR will cause, the oxygen concentration in the total charge to decrease. According to [3; T. Jacobs]: "By reducing the oxygen concentration, the mixing time between the direct –injected fuel and the fresh oxygen increases. This is expected to increase the ignition delay and reduce the burn rate once diffusion combustion starts, in case all other parameters are kept constant." However when applying EGR the A/F ratio may decrease, "a decreasing A/F ratio has a tendency to shorten ignition delay" [3;T. Jacobs] (also see [57]).
When applying EGR; the gas temperature in the cylinder may increase in case of no, or insufficient cooling of the EGR-gas prior to recycling it back into the combustion chamber or by creating large exhaust back-pressure, increasing the internal residual fraction [3]. Increased gas temperature has a tendency to shorten ignition delay.
The increased temperature and reduced A/F ratio can offset the dilution effect on ignition delay [3]. In conclusion: depending on the extent in which EGR affects oxygen concentration, A/F ratio and gas temperature, the ignition delay may increase or decrease, or remain constant.

- *Combustion rate*

Due to the dilution effect caused by applying EGR, the combustion rates can decrease. According to [4; Ladommatos]: "The reduction in the local oxygen availability and the presence of $CO_2$ and $H_2O$ reduces the probability that the fuel and oxygen molecules meet and react, resulting in lower reaction rates". As a consequence, burn duration may increase [3]. However according to [59; Ladommatos] an increase in ignition delay caused by the use of EGR, may cause the combustion process to shift towards the expansion stroke resulting in "earlier quenching of the combustion process, that is, shorter combustion duration".

- *Incomplete combustion*

When applying EGR, the dilution effect as a consequence of lower oxygen concentration, increases the chances for incomplete combustion. A higher engine inlet temperature caused by hot EGR gas in case of no or insufficient EGR cooling will however decrease chances for incomplete combustion. Therefore the level of incomplete combustion may increase in effect of applying EGR, depending on the level of EGR and EGR cooling. [3, 4]

- *Performance*

Applying EGR may lead to a lower IMEP. According to [3; T, Jacobs] from tests in which EGR is applied on a HD diesel engine it comes forward that "both Gross IMEP and Net IMEP decrease with increasing EGR. The decreasing Gross IMEP indicates a decrease in combustion work. The decreased combustion work is the consequence of combustion degradation due to lower combustion temperatures and changes in A/F ratio".
According to [60; Pickett], "The rate of heat release also tends to decrease with decreasing ambient oxygen concentration".

Other emissions

- *Soot*

The use of EGR can have an influence on the emission of soot and particulate from a HD DI diesel engine. The "level of particles appearing in the exhaust depends on the competing processes of soot particle generation and growth on one hand and oxidation on the other" [4; Ladommatos]. In the oxidation process the availability of oxygen is of importance as well as local temperature. In relation to oxygen availability the A/F ratio is often mentioned to be of importance to soot and particulate emission of HD diesel engines. Also the way in which combustion takes place, premixed or diffuse burn, can have an effect on the particulate and soot emission. It is indicated by [4; Ladommatos] amongst others, that the premixed burn phase in a DI CI engine is relatively soot free compared to the diffusive burn phase. However J. Dec [58; Dec] points out that "all of the premixed combustion is fuel rich, in the range of an equivalence ration of 4", and

that "soot occurs throughout the jet cross-section, rather than only in a shell near the diffusion flame around the jet periphery".
In case of applying *additional-EGR* with a constant A/F ratio, compared to engine operation without EGR, this will cause an increase in soot emission compared to engine operation without EGR [6]. This is the result of the lower flame temperature caused by the EGR, which "reduces the oxidation rate of soot precursors and soot particles, resulting in greater soot formation" [4 Ladommatos]. The increased premixed burn duration that may be caused by the application of EGR through increased ignition delay, may possibly counter the increased soot formation.
In case of applying *replacement-EGR* with a decreasing A/F ratio, compared to engine operation without EGR, this will cause relative high soot emission, compared to engine operation without EGR and compared to engine operation with additional EGR [6]. An explanation can be found in the lower flame temperature caused by the EGR, in the same way as with applying additional-EGR. In addition to that, the oxygen availability decreases in case of replacement-EGR. According to [4; Ladommatos] "EGR reduces local oxygen availability, leading to lower oxidation rate of soot precursors and soot particles, resulting in greater soot formation". Here also the increased premixed burn duration that may be caused by the application of EGR through increased ignition delay, may possibly counter the increased soot formation.
In case EGR is applied without sufficient or no cooling of the EGR gas (hot EGR), this may cause an increase in inlet charge temperature. According to [4; Ladommatos]: "It is possible that the increase in the inlet charge temperature due to the use of EGR can result in greater soot formation due to an increase in the pyrolysis rate of the fuel molecules in the fuel-rich regions of the spray core. Because of little oxygen availability in these regions, soot particles may have a chance to grow and may ultimately escape oxidation." Also according to [4; Ladommatos]: "the use of EGR tends to raise the temperature of the inlet charge and tends to reduce the ignition delay period. This, in turn, reduces the amount of fuel burned during the relatively soot-free premixed-combustion period. More fuel is thus burned in diffusion-controlled combustion which is responsible for most of the soot production." This reasoning however does not correlate well with findings from Dec [58] who indicates that the premixed combustion phase is also fuel rich and soot is not only formed near the diffusive (fuel-rich) flames. Because of the complexity of this subject this will not be further assessed in this report. Both views from Ladommatos and Dec are considered.
In a diesel engine "the impact of EGR on soot at high engine load is particularly detrimental, firstly, because the engine is already working at low air fuel ratio and, secondly, because the EGR is poor in oxygen content" [4; Ladommatos]. Therefore "in practice EGR is either not used or is used in small quantities at higher engine loads" [4; Ladommatos].
- *HC*

The emission of unburned Hydrocarbons or HC's may be affected by the use of EGR. The dilution effect of EGR on diesel combustion, earlier discussed, increases the chances of incomplete combustion, causing HC emission. The chemical effect of EGR on diesel combustion leads to only a slight increase in unburned HC. The thermal effect of EGR on diesel combustion has virtually no effect on unburned HC emission [4]. In the same way as with soot emission the increase of inlet charge temperature, due to the use of un-cooled EGR, may have an influence on the HC emission. In general a higher inlet charge temperature in a DI CI diesel engine, will cause a decrease in HC emission EGR [4]. In case of cooled EGR, the dilution effect of EGR will prevail causing an increase in HC.

<u>Fuel consumption</u>
It is generally accepted that the use of EGR in a diesel engine will have a negative effect on the fuel consumption, due the decrease in brake thermal efficiency, compared to engine operation without EGR.
The decrease in brake thermal efficiency is partly due to combustion degradation caused by EGR. According to [3; Jacobs] combustion degradation is "due to lower combustion temperatures and changes in A/F ratio". This causes a reduction in indicated work compared to engine operation without EGR and equal amount of fuel injected.
Other causes for the decrease in brake thermal efficiency are due to increased pumping work, or losses, caused by the application of EGR. The increase in pumping work comes from extra piping

that feed the EGR gas back into the engine and measures to generate the desired EGR flow such as increasing exhaust backpressure. Also EGR cooling may lead to an increase in pumping work by increased flow resistance of the EGR-cooler(s). In addition more power will be required for driving the coolant-water pump and cooling fan(s).
The level of increase in pumping work depends, amongst others, on the level of EGR applied, the type of engine and the lay-out of the engine i.e. the way in which EGR is applied. Furthermore the brake thermal efficiency and the contribution of the two causes (increased pumping work and combustion degradation) too that may vary with different engine operations and settings. However according to [3; Jacobs], when applying EGR on a HD diesel engine; "negative side-effects of increased EGR on indicated efficiency appear closely related to the overall A/F level. In other words when the engine operates with overall low A/F ratios, relative changes of mixture composition due to EGR have much more impact."
Test performed by [3; Jacobs] on a HD CI engine with increasing EGR levels up to 25% for three engine speeds, keeping engine load constant at each speed, indicate that: "Gross IMEP remains nearly constant, indicating little degradation of combustion." "Net IMEP steadily decreases with increased EGR, indicating an increase in pumping losses". The brake thermal efficiency decreases with increasing EGR level at each engine speed and set load.
Also from tests performed by [3; Jacobs] on a HD diesel engine with EGR it is concluded that: "Combustion deterioration is the predominant reason for efficiency losses under low speed- mid load conditions where relatively low boost pressure levels might lead to critically low A/F values. For conditions characterized by higher overall A/F ratio, e.g. low speed- low load and mid speed (and high boost)- mid load, most of the fuel economy deterioration can be attributed to the increase in pumping work."

Heat rejection & cooling

The possible increase of inlet charge temperature as a consequence of applying EGR initiates thermal throttling and reduces the volumetric efficiency of the engine. Cooling the EGR-gas prior to recycling the EGR-gas back into the engine is often applied in order to restore the volumetric efficiency of the engine and in order to increase the effectiveness of the EGR in reducing NOx emissions [4]. EGR cooling will greatly increase the heat-rejection of the engine.
Applying EGR may increase or decrease the heat rejection from the combustion chamber. Due to lower combustion temperature as a consequence of EGR, the heat-rejection from the combustion chamber decreases. However in case of no, or insufficient EGR cooling, prior to recycling the EGR-gas back into the combustion chamber, the mixture temperature throughout the cycle may increase, causing an increase of heat rejection from the combustion chamber. This heat rejection from the combustion chamber may be in the form of convection to the combustion chamber.
According to [3; Jacobs] "the exhaust energy decreases with the increase of EGR, primarily as a result of lower exhaust flow rate, since a fraction of exhaust energy is re-circulated. This is often followed by a decrease in exhaust temperature". However it may be desirable to increase exhaust backpressure in order to control EGR flow and according to [3; Jacobs] "under certain conditions the exhaust back pressure may actually lead to higher exhaust temperatures".
According to [8; MTZ] heat rejection may increase by 100 kW for a typical 12 liter Heavy-Duty (HD) diesel engine with an average EGR level of 40% (required for euro5, according to [8]), in comparison to the current Euro 3 level. The total heat-rejection of a 12-liter HD diesel engine in Euro 5 setup would increase to 125 kW [8].

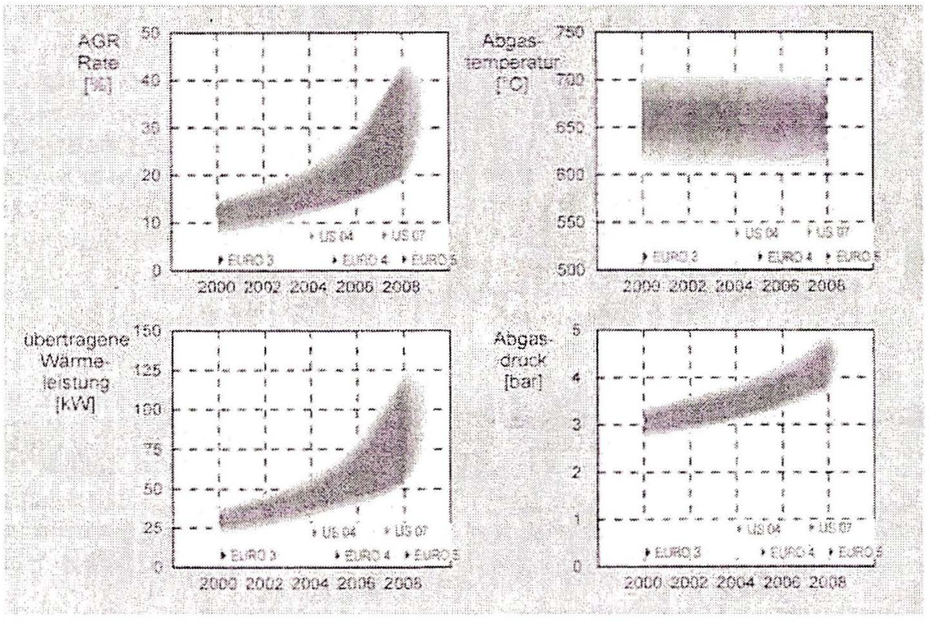



*figure 2.4 [8]; Future development of the boundary conditions EGR-rate, exhaust gas temperature, amount of heat to be dissipated and exhaust gas pressure.*

## 2.4 Considerations for further research concerning the effects of EGR on diesel combustion

For further research in the form of simulations several of the found effects of EGR on diesel combustion are considered for model construction and optimization. The most important considerations are given here.

High EGR levels can be very effective in lowering NOx emission. However AF ratios should preferable not suffer from applying EGR. Low AF ratios cause combustion degradation and thus low fuel economy. In general PM emission increase at decreasing AF ratio. In order to maintain AF ratios, additional EGR should be applied. A minimum AF ratio should be determined in order to prevent excessive loss of combustion efficiency.

In order to achieve high-EGR cooling of the EGR-gas is required. Cooling of the EGR-gas reduces the loss of volumetric efficiency and increases the effectiveness of the EGR-gas in reducing NOx emissions. Pumping losses due to EGR-cooling have to be taken into account.

The effects of EGR on ignition delay and the combustion rate require modifications to conventional (non-EGR) combustion models with simulations. In order to do this the increase in ignition delay and decrease in combustion rate at high levels of EGR should be further investigated. Furthermore the start of injection could be advanced in order to compensate for the increase in ignition delay.

Soot and particulate emissions most likely increase with the use of EGR, due to low oxygen availability and low temperatures. However a larger degree of premixed combustion may counter the increase in soot and particulate emission. There are however contradicting views on this subject (Ladommatos vs. Dec). Because of the complexity of this subject this will not be further assessed.

# 3. Engine modifications for applying EGR

In this chapter findings from literature are presented concerning the application of EGR on Internal Combustion diesel engines. It is aimed at giving a complete overview of the various systems possible for supplying EGR to an engine, plus additional modifications beneficial to engines with EGR.

## 3.1 General engine modifications

Engine (turbo) charging

As earlier described EGR can be applied either by replacement-EGR or by additional-EGR, keeping A/F ratio constant. It is stressed in most literature on this subject that the most beneficial way of utilizing EGR in DI diesel engines is the use of additional EGR, keeping A/F as high as possible [89, 90].
According to [7; Pfeifer] in order too achieve high NOx reduction in CI-DI-diesel engines: "the oxygen concentration must be as low as possible while the locally required in-cylinder oxygen mass required to completely combust the injected fuel must remain constant. Practically this can be achieved by high EGR rates to realize the low oxygen concentration and at the same time increasing the intake mass to keep the total oxygen/fuel ratio constant." Others [4, 6] stress that additional-EGR in DI diesel engines "allows exhaust NOx emissions to be reduced substantially with little penalty of increased particulate emissions", [4; Ladommatos], compared to replacement-EGR. Furthermore according to [3; Jacobs] when maintaining "high enough boost and hence A/F ratio, combustion deterioration with increased EGR is minimal" unlike applying EGR with lower A/F ratio's. According to [33; Morgan, Ricardo] "combustion efficiency deteriorates significantly at air fuel ratios below 23:1 resulting in a reduction in the rate of increase of power with increase fuelling".

Applying additional EGR with high A/F ratios, means increasing the total mass of in-cylinder charge. In order to do this the boost pressure needs to be increased until approximately 4 to 5 bar for a typical 12 Liter HD diesel engine in Euro 5 setup with 30% EGR applied at full load [8]. Increasing boost pressure with the use of EGR, will lead to high demands on the (turbo) charging system. Especially at low loads, where the energy content of the exhaust gas is low (low temperature), and in case EGR gas is branched off in-between engine and turbine resulting in less exhaust-gas being fed through the turbine, it proves to be difficult to produce high boost pressure (>4 bar) with a (single) turbocharger, as less work is delivered to the turbine while more work is demanded by the compressor. In most literature on the subject of increasing levels of EGR (15% and higher) in HD diesel engines, it can be found that the efficiency of the turbochargers needs to be high and the pressure ratio needs to be increased (>4). Furthermore the operating envelope of the turbo-charging system should be wide, enabling it to perform well at the entire engine envelope. It is indicated by [89] that the maximum EGR range is up to 22% for a single stage VTG (Variable Turbine Geometry) turbo, which is insufficient. A solution to these high turbo-charging demands may be found in the application of 2-stage turbo-charger systems, placing two turbo-chargers in series.

As both VTG turbochargers and two-stage turbochargers are capable of delivering relative high boost pressures and have wide operating envelopes, both types of turbocharging are shortly explained.

*VTG (Variable Turbine Geometry)*
In order to create a wide turbo operating field the compressor map should be fully utilized. Ideally this would mean a small turbine at low engine speed and a large one at high engine speed. A variable geometry of the turbine can fulfill this need. Variable turbine geometry is achieved by varying the area of a nozzle, a set of guide vanes that control the flow through the turbine. There are two types of VTG: the sliding wall and rotating vanes. With the sliding wall system the vanes slide axially varying the nozzle area. With the rotating vane system the vanes rotate in order open or close the vanes. By closing the vanes the turbine inlet pressure is increased, which increases turbine power and drives higher engine boost pressure.

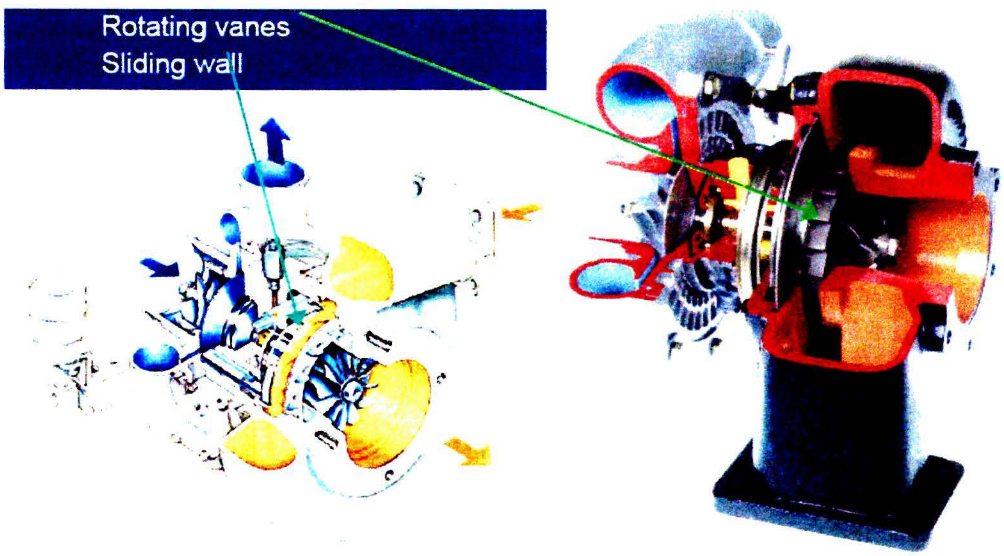



*figure 3.5* *VTG turbos, sliding wall and rotating vane system*

*Two stage turbocharging*
With two-stage turbocharging the turbochargers are connected in series i.e. one turbo boosts the other one. Relative high boost pressures are achievable, beyond those of conventional single stage turbochargers. Furthermore the operating envelope can be relative wide i.e. high boost pressure possible at relative low flow without surge problems.

For almost every application the HP turbocharger has a controlled turbine, either wastegate or VTG. Further possibilities are HP compressor bypass, interstage cooling (charge cooling in-between compressors) an LP turbine wastegate. With the HP turbo the overall boost level is controlled. With the LP turbine the ratio between HP & LP pressure ratio is determined.

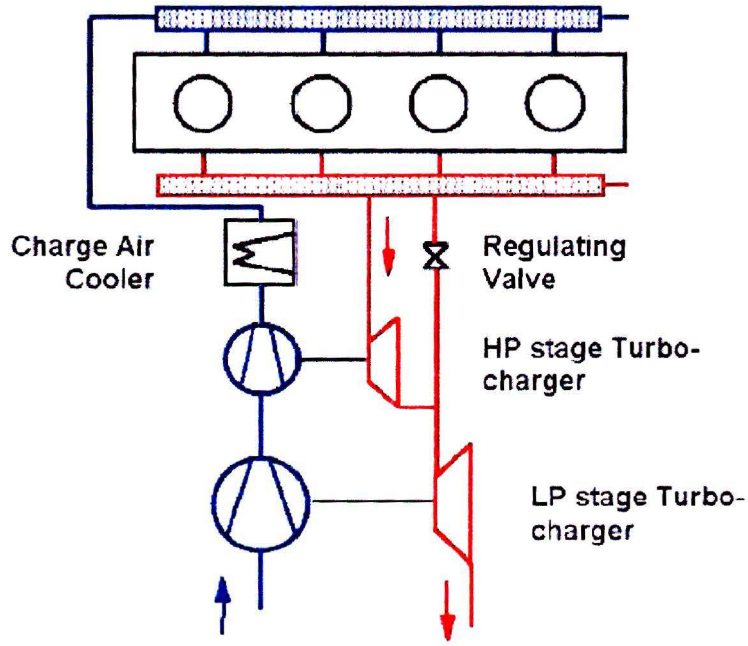



*figure 3.2 two-stage turbo lay-out for HD engines (without EGR)*

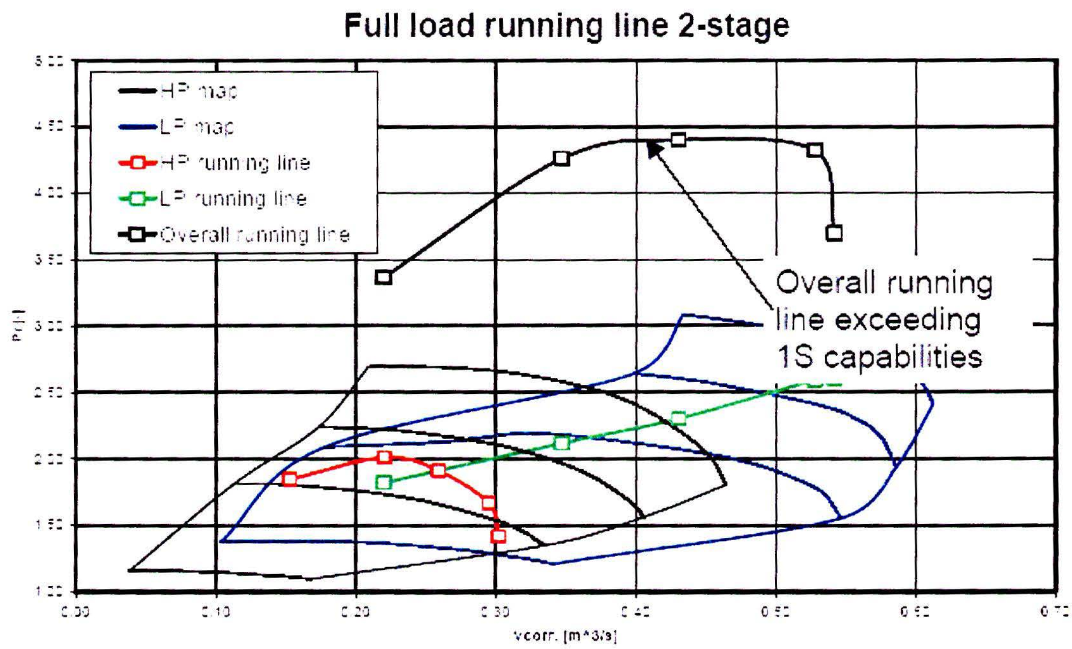



*figure 3.3 compressor map of two stage turbo-charging*

Cooling

EGR cooling is often required when applying high levels of EGR to HD diesel engines. The EGR cooling improves the effectiveness of the EGR-gas in reducing NOx emission and improves the volumetric efficiency of the engine opposed to applying un-cooled (hot) EGR. According to [4, Ladommatos]; "in practice, the increased engine volumetric efficiency gives considerable flexibility to the engine manufacturer for better exhaust NOx and smoke control". Also as the cooled EGR-gas is more effective than hot EGR, in some cases less EGR will be needed, reducing the need for high boost pressure, compared to applying hot EGR. Cooling the EGR gas may decrease total pumping power and hence pumping loss, increasing engine efficiency as less volume needs to be displaced. According to [15 Tomazic] "Particularly at higher exhaust recirculation rates, the fuel consumption can be improved by cooling the EGR". Furthermore EGR cooling can reduce the thermal load on the engine and its components.
According to [3, Jacobs]; "The easiest and most widely considered way to implement cost-effective EGR cooling is to use the engine coolant to reduce the exhaust gas temperature". EGR coolers are commercially available for HD diesel engine and are water-gas heat-exchangers to fit on HD diesel engines and make use of the engine coolant and pump. As the EGR gas may have a certain level of acidity the coolers need to be constructed out of corrosive resistant material such as stainless steel. Also, possible soot deposits in the EGR cooling-system need to be taken into account. According to [8] a controllable 'bypass' on the EGR cooler may be required to let the EGR-gas bypass the cooler at low engine load and speed, as is already applied on Low Duty diesel engines. At low engine load and speed the EGR-gas may otherwise be cooled beyond the desired level. Overcooled EGR-gas will cause the HC emission to increase significantly and promote condensation of acids from the EGR-gas.
As the total heat rejection of the engine increases, the capacity of the engines cooling system needs to be increased (in case EGR-cooling shares the engine cooling system), including a larger radiator with increased airflow, this will also increase pumping work or pumping losses due to cooling.

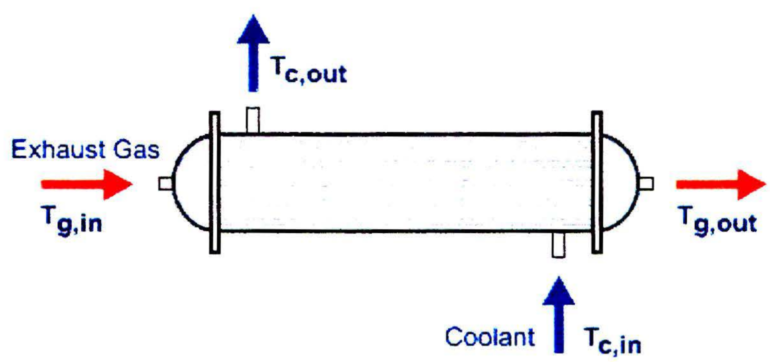



*figure 3.4* *Schematic diagram of EGR Cooler*

## 3.2 Classification of engine configurations for applying EGR

EGR can be re-circulated through to engine in several ways, meaning that the way in which an engine with EGR is configured can differ, due to differences in the EGR system. Difference can be made between 'external route EGR' and 'internal route EGR' configurations for Heavy-Duty diesel engines. For both the 'external route EGR' and the 'internal route EGR' further classification can be made with respect to turbo-charging solutions and optional features such as additional valves. An overview of the classification can be found in the figures 3.5 and 3.6 based on configurations found in literature. Also given in the figures 3.5 and 3.6 are the combinations of EGR configurations with (turbo) charging-systems and optional features, as found in literature. Here the definitions are briefly explained for figure 3.5 and 3.6. In paragraph 3.3 the several configurations are described in more detail.
In appendix 1.1 several "miscellaneous" EGR configurations are given. These configurations are mentioned in literature without further explanation or reference, and are considered to be unrealistic or too impractically and will not be further assessed.

*Figure 3.5*
- External route; The EGR gas is lead from the exhaust system of the engine via an EGR-line, which is external of the engine block, to the inlet system of the engine. External EGR can be sub-divided into *'internally driven*' and *'externally driven'*.

- Internally driven; the EGR-gas flows trough the EGR- line under the influence of differential pressure obtained out of the pressure differential over the engine. Meaning the pressure difference between inlet and outlet manifold. Internally driven EGR can be divided into '*low-pressure*' and '*high-pressure'*.
    - Low-pressure; the EGR-line starts downstream of the turbocharger- turbine and ends upstream of the turbocharger-compressor.
    - High-pressure; the EGR-line starts upstream of the turbocharger- turbine and ends at the engine inlet manifold.

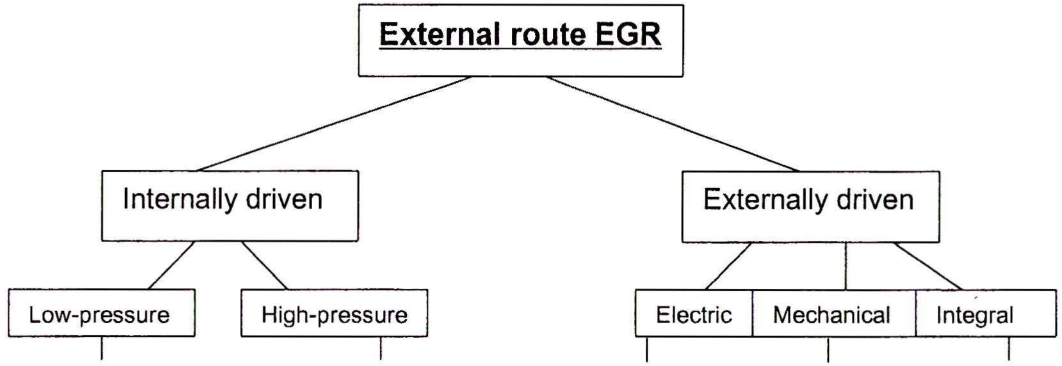
- Externally driven; the EGR-gas is pumped through the EGR-line by means of an auxiliary pump, which is called EGR-pump. Externally driven EGR is also referred to a 'Pumped' or 'Forced' EGR. The EGR pump can be driven;
    - Electric; by electric motor
    - Mechanical; directly by the IC engine
    - Integral; by an exhaust turbine or turbo-charger (in essence a form or turbo-compounding)

  Note: Hydraulic driven EGR-pumps are not found in literature, but are in principle possible.

Further classification in figure 3.5 concerns turbo-charging solutions and optional features.

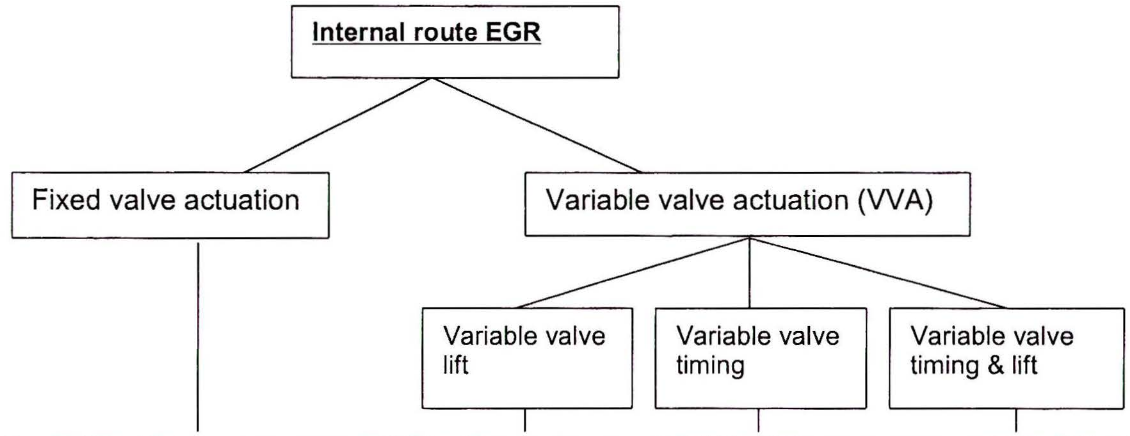
*Figure 3.6*
- Internal route; the EGR-gas is either kept in the cylinder or is driven back into the cylinder from the exhaust manifold through the exhaust ports. This accomplished by means of;
    - Fixed valve actuation; fixed valve timing and lift
    - Variable valve actuation (VVA);
        - Variable valve lift
        - Variable valve timing
        - Variable valve lift & timing

In the following chapters each of these EGR configurations will be discussed. In many cases, EGR can benefit from other techniques aimed at lowering exhaust-gas emissions, and the other way around as well.




| Charging Systems | | Low-pressure | High-pressure | Electric | Mechanical | Integral |
|---|---|---|---|---|---|---|
| Single Stage | Fixed geometry turbo | X | X | X | X | X |
| | VTG | X | X | X | X | X |
| Two Stage | 2× fixed geometry turbo | | X | X | X | |
| | VTG + fixed geometry | | X | X | X | |
| | VTG + roots | | | X | X | |
| Compound | Turbo compound | | X | | | |
| **Options** | Hot/ Cold EGR valve | X | X | X | | |
| | Backpressure control throttle valve | X | | | | |
| | PM filter | X | | | | |
| | Reed valves | | X | | | |
| | Venturi EGR mixer | | X | | | |
| | Single EGR cooler/ Double EGR coolers | X | X | X | | |
| | EGR-cooler bypass | | X | | | |
| | Air intake throttle | | X | | | |

*figure 3.5; External route EGR-systems overview.*
***X**: cases found in literature*




| | | | | | |
|---|---|---|---|---|---|
| Light duty | | X | X | X | X |
| Heavy Duty + EGR | Jacobs; Jake Brake | | | | X |
| | Caterpillar; ACERT | | | | (X) |
| | Mack; ACET | | | | (X) |

*figure 3.6; Internal route EGR-systems overview as found in literature*

## 3.3 Explanation of EGR configurations and findings from literature

In this paragraph several EGR configurations are described as found in literature. The EGR-configurations are;

- External route EGR
  - Internally driven-high-pressure
  - Internally driven -low-pressure
  - Externally driven
- Internal route EGR

All described configurations are as found in literature, however often many variations are possible.

### 3.1.1 External route EGR

#### *3.1.1.1 External route; Internally driven- high-pressure*

External routed, internally driven high –pressure EGR, or short-route-EGR, is the more conventional method of applying EGR, by means of removing exhaust under high-pressure and temperature upstream of the turbine and reintroducing it into the intake system downstream the compressor (Figure 3.7). External route internally driven high-pressure EGR, may be abbreviated to 'High-pressure' in the rest of this thesis.

- *lay out high-pressure;*

Figure 3.7 [13] shows a layout of a typical high-pressure-EGR, with an EGR cooler, EGR-valve and turbo compressor. EGR-gas flows through the EGR-line which starts at the exhaust manifold of the engine upstream of the turbine and stops at the entrance of the intake manifold or air collector. In the layout presented here, an EGR-valve (AGR-ventil) and an EGR cooling element (Abgaskühler) are taken up in the EGR-line.

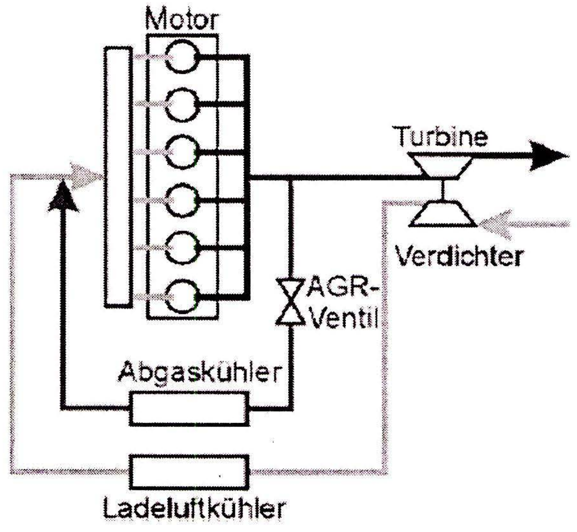



*figure 3.7 [13] ; Lay-out of a high-pressure EGR system*

Although the basic layout of a high-pressure EGR system is the same in all cases, the layout may vary in detail. Several different measures are possible in order to generate a negative pressure difference between intake- and exhaust system and thus over the EGR-line, in order to stimulate EGR-flow. The control requirements of a high-pressure EGR system also have an effect on the layout, for example implementation of additional valves. Furthermore layout variations may depend on the desired levels of EGR or to what extend it is aimed to keep the A/F ratio constant.
Layout differences are possible in the charging system, the cooling system and the control system including means to achieve negative pressure differential between the intake- and exhaust system.

*Charging*
Many high-pressure EGR systems as recently described in literature make use of VTG (variable turbo geometry) turbo charges, which are commercially available [11, 12, 15, 17, 18, 89, 90, 92]. Besides better performance of a VTG opposed to a fixed geometry (single) turbocharger, the VTG is used in high-pressure EGR systems for its ability to control the EGR flow by adjusting exhaust-back pressure. VTG turbochargers are currently commercially available for Heavy-Duty diesel engines.
Because of the limited maximum pressure- ratio achievable with a single VTG, particularly at low engine speed (exhaust mass flow) it is most likely unable to achieve the high boost pressure at low engine speed desired in some of the more advanced High-pressure EGR systems designed for the HD EURO 5 emission standard [7, 8, 11, 19, 22].
In most literature it is therefore recommended that 2-stage turbocharging should be used, in order to achieve relative high levels of EGR (>25 %) [7] and keeping the AF ratio as high as possible (>25) [11] over the entire engine-envelope [7, 11, 13, 15, 20, 22]. A 2-stage Turbo-charging-system is capable of reaching the required high boost pressure at low engine speed and still is capable of maintaining high performance over a wide envelope. The disadvantage of a 2-stage turbo-charging system is its complexity and size. Currently 2-stage turbo-charging systems are only just becoming commercially available for Heavy-Duty (on high-way) purposes [22, 91].
An alternative, charging system beneficial to high levels of EGR makes use of a mechanical compressor (roots), possible in combination with a turbo charger. A mechanical compressor can provide higher boost pressure and better transient response at low engine speed opposed to more conventional turbo-charging systems [11]. Another advantage of a mechanical compressor over more conventionally turbochargers is it relative simplicity. The mean disadvantage is its relative low efficiency leading to relative high BSFC (Brake Specific Fuel Consumption) of the engine [11].
A so-called 'High-pressure Ratio, Wide Flow Range Compressor' [24] is also a charging system that can provide high-boost pressures over a wide engine envelope. The charging system consists of a turbo-charger with a two-stage compressor-wheel, which enables two-stage compression packaged in a single rotating assembly [24]. As a consequence of the two-stage compression the total compressor-map is relatively large; due to higher pressure ratio a larger surge margin is obtained. Furthermore the two-stage compression enables relative low compressor tip-speeds at high pressure ratios and flow rates, thus provoking long compressor life. Although little (un-biased) information is available on this type of charging system, disadvantages are likely to be found in the complexity of the two-stage compressor wheel and housing.
Turbo-compounding can be used in combination with EGR [9; Scania]. According to [9; Scania] high engine efficiency can be reached with turbo-compounding. However turbo-compounding will not improve turbo performance by increasing pressure ratios as is required for High EGR applications. Turbo compounding could possibly be helpful in controlling EGR as the compound-turbine provides exhaust-backpressure control. In general turbo-compounding can be regarded as complex and expensive.

*Cooling*
It is clear from literature that for future EGR systems, cooling the EGR is a necessity for 'high EGR' applications. In some cases two EGR coolers are applied which are set in parallel [14], however in most cases a single- unit EGR cooler is used, for packaging reasons [11, 12, 13]. It may be desirable to let the EGR-gas bypass the EGR-cooler in some conditions to avoid too low charge temperature, for example at cold start. A too low charge temperature increases HC and CO emissions and increases condensation of exhaust species from the EGR-gas. Therefore a controlled-bypass valve may be implemented in the EGR-cooler design [13, 19, 20]. Depending on the design, an EGR-bypass may increase the volume of the EGR-cooling device since the un-cooled EGR-gas needs to be fed through a separate channel [8, 19, 20]. Currently EGR-coolers for Heavy-duty application are commercially available as a single-unit to fulfill the EGR cooling requirements up to EURO 5 regulations. In case of two-stage turbo-charging "the addition of an extra intercooler between the low-pressure and high-pressure compressors will reduce compressor inlet temperature and improve efficiency" [11].

*Control*
Controlling EGR flow is required in order to obtain the target EGR level at the right moment, meeting emission regulations with minimum penalties in performance and fuel consumption. Controlling EGR means limiting EGR-flow in some conditions and stimulating EGR-flow in other conditions. It is typical with Heavy-duty diesel engines that a positive pressure differential between intake and exhaust system pressure may exist, i.e. the boost pressure is higher than the exhaust back-pressure. This may prove "recirculation of exhaust into the intake system without additional modifications difficult, if not impossible" in case of the external route, internally driven High-pressure EGR system [15; Tomazic, FEV]. In order to increase EGR flow control-devices and strategies, capable of increasing the negative pressure differential of the EGR-line are required.

In order to be able to limit the EGR-flow, it is common to apply an EGR-valve(s) in the EGR-line. Limiting the amount of EGR may be required in case of transient engine behavior or high load conditions. In many cases an EGR-control-valve is implemented in the EGR-line in front of (upstream) the EGR cooler (hot EGR valve) [12, 13, 15, 16], or after (downstream) the EGR cooler (cold EGR-valve) [11, 14, 17]. The disadvantage of the EGR-valve(s) taken up into the EGR line is the pumping loss over the valve, especially as the pressure differential over the EGR line already needs to be increased artificially in some (other) conditions [13]. It is not always made clear in literature what the advantages or disadvantages of mounting the EGR valve upstream or downstream of the EGR cooler are. Clearly, mounting the EGR valve upstream of the EGR cooler (hot EGR valve) will cause a higher thermal load on the valve compared to mounting it downstream of the EGR cooler. Fouling however is less compared to downstream mounting. Mounting the EGR-valve downstream of the EGR cooler will increase the dead volume between engine exhaust-manifold and the turbine, since part of the EGR line and EGR cooler are in contact with the exhaust manifold. This may have a negative effect on the turbo performance at transient engine behavior. According to [13] placing the EGR-valve upstream of the EGR cooler is therefore preferable. However mounting the EGR-valve downstream of the EGR-cooler could reduce the required EGR-valve size or reduce pressure loss over the valve, as the density of the gas is increased.
The second requirement in the control of EGR is to achieve a negative pressure differential between intake- and exhaust system, as already indicated. This can be done in several ways. In most cases the ability of the VGT to control back-pressure is utilized for this purpose [11, 12, 15, 17, 18]. By controlling the vanes of the VGT the exhaust back-pressure can be increased to create the required negative pressure differential over the intake- and exhaust manifold and thus over the EGR-line. Using the VTG to increase back-pressure is an effective method in delivering sufficient EGR to the engine, but may lead to a decrease in engine efficiency through an increase in pumping work. Also engine transient behavior may suffer from this method of increasing back-pressure [3, 11, 14]. It is also possible to lower the pressure downstream of the EGR-line, by applying a venturi type of mixer to add the EGR-gas to the fresh-air stream to the engine [13, 15, 19, 21]. In addition a venturi-mixer mounted on the inlet manifold enables good mixing of the

EGR-gas with the fresh-air charge. Disadvantage of a venturi-mixer is the possible pressure loss over the mixer in the intake system. [13, 21]
Another way of lowering the pressure downstream of the EGR-line is the use of a throttle valve on the air-intake of the engine (prior to EGR-injection) lowering the pressure in the air intake of the engine and as such increasing negative pressure differential over the EGR-line [15]. However throttling the fresh air intake of the engine will cause the A/F ratio of the engine to decrease with a negative effect on soot emission. Therefore this method, of ensuring sufficient EGR flow, is rejected in most cases found in literature (except for experimental purposes) [3, 4, 7].
Non-return valves, better known as 'Reed-valves' can be applied in order to provoke EGR-flow through the EGR-line. The reed valve(s) should be mounted in the high-pressure EGR line (s) connecting the exhaust-system (upstream of the turbine) and the intake system (downstream the compressor). Reed-valves make use of the consecutive pressure peaks in the exhaust manifold generated by the individual cylinders during the blow-down process and exhaust stroke. In principal the reed valve opens as soon as the dynamic exhaust pressure exceeds the intake pressure. The closure occurs subsequently when the exhaust pulse drops below the intake system pressure. Typically the self-controlled valves are installed in each of the two streams of the twin-flow turbine to match the flow requirements of the re-circulated exhaust [15]. The reed valves although simple in design, may prove to deliver insufficient EGR. In order to deliver sufficient EGR, the VTG should be closed so that the backpressure is increased allowing the exhaust pressure pulses to overcome the opening pressure of the valve. This increases the engine work during the exhaust phase leading to low overall engine efficiency [11].

- *Achievements*

Based on engine tests and simulations with realistic engine models, some relevant achievements of the high-pressure external EGR system for Heavy-Duty diesel engines will be given here as found in literature. However as each test or simulation is conducted under different circumstances and with different setting it will be difficult to compare the achievements. Please note, only tests of high-pressure EGR without any additional technique such as exhaust-gas after-treatment and advanced fuel injection strategies are assessed in this chapter.

In literature source [11; Iveco] simulations where conducted with a realistic engine model of a baseline Heavy-Duty diesel engine (12,9L, 400kW). Amongst the simulations are simulations with a high-pressure EGR configuration. The lay-out consists of a single EGR line with a single EGR-cooler, a non-return valve (reed valve), and a controlled EGR valve downstream of the EGR-cooler. Furthermore a VTG turbo-charger and intercooler are applied.
From the simulations it can be concluded that; "It is possible to achieve EURO 4 (NOx) limits while maintaining air excess ratio, but EURO 5 (NOx) is not possible with the current single stage turbocharger". Meaning that NOx levels are at <3.50 (g/kWh) and <2.00 (g/kWh) respectively. It's not clear whether the Particulate Matter emission also complies with EURO 4&5 limits. When a mechanical charger is used in series with a single stage turbocharger it is possible to achieve the desired air excess ratio (λ: 1.7) and EGR (±20%) for EURO 5 but at the expense of much higher BSFC (>20%) and much higher heat rejection up to a total of 375 kW which is about 140 kW more than with a equivalent EURO 3 engine. "A two stage turbo-charging solution can achieve the desired EGR rates and air excess ratios without any increase in BSFC over current EURO 3 engines." "If EGR is to be used to achieve EURO 5 NOx levels higher peak cylinder pressures, up to 200 bar, must be expected. The quantity of rejected heat that must be handled will increase by up to 50%"

From literature source [8; MTZ] it may be concluded that with high levels of EGR (40%) EURO 5 emission standards are attainable with a Heavy Duty 12 liter engine, although no test or simulations are performed. It is indicated that in achieving EURO 5 by means of high-pressure EGR, two-stage turbo charging at 4 to 5 bar boost pressure is required. Up to 120 kW of heat should be cooled from the EGR gas by means of a single unit EGR cooler, preferable with a controlled bypass.

From literature source [14; Mitsubishi FUSO] it can be concluded that during transient engine operation, with the application of high-pressure EGR at 23% EGR on a Heavy duty engine, the PM emission can be reduced by 40% when the EGR is partially or completely cut for 1 to 5 seconds, depending on the transient. Cutting EGR during transients will thus prevent smoke spikes and improve drivability [92]. According to [21; TNO] however, it is indicated that at relative low transients, EGR should be maintained in order to maintain low NOx emission.
According to [7; FEV] a combustion system aiming at complying US 2010 regulations will include: cooled EGR with recirculation rates of up to 50%, high-pressure boosting systems (most reasonably two stage turbo charging with dual stage inter-cooling), a flexible injection system with pressure levels of 2.500 bar and beyond and the capability of multiple split main and post injections. Consequently the engine architecture should change since peak in-cylinder pressures will be 200 to 230 bar.
In literature source [12; SwRI] NOx levels of 2.1 g/kW-hr are achieved over a FTP cycle with a high-pressure EGR system on a HD diesel engine (10-15 L). Giving a reduction of 20% compared to the baseline engine. "Obtaining sufficient EGR levels at full load and part load will require the increase of boost pressures in order to maintain current power densities". High-pressure ratio VTG turbochargers can be used for this purpose. Peak cylinder pressures will consequently reach 180- 200 bar. Therefore special attention should be given to the design of engine- block and drive-train.

From the tests and simulation results it can be concluded that EURO 5 engine out emission is achievable with a high-pressure EGR system without any additional techniques such as exhaust-gas after treatment or advanced fuel injection. It will be necessary to make use of two-stage turbo charging to achieve high A/F ratios in the region of 1.7 till 2 even at low engine speeds. EGR should be up to 30 till 40%. Heat rejection from the EGR-cooling will increase to approximately 120 kW. When using high boost pressure the peak cylinder pressure will increase up too 200 bar. In order to comply with US 2010 peak cylinder pressure may become as high as 230 bar. Also the fuel injection system will have to be more advanced, particular the injection pressure should be raised. More on advanced fuel injection systems can be found in paragraph 3.4.

3.1.1.2 External route; Internally driven- Low-pressure EGR

External route, internally driven Low-pressure EGR or long route EGR, operates by removing exhaust under relative low-pressure and temperature downstream of the turbine and reintroducing it into the intake system upstream of the compressor. External route, internally driven Low-pressure EGR, may be abbreviated to 'Low-pressure' in the rest of this thesis.

- *Lay out*

Figure 3.8 shows a layout of a low-pressure-EGR system, with an EGR cooler, EGR-valve and turbo compressor. The EGR-gas flows through the EGR-line, which starts at the exhaust system, downstream of the turbine and ends at the intake-system, upstream of the compressor. In the layout of figure 3.8, an EGR-valve and EGR-cooler are taken up in the EGR-line.

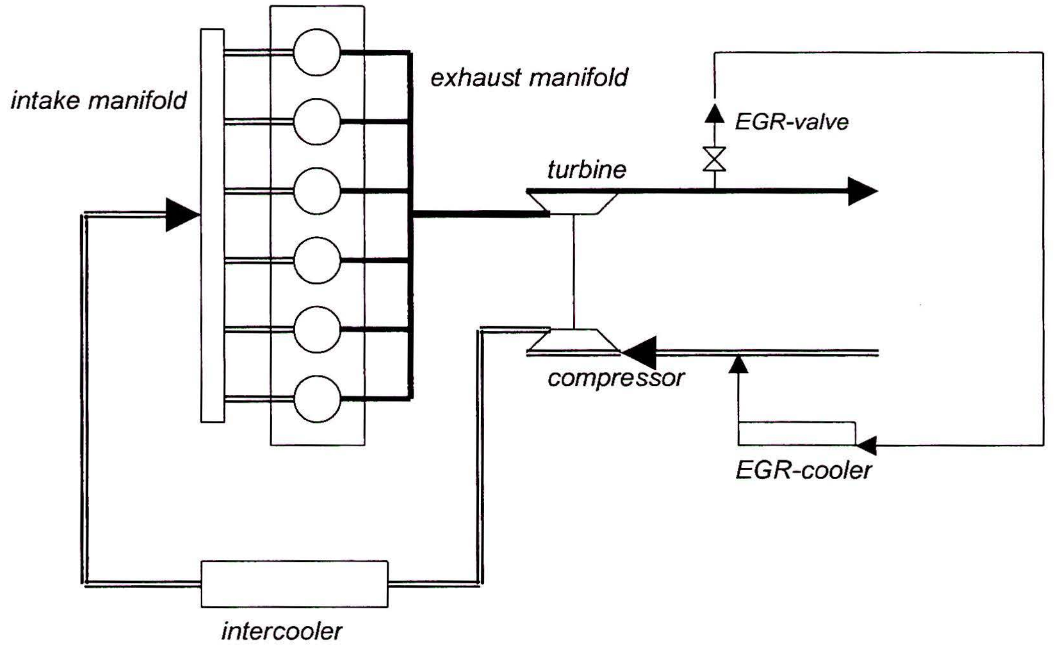



*figure 3.8; Lay-out of a external route, internally driven, low-pressure EGR system*

The basic layout of a low-pressure EGR system is the same in all cases. However there may be differences. In contrast to high-pressure EGR, most configurations of low-pressure EGR as presented in literature have little differences. Some features of the low-pressure EGR system will be further discussed

*Charging*

The charging system of the low-pressure EGR system in all of the in literature observed versions consists of a VTG turbocharger and charge cooler [11, 15, 17].

In contradiction to the High-pressure configuration the turbine is fed by the entire exhaust-gas mass flow, leading the turbine to operate in "a better efficiency zone" [19]. As such according to [29, DieselNet] the Low-pressure configuration has the potential of "lower fuel consumption" opposed to its High-pressure counter-part.

Since EGR- gas is taken off relative far downstream in the exhaust-system relative long piping is required for the EGR-line, which increases packaging efforts. Because of the relative long piping and thus relative large volume of the EGR-system, the dynamic response of EGR-flow during transient operating conditions may be poor and "harder to manage" [15, 19].

As the EGR-gas is fed through the compressor after injection into the intake system, this may create problems due to the EGR-gas composition. The PM in the EGR-gas may cause fouling of compressor and intercooler. Acid component in the EGR-gas may cause damage (corrosion) to compressor and intercooler [3,11, 17].
As earlier indicated the AF ratio should preferable be maintained when applying EGR, meaning that in case of low-pressure EGR besides the 'normal' fresh-air charge, also relative hot EGR-gas needs to be compressed by the compressor. As a consequence the turbo flow, both compressor and turbine, is relatively high. Also a high-pressure ratio is required. This stresses the need for a relative large size turbo, which may result in large turbo inertia.
Turbo durability is likely to be a problem as a consequence of EGR injection and relative high performance requirements. "Special coating and/ or materials may be required" [17; Edwards, Ricardo] in the turbo design, in order to withstand damage caused by aggressive EGR-components, relative high temperatures and load.
Even though the EGR is cooled, it is at a higher temperature than the inlet air. This increases the compressor entrance temperature, which increases the work the compressor must do per kg mass of air delivered. Therefore the low-pressure EGR system may require a significant increase in the compressor-operating field especially at higher engine speeds [11].

*Cooling*
In all of the in literature observed Low-pressure EGR engine-configurations, a single but large unit EGR-cooler is adapted for cooling of the EGR gas [11, 15, 17]. As already indicated the cooled-EGR temperature is of influence on the compressor entrance temperature and as such on turbo-performance. EGR-cooling at high EGR levels, may prove critical for turbo-performance. However too low EGR temperature increases the possibility of condensation of the EGR-gas.
In a low-pressure EGR-system both the EGR-cooler and the 'normal' intercooler cool the EGR-gas, creating a relative cool charge after the compressor, increasing the EGR's effectiveness in reducing NOx [11]. At High EGR levels and relative high AF levels the total charge flow through the intercooler (fresh air+ EGR) is relative high as well. This may require a relative large intercooler opposed to non-EGR or external-route internally driven High-pressure EGR configurations. In the same way as with the turbo-compressor the intercooler may suffer contamination and chemical aggression from EGR-components being fed through.

*Control*
Control over the EGR-flow is obtained by operating the EGR valve. In the same way as with high-pressure EGR, the EGR valve may be placed upstream of the EGR-cooler (hot EGR valve) [15, 17] or down-stream of the EGR-cooler (cold EGR valve)[11]. A throttle valve can be used in the exhaust system, downstream of the EGR take off, in order to increase backpressure when necessary to attain sufficient EGR flow [11].
In comparison to the high-pressure variant of external EGR, a low-pressure EGR configuration is easy to operate [15]. The requirede negative pressure differential will more naturally exist between the exhaust system downstream of the turbine and the intake system upstream of the compressor. This reduces the need for artificially raising the pressure differential over the EGR-line, by means of excessive exhaust-backpressure [3, 19]. The VTG rack-position is not likely to have a large effect on the EGR flow and level. The VTG (when used) can be operated independently of the EGR-system (EGR-valve) [15].

*Filtering*
As already indicated contamination of the compressor and intercooler by the soot contained in the EGR-gas may cause problems. Therefore, it is indicated in some literature source [12, 15, 19, 23] that a DPF (Diesel Particulate Filter) should be taken up in the exhaust system, upstream of the EGR take off, after the turbine.

- *Achievements*

In literature source [11; Iveco] a low-pressure EGR system on a Heavy Duty diesel engine is simulated with a realistic engine model. It is concluded that EURO 4 and EURO 5 NOx levels are achievable by means of a low-pressure EGR system on a HD diesel engine. EURO 4 NOx level could be achieved without increasing BSFC over the current EURO 3 level. To achieve EURO 4 NOx levels (3,50 g/kW-hr) ±8% EGR is required at an AF ratio of 25 (λ: 1.7), peak cylinder pressures are increased to over 170 bar. The total heat rejection is increased by approximately 15% over the EURO 3 level.
EURO 5 NOx levels are achievable, however at lower AF ratios: 20.3 to 21.8 (λ: 1.4-1.5). Approximately 18% EGR is required. It is indicated that for a low-pressure EGR system a compressor with a wide operating field is required. Heat rejection increases drastically too approximately 65% over the EURO 3 level to a maximum of 435 kW.
Unfortunately it is not clear from this source, what levels of PM are accompanying the low NOx levels (EURO 4 and 5).
According to [12; SwRI] 2.0 g/kW-hr NOx and 0.01 g/kW-hr PM is achievable over the heavy-duty FTP cycle on a 12L engine (conventional combustion). These results could comply with EURO 5 emission regulations. However use is made of a DPF in the exhaust prior to extracting EGR.

From the above mentioned achievements it can be concluded that EURO 5 engine out NOx emissions are achievable with a low-pressure system, without further additional techniques, such as after treatment. In the same way as with high-pressure EGR, boost-pressures are to increase in order to maintain a minimum A/F ratio (1.4- 1.5). As a consequence peak cylinder pressures also increase to well over 170 bars (the same as with High-pressure EGR). In order to obtain high boost pressures the compressor should have a wide operating field. Although not clearly indicated in the papers, it is likely that two-stage turbo-charging systems are required in order to reach the high boost-pressures, in the same way as with high-pressure EGR.
In case DPF filters are taken up in the low-pressure EGR system the levels of PM can be brought to acceptable levels, EURO 5 and lower. Furthermore it seems that the same levels of NOx reductions are achievable with low-pressure EGR as with high-pressure EGR, however at slightly lower EGR levels (-2%). This may be due to lower intake charge temperatures, however this is not confirmed in the papers. Depending on the level of EGR cooling the total Heat rejection of the engine will increase. As earlier indicated the level of EGR cooling also has an influence on the compressor work needed, a trade off may exist between the rejected heat from EGR cooling and compressor work.

### 3.1.1.3 External route; Externally driven

The external route, externally driven EGR system work in a similar way as the external route, internally driven, High-pressure EGR-system; the exhaust gas is removed upstream of the turbine and reintroduced into the intake system downstream of the compressor. However in contrast to High-pressure EGR, with externally driven EGR an EGR-pump is taken up in the EGR line (figure 3.9). External route, externally driven EGR, may be abbreviated to 'externally driven' in the rest of this thesis.

- Lay out

Figure 3.9 shows a lay-out of a external route, externally driven-EGR system on a diesel engine. In the lay-out an EGR-cooler, EGR-valve, turbocharger and intercooler, and the EGR-pump can be seen as a sketch. The EGR-line starts at the exhaust-manifold, upstream of the turbine and ends at the inlet-manifold, downstream of the compressor. In the lay-out presented here a EGR-valve, EGR-cooler and EGR-pump are taken up in the EGR-line.

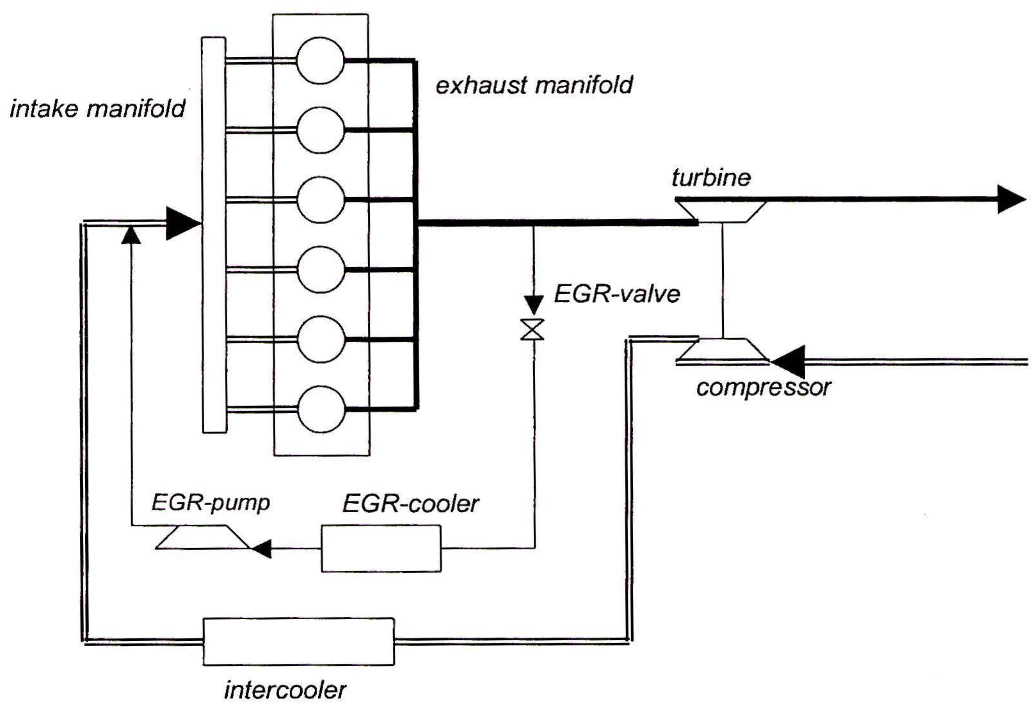



*Figure 3.9; Lay-out of a external route, externally driven EGR-system*

Differences in lay-out may be possible, several lay-out differences and features as found in literature will be discussed here.

*EGR-pump*

The main function of the EGR-pump is to pump exhaust gas from the exhaust system into the intake system of the engine regardless of the existing pressure differential between intake- and exhaust-system. The EGR-pump thus overcomes the problem of creating a negative pressure differential over the EGR-line as exists with high-pressure EGR-systems. As a result the EGR-

pump enables EGR flow without the need for increasing exhaust backpressure, as is common with the external route internally driven high-pressure EGR-systems, "mitigating the negative impact on fuel economy experienced by back-pressure based systems" [24].
Several variants of EGR-pumps and drives are possible. The EGR-pump may be mechanically or electrically driven and may be a compressor or blower (roots) [15]. Also an EGR-pump integrated in a conventional turbocharger or VTG, also referred to as 'integral EGR pump' is developed [15, 24]. The integral EGR-pump has the advantage that it is integrated in the housing of the turbocharger, as such it poses less of a packaging problem. It is indicated that, as the integral EGR-pump is driven by the turbocharger, it will lead to better overall fuel efficiency than back-pressure based, high-pressure EGR, systems [24]. It is not clear however in what way the turbocharger- or VTG-performance is influenced by the integration of the EGR-pump, but it can be suspected that transient behavior may suffer.
The disadvantages of the external route, externally driven EGR system are its suspected overall low efficiency due to the additional power required to drive the EGR-pump [11]. Furthermore the durability of an EGR-pump is questioned, as it will suffer form relative high temperature gas, together with soot and acid as part of the gas composition. Another issue is the packaging of the pump and its driveline, which may be big and difficult to implement on a Heavy-duty diesel engine [3, 15, 17].

*Control*
As earlier indicated the EGR-valve is used to control the EGR-flow rate. The position of the EGR-valve may be upstream or downstream of the EGR-cooler, in the same way as this is possible with high-pressure EGR systems. In case of a controllable EGR-pump such as an electric driven EGR-pump, the EGR-flow can be controlled by both the EGR-valve and the EGR-pump. The fresh-air flow is controlled by the VTG or conventional turbocharger, such as a waste-gate turbo. The control of EGR-flow and fresh-air flow is independently. [24] This independent control of EGR and fresh-air flow (AF ratio) can be beneficial in reducing the effect of EGR on particulate emissions during transient engine operation [24]. However it is believed that the separate control of the EGR-pump may prove to be complicated in practice [3].

*Cooling*
A single unit EGR-cooler is used in all cases [11, 24]. The EGR-cooler is placed in the EGR-line, upstream of the EGR-pump mitigating the thermal burden on the EGR-pump. Too low EGR temperature however increases the possibility of condensation of EGR-gas.

*Charging*
Many charging solutions, which can be adapted for the High-pressure EGR configuration, can also be used for the external route externally driven configuration.
A VTG turbocharger (§3.1) is used in most externally driven EGR-configurations. In case of an integral EGR-pump the EGR-pump may be integrated in a 'conventional' turbocharger (waste-gate) or in a VTG. A VTG gives better control over the fresh-air charge [3, 11, 24]. An integral EGR-pump needs to be designed together with the turbocharger or VTG and cannot be regarded as separate units.
In order to achieve high boost pressures over a wide engine envelope, for the purpose of maintaining a high A/F ratio, it is possible to make use of a mechanical compressor possibly in combination with a turbo charger. The mechanical driven charger is capable of providing high boost pressures and thus A/F ratios, even under high load and low engine speed, giving good transient response, however at the expense of higher BSFC. An advantage of the mechanical compressor is its relative simplicity. Another charging system that provides high boost pressures over a wide engine envelope that may be used in an external route EGR configuration is two-stage turbo-charging (also see §3.1). The same advantages and disadvantages apply as in use with the high-pressure EGR configuration. The two-stage turbo-system is believed to give very good performance at a wide engine envelope, but the system is complex and just becoming commercially available [11]. A so-called 'High-pressure ratio, wide flow range compressor' is also earlier discussed with the use in High-pressure EGR systems; the same characteristics apply in use with an externally driven-EGR configuration.

- *Achievements*

Unfortunately not many test- or simulation- results are available of externally driven-EGR systems.
It is indicated by [24; Garret] that in using an integral EGR-pump, EGR-rates of 12-16% are possible while maintaining almost the same pressure differential over intake and exhaust manifold, as without the use of EGR. Higher levels of EGR are feasible depending on the compressor design of the integral EGR-pump. Meaning that although the pressure in the intake-manifold is higher than the pressure in the exhaust-manifold, high levels of EGR are made possible by the EGR-pump. It is claimed that [24; Garret]; "only a very small loss in the positive pumping power of the engine is experienced while driving high percentages of EGR". However measurement data presented in the same paper show a maximum achieved EGR percentage of 15.7%, which is considered medium level EGR in the context of this report. It is uncertain if the claims made by Garrett are also valid for EGR levels in the range of 30- 70% i.e. high-EGR.

From [11, Iveco] it can be concluded that, EURO 4 and 5 NOx levels are achievable with the use of an EGR-pump based on simulation results. Unfortunately it is not given by [11, Iveco] what type of pump is used as EGR-pump and how the pump is driven. The illustrations in this paper suggest that a radial (turbo) compressor is used as EGR-pump.
When a single stage VTG-turbocharger is used in combination with the EGR-pump EURO 4 NOx emission can be achieved without increasing BSFC. Peak cylinder pressures will be no less than 170 bar at approximately 12% EGR and an AF-ratio of 25 ($\lambda$; 1.7). Also in case of a single stage VTG-turbocharger EURO 5 NOx emission can be achieved, however at lower AF ratios: 20.3-21.8 ($\lambda$; 1.4-1.5). Furthermore when using a single VTG it can be seen that at low engine speed, the EGR-pump causes high BSFC compared to a High- and Low- pressure EGR system.
In case of using a mechanical charger in combination with the EGR-pump both EURO 4 and 5 NOx levels can be achieved with minimal AF-ratio of 25 ($\lambda$; 1.7), and a boost pressure in the region of 4 bars. Peak cylinder pressures are never less than 170 bars, and also BSFC and heat rejection are greatly increased, opposed to the use of a single turbocharger an EGR-pump. At EURO 5 NOx emission the BSFC is much higher (>15%) than at the EURO 4 NOx level. The same applies for the heat-rejection, which increases by approximately 25% over the EURO 4 level, and about 60% over the EURO 3 level.
In case of applying a two-stage turbo-system, with a maximum pressure ratio of 4.2, both EURO 4 and 5 NOx levels can be achieved. EGR levels are up to 20% while maintaining A/F ratio and without any increase in BSFC over current EURO 3 engines. However, heat rejection is increased over the EURO 3 engine by approximately 60%.

From the little test results available, it comes forward, that EURO 4 and 5 NOx emissions are achievable, however it is not clear whether PM emissions are also up to EURO 4 and 5 level. Boosting solutions associated with high-pressure EGR may also apply to externally driven EGR configurations, with the two-stage turbo-system giving the best results, meaning a relative high A/F ratio. At some engine operating points, particular low speed, the pumping work of the EGR-pump is believed to cause relative high BSFC [11; Iveco]. An integrated EGR pump is capable of delivering medium levels of EGR (12-16% possibly higher) with relative low pumping losses and hence good engine efficiency.

### 3.1.2 Internal route EGR

Internal route EGR is a method of applying EGR in which the re-circulated exhaust gas does not leave the engine, but is kept inside. It is possible though that the re-circulated exhaust gas enters the exhaust manifold but is sucked back into the combustion chamber. The way in which internal route EGR is accomplished is by timing the valves of the engine in such a way that not all exhaust gas is driven out of the engine or by timing the exhaust-valves for reopening after the engine expansion-stroke in order to re-intake the exhaust-gas from the exhaust-manifold. Timing the valves in this way can be done by either Fixed Valve Actuation giving the valves a fixed timing with large overlap or by Variable Valve Actuation (VVA) giving variable valve timing. VVA may also allow for variable valve lift as is often used in light duty applications for performance purposes.
Internal route EGR, may be abbreviated "Internal EGR" in the rest of this thesis.

- *Valve timing*

As already indicated, internal EGR is accomplished by means of engine valve timing. Several options in valve timing are possible in order to retain exhaust gas into the cylinders. Most effective and thereby most used methods for HD diesel engines are [15, 17, 28];

- Negative valve overlap, this can be a combination of early exhaust valve closing and late inlet valve opening, or by only an early exhaust valve closing (325° to 369° ca), or by earlier phasing of the exhaust valve event (starting exhaust valve opening at 70° to 125° ca).
- Reopening of the exhaust valve after it has closed, during the engine intake stroke, simultaneously opening intake and exhaust valve in order to suck exhaust gas from the exhaust manifold back into the cylinder.
- Large valve overlap between the intake and the exhaust valve, usually by late closing of the exhaust valve

Figure 3.10, shows diagrams of the above valve timings;

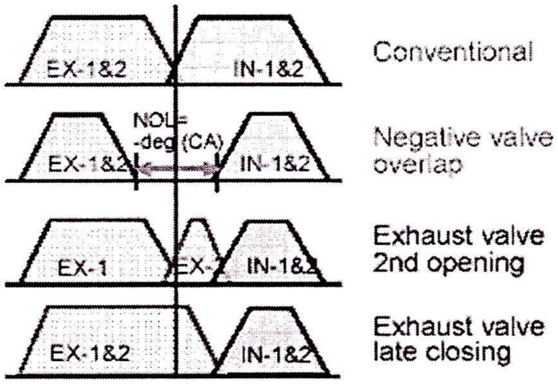



*Figure 3.10 [28]; Valve timing diagrams*

From simulations conducted by [17] it appears that the method of reopening the exhaust valve gives the most promising results, in terms of levels of EGR, engine performance and exhaust temperature, and is therefore preferable. However according to [17; Edwards]; "Second exhaust valve opening (2EVO) relies on exhaust manifold pressure pulsations to achieve the required internal-EGR level. These pulsations are a strong function of the manifold geometry, therefore the cylinder to cylinder variation in EGR level is likely to be greater than with an external-EGR circuit unless careful attention is paid to the manifold design."

- *Lay-out*

As internal EGR depends on valve timing, several valve-train systems will be discussed here, which are suitable for accomplishing internal EGR. The valve train systems are:
- Fixed valve timing
- Hydraulic activated VVA
- Electro-mechanical VVA

Many other systems are available in order to accomplish variable valve timing, such as a second camshaft, 3D cam profile and even extra valves that can provide EGR. However many of these systems are not yet applied for accomplishing internal EGR (except for the extra EGR intake valve). Also many of these other variable valve-timing systems are still far from operational for (Heavy Duty) Diesel engines or have serious drawbacks. Therefore these systems will not be discussed here. Besides a suitable valve-train system other aspects of internal EGR, charging and cooling, are discussed here.

*Fixed valve timing*

The fixed valve timing approach to internal EGR relies on relative large overlap between the intake and the exhaust valve, usually by late closing of the exhaust valve. Or a negative valve overlap can be used, which combines exhaust valve early closing and inlet valve late opening [28]. The desired valve timing is accomplished by a fixed cam-shaft phasing or fixed cam profile, or a combination of the two. Depending on the boost level and exhaust gas backpressure as well as the overlap duration and corresponding valve lifts, the EGR fraction can vary substantially. Especially in engines with high thermal load, where the boosted air is also used to cool the cylinders during valve overlap, the internal EGR-rate at low speed and load can be too high [15]. Furthermore, due to the constant cam lobe profile optimized for certain operating conditions, compromises, such as increased pumping losses caused by retarded exhaust cam phasing or reduced volumetric efficiency due to early intake closure of the intake valve must be accepted. In principle secondary exhaust valve lift, in order to re-intake exhaust gas during the engine intake stroke, could be made possible in a fixed valve timing system if a additional profile on the existing exhaust cam would be used. However sufficient lift would be difficult to achieve [17].
In summary the major advantage of a fixed valve timing system is its relative simplicity, no extra control of the system is required. Major disadvantages are the expected high variations in EGR-fraction and therefore high variations in engine-out emissions. As the system is fixed and set for certain engine conditions, the maximum EGR-rate may be limited. Furthermore as the system is fixed and compromises are made in its set-up, it may result in a major decrease of engine efficiency in some engine operating points.

*Hydraulic VVA*

In a hydraulic VVA system, (small) hydraulic cylinders actuate the valves. In most cases for HD diesel engines the actuation is only partially hydraulic and the hydraulic cylinders are embedded in a traditional, cam driven valve train. The hydraulic system is then only used to alter the normal valve timing given by the fixed cam profile, to accomplish internal EGR.
Some HD diesel engine manufacturers are already applying hydraulic VVA, amongst which is Caterpillar (ACERT™) [25, 26, 27].
A hydraulic VVA system for the purpose of accomplishing internal EGR that is well described in literature is the IEGR system from Jacobs Vehicle Systems. This system applies technology similar too the well-known Jake Brake® engine-retarder. In this Hydraulic VVA system, use is made of master-slave hydraulic cylinders embedded in a traditional, cam-driven valve train. In this way the exhaust valve can be made to follow the movement of the intake valve.
Figure 3.11 schematically shows the lay-out of the Jacobs IEGR system. In this figure it can be seen that a *master* cylinder-piston is placed on the intake rocker arm and a *slave* cylinder-piston is placed on the exhaust rocker arm. With the help of solenoids, control valves and hydraulic lines; the slave piston can be actuated by its master piston on the intake rocker arm, letting the exhaust valve follow a controlled movement laid on by the movement of the intake valve. This enables reopening of the exhaust valve at opening of the intake valve at the engine intake stroke, thus providing internal EGR. The electronic control of the solenoid controls to what extend the

exhaust valve follows the movement of the intake valve by controlling the flow of hydraulic fluid between the master and slave cylinder-pistons.
Also it can be seen in figure 2.7 that a *master* cylinder-piston is placed on the injector rocker arm, this enables engine braking, by opening the exhaust valve near TDC of the engine.

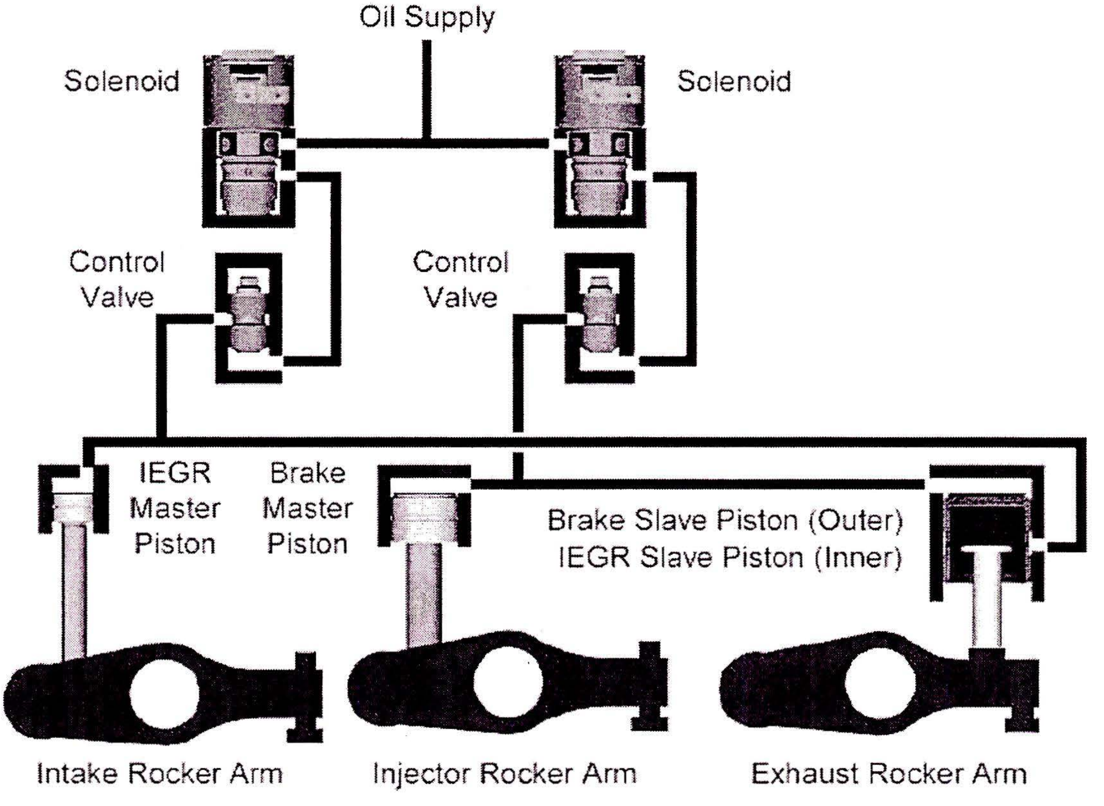



*Figure 3.11 [5; Jacobs] System schematic of On-Off internal EGR, plus braking system*

It is indicated by [5; Jacobs Vehicle Systems] that in applying the Jacobs hydraulic VVA system as described above, in case of 20% EGR, the turbo nozzle area should be reduced by 30% compared to the baseline engine without EGR.
According to [5], internal EGR in combination with a retarded injection timing can provide a 30% reduction in diesel NOx emissions and is an attractive solution for meeting NOx emission levels in the range of 3.4 and 4.0 g/kWh for HD diesel engines, especially for off-road and vocational applications.
From engines tests performed with the Jacobs hydraulic VVA system on a HD diesel engine it comes forward that over a 13-mode ESC test cycle a reduction of 24 % NOx is possible opposed to the baseline value. With that the BSFC increases by 1%, a large increase in PM emission is also witnessed (over 400%) compared to the baseline value. Baseline values are from the same engine without the use of EGR.
The maximum EGR level realized at engine tests is 16%. The amount of EGR realized, amongst others, depends on the amount of valve lash that the exhaust valve experiences at reopening after the engines expansion stroke. A lower valve lash appears to result in larger EGR levels than a large valve lash. At high engine speed and load (1800rpm) high back-pressure in the exhaust collector results in a relative high EGR level. At mid engine speed (1500 rpm) “the gas starts to flow back into the exhaust port at the beginning and end of the valve lift” [5]. Meaning that some of the EGR-gas escapes back into the exhaust manifold. At low engine speed (1200rpm) “the back pressure is lower than the boost pressure” [5]. This again causes gas to flow back into the

exhaust port at the beginning and end of the 2nd exhaust valve lift. Some of fresh air charge may even escape into the exhaust manifold. Based on simulations, an optimized turbocharger would show greater NOx reduction and generally less increase in BSFC than found at engine testing. The smaller turbine nozzle area (-30%) increases back-pressure without adversely affecting BSFC and eliminates the boost flow directly to the exhaust manifold.

It is also indicated by [5] that in order to reduce the excessive PM emission it may be needed to turn off internal EGR at high engine load, de-rate the engine, or combine the hydraulic VVA internal EGR system with particulate after-treatment.

*Electro-mechanical VVA*
Electro-mechanical VVA is not ready to be applied on a HD diesel engine, however it is indicated that this form of VVA may be a good alternative for hydraulic VVA for HD diesel engines. "More specifically, a hydraulic mechanism is vulnerable to environmental issues when the ambient and engine temperatures are low, and concerning any kind of mechanical system, a typical unsolved technical problem is the increased number of parts and the increased complexity of the system" according to [28; Urata, Honda R&D].
With electro-mechanical VVA a cam-less valve actuation system would be realized. The lay-out consists of electro-magnetic actuators that actuate each valve on the cylinder-head. In figure 3.12 a scheme of an electro-mechanical VVA system can be seen complete with its control or management system. In this case each valve (16) of the engine, both intake and exhaust, are actuated by an electro-magnetic actuator which is placed on top of the cylinder head, inline with the valve. As can be seen there are actually two electromagnets placed on each valve, the upper electromagnet is for valve closing, the lower electromagnet is for opening of the valve. A lift sensor on each valve provides feedback to the ECU and driver, which controls the electromagnets. As can be seen the valve management system is embedded in the conventional engine management system. The electromagnets operate on a low voltage of 42 V the total electric power required to operate a engine valve train are slightly less (-6%) then the mechanical power required to operate a conventional slipper-type valve train. However the electric power required for Electro-mechanical VVA is 200% more than the mechanical power needed for a roller-type mechanical valve train [28].

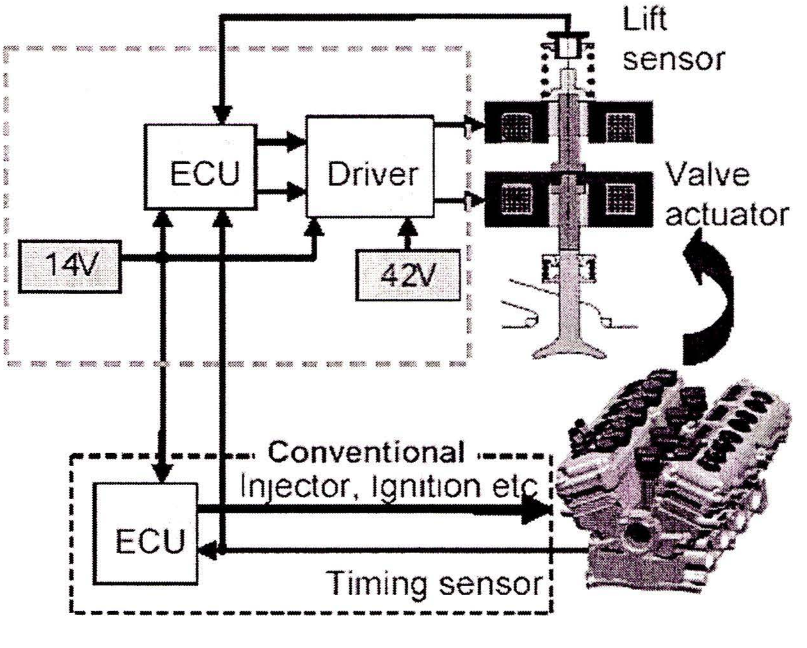



*Figure 3.12 [28]; Electromagnetic/ mechanical valve management system*

Because the electro-mechanical variable valve timing systems are cam-less they provide high flexibility in controlling internal EGR and overall engine control [15, 28]. In order to accomplish internal EGR with an electro-mechanical VVA system, all three possible valve timing strategies, large overlap, negative overlap and reopening exhaust valve, are possible. From [28] it can be concluded that 40% internal EGR is achievable using a negative valve overlap of -190° ca. Meaning inlet-valve opening at 95° ca. and exhaust-valve closing at 265° Ca. Unfortunately no test- or simulation results are available of an electromechanical VVA on a HD diesel engine.

*Charging*

As already indicated the manifold design is critical for pressure pulsations, responsible for the internal EGR at exhaust valve reopening. In this context it should be noted that the rate of internal EGR is driven by the pressure difference across the exhaust valve when it is opened. A negative pressure differential over the exhaust-valve, meaning high- exhaust backpressure, is required to obtain internal EGR.

If the turbine nozzle area is relative large, this can result in relative low exhaust backpressure, particularly at low engine speeds. The pressure differential over the exhaust-valve can become positive. As a result EGR levels become low and fresh charge may escape into the exhaust manifold at valve overlap and. This will have a negative effect on engine efficiency. Although VVA could reduce this problem by adjusting valve lift, the best approach is to resize the turbocharger, giving more exhaust backpressure. Another reason for turbocharger resizing is the reduction in turbine mass flow rate, due to EGR. Also higher boost pressures are desirable. This stresses the need for a turbo with a higher-pressure ratio and lower mass flow rate over the turbine when applying internal EGR [5].

It seems typical that in case of 20% internal EGR the turbine nozzle should be downsized by 30% [5]. It is also suggested that a light restriction in the exhaust system may be required to optimize performance by increasing exhaust backpressure [5].

*Cooling*

Little can be found in literature on EGR-gas cooling with the use of internal EGR. As the EGR-gas does only briefly leave the cylinders or does not leave the cylinders at all, it is nearly impossible to cool the EGR-gas. In some cases however it is indicated that the manifold could be water-cooled to accomplish some degree of EGR-gas cooling, however it also indicated that this would be far from sufficient to overcome all problems associated with hot-EGR, particularly at high EGR levels up to 50% (part load). The total heat-release through the engines conventional cooling system (engine block cooling) may be increased in order to lighten the thermal load on the engine.

- *Advantages and disadvantages of Internal route EGR*

The advantages of internal EGR are its very short response, since the EGR-gas does not have to travel outside the engine but is directly available where it is needed [3, 17]. Since no extra piping is required outside the engine, the internal EGR design will give little to no increase in engine packaging size [5, 17]. Furthermore no extra piping required with internal EGR, may also lead to lower initial costs, lower maintenance and higher reliability and better tolerance to high-sulfur fuel, causing fouling and corrosion, opposed to external EGR [5]. If use is made of VVA, this will allow for tuning of the valve-timing to the engine's present condition, for example; increased effective compression ratio to improve cold start, light load fuel economy and emissions [17].
The main disadvantage of internal EGR, is that it does not allow for cooling of the EGR-gas. If un-cooled EGR-gas is applied to the engine this will give a reduction in the engines volumetric efficiency, and thus result in a fuel economy penalty and overall lower engine performance [3]. According to [5] the fuel economy is typically 5% lower with (un-cooled) internal EGR than with cooled EGR, at the same NOx emission level, also a loss of power at full load may be witnessed.
As earlier described cooling the EGR-gas makes the EGR-gas more effective in reducing NOx emission, the absence of EGR-cooling therefore makes internal EGR less effective in reducing NOx emission than external cooled EGR configurations [3].
Furthermore the lack of cooling of the EGR-gas can result in high thermal stresses in the cylinder-head in case high levels of EGR (+40%) are used. In order to keep these thermal stresses to a acceptable level, the heat rejection through the engines conventional cooling circuit may have to be increased [7]. Also thermal throttling may occur due to the high gas temperatures, this can reduce the A/F ratio and thereby cause excessive PM emission [5, 7]. In this context the distribution of the EGR-fraction over all the cylinders needs to be equal, to prevent reduction in A/F ratio at certain individual cylinders [15]. In some cases it is indicated that in order to limit excessive PM emission it is necessary to turn off internal EGR at high engine load, to de-rate the engine, or to combine internal EGR system with particulate after-treatment [5].
Although internal EGR may provide relative small engine-packaging, complexity of VVA type internal EGR systems may be a disadvantage [17]. The level of EGR applied in an internal EGR-system is (indirectly) limited by the lack of EGR-gas cooling, further limitations are the A/F ratio (smoke limit), peak cylinder pressure, exhaust gas temperature to turbine, and turbocharger speed [5]. Engine durability may be reduced due to relative high thermal stress and contamination of the engines lubrication oil by soot from the EGR-gas [17].
Although the lack of EGR cooling is considered to be a major disadvantage of the internal EGR configuration, some tests show that hot EGR can reduce both NOx and PM emission with no adverse effect on fuel economy at low to medium engine load and EGR rate up to approx. 20%.
It is even quoted by [5; Jacobs Vehicle Systems] that; "uncooled EGR increases combustion efficiency at lower load points resulting in lower PM emissions".
Given the advantages and disadvantages of internal EGR, one manufacturer, Mack, applies internal EGR specifically for vocational trucks. These trucks often perform in a stop and go-manner over shorter distances, where operational hours are a more important concern than miles traveled. According to Mack, the approach of internal EGR provides a consistent level of emission benefits and performance in the varying environments in which vocational trucks operate [26].

- *Conclusions regarding literature study of internal route EGR*

It can be concluded that internal routed EGR has major disadvantages over external routed EGR, however there are also some advantages in favor of internal EGR. In contrast to external EGR no test-data is found that proves that internal EGR is capable of achieving NOx emission lower than 3.4 g/kWh. It is suspected that internal EGR is only suitable for less stringent emission requirements for example non-road emission standards. In case of more stringent emission regulations such as EURO 5 and US 2010, internal EGR maybe combined with other techniques in order to reduce emission, such as exhaust-gas after-treatment or external EGR. From the different valve-train possibilities for achieving internal EGR, the hydraulic VVA seems most suited for usage on HD diesel engines at this moment as fixed valve timing posses major drawbacks in terms of performance and electro-mechanical VVA is still not fully operational for HD diesel engines, while hydraulic VVA is already in use on HD diesel engines (Caterpillar).

## 3.4 Additional engine modifications and techniques in combination with EGR

In many cases EGR is applied on HD diesel engines in combination with techniques and methods other than conventionally used on HD diesel engines to further reduce engine out emissions. In some cases these other techniques and methods benefit from the use of EGR in reducing emission and in some cases the use of EGR becomes more effective in reducing NOx emission with the use of the additional techniques and methods. The additional techniques and methods found in literature are; alteration of injection timing, injection rate shaping, the application of the Miller cycle and HCCI or PCCI like combustion.
Exhaust-gas after-treatment will not be discussed here.

### 3.1.3 Injection timing and pressure

It is well understood that increasing injection pressure, is beneficial to lowering PM emissions as the injected fuel becomes better mixed (finer droplets and deeper penetration), this will promote oxidation and result in lower PM, whiteout negative influence on NOx emission. In case of High EGR level, with additional EGR and relative high AF levels, high boost pressures are required for the increased cylinder charge. The high boost pressures "and therefore increased in-cylinder pressure level, will alter the injection spray shape, reducing the penetration depth and therefore deteriorate the air utilization". "Overall higher injection pressures, in the range of 2000 bar and higher, compensate for the reduction of spray penetration due to the increased in-cylinder pressure" [7; FEV Motorentechnik].
In order to lower NOx emissions it is indicated that fuel injection timing could be retarded, however with a BSFC penalty involved [4; Ladommatos, 17; Ricardo]. According to [2; Heywood] "As injection timing is retarded, so the combustion process is retarded; NO formation occurs later, and concentrations are lower since peak temperatures are lower". "Retarding timing generally increases smoke, though trends vary significantly between different types and designs of diesel engine. Mass Particulate emissions increase as injection is retarded" [2; Heywood].
In order to lower PM emission with conventional diesel combustion it is indicated that fuel injection timing should be set earlier, increasing the "relatively soot-free premixed-combustion period" [4; Ladommatos, 31]. However a larger premixed combustion phase may promote NOx emission.
In applying high levels of EGR, to lower NOx emission, it is indicated in literature [15; FEV Motorentechnik] that; in order to counter a retarded combustion event due to an extended ignition delay, fuel injection timing should be set earlier. As such countering deterioration of fuel economy generally caused by the retarded combustion event.
Injecting the fuel in several stages, for example by giving a early pilot injection, can promote lower PM as more fuel is combusted in a premixed combustion, but it can also lower NOx. "With both pilot injection timings (5° & 17° CA. BTDC) the NOx formation rates during the main combustion are distinctly lower than the baseline without pilot injection". "Between about 8 and 25 deg CA ATDC, which is the initial combustion phase of the main injection of all cases, the gas temperature is lower with pilot injection than the baseline without pilot injection" [30; AVL List]. The lower gas temperature explains the lower NOx formation.
It is found that Split main injection can cut the "soot emission by up to 40% in all load points due to its effect of introducing new mixing energy in the combustion chamber" [7; FEV Motorentechnik].

### 3.1.4 Injection rate shaping

With the modern fuel injection system, a larger degree of freedom is obtained in fuel injection pressure and timing. Utilizing this larger degree of freedom, can be done in the pressure build-up when injection fuel.
In order to lower engine out NOx emissions it may be beneficial to inject the fuel in a way so that a so-called 'boot-shape' injection-pressure-diagram appears. Meaning that as the injection phase evolves more fuel is injected at higher pressure. According to [15; FEV Motorentechnik 2002]

"this technology provides low injection rates to suppress initial NOx formation caused by the mixture preparation during ignition delay".
According to [30; AVL List] the boot-shape injection rate shaping strategy does result in a NOx emission reduction of 14% at constant BSFC opposed to a standard injection without rate shaping at a typical operating point of a 2 liter single cylinder research engine.
However according to [7; FEV Motorentechnik 2003]; "Injection rate shaping has shown no additional benefits in terms of NOx reduction when high EGR-rates are applied. The block-like but multiple split injection profile seems to be most beneficial for low soot emissions at high EGR-rates".

### 3.1.5 Miller cycle

The miller cycle is characterized by a reduction of the effective compression ratio compared to the geometric expansion ratio of the engine. This can be realized by "advancing the closure of the intake valve or variable valve lift, resulting in an expansion upstream the compression stroke and therefore a lower temperature level of the intake charge to reduce NOx emissions" [15; FEV Motorentechnik].
However in order for the Miller cycle to be effective over the entire engine operating range, the application of the Miller cycle requires a variable intake process. Thus a VVA-system with variable intake valve actuation will be required. Also as a consequence of shortened intake duration or valve lift, a power loss is to be expected. In order to overcome this power loss boost pressure will have to be increased. It is indicated by [15] that incase of applying a two-stage turbo-charging system to increase boost pressure in a Miller cycle, "NOx reductions of up to 35 % can be achieved with a simultaneous increase in thermal efficiency".
The Miller cycle could be beneficial in the application of internal EGR. According to [17; Ricardo] "the operating cycle with a short effective compression stroke and long expansion stroke can therefore be regarded as a cycle with internal cooling or with extended expansion. This cycle could provide a solution to the thermal issues associated with an Internal EGR engine". However also according to [17; Ricardo] "the fact that little durability and cost information is available for a Miller cycle plus Internal EGR engine it must be considered that the potential of such an engine at Euro 3 is insufficient to warrant further investigation" and "expansion cooling suitable for higher EGR levels (e.g. Euro 4) does not seem feasible (it would require greater than 6 bar boost pressure and lead to excessively high valve accelerations)". According to [15; FEV Motorentechnik] "the realization of a variable intake valve mechanism is considering the cost, packaging, reliability and durability of a HD diesel engine, still represents the largest hurdle".
As already indicated high-EGR in combination with relative high AF levels demands an increase in engine charge and relative high boost pressures. Applying Miller would further increase the required boost pressure. In most external route EGR configurations achieving sufficient boost pressure for high-EGR is already an issue. It is therefore likely that applying a Miller-cycle will lead to practically unobtainable boost-pressure requirements.

### 3.1.6 HCCI & PCCI

*HCCI*

HCCI or Homogeneous Charge Compression Ignition is a combustion process aimed at provoking premixed diesel combustion instead of conventional diffusive diesel combustion. In order to create a homogeneous mixture use is made of very early diesel injection or port injection. In case of direct injection modifications to the injection angle and piston bowl are needed to avoid wall impingement of the fuel (NADI). In most cases use is made of EGR to keep the HCCI combustion process controllable and to diminish excessive NOx emission. HCCI in combination with High-EGR can result in ultra low PM and NOx emission. The low PM is most likely due to the premixed combustion and the low NOx due to the EGR.
However, today HCCI faces many problems and drawbacks, such as loud noise, high HC emissions and difficulties in control particularly over a wide engine envelope (high engine load and speed).

*PCCI*

PCCI or Premixed Charge Compression Ignition is a combustion process in which a large amount of fuel is injected and mixed during the engine's compression cycle. The homogeneous mixture that is formed combusts in a premixed combustion process. The premixed combustion process is then followed by a conventional fuel injection and diffusive combustion process. The difference between PCCI and HCCI is that in HCCI it is aimed at combusting all injected fuel in a premixed combustion process, while in PCCI only part of the fuel is combusted in a premixed combustion process. The aim of PCCI however is the same as with HCCI; reducing NOx and PM emissions by means of premixed low temperature combustion instead of diffusive burn.

The pilot injection plays a dominant role in PCCI, combustion, however it is also possible to create PCCI combustion with a very late and fast injection of fuel after TDC [32]. The advantage of such a late injection is that the mixture is not able to combust prior to reaching TDC as may be a problem when applying pilot injection. The late injection strategy provides a more controllable ignition process. The disadvantage is that a large amount of fuel needs to be injected vary fast to be able to form a homogeneous mixture prior to ignition and as combustion takes place after TDC efficiency decreases.

PCCI is often mentioned in combination with high EGR levels for the same reasons as with HCCI, EGR provides a low-temperature combustion reducing NOx and may help in controlling the combustion process. PCCI faces similar problems as associated with HCCI although to less extend.

## 3.5 Conclusions regarding EGR configurations based on literature findings

Based on conducted literature research conclusions are drawn regarding the different EGR-configuration.The conclusions form a basis for further research in the form of calculations and simulations. From literature-research it can be concluded that:

In any form of high-EGR application it is desirable to maintain a respectable AF level in order to prevent excessive PM and fuel economy deterioration. As such the total cylinder charge needs to be increased compared to non-EGR. Increasing the cylinder charge mass demands severe increase of boost pressure, and thus increased turbo power. EGR cooling becomes an important aspect in case of High-EGR application.

Several engine configurations are possible for supplying EGR on HD diesel engines.
The external routed internally driven High-pressure is by far the most popular and best-documented EGR- configuration. External routed internally driven High-pressure configuration is used for low EGR purposes in almost all cases. In most cases a VTG turbo-charger is adapted for the purpose of EGR control. It is found that in order meet future emission regulation (Euro 5 & US 2010) by means of High-EGR, the external route Internally driven High EGR configuration is suitable, however featuring two-stage turbo-charging and high capacity EGR-cooling.
External route internally driven Low-pressure EGR is found to be less popular for current EGR applications. However it is indicated that in comparison to the more popular High-pressure configuration, the low-pressure configuration has the potential of lower fuel consumption, due to a better use of the turbo. It is also indicated that cooling of the EGR-gas is critical for turbo-performance as this affects the compressor entrance temperature. The major drawback of the Low-pressure configuration is the turbo durability issue. Simulations have shown that Euro 5 NOx levels are achievable with a Low-pressure EGR system. Here also the need for high boost pressures is stressed in order to maintain a sufficient AF level.

External route externally driven EGR systems are found with various EGR-pump options. The external route, externally driven EGR systems overcome the issue of increasing exhaust-backpressure in order to create EGR flow from the exhaust manifold to the inlet manifold. It is indicated that Euro 5 NO emission levels are achievable however with overall higher BSFC. With the use of an EGR-pump for high EGR levels engine-packaging and fuel efficiency problems are to be expected For example; if an EGR level is required of 50% and boost pressure is in the order of 4-5 bar in order to maintain a sufficient AF-ratio (order 23:1 [33]), it can be estimated that this would require a big, possibly two-stage EGR-pump or positive displacement pump (piston pump), considering the high-pressure difference between exhaust manifold and inlet manifold. The high mass-flow rate, 50% of the total charge, requires a high capacity and thus big pump. In case of a piston pump it will be easy to see that at 50% EGR the EGR-pump may be up to 50% the size of the engine block. It is likely that BSFC is to suffer under the usage of such big external pumps.

Internal route EGR has attracted the interest of some HD diesel engine manufactures and offers some advantages in terms of packaging, reliability and response. However it is indicated that internal route is not suited for High-EGR up to 50%, mainly due the lack of EGR cooling options. Even in combination with a Miller-cycle in order to provide expansion-charge cooling, it is indicated that internal EGR is not suited for High EGR levels.

# 4. Simulations

In order to make a qualitative comparison of several EGR configurations, simulations are performed. It is aimed at comparing the different EGR configurations for their suitability for High-EGR application under realistic conditions for future Heavy Duty Diesel production engines. The focus in these simulations is on the charging of the engine. Prediction of engine emissions is not part of the simulations.
Several EGR configurations are selected for simulation based on findings in literature. Target values for EGR and $\lambda$ are determined to meet a targeted NOx emission level (US 2010). A preparatory study is performed in order to obtain an indication of the required turbo-performance and system limitations for high levels of EGR. A base-line engine model is constructed in GT-power. This engine model is subjected to validation with engine measurement data. A VTG-sweep is performed in order to gain insight in the control function of the VTG-turbo for EGR applications. A simulation case-setup is made in order to apply for all simulations for good comparison. The selected EGR configurations are modelled, based on the base-line engine model and tested according to the predetermined case-setup. Some level of optimization is applied for each configuration. Finally the simulation-results for each EGR configuration are compared.

## 4.1 Selection of EGR configuration for simulations

Based on the findings from literature a number of EGR configurations are selected for further research. The selected configurations are:

- External route Internally driven- High-pressure
    - single stage- VTG turbo
    - two stage- VTG & fixed geometry turbo
- External route Internally driven- Low-pressure-
    - two stage- VTG & fixed geometry turbo

The internally driven high-pressure EGR configuration is by far the most popular and well-documented EGR configuration and it is indicated that relative good results for medium EGR levels (20%) are achieved. The internally driven high-pressure EGR configuration offers good potential for achieving high levels of EGR based on results achieved with medium EGR levels; it is therefore selected for further research. The high-pressure EGR configuration with the use of a VTG turbo-charger is the most favorable high-pressure EGR configuration, because of the control function of the VTG in controlling the EGR and AF level. It is also found that in order to reach the desired high EGR levels in combination with acceptable AF ratios, high boost pressures are required and therefore two-stage turbo-charging may be essential. Therefore the simulations are conducted with both a single stage-VTG- high-pressure- internally driven-EGR configuration and a two stage-VTG & fixed geometry turbo- high-pressure- internally driven-EGR configuration.

The low-pressure EGR configuration is mentioned in literature as a good alternative to the more conventional high-pressure configuration, offering some potential benefits over the high-pressure EGR configuration and is therefore selected for further research. Initially the same two-stage turbo configuration as selected for the high-pressure EGR configuration is chosen for the low-pressure configuration, minimizing the effects of turbo-properties in the comparison of the High and low-pressure EGR configurations. However the turbo-system is scaled different in order to compensate for the difference in mass-flow through the turbo-system compared to the high-pressure EGR configuration.

Internal route EGR configurations offer some major benefits over external route EGR configurations and have attracted the interest of some major engine manufactures. However for high-EGR application internal route EGR is found to be unattractive, mainly because of difficulties in cooling the EGR gas or cylinder charge. Therefore internal route EGR configurations will not be further examined in this thesis.

Externally driven EGR configurations are not further examined because of the major impracticability of such a system. Based on the increased total engine mass flow and the large EGR mass-flow, for high EGR levels and relative high AF-ratios, it is estimated that the pressure ratio over the 'EGR-pump' is in the order 4-5 bar in combination with a mass-flow over 0.45 kg/s for a 12.9L HD DI diesel engine (50% EGR). This would require a big and possibly two-stage compressor system or a positive-displacement pump (piston pump), resulting in a major increase in engine-size up to 50%. Furthermore it is expected that BSFC is to suffer under the usage of such big externally driven pumps.

Initially a minimum of options is selected for the external route EGR configurations. The options that are considered indispensable for a diesel engine with high EGR are the EGR valves, EGR cooler(s) and a backpressure control valve. The EGR valves offer control over the EGR level and AF in addition to the VTG-turbo. Although it is not preferable to throttle the EGR flow, because of the BSFC penalty involved, the EGR valves may be required for some operating points and transients. As is clearly stressed in the literature study (Chapter 3) cooling the EGR-gas is required in order to reach high levels of EGR. For the Low-pressure EGR configuration a certain restriction in the exhaust-system is required in order to force exhaust-gas to the inlet system of the engine. By implementing a back-pressure exhaust-valve the restriction in the exhaust-system can be varied to reach optimum performance and control EGR-flow as varying the VTG-vane position has little influence on the EGR level in a Low-pressure EGR configuration.
Options such as an air-intake throttle, reed valves and venturi-mixer pose disadvantages in terms of packaging, costs, BSFC and performance and are therefore not further examined. However the reed valves and the venturi-mixer remain interesting options for achieving high-EGR, as they may add some level of improvement in EGR flow. The use of PM filters and a EGR-cooler-bypass may eventually become required in order to reduce respectively PM and HC emissions and to meet (turbo) durability demands, but the application of these devices is outside the scope of this thesis.

## 4.2 Simulation target values

For further research in the form of simulation, target values for NOx, AF and EGR are defined. Also a target value for an additional parameter FS (Fraction Stoichiometric) is defined. The target values are given in table 4.1 and are further explained below.

| *Parameter* | *Target value* | |
|---|---|---|
| NOx | 0.27 | g/kWh |
| AF | ≥ 20.3 | kg/kg |
| FS | ≥ 0.36 | - |
| EGR | 50 | % |

*Table 4.1*

The NOx target value is chosen 0.27 g/kWh based on the US federal (EPA) 2010 emission standard for heavy-duty diesel truck and bus engines, which is currently the most progressive emission regulation for Heavy Duty diesel engines.

The AF target is based on earlier findings by TNO automotive and on literature study. From the TNO report [82] it comes forward that for HD diesel engines combustion efficiency decreases and BSFC increases exponentially below $\lambda$ = 1.4 values. Similar results can be found from [33]. Furthermore according to TNO [82] EGR levels up to 70% with a $\lambda$ value as low as 1.4 gives satisfying result in terms of low NOx however at very low BMEP's. For further research the minimal target $\lambda$ value is set at 1.4 which equals AF = 20.3. It is aimed at combustion efficiencies above 99%.

The target EGR value is determined with the help of FS based NOx prediction (appendix 2.1);
FS (Fraction Stoichiometric) represent the mass fraction (or percentage) of inert gas in a gas volume.

$$FS = \frac{m_{inert}}{m_{tot}} \qquad \text{eq. 4.1}$$

$m_{inert}$ : the mass of inert gas

$m_{tot}$ : the total gas mass in a volume

In an engine the inert gas fraction in the cylinder charge created by EGR lowers the local $O_2$ concentration. This lowers the NO formation rate by the effects described in chapter 2. With EGR the inert gas in the cylinder charge contains $CO_2$ and $H_2O$ from the exhaust gas, which is of importance for the thermal effect of EGR on NO formation. Because of this a relationship exist between the FS value and engine out NOx emissions. As such the FS value can be used as a parameter for predicting engine-out NOx. However the relationship between FS (or oxygen concentration) and engine-out NOx emissions is not unambiguous. Other factors, such as inlet charge temperature, fuel injection timing and pressure and engine geometry are of influence on NOx emissions as well. The use of FS in predicting NOx emissions is only based on measurement-results from engine testing and can only give an indication of the NOx emissions to be expected at a certain EGR level and AF ratio for a certain engine at similar engines operation as during measuring.

The FS value of the cylinder charge can be calculated based on the EGR level and the real- and stoichiometric AF ratio. The complete derivation of equation 4.2 is given in appendix 2.1;

The expression for FS, where FS is a function of EGR and AF;

$$FS = \frac{(AF_{stoich} + 1) \cdot EGR/100}{AF_{real} + 1} \qquad \text{eq 4.2}$$

The relation between FS and Oxygen concentration;

$$\%O_2 = (1 - FS) \cdot 21\% \qquad \text{eq 4.3}$$

In this case the FS parameter is used in order acquire a realistic target value for the EGR level given future emission standards and in order to make quantitative comparisons between simulation results. Determining the required FS level for reaching the desired NOx emission level of 0.27 g/kWh (US 2010), must be regarded as an estimation, since it relies on extrapolation of measurement results and because FS in not the only factor of influence on NOx emissions. However the aim of the following simulations is to determine the suitability of the selected EGR configurations for high-EGR levels in general. Herein the US 2010 NOx level serves as a target value for determining a realistic high EGR level. No attempts will be made to predict NOx emissions, due to limitations of the models in generating reliable NOx emission predictions (§4.6).

Based on measurement-results from previous engine tests at TNO, two relation-ships are derived, which describe NOx emission in g/kWh as a function of FS. As FS is a function of EGR and λ, NOx emission can be plotted as a function of EGR and λ (see appendix 2.1). In figure 4.1 a fitted relationship between NOx and FS can be seen based on measurement results.

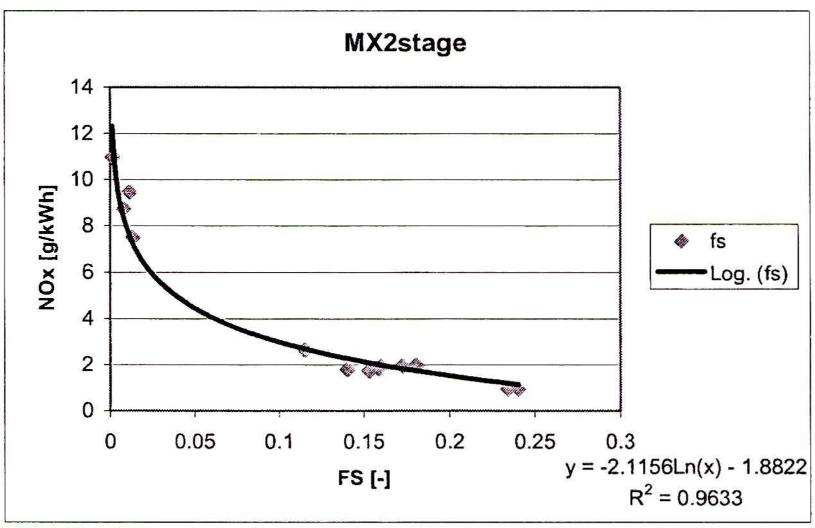



*figure 4.1 NOx vs. FS*

With the help of the derived FS vs. NOx relationship (A 2.1.2) plotted above, extrapolation of the measurement results is performed. From this extrapolation it follows that the target FS value should be 0.36 in order to achieve a NOx emission of 0.27 g/kWh (see appendix 2.1). For a FS value of 0.36 and an AF-ratio of 20.3 an EGR level of 50% is required. These are the minimal target values for both AF and EGR; higher AF values demand higher EGR values and visa-versa in order to reach the target FS value and subsequently the desired NOx value.

## 4.3 Preparatory EGR and turbo study for high EGR

A preparatory study on turbo requirements for high EGR is performed in advance of simulations in order to obtain an indication of the required turbo-performance and limitations. The preparatory study can be found in the appendix 2.2, here only a summary is given.

In order to determine turbo requirements for usages in combination with High-EGR, several calculations are performed. As a base-line engine a DAF MX engine is used rated at a maximum of 390 kW, the same as used in recent engine measurements with a DAF MX engine at TNO and as is used in the simulations.

Calculations are performed for an external routed, internally driven 'High-pressure' EGR configuration. Calculation's are performed for 12 points of the 13mode ESC steady state engine test cycle (all but idle), with EGR levels between 40 and 70% at λ=1.4 (AF=20.3). Several assumptions are made, which are listed in table T2.2.3 in appendix 2.2.

Given the engine speed and load (ESC test cycle, appendix 3.2), the required BMEP is calculated. Also the required fuel and fresh air mass flows are calculated. For the mentioned EGR level the total mass flow through the engine is calculated. The required manifold pressure and boost pressure is calculated and finally the pressure ratios over compressor and turbine as well as turbo work are calculated. Please note, no combustion is calculated or simulated during these calculations, instead several assumptions are made.
In result of the calculations an indication of the required pressure-ratio's over turbo-compressor and turbine are plotted against the mass flow through these parts for the given EGR configuration and engine work-points. Such a plot for 50% EGR is given below figure 4.2.

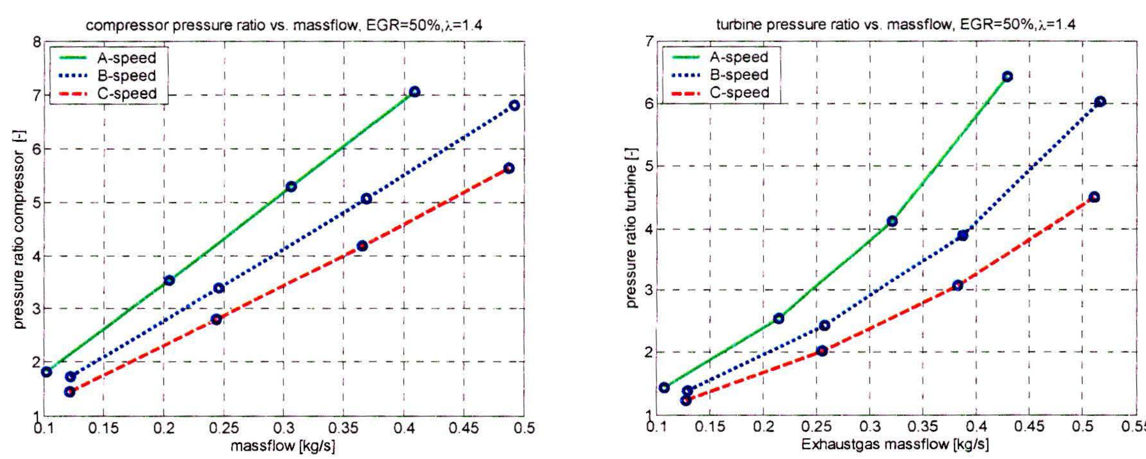



*Figure 4.2; required compressor and turbine map, at three engine speeds for 50% EGR and λ 1.4. (A=1210rpm, B=1525rpm, C=1830rpm). Assumed compressor efficiency: 75%, assumed turbine efficiency: 70%.*

Based on the results from calculation (appendix 2.2) it can be concluded that: in order to achieve 70% EGR with λ 1.4 and estimated compressor and turbine efficiencies (75% & 70% respectively) the pressure ratio over the turbine needs to become unrealistically high (over 20). In this case higher turbo-efficiency is required in the form of turbine efficiency of 80% or more.
At 50% EGR and λ 1.4 the required pressure ratios of both the compressor and turbine are high, up to 7 (figure 4.2) at relative low mass-flow (compared with available turbo-maps), but seem feasible, considering two-stage turbo-charging.

As such, an EGR level of 70% seems very difficult to achieve at high engine loads even at minimal lambda value. 50% EGR can be considered a maximum EGR target level for high engine loads. At lower engine loads the EGR level and/ or lambda should be higher. As most commercially available single stage turbochargers (either fixed or variable geometry) are not capable of reaching the desired high-pressure ratios (seven and higher) a two-stage turbo-charger system could be considered as the only option. The temperature of the EGR –gas is of influence on the required boost-pressure. Although it is desirable to maintain as low as possible EGR-gas temperatures, particularly at high engine speed and load, it will be difficult to maintain such a low EGR-gas temperature as high cooling capacity is required and condensation may occur. The feasibility and desirability of such a low EGR-gas temperature will be assessed.

## 4.4 Composition GT-power models

"GT-power is base on one-dimensional gas dynamics, representing the flow and heat transfer in the piping and in the other components of an engine system." "In addition to the fluid flow and heat transfer capabilities, the code contains many other specialized models required for system analysis" [87; GT-POWER User's Manual]
A base-line engine model is constructed based on an existing engine-model from DAF Trucks NV in GT-power. The base-line engine-model is used for validation and basic tests with a VTG turbocharger. Several engine-models featuring the selected EGR configurations are derivatives of the base-line engine-model. Only the construction of the base-line engine is described here, all other engine-models are basically the same with exception to the EGR- and turbo-charging system.

Crank-train
The base-line engine-model is constructed to resemble a production DAF MX engine, however equipped with a two stage turbo-system as this is required for high EGR applications (see §4.3). The DAF MX engine is a Heavy Duty 12.9 liter, 6-cylinder inline diesel engine. More engine-specifications can be found in appendix 3.1, table T3.1.1.

In the GT-power engine-model the engine block and Valve train are unchanged copies of the original engine-model supplied by DAF Trucks N.V. Some general information on engine-block and valve-train can be found in appendix 3.1, table T3.1.2.

Turbo system
The two-stage turbo-system in the engine-model consists of two turbo-chargers in series as is schematically shown in figure 4.3.

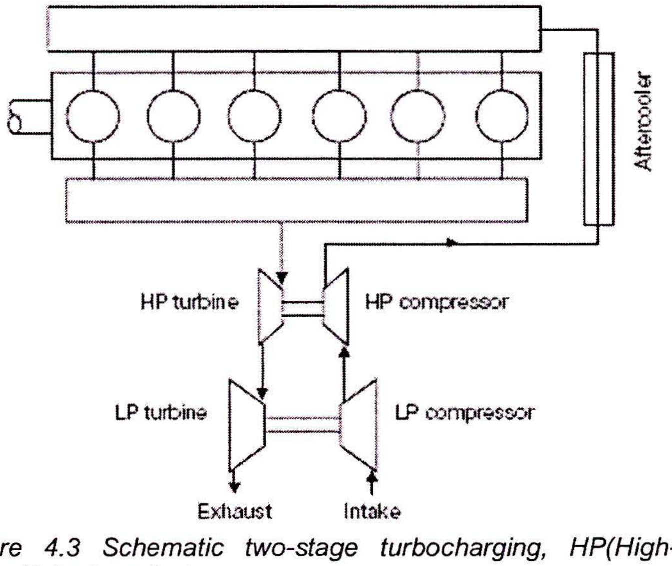



*Figure 4.3 Schematic two-stage turbocharging, HP(High-pressure) and LP(Low-pressure) [www.Holset.com]*

SAE turbo-maps are used for turbo specification in the GT-power model. In the base-line engine-model, for the High-pressure turbo use is made of the Holset HE431 VTG turbo-maps. For the Low-pressure turbo use is made of the Garett GT50 (108/56 compressor) turbo map. The Holset turbo has Variable Turbine Geometry (VTG) whereas the Garett turbo is a twin entry fixed geometry turbo. Both turbo's are designed for Heavy Duty diesel engines. Turbo-maps can be found in appendix 2.2. It is possible in GT-power to scale the turbo-maps by mass flow rate ('Mass multiplier'). According to GT power "The code calculates the instantaneous pressure ratio and speed and looks up the mass flow rate in the map. The rate is then multiplied by this attribute before it is imposed." This feature is used for performing simulations with smaller or bigger turbo's based on the original turbo-maps of available turbo's.

Cooling
The actual engine cooling circuit is not modelled in GT-power, instead separate models are implemented for heat-transfer from the different engine parts. Part-temperatures are user defined and at constant level for most parts (head, piston, cylinder).

The intercooler, cooling the compressed air (+EGR possibly), is modelled in GT-power by means of multiple (1000) identical pipes, and is an unchanged copy from the original engine-model, supplied by DAF Trucks N.V. The wall temperature over the entire length of the pipes is kept at a constant average level (313.15 K) as defined by DAF Trucks N.V.
The EGR-cooler, cooling the EGR mass flow, is modeled in an identical way as the intercooler by means of multiple (108) identical pipes and is also an unchanged copy from the original engine-model by DAF Trucks N.V The average wall temperature over the entire length of the pipes is kept constant, initially 398.15 K as defined by DAF Trucks N.V. Although attempted, it was not possible to create a more accurate model of the EGR-cooler, which would calculate the heat transfer process between the EGR-gas, cooler-wall and cooling water accurately. This requires the use of GT-cool, which was not available at the time.

EGR system
In the base-line engine model the EGR system is modeled as an external routed, internally driven, high-pressure system and is (initially) an unchanged copy of the EGR-system from the original engine-model from DAF. The EGR-system in the engine-model will be described from exhaust-manifold to inlet manifold, also see appendix 3.1, figure E3.1.1 and E3.1.2. As the exhaust-manifold on the MX engine is divided in two sections to fit a twin-entry turbo the EGR line is branched off at each of the two sections of the manifold giving two separate EGR-lines. Two EGR valves are implemented on both of the EGR-lines after which the two separate lines are merged in to one. The remaining single EGR line is connected to the EGR-cooler. After the EGR cooler, a venturi-flowmeter is implemented in the EGR line after which the EGR-line ends at the beginning of the inlet manifold, together with the end of the boost-pipe (pipe from compressor to inlet-manifold).

Piping
The piping of the engines inlet- and exhaust-system and the EGR piping are modelled by individual pieces of pipe and fittings connected together. Each individual piece has its own specifications, such as size and temperatures. Most pipes have constant wall temperatures, however some pipes have a 'Heat Conduction Object' (HTO) implemented, which recalculates the wall temperature based on heat transfer of the pipe to its surrounding. Some general information about the pipes and parts in the inlet-, exhaust- and EGR system can be found in table T3.1.3 in appendix 3.1. More detailed information can be found in de GT-power model itself as it is to comprehensive to mention all of the specifications in this report.

Combustion model
The 'Wiebe' combustion-model is used, as a combustion model. More on this can be found in paragraph 4.5.

Fuel injection system
The fuel injection system is modeled by means of a standard injector object in GT-power and is an unchanged copy from the original engine-model from DAF. However some set-up changes have been made. The original injection-profile as implemented by DAF has been maintained in the base-line engine model. However as the Wiebe combustion-model is used the injection profile will not influence the combustion rate. Detailed information about the injection-system can be found in the GT-power model.

Control systems
In order to optimize engine performance during the simulations several engine control functions are implemented in the engine-model. The control functions are on the fuel-injection system and on the VTG position.
For a steady state engine work-point it is desirable to keep engine speed and torque at constant predetermined level. In the base-line engine model the engine speed is fixed and torque is controlled by adjustment of the mass of injected fuel. The torque control system is modeled in the base-line engine-model with a Torque sensor on the engine crank-train and a PI controller. The PI controller controls each of the six injectors by adjusting the 'Injected Mass', within the hardware system limitations, attempting to reach the reference torque level. A so-called 'smoke-limiter' is activated which reduces the mass of injected fuel if a certain AF-ratio level is under exceeded. In this case the minimum AF-ratio is set at 14.5 which equals $\lambda=1$.
The VTG-position is of influence on the EGR level and AF level and subsequently FS level, particularly in an 'External routed, internally driven, High-pressure' configuration, also see paragraph 4.6. In order to reach a certain FS, EGR or AF level, the VTG-rack position can be adjusted by means of a control system. In the base-line engine-model the VTG-rack position is controlled with FS as control parameter. The real-time actual FS level is determined by several sensors and control-tools which determine EGR level and AF-ratio, from which the FS level is calculated. A PI controller adjusts the VTG-rack position until the reference FS level (or limitations of the VTG) is reached. A step-by-step explanation of this FS control-system can be found in appendix 3.1. Besides FS it is also possible to use the AF-ratio as a control parameter.

Convergence
In order to determine if a flow solution has reached steady state during simulation, steady state convergence criteria need to be met. The simulation can be stopped if all convergence criteria are met for every connection and flow sub-volume element in the model.

Here the simulation steady state convergence criteria are based on flow and pressure.

The steady-state convergence tolerance of flow, so-called "Fluid flow Convergence Tolerance" is set at 0.001 or 0.1%. Meaning that the relative change of the average mass flow rate is no greater than the "Fluid flow convergence tolerance" (0.1%) during cycle update. The relative change of the average mass flow rate is calculated as followed:

$$df = \frac{|(f_{new} - f_{old})|}{|f_{new}|} \qquad \text{eq; 4.4}$$

in which;

| | |
|---|---|
| $df$ | = relative change of average mass flow rate |
| $f_{new}$ | = the average mass flow rate of the most recent cycle update |
| $f_{old}$ | = the average mass flow rate of the previous cycle update |

The steady state convergence tolerance of pressure is not directly used to determine convergence. However, if the largest pressure change (dpmax) is greater than or equal to 1%, then the flow criteria above will be made four-times more strict:

$$|(f_{new} - f_{old})| / f_{new} < (0.25) \times Fluid\ flow\ convergence\ tolerance \qquad \text{eq; 4.5}$$

The pressure change is calculated as followed; $dp = \frac{|(P_{new} - P_{old})|}{P_{\max}}$ eq; 4.6

In which;

$dp$ = pressure change

$P_{new}$ = instantaneous pressure of the most recent cycle update

$P_{old}$ = instantaneous pressure of the previous cycle update

$P_{max}$ = maximum pressure of the last cycle update

The variable dpmax is the largest change for all sub-volumes.

## 4.5 Wiebe combustion model

As earlier mentioned in paragraph 4.4, the 'Wiebe' combustion-model is used as the combustion model in all constructed GT-power models. In GT-power the Wiebe combustion model consists of a three-term polynomial, which fits the combustion process in terms of Heat Release Rate.

General three-term Wiebe function:

$$x_b(\vartheta) = \alpha x_1(\vartheta) + \beta x_2(\vartheta) + (1-\alpha-\beta)x_3(\vartheta) \qquad \text{eq; 4.7}$$

$$x_i(\vartheta) = 1 - \exp\left[-a_i\left(\frac{\vartheta - \vartheta_{0i}}{\Delta\vartheta_i}\right)^{\omega_i+1}\right] \quad i = 1,2,3 \qquad \text{eq; 4.8}$$

| | | |
|---|---|---|
| $x$ | = burn rate (≈Heat Release Rate) | [1/deg] |
| $\vartheta$ | = crank angle | [deg] |
| $\alpha$ | = Wiebe parameter, premix fraction | |
| $\beta$ | = Wiebe parameter, tail fraction | |
| $a$ & $\omega$ | = adjustable Wiebe parameters | |
| $\vartheta_0$ | = start of combustion | [deg] |
| $\Delta\vartheta$ | = combustion duration | [deg] |

Subscripts;

| | |
|---|---|
| *b* | = total combustion process (complete burn) |
| *i* | = part of combustion process |
| *1* | = premix part of combustion process |
| *2* | = main part of combustion process |
| *3* | = tail part of combustion process |

In the equations 4.7 and 4.8 in total 14 Wiebe-parameters describe the diesel combustion. In GT-power the so-called 'DI-Wiebe' model reduces the number of Wiebe-parameters to 9. This is accomplished by using several parameters double and by calculating parameters based on other parameters. This is explained in appendix 2.1.
The DI-Wiebe model is a non-predictive model (unless additional models are included to predict the Wiebe-parameters). By leaving some of the user defined Wiebe-parameters at default values instead, the DI-Wiebe model becomes 'semi-predictive'. Meaning that the model will try to choose the Wiebe parameters automatically depending on injection profile, air-to-fuel ratio, in-cylinder pressure and temperature [GT-power help navigator]. "According to Gamma Technologies, this method is only applicable for full load conditions without EGR" [83, TNOreport| 05.OR.VM.021.1/AB| June 23, 2005].

In case of high-EGR the Wiebe-parameters in the DI-Wiebe model need to be user defined. As earlier described (chapter 2) EGR is of influence on the combustion process as it may increase combustion delay and increase combustion duration.

In order to determine the required Wiebe-parameters the combustion process is investigated, making use of cylinder pressure data from a similar DAF MX engine. The influence of high levels of EGR and FS on diesel combustion is investigated, also see appendix 2.3. Unfortunately only a limited body of measurement-data is available from experiments with High EGR. As such it is not possible to predict the Wiebe parameters at high levels of EGR and FS.
However certain trends are visible; ignition delay and combustion delay increases at increasing EGR and FS level. Furthermore the combustion process shifts towards the expansion stroke and peak pressure and ROHR (Rate Of Heat Release) are reduced under the influence of EGR and FS. At more extreme FS and EGR levels, two-stage –ignition was witnessed and incomplete

combustion. Naturally the mass of fuel injected is of major influence on ignition delay and combustion duration.

As it proved not possible to predict the true Wiebe-parameters for High-EGR, use is made of more conventional Wiebe setting in all cases, which can be found in the appendix 3.1 (composition GT-power models). These conventional Wiebe-settings are based on setting provided by DAF as part of the original engine-model for full load conditions. In using these conventional Wiebe-parameters the applied combustion process differs from the true combustion process in that the ignition delay and combustion duration are underestimated at high EGR and FS conditions.
It is estimated that in reality; increased combustion duration and elongation of ignition-delay, (as a consequence of High EGR) can be compensated for by advancing the start of injection. This under the assumption that a fuel injection system is available which enables a large degree of freedom in injection timing, such as common-rail or smart injectors (Delphi). In figure 4.4 it is illustrated how advancing injection timing can compensate for the effects of EGR and FS on the combustion process. On the left the combustion process is given as is typical under the influence of EGR and FS, the combustion process is shifted towards the expansion stroke. On the right advance of start of injection (SOI) compensates for this shift in combustion process as the center of each curve lines up at the same degree CA.

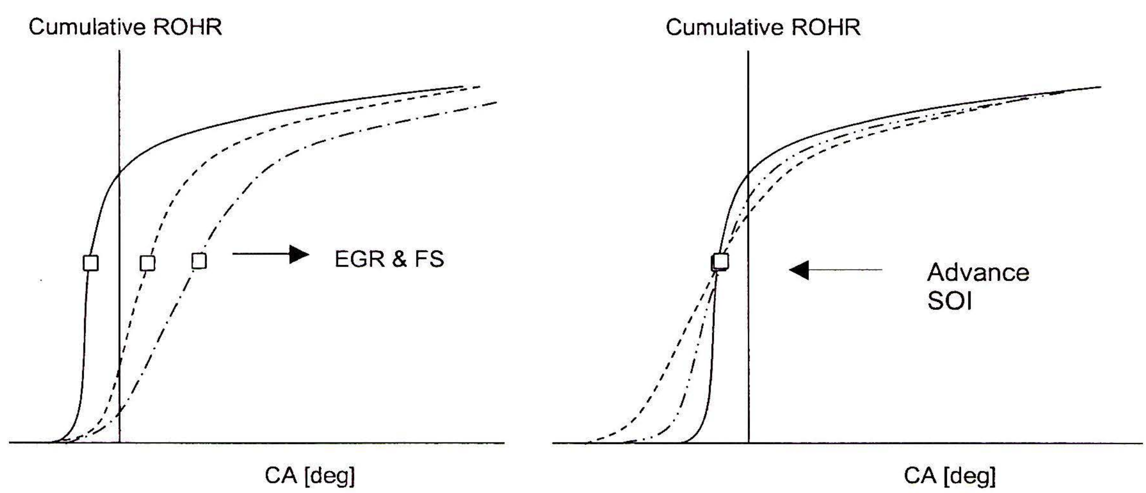



*Figure 4.4 left; representation effect of EGR and FS on cumulative rate of heatrelease. Right; advance of Start of Injection as compensation for effect of EGR and FS on combustion process.*

However advancing start of injection is limited if conventional diesel combustion is to be maintained (diffusive burn). A large premixed combustion-phase is provoked by early injection. The combustion process shifts from conventional diesel combustion towards PCCI and HCCI like combustion processes at a severe increase of the start of injection advance-angle. Also if start of injection is set earlier then approximately 40 deg BTDC [48] problems such as excessive wall impingement of the fuel spray can be encountered and require modifications to the injection system and piston bowl (NADI) [19].
So far, based on limited measurement data (see appendix 2.3) it is found that combustion duration increases in the order of 10°CA at High EGR (up to 70%). Advancing SOI by 10°CA in addition to the current used SOI of 5°CA BTDC (see § 4.8), would be feasible without the need of hardware modifications. The portion of pre-mixed combustion may however increase at advanced SOI, as conditions for ignition will only be met under sufficient pressure and temperature, ignition delay may increase.

In conclusion: for current simulation results, which are the product of simulations with constant, conventional, Wiebe parameters and conventional SOI (5°CA BTDC), several reservations need to be made;

- Exhaust gas temperature is underestimated, as in reality (true combustion process) at high levels of EGR the combustion shifts more towards the expansion stroke of the engine.
- Exhaust pressure is underestimated, for the same reasons of underestimation of exhaust gas temperature.
- BSFC is underestimated; in reality (true combustion process) the combustion process is shifted more towards the expansion stroke. As a consequence the IMEP decreases which will most likely have a negative effect on BSFC.
- Turbo-power and boost pressure are likely to be underestimated; given above reservation more power will be available to the turbine(s) and thus boost-pressure may be higher in reality.

## 4.6 GT-power base-line engine-model validation

In order to validate the base-line engine model, several simulations are performed with the base-line engine-model in order to copy already conducted real-life engine tests. The results from simulations and measurement data from real-life engine tests are compared. The complete validation-report can be found in appendix 3.3, Only a summary is given here.

The available measurement data comes from engine tests conducted by TNO with an experimental DAF MX engine with a two-stage turbocharging system. The experimental engine is basically the same as the engine modeled in the base-line GT-power engine-model. As such the turbocharging-system and EGR-system in the base-line engine-model is the same as used on the experimental test engine.
Four measurement-points are selected from the available measurement-data. The points selected are 4 ESC (steady-state) operating-points and are selected for their relative high EGR and AF-ratio levels and subsequently relative high FS levels.

In the simulation case-setup the engine speed, engine load, reference FS value and Start of injection are based on the measurement-points form measurement-data. For the validation simulations, engine load and FS are the two control parameters in controlling respectively fuel injection and VTG-rack position. The final (converged) actual engine load and FS level as well as other parameters, such as the EGR, AF-ratio and BSFC follow as results from the simulations.
Validation is conducted by comparing simulation-data with measurement-data for the following parameters; load, FS, EGR, AF, NOx, BSFC, manifold-temperature and -pressure and EGR temperature. In table 4.2 only the average value and standard deviation are given of the differences found for each validation parameter over the four measurement-points.

| **Statistics** | Average [%] | Standard.dev |
|---|---|---|
| load | 0.25 | 1.02 |
| FS | -1.03 | 0.26 |
| EGR | -0.10 | 3.26 |
| AF | 1.07 | 3.56 |
| Nox prediction | 3.58 | 52.79 |
| BSFC | -3.38 | 2.50 |
| Manifold. Pres | -11.47 | 11.31 |
| Manifold. Temp | -7.78 | 14.85 |
| EGR temp | -24.00 | 8.53 |

*Table 4.2; average and standard deviation of the differences found at validation for each validation parameter. Temp.=temperature, Pres.=pressure. EGR temp.= EGR-gas temperature downstream of EGR cooler.*

In conclusion to the conducted comparison (appendix 3.3), some differences are apparent between data from simulation and data from measurement for the same case (steady state operation point). For the parameters 'load, FS, EGR, AF and EGR temp' the differences can be explained as caused by controller inaccuracy and different pre-set reference values. Particularly the wall temperatures of the EGR-cooler and intercooler seem badly chosen, but can be improved. The NOx prediction from the model seems unreliable. The largest differences are found in the manifold pressure and manifold temperature and EGR temperature. To some extend the differences in manifold pressure are as a consequence of the difference in manifold temperatures which may be caused by inaccurate settings of EGR-cooler and intercooler wall temperature.

The differences in BSFC are also to some extend related to the differences in manifold temperature, due to inaccurate settings of EGR-cooler and intercooler wall temperature. However at low engine speed (1214 rpm.) BSFC prediction only deviated by 0.21%.
Unfortunately the available measurement data is limited. No in-cylinder pressure trace is available and the real VTG-rack position is also not available. Furthermore the measurement inaccuracy is unknown. Therefore it is difficult to determine exactly what causes the differences, particularly in BSFC, between simulation and measurement. Possible causes may be turbo-map inaccuracy, combustion model inaccuracy, differences in pumping loss possibly caused by differences in temperatures of pipe work and coolers. Also measurement inaccuracy could be the cause.

Although it is uncertain what exactly causes the model inaccuracy, it is known that the use of the Wiebe-combustion model can cause inaccuracies with the use of EGR (§4.5). As substantial levels of EGR (>20%) are used during validation this is a likely cause.

During the validation-simulations a conventional Wiebe combustion model is used which does not take into account the effects of EGR on the combustion process. In §4.5 it is already indicated that the use of a conventional Wiebe combustion model with high levels of EGR may lead to:

- Too low predicted exhaust-gas temperature
- Too low predicted exhaust-gas pressure
- Too low predicted BSFC
- Too low predicted turbo-power and boost-pressure

All of the effects described above are found in the differences between the simulation results and the measurement results. Leading to believe that the inaccuracy of the Wiebe combustion model with the use of EGR is the main cause for the found model inaccuracy.

Although attempted in §4.5 it was not possible to improve the Wiebe combustion model for high-EGR purposes due to insufficient measurement data. Therefore a certain level of inaccuracy in BSFC is inevitable with simulations at high levels of EGR due to inaccuracies in the combustion model. For the following simulations the focus is on the EGR flow, the AF ratio and the FS level. The simulation model can predict these three values with good accuracy. Therefore, despite the differences between simulation- and measured-data it is decided to continue with the current model.
The predicted BSFC in the following simulations must be regarded as indicative, due to model inaccuracy.

## 4.7 GT-power base-line engine-model VTG-sweep

As is indicated in literature a VTG (variable turbo geometry) turbocharger is beneficial in controlling EGR flow, particularly in external routed, internally driven, high-pressure EGR systems, by controlling turbine power and in effect of that; exhaust-manifold pressure. In order to gain more insight in the control-function of the VTG turbo for high EGR applications a simulation with a VTG-sweep is conducted. Meaning that the VTG-rack position is varied under identical conditions, constant load, speed, SOI etc. The effects of VTG-rack position on EGR, AF and FS are shown in figure 4.5.The complete VTG-sweep report is given in appendix 3.4 (VTG position test), only a summary is given here.
The simulation is conducted with the base-line GT-power engine-model as earlier described, however FS is not used as control-parameter in adjusting VTG-rack position. Instead the VTG-rack position is fixed. As such the EGR, AF-ratio and subsequently FS are not controlled and follow as results from the simulations. As a case-setup eight cases are used all at identical engine speed and load. The VTG-rack positions are varied from 0.1 till 0.7 divided over the eight cases in the case-setup. All other conditions are equal in all cases.

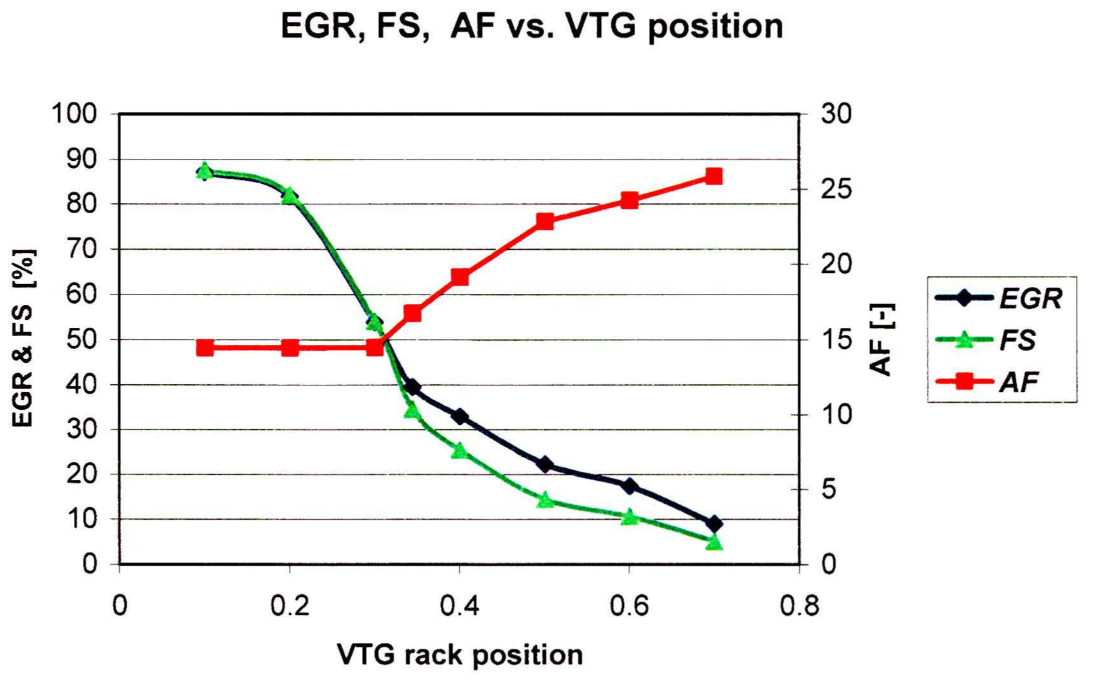



*Figure 4.5 Simulation result VTG sweep simulation, EGR, AF and FS as a function of VTG-rack position*

The results from the VTG-sweep simulation show that high EGR levels are achieved at nearly closed VTG-positions with a 'external routed, internally driven, high-pressure EGR configuration'. The further the VTG is closed the more EGR is obtained. Also the further the VTG is closed the lower the AF ratio becomes. The constant AF level of 14.5 at the lowest VTG positions is maintained by cutting the fuel-supply (smoke-limiter). The FS value is a function of EGR and AF and increases with lower VTG-rack positions.
In conclusion to the result it is apparent that an unambiguous relationship exits between the levels of EGR, AF and FS and the VTG-rack position in an 'external routed, internally driven, high-pressure EGR configuration'.

As such, for stable engine operation with a 'external routed, internally driven, high-pressure EGR configuration' at specific engine speed and load a specific FS value can only be found at a specific VTG-rack position at a specific combination of EGR and AF. Changing the VTG position while maintaining engine speed and load constant will change the FS value as described above.
The VTG-control strategy can be based on EGR, AF or FS level, in order to achieve the desired level for this parameter for an 'External routed, internally driven, High-pressure EGR configuration'.

## 4.8 Simulation setup

For good comparison of the results from the different simulations used in comparing different EGR configurations (EGR-configuration simulations), a uniform case-setup is made, meaning that several parameters are kept at constant level for all simulations. A list of these constant parameters is given here. Several other parameters in the case-setup of the simulation model may be varied for model optimization or used as initial values.

The uniform case-setup is based on the ESC 13 mode test-cycle. However in order reduce the simulation duration the number of engine operating points are reduced to six (6mode test cycle, appendix 3.2), which results in six cases applied in all of the EGR-configuration simulations. The six operating points are applied to a DAF MX 390 kW engine and as such determines engine speed and load for each of the six cases. In table 4.3 the uniform case-setup is given.

| *CASE* | | *1* | *2* | *3* | *4* | *5* | *6* |
|---|---|---|---|---|---|---|---|
| *ESC* | | *A100* | *B100* | *C100* | *A50* | *C50* | *Rosi* |
| RPM (constant) | Rpm | 1212 | 1530 | 1830 | 1212 | 1830 | 1350 |
| LOAD (constant) | Nm | 2557 | 2442 | 2015 | 1279 | 1008 | 707.4 |
| | | | | | | | |
| DESIRED_AF (constant) | - | 20.3 | 20.3 | 20.3 | 20.3 | 20.3 | 20.3 |
| FS (constant) | - | 36 | 36 | 36 | 36 | 36 | 36 |
| MINIMUM_AF (constant) | - | 14.5 | 14.5 | 14.5 | 14.5 | 14.5 | 14.5 |
| SOI (constant) | ° TDC | -5 | -5 | -5 | -5 | -5 | -5 |
| TNIK (constant) | K | 313.15 | 313.15 | 313.15 | 313.15 | 313.15 | 313.15 |
| TWALL (constant) | K | 800 | 800 | 800 | 800 | 800 | 800 |
| | | | | | | | |
| INITIAL-SPEED (initial) | Rpm | 100000 | 90000 | 90000 | 90000 | 90000 | 90000 |
| INITIAL-SPEED-LP (initial) | Rpm | 50000 | 80000 | 80000 | 80000 | 80000 | 80000 |
| M_INJ (initial) | Mg/cycle | 280 | 300 | 300 | 140 | 100 | 70 |
| IN_RACK (initial) | - | 0.5 | 0.5 | 0.5 | 0.5 | 0.5 | 0.5 |
| | | | | | | | |
| SCALE-HP-COMPR (variable) | - | var. | var. | var. | var. | var. | var. |
| SCALE-HP-TURB (variable) | - | var. | var. | var. | var. | var. | var. |
| SCALE-LP-COMPR (variable) | - | var. | var. | var. | var. | var. | var. |
| SCALE-LP-TURB (variable) | - | var. | var. | var. | var. | var. | var. |
| | | | | | | | |
| TEGR | K | var. | var. | var. | var. | var. | var. |

*Table 4.3 Uniform case-setup*

In the uniform case-setup as given in table 4.3 it can be seen that engine speed and load conditions are in agreement with the six ESC operating points for the DAF MX engine (except for Rosi). Most of the operating points are at full load which is most difficult for the turbocharger(s) as can be seen in the 'Preparatory EGR and turbo study for high EGR' (4.3).
Depending on the control-strategy used, the target AF-ratio (desired_AF) or FS (FS) value can be used for adjusting the VTG-rack position as earlier described, both values are set as constant values in the case-setup for all EGR-configuration simulations. In most of the EGR-configuration simulations the AF-ratio is used as a control-parameter for adjusting VTG-rack position, the target FS value from the case-setup is then left unused. The minimum AF-ratio (minimum_AF) is used for the so-called 'smoke-limiter' and is a constant value for all EGR-configuration simulations. The start of injection (SOI) is set at a constant of 5° before TDC for all EGR-configuration simulations.

The intercooler temperature (Tnik) and the temperatures of the exhaust-manifold and hot parts of the EGR –line (Twall) are set constant for all EGR-configuration simulations.
The initial values given in the case-setup may vary somewhat between the simulations, but do not effect the final results of the simulations. The initial speed of high-pressure turbo (initial-speed) and low-pressure turbo (initial-speed-LP) vary depending on the engine operating-point and are generally set relative high in order to speed-up simulation convergence. The initial values for the mass of fuel injected (M_inj) are set based on the engine speed and load and are close to the actual required mass of fuel for the given conditions. The initial VTG-rack positions (In_rack) are generally set at 0.5 which is between minimum and maximum VTG-rack position, respectively 0.25 and 1.
The turbo scale factors (mass multiplier) for High-pressure compressor (Scale-HP-compr) and turbine (Scale-HP-turb) and Low-pressure compressor (Scale-LP-compr) and turbine (Scale-LP-turb) are used as variables in optimizing a given EGR-configuration.
The average EGR-cooler wall temperature (Tegr) is also used as an optimization variable. Sweeps are performed with the EGR-cooler wall temperature in order to determine the effects of different EGR-cooler temperatures.

Based on the simulation outcomes of the six cases it is aimed at giving a good prediction of the outcomes for a full 13 mode test for the same engine, therefore a weighed average value is calculated, combining the weighing factors as prescribed for the ESC 13mode test. A explanation of the calculation of this average can be found in appendix 3.2 *6mode test cycle*.

Some level of model optimization is required, as it is aimed to compare the best results of each EGR-configuration. Model optimization is primarily conducted by turbo-scaling. Initially simulations are performed without turbo-scaling (100%) after which turbo-scaling may be applied in an iterative way. Also adjustment of the EGR-cooler wall temperature is used as model optimization, after an optimum turbo-scale is determined. However as the model validation indicates, the GT-power engine model is not 100% accurate, model-optimization is therefore not conducted to great precision.

# 5. Simulation results

As already explained in chapter 4, in total three engine configuration are simulated in GT-power. The details of the different simulation-models are described and the way in which the simulation is conducted. For each configuration a short optimization process in conducted in order to find reliable and realistic results. Optimization is conducted by varying the turbo-scale, EGR cooler temperature, number and size of both intercooler and EGR-cooler.
The results of each of the simulations are given and discussed. However as the main objective is to determine the suitability of the different configurations for High-EGR application, the focus is on those results most important in applying EGR.
Please note that all simulation-results must be regarded as predictive and indicative as several assumptions are made particularly concerning the combustion model. The reservations made in §4.5 apply to all simulation-results. The following simulation-results only apply for comparing high-pressure and low-pressure internally driven EGR configurations.

## 5.1 Internally driven- high-pressure- single stage- VTG turbo

Details regarding the model
The 'internally driven- high-pressure-single stage- VTG turbo' simulation model is the same as the base-line engine model described in paragraph 4.4, however with only 1 turbo-charger. The single turbo-charger is the Holset HE431 VTG used as high-pressure turbo in the two-stage setup of the base-line engine model. As the flow through compressor and turbine decrease at increasing EGR levels in an internally driven- high-pressure configuration, it is estimated that the turbo may be oversized at high EGR levels. Therefore the option is used to downscale the turbo map by mass flow rate as means of optimization.

Details regarding case-setup and settings
In total two simulations where run; one with original turbo maps and no scaling applied. The second with the turbo-map "downscaled" with respect to mass-flow rate. Scaling of the turbo is performed with the help of the 'Mass multiplier' factor in GT-power, explained in paragraph 4.4. The downscaled turbo is comparable with a smaller turbo opposed to the original un-scaled turbo. The case set-up in both simulations is basically the same as in the 'Uniform case-setup' given in paragraph 4.8. The turbo-scale variables (mass multiplier) are different. In table 5.1 below, the optimization parameters and additional setting are given for both simulations.

| **Setting/ parameter** / **Simulation** | *Cases* | *VTG control parameters* | *HP turbo scale multiplier* | *LP turbo scale multiplier* | *EGR-cooler wall temp. [K]* | *EGR-cooler size* | *Second inter-cooler ?* |
|---|---|---|---|---|---|---|---|
| Simulation 1 | 6 mode | FS (0.36) | 1 | n.a. | 398.15 | Base-line | No |
| Simulation 2 | 6 mode | FS (0.36) | 0.7 | n.a. | 398.15 | Base-line | No |

*Table 5.1; Optimization parameters and settings for the different simulations*

As can be seen in table 5.1 for each simulation all six operating points of the 6mode test cycle are simulated. FS (fraction stoichiometric) is used as a control parameter for the VTG-rack position. As already indicated in simulation 1 the turbo-maps of the high-pressure (HP) turbo are used as original. In simulation 2 the turbo-maps of the high- pressure turbo are used together with a "mass multiplier" of 0.7. The low-pressure (LP) turbo is removed from the model. All other optimization parameters and settings are kept at constant, base-line, values for both simulations. No second intercooler was applied.

Results

In figure 5.1 a sample of the simulation results are given. As for these simulations 'FS' was used as a control-parameter for the VTG, the resulting average FS value is very close to its reference value (36%) in both simulations. However in both simulations the resulting average AF values are far below the desired value of 20.3 ($\lambda$=1.4). This indicates that the EGR gas replaces too much fresh air and the boost pressure needs to be increased in order to counter this effect. However the turbo-maps are already exceeded in both simulations in most cases. This means that the turbo-operating points are outside the turbo-map and the simulation is continued based on extrapolation of the turbo-map.

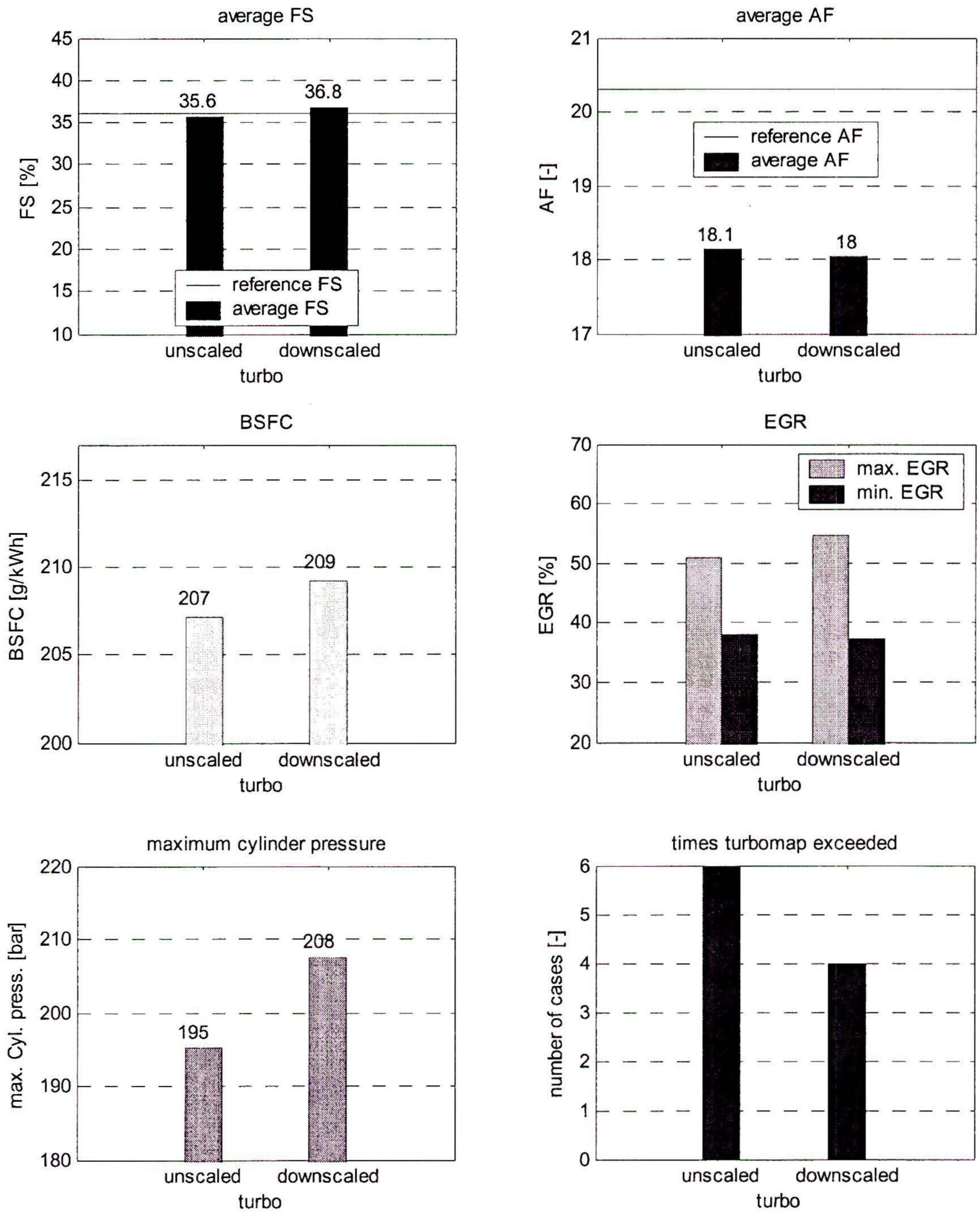



*Figure 5.1; Comparison original (un-scaled) and downscaled turbo for a internally driven- high-pressure- single stage- VTG turbo engine configuration. Simulation results; weight average over 6-mode test cycle. Turbo 'downscaled' by 30% with use of a 'Mass multiplier' factor of 0.7 in GT-power (§4.4). Simulation nr. 1-2. DAF MX 12.9L engine 390kW, Holset HE431 turbo.*

In the figures 5.2 and 5.3 the speed-maps are given together with the operating points for each of the 6 cases from the 6-mode test cycle (§ 4.8).

As can be seen from figure 5.2 compressor stall occurs at simulation 1. More precisely the compressor stalls for case 1, 4 and 6(far left, on the surge-line). Also the turbine-speed-map is exceeded in case 1,2, 3 and 5. However this is not shown as a VTG turbine does not permit to draw one single speed-map since it changes with change of rack-position. Also see appendix 4.1.

As can be seen from figure 5.3 the (scaled) compressor operating points are within the compressor map in all cases. The turbine-speed-map is exceeded in case 1,2, 3, 4 and 5, see appendix 4.1.

More simulation data can be found in the appendix 4.1 however for some cases results are achieved with turbo operating points interpolated outside the turbo-maps.

Conclusions regarding simulations with Internally driven- high-pressure- single stage- VTG turbo configuration

As was expected, it is clear that the single stage VTG turbo is not capable of delivering the desired FS values even at lower than desired AF ratio. With a down-scaling applied to a original single stage VTG turbo in terms of mass-flow, improvement is reached over the original (non-scaled) turbo when trying to achieve high EGR and high FS levels. The compressor of the turbo does not exceed its compressor-map. However the turbine does exceed its map, the maximum turbine speed is exceeded. Also the desired AF level could not be reached, on average. Further downscaling of the turbo, will cause the turbine speed to increase even further and in more cases and as such exceed the turbo-maps further.

It can be concluded that in order to reach the defined target-values (§4.2) a turbo-system is needed which is capable of delivering a higher compressor-ratio at lower mass-flow, such as two-stage turbo-charging.

Improving compressor design for delivering high-pressure ratio's at low mass-flow, would (theoretically) be possible using a (further) backward curvature compressor wheel and recirculation of compressor flow back to the inlet of the compressor. By giving the blades on the compressor wheel a backward curvature the wheel takes over a larger portion of the overall pressure rise. This increases the surge margin, as the diffuser is less likely to stall. Recirculation increases the inlet flow of the compressor, shifting the surge limit to a lower flow. However backward curvature compressor wheels require higher tip speeds at the same pressure ratio, increasing the material loading. A titanium compressor wheel is required which is costly. Recirculation of compressor flow lowers efficiency. Two-stage turbo-charging therefore is more attractive.

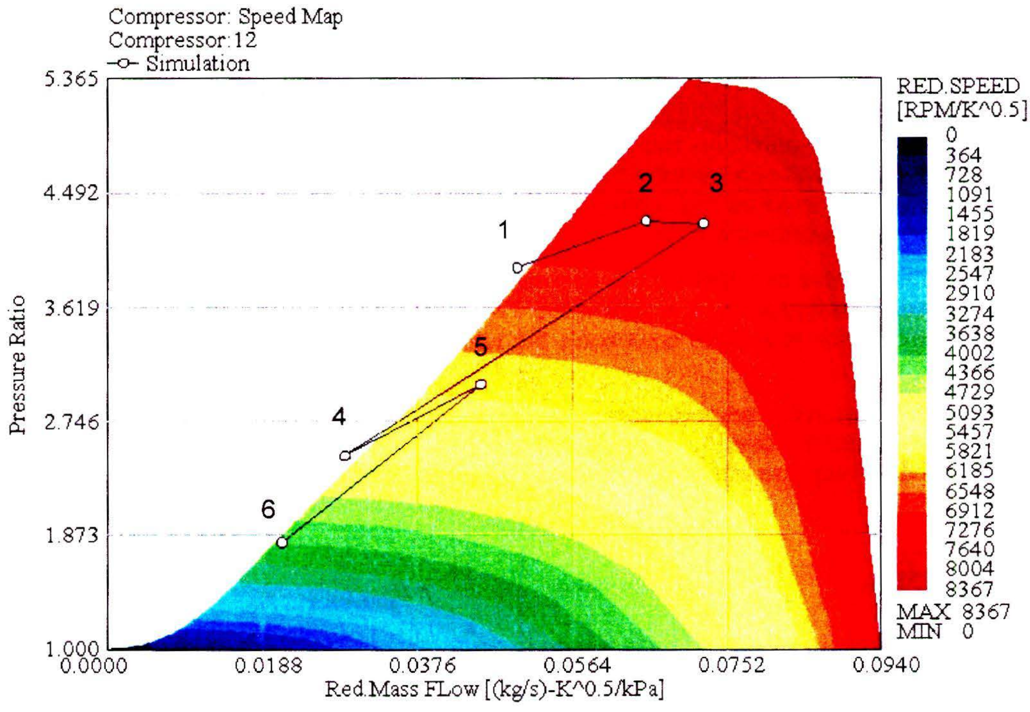



*figure 5.2; Speed-map HP-turbo compressor, internally driven- high-pressure- single stage- VTG turbo (Holset HE431), Simulation 1, cases 1 to 6.*

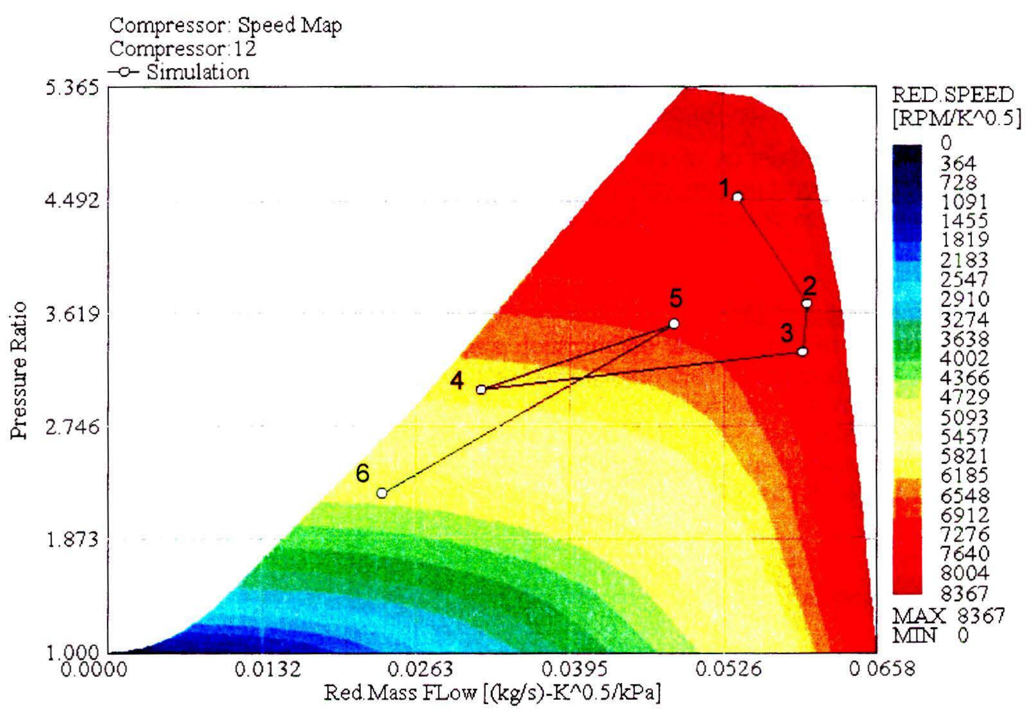



*figure 5.3; Speed-map HP-turbo compressor, internally driven- high-pressure- single stage- VTG turbo (Holset HE431). Turbo 'downscaled' by 30% with use of a 'Mass multiplier' factor of 0.7 in GT-power (§4.4). Simulation 2, cases 1 to 6.*

## 5.2 Internally driven- high-pressure- two stage- VTG turbo & fixed geometry turbo

Details regarding the model

The 'internally driven- High-pressure- two stage- VTG turbo & fixed geometry turbo' simulation model is the same as the base-line engine model described in paragraph 4.4. Turbo-maps from a Holset HE431 VTG are used as high-pressure turbo. Turbo-maps from a Garett GT 50 (fixed geometry) are used as low-pressure turbo.

As the flows through compressor and turbine decreases at increasing EGR levels in an internally driven- High-pressure configuration, it is estimated that both turbo's are oversized at high EGR levels. Therefore the option is used to downscale the turbo maps by mass flow rate as means of optimization.

Because of the use of two turbo-compressors in series, the inlet temperature of the high-pressure compressor will be raised opposed to a single turbo configuration. The increase of compressor inlet temperature has a negative effect on turbo performance. This can be explained with the help of equation 5.1:

$$W_{compressor} = \dot{m}_{air} \cdot Cp_{air} \cdot 1000 \cdot T_{entrance} \cdot \left(\tau_{compressor}^{\kappa-1/\kappa} - 1\right) \cdot 1/\eta_{compressor} \qquad \text{eq. 5.1}$$

As can be seen in equation 5.1 above, an increase in compressor entrance temperature ($T_{entrance}$) as a consequence of hot EGR-gas supplied to the compressor, will cause the required compressor power ($W_{compressor}$) to increase or the achievable pressure ratio ($\tau_{compressor}$) and mass-flow ($m_{air}$) to decrease. Therefore a second intercooler in-between both compressors, reducing the LP-compressor entrance temperature, may be desirable. A second intercooler is integrated in some of the simulations for 'Internally driven- High-pressure- two stage- VTG turbo & fixed geometry turbo'. The second intercooler is modelled between the low-pressure compressor outlet and the high-pressure compressor inlet and is of the same size as the single intercooler in the base-line engine model. The wall-temperature of the second intercooler is set at an estimated 300K, which is lower than the main (base-line) intercooler (313.15K).

In order to determine the influence of the EGR-cooler on the overall engine performance, the EGR-cooler size and temperature are varied over several simulations. Doubling the amount of pipes in the EGR-cooler, thus doubling the total surface, increases the EGR-cooler size.

Details regarding case-setup (1 t/m 17)

In order to optimize the model, several optimization steps are taken in which -parameters are varied and sweeps are conducted. This results in 16 simulations with different parameter setting however all simulations are conducted over the 6-mode test (appendix 3.2). An overview of the simulations is given in table 5.3. After each optimization step the most optimal setting is used in the next step. The optimization steps concern the following sweeps and variations in optimization parameters, further explained below;

- 2nd intercooler
- Turbo scale
- EGR-cooler temperature
- EGR-cooler size

*2nd intercooler*, As pointed out above under 'Details regarding the model' a second intercooler in-between the 2-stage compressors is likely to give better performance. The effect of a second intercooler is tested in simulation 3 and 4. Initially the same turbo downscaling is used as used with the single-stage simulations, which is a mass multiplier of 0.7. All other settings are kept at constant base-line value.

*Turbo scale*; Simulations 5 till 8 and simulation 4.In order to find the optimal turbo size 4 different turbo scale settings are tested, for both turbo-chargers (compressor and turbine). The scale factors (= GT-power mass multiplier) are varied from 0.5 till 0.8.

*EGR-cooler temperature;* Simulations 9 till 12 and simulation 4. A sweep in average EGR-cooler wall temperature is performed with the temperatures [313-323-333-373-398.15]. Average EGR-cooler wall temperatures below 313K (≈40°C) are considered unrealistic because off condensation effects, which need to be avoided.

*EGR-cooler size;* Simulations 13 till 17. The same sweep in average EGR-cooler wall temperature is conducted however with double the EGR-cooler surface area.

The simulation, which provides to best results, primarily highest FS level, is selected for further comparison with results from the external route internally driven low-pressure configuration.

| Setting/ parameter<br>Simulation | Cases | VTG control parameters | HP turbo scale multiplier | LP turbo scale multiplier | Tegr EGR-cooler wall temp. [K] | EGR-cooler size | Second inter-cooler? |
|---|---|---|---|---|---|---|---|
| *2nd intercooler* | | | | | | | |
| Simulation 3 | 6 mode | AF (20.3) | 0.7 | 0.7 | 398.15 | Base-line | No |
| Simulation 4 | 6 mode | AF (20.3) | 0.7 | 0.7 | 398.15 | Base-line | yes |
| *turbo scale* | | | | | | | |
| Simulation 5 | 6 mode | AF (20.3) | 0.5 | 0.5 | 398.15 | Base-line | yes |
| Simulation 6 | 6 mode | AF (20.3) | 0.6 | 0.6 | 398.15 | Base-line | yes |
| Simulation 4 | 6 mode | AF (20.3) | 0.7 | 0.7 | 398.15 | Base-line | yes |
| Simulation 7 | 6 mode | AF (20.3) | 0.7 | 0.8 | 398.15 | Base-line | yes |
| Simulation 8 | 6 mode | AF (20.3) | 0.8 | 0.8 | 398.15 | Base-line | yes |
| *EGR-cooler temperature* | | | | | | | |
| Simulation 9 | 6 mode | AF (20.3) | 0.7 | 0.7 | 313.00 | Base-line | yes |
| Simulation 10 | 6 mode | AF (20.3) | 0.7 | 0.7 | 323.00 | Base-line | yes |
| Simulation 11 | 6 mode | AF (20.3) | 0.7 | 0.7 | 333.00 | Base-line | yes |
| Simulation 12 | 6 mode | AF (20.3) | 0.7 | 0.7 | 373.00 | Base-line | yes |
| Simulation 4 | 6 mode | AF (20.3) | 0.7 | 0.7 | 398.15 | Base-line | yes |
| *EGR-cooler size* | | | | | | | |
| Simulation 13 | 6 mode | AF (20.3) | 0.7 | 0.7 | 313.00 | Double Base-line | yes |
| Simulation 14 | 6 mode | AF (20.3) | 0.7 | 0.7 | 323.00 | Double Base-line | yes |
| Simulation 15 | 6 mode | AF (20.3) | 0.7 | 0.7 | 333.00 | Double Base-line | yes |
| Simulation 16 | 6 mode | AF (20.3) | 0.7 | 0.7 | 373.00 | Double Base-line | yes |
| Simulation 17 | 6 mode | AF (20.3) | 0.7 | 0.7 | 398.15 | Double Base-line | yes |

*Table 5.3; Optimization parameters and settings for the different simulations. 'internally driven- high-pressure- two stage- VTG turbo & fixed geometry turbo'.*

Results

- *2nd intercooler*

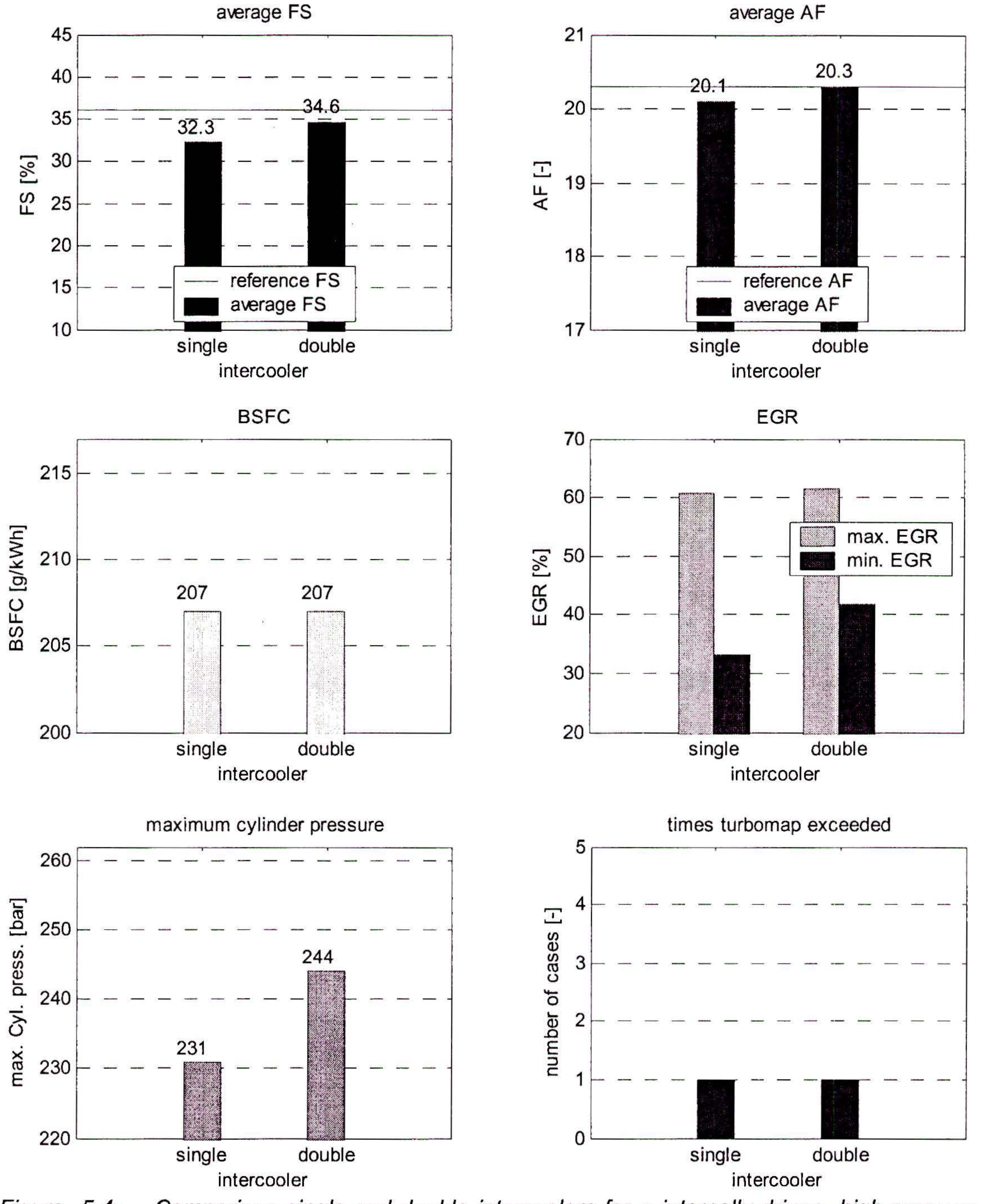



*Figure 5.4; Comparison single and double intercoolers for a internally driven- high-pressure- two stage- VTG turbo & fixed geometry turbo engine configuration. Simulation results; weight average over 6-mode test cycle. Turbo 'downscaled' by 30% with use of a 'Mass multiplier' factor of 0.7 in GT-power (§4.4). Simulation nr. 3-4. DAF MX 12.9L engine 390kW, Holset HE431 VTG & Garrett GT50 turbo.*

Applying a second intercooler improves performance in terms of AF and FS. Higher AF ratio and at the same time higher FS level are achieved. The maximum cylinder pressure increases substantially due to the increase in boost-pressure, max. 4.8 bar till max 5.0 bar, with the use of a second intercooler. Fuel consumption is unaffected. A possible explanation for this may lay in counteracting positive and negative effects on combustion and BSFC due to changes in AF ratio, boost pressure and pumping losses. The intake manifold charge temperature remains almost equal with and without the second intercooler, approximately 370K, and as such is unlikely to affect BSFC. The increased AF ratio and higher boost-pressure may have a positive effect on efficiency and thus lower BSFC. The higher AF ratio and boost-pressure result in a higher oxygen content and end of compression temperature, which may have a positive effect on combustion. For example; lower ignition delay, higher combustion rate, and higher combustion efficiency. The absolute average PMEP increases from 0.7 till 0.8 as a consequence of the second intercooler. This indicates an increase in pumping loss, which also has a negative effect on BSFC.
Unfortunately the HP-turbine exceeds maximum speed in both simulations at A100. In further simulations with two-stage turbo-charging a second intercooler is implemented in the models as standard.

*turbo-scale*

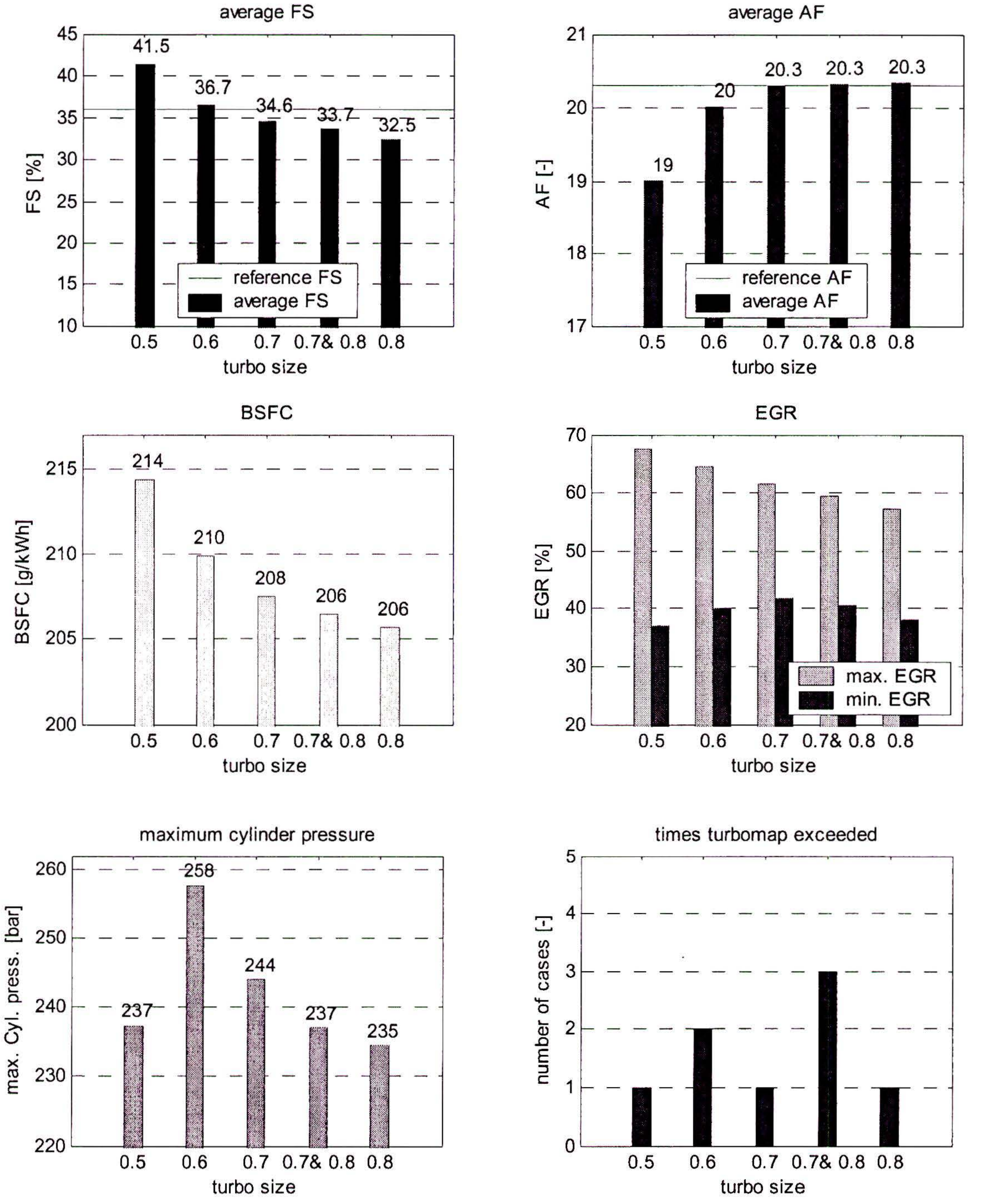



*Figure 5.5; Comparison turbo scale, high and low-pressure turbo for a internally driven- High-pressure- two stage- VTG turbo & fixed geometry turbo engine configuration. Simulation results; weight average over 6-mode test cycle. Turbo size expressed as mass multiplier factor used for turbo scaling in GT-power (§4.4). Simulation nr. 4-8. DAF MX 12.9L engine 390kW, Holset HE431 VTG & Garrett GT50 turbo.*

Trends are visible; FS, BSFC and maximum EGR levels decrease at increased turbo size. AF ratio increases at increasing turbo size. These trends can be explained by the increase of backpressure at increasingly smaller turbos. Increased backpressure increases pumping loss, indicated by an increasing in absolute PMEP from 0.7 till 1.1 bar for turbo size 0.5 and 0.8 respectively. In an 'internally driven- High-pressure' engine configuration, increased backpressure increases EGR and thus FS. Increased EGR and FS levels have a negative effect on efficiency and thus BSFC also see § 2.3. Please notice that at increasing AF levels the FS and EGR levels are decreasing. In case of large turbos (0.7 & 0.8) the AF level could be increased however at the expense of FS and EGR, which are not on target level.

Maximum cylinder pressure seems to decrease at increasing turbo size and is related to the maximum boost pressure realized. At increasing turbo size, boost-pressure decreases and thus maximum cylinder pressure decreases as well. At very small turbo-size (0.5) boost pressure is relatively low, due to relatively low turbo efficiency. At turbo-size 0.6 the highest boost-pressure was found at 5.42 bar.
Maximum cylinder pressures are very high in all cases, but are found at only single engine operating-points of the 6-mode test cycle. The maximum allowable peak cylinder pressure is considered to be 250 bars. Therefore the maximum cylinder pressure found for a turbo-size of 0.7 (244 bar) is within reason. Maximum cylinder pressure is however dependent on boost pressure. Given the reservations made in §4.5 concerning boost pressure, the predicted maximum cylinder pressure, may be unreliable and become higher in reality.
The turbo-maps are exceeded at least once in each simulation. This is caused by the HP-turbine, which exceeds its maximum speed. In case the LP turbo (both compressor an turbine) are scaled larger than the HP turbo (both compressor an turbine), the HP turbine suffers as it exceeds maximum speed in three cases (A100, B100 & C50).

For further research a turbo-size of 0.7 or 0.8 are reasonable options, since they both provide the desired AF ratio and come close to reaching the desired FS level. Furthermore BSFC and maximum cylinder pressure are relatively low. The turbo-maps are only exceeded in one case, unlike with a turbo-size of 0.7 & 0.8 (HP-LP turbo). Based on the obtained FS and AF levels from both the simulation with turbo-size of 0.7 or 0.8, a turbo-size of 0.7 is chosen for further simulations as it comes closest to the desired FS level.
However as can be seen in figure 5.6 below, the manifold temperatures for a 0.8 turbo-size are generally lower that for a 0.7 turbo-size. This lower manifold temperature may compensate for the lack of FS level found for a 0.8 turbo-size, as it may suppress NOx emissions in addition to EGR. Because of this and because of the relative low BSCF level, a turbo-size of 0.8 remains an interesting option.

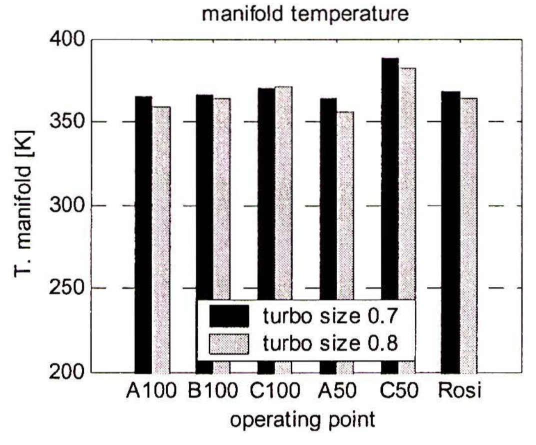



*Figure 5.6 manifold temperatures at 6-mode test for turbo size 0.7 and 0.8. Simulations 4 and 8. Turbo size expressed as mass multiplier factor used for turbo scaling in GT-power (§4.4). DAF MX 12.9L engine 390kW, Holset HE431 VTG & Garrett GT50 turbo.*

- *EGR-cooler temperature*

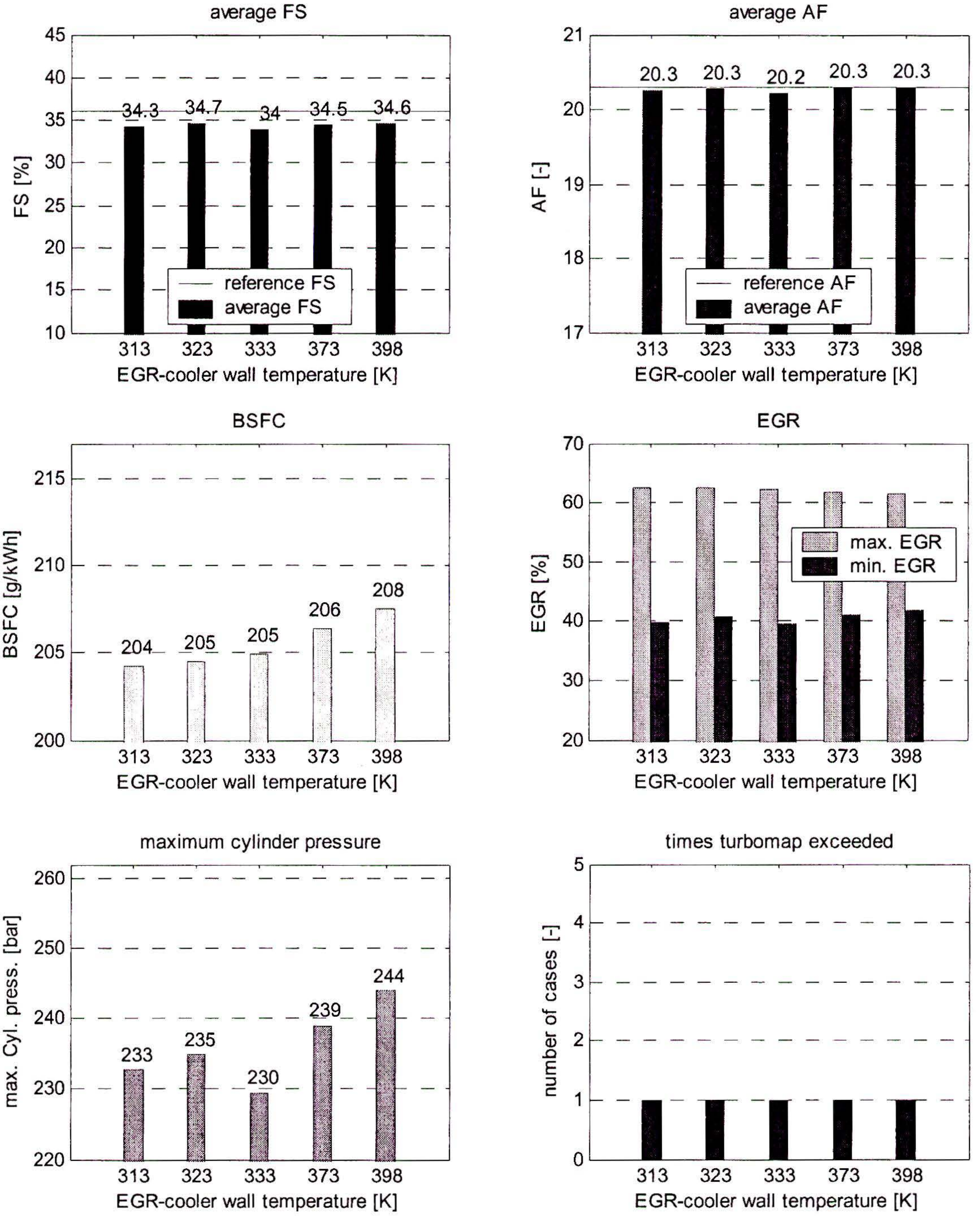



*figure 5.7; Result EGR-cooler wall temperature sweep for a internally driven- high-pressure- two stage- VTG turbo & fixed geometry turbo engine configuration. Simulation results; weight average over 6-mode test cycle. Simulation nr. 9-12 + 4. DAF MX 12.9L engine 390kW, Holset HE431 VTG & Garrett GT50 turbo with mass multiplier of 0.7 applied.*

As can be seen in figure 5.7 the engine performance is slightly affected by the change in EGR-cooler temperature. The desired FS level of 36 is not met in any case. However fuel consumption decreases at lower EGR-cooler temperatures. The higher density of the EGR-gas may explain this. Higher density EGR-gas reduces pumping loss, as less exhaust backpressure is required by the VTG in order to supply the same mass percentage EGR. This is confirmed by the decrease in absolute PMEP from an average of 0.8 bar at a EGR-cooler wall temperature of 398.15K, to an average of 0.72 bar at an EGR-cooler wall temperature of 313K. Less pumping work has a positive effect BSFC.
Maximum cylinder pressure seems to decrease at decreasing EGR-cooler temperature. In all cases the turbo-maps where exceeded, at the A100 engine operating point the High-pressure turbine exceeds its maximum turbine speed. The high VTG turbine speed is created by the VTG-rack position, which is set in order to reach the target AF-ratio. Further "opening" of the VTG-rack-position (rack-position towards 1), would most likely reduce turbo-speed and increase the AF ratio, however with a negative effect on the EGR-level as less exhaust-backpressure is created. As a consequence the FS level would decrease.
The simulation software is capable of finishing the calculation based on extrapolation of the turbo-maps. In this case, this results in overall good performance in terms of FS, AF and Fuel consumption. Also the maximum cylinder pressures, all found at the A100 engine operating point, are based on calculations with extrapolated turbo-maps. The found results are only valid in case turbo-charger performance can be increased up too the extend to which the turbo-maps are extrapolated. With the current turbo-charger set-up the found results are thus unreliable.
As turbo-limitations are consequently met at the A100 engine operating point, reducing the target FS level for this operating point, may result in less performance in terms of FS but without exceeding the turbo-limitations.

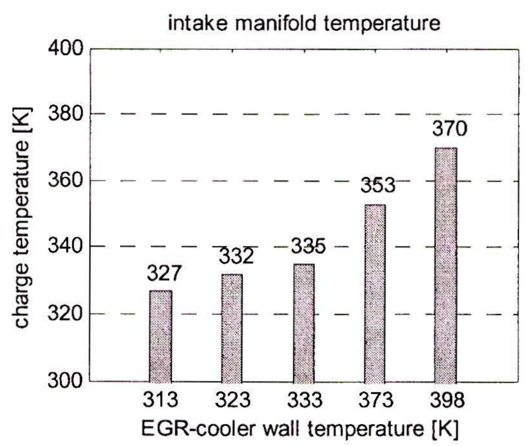



*figure 5.8; Intake manifold charge temperature at EGR-cooler wall temperature sweep. For an internally driven- high-pressure- two stage- VTG turbo & fixed geometry turbo engine configuration. Simulation results; average over 6-mode test cycle. Simulation nr. 9-12 + 4. DAF MX 12.9L engine 390kW, Holset HE431 VTG & Garrett GT50 turbo with mass multiplier of 0.7 applied.*

As can be seen in figure 5.8 the intake manifold temperature decreases at decreasing EGR-cooler wall temperature. The temperature of the EGR gas after the EGR-cooler is of direct influence on the manifold charge temperature. A lower manifold charge temperature is beneficial for suppressing NOx emissions in addition to EGR.

- *EGR-cooler size*

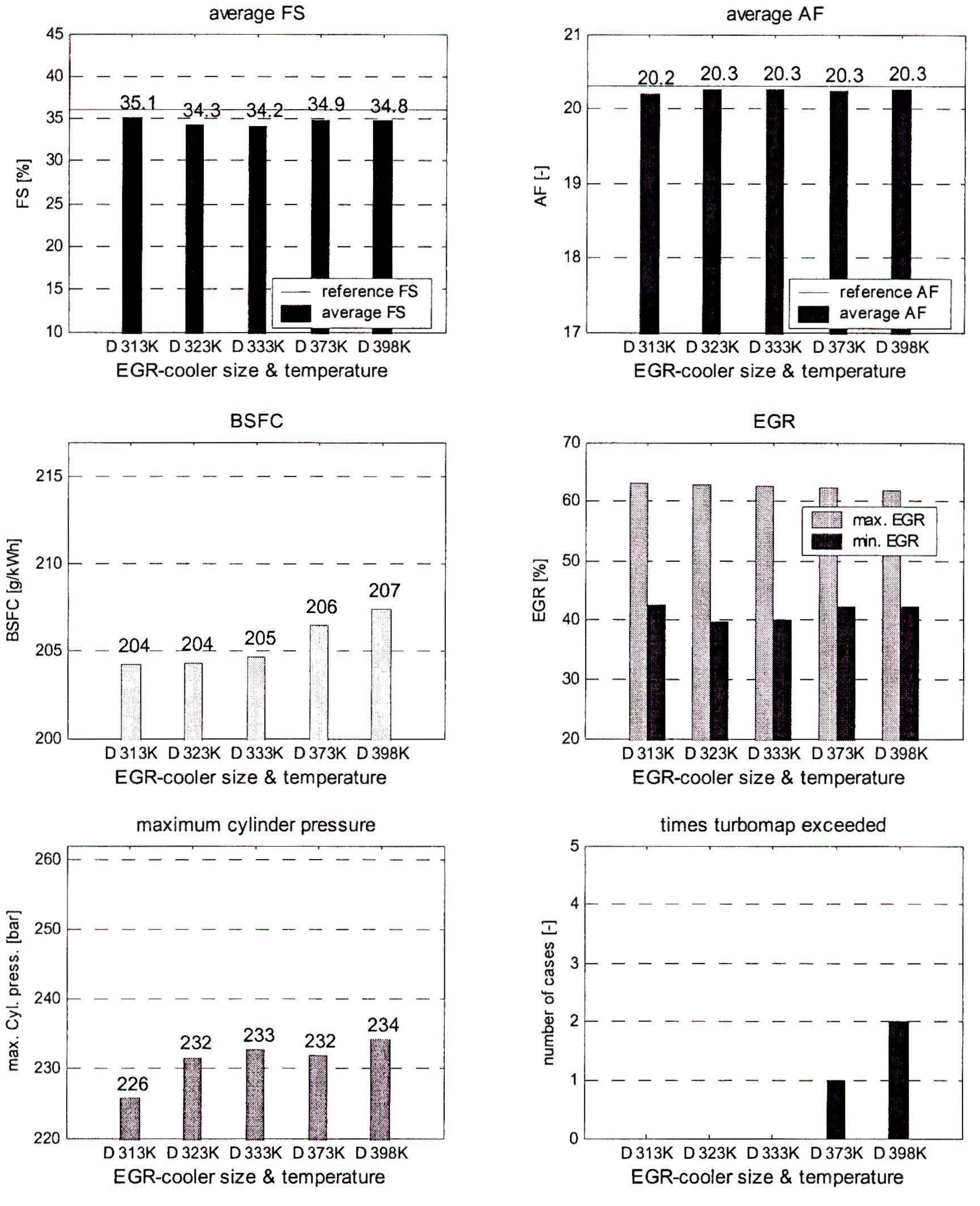



*figure 5.9; Result EGR-cooler wall temperature sweep with double the standard EGR-cooler size for a internally driven- high-pressure- two stage- VTG turbo & fixed geometry turbo engine configuration. Simulation results; weight average over 6-mode test cycle. Simulation nr. 13-17. DAF MX 12.9L engine 390kW, Holset HE431 VTG & Garrett GT50 turbo with mass multiplier of 0.7 applied. D = double the standard EGR-cooler size.*

As can be seen in figure 5.9 doubling the EGR cooler size has little influence on the engine performance opposed to the application of the base-line EGR cooler size. Fuel consumptions decreases slightly with a larger EGR cooler size opposed to a single EGR-cooler. An explanation for this maybe found in the reduced flow resistance as a consequence of double the size EGR-coolers. The lower flow resistance reduces pumping loss. This is confirmed by the absolute PMEP, which is approximately 0.02 bar less then for a single EGR-cooler together with a more 'open' position of the VTG.
However the desired FS level of 36 is not met in any case. Also the maximum cylinder pressure decreases with the use of a larger EGR cooler. Most importantly the turbocharger does not exceed its turbo-maps at low EGR-cooler temperatures (313, 323, 333 K) in combination with the double EGR-cooler size. Cooling the EGR-gas increases its density; this reduces the need for high boost pressures in order to achieve both a high AF-ratio and a high EGR level. A lower required boost-pressure allows the VTG to operate with a more "open" rack-position and still achieving the targeted AF–ratio. A more "open" rack-position of the VTG results in a lower turbo-speed and thus over-speed of the HP turbine is prevented. Also an increase in EGR-cooler size reduces the pressure drop over the EGR-line, reducing exhaust backpressure and thus turbine speed.
At the lowest EGR-cooler temperature (313K) the desired AF level cannot be reached. This is explained by the severe cooling of the EGR-gas in this case. Cooling the EGR-gas increases its density and allows increased flow of EGR-gas. Also increased EGR-cooler size reduces the total pressure loss over the (total) EGR cooler, which may lead to an increase in flow. An average EGR-cooler wall temperature of 313K and double the standard EGR-cooler size leads to excess EGR-flow at the C100 engine operating point, causing the AF-ratio to drop below the desired level of 20.3. The VTG control over the AF-ratio has reached its limits, as the VTG-rack position is fully open, resulting in minimum exhaust-backpressure. The EGR flow could be limited by means of the EGR-valves in order to maintain the desired AF ratio of 20.3. However applying the EGR-valve has a detrimental effect on fuel-economy.
The intake manifold charge temperature decreases with the use of double the size EGR-cooler opposed to a single size EGR-cooler. As such intake manifold charge temperatures are 322K and 326K at an average EGR-cooler wall temperature of 313K and 323K respectively. A lower manifold charge temperature is beneficial in lowering NOx emissions. Although a EGR-cooler wall temperature of 313K is expected to provided the lowest NOx emission based on the highest FS level and the lowest intake charge temperature, the AF ratio is below the target level. Therefore considering performance in terms of FS- and AF levels and Fuel economy as well as turbo-limitations, the simulation with an average EGR-cooler wall temperature of 323 K (simulation 14) and with double the standard size EGR- and inter-cooler is considered the best configuration for the External routed- Internally driven- High-pressure EGR configuration. The results from this simulation case will be used in further comparison with results from the External route-internally driven- Low-pressure EGR configuration simulations.

Explanation regarding internally driven high-pressure EGR simulation final results

The results of the internally driven high-pressure EGR configuration simulations are unsatisfactory, as they do not meet the target values. The lack of performance in terms of FS and EGR is largely due to turbo and limitations. In order to achieve the targeted FS level at minimum AF-ratio, a relative high boost-pressure is required. However the total flow over the turbines is reduced as a consequence of High-EGR, reducing available turbo-power. Additional cooling of the charge reduces the need for high boost-pressure as the density of the charge is increased.
With the current turbo-charger configuration it is not found possible to reach sufficient boost-pressure. Downsizing the turbo results in a reduction of the AF-ratio as the smaller turbine-housings will create more exhaust-backpressure, increasing the EGR-level at the expense of fresh-air, hence AF. The possibility of turbo over-speed most likely increases. A larger turbo size reduces the EGR level and thus FS as less exhaust backpressure is obtained. Furthermore, the VTG-rack position would have to be "closed" more of the time in order to create backpressure, this may lead to poor turbo efficiency. Also see results obtained by turbo-scaling in this paragraph. As is also indicated in the 'Preparatory EGR and turbo study' (appendix 2.2); a turbo-

charger system with high turbo efficiency is beneficial in achieving high boost-pressures at relative low flow.
Increasing EGR-cooling helps in reducing the need of high boost-pressure and is likely to result in higher EGR and thus FS levels. However as is found; increased EGR-cooling results in excess EGR at some operating points, reducing the AF-ratio. The EGR-valves should than be used in order to limit the amount EGR flow, with a penalty in BSFC.

## 5.3 Internally driven- low-pressure- two stage- VTG turbo & fixed geometry turbo

Details regarding the model

The 'internally driven-Low-pressure-two stage- VTG turbo & fixed geometry turbo' simulation model is basically the same as the base-line engine model described in paragraph 4.4 except for the EGR-, cooling- and control- systems. Each of these systems is described in more detail.

Preliminary to the construction of the final 'internally driven-Low-pressure-two stage- VTG turbo & fixed geometry turbo' simulation model, some early simulations with a more general internally driven-Low-pressure-EGR model are conducted for a single full-load engine operating point in order to gain some insight on the characteristics of the internally driven Low-pressure EGR configuration. The exact details and results are not given here, as they are merely explorative. Some results however are used for the purpose of model construction and simulation case-setup.

Early simulations with an internally driven-Low-pressure-EGR model indicated that turbo-performance suffered severely from the hot EGR gas fed through the compressor and engine efficiency was low.

$$W_{compressor} = \dot{m}_{air} \cdot Cp_{air} \cdot 1000 \cdot T_{entrance} \cdot \left(\tau_{compressor}{}^{\kappa-1/\kappa} - 1\right) \cdot 1/\eta_{compressor} \qquad \text{eq. 5.2}$$

As can be seen in equation 5.2 above, an increase in compressor entrance temperature ($T_{entrance}$) as a consequence of hot EGR-gas supplied to the compressor, will cause the required compressor power ($W_{compressor}$) to increase or the achievable pressure ratio ($\tau_{compressor}$) and mass-flow ($m_{air}$) to decrease. Therefore the temperature of the EGR-cooler, pipe-work and components are chosen relatively low in order to reduce loss of turbo performance. The EGR-cooler is chosen extremely big for sufficient cooling of the EGR-gas. Further, the EGR-line and its components have large inner diameters in order to reduce friction losses and thereby reduce efficiency losses. Because of these extreme design choices, this model can be considered as an "extreme case" in which the potential and design-consequences of an internally driven-Low-pressure EGR-system are accessed. It is not aimed at optimizing the design of the several components for packing and cost criteria.

- *EGR*

The EGR-line is modeled external of the engine block and runs from the LP (Low-pressure)-turbine exit to the LP-compressor entrance.

The EGR line is described from start to finish; Downstream of the LP-turbine a T-split, splits the exhaust pipe in the EGR-line and the final exhaust-system. An EGR-valve is implemented in order to control the EGR-flow rate. In addition a exhaust back pressure valve is implemented downstream in the exhaust system. The EGR line features an EGR-cooler, which is described in further detail under 'cooling'. No venturi-flowmeter is implemented in order to avoid friction loss due to this device. All piping and components in the EGR-line are at least 80 mm in diameter, in order to minimize friction loss. As the exit of the EGR-line, at the LP-compressor entrance, is fixed at 80 mm in diameter it is not sensible to make the overall EGR-pipe diameter any larger in order to reduce friction losses even further.

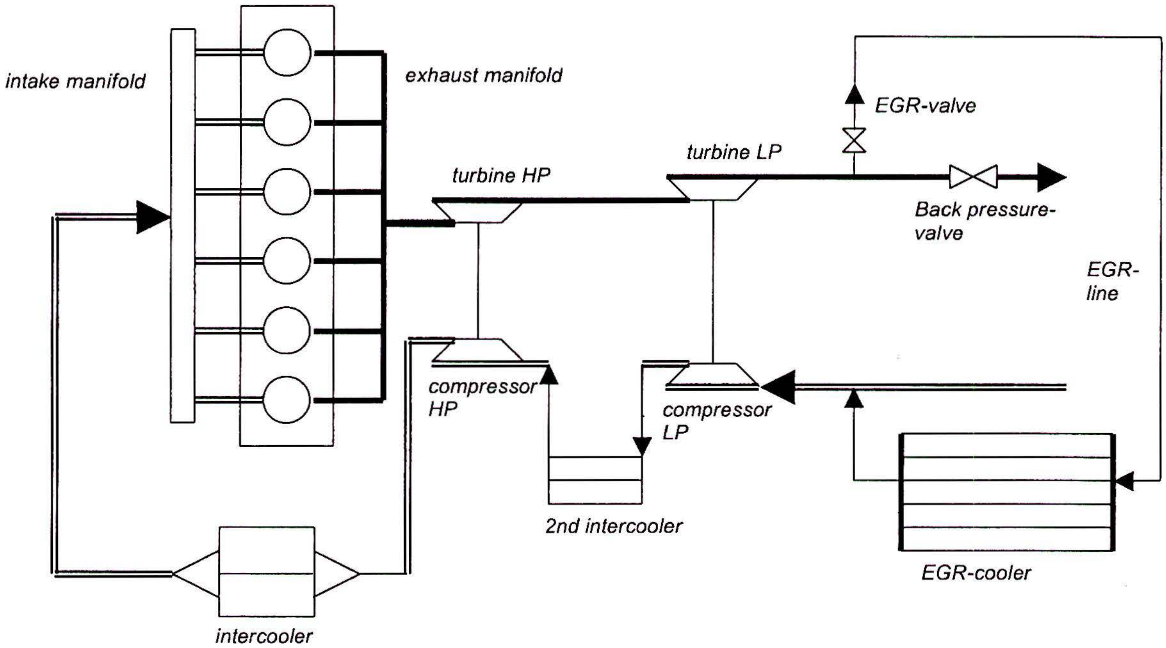



*figure 5.10; Lay-out of external route, internally driven, low-pressure EGR simulation model*

- *Cooling*

The EGR-cooler is modelled extremely big; five times the size of the standard EGR-cooler as supplied by DAF. This means that the number of identical pipes in the model is multiplied by five and thus the total surface area of the EGR-cooler is five times bigger than standard. It is estimated that extreme size EGR-coolers are required for cooling the EGR-gas in order to maintain turbo performance. The temperature of the EGR-cooler is varied. Initially an extremely low EGR-cooler wall temperature is used of 293K, about ambient. Condensation of EGR species is ignored here. However it is estimated that condensation problems are largely avoided at EGR-gas temperature above 313K, therefore in further optimization the average EGR-cooler wall temperature is not set lower than 313K keeping simulations realistic.
A second intercooler is implemented in-between both compressors as it increases (HP) turbo-performance by lowering the HP-compressor entrance temperature (paragraph 5.2) in the same way as with the simulations of the High-pressure engine configuration. The second intercooler has a constant wall temperature of 300K, again the same as use for previous (High-pressure) simulations. From early simulations with an internally driven-low-pressure-EGR model it was concluded that the intercoolers cause most of the flow restriction in the engine intake system. As the total mass-flow through both intercoolers increases opposed to a non-EGR or internally driven high-pressure EGR configuration, the flow resistance over the intercoolers needs to be decreased in order to prevent excessive pressure loss. As 50% EGR is the target EGR level the flow over the intercoolers is doubled, it is estimated that the flow resistance should therefore be halved. The number of identical pipes of the standard intercooler is therefore doubled and the length of the pipes halved. This results in more pipes being stacked in parallel with a shorter length and thus lower flow resistance. This is illustrated below in figure 5.11. The total cooling surface area of both intercoolers remains the same as for the standard intercooler.

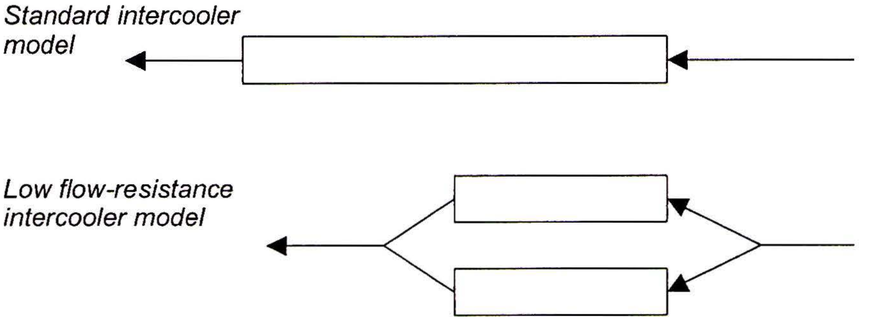



*figure 5.11 illustration of Intercooler model with low flow resistance for Internally driven Low-pressure EGR engine configuration.*

- *Control*

In figure 5.12 the relationships between VTG rack position and EGR, AF and FS are given for an 'internally driven low-pressure' EGR-configuration at a full load engine operating point, as derived from preliminary simulations. As can be seen the EGR level remains fairly constant over the sequenced VTG-rack positions. The AF and subsequently the FS levels do vary over the sequenced VTG-rack positions with an optimum in AF at VTG-rack position 0.5. However the relationships shown in figure 5.12 are found at only one engine operating point and relationships may vary for different engine operating points. The optimum VTG position needs to be determined by executing a VTG-rack position sweep at each engine operating point. However as can be seen in figure 5.13 best fuel economy and efficiency is typically found at a VTG-rack position of 1, meaning a fully open VTG giving the least exhaust backpressure.

In contrast to an 'internally driven-High-pressure EGR configuration' the VTG-rack position has little to no influence on the EGR flow rate in a internally driven-Low-pressure, because EGR-gas is separated from the exhaust-gas flow downstream of the turbine(s). Therefore the VTG-rack position cannot be used as a means of controlling the EGR flow.
Instead of controlling EGR flow with the VTG-turbo, the EGR-flow is controlled making use of the EGR valve and the exhaust backpressure valve, allowing more or less EGR-flow. The exhaust-backpressure valve is implemented in the exhaust-system downstream of the EGR-line. A PI controller is implemented in order to adjust both valve positions based on a control parameter. AF (Air Fuel ratio) is used as a control parameter in all cases, since this is one of the most important target values. If the AF level falls below the target level (20.3), less EGR is wanted and so the EGR-valve closed. If the AF is above the target level more EGR can be allowed and the EGR-valve is opened an eventually the Exhaust-backpressure valve is closed. In the controller-mechanism both valve positions are linked in such a way that the exhaust backpressure valve is only allowed to close for increased EGR flow, if the EGR-valve is already fully open.

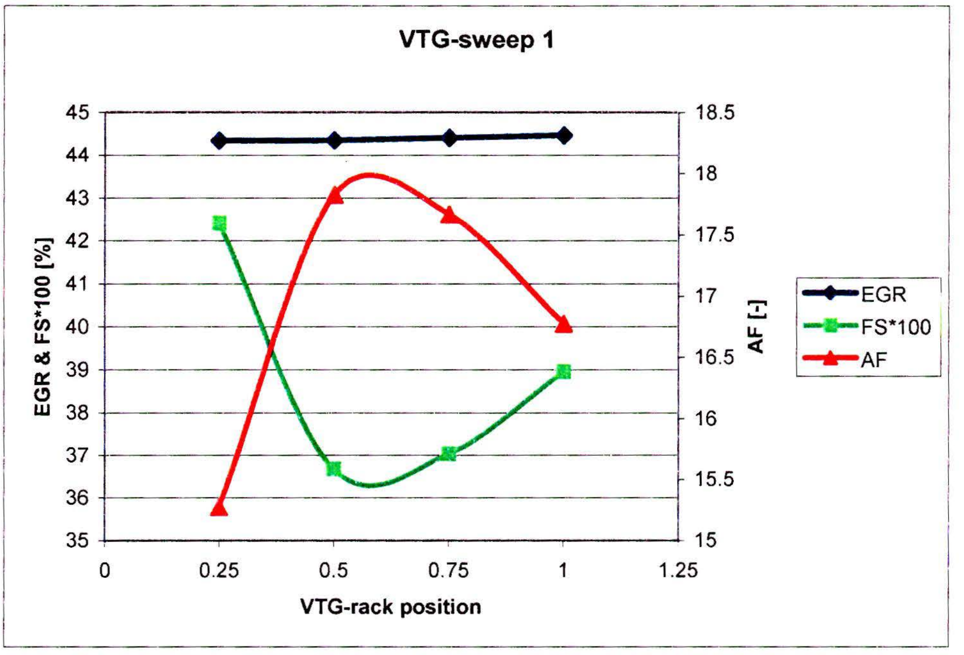



*figure 5.12* *Typical relationship between EGR, AF, FS and the VTG-rack position for a 'internally driven Low-pressure' EGR-configuration. Engine operating point; B100, turbo size 1.*

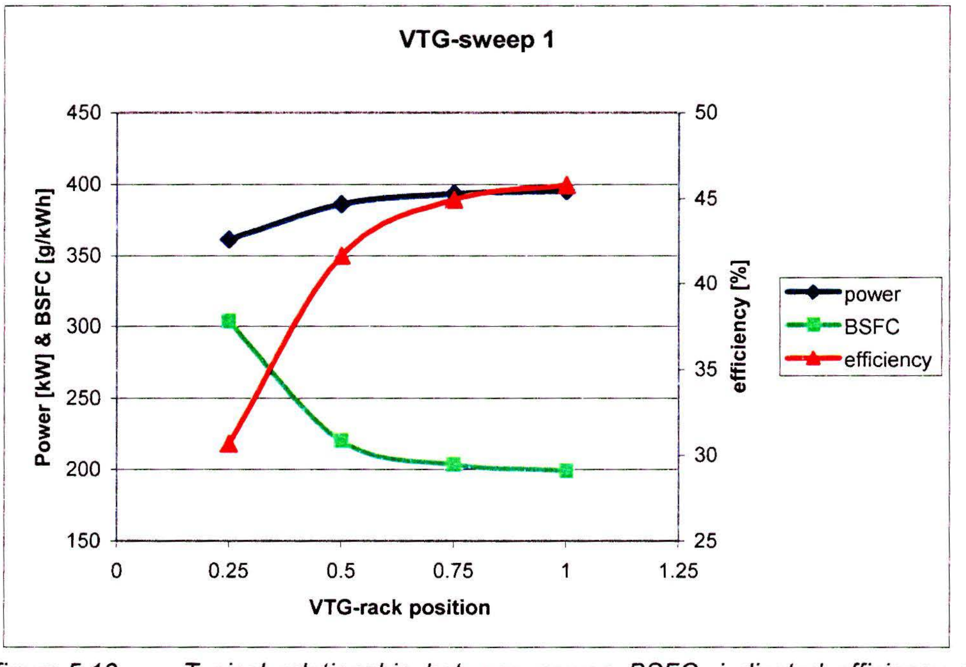



*figure 5.13* *Typical relationship between power, BSFC, indicated efficiency and the VTG-rack position for a 'internally driven Low-pressure' EGR-configuration. Engine operating point; B100, turbo size 1.*

Details regarding case-setup

In order to gain some level of optimization, several optimization steps are performed. In each optimization step parameters are varied. This results in 6 simulations for the external route internally driven low-pressure configurations. An overview of the conducted simulations and the parameter setting is given in table 5.4. After each optimization step the most optimal setting is used in the next step. The optimization steps concern to following sweeps and variations in optimization parameters, further explained below;

- Turbo scale
- VTG rack-sweep
- EGR-cooler temperature

*Turbo scale;* In simulation 14 a sweep is conducted in turbo size, both low-pressure and high-pressure turbo, for one (full load) engine operating point (B100), in order to find an optimal turbo-size. In order to eliminate the influence of the VTG-rack position, this parameter is kept constant at 1, which means the VTG creates the least exhaust backpressure. Both the EGR-valve and exhaust backpressure valve are fully open.

*VTG rack-sweep & EGR-cooler temperature;* The optimization steps 'VTG rack-sweep' and 'EGR-cooler temperature'. The simulations 19 till 23 are simulations over the full 6mode-test with a different average EGR cooler temperature for each simulation.
For the simulations 19 till 21 a VTG-rack position sweep is conducted in order to find an optimal setting for the VTG for each engine-operating point. For the sake of simulation duration only four VTG-rack positions are tested, these are [0.25-0.5-0.75-1.0]. During simulation the EGR-valve and exhaust-backpressure valve are controlled by PI-controller in order to maintain minimum AF ratio as described above. From the simulation results the cases with the best performance in terms of FS are retrieved as the most optimal cases, with the best accompanying VTG setting, for each 6mode test-point. Over these 6 'optimal cases' for the 6mode-test, average values are calculated.
It is found that for the simulation 19 till 21 a VTG-rack position of 0.75 gives the most optimal performance in terms of FS for all 6 operating points of the 6mode-test, except for the A50 operating point. At A50 a VTG-rack-position of 0.5 is found to give the highest FS level. As a consequence of these findings the VTG-rack positions are set at 0.75 and 0.5 (A50) as fixed for the simulations 22 and 23.

The simulation, which provides to best results, primarily highest FS level, is selected for further comparison with results from the external route internally driven high-pressure configuration.

| **Setting/ parameter** / **Simulation** | *Cases* | *VTG control parameters* | *HP turbo scale multiplier* | *LP turbo scale multiplier* | *Tegr: EGR cooler temp. [K]* | *EGR-cooler size* | *Second inter-cooler ?* | *EGR & exhaust valve control* |
|---|---|---|---|---|---|---|---|---|
| *Turbo scale* | | | | | | | | |
| Simulation 18 | B100 | Fixed rack pos; 1 = open | Sweep; [0.7-0.8-0.9-1-1.25] | Sweep; [0.7-0.8-0.9-1-1.25] | 293 | 5 * Base-line | yes | Fully open |
| *VTG rack-sweep & EGR-cooler temperature* | | | | | | | | |
| Simulation 19 | 6 mode | Sweep; [0.25-0.5-0.75-1.0] | 1 | 1 | 313 | 5 * Base-line | yes | AF (20.3) |
| Simulation 20 | 6 mode | Sweep; [0.25-0.5-0.75-1.0] | 1 | 1 | 323 | 5 * Base-line | yes | AF (20.3) |
| Simulation 21 | 6 mode | Sweep; [0.25-0.5-0.75-1.0] | 1 | 1 | 333 | 5 * Base-line | yes | AF (20.3) |
| Simulation 22 | 6 mode | Fixed rack pos; 0.75 0.5 @A50 | 1 | 1 | 373 | 5 * Base-line | yes | AF (20.3) |
| Simulation 23 | 6 mode | Fixed rack pos; 0.75 0.5 @A50 | 1 | 1 | 398 | 5 * Base-line | yes | AF (20.3) |

*Table 5.4; Optimization parameters and settings for the different simulations. 'Internally driven-Low-pressure-two stage- VTG turbo & fixed geometry turbo'*

Results

- *Turbo-scale*

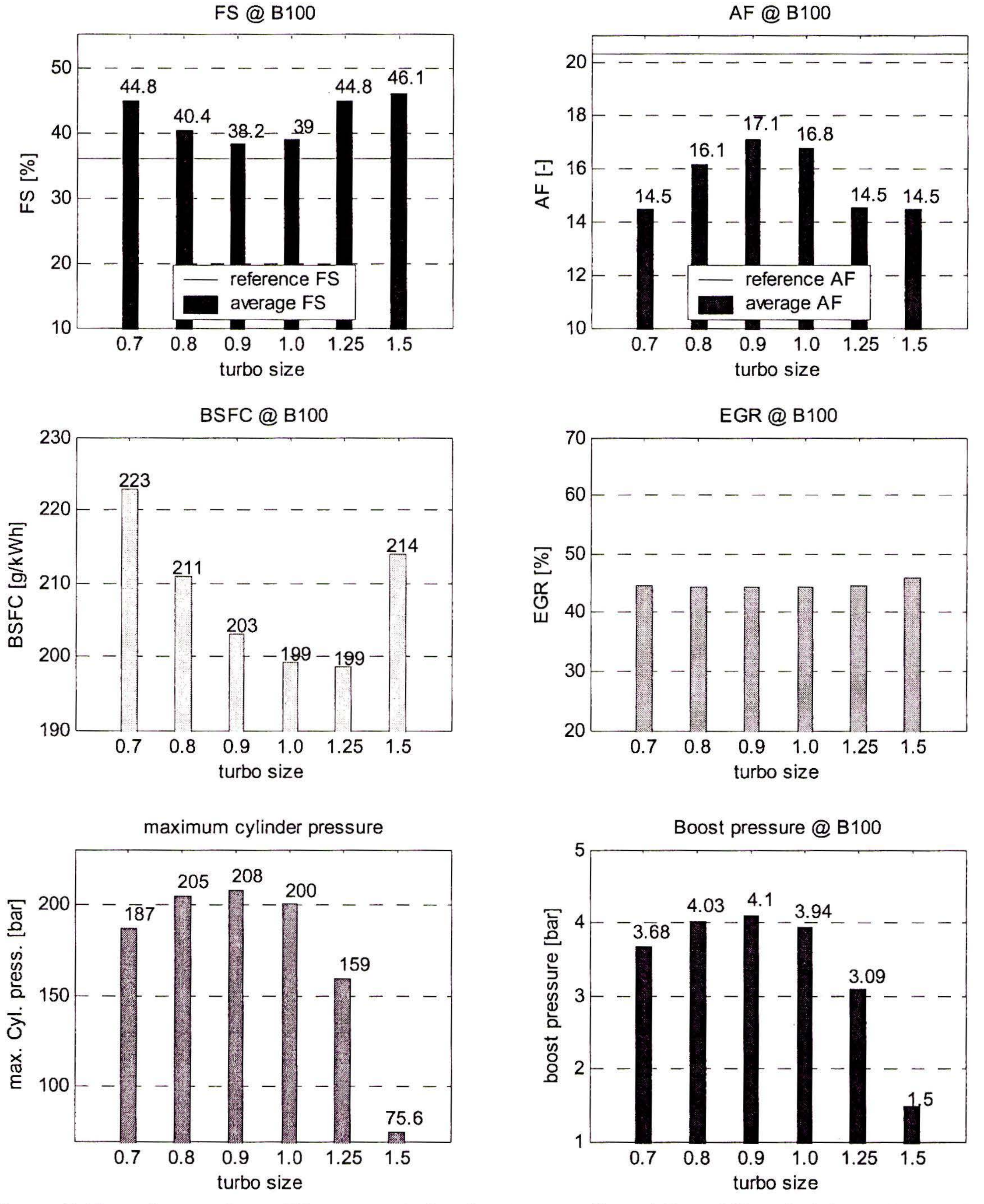



*Figure 5.13; Comparison different scaled turbochargers (both LP as HP turbo) for a, internally driven- low-pressure- two stage- VTG turbo & fixed geometry turbo engine configuration. Simulation results; single 6-mode point B100. Turbo scale sweep [0.7-0.8-0.9-1.0-1.25-1.5] @ average EGR-cooler temperature of 293K, 5*standard EGR cooler size, fully open EGR- and Back-pressure-valve. Engine: DAF MX 12.9L, 390kW, Holset HE431 VTG & Garrett GT50 turbo. Simulation nr. 18.*

Figure 5.13 shows the influence of turbo-size on engine performance at a full load engine operating point at fully open VTG-rack position (least backpressure). It can be seen that the desired FS level of 36 is reached in all cases however with far too low AF levels. The low AF is partly due the fully open position of the EGR valve allowing a relative high EGR flow, which replaces fresh air. For small and large turbo-size, 0.7, 1.25 and 1.5 respectively, the AF ratio is 14.5 and the smoke limiter is activated. The smoke limiter is described in paragraph 4.8, when activated it cuts the fuel supply in order to avoid AF ratios below 14.5, subsequently lowering power. Turbo-sizes 0.9 and 1.0 produce FS and AF levels which are closest to the target values. Also BSFC and Indicated efficiency are best for turbo-size 0.9 and 1. A turbo-size of 1.25 gives the lowest BSFC, however at very low AF level, due to a too low boost pressure. As can be seen the turbo-size has little to no influence on the EGR level. Please note that both the EGR-valve and the exhaust-backpressure valve where fully open in all cases. Overall it can be concluded that even with the low EGR-cooler temperature and large size EGR-cooler, performance in terms of AF is poor, at least for this particular engine operating point (B100). High boost pressures are needed in order to supply sufficient fresh air and EGR-gas to the engine for meeting the target AF and FS values.

A turbo-size of 1 (100%), results in relative good FS and AF levels and provides good BSFC and the best indicated efficiency. Therefore turbo-size of 1 for both HP- and LP -turbo, is chosen in following simulations. This means that the turbo-maps are used without scaling.

- *VTG rack-sweep*

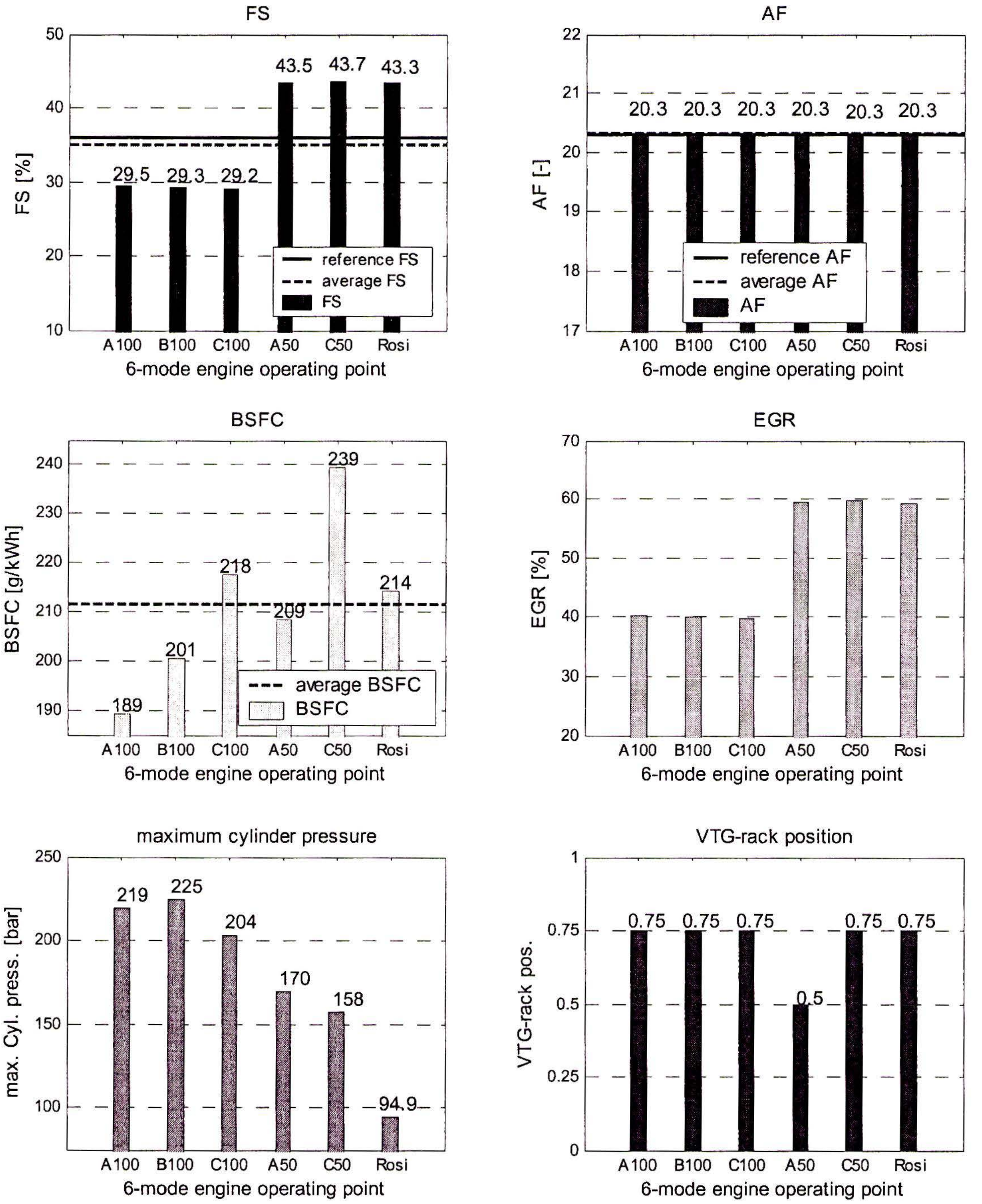



*Figure 5.14; Results at optimal VTG-rack positions for a, internally driven- low-pressure- two stage- VTG turbo & fixed geometry turbo engine configuration. Simulation results: 6-mode test cycle. VTG-rack positions tested: [0.25-0.5-0.75-1.0], @ EGR-cooler temperature of 313K. EGR valve and backpressure valve position optimized for AF of 20.3. Engine: DAF MX 12.9L, 390kW, Holset HE431 VTG & Garrett GT50 turbo with mass multiplier of 0.7 applied. Simulation nr. 19.*

In figure 5.14 the best results of the VTG-rack-position sweep from simulation 19 are given (internally driven- low-pressure- two stage- VTG turbo & fixed geometry turbo). The main criteria in selecting the best results are AF at least 20.3, FS as high as possible and turbo operation within the turbo-maps (no stall or over-speed). As can be seen in figure 5.14 for most operating points of the 6mode test a VTG rack-position of 0.75 is found to give the best results. In most operating points a VTG-rack position of 0.5 or 0.25 result in turbo-stall or over-speed (outside turbo-maps) of the HP-turbo. These positions can therefore not be used at steady state operating points and results are ignored. It needs to be said that at a VTG-rack position of 1, BSFC is generally lower as already indicated in figure 5.13, however at the expense of AF levels, also indicated in figure 5.12. The same optimal VTG-rack positions are found for the simulations 20 and 21.

As can also be seen in figure 5.14 that the AF target level of 20.3 is met in all operating points, this because the AF level is used as control parameter in controlling the EGR- and backpressure-valve. However as a consequence of maintaining this minimum AF level the FS target level cannot be reached at full load operating points (A100, B100, C100). This means that at full load it is not possible to increase EGR up to the required level without degradation of the AF level. The EGR flow is restricted by means of the EGR-valve. At above given full-load conditions the EGR-valve is slightly closed limiting EGR flow. The Exhaust-backpressure valve is left fully open at all full-load operating points (in contrast to the part-load points). At part load (A50, C50 & Rosi) more EGR can be allowed while meeting the target AF level. The target FS level (reference) can be met at part load conditions. The average FS level over the 6 test points is however is 34.9%, which is below the target FS level of 36%.

BSFC is relatively high, particularly at the C50 point. In all three part-load operating points (A50, C50 & Rosi), the EGR-valve is fully open and the exhaust-backpressure valve is activated in order reach the desired EGR level. On average the backpressure valve is closed at 40° for the three part-load operating points (90° being fully open and 0° being fully closed). Increasing exhaust-backpressure is known to decrease engine efficiency and increase BSFC. The average BSFC over the 6 test points is 212 g/kWh.

The maximum cylinder pressure reaches 225 bar at the B100 operating point, which is below the maximum allowed peak cylinder pressure of 250 bar for the MX engine. Maximum cylinder pressure is however dependant on boost pressure. Given the reservations made in §4.5 concerning boost pressure, the predicted maximum cylinder pressure, may be unreliable and become higher in reality.

- *EGR-cooler temperature*

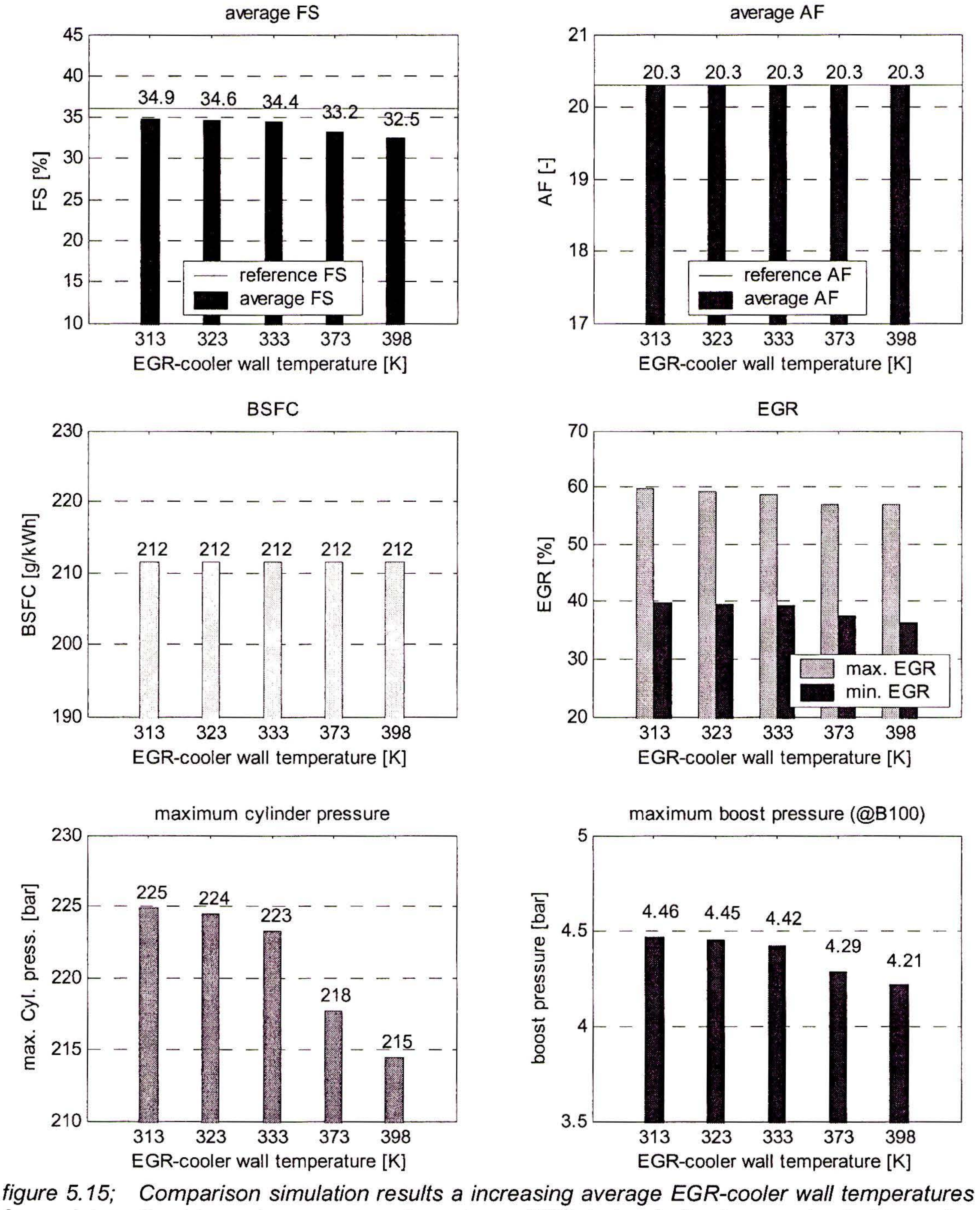



*figure 5.15; Comparison simulation results a increasing average EGR-cooler wall temperatures for a, internally driven- low-pressure- two stage- VTG turbo & fixed geometry turbo engine configuration. Simulation results; weight average over 6-mode test cycle. VTG-rack position @ 0.75 for all 6mode points except A50. VTG-rack; 0.5 @ A50. Average inlet manifold temperature constant 315K. Simulation nr. 19-23.*

In figure 5.15 the simulation results are given for increasing average EGR-cooler wall temperature in a low-pressure configuration. It can be seen that at increasing EGR-cooler wall temperature the FS level decreases. Subsequently the highest FS level is found at the lowest EGR-cooler wall temperature, which is an FS level of 34.9 % at an average EGR-cooler wall temperature of 313K. The highest FS level does not meet the target FS level of 36%.
At higher EGR-cooler temperatures the compressor entrance temperature increases. As can be expected based on equation eq. 5.2, the pressure ratio over the compressor decreases and thus boost pressure decreases. At constant AF level the lower boost pressure affects the EGR level, which causes the FS level to decrease. The maximum cylinder pressure also decreases in affect of the lower boost-pressure at increasing EGR-cooler wall temperature. The fuel consumption appears unaffected by the increasing EGR-cooler wall temperature. However as is mentioned in the figure label, the inlet manifold temperature is found constant at about 315K for all simulations and individual engine operating points. This constant manifold temperature of 315K and the constant AF ratio may explain the almost constant BSFC. Also at increasing EGR-cooler temperature the PMEP level remains almost constant (approximately -1.4 bar), indicating almost constant pumping loss.
The lower FS level at increasing EGR-cooler wall temperature equals a higher oxygen concentration, which may have a positive effect on efficiency and thus BSFC. However the lower boost pressure may offset this effect by decreasing the total cylinder charge and density. Furthermore the lower boost pressure causes the end of compression temperature to decrease, which may have a negative effect on BSFC. Please note however that the used Wiebe combustion model may lead to inaccurate calculation of cycle efficiency with high levels of EGR, as the combustion model may not accurately respond to changes in oxygen concentration and density. Also see §4.5. This may also explain the constant BSFC.

The constant manifold temperature is result of EGR-gas being cooled by both the EGR-cooler and the intercoolers and the high combined cooling capacity of the intercoolers and EGR-cooler. Unlike with the high-pressure configuration in the low-pressure configuration the EGR-gas is not fed directly into the intake manifold after it passes the EGR-cooler. Because of this the average EGR-cooler wall temperature has no direct effect on the intake manifold charge temperature with the low-pressure configuration. In the simulated low-pressure configuration the EGR-gas is extensively cooled by the large EGR-cooler after which it is mixed in with the fresh air stream. The mixture is then being compressed and cooled by both intercoolers (main & in-between the compressors). The intercoolers have the capacity to cool the charge (air + EGR) down to almost the average intercooler wall temperature (314K) under all conditions. It may be concluded that the intercoolers are oversized or the average intercooler wall temperature set to low. However, the total intercooler cooling surface area and average wall temperature are identical to that of the standard intercoolers as used in the Internally driven high-pressure EGR engine configuration simulations as earlier described under 'Details regarding the model' in this paragraph.

As the highest FS level is found at an average EGR-cooler wall temperature of 313K, the results of this simulation (19) is used in further comparison with the best result from simulations with the Internally driven high-pressure EGR engine configuration.

Explanation regarding internally driven low-pressure EGR simulation final results

The results of the internally driven low-pressure EGR configuration simulations are unsatisfactory, as they do not meet the target values. Particularly at full load conditions FS and EGR levels are relative low, see figure 5.14. The reason for the relative poor performance in terms of EGR and FS at full load conditions is due to the fact that in a Low-pressure EGR configuration the mass-flow through the compressor is increased when applying 'additional EGR' (§ 2.2) as is case here. The compressor (s) need to deliver a higher mass-flow at a higher pressure ratio opposed to an externally driven high-pressure EGR configuration or no EGR. Not only the fresh-air but also the EGR-gas needs to be compressed by the compressors. The required turbo-power increases because of this. Particularly at full load conditions mass-flows are relatively high and as such the required turbo-power is high. However the available turbo-power from the turbines is limited, and so the required mass-flow and boost-pressure for reaching the targeted EGR, FS and AF levels may not be achievable, as is the case here.

In addition to that, the compressor-entrance-temperature may increase in a Low-pressure EGR configuration. This has a negative effect on turbo-performance as earlier explained. In this case the large EGR-cooler and low EGR-cooler temperatures greatly reduce this negative effect by cooling the EGR-gas. However in order to reduce the possibility for condensation to occur, the EGR-cooler wall temperature is set not lower than 313K. At an average EGR-cooler wall temperature of 313K, the average compressor-entrance temperature over the 6-mode test cycle is 309K. The ambient air temperature is 293K.

A third factor of influence is the specific heat value. In a Low-pressure EGR configuration the compressors need to compress a mixture of fresh-air and exhaust-gas. As a result of that the specific heat value (Cp) of the mixture is slightly higher than that of pure air. A higher specific heat value has a negative effect on turbo-performance as can be seen in equation 5.2, however this influence is relatively small. The specific heat value typically rises to 1040 [kJ/kgK], as found in simulations. The specific heat value of air is 1005 [kJ/kgK] at 293K.

A last factor of influence is the increased pressure loss over the piping and intercooler(s) in-between the LP-compressor and inlet manifold. Because of the higher mass-flow over this section, (as a consequence of the internally driven low-pressure EGR configuration) the pressure loss will increase, opposed to internally driven high-pressure EGR or non-EGR configurations. In this case, the flow resistance of both intercoolers (main & in-between the compressor) has already been reduced. This greatly eliminates an increase in pressure loss induced by increased mass-flow over the intake system. Total engine pumping losses are however still significant.

# 6. Comparison configurations

From literature study it is concluded that from all the EGR-configurations the internally driven configurations are best suited for High EGR applications. For both external route internally driven high-pressure and the external route internally driven low-pressure EGR configuration simulations are performed.
In this chapter the simulation results of the externally driven internally driven high-pressure EGR configuration will be compared with those of the internally driven low-pressure EGR configuration. Besides the results also the differences in hardware (turbo, coolers etc.) between the two configurations as used in the simulations, will be discussed. Finally it will be concluded which configuration is most suited for high EGR applications.

## 6.1 Comparison simulation results

For both the internally driven high-pressure and low-pressure EGR-configurations only the best simulations results in terms of EGR, FS and AF are compared. The simulations of which the results are part of the comparison can be found in table 6.1. The simulation results only give an indication of the engine performance for the given engine configurations as many assumptions and reservations are made during the simulations. In comparing the results the given reservation are not taken into account as they apply to all simulation-results in the same way.

| **Setting/ parameter** / **Simulation** | *Cases* | *VTG control parameters* | *HP turbo scale multiplier* | *LP turbo scale multiplier* | *Tegr: EGR cooler temp. [K]* | *EGR-cooler size* | *Second inter-cooler ?* | *EGR & exhaust valve control* |
|---|---|---|---|---|---|---|---|---|
| Simulation 14 | 6 mode | AF (20.3) | 0.7 | 0.7 | 323 | Double Base-line | yes | - |
| Simulation 19 | 6 mode | Fixed rack pos; 0.75 0.5 @A50 | 1 | 1 | 313 | 5 times Base-line | yes | AF (20.3) |

*Table 6.1: Simulations, used for comparison between Internally driven-low-pressure and internally driven- high-pressure EGR engine configuration.*

In figure 6.1 the average simulation results of both the internally driven high-pressure and low-pressure EGR-configurations are given.

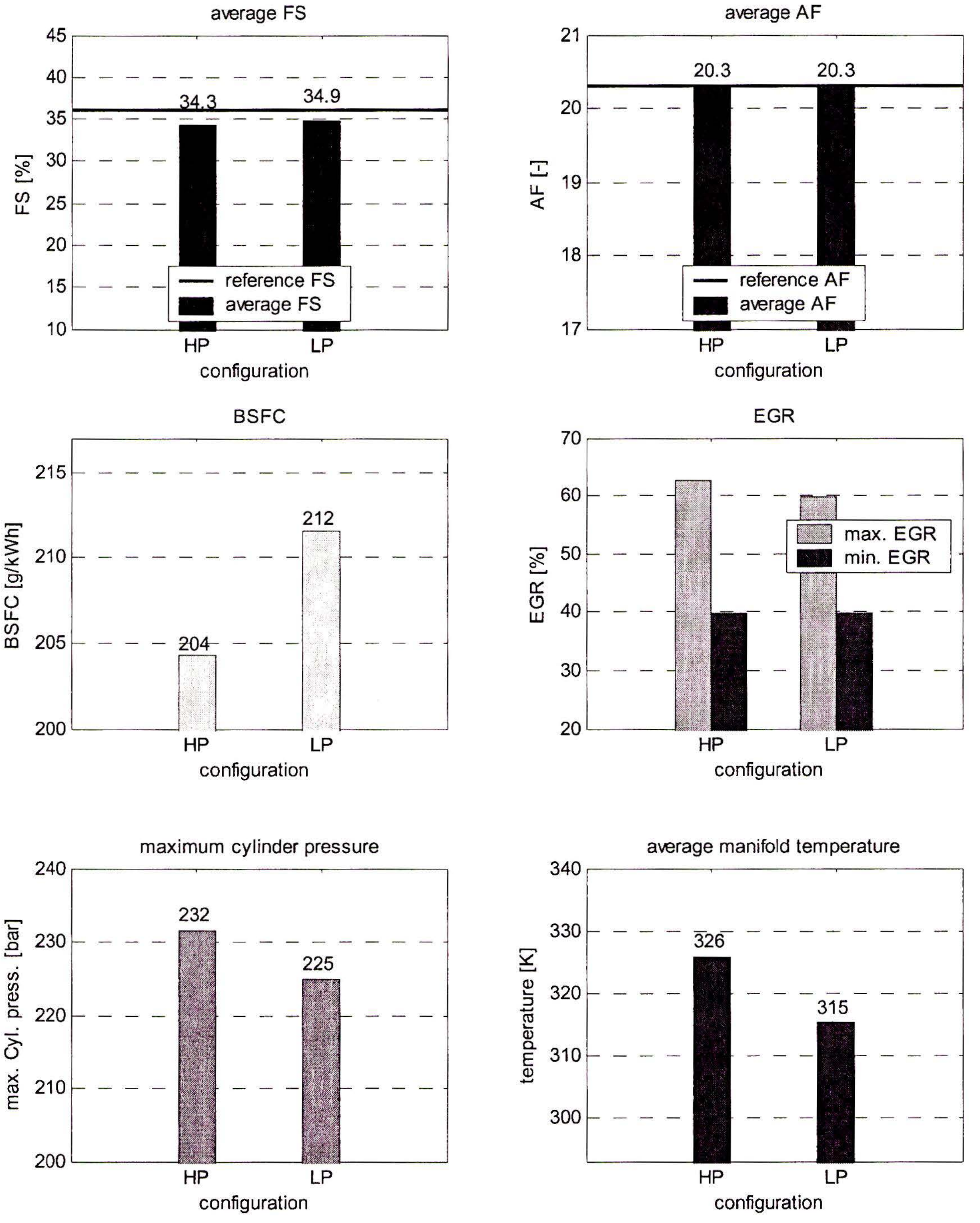



*figure 6.1 comparison simulation result internal-driven high-pressure (HP) and low-pressure (LP) EGR configuration. Simulation results; weight average over 6-mode test cycle. Simulation nr. 14-19.*

It can be seen from figure 6.1 that both configurations, high-pressure and low-pressure, give similar results in terms of average-FS and average-AF levels. In both cases the minimum AF level of 20.3 is maintained as supposed. In both cases the achievable average FS level is less than the desired 36%, at 34.3% and 34.9% high- and low –pressure respectively. Applying equation A2.1.2 a conservative prediction of NOx emission can be given. At FS 34.3% this amounts to 0.38 g/kWh NOx. At FS 34.9% this would give 0.34 g/kWh NOx. Again the given NOx-emission levels are only indicative as other factors, such as the manifold temperature, are of influence on engine-out NOx emissions besides the FS level. Also see §4.2.

Meaningful difference can be seen in BSFC for the high-and low-pressure EGR configuration. The low-pressure EGR configuration has a higher average BSFC than the high-pressure EGR configuration. The higher BSFC of the low-pressure EGR configuration opposed to the high-pressure configuration can be explained by higher average pumping losses found for the Low-pressure configuration, based on the calculated PMEP. The average PMEP over the 6-mode test cycle for the high-pressure configuration (simulation 14) amounts to -0.73 bar. The average PMEP over the 6-mode test cycle for the low-pressure configuration (simulation 19) amounts to -1.41 bar. A higher absolute PMEP indicates higher pumping losses. The higher pumping losses for the low-pressure configuration opposed to the high-pressure configuration are most likely a result of longer pipe-work required for this configuration. Also the higher mass flow rate through the engine, large EGR-cooling surface area and the use of exhaust backpressure and EGR valves may have resulted in higher pumping losses during the low-pressure EGR configuration simulations.

Only a limited level of model optimization is reached for both configurations. In the case of low-pressure EGR, only four VTG-rack positions are tested, more optimal VTG-rack positions may be available, which may result in lower BSFC for the low –pressure EGR configuration.

Overall the EGR levels are higher for the low-pressure EGR configuration compared to the high-pressure EGR configuration. This explains the slightly higher FS level for the low-pressure EGR configuration, compared to the high-pressure configuration. The maximum EGR level is higher for the high-pressure EGR configuration (C100).

As both AF and FS are about equal for both the high-and low-pressure EGR configuration, this means that the filling of the cylinders is about equal as well. This results in an average maximum cylinder pressure, which differs only 7 bars between the two configurations. However given the reservations made in §4.5 concerning boost pressure, the predicted maximum cylinder pressure, may be unreliable.

The found manifold temperatures differs 11K between the high-pressure and low-pressure EGR configurations. This is explained by the EGR-gas being cooled both by the EGR-cooler and by the intercoolers in the low-pressure configuration, unlike with the high-pressure configuration. With the high-pressure configuration the EGR-gas is only cooled by the EGR-cooler. As the EGR-gas is fed directly into the inlet manifold with the high-pressure configuration the manifold temperature is greatly affected by the average EGR cooler wall temperature. A higher average EGR-cooler wall temperature results in a higher inlet manifold temperature. With the low-pressure configuration however the EGR-gas is extensively cooled by both the big EGR-cooler and both intercoolers, as such the charge temperature in the inlet manifold is at a relative low level (315K) and remains constant under all simulated conditions. A low manifold temperature is beneficial for lowering NOx in addition to EGR. In the current set-up, the lower manifold temperature of the low-pressure EGR configuration could result in a lower NOx emission compared to the high-pressure EGR configuration despite the almost equal FS levels.

## 6.2 Comparison hard-ware

Several difference in hardware as modelled in the simulation models are compared for the high-pressure and low-pressure configuration. The hardware as modelled in the simulation models gives a good indication of the hardware requirements for achieving high levels of EGR with both the high-pressure and the low-pressure configuration.
Clearly 2-stage turbo-charging is required in order to reach the desired EGR, FS and AF levels. In the high-pressure configuration the turbos need to be smaller (30% smaller in terms of mass-flow) than for the low-pressure configuration because of the differences in mass-flow through the turbos (both turbine and compressor). As for simulation purposes the turbos are scaled with respect to mass-flow, physical turbo-chargers with equal performance are to be found. For the high-pressure configuration the VTG is helpful in controlling the EGR level. For the low-pressure configuration the VTG serves no purpose in controlling the EGR level.

The main difference in both configurations is in the cooling. An extra intercooler in-between the compressor serves overall performance in both configurations. In both cases the standard DAF EGR-cooler is increased in size and the average EGR cooler wall temperature set relatively low. For the high-pressure configuration the given performance is achieved with double the size of the standard EGR-cooler and an average EGR-cooler wall temperature of 323K.
For the low-pressure configuration an EGR-cooler five times the standard size with average wall temperature of 313K, gives the best results. Lower EGR-cooler temperatures (<313K) although likely to give better performance in terms of FS and EGR are considered unrealistic because of increased probability of condensation to occur. A smaller size EGR-cooler would result in a higher compressor entrance temperature and as such, worse performance in terms of EGR, FS and AF. It can therefore be concluded that for both the high-pressure and low-pressure configuration the average EGR-cooler temperature needs to be set low in the order 313- 323K. The EGR-cooler size needs to be bigger in case of the low-pressure configuration opposed to the high-pressure configuration with a factor 5:2.

Attention should be paid to the pressure loss over the intercoolers in case of a low-pressure EGR configuration because of the relative high mass-flows involved. For both the low-pressure and high-pressure configuration the intercooler surface area is kept as standard used on the MX engine. The second intercooler in between the 2stage compressors is identical to the main intercooler for both engine configurations. The surface area of both intercoolers is found to be sufficient for both configurations provided that the average intercooler wall temperature can be kept at 314K. However further optimization for the intercoolers size and temperature may be possible.

Next to the cooling aspects extra piping is required for a low-pressure configuration compared to the high-pressure configuration. In particular the EGR pipe needs to have a large diameter in order of 80mm, to minimize pressure loss. In reality the EGR-pipe may needs to be longer as the massive EGR-cooler may not fit inside the engine compartment.
In order to control the EGR flow the high-pressure configuration can rely on the VTG-turbo charger and addition EGR valves, which provide adequate control freedom. For the low-pressure turbo the control of EGR flow relies on the EGR-valves and an additional exhaust-backpressure valve.

Although fuel injection is left out of the equation during simulations by keeping timing constant, it may be desirable for both configurations to make use of a fuel injection system, which provides a large degree of freedom in injection timing. Such as common rail or smart injectors. Advancing injection timing may be required in order to compensate for increasing ignition delay and combustion duration. Also see §4.5.

## 6.3 Conclusion concerning simulations

The conducted simulations only provide an indication of real engine performance for the two tested engine configurations. Both EGR configurations (high- and low-pressure internally driven) give almost equal simulation results for EGR, FS and AF levels. However the low-pressure EGR configuration produces slightly higher FS and EGR levels. The high-pressure EGR configuration gives better BSFC opposed to the low-pressure configuration. However it can't be ruled out that further optimization of the low-pressure EGR configuration, for example more optimal VTG-rack positions (only four positions are tested), could provide somewhat better results in terms of BSFC. Also it can't be ruled out that in case the EGR-valves would be applied in the high-pressure EGR configuration together with a lower EGR-cooler temperature it may be possible to reach higher FS levels at minimum AF-ratio of 20.3. However a penalty in BSFC is to be expected, due to EGR-throttling at certain engine operating points. The lower average manifold temperature for the low-pressure configuration could be beneficial in lowering NOx emissions.

Meaningful difference can be found in the required hardware for both EGR configurations. Both simulations require two-stage turbo-chargers, however with 30% less mass-flow in case of the high-pressure EGR-configurations. Also both configurations demand increased cooler capacity as is natural when applying High levels of EGR. However in case of the low–pressure EGR configuration extreme cooling capacity is required (five times the current standard size) for cooling the EGR-gas in order to minimize loss of turbo-performance. Also more (longer) pipe-work will be required for the low-pressure EGR-configuration. It can be concluded that the low-pressure EGR configuration poses more packaging problems compared to the high-pressure EGR configuration due to larger required EGR-cooler surface area and longer piping. The required longer piping, larger EGR-cooler surface area and control by means of EGR- and backpressure valves are expected to cause higher pumping losses for the low-pressure configuration opposed to the high-pressure EGR configuration. This results in worse BSFC.

In table 6.2 an overview is given of the most important advantages and disadvantages of both the high-pressure and low-pressure engine configuration.

| | + | - |
|---|---|---|
| External route internally driven High-pressure | (more) acceptable packaging | FS below target |
| | Lower BSFC | |
| External route internally driven Low-pressure | Higher FS | FS below target |
| | Lower manifold temperature | Large packaging |
| | | Higher BSFC |

*Table 6.2: Overview of most important advantage and disadvantages of the 'external route internally driven high-pressure' and 'external route internally driven low-pressure' engine configuration, based on the conducted simulations for high-EGR.*

The large packaging of the low-pressure configuration is considered a major disadvantage of this configuration together with the higher BSFC. Because of the more acceptable packaging, the lower BSFC and the only marginally lower FS level of the 'external route internally driven high-pressure' opposed to the 'external route internally driven low-pressure' configuration, it can be concluded that the high-pressure configuration is better suited for High EGR application than the low-pressure configuration.

# 7. Conclusions high-EGR

Based on a literature study the effects of EGR on diesel combustion are described. Most important herein is the capability of EGR to lower local flame temperature and thereby lower engine-out NOx emission.

In order to apply high levels of EGR (30-70%) in a Heavy Duty CI diesel engine, engine modifications over the conventional HD engine construction and configurations are required, most important herein are the turbo configuration and cooling. The EGR gas can be routed in several ways around or through the engine. This results in several different engine configurations for the application of high-EGR. The different engine configurations are classified in two main groups, 'external route' and 'internal route'.

From literature it can be concluded that external route internally driven (no externally driven pumps) EGR systems are most favorable for applying 'high-EGR'. However 2-stage turbo-charging is considered essential if acceptable AF (air-fuel) ratios are to be maintained (order 23:1). Internally driven EGR systems can be subdivided in high-pressure (short route) and low-pressure (long route).

In order to reach the US 2010 NOx emission limit of 0.27 g/kWh NOx, it is concluded that an average EGR level of 50% is required. In order to maintain acceptable combustion efficiency (>99%) the average $\lambda$ level should not be lower than 1.4, which equals an AF-ratio of 20.3. An additional parameter FS (Fraction Stoichiometric), which is a function of EGR and AF should be at least 36% at 50% EGR and AF is 20.3.

The following results and conclusions are all based on external route internally driven EGR configurations, high-pressure and low-pressure, simulated in GT-power. Please note, the simulation results give only an indication of engine performance as several assumption and reservations are made during simulation. Furthermore only a limited level of optimization is searched during simulations.

- An external route internally driven high-pressure EGR configuration with a two-stage turbocharging system by means of a VTG turbo and a fixed geometry turbo is capable of giving the following performance over a 6-mode test cycle at a constant AF ratio of 20.3:
  - FS: 34.3%
  - BSFC: 204 g/k/Wh

  This configuration features;
  - Double the size EGR-coolers as standard on the MX-US engine with an average EGR-cooler wall temperature 323K.
  - Extra intercooler between the two compressors
  - Two-stage turbocharging system, with turbos down- scaled in mass-flow by 30 % (both turbines and compressors).

- The external route internally driven low-pressure EGR configuration two-stage turbocharging system by means of a VTG turbo and a fixed geometry turbo is capable of giving the following performance over a 6-mode test cycle at a constant AF ratio of 20.3:
  - FS: 34.9%
  - BSFC: 212 g/kWh

  This configuration features;
  - Factor five bigger EGR-cooler opposed to the standard size, with an average wall temperature of 313 K
  - Extra intercooler between the two compressors
  - Two-stage turbocharging system, with original (not scaled) turbos.
  - Low flow resistance EGR-line and intercoolers
  - Exhaust backpressure control valve

- An external route internally driven high-pressure EGR configuration with a single stage turbocharging system by means of a VTG turbo is not capable of reaching the desired EGR level (50%) even at lower than targeted lambda value (1.4), this because of turbo limitations. In conclusion to this a turbo is needed which is capable of delivering a higher pressure-ratio at lower mass-flow. Downscaling the turbo by its mass flow does improve its performance for high EGR, but not sufficiently.

For both the low-pressure and high-pressure EGR configuration the average maximum cylinder pressures are occasionally above 220 bar up to 232 bars, which is considered acceptable (<250 bar). However the reliability of these findings is not established.
The intake manifold charge temperature is typically lower for the low-pressure configuration compared to the high-pressure configuration, on average 315K and 326K, low-pressure and high-pressure configuration respectively. A low intake manifold charge temperature is beneficial for lowering engine-out NOx emissions.
For both configurations cooling of the EGR-gas is of importance for supplying EGR. A lower EGR-cooler wall temperature and increased EGR-cooler surface area increase EGR flow. Lower EGR-cooler wall temperature is beneficial to BSFC and lowers the intake manifold charge temperature in the high-pressure EGR configuration.

The lack of performance of the High-pressure EGR configuration in terms of FS is due to limitations in turbo-performance. The applied turbo-configuration is not capable of achieving high enough boost-pressure at relative low exhaust-flow over the turbines. Higher turbo-efficiency are needed.

The lack of performance of the low-pressure EGR configuration in terms of FS and BSFC can be explained to have several causes. These are;

- Increased mass-flow trough the compressors requires increased turbo-power, which is not always available, especially at high load.
- Hot EGR-gas mixed with fresh air may increase compressor entrance temperature, reducing turbo performance, increasing turbo power demand.
- Increased specific heat value (Cp) of the fresh-air & EGR mixture supplied to the compressors reduces turbo performance, increasing turbo power demand.
- Increased pumping loss due to the increased mass-flows over the intake system, longer pipe work, larger cooler surface area and the use of EGR- and exhaust backpressure valves for control. This is the main cause of the higher BSFC compared to the high- pressure configuration.

The external route internally driven low-pressure engine configuration is capable of achieving a slightly higher FS level. However, the large packaging of the external route internally driven low-pressure engine configuration is considered a major disadvantage of this configuration, together with a higher BSFC. Therefore the external route internally driven high-pressure engine configuration is considered better suited for achieving high levels of EGR.

Overall it can be concluded that based on literature study and conducted simulations, the internally driven high-pressure EGR configuration is most favorable for applying high levels of EGR (30-70%) on a Heavy-Duty Compression-Ignition engine.

# 8. Recommendations

In case of external route internally driven high-pressure EGR turbo-chargers, which fit the required turbo-demands need to be found. The requirements are 30% decrease in mass flow, thus smaller turbos.

The transient behavior of both the external route internally driven high-pressure and low-pressure configuration needs to be investigated. Transient behavior may suffer from some design-requirements for both high-EGR engine configurations.

The combustion model applied in all simulation-models makes use of constant 'Wiebe-law' settings for all engine-operating points. However the Wiebe-settings, particularly ignition delay and combustion duration may vary over the different operating points and EGR-levels. Although the influence of high levels of EGR on combustion is assessed in this thesis, it was not possible to integrate the results in the simulation models, due to a lack of measurement data. Instead several reservations are made. It is recommended to further investigate the effect(s) of high levels of EGR on combustion, particularly at high load conditions (high BMEP). The results of this investigation could than be used to improve the combustion-model, and as such provide more reliable results.

In the current simulation setup the engine is rated at 390kW, which is the maximum rating for the DAF-MX engine. In further research a lower engine rate could be used. With a lower rate the target levels for FS and AF could most likely be met more easily.

In case the high average maximum cylinder pressure poses a problem for reliable engine operation, several options for reducing the maximum pressure are available. These options are; alteration of the injection timing (later) or injection rate-shaping. Provided that a fuel injection system is available to allow for such large degree of freedom. The second option for reducing maximum cylinder pressure is the application of a Miller cycle. However a Miller cycle is expected to further increase the required turbo-power, which is already relatively high.

With respect to cooling several assumptions are made. It needs to be investigated to what extend the temperature levels used in simulation are practically feasible. A second cool-water-loop or a second EGR-air cooler may prove to be required for sufficient EGR-cooling. Given the required cooling capacities, this will require considerable cooling power for fans and water pumps to be driven. In further simulations this cooling power should be accounted for as it is of influence on fuel consumption. A more detailed EGR-cooler model may be required (as earlier indicated, this was not possible for this thesis).

In these simulations the engine-out emissions are not taken into account, however it is concluded from literature that 'high EGR' although beneficial to lowering NOx emission is likely to increase PM emissions. As such the effects of 'High EGR' under the simulated conditions on PM should be investigated, also the possibility of DPF's to reduce PM emissions for 'high EGR' should be assessed.

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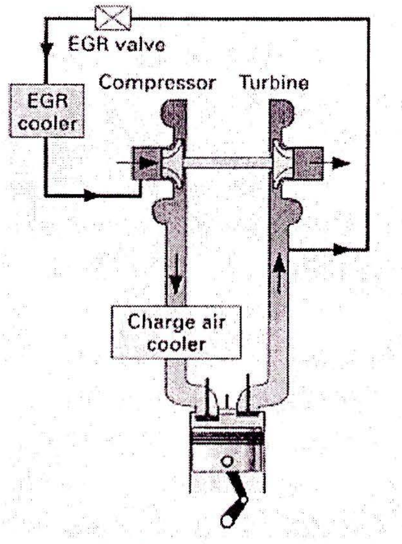
## Appendix content

# Appendix 1 Literature

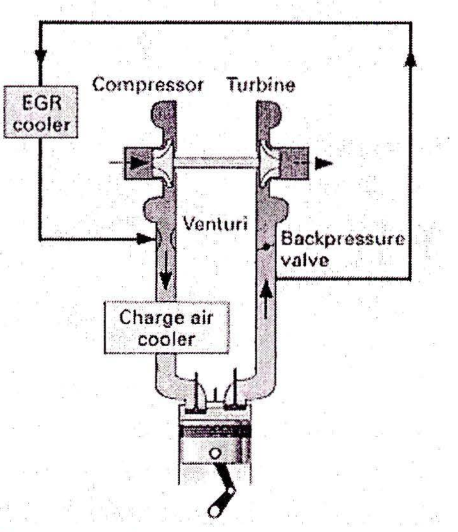
## Appendix 1.1 Miscellaneous EGR systems




Turbine in → compressor in
'Hybrid HPL/ LPL' (dieselnet)

(Paramins Post, Autumn 1997, *EGR- the key to diesel's future?*)
(www.dieselnet.com, 2005.01, *Exhaust Gas Recirculation*)

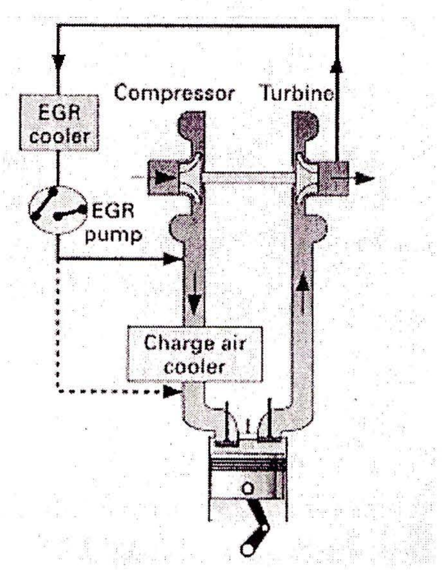
Associated problems:
- Low efficiency
- Turbocharger unbalance (unequal flows turbine/ compressor)
- Fouling




Turbine in → Intercooler in

(Paramins Post, Autumn 1997, *EGR- the key to diesel's future?*)

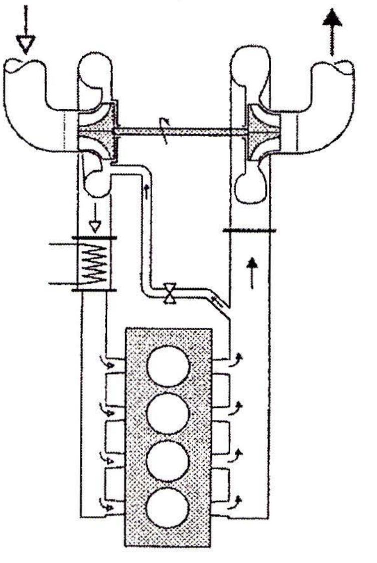
Associated problems:
- Fouling of the charge air cooler




Turbine out → pump → compressor out

(Paramins Post, Autumn 1997, *EGR- the key to diesel's future?*)

Associated problems:
- Difficult to realize with high-EGR because of the high flow rates through the 'EGR-pump' and high pressure ratio over the 'EGR-pump'
- Low efficiency



Turbine in → compressor diffusor
(SAE 2000-01-0226; MTC AB)

'EGR turbo' (SAE 980775; Ishikawajima Harima Heavy Industries Co., Ltd)

Associated problems:
- Fouling of the intercooler
- Insufficient cooling of the EGR-gas
- Insufficient EGR flow

# Appendix 2 Extra research and calculations on behalf of model construction and set-up

## Appendix 1.2 Calculation EGR target value

Conditions

- NOx target: 0.27 g/kWh from US federal (EPA) 2010 emission standards for heavy-duty diesel truck and bus engines.
- Minimum AF-ratio = 20.3, ($\lambda$ =1.4)

Data

Engine-measurement-data, from a DAF MX two-stage turbocharged engine rated at 390kW is used and from a DAF XEC engine rated at 355 kW, TNO (01.OR.VM.001.1/MVE).

Calculation of FS based NOx relationships

The expression for FS, where FS is a function of EGR and AF or $\lambda$

$$FS = \frac{(AF_{stoich}+1) \cdot EGR/100}{AF_{real}+1} \qquad A\ 2.1.\ 1$$

For each measured work-point of the engine the FS can be calculated and plotted as a function of the measured NOx emission in g/kWh. A function can be fitted which gives the relationship between FS and NOx. The best fit is achieved by a logarithmic function.

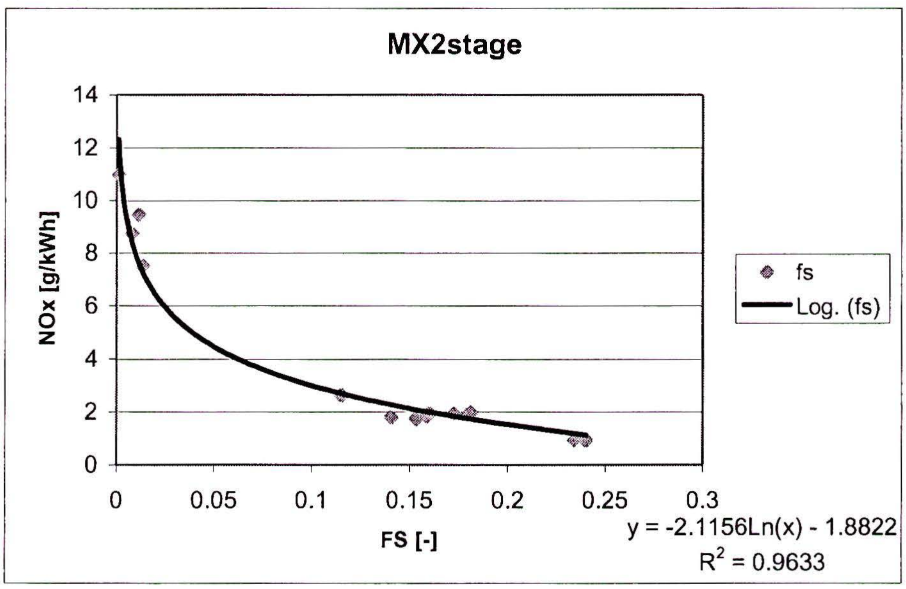
In this case, above calculations are performed for measurement data of the DAF MX (390kW) and for the measurement data of the DAF MX (390kW) supplemented with data from the DAF XEC (355kW), resulting in two FS←→ NOx relationships:

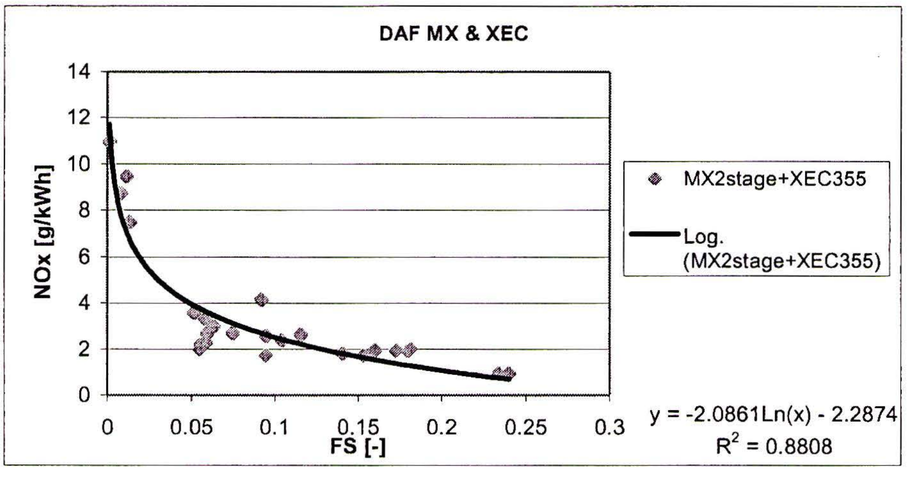
*DAF MX;* $NOx = -2.1156\ln(FS) - 1.8822$ *A 2.1. 2*

*DAF MX & XEC;* $NOx = -2.0861\ln(FS) - 2.2874$ *A 2.1. 3*

With NOx [g/kWh] and FS [-].

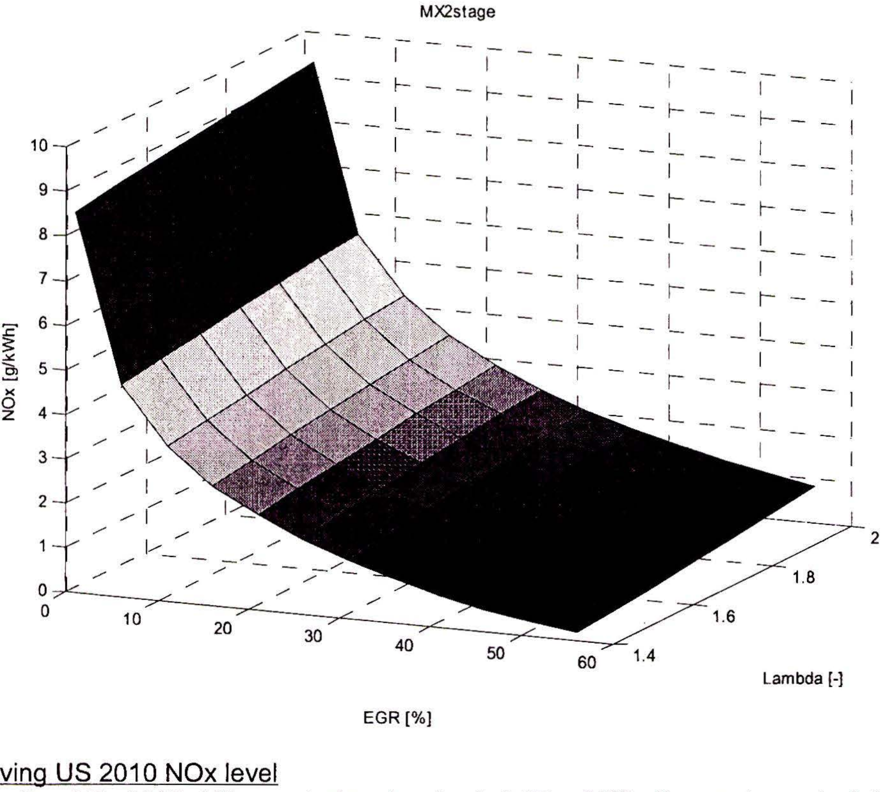
*E 2.1.1; NOx vs. FS*






In both the functions A2.1.2 and A2.1.3 the Variable FS can be replaced by expression A2.1.1, as such giving the relationships between NOx, EGR and $\lambda$. A 3D surfaces plot expresses this relationship, see under E2.1.2;

*E2.1.2; NOx vs. EGR & $\lambda$*




<u>Achieving US 2010 NOx level</u>

Given the US 2010 NOx emission level of 0.27 g/kWh it can be calculated for the two relationships A2.1.2 & A2.1.3 at which FS level this low NOx level is reached:

| | | |
|---|---|---|
| (A 2.1.2) | MX | FS = 0.36 |
| (A 2.1.3) | MX + XEC | FS = 0.29 |

A range of combinations of EGR and $\lambda$ can be found from the NOx-EGR-$\lambda$ relationships which should give a NOx level below 0.27 g/kWh;

*E2.1.3; EGR & λ value for NOx levels below 0.27 g/kWh*

**MX 2stage** — As derived from Matlab file "FS_NO.m"

| EGR [%] | Lambda [-] | FS [-] | NO_low [g/kWh] |
|---|---|---|---|
| 11 | 0,3 | 0,3667 | 0,2404 |
| 16 | 0,4 | 0,4 | 0,0563 |
| 26 | 0,7 | 0,3714 | 0,2131 |
| 31 | 0,8 | 0,3875 | 0,1235 |
| 36 | 0,9 | 0,4 | 0,0563 |
| 36 | 1 | 0,36 | 0,2792 |
| 41 | 1 | 0,41 | 0,0041 |
| 41 | 1,1 | 0,3727 | 0,2057 |
| 46 | 1,2 | 0,3833 | 0,1463 |
| 51 | 1,3 | 0,3923 | 0,0974 |
| 51 | 1,4 | 0,3643 | 0,2542 |
| 56 | 1,4 | 0,4 | 0,0563 |
| 56 | 1,5 | 0,3733 | 0,2023 |
| 61 | 1,5 | 0,4067 | 0,0213 |
| 61 | 1,6 | 0,3812 | 0,1579 |
| 61 | 1,7 | 0,3588 | 0,2861 |
| 66 | 1,7 | 0,3882 | 0,1195 |
| 66 | 1,8 | 0,3667 | 0,2404 |

**MX & XEC**

| EGR [%] | Lambda [-] | FS [-] | NO_low [g/kWh] |
|---|---|---|---|
| 6 | 0,2 | 0,3 | 0,2242 |
| 16 | 0,5 | 0,32 | 0,0896 |
| 21 | 0,7 | 0,3 | 0,2242 |
| 26 | 0,8 | 0,325 | 0,0572 |
| 31 | 1 | 0,31 | 0,1558 |
| 36 | 1,1 | 0,3273 | 0,0427 |
| 36 | 1,2 | 0,3 | 0,2242 |
| 41 | 1,3 | 0,3154 | 0,1199 |
| 41 | 1,4 | 0,2929 | 0,2745 |
| 46 | 1,4 | 0,3286 | 0,0344 |
| 46 | 1,5 | 0,3067 | 0,1784 |
| 51 | 1,6 | 0,3187 | 0,0977 |
| 51 | 1,7 | 0,3 | 0,2242 |
| 56 | 1,7 | 0,3294 | 0,0291 |
| 56 | 1,8 | 0,3111 | 0,1483 |
| 56 | 1,9 | 0,2947 | 0,2611 |
| 61 | 1,9 | 0,3211 | 0,0827 |
| 61 | 2 | 0,305 | 0,1897 |
| 66 | 2 | 0,33 | 0,0254 |

Given the minimal λ value of 1.4 (TNO; 04.OR.VM.040.2/MvA) the EGR level should be 50% and 40% respectively for relationship A2.1.2 and A2.1.3 in order to reach a NOx level of 0.27 g/kWh.

Conclusions on the use of FS based NOx prediction

Two relationships are found representing the relationship FS (Fraction Stoichiometric) and NOx [g/kWh] and subsequently two relationships between NOx, EGR and $\lambda$.

In order to reach the US 2010 NOx level of 0.27 g/kWh, FS should be, depending on which relationship is used: FS = 0.36 or FS = 0.29.

When applying a minimal $\lambda$ value of 1.4, EGR levels should be between 40 and 50%, other combination of EGR and $\lambda$ are possible in order to reach the desired US2010 NOx level.

Remarks

The DAF MX engine is the engine to be used in further research on the subject of 'High EGR'. As such the measurement data from the DAF XEC engine may not be valid to be used in this context, leaving only one relationship between NOx and FS (A2.1.2).

Further validation of the relationships may be required. Amongst the factors that are of influence on the validity of the relationships are the type of engine and the injection parameters. However as an indication of the required amount of EGR and $\lambda$, the current relationships are already satisfactory.

*Derivation FS*

The FS is defined as; $FS = \frac{m_{inert}}{m_{tot}}$ A 2.1.4

Assuming $m = \dot{m}$

The FS value of the inlet charge ($FS_{inlet}$) is:

$$\Rightarrow FS_{inlet} = \frac{m_{egr}.FS_{exhaust} + m_{air}.FS_{air} + m_{egr\_scaveging}.FS_{scaveging}}{m_{tot}};$$ A 2.1.5

Under the assumption that the fresh air contains no inert gas and no EGR is created due to poor scavenging: $FS_{air} \cong 0; m_{egr\_scaveging} \cong 0$

The FS value of the exhaust flow ($FS_{exhaust}$) is:

$$\Rightarrow FS_{\text{exhaust}} = \frac{m_{inert}}{m_{tot}} = \frac{m_{stoichiometric}}{m_{tot}} \quad \text{(post combustion)}$$ A 2.1.6

This can be written as

$$\Leftrightarrow FS_{\text{exhaust}} \cong \frac{(m_{fuel} + m_{air})_{stoich}}{m_{tot}} = \frac{m_{fuel} + m_{fuel}.AF_{stoichiometric}}{m_{fuel} + m_{fuel}.AF_{real}} = \frac{m_{fuel}.(AF_{stoichiometric} + 1)}{m_{fuel}.(AF_{real} + 1)}$$ A 2.1.7

(pre combustion)

The $FS_{inlet}$ now becomes:

$$\Leftrightarrow FS_{inlet} = \frac{m_{egr}.FS_{\text{exhaust}}}{m_{tot}} = \frac{\frac{\%EGR}{100\%}.m_{tot}.FS_{\text{exhaust}}}{m_{tot}} = \frac{\%EGR}{100\%} FS_{\text{exhaust}}$$ A 2.1.8

$$\Leftrightarrow FS_{inlet} = \frac{\%EGR}{100\%} \frac{m_{fuel}.(AF_{stoichiometric} + 1)}{m_{fuel}.(AF_{real} + 1)} \boxed{\Leftrightarrow FS_{inlet} = \frac{(AF_{stoichiometric} + 1) \cdot \frac{\%EGR}{100\%}}{AF_{real} + 1}}$$ A 2.1.9

## Appendix 2.2 Preparatory EGR and turbo study

*Introduction*

In order to determine turbo requirements for usages in combination with high EGR, several calculation are performed. As a base-line engine a DAF MX engine is used rated at a maximum of 390 kW, the same as used in recent engine measurements with a DAF MX engine at TNO. Initial values for EGR and λ are based on earlier work at TNO (04.OR.VM.040.2/MvA) and calculations performed with measurement data from DAF engines (appendix 1.1). Also earlier literature research gives an indication of the desired EGR levels for future emission levels (US 2010). Calculations are performed for a 'High pressure' EGR configuration.

Calculation's are performed for 12 points of the 13mode ESC steady state engine test cycle (all but idle). In table T2.2.1 the 12 work points are given for the DAF MX engine.

*Engine*

Some general information about the engine is given in table T2.2.1:

| Component | Description | Range |
|---|---|---|
| General Engine Performance | ▪ Power rated in kW at n=1405 - 2100rpm<br>▪ Power rated in hp at n=1400 - 2100 rpm<br>▪ Torque max in N.m. | 265 - 380<br>350 - 510<br>1830 - 2510 |
| General Engine | ▪ 4 Cycle-diesel; 6 cyl in-Line; turbocharged and intercooled<br>▪ Bore<br>▪ Stroke<br>▪ Displacement<br>▪ N idle<br>▪ Moment of Inertia<br>▪ Max. No load Governed speed<br>▪ Maximum Overspeed Capability<br>▪ Max cycle pressure | <br>: 130 (mm)<br>: 162 (mm)<br>: 12.9 ($dm^3$)<br>: 600 +/- 100 rpm<br>: (t.b.e) kg-$ms^2$<br>: 2200 +/- 25rpm<br>: (t.b.e) rpm<br>: 20 MPa |

*Table T2.2.1 General engine specifications*

*Conditions*

The 12 calculation points are given in table T2.2.2:

| **DAF MX 390 kW** | | | | | |
|---|---|---|---|---|---|
| *speed* | | *load* | | | |
| | [rpm] | 100% [Nm] | 75% [Nm] | 50% [Nm] | 25% [Nm] |
| *A* | 1210 | 2557 | 1920 | 1279 | 640 |
| *B* | 1525 | 2442 | 1832 | 1221 | 610 |
| *C* | 1830 | 2015 | 1511 | 1008 | 504 |

*Table T2.2.2 Work-points*

Initially 70% EGR with a λ value of 1.4 is maintained for all work points. This initial high EGR level is chosen in order to assess a wide range of EGR levels.
According to TNO (04.OR.VM.040.2/MvA ) EGR levels up to 70% with a λ value as low as 1.4 gives satisfying result in terms of low NOx and combustion efficiency however at very low BMEP's. From FS (Fraction Stoichiometric) calculations with DAF measurement-data it comes forward that in order to reach the US 2010 NOx level with an acceptable λ value (1.4) in terms of combustion efficiency, EGR levels should be between 40 and 70 %, also see Appendix 2.1 on FS based NOx prediction. Further assumptions and estimates are given in 'Variable estimates and assumptions'

*Calculation*

All calculation are performed in Matlab, by constructing a model. A list of symbols can be found near the end of this study

> *BMEP and power*

Given the engine load and the engines displacement the required BMEP can be calculated;

$$BMEP = \frac{M \cdot 2\pi}{Vs \cdot \frac{1}{2} \cdot 100} \qquad \text{A 2.2.1}$$

The engine load and speed produce its power-output.

> *Fuel mass flow*

$$\dot{m}_f = \frac{BMEP \cdot 10^5 \cdot VN}{H \cdot 10^6 \cdot \eta_{eng}/100} \qquad \text{A2.2.2}$$

*VN* represents the theoretical volume flow at N engine speed, calculated as:

$$VN = Vs \cdot 10^{-3} \cdot N/120 \qquad \text{A2.2.3}$$

Engine efficiency; $\eta_{eng} = \eta_T \cdot \eta_{comb}$ A2.2.4

> *Charge mass flow*

Given the 'Fuel mass flow', the AF ratio (dry) and humidity the total 'Fresh air mass flow' can be calculated;

$$\dot{m}_{air} = \dot{m}_f \cdot AF_{real} \cdot \left(1 + Ha \cdot 10^{-3}\right) \qquad \text{A2.2.5}$$

Herein; $AF_{real} = AF_{stoich} \cdot \lambda$ A2.2.6

Given the 'EGR percentage' and the 'Fresh air mass flow', assuming additional EGR-control the 'Total charge mass flow' can be calculated;

$$\dot{m}_{charge} = \dot{m}_{air} \cdot \left(100/(100 - EGR)\right) \qquad \text{A2.2.7}$$

> *Manifold pressure*

Given the 'Total charge mass flow', the theoretical volume change (*VN*) and the volumetric efficiency of the engine; the required density of the charge can be calculated:

$$\rho = \frac{\dot{m}_{charge}}{VN \cdot \eta_{vol}/100} \qquad \text{A2.2.8}$$

The temperature of the total charge can be calculated based on the estimates for the air-temperature entering the manifold (*Tair*) and the temperature of the EGR-gas (*Tegr*) entering the manifold.

$$T_{charge} = \frac{Q_{charge}}{Cp_{charge}} \qquad \text{A2.2.9}$$

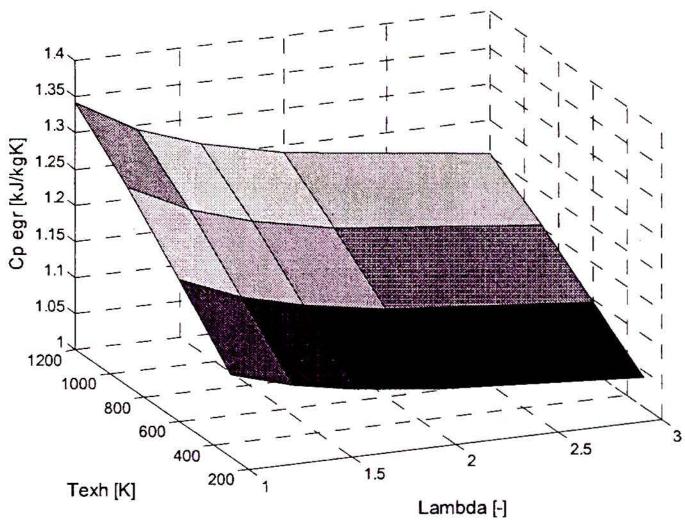
With: $Q_{charge} = Q_{air} + Q_{egr} = \left(Y_{air} \cdot Cp_{air} \cdot T_{air}\right) + \left(Y_{egr} \cdot Cp_{egr} \cdot T_{egr}\right)$ A2.2.10

And: $Y_{air} = 100 - EGR/100$ A2.2.11

$Y_{egr} = EGR/100$ A2.2.12

With: $Cp_{charge} = Y_{egr} \cdot Cp_{egr} + Y_{air} + Cp_{air}$ A2.2.13

The specific heat value of EGR ($Cp_{egr}$) is determined by interpolation based on Lambda ($\lambda$) and exhaust-gas-temperature (Tegr) from a look-up table derived from DynaMo.

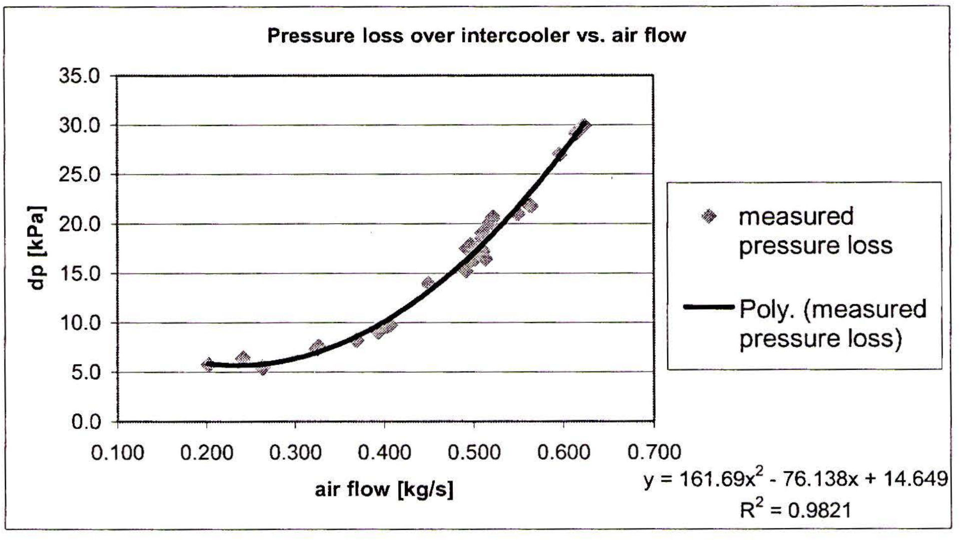



Knowing the density of the charge and the temperature; the manifold pressure can be calculated:

$P_{man} = \rho \cdot T_{charge} \cdot R$ A2.2.14

(With: *R* the individual gas constant)

The 'turbo-boost pressure' (*$P_{boost}$*) can be calculated by adding the pressure losses over the intercooler and piping (*dP*). This pressure loss can be derived from a function based on engine measurement data. In this function the pressure-loss is a function of the 'fresh air mass flow':




Measurement points are derived from test results from a DAF MX 2 stage engine. A polynomial fit is made through the measurement point with a correlation coefficient ($R^2$) of 0.98.

$$dp = \left(1.59 \cdot \dot{m}_{air}^2 - 0.76 \cdot \dot{m}_{air} + 0.15\right) \cdot 10^5 \qquad \text{A2.2.15}$$

$$P_{boost} = P_{man} + dp \qquad \text{A2.2.16}$$

> Compressor & turbine

The compressor pressure ratio can be calculated:

$$\tau_{compressor} = \frac{P_{boost}}{P_{ambient}} \qquad \text{A2.2.17}$$

The required compressor power:

$$W_{compressor} = \dot{m}_{air} \cdot Cp_{air} \cdot 1000 \cdot T_{amb} \cdot \left(\tau_{compressor}{}^{\kappa-1/\kappa} - 1\right) \cdot 1/\eta_{compressor} \qquad \text{A2.2.18}$$

Assuming a 'High pressure' EGR system, the mass flow through the turbine can be regarded equal to the 'Fresh air mass flow' plus 'Fuel mass flow'.

$$\dot{m}_{turbine} = \dot{m}_{air} + \dot{m}_f \qquad \text{A2.2.19}$$

Furthermore assuming the turbine power to be equal to the compressor power the pressure ratio over the turbine can be calculated.

$$\tau_{turbine} = \left( - \frac{W_{turbine}}{\dot{m}_{turbine} \cdot Cp_{egr} \cdot 1000 \cdot T_{exh} \cdot \eta_{turbine}} \right)^{-\left(\kappa_{egr}/\kappa_{egr}-1\right)} \qquad \text{A2.2.20}$$

With: $W_{turbine} \approx W_{compressor}$

The value '$\kappa_{egr}$' in (A2.2.20) can be determined by interpolation based on Lambda ($\lambda$) and exhaust-gas-temperature (Tegr) from a look-up table derived from DynaMo ( in the same way as '$Cp_{egr}$') .

*Results:*
In figure E2.2.1 and E2.2.2 the desired compressor ratio vs. mass-flow are given for both 50% and 70% EGR for all test conditions presented in table T2.2.2. A fixed lambda value of 1.4 is maintained. Also the desired pressure ratio over the turbine is given as a function of mass-flow again for 50% and 70% EGR, all for a 'High pressure' EGR configuration.

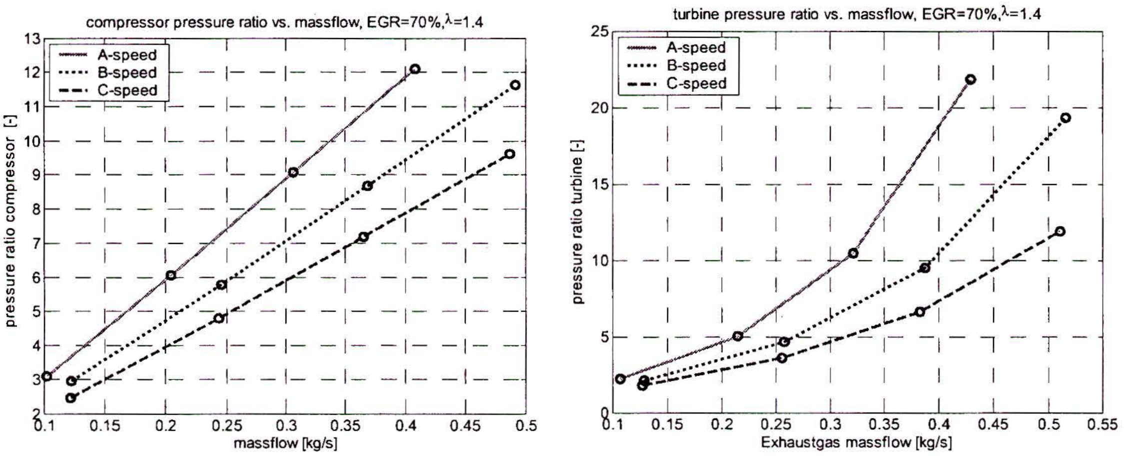



*Figure E2.2.1; Required compressor and turbine map, at three engine speeds for 70% EGR and λ 1.4.*

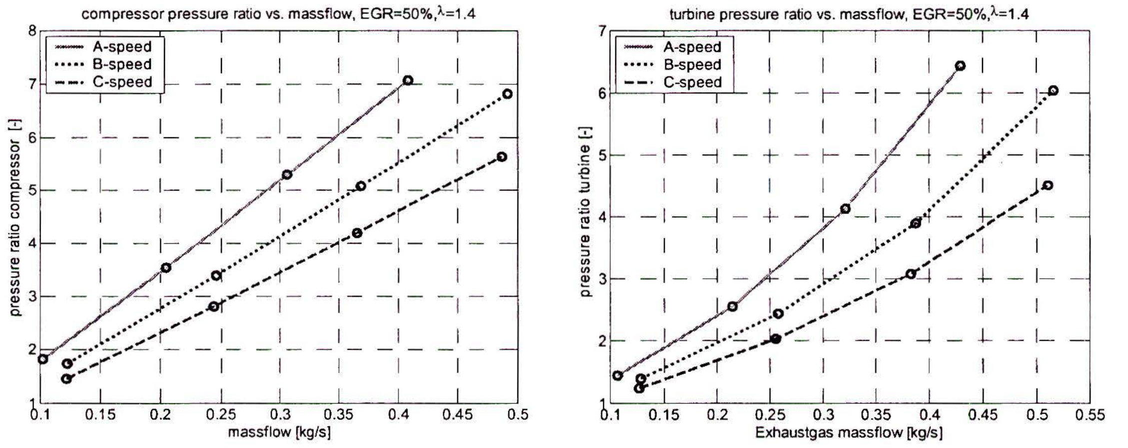



*Figure E2.2.2; Required compressor and turbine map, at three engine speeds for 50% EGR and λ 1.4.*

As can be seen in Figure E2.2.1, very high compression ratios are required to reach 70% with minimum lambda of 1.4 at high engine loads. Although this calculation does not represent a turbo-match it can be seen that, based on realistic compressor and turbine efficiencies, the turbine pressure ratio is required to be unrealistically high at 70% EGR and lambda 1.4. If turbo efficiency is increased the turbine pressure ratios are brought towards more realistic levels, as can be seen under 'Plots' (D3) where turbine efficiency was assumed 80%, instead of 70%.

In Figure E2.2.2, also relative high compression ratios can be seen, but much lower compared with 70% EGR (figure E2.2.1). In this case the required turbine pressure ratios are also more realistic and slightly less than the compression ratio of the compressor.

Under 'Plots' (in this appendix); turbo-power and BMEP values, as calculated, are given, it can be seen that the required turbo power is much lower in case of 50% EGR ($\lambda$ 1.4) opposed to 70 % EGR ($\lambda$ 1.4).

*Conclusions:*
In order to reach future NOx emission targets for HD diesel engine, for example US 2010 by means of EGR, high levels of EGR are required. Based on earlier research and measurements, these high EGR levels are in the range of 40 to 70 % with relative low lambda levels of minimal 1.4.
High levels of EGR require high compression ratios and high-pressure ratios over the turbine even at minimal lambda value. With 70% EGR and estimated compressor and turbine efficiencies (based on available turbo-maps) the pressure ratios over the turbine are unrealistically high. In this case higher turbo-efficiency is required.
With 50% EGR the pressure ratios of both the compressor and turbine are high, but seem feasible, considering two-stage turbo-charging.

*Recommendations:*
A EGR level of 70% seems very difficult to achieve at high engine loads even at minimal lambda value, therefore it should be considered to set EGR level for high engine loads at maximum 50% or lower. At lower engine loads it can be considered to increase EGR level or lambda.
A turbocharger that fits the general requirements in terms of pressure ratios and mass-flow should be looked for and be matched for the DAF MX engine.
As most single stage turbocharger (either fixed or variable geometry) are not capable of reaching the desired high-pressure ratios (seven and higher) a two-stage turbo-charger system could be considered as the only option.
Remark: in above described calculations numerous assumptions are made, a critical assumption is the temperature of the EGR-gas witch is set at 100°C. Particularly at high engine speed and load it will be difficult to maintain such a low EGR temperature and cooling power will be relative high. The feasibility and desirability of such a low EGR-gas temperature may be reconsidered.

Symbols

| | | |
|---|---|---|
| BMEP | Brake Mean effective pressure | [bar] |
| M | Torque | [NM] |
| N | Engine speed | [rpm] |
| Vs | Total engine displacement | [L] [$dm^3$] |
| $\dot{m}$ | Mass flow | [kg/s] |
| H | Lower combustion value | [MJ/kg] |
| VN | Theoretic volume flow per second | [$m^3$/s] |
| Ha | Absolute humidity of ambient air | [g/kg] |
| $\eta$ | Efficiency | [%] |
| $\eta_{eng}$ | Engine efficiency | [%] |
| $\eta_T$ | Thermal efficiency | [%] |
| $\eta_{vol}$ | Volumetric efficiency | [%] |
| $\eta_{comb}$ | Combustion efficiency | [%] |
| $AF_{real}$ | Actual Air Fuel ratio | [kg/kg] [-] |
| $AF_{stoich}$ | Stoichiometric Air Fuel ratio | [kg/kg] [-] |
| $\lambda$ | Lambda | [-] |
| EGR | EGR mass percentage | [%] |
| $\rho$ | Density | [$kg/m^3$] |
| T | Temperature | [K] |
| Q | Specific heat per mass | [kJ/kg] |
| Cp | Specific heat value | [kJ/kgK] |
| Y | Mass fraction | [-] |
| P | Pressure | [Pa] |
| W | Work | [W] |
| R | Individual gas constant | [J/kgK] |
| $\kappa$ | Ration of specific heat | [-] |
| $\tau$ | Pressure ratio | [-] |
| dp | Pressure loss over intercooler and piping | [Pa] |

Subscripts

| | | |
|---|---|---|
| *f* | Fuel | |
| *air* | Air | |
| *charge* | Manifold charge | |
| *egr* | EGR gas | |
| *exh* | exhaust gas | |
| *turbine* | turbine | |
| *compressor* | compressor | |
| *man* | manifold | |
| *ambient* | ambient | |
| *boost* | turbo | boost |

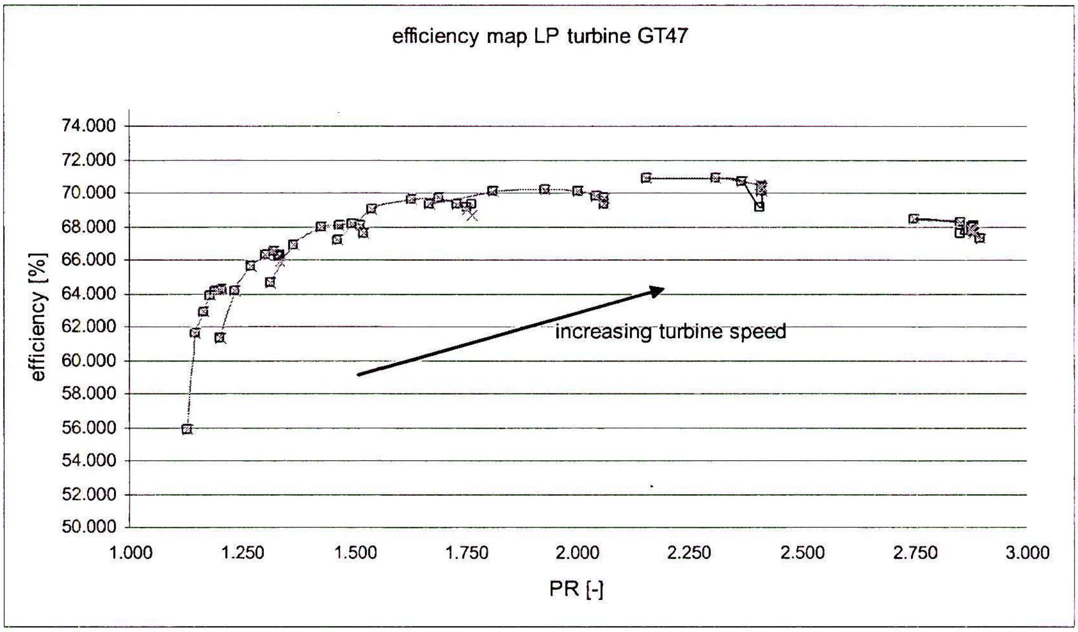
Variable estimates and assumptions

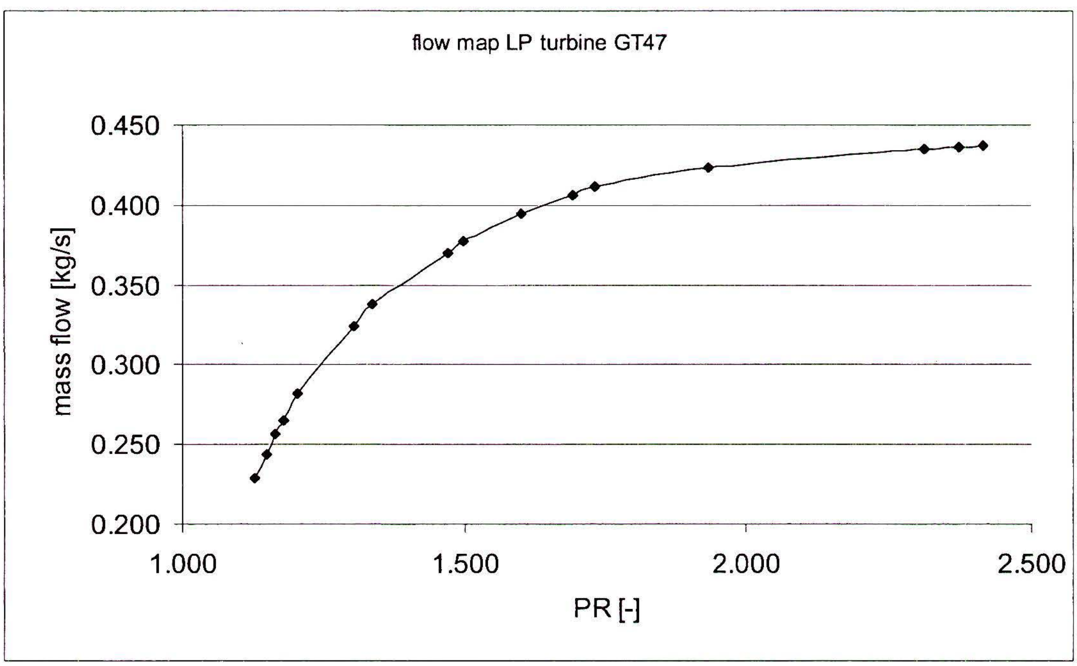
| ***Variable*** | ***Value*** | ***Unit*** | ***Description*** | ***Reference*** |
|---|---|---|---|---|
| EGR | 40~70 | % | EGR mass | TNO & DAF |
| $\lambda$ | 1.4 | - | Lambda | TNO |
| H | 43.1 | MJ/kg | Lower combustion value | TNO measurement 15-22 March 2005; DAF MX 2stage |
| AFstoich | 14.57 | Kg/kg | Stoichiometric Air Fuel ratio | TNO measurement 15-22 March 2005; DAF MX 2stage |
| $\eta_{com}$ | 93.4 | % | Combustion efficiency | TNO (worst case, $\lambda$=1.4) |
| $\eta_{vol}$ | 89.3 | % | Volumetric efficiency | TNO measurement 15-22 March 2005; DAF MX 2stage |
| $\eta_T$ | 40.4 | % | Thermal efficiency | TNO measurement 15-22 March 2005; DAF MX 2stage |
| Ha | 5 | g/kg | Absolute Humidity | TNO measurement 15-22 March 2005; DAF MX 2stage |
| Tair | 60<br>333 | °C<br>K | Air temperature downstream intercooler | advise by Ruud Verbeek (TNO) |
| Tegr | 100<br>(→200)<br>373 | °C<br><br>K | EGR temperature downstream EGR cooler | advise by Ruud Verbeek (TNO) |
| Cp_air | 1.05 | kJ/kgK | Specific heat value Air | www.engineeringtoolbox.com |
| Cv_air | 0.72 | kJ/kgK | Specific heat value Air | www.engineeringtoolbox.com |
| R | 286.9 | J/kgK | Individual gas constant | www.engineeringtoolbox.com |
| T_amb | 20<br>293 | °C<br>K | Ambient air temperature | TNO measurement 15-22 March 2005; DAF MX 2stage |
| P_amb | 102.3<br>$102.3*10^{-3}$ | kPa<br>Pa | Ambient air pressure | TNO measurement 15-22 March 2005; DAF MX 2stage |
| T_exh | 580<br>853 | °C<br>K | *Estimated* exhaust gas temperature | TNO measurement 15-22 March 2005; DAF MX 2stage |
| $\eta_{compressor}$ | 0.75 | % | Compressor efficiency | DAF turbo maps |
| $\eta_{turbine}$ | 0.70 | % | Turbine efficiency | DAF turbo maps |
| Vs | 12.9 | $dm^3$ | Total engine displacement (swept volume) | DAF |

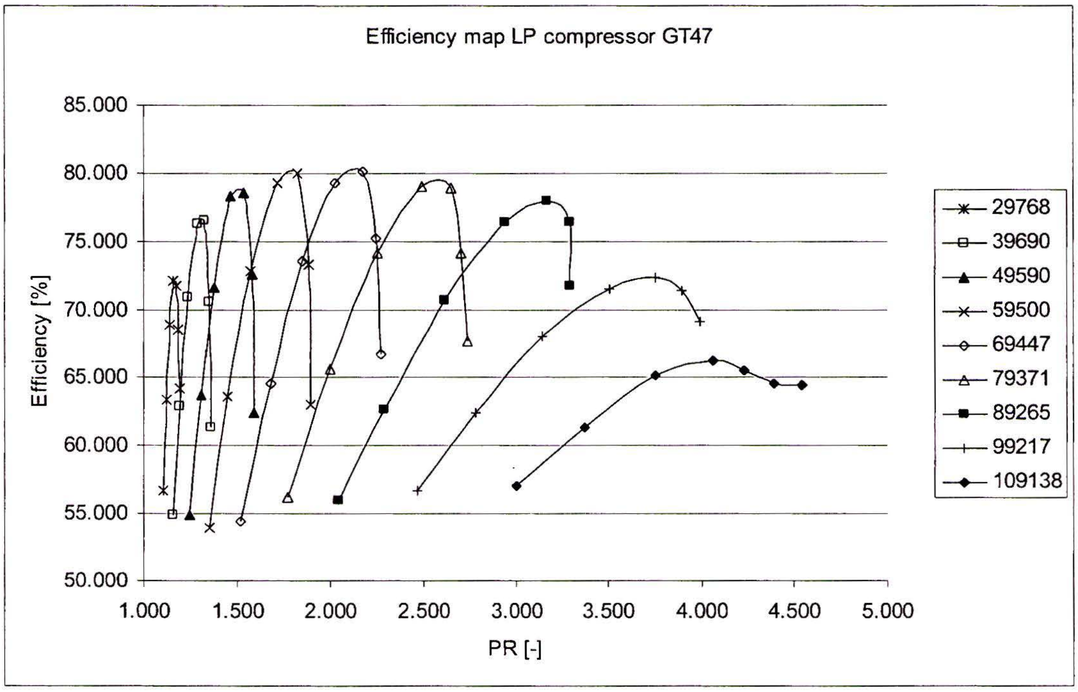
*Table T2.2.3; Variable estimates and assumptions*

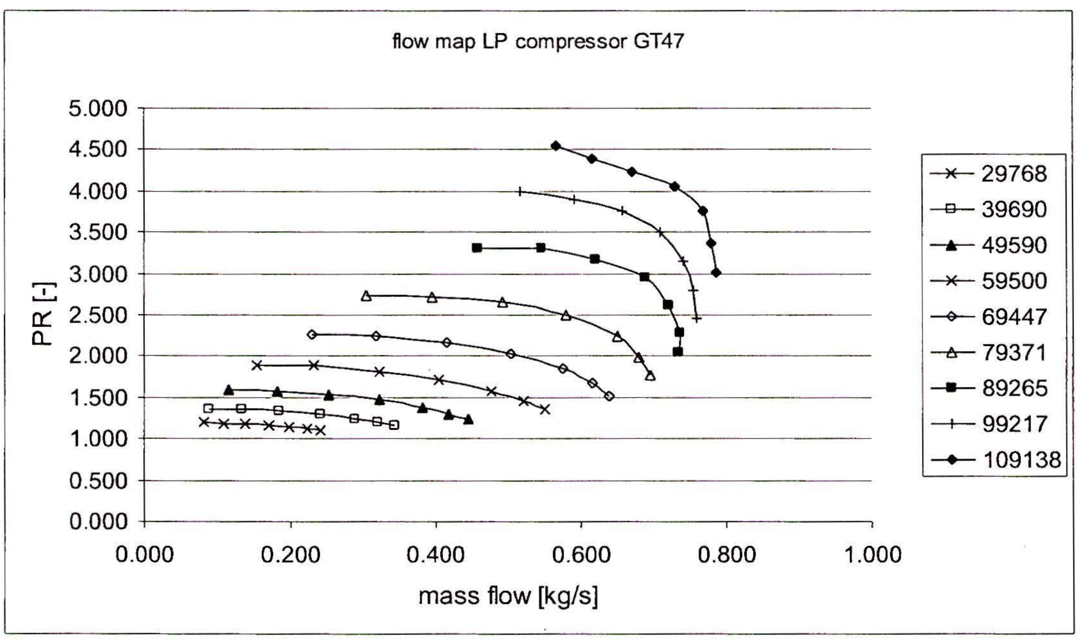
Turbo maps

*GT_47 with 102/53 compressor (Garett);*

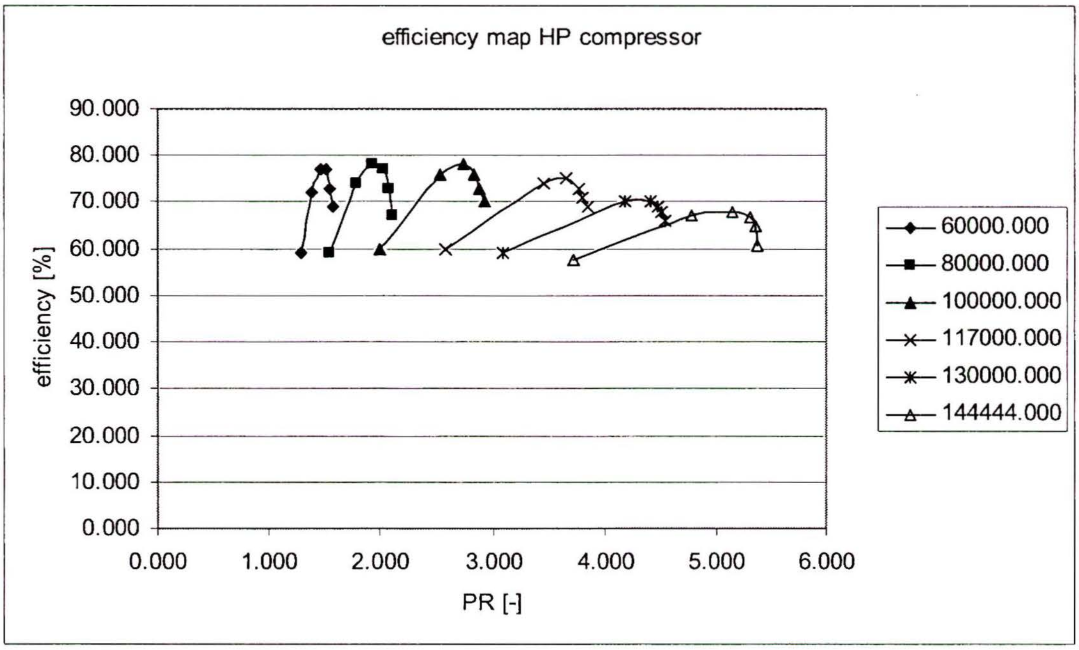


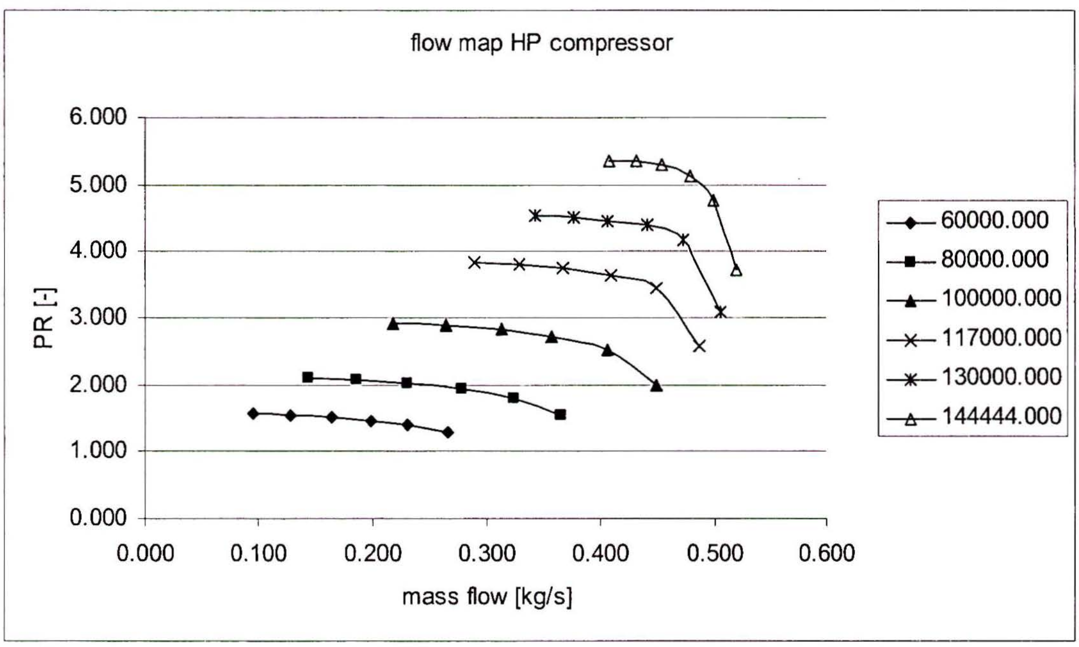




Efficiency map LP compressor GT47
Efficiency [%]
85.000
80.000
75.000
70.000
65.000
60.000
55.000
50.000
1.000
1.500
2.000
2.500
3.000
3.500
4.000
4.500
5.000
PR [-]
29768
39690
49590
59500
69447
79371
89265
99217
109138


flow map LP compressor GT47
PR [-]
5.000
4.500
4.000
3.500
3.000
2.500
2.000
1.500
1.000
0.500
0.000
0.000
0.200
0.400
0.600
0.800
1.000
mass flow [kg/s]
29768
39690
49590
59500
69447
79371
89265
99217
109138

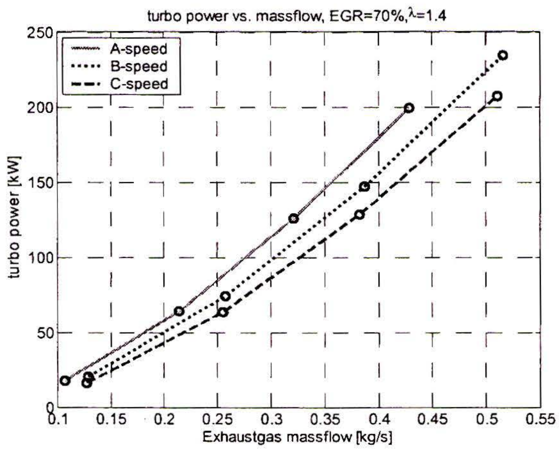
*Holset HE431 VTG*

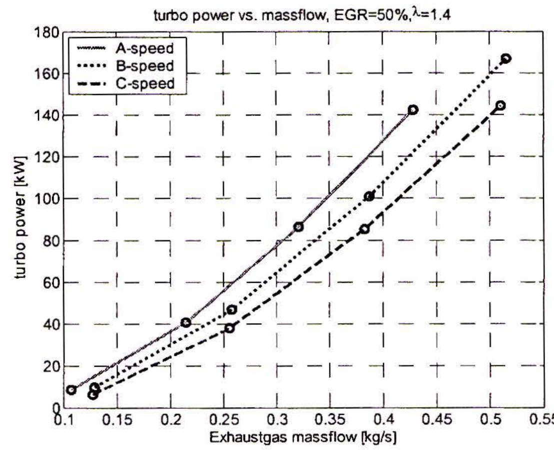


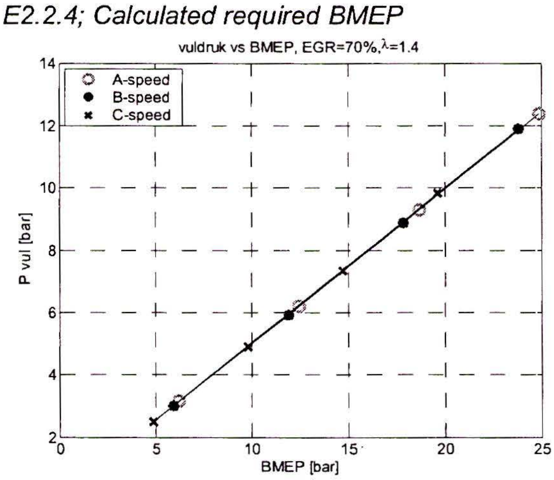
## Plots

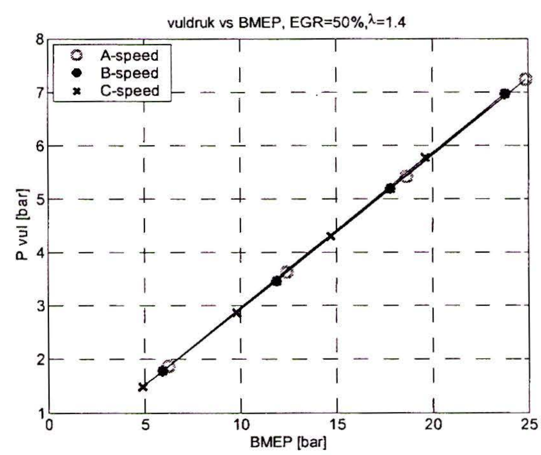
### *E2.2.3; Calculated required Turbo power*



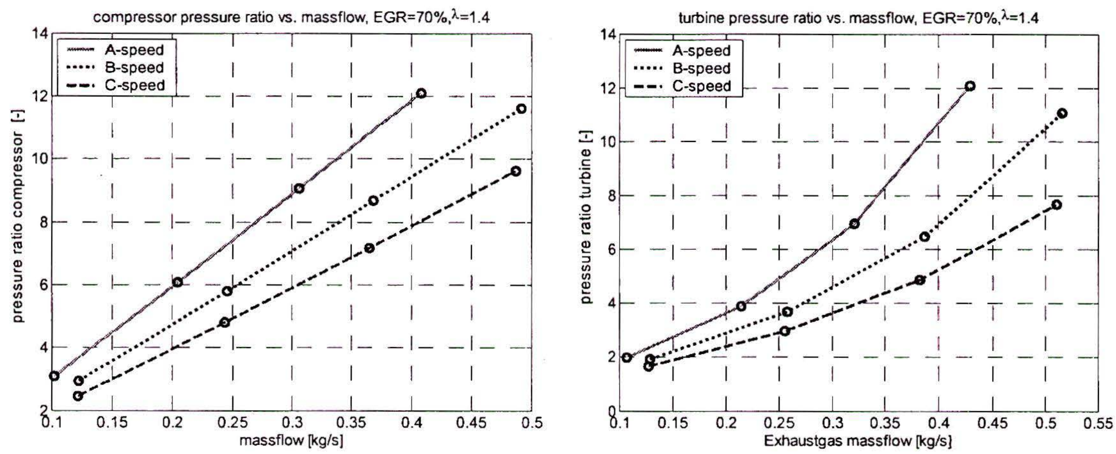




### *E2.2.4; Calculated required BMEP*

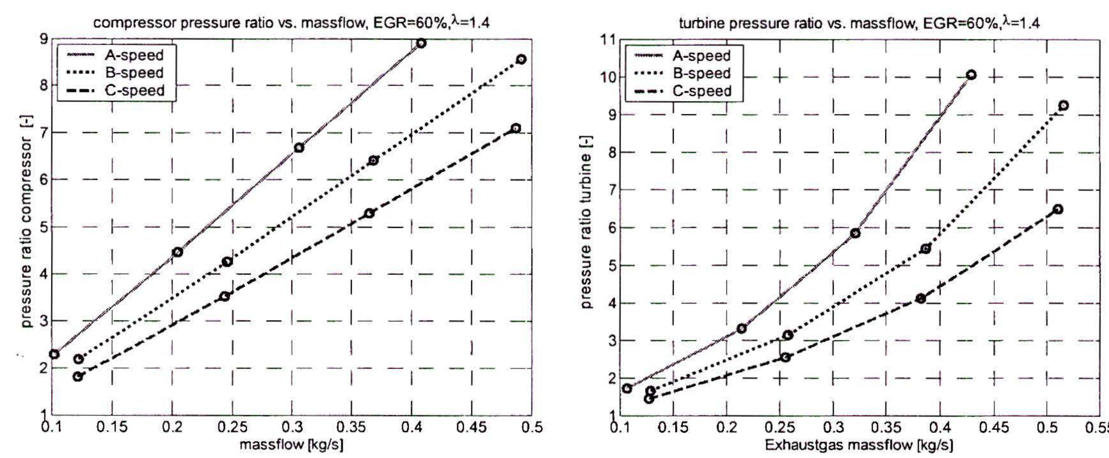


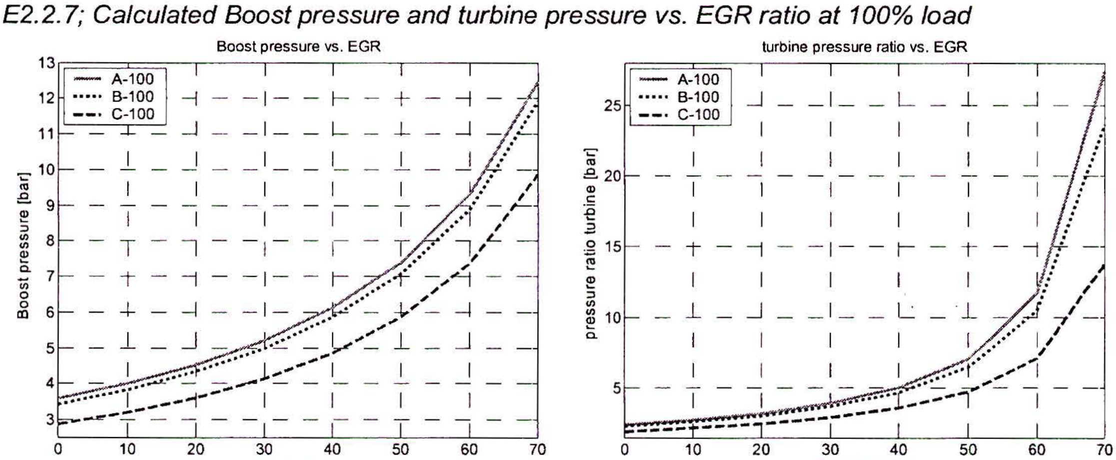
*E2.2.5; Calculated Turbo maps with high turbine efficiency 80% (75% compressor)*




*E2.2.6; Calculated Turbo maps 60% EGR*




*E2.2.7; Calculated Boost pressure and turbine pressure vs. EGR ratio at 100% load*



## Appendix 2.3 Combustion process at High EGR;

In the GT-power models use is made of the so-called Wiebe-combustion model. The Wiebe-combustion model represents the combustion process. The combustion process is of influence on the prediction of engine efficiency, cylinder peak pressure and exhaust pressure and temperature. Exhaust pressure and temperature are of influence on the amount of energy delivered to the turbine, which in turn is of influence on boost-pressure and thus the charging of the engine. As already indicated the charging of the engine is critical to achieve high levels of EGR. Therefore it is important to construct the Wiebe-model as close as possible to the true combustion process. In case of high-EGR it is expected that the level of EGR and/or FS is of influence on the combustion process (see chapter 2). Therefore the influence of EGR and FS on the combustion process is investigated with the help of measurement data from earlier conducted measurements. In addition a small literature-study is performed on this subject.

### Available measurement data

The available measurement data is the result of earlier research by TNO conducted on a DAF MX, heavy-duty diesel engine without turbocharger. Unfortunately the body of measurement-data comprising High EGR (+30%) is only limited to low load engine-operating points. Below an overview is given of the available measurement data. From each measurement the cylinder-pressure and injection-pressure is available and for some also injector needle-lift.

Available measurement data [TNO] (pressure traces, atmospheric DAF MX engine):

| Speed; BMEP | ±0% EGR | 18% EGR | 23% EGR | 27% EGR | 40% EGR | 50% EGR | 70% EGR | 73% EGR |
|---|---|---|---|---|---|---|---|---|
| A; 1bar | X | | | | X | | X | X |
| A; 2bar | X | | | | X | X | X | |
| B; 4.8 bar | X | X | X | X | | | | |

*Table T2.3.1 Available measurement-data at High EGR*

### Calculation

With the use of GT-power calculations are performed in order to determine the Rate of Heat Release (ROHR). The so-called "EngHeatRel" object in GT-power calculates a heat release curve from measured cylinder pressure data. In this case the gross apparent ROHR is calculated, based on the net apparent ROHR and assumed heat transfer. As such the cumulative (gross) ROHR equals 1 (100%) at complete combustion of the injected fuel. In GT-power a so-called 'Cumulative Heat Release Adjustment' is applied, which adjusts the level of heat transfer (heat transfer multiplier) in order to try to obtain a cumulative heat release of 1.0. According to GT-power "The heat transfer multiplier is used as the adjustment factor because heat transfer is generally a true unknown in the system." A filter is applied in order to remove measurement disturbance from the measurement-data. However in order prevent any loss of information due to severe filtering only a mild filter is used here, resulting in ROHR curves which are not completely smooth.

Based on the apparent ROHR and the cumulative ROHR the start of combustion (SOC) and the 1, 10, 50 and 90% burnt moments are given in degrees CA relative to TDC. Ignition delay and combustion duration are calculated and used in comparison for different levels of EGR and FS.
Ignition delay is defined as the difference between start of combustion and start of injection in °CA. (ignition delay = |SOC – SOI|). The start of combustion (SOC) is given by GT-power as "the crank angle (with respect to TDC of the power stroke) at which fuel is first consumed in combustion". The resolution of the crank angles is 0.5°. As such it may be possible that the calculated 1% burnt is less than 0.5°CA before the start of combustion.

## Results

Measurement data and results from calculations are given below for each measurement series; speed A-1bar BMEP, speed A-2bar BMEP and speed B-4.8bar BMEP. Besides the EGR level, the FS level is given, also see paragraph 4.2 and appendix 2.1.

$$FS = \frac{(AF_{stoich} + 1) \cdot EGR/_{100}}{AF_{real} + 1} \qquad \text{A 2.3.1}$$

- *Speed A, 1bar BMEP:*

In figure E1.3.1 the cylinder pressure is given for different EGR/ FS levels. It is clear that as a result of increasing EGR and FS the peak cylinder pressure is reduced.

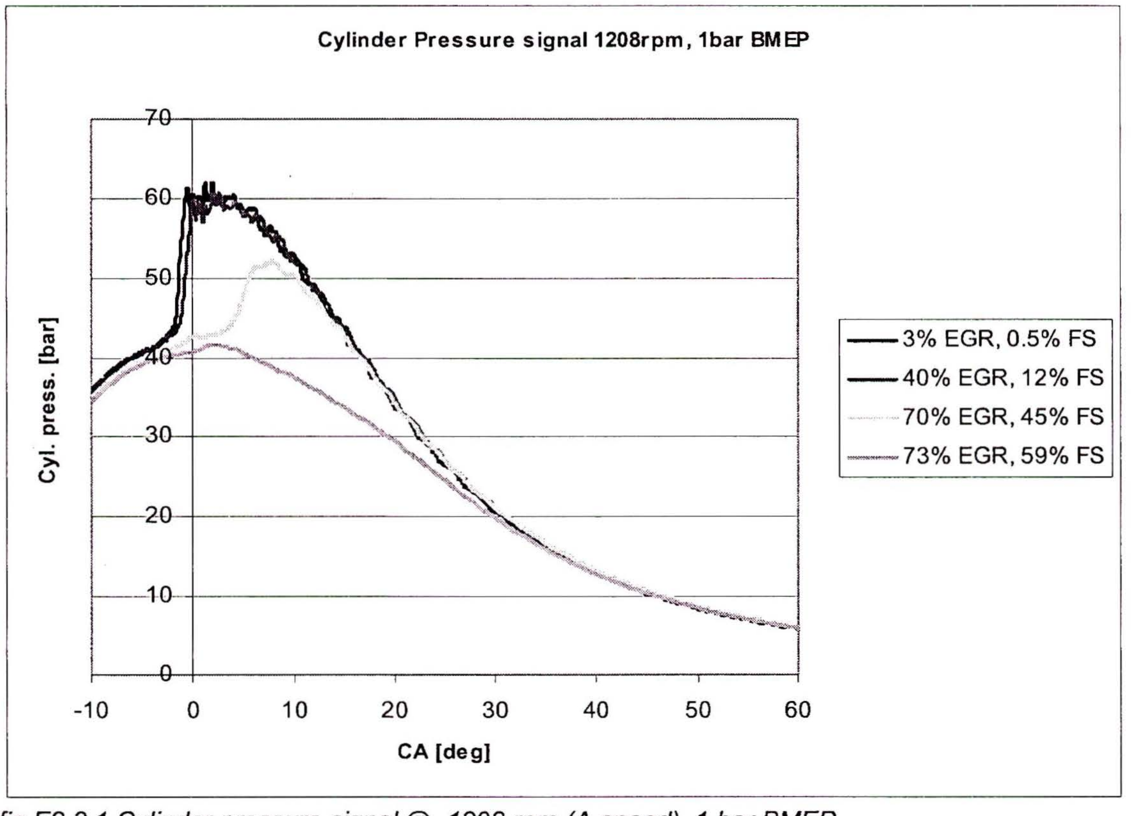



*fig E2.3.1 Cylinder pressure signal @ 1208 rpm (A-speed), 1 bar BMEP*

In figure E2.3.2 the injection line-pressure signal is given as well as the injector needle lift signal at different EGR and FS levels. As expected the injection pressure and lift signals are practically the same in all cases and vary only a little, creating equal conditions. The slight pressure rise at 40°CA may be due to pressure waves in the fuel-line.

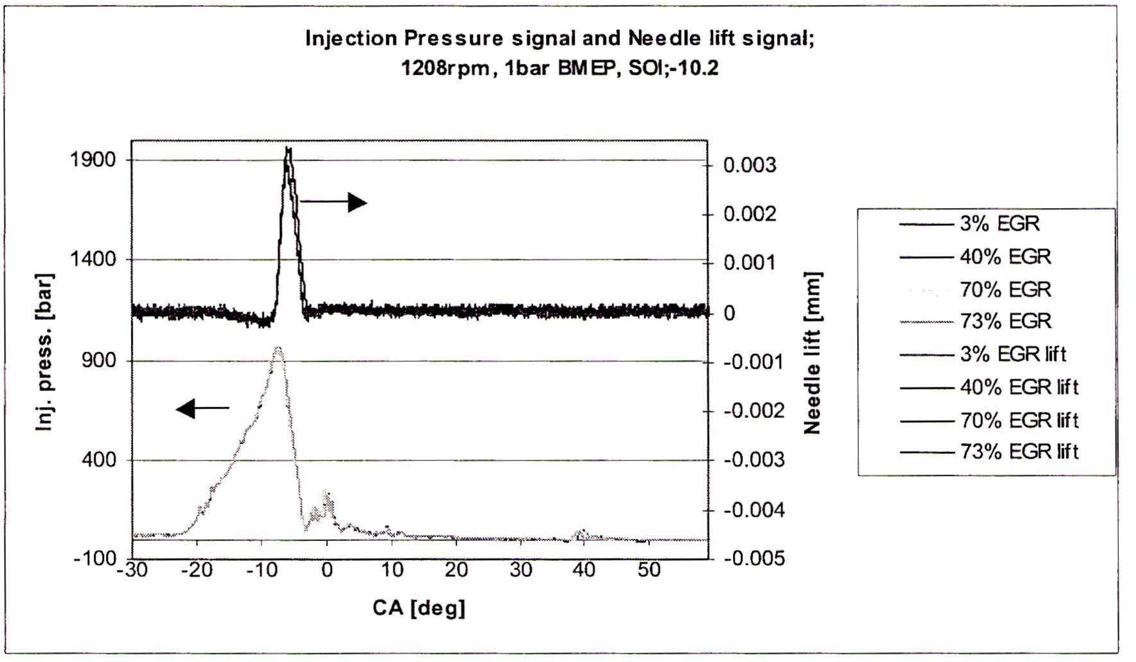



*fig E2.3.2 Fuel injection pressure signals and injection needle lift signals @ 1208 rpm (A-speed), 1 bar BMEP*

In figure E2.3.3 the normalized heatrelease rate is given for the different EGR and FS levels. As a consequence of increasing EGR & FS the heatrelease rate decreases, indicating a slower combustion process. The heatrelease shifts towards the expansion stroke at increasing EGR & FS level. At 70 and 73% EGR, combustion starts after the fuel-injection has ended meaning almost or complete pre-mixed combustion.
At 70 and 73% EGR the small hump before the main heatrelease could indicate a two stage ignition process as described by; [73] and also mentioned by [2, 72]. The two stage ignition process features a cool flame, followed by a 'hot flame' or high-temperature explosion where the reaction accelerates rapidly after ignition [2].

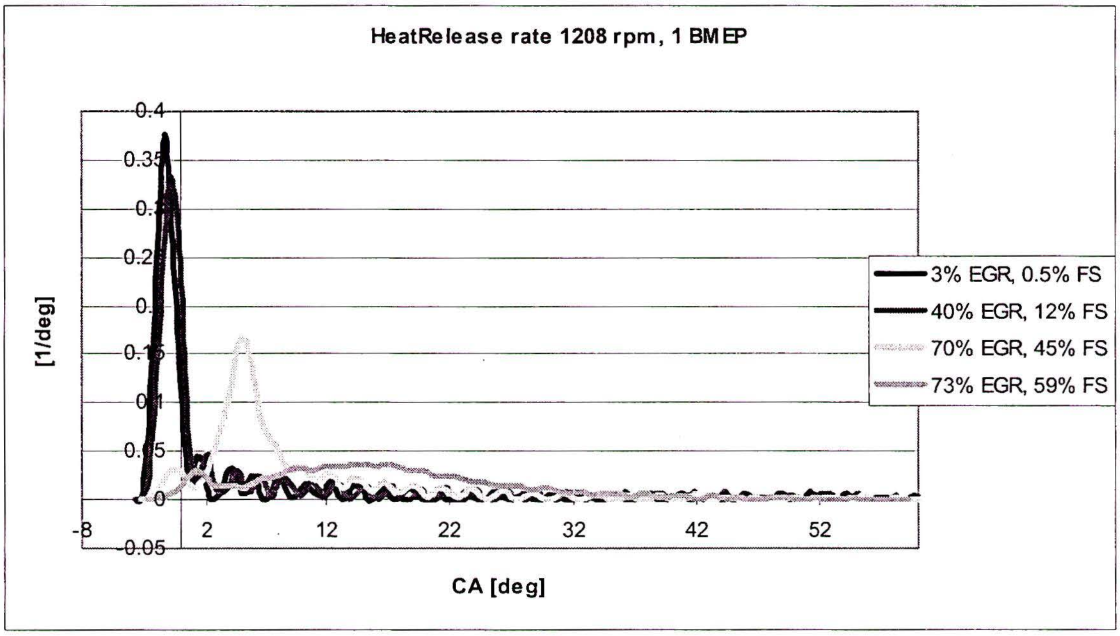



*fig E2.3.3 Normalized Heat Release @ 1208 rpm (A-speed), 1 bar BMEP*

Figure E2.3.4 shows the normalized cumulative heatrelease rate for different EGR and FS levels. It can be seen that at 73% EGR, combustion is incomplete as combustion ends at a normalized cumulative heatrelease level of less than 1. Also at increasing EGR & FS level the combustion process shifts towards the expansion stroke as earlier indicated. At 70% a late rapid increase of the cumulative heatrelease can be seen compared to combustion at lower EGR & FS levels this may be due to the large portion of pre-mixed combustion in comparison to the lower EGR & FS levels.

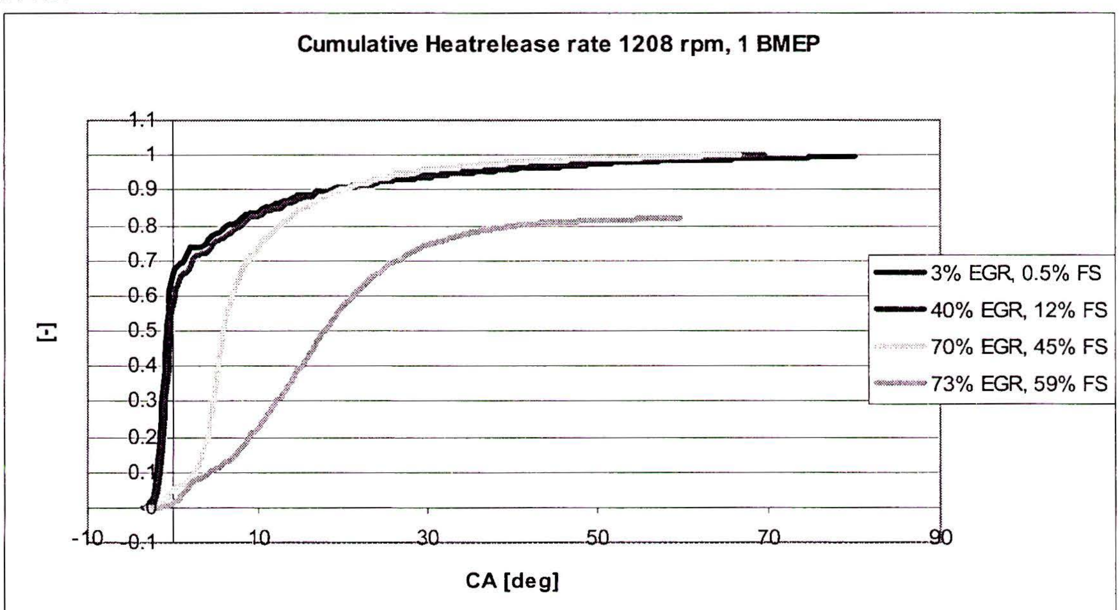



*fig E2.3.4 Cumulative normalized Heat Release @ 1208 rpm (A-speed), 1 bar BMEP*

In figure E2.3.5 combustion duration and ignition delay are given for different EGR & FS levels. It needs to be noted that at 73% EGR the combustion is incomplete and therefore the results regarding combustion duration up to 90% burnt may be unreliable and will therefore be ignored.

As can be seen ignition delay increases slightly at increasing EGR & FS level. Combustion duration for 10-50% burnt increases also at increasing EGR & FS, maximum is 2°CA between 3 and 70% EGR (0.5% - 44.5% FS). Combustion duration 10-90% seems to increase at increasing EGR & FS, except at 70% EGR where combustion duration 10-90% decreases. This may be due to the large portion of pre-mixed combustion as discussed above at figure E2.3.4. Maximum increase in combustion duration 10-90% burnt is 1.6° CA between 3 and 40% EGR (0.5% - 12.1% FS). Exact values can be found in table T2.3.2.

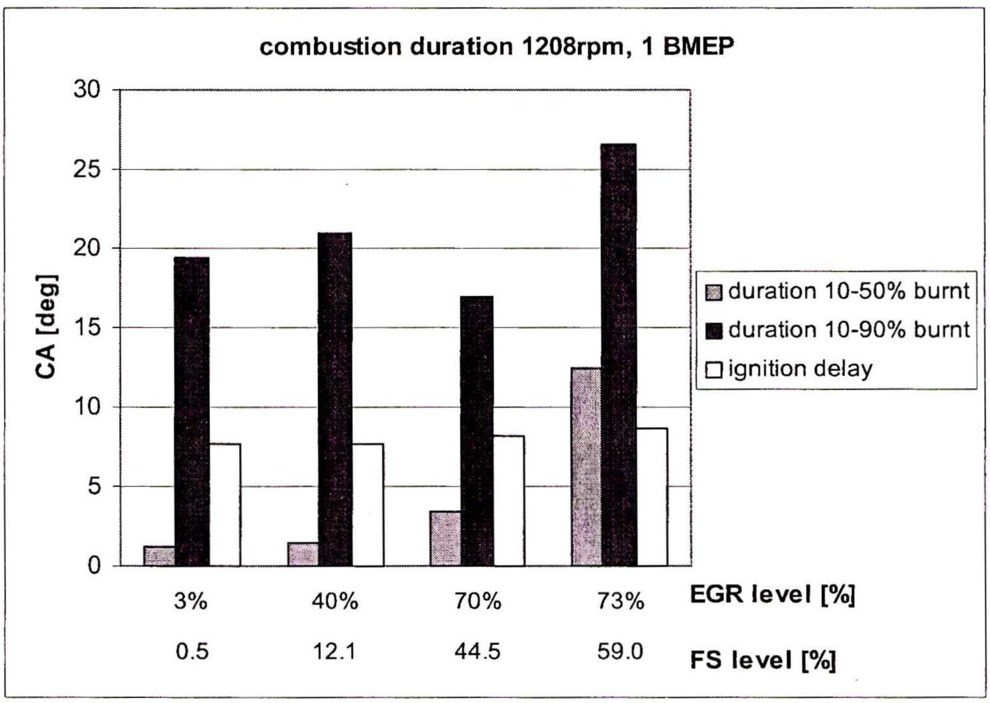



*fig E2.3.5 Combustion duration 10-50% burnt and 10-90% burnt @ 1208 rpm (A-speed), 1 bar BMEP*

| | | | | | |
|---|---|---|---|---|---|
| Speed | rpm | 1208 | 1208 | 1208 | 1208 |
| EGR | % | 2.8 | 39.95 | 69.93 | 72.84 |
| λ | | 5.69 | 3.45 | 1.61 | 1.25 |
| FS | % | 0.5 | 12.1 | 44.5 | 59.0 |
| SOI | °CA | -10.2 | -10.2 | -10.2 | -10.2 |
| M_fuel | mg/cycle | 25.62 | 25.71 | 26.6 | 30.23 |
| BMEP | bar | 1.05 | 1.12 | 1.12 | 0.93 |
| Avr. EGR | | 3% | 40% | 70% | 73% |
| SOC | °CA | -2.50 | -2.50 | -2.00 | -1.50 |
| 1% Burnt | °CA | -2.82 | -2.45 | -1.40 | -0.28 |
| 10% Burnt | °CA | -2.05 | -1.68 | 2.53 | 3.01 |
| 50% Burnt | °CA | -0.80 | -0.25 | 5.90 | 15.39 |
| 90% Burnt | °CA | 17.35 | 19.35 | 19.49 | 29.64 |
| duration 10-50 | °CA | 1.25 | 1.43 | 3.37 | 12.38 |
| duration 10-90 | °CA | 19.4 | 21.03 | 16.96 | 26.63 |
| ignition delay | °CA | 7.70 | 7.70 | 8.20 | 8.70 |

*Table T2.3.2 Measurement data and data regarding the combustion-process @ 1208 rpm (A-speed), 1 bar BMEP for four EGR & FS levels.*

- *Speed A, 2bar BMEP*

The results for the measurement series at speed A, 2bar BMEP are similar to those of above described measurement series at speed A, 1bar BMEP.

In figure E2.3.6 the cylinder pressure is given for different EGR/ FS levels. It is clear that as a result of increasing EGR and FS the peak cylinder-pressure is reduced.

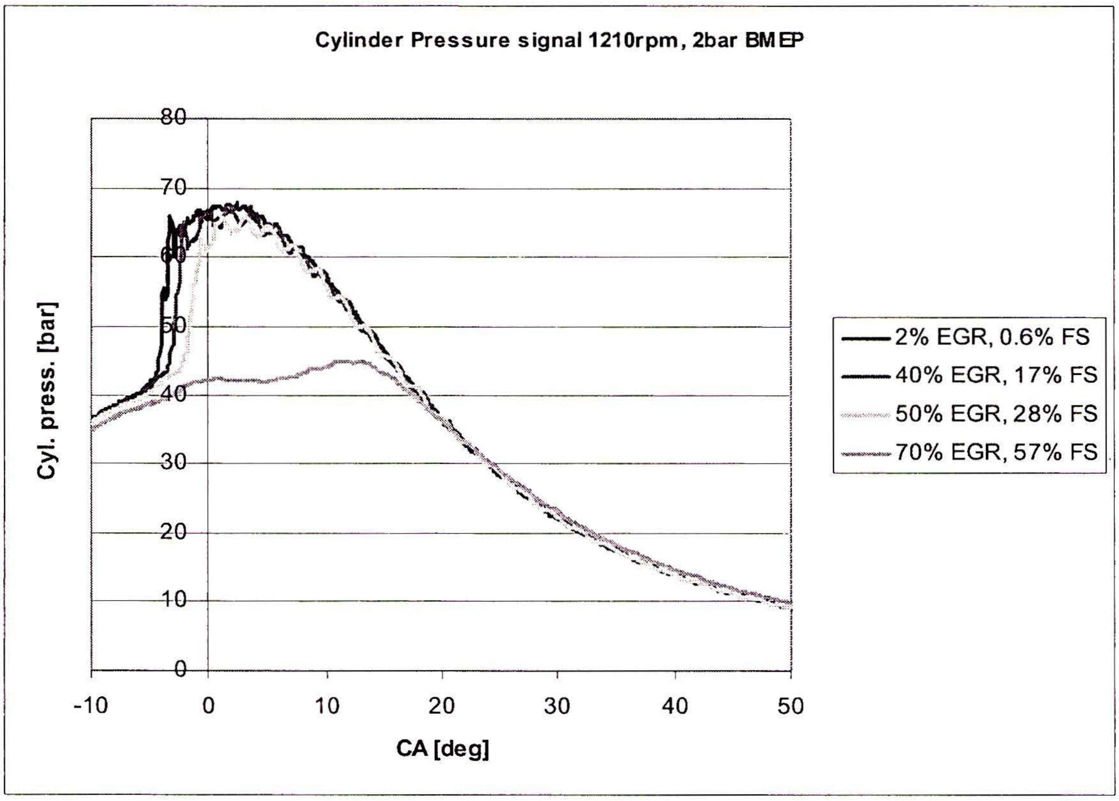



*fig E2.3.6 Cylinder pressure signal @ 1210 rpm (A-speed), 2 bar BMEP*

In figure E2.3.7 the injection line pressure signal is given at different EGR and FS levels. The pressure signal is practically the same in all cases and varies only a little, creating equal conditions. The slight pressure rise at 45°CA may be due to pressure waves in the line.

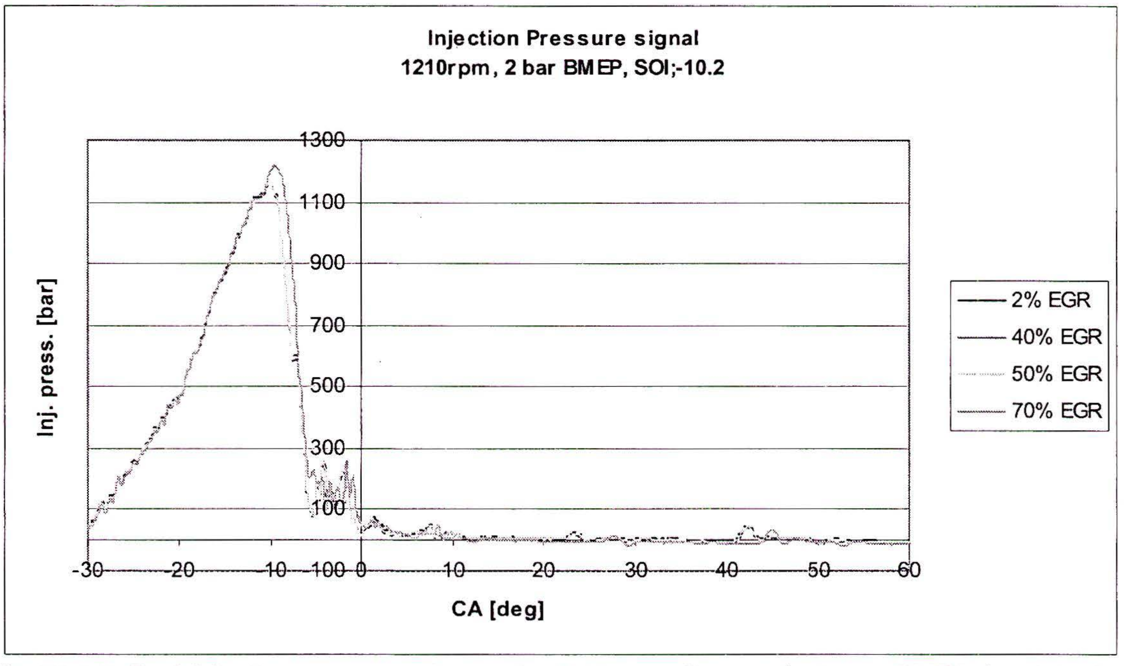



*fig E2.3.7 Fuel injection pressure signals @ 1210 rpm (A-speed), 2 bar BMEP (no injection needle lift signals available) (Needle lift not available)*

In figure E2.3.8 the heatrelease rate is given for the different EGR and FS levels. As a consequence of increasing EGR & FS levels, the heatrelease rate decreases indicating a slower combustion process. The heatrelease shifts towards the expansion stroke at increasing EGR and FS levels. At 70% EGR (57% FS) combustion starts after fuel injection has ended, meaning almost or complete stratified charge combustion.
Here also the two-stage ignition process can be witnessed at 70% EGR (57% FS) in the same way as described at figure E2.3.3.

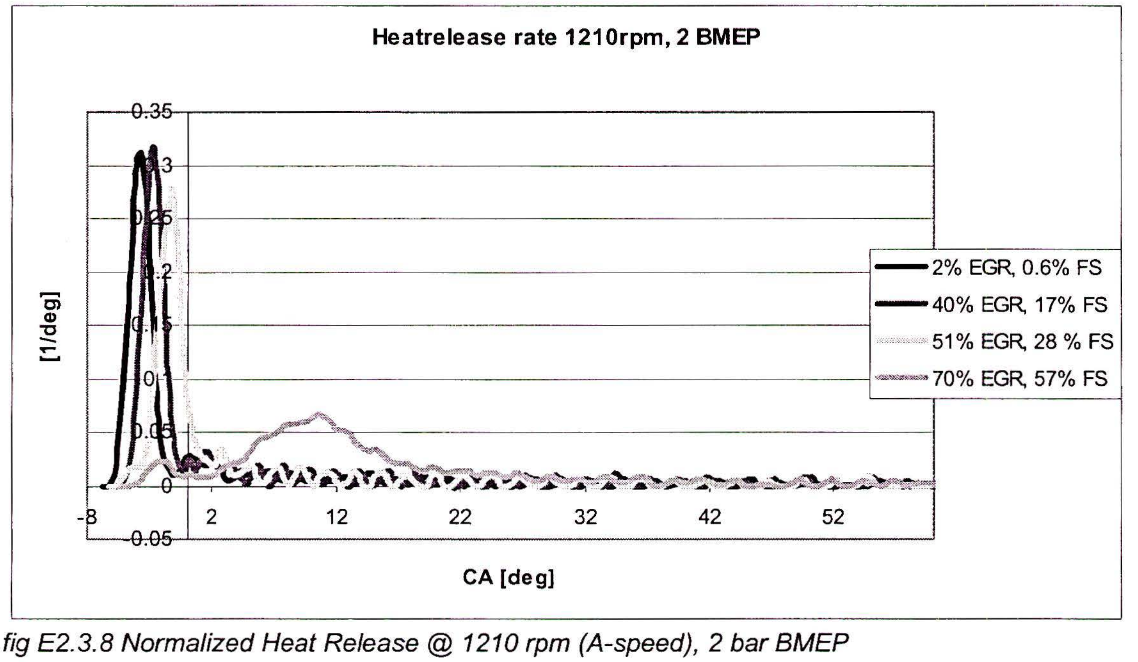



*fig E2.3.8 Normalized Heat Release @ 1210 rpm (A-speed), 2 bar BMEP*

Figure E2.3.9 shows the normalized cumulative heatrelease rate for different EGR and FS levels. Here it appears that at 70% EGR (57% FS) combustion is not entirely complete (exhaust-valve opens at 90° CA ATDC). The combustion process shifts towards the expansion stroke at increasing EGR and FS, however shift is small (approximately 3.5°) up to 50% EGR (28% FS). The combustion process shifts more substantially (+15°) at 70% EGR (57% FS)

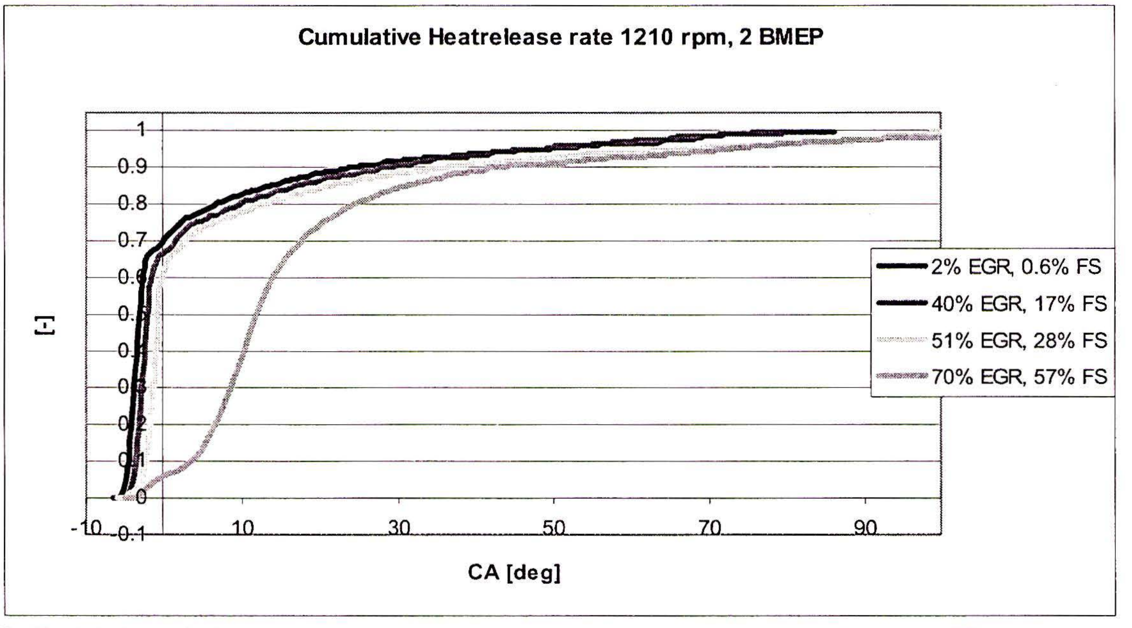



*fig E2.3.9 Cumulative normalized Heat Release @ 1210 rpm (A-speed), 2 bar BMEP*

In figure E2.3.10 combustion duration and ignition delay are given for different EGR & FS levels. At 70% EGR (57% FS) the combustion is incomplete and therefore the results regarding combustion duration up to 90% burnt may be unreliable and will be ignored.

As can be seen that ignition delay increases slightly at increasing EGR & FS level. Combustion duration, both for 10-50% burnt and 10-90% burnt increase at increasing EGR & FS. Maximum increase in combustion duration, 10-90% burnt, between 2% and 51% EGR (0.6% - 27.9% FS) is 8.5°CA. Exact values are given in table T2.3.3.

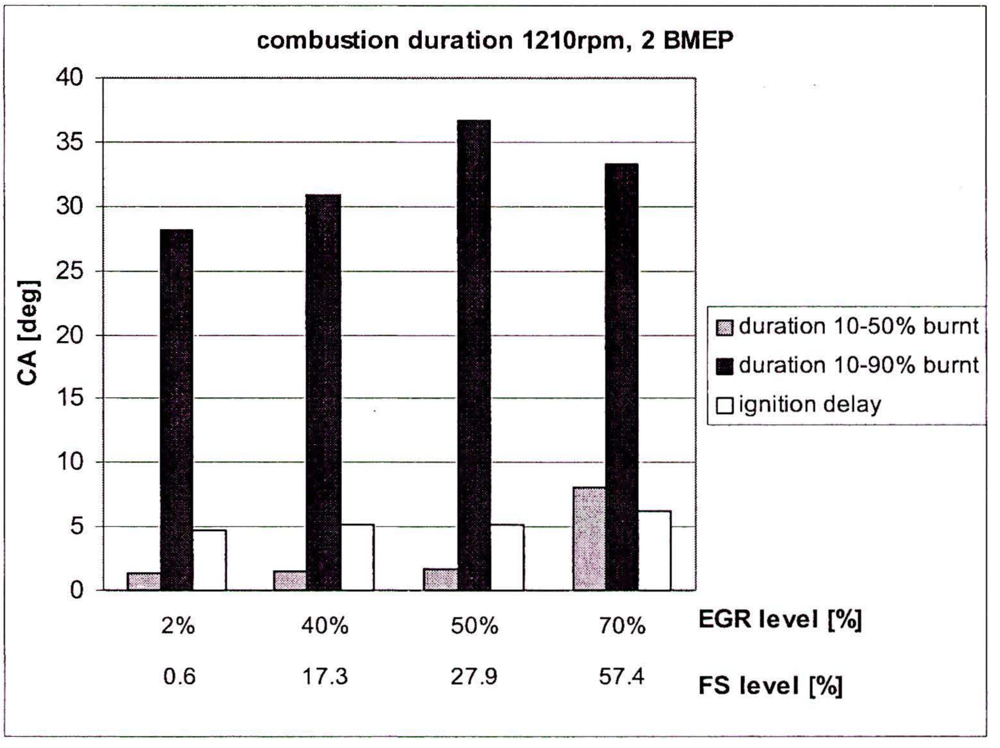



*fig E2.3.10 Combustion duration 10-50% burnt and 10-90% burnt @ 1210 rpm (A-speed), 2 bar BMEP*

| Speed | rpm | 1210 | 1210 | 1210 | 1210 |
|---|---|---|---|---|---|
| EGR | % | 2.3 | 39.9 | 51.4 | 68.1 |
| λ | | 3.9 | 2.4 | 1.9 | 1.2 |
| FS | % | 0.6 | 17.3 | 27.9 | 57.4 |
| SOI | °CA | -10.2 | -10.2 | -10.2 | -10.2 |
| M_fuel | mg/cycle | 36.41 | 36.46 | 36.36 | 42.33 |
| BMEP | bar | 2.02 | 2.12 | 2.14 | 2.13 |
| Avr. EGR | | 2% | 40% | 50% | 70% |
| SOC | °CA | -5.50 | -5.00 | -5.00 | -4.00 |
| 1% Burnt | °CA | -5.42 | -4.48 | -4.50 | -2.48 |
| 10% Burnt | °CA | -4.65 | -3.67 | -2.43 | 3.57 |
| 50% Burnt | °CA | -3.23 | -2.21 | -0.72 | 11.63 |
| 90% Burnt | °CA | 23.51 | 27.14 | 34.22 | 36.95 |
| duration 10-50 | °CA | 1.42 | 1.46 | 1.71 | 8.06 |
| duration 10-90 | °CA | 28.16 | 30.81 | 36.65 | 33.38 |
| ignition delay | °CA | 4.70 | 5.20 | 5.20 | 6.20 |

*Table T2.3.3 Measurement data and data regarding the combustion-process @ 1210 rpm (A-speed), 2 bar BMEP for four EGR & FS levels.*

- *Speed B, 4.8 bar BMEP:*

The results for the measurement series at speed B, 4.8bar BMEP are similar to those of above described measurement series at speed A, however at speed B, data is limited to 27% EGR and 20% FS.

In figure E2.3.11 the cylinder pressure is given for different EGR/ FS levels. It is clear that as a result of increasing EGR and FS the peak pressure is reduced. Given the limited maximum EGR and FS levels the reduction in peak pressure is limited to approximately 4 bar.

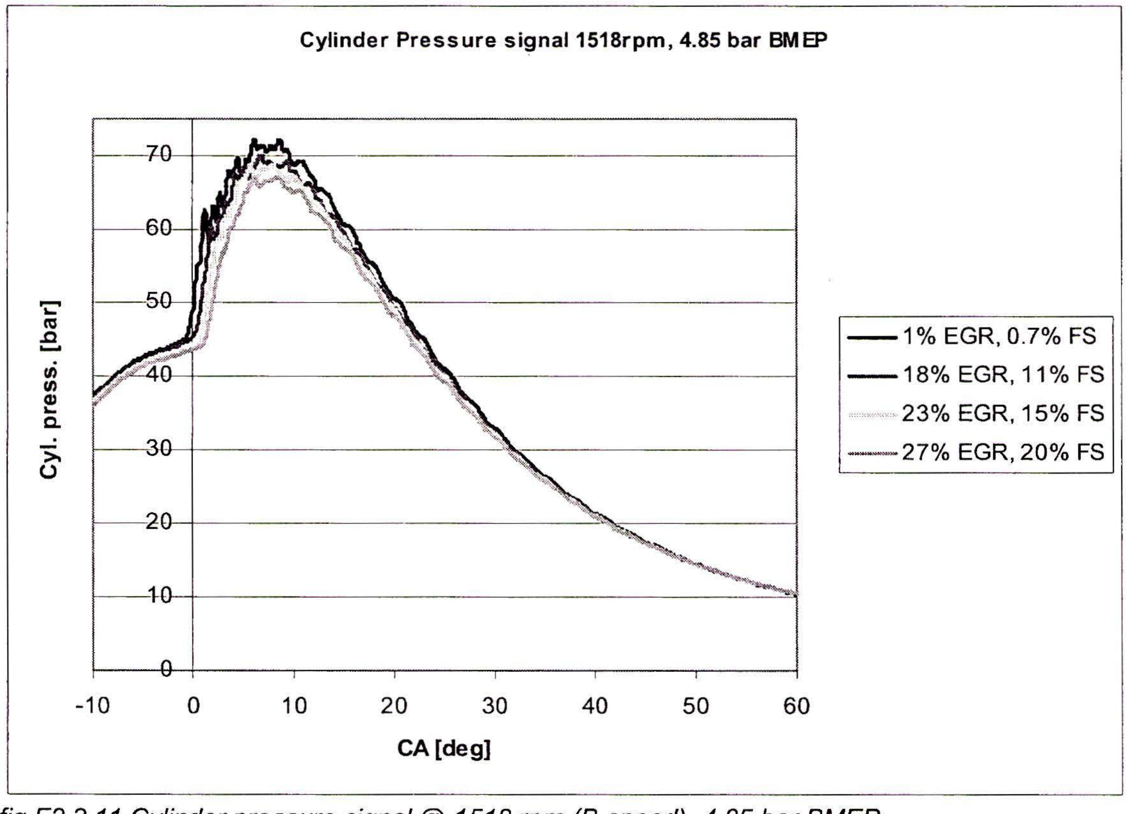



*fig E2.3.11 Cylinder pressure signal @ 1518 rpm (B-speed), 4.85 bar BMEP*

In figure E2.3.12 the fuel injection line pressure signal is given as well as the injector needle lift signal at different EGR and FS levels. As desired the pressure and lift signals are practically the same in all cases and vary only a little, creating equal conditions. The slight pressure rise at 30 and 50°CA may be due to pressure waves in the fuel-line.

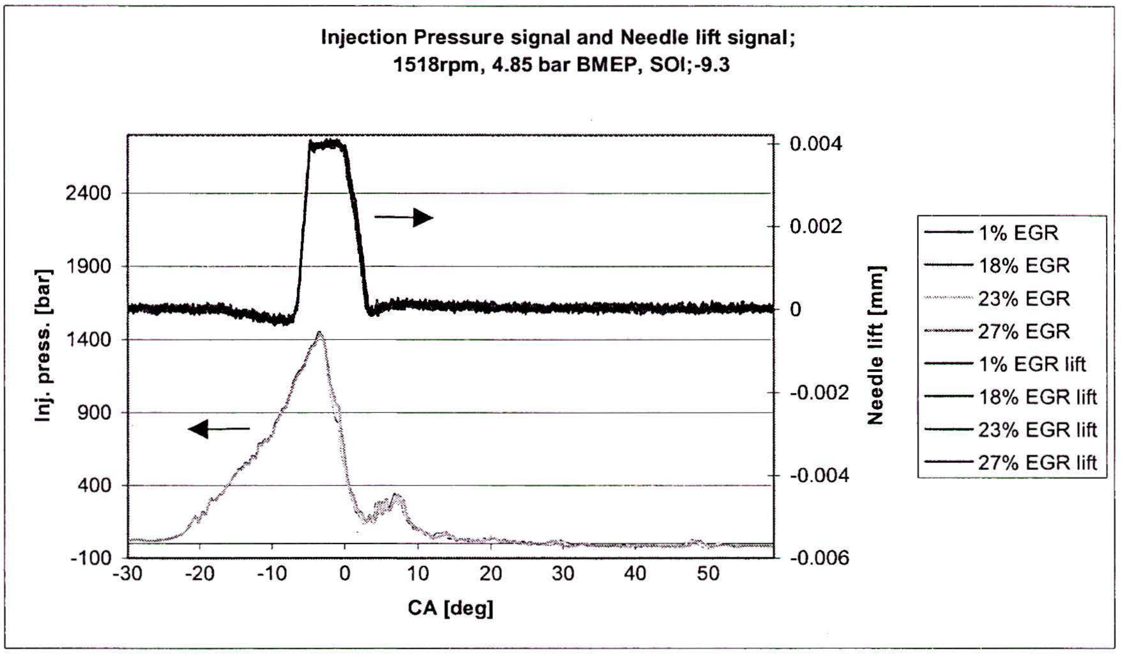



*fig E2.3.12 Fuel injection pressure signals and injection needle lift signals @ 1518 rpm (B-speed), 4.85 bar BMEP*

In figure E2.3.13 the heatrelease rate is given for the different EGR and FS levels. As a consequence of increasing EGR & FS the heatrelease rate decreases. Also the heatrelease shifts towards the expansion stroke at increasing EGR and FS level. The higher the EGR & FS level the larger the portion of pre-mixed combustion. Two-stage ignition process cannot be witnessed clearly here.

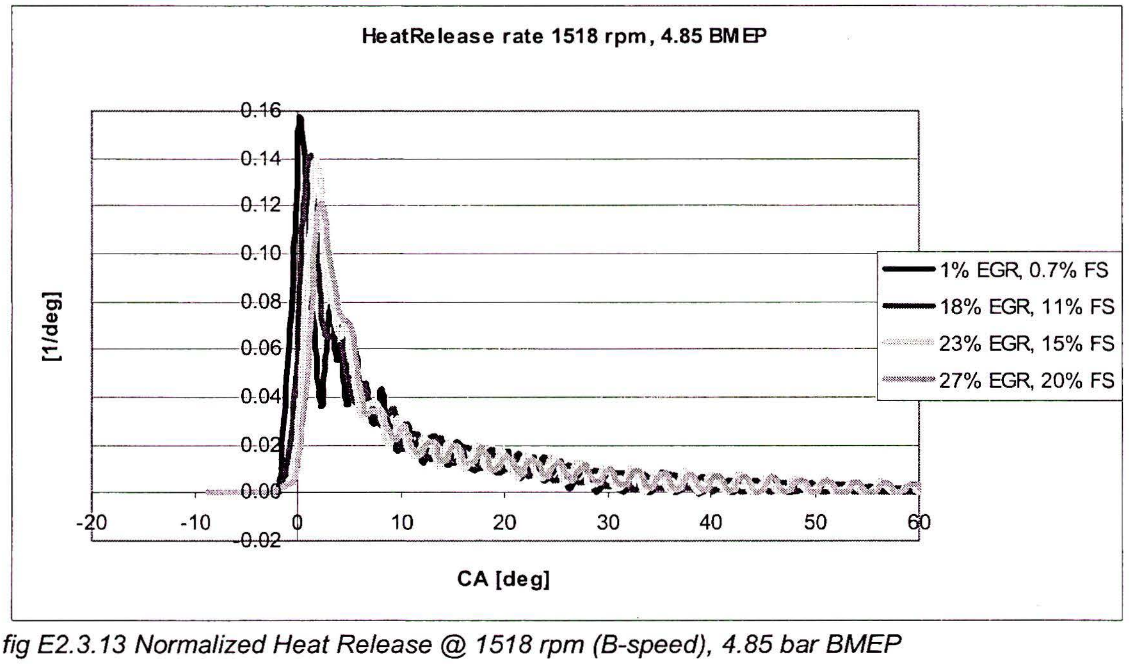



*fig E2.3.13 Normalized Heat Release @ 1518 rpm (B-speed), 4.85 bar BMEP*

Figure E2.3.14 shows the normalized cumulative heatrelease rate for different EGR and FS levels. The combustion process shifts towards the expansion stroke at increasing EGR and FS.

However the shift in combustion process is small, approximately 2° between 1% and 27% EGR (0.7% - 20% FS) at start of combustion and increases during the combustion process, as a consequence of slower combustion at increasing EGR & FS.

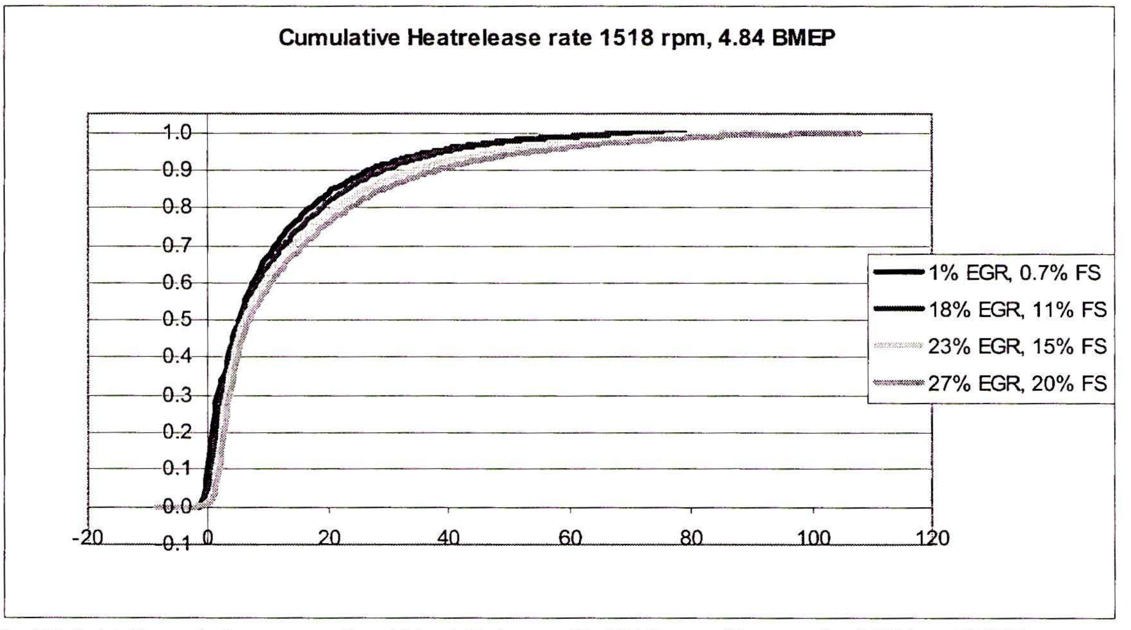



*fig E2.3.14 Cumulative normalized Heat Release @ 1518 rpm (B-speed), 4.85 bar BMEP*

In figure E2.3.15 combustion duration and ignition delay are given for different EGR & FS levels.

As can be seen ignition delay increases slightly at increasing EGR & FS level, 1.5°CA between 1% and 27% EGR (0.7-19.9% FS). Combustion duration 10-50% burnt remains fairly constant at increasing EGR & FS level up to 27% EGR (19.9% FS).
Combustion duration 10-90% burnt increases at increasing EGR & FS. Maximum increase 10-90% burnt between 1% and 27% EGR (0.7-19.9% FS) is 9°CA. Exact values are given in table T2.3.4.

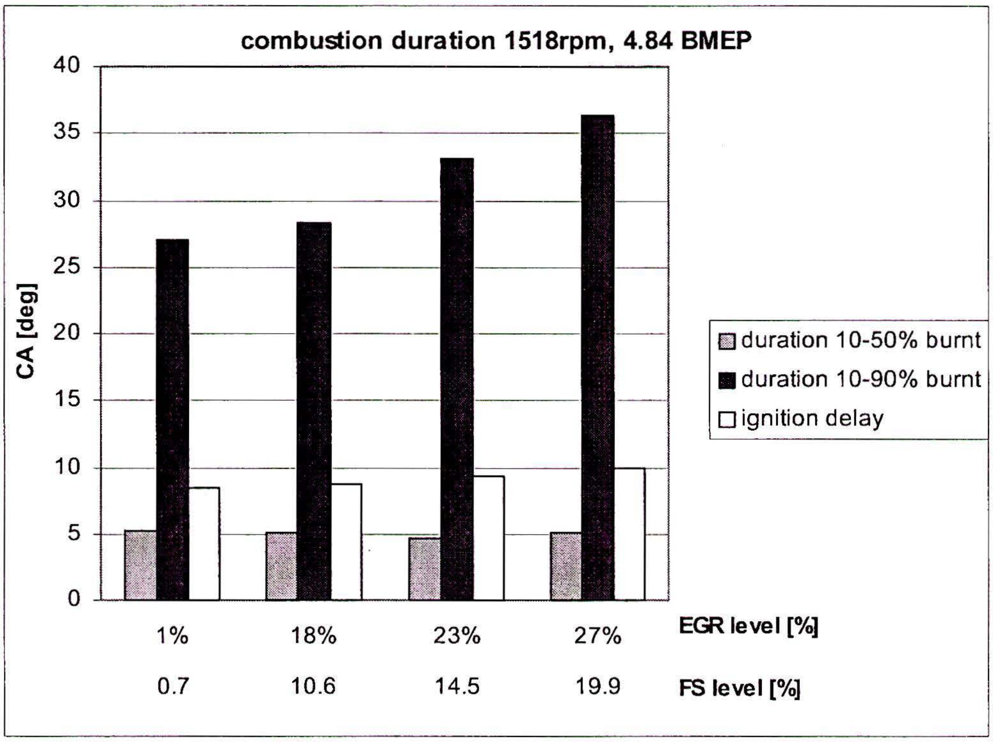



*fig E2.3.15 Combustion duration 10-50% burnt and 10-90% burnt @ 1518 rpm (B-speed), 4.85 bar BMEP*

| | | | | | |
|---|---|---|---|---|---|
| Speed | rpm | 1518 | 1518 | 1518 | 1518 |
| EGR | % | 1.41 | 17.75 | 22.6 | 27.32 |
| λ | | 2.11 | 1.72 | 1.6 | 1.4 |
| FS | % | 0.7 | 10.6 | 14.5 | 19.9 |
| SOI | °CA | -9.3 | -9.3 | -9.3 | -9.3 |
| M_fuel | mg/cycle | 69.16 | 69.56 | 69.94 | 72.46 |
| BMEP | bar | 4.86 | 4.86 | 4.84 | 4.88 |
| Avr. EGR | | 1% | 18% | 23% | 27% |
| SOC | °CA | -0.90 | -0.50 | 0.10 | 0.60 |
| 1% Burnt | °CA | -1.20 | -0.59 | -0.15 | 0.13 |
| 10% Burnt | °CA | -0.05 | 0.67 | 1.29 | 1.77 |
| 50% Burnt | °CA | 5.25 | 5.73 | 6 | 6.87 |
| 90% Burnt | °CA | 27.01 | 28.95 | 34.49 | 38.15 |
| duration 10-50 | °CA | 5.3 | 5.06 | 4.71 | 5.1 |
| duration 10-90 | °CA | 27.06 | 28.28 | 33.2 | 36.38 |
| ignition delay | °CA | 8.40 | 8.80 | 9.40 | 9.90 |

*Table T2.3.4 Measurement data and data regarding the combustion-process @ 1518 rpm (B-speed), 4.85 bar BMEP for four EGR & FS levels.*

**Trends in ignition delay and combustion duration;**
Combustion duration and ignition delay are plotted for the three measurement series in CA vs. FS level. Also combustion duration and ignition delay are plotted versus the mass of fuel injected.

In figure E2.3.16; ignition delay is plotted for three measurement series against FS. It can be seen that ignition delay increases at increasing FS level (increasing EGR, decreasing λ). Please note that SOI (Start of Injection) is 10.2° BTDC for both 1208rpm and 1210rpm and 9.3° BTDC at 1518rpm. At 1518rpm and 4.84 bar BMEP ignition delay is more sensitive to the increase in FS opposed to the other two measurement series (1208 and 1210 rpm). This may be due to higher engine speed and load and thus higher mass of fuel injected or different SOI.

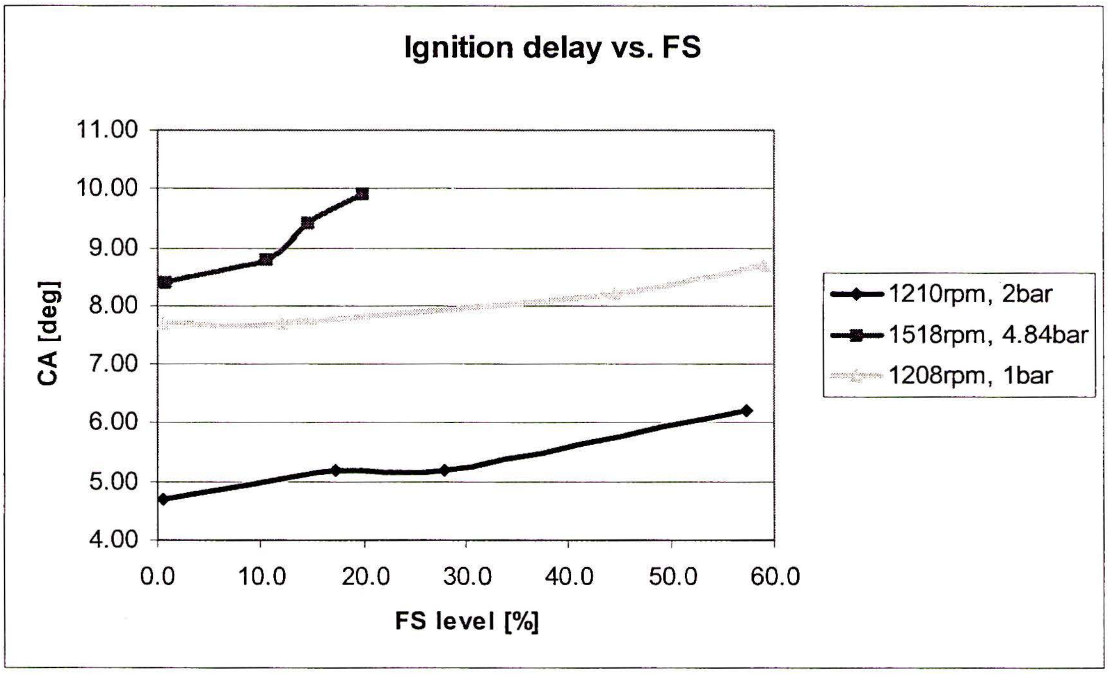



*fig E2.3.16 Ignition delay vs. FS (A; 2bar-B; 4.8bar- A; 1bar)*

In figure E2.3.17; combustion duration between 10 and 50% burnt is plotted for three measurement series against FS. Combustion duration 10 and 50% burnt remains fairly constant at given engine speed and BMEP and varying FS levels below 30. It can be seen that combustion duration increases more significantly at FS levels above 30% (@ 50 % EGR, λ=1.7), up to app. 10°CA at 55% FS (70%EGR, λ=1.30). Combustion duration 10 and 50% burnt at speed-B is 5° CA longer as opposed to speed-A and may depend on the mass of fuel injected, more fuel; longer combustion duration.

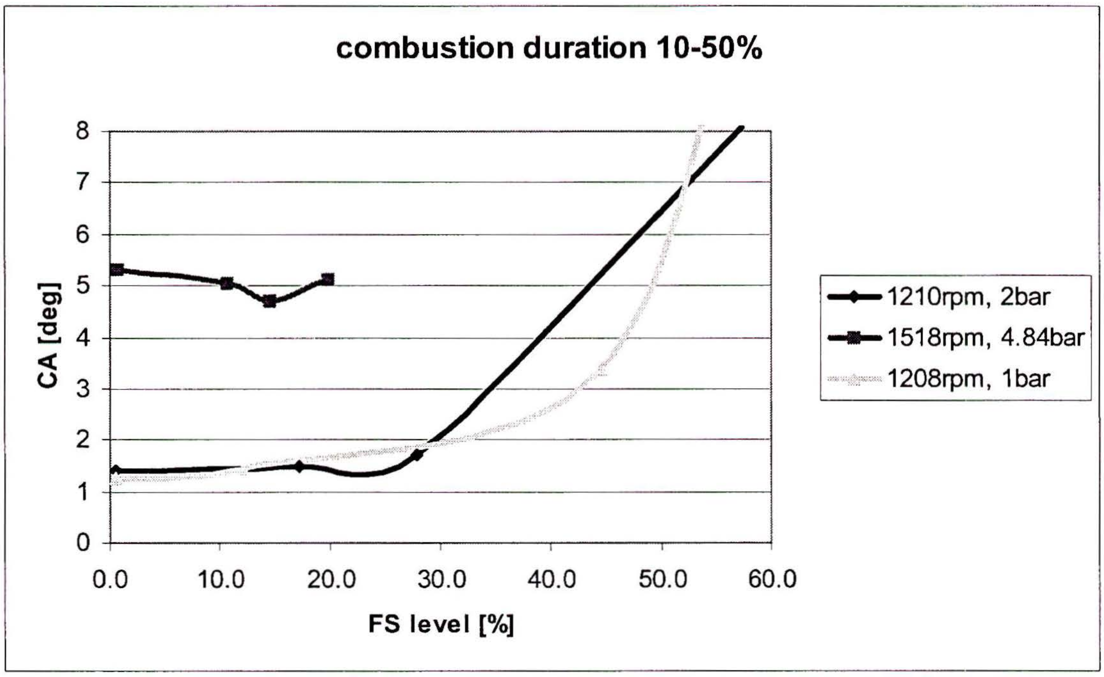



*fig E2.3.17 Combustion duration for 10-50 % burnt vs. FS (A; 2bar-B; 4.8bar- A; 1bar)*

In figure E2.3.18; combustion duration between 10 and 90% burnt is plotted for three measurement series against FS. As earlier indicated at high EGR and FS levels incomplete combustion was found and therefore the found combustion duration between 10-90% burnt may be inaccurate, this area is marked shaded in figure E2.3.18 and will be ignored.
A certain trend is visible for combustion duration between 10 and 90% burnt versus FS. The combustion duration (10-90% burnt) increases exponentially till a certain FS limit, after which it may decrease at increasing FS level for 1210 rpm and 1518rpm. The decrease in combustion duration at increasing FS level may be explained by the large degree of pre-mixed combustion at high FS levels.
However, at 1208rpm and very low BMEP (1bar) combustion duration (10-90% burnt) successively increases slightly and reduces slightly at increasing FS level. Furthermore combustion duration (10-90% burnt) for 1208rpm is shorter overall compared to 1210 and 1518rpm. One and other may be related to the low mass of fuel injected at 1208rpm and the very low load (1 bar BMEP).

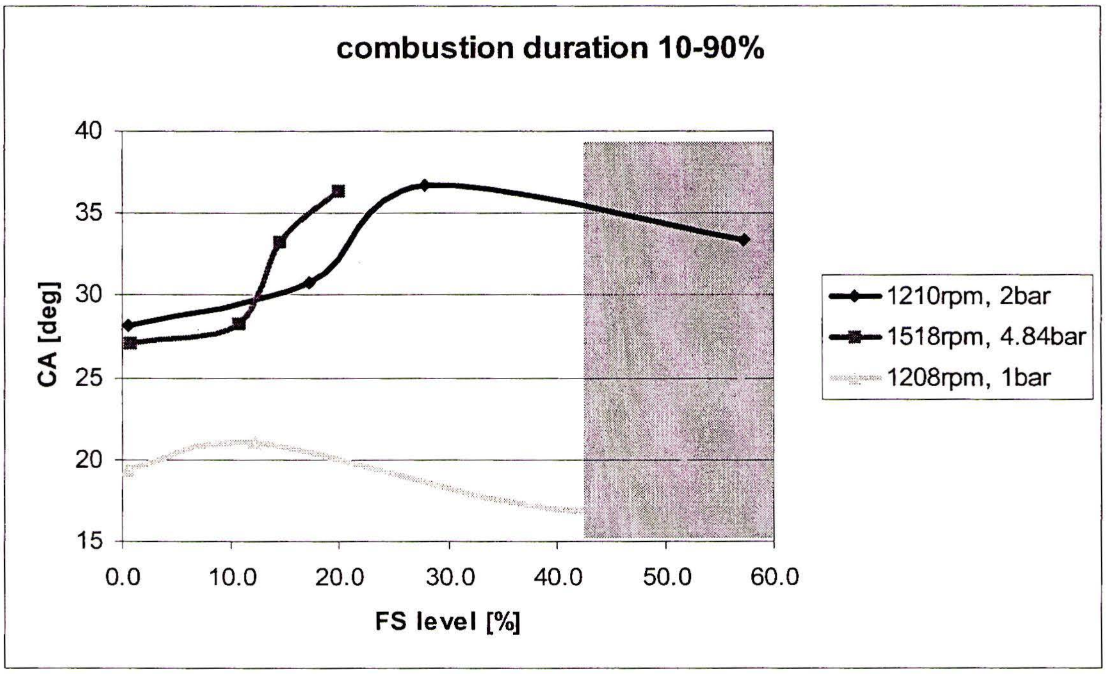



*fig E2.3.18 Combustion duration for 10-90 % burnt vs. FS (A; 2bar-B; 4.8bar- A; 1bar)*

At increasing EGR and FS level the mass of fuel that is injected is increased in order to maintain a constant BMEP. The increase in fuel mass compensates for the power loss as a consequence of increased EGR & FS.

In figure E2.3.19 combustion duration and ignition delay are plotted against the mass of fuel injected for measurement series of 1208rpm and 1 bar BMEP. As earlier identified ignition delay increases only slightly. A good correlation can be seen between combustion duration, 10-50% burnt, and the mass of injected fuel, as it increases almost linearly with the increase of mass of fuel. Combustion duration 10-90% burnt initially increases at increasing mass of fuel after which it decreases. This decrease may be as a consequence of the increase of pre-mixed combustion at increasing EGR & FS levels.
Combustion duration between 10-90% burnt at mass of fuel injected between 27 and 30 mg/cycle is to be ignored as incomplete combustion was found at the latest operating point.

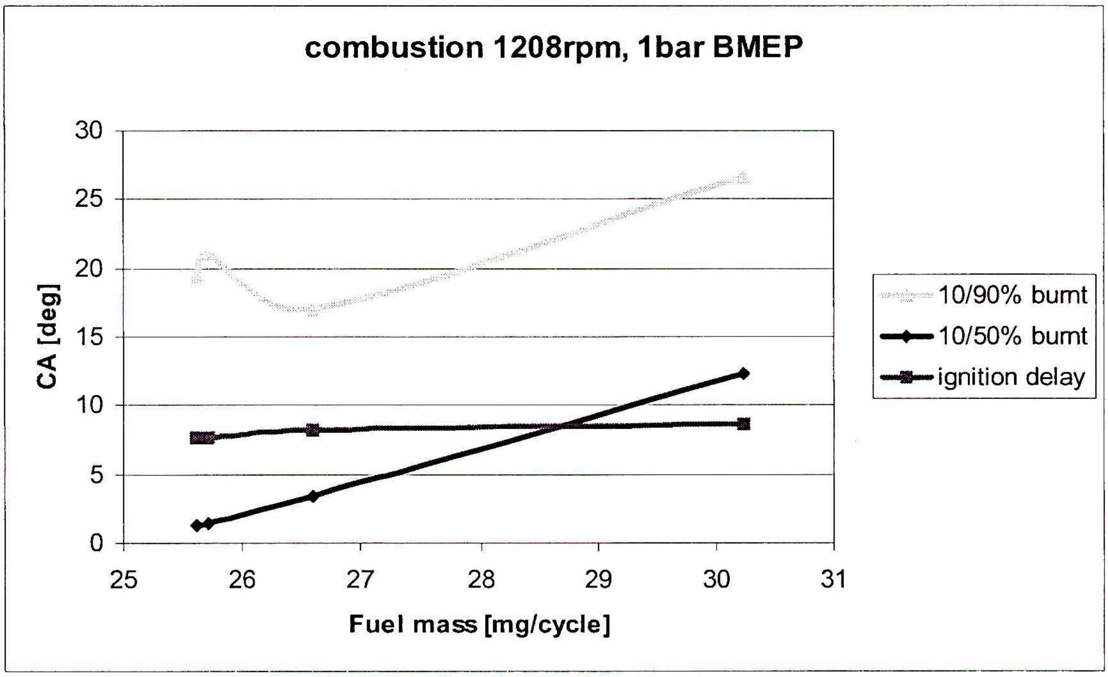



*fig E2.3.19 Combustion duration for 10-90 % and 10-50% burnt and ignition delay vs. Fuel mass (A; 1bar)*

In figure E1.3.20 combustion duration and ignition delay are plotted against the mass of fuel injected for measurement series of 1210rpm and 2 bar BMEP. Here also ignition delay increases only slightly. Again combustion duration, 10-50% burnt, increases almost linearly with the increase of mass of fuel.
No good correlation between combustion duration 10-90% burnt and mass of fuel injected can be concluded, as combustion duration (10-90% burnt) increases even as mass of fuel injected decreases. Combustion duration between 10-90% burnt at mass of fuel injected between 36.5 and 42 mg/cycle is to be ignored as incomplete combustion was found at the latest operating point.

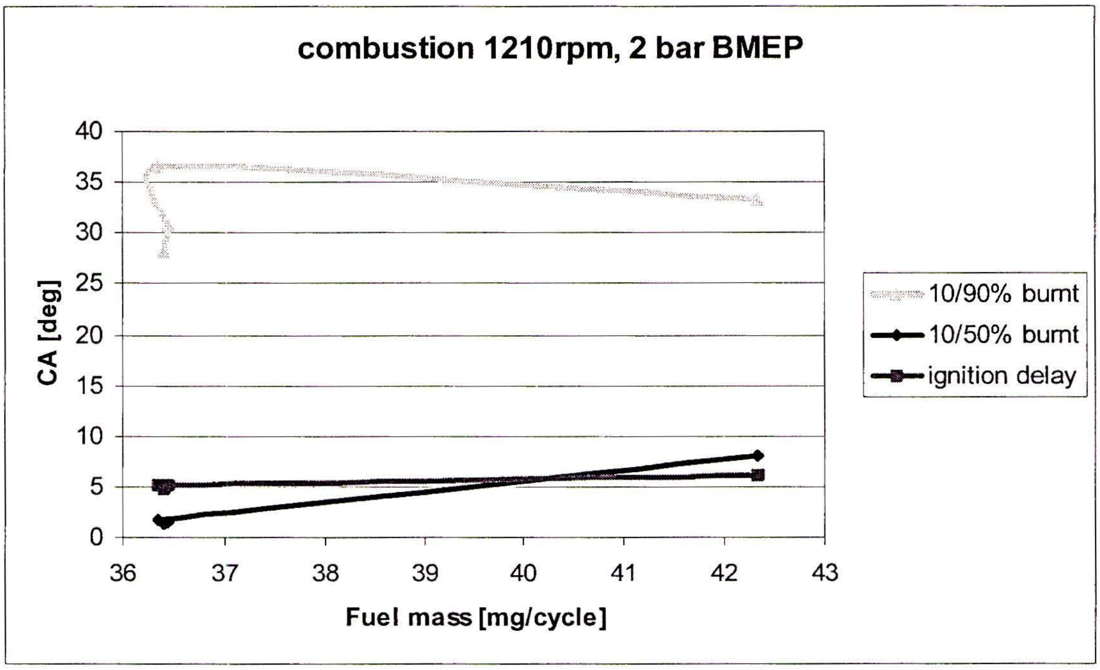



*fig E2.3.20 Combustion duration for 10-90 % and 10-50% burnt and ignition delay vs. Fuel mass (A; 2 bar)*

In figure E2.3.21 combustion duration and ignition delay are plotted against the mass of fuel injected for measurement series of 1518rpm and 4.8 bar BMEP. At this measurement series the maximum EGR and FS levels was limited to 27% and 20% respectively and therefore the mass of fuel injected is increased only slightly.
Ignition delay remains fairly constant. Combustion duration, 10-50% burnt, also remains almost constant. Combustion duration 10-90% burnt increases at increasing mass of fuel injected.

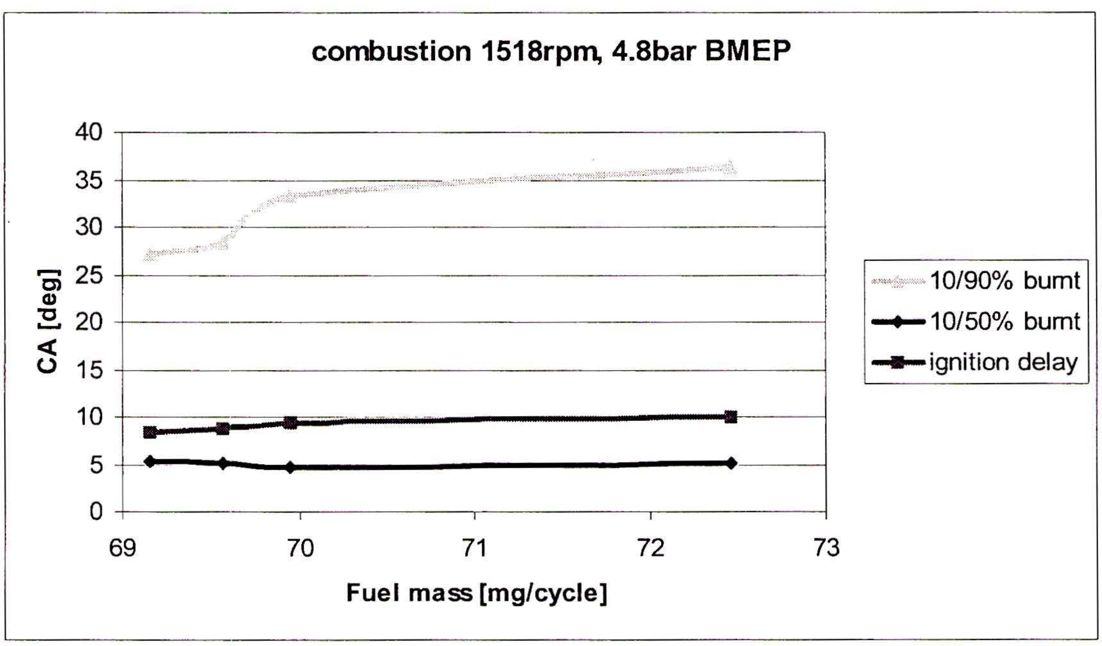



*fig E2.3.21Combustion duration for 10-90 % and 10-50% burnt and ignition delay vs. Fuel mass (B; 4.8 bar)*

Conclusions based on measurement data and performed calculations:
The available body of measurement data is limited and therefore can only give limited insight on the influence of EGR and FS on combustion. Furthermore in each measurement series, besides the increase in EGR and FS, the mass of injected fuel is increased at the same time, as such it is not possible to determine precisely to what extend combustion changes are the effect of FS or mass of fuel. In addition to this it also not possible to determine precisely to what extent combustion changes are the effect of EGR or $\lambda$.

Ignition delay is dependent on many factors among which are lambda and EGR level, also referred to as FS, and mass of fuel injected. At increasing FS and mass of fuel level, ignition delay increases, at certain (relative) constant engine speed and load and injection timing. It appears that at higher engine speed and load, and hence higher mass of fuel injected, ignition delay becomes more sensitive to FS. Maximum increase is found to be 1.5°CA from 0.7 to 20 % FS (1.4 to 27% EGR) at 1518rpm.

Combustion duration is sensitive to injected fuel mass, lambda and EGR level (FS).

Combustion duration between 10 and 50% burnt is relative insensitive to changes in EGR-level (and lambda) below 50% EGR and 30% FS. At FS levels of over 30% (+50% EGR) combustion duration between 10-50% burnt rises rapidly, up to 7°CA at 55% FS (70%EGR, $\lambda$=1.30).

Combustion duration between 10 and 90% burnt is more sensitive to an increase in FS (higher EGR and lower $\lambda$). Particularly at higher engine speed and load and thus higher injected fuel mass. A certain trend appears to be visible at which combustion duration between 10 and 90% burnt increases exponentially, after which it decreases at increasing FS level. The maximum increase in combustion duration (10-90% burnt) is found to be 9°CA (1518rpm 4.8bar BMEP)

It appears that by increasing injected fuel mass combustion duration is prolonged, primarily between 10-50% burnt. Increasing injected fuel mass by approximately a factor 2 (1210 rpm, vs. 1518 rpm) the combustion duration 10-50% burnt increases by approximately 4.0 deg CA. Combustion duration 10-50% burnt seems to increase linearly with the increase in injected fuel mass at increasing FS levels and constant engine speed and load. Furthermore increasing injected fuel mass appears to make ignition delay and combustion duration between 10-90% burnt more sensitive to increase in FS (EGR/$\lambda$).

As a consequence of increased ignition delay and combustion duration at increasing FS levels (EGR/$\lambda$) the combustion process shifts towards the expansion stroke and combustion becomes incomplete at more extreme FS levels +50%. Other effects of increase of FS and EGR are the decrease in peak cylinder pressure and reduction in ROHR.

Findings from literature regarding combustion process and the influence of EGR:

In literature it can be found that combustion duration is prolonged (combustion rate reduced) and the combustion retarded as a consequence of EGR [2, 3, 71, 59]. The general effects of EGR are described in chapter 2.

Combustion rate depends on the probability that the fuel and oxygen molecules meet and react [4]. As such parameters that are of influence on combustion duration are the injected fuel mass, lambda, EGR level, injector and combustion chamber design (injection pressure, swirl etc..), temperature.

Unfortunately, the magnitude of the effect of EGR on combustion duration for CI engines is not extensively described in literature (High EGR, +30%). From SAE 2001-01-3497 it can be estimated that for a heavy-duty CI diesel engine at 1460rpm, full load (bmep = 18.7 bar) at 1600 bar fuel injection pressure, SOI 3° BTDC; combustion duration increases with 3° and 12° CA ($\theta$ 10-50% and $\theta$ 10-90% burnt period respectively) from 0 to 20% EGR.

| *Engine; Heavy Duty CI 2.147dm$^3$/cyl* | | | |
|---|---|---|---|
| *speed* | rpm | 1460 | 1460 |
| *EGR* | % | 0 | 20 |
| *SOI* | ° CA | -3 | -3 |
| *BMEP* | bar | 18.7 | 18.7 |
| θ 10-50% | ° CA | **8** | **11** |
| θ 10-90% | ° CA | **20** | **32** |

*Table T2.3.5 Indicated combustion duration derived from Rate of Heat Release; figure 14 [77]*

Heywood [2] mentions burn-durations of 20° to 100° CA for 0 to 100% burn in case of CI engines, in which 100° CA combustion duration (0 -100% burn) is referred to as "very long combustion process". Also a combustion interval of order 40 to 50° CA is mentioned for CI engines "to maintain high fuel conversion efficiency".

Remarks in relation to simulations for high EGR;
The intended simulations are to be performed for much higher BMEP's (up to 25 bar) and higher engine-speeds as used in the above assessed cylinder-pressure measurement. The engine to be modeled for the simulation purposes is a DAF MX with a 2stage turbo system. The engine from which the cylinder-pressure data is available is an atmospheric DAF MX engine. As such the influence of swirl, injection pressure and cylinder temperature are assumed to be negligible.

Because of the limited amount of measurement data available at high EGR engine-operating points, and because combustion duration is depend on at least 3 variables (injected fuel mass, lambda, EGR level) it is difficult to give reliable interpolation and extrapolation of combustion duration at increasing EGR levels, BMEP (up to 25 bar) and engine speed. Therefore it is not possible to give a reliable prediction of the influence of high EGR and FS levels on combustion duration and ignition delay for intended simulations, based on measurement data.

However it seems proven that ignition delay does increase with the use of high levels of EGR and FS in the order of 1.5° CA. Combustion duration does also prolong significantly in the order of 10° CA with the use of high levels of EGR (up to 70%). The combustion process shifts further towards the expansion stroke. Naturally the mass of fuel injected is of major influence on ignition delay and combustion duration. Based on literature typical combustion duration for CI engines are in the order of 40-50° CA, depending on the injected fuel mass.

*References:*

[2, 3, 59, 71, 76, 77, 78]

# Appendix 3 Model set-up and performance

## Appendix 3.1 Composition GT-power models

Supplement to *§4.4 composition GT-power models*

| Component | Description | Range |
|---|---|---|
| General Engine Performance | ▪ Power rated in kW at n=1405 - 2100rpm<br>▪ Power rated in hp at n=1400 - 2100 rpm<br>▪ Torque max in lb.ft<br>▪ Torque max in N.m. | 265 - 380<br>350 - 510<br>1350 - 1850<br>1830 - 2510 |
| General Engine | ▪ 4 Cycle-diesel; 6 cyl in-Line; turbocharged and intercooled<br>▪ Bore<br>▪ Stroke<br>▪ Displacement<br>▪ N idle<br>▪ Moment of Inertia<br>▪ Max. No load Governed speed<br>▪ Maximum Overspeed Capability<br>▪ Max cycle pressure | <br>: 130 (mm)<br>: 162 (mm)<br>: 12.9 ($dm^3$)<br>: 600 +/- 100 rpm<br>: (t.b.e) $kg\text{-}ms^2$<br>: 2200 +/- 25rpm<br>: (t.b.e) rpm<br>: 20 MPa |

*Table T3.1.1 General engine specifications DAF MX*

| Engine geometry | |
|---|---|
| Bore [mm] | 130.0 |
| Stroke [mm] | 162.0 |
| Connecting Rod Length [mm] | 262.0 |
| Piston Pin Offset [mm] | 0.00 |
| Displacement/Cylinder [liter] | 2.150 |
| Total Displacement [liter] | 12.902 |
| Number of Cylinders | 6 |
| Compression Ratio | 17.00 |
| Bore/Stroke | 0.802 |
| IVC [CA] | -157 |
| EVO [CA] | 130 |
| IVO [CA] | 343 |
| EVC [CA] | 371 |

*Table T3.1.2 Engine geometry GT-power models. 0° CA = TDC compression stroke*

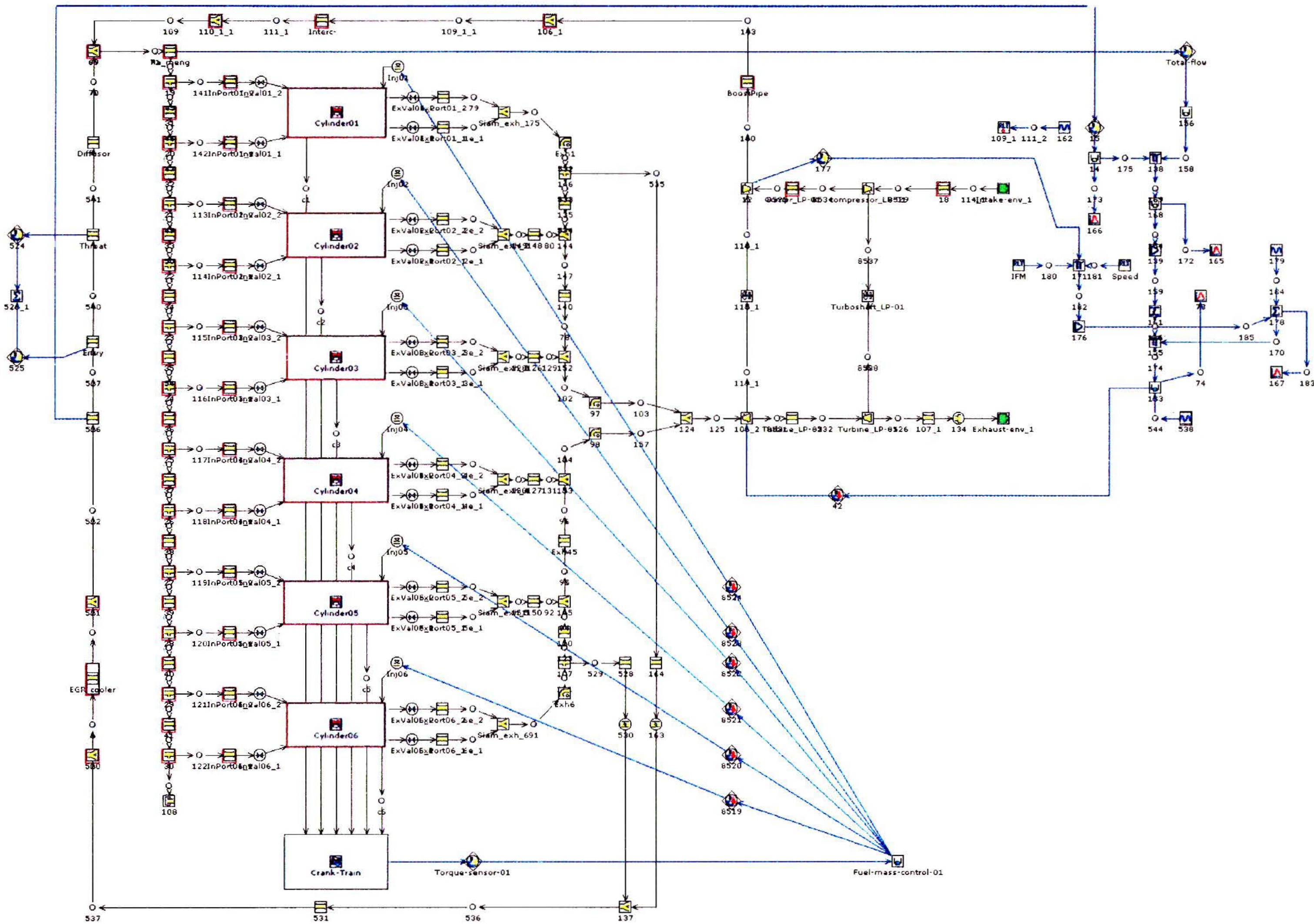



*Figure E3.1.1 Impression of the base-line GT-power engine-model*

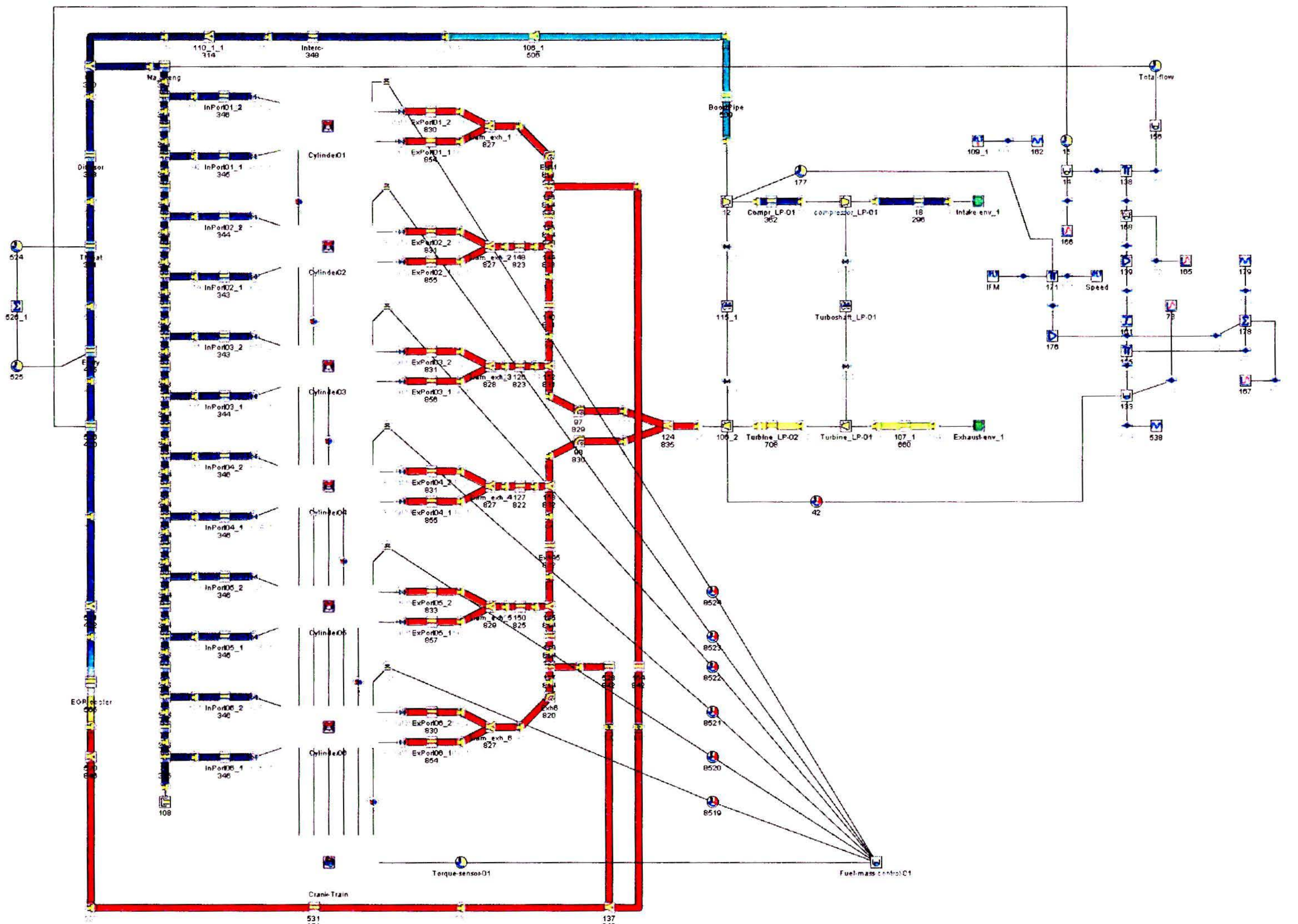


Figure E3.1.2 Impression of the base-line GT-power engine-model, red=exhaust & EGR piping, blue=air & cooled EGR piping.

## General specifications per pipe section of, GT-power base-line engine-model

In table T3.1.3 the initial wall temperature, heat conduction, initial state and surface roughness is given for the parts within a certain section of pipe work in the base-line engine model. The initial wall temperature is recalculated during simulation if the 'Heat Conduction Object' (HTO) is selected, if not selected (Ign.) the initial wall temperature remains constant during the entire model calculation. The 'initial state ' gives the initial conditions of the medium within the system. Surface roughness is given for some part and left 'default' at others.

| *Section* | *Initial wall temperature [K]* | *Heat conduction* | *Initial state* | *Surface roughness [mm]* |
|---|---|---|---|---|
| Inlet manifold | 353.15 | Ign. | Intake-initial | Def. |
| Inlet ports | 353.15 | Ign. | Intake-initial | 0.25 |
| Mixer | 313.15 | Ign. | Intake-initial | Def. |
| EGR cool section | 398.15 | Ign. | Exhaust-initial | Def. |
| EGR cooler | 398.15 | Ign. | Intake-initial | Def. |
| EGR hot section | 800 | HTO | Intake-initial | Def. |
| Exhaust ports | 575 | Ign. | Exhaust-initial | 0.25 |
| Exhaust manifold | 800 | HTO | Exhaust-initial | Def. |
| Exhaust muffler | 650 | HTO | Exhaust-initial | Def. |
| Inlet air filter | 293.5 | Ign. | Intake-initial | 0.5 |
| In between compressors | 353.15 | Ign. | Intake-initial | Def. |
| Boost pipe | 423.15 | Ign. | Intake-initial | 0.5 |
| Intercooler inlet | 450 | Ign. | Intake-initial | Def. |
| Intercooler | 313.15 | Ign. | Intake-initial | Def. |
| Intercooler outlet | 322 | Ign. | Intake-initial | Def. |

*Table T3.1.3 general specifications per pipe section, base-line engine model*

More specifications and information about the base-line engine-model regarding; HTO, ambient temperature (outside the engine compartment) and the engine's cylinder general set-up.

- Heat Conduction Object (HTO); free convection,
  External temperature (convective temperature of the ambient surrounding) = 313.15 K
  Radiation sink temperature = 300 K

- Ambient temperature 293.15 K
  Pressure 1 bar

- Cylinders; Wall temperature; Head 510 K
  Piston 510 K
  Cylinder 430 K
  Heat transfer mode; Woschni
  Combustion object Wiebe

## Wiebe combustion-model setting

In table T3.1.4 the Wiebe combustion model settings as used in all of the conducted simulations are given. If 'def' is selected, meaning default, this parameter will be chosen by GT-power as is described in §4.5. More information regarding the Wiebe combustion model can be found §4.5 and appendix 2.3. Please notice that in tableT3.1.4 the first 9 values are the GT-power Wiebe-parameters. Below it is given how the 14 Wiebe-parameters of the three terms Wiebe-function are determined.

| | | |
|---|---|---|
| *main* | | |
| Ignition Delay | | def |
| Premixed Fraction | | def |
| Tail Fraction | | def |
| Premixed Duration | | 10 |
| Main Duration | | 25 |
| Tail Duration | | 26 |
| Premixed Exponent | | 0.5 |
| Main Exponent | | 1.2 |
| Tail Exponent | | 2.36 |
| | | |
| *options* | | |
| Number of Temperature Zones | | two-temp |
| Fraction of Fuel Burned | | 1 |
| Ignition Delay Multiplier | | def |
| Premix Duration Multiplier | | def |
| Main Duration Multiplier | | def |
| Tail Duration Multiplier | | def |
| NOx Reference Object | | EngCylNOx |

Table T3.1.4 Wiebe combustion-model settings as used

Referring to the 14 parameters in the equations 4.7 and 4.8 (§4.5);

$\vartheta_{01}$ = SOI (start of injection)– Ignition delay

$\vartheta_{02}$ = $\vartheta_{01}$

$\vartheta_{03}$ = $\vartheta_{01}$

$\alpha$ = Premixed fraction

$\beta$ = Tail fraction

$\Delta\vartheta_1$ = Premixed duration

$\Delta\vartheta_2$ = Main exponent

$\Delta\vartheta_3$ = Tail exponent

$\omega_1$ = Premixed exponent

$\omega_2$ = Main exponent

$\omega_3$ = Tail exponent

$a_1$ = $f(\Delta\vartheta_1;\omega_1)$

$a_2$ = $f(\Delta\vartheta_2;\omega_2)$

$a_3$ = $f(\Delta\vartheta_3;\omega_3)$

**Step by step explanation of this FS control-system, base-line engine-model**

1. A sensor measures EGR mass flow in kg/s between the EGR-cooler and venturi-flowmeter
2. A sensor measures the total mass flow in kg/s at the entrance of the inlet manifold
3. The EGR level is calculated by dividing the filtered EGR mass flow sensor signal by the filtered total mass flow sensor signal. $\frac{EGR\ mass\ flow}{total\ mass\ flow} = EGR$
4. A sensor measures fresh air mass flow in kg/s at the HP compressor.
5. A sensor measures the Fuel injection rate in mg/cycle at the injector of cylinder nr.1
6. A sensor measures the average engine speed in rpm. at the engine crank-train.
7. A measure for the AF ratio is calculated making use of the fresh air mass flow, the fuel injection rate, and the average engine speed a gain factor is used to make the dimensions correct and obtains the number of cylinders and number of injection per revolution; $\frac{Air\ mass\ flow}{fuel\ rate \times engine\ speed} \cdot gain = AF$
8. Dividing the EGR level by the AF level results in a measure for the FS, a gain factor is used in order to calculated the FS factor more accurate; $\frac{EGR}{AF} \cdot gain \approx FS$
9. A PI-controller is used in order to determine the required VTG position. If the actual FS level is lower than the reference FS level the VTG position is further closed (towards 0), if the actual FS in higher then the reference FS level the VTG position is further opened (towards 1), all within VTG boundaries. P value of (-) 0.02 and a I value of (-) 0.1.

## Appendix 3.2 6-mode test cycle

In order to reduce simulation duration a limited number of engine operating points are used during simulations. The selected engine operating points for the simulation of the selected EGR configurations consists of 6 operating points (6-mode) based on the ESC 13-mode test cycle and the Rosi engine operating-point.

The "ESC (European Steady-State Cycle) 13-mode test cycle" is determined by 13 steady and 3 random modes (engine operating points). Emission values are obtained with the weighted mean of emissions on each of the 13 modes. The 13 test-modes are spread over four engines speeds, categorized as A, B, C speed and idle speed. At each engine speed, except for idle, four load points are determined, 25%, 50%, 75% and 100% load [34]. For the DAF MX engine ESC 13 mode engine speeds A, B, C are at 1212, 1530, 1830 rpm respectively.
The "Rosi" working point corresponds to working conditions typical for HD truck application. These conditions come forward from maintaining a cruising speed of approximately 85 km/h on flat road with a HD vehicle. As this is the most common driving condition for HD vehicles, the Rosi working point is often used in the Netherlands for determining a typical fuel consumption level for this engine application. For the DAF MX engine the Rosi working point is at 1350rpm with a power output of 100 kW.

In selecting 6 operating points for simulation purposes it was aimed at spreading the operating points evenly over the ESC 13-mode test cycle plus Rosi operating points. Also it was decided to include all 100% load points of the ESC 13-mode test cycle as these are found to be most difficult in applying High EGR (preparatory EGR and turbo study for High EGR, appendix 1.2). This because of the already high flow and low AF levels at these operating points, making it difficult to apply high levels of EGR maintaining the desired output power with acceptable BSFC and emission penalty. As a result the selected 6 operating points used in simulation correspond with all 100% load points from the ESC 13mode test cycle; A100, B100 and C100 and in addition A50, C50 and Rosi (1350rpm, 100kW).

| *6-mode* | *Speed [rpm]* | *Load [Nm]* |
|---|---|---|
| **A-100** | 1212 | 2557 |
| **B-100** | 1530 | 2442 |
| **C-100** | 1830 | 2015 |
| **A-50** | 1212 | 1279 |
| **C-50** | 1830 | 1008 |
| **Rosi** | 1350 | 707.4 |

*Table T3.2.1; 6-mode test cycle applied to DAF MX max. 390kW*

Although numerous points are missing compared to the ESC 13-mode test cycle it is still desirable to calculate a weighted mean (ESC) value for several parameters, based on the 6 operating points instead. In order to do this the weight-factors for each of the 6 operating points need to be defined. In defining the weight-factors for the 6 operating points use is made of the existing 13-mode ESC weight factors, which are rearranged over the 6 points. As the selected 6 operating points already correspond to ESC 13-mode points, the weight-factors belonging to these points in the 13 mode ESC are maintained. The remaining weight-factors of the other operating points in the 13-mode ESC are divided over the 6 operating points and added as illustrated below in figure E3.2.1. The weight-factor of idle is neglected. In figure E3.2.2 the weight-factors per operating point are given for the constructed 6-mode test cycle

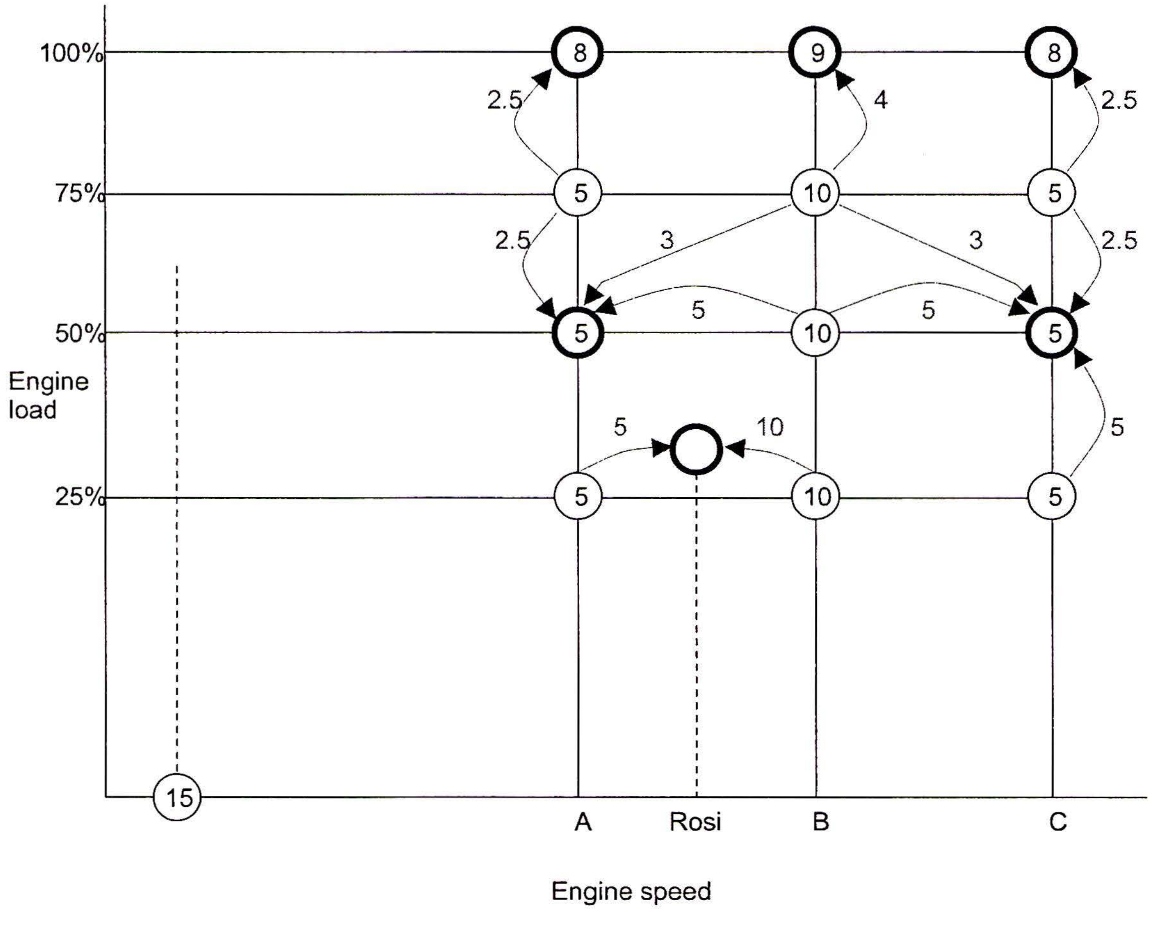



*Figure E3.2.1; distribution of the weight factors from 13-mode to 6-mode test cycle*

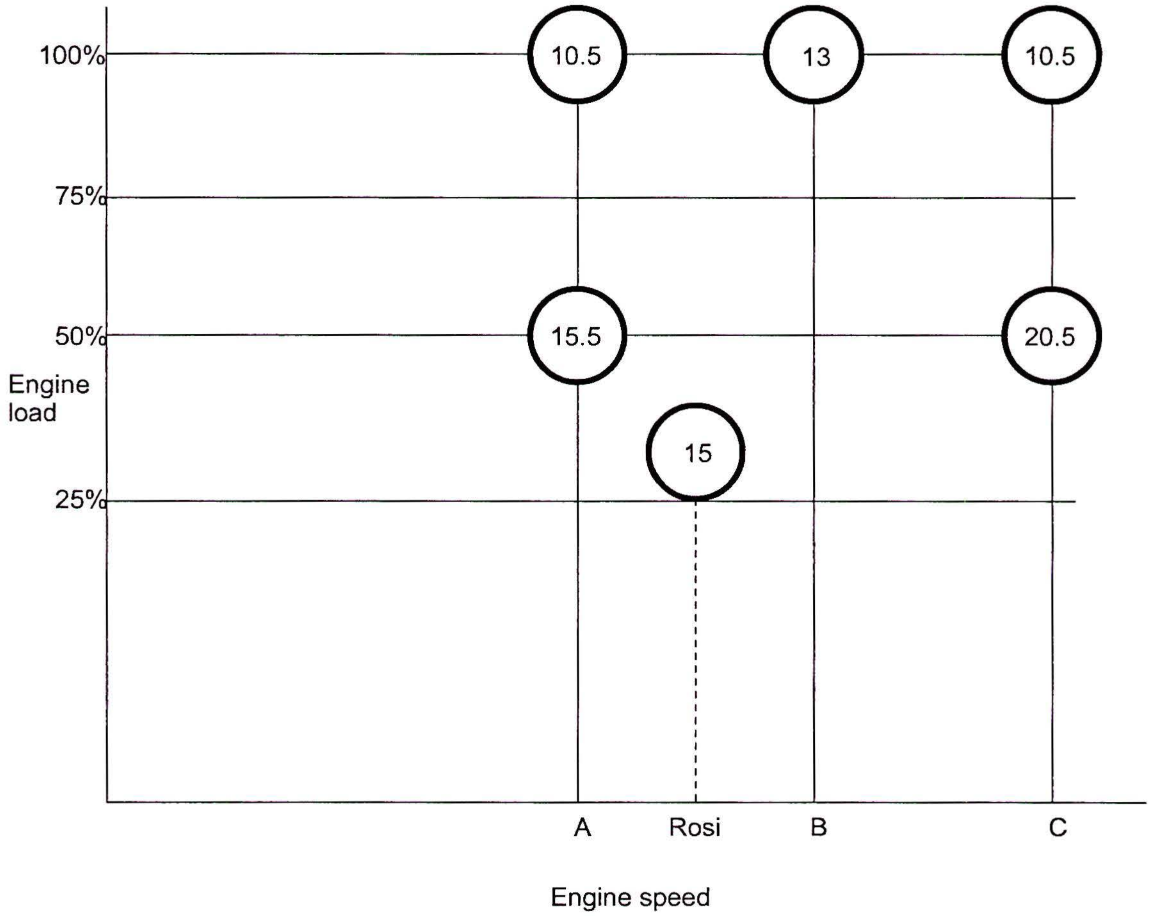



*Figure E3.2.2; weight factors for a 6-mode test cycle*

Average values for a given variable, over the 6mode test cycle are calculated as;

$$Average = \frac{\sum_{j=6} Variable_j \times PO_j \times WF_j}{\sum_{j=6} PO_j \times WF_j}$$

A 3.2.1

In which;

| | | |
|---|---|---|
| *j* | = case number, 6 in total |
| *PO* | = engine power [kW] |
| *WF* | = weight factor |

## Appendix 3.3 Model validation

GT-power-File; EGR_HS_2stage7_val_4
Engine; DAF MX 12.9 L, 2 stage HOLSET HE431 VTG & GT50,
EGR; External routed internally driven High pressure
Simulation Case setup;

| | Min AF | Min FS | RPM | Load [Nm] | Tegr_wall [K] | Tnik_wall [K] | SOI ° |
|---|---|---|---|---|---|---|---|
| 1) | 14.5 | 15.42 | 1214 | 2565 | 398.15 | 313.15 | 0.5 |
| 2) | 14.5 | 23.59 | 1524 | 1156.4 | 398.15 | 313.15 | -6 |
| 3) | 14.5 | 15.38 | 1822 | 2012.6 | 398.15 | 313.15 | -5 |
| 4) | 14.5 | 18.19 | 1906 | 1941.4 | 398.15 | 313.15 | -8 |

*Table T3.3.1. simulation case-setup*

Note; no use is made of the EGR-valves (open)
FS is controlled by a PI-controller and adjustment of the VTG-rack position
Engine torque is controlled by a PI-controller and adjustment of the fuel injection rate

Based on performed engine tests conducted by TNO, a simulation is performed with the two-stage engine model. In the simulation case-setup the engine speed, engine load, reference FS value and Start of injection are based on the measurement-points from the measurement-data. Four steady-state engine operating-points are simulated and compared with the measurement-data. The simulation case-setup is given in table T3.3.1.
The simulation-data and comparison with measurement-data can be found under 'DATA validation'. From the simulation data it can be concluded that in all cases the system is converged to a equilibrium and that in all cases, the turbo operating-points are within the current turbo-maps. Comparison between simulation data and measurement data is given below for load, FS, EGR, AF, NOx, BSFC, Manifold temperature and pressure and EGR temperature.

Results

| **Load** | | | |
|---|---|---|---|
| *case* | Measurement [Nm] | Simulation [Nm] | Difference [%] |
| 1 | 2565.00 | 2570.84 | 0.23 |
| 2 | 1156.40 | 1157.51 | 0.10 |
| 3 | 2012.60 | 1994.57 | -0.90 |
| 4 | 1941.40 | 1972.08 | 1.58 |

*Table T3.3.2. comparison engine Load-conditions for measurement and simulation*

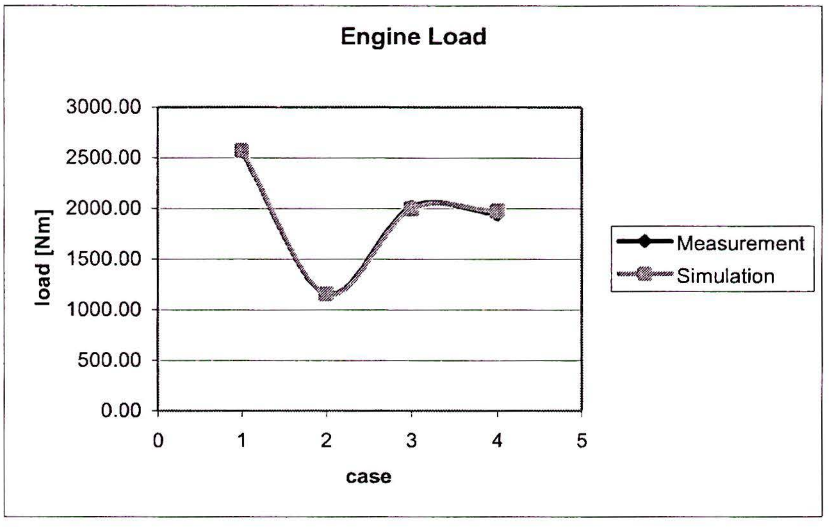



*Figure E3.3.1. comparison engine Load-conditions for measurement and simulation*

It can be seen from table T3.3.2 and figure E3.3.1 that there is little difference in the load conditions between measurement and simulation. The difference is caused by the controller implemented into the model (simulation) which lacks accuracy to maintain the pre-set load values as given in the case-setup.

| **FS** | | | |
|---|---|---|---|
| *case* | Measurement [%] | Simulation [%] | Difference [%] |
| 1 | 15.42 | 15.25 | -1.09 |
| 2 | 23.59 | 23.41 | -0.76 |
| 3 | 15.38 | 15.24 | -0.90 |
| 4 | 18.19 | 17.94 | -1.37 |

*Table T3.3.3. comparison FS for measurement and simulation*

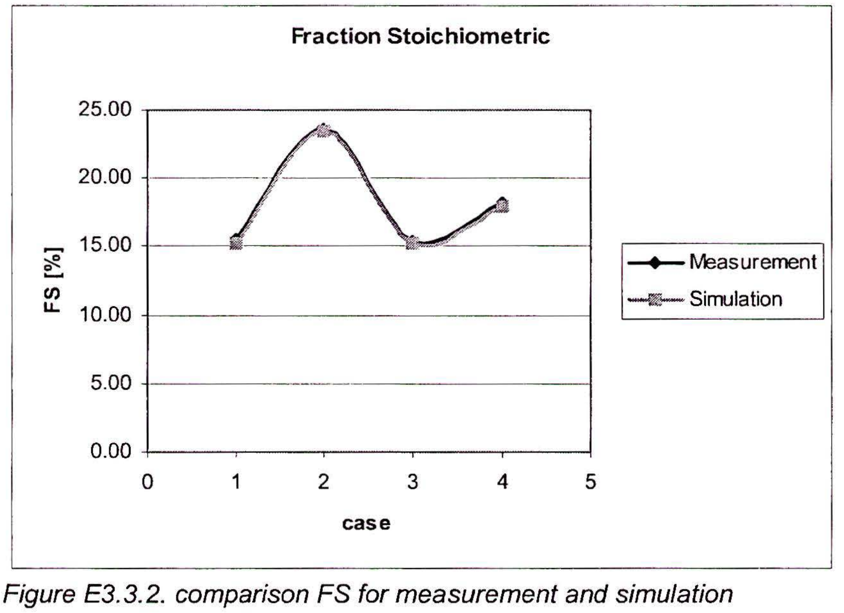



*Figure E3.3.2. comparison FS for measurement and simulation*

As can be seen from table T3.3.3 and figure E3.3.2 the difference in FS value between measurement and simulation is small. The difference is primarily caused by the lack of accuracy of the 'FS-controller' in the model.

| EGR | | | |
|---|---|---|---|
| *case* | Measurement [%] | Simulation [%] | Difference [%] |
| 1 | 22.40 | 23.50 | 4.68 |
| 2 | 35.70 | 35.30 | -1.13 |
| 3 | 23.20 | 22.90 | -1.31 |
| 4 | 27.30 | 26.60 | -2.63 |

*Table T3.3.4. comparison EGR for measurement and simulation*

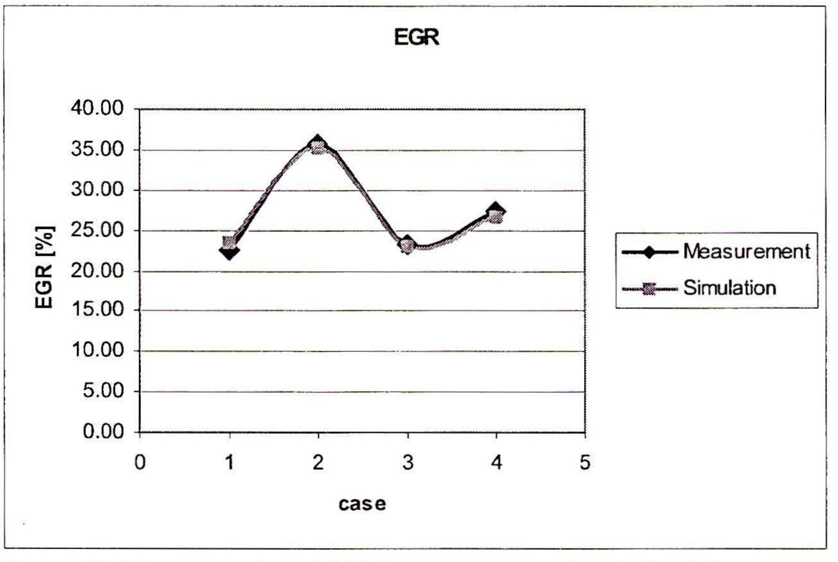



*Figure E3.3.3. comparison EGR for measurement and simulation*

From table T3.3.4 and figure E3.3.3 it can be seen that the achieved EGR level from the simulation is in relative good correlation with the EGR level from the measurement-data. In case 1 the difference in EGR level from simulation and measurement is the biggest at 4.7%, this may be caused to some extend, by the difference in FS value ( 1.1 % see table T3.3.3.).

| AF | | | |
|---|---|---|---|
| *case* | Measurement [-] | Simulation [-] | Difference [%] |
| 1 | 21.62 | 23.00 | 6.38 |
| 2 | 22.56 | 22.48 | -0.37 |
| 3 | 22.49 | 22.39 | -0.45 |
| 4 | 22.37 | 22.08 | -1.29 |

*Table T3.3.5. comparison EGR for measurement and simulation*

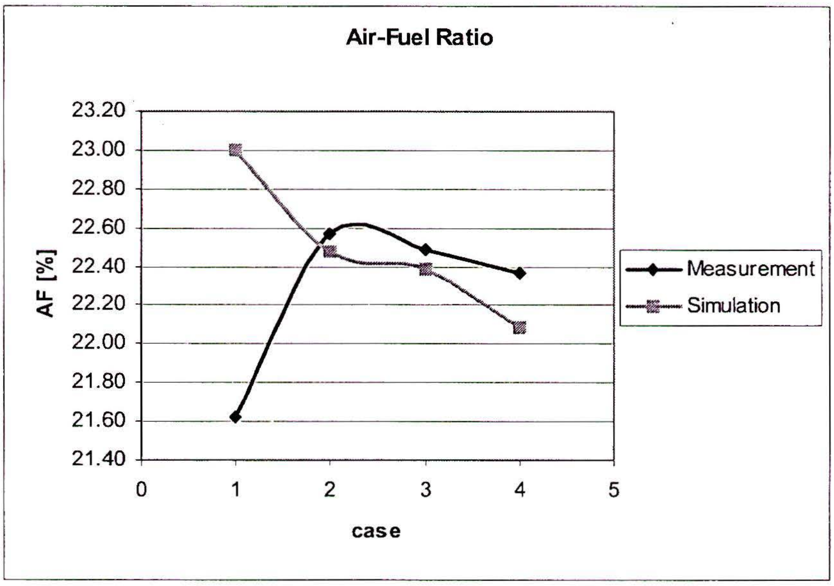



*Figure E3.3.4. comparison EGR for measurement and simulation*

From table T3.3.5 and figure E3.3.4 it can be seen that the achieved AF ratio in simulation and from measurement are in good correlation with each other. The apparent big difference in figure E3.3.4 is partly due to the scaling on the y-axis. In case 1 the difference in AF ratio from simulation and measurement is the biggest, in the same way as for EGR (see table T3.3.4.) the difference in AF ratio in case 1, may be explained to a small extend by the difference in FS value. From both table T3.3.4 and T3.3.5 it can be seen that the FS value from table T3.3.3 are achieved by almost the same combinations of EGR and AF.

| **NOx** | | | |
|---|---|---|---|
| *case* | Measurement [g/kWh] | Simulation [g/kWh] | Difference [%] |
| 1 | 1.83 | 1.94 | 6.01 |
| 2 | 0.93 | 0.32 | -66.02 |
| 3 | 1.72 | 2.79 | 62.21 |
| 4 | 1.98 | 2.22 | 12.12 |

*Table T3.3.6. comparison NOx emission for measurement and simulation*

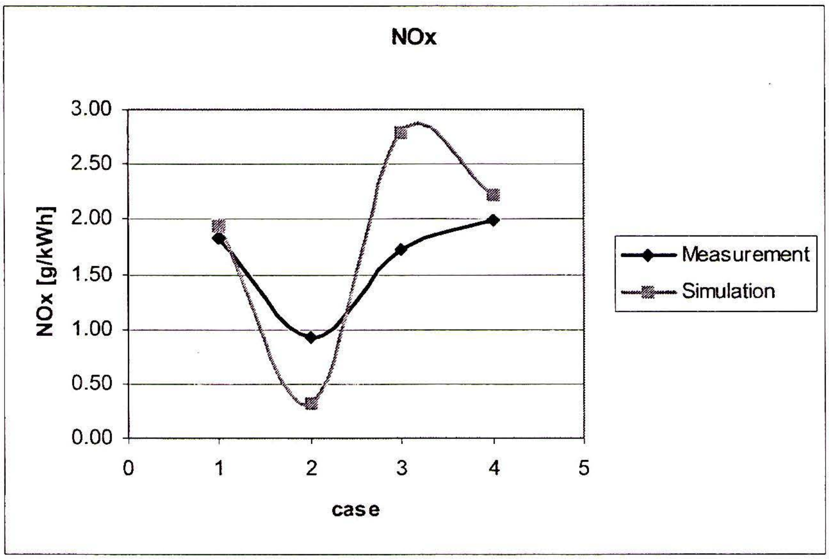



*Figure E3.3.5. comparison EGR for measurement and simulation*

From figure E3.3.5 it can be seen that some correlation is apparent for the NOx prediction from the simulation and the measured NOx emission. However in table T3.3.6. it can be seen that large differences between model predicted NOx emission and measured NOx are present. The reason for this large difference may be the inaccuracy of the NOx prediction model within GT-power. For all cases the so-called Wiebe combustion model is used, in which the NOx prediction module is activated. It is not clear whether the more sophisticated GI-jet combustion model provides a more accurate NOx prediction.

| BSFC | | | |
|---|---|---|---|
| *case* | Measurement [g/kWh] | Simulation [g/kWh] | Difference [%] |
| 1 | 201.00 | 200.57 | -0.21 |
| 2 | 208.40 | 203.09 | -2.55 |
| 3 | 212.06 | 200.20 | -5.59 |
| 4 | 209.35 | 198.57 | -5.15 |

*Table T3.3.7. comparison BSFC (Brake Specific Fuel Consumption) for measurement and simulation*

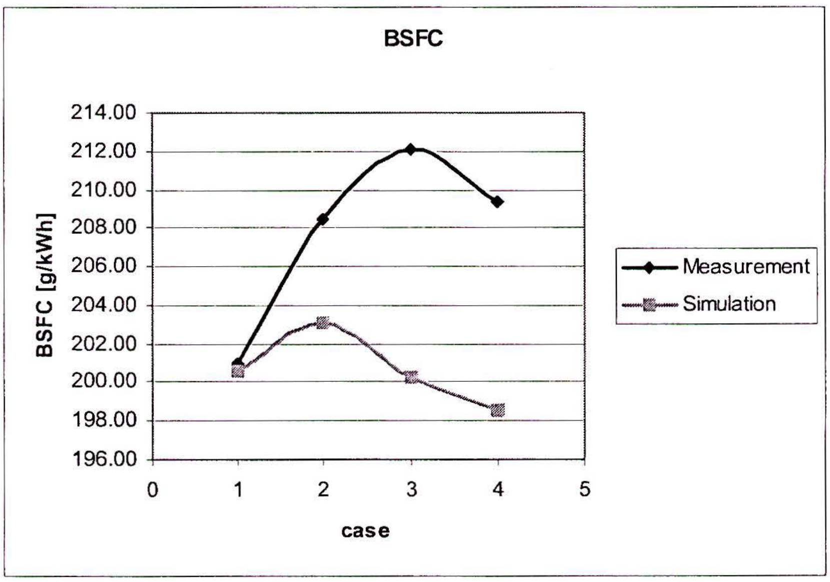



*Figure E3.3.6. comparison BSFC (Brake Specific Fuel Consumption) for measurement and simulation*

From table T3.3.7 it can be seen that the model predicted BSFC from simulation and BSFC from measurement are in good correlation with each other. In all cases the model predicts a lower BFSC and in case 3 and 4 the differences in BSFC are the biggest (5.59 & 5.15 %). Apparently the model assumes better efficiency in all cases.

| Manifold pressure | | | |
|---|---|---|---|
| *case* | Measurement [Bar] | Simulation [Bar] | Difference [%] |
| 1 | 4.13 | 4.33 | 4.78 |
| 2 | 2.63 | 2.30 | -12.62 |
| 3 | 3.86 | 3.18 | -17.65 |
| 4 | 4.09 | 3.26 | -20.40 |

*Table T3.3.8. comparison Manifold pressure for measurement and simulation*

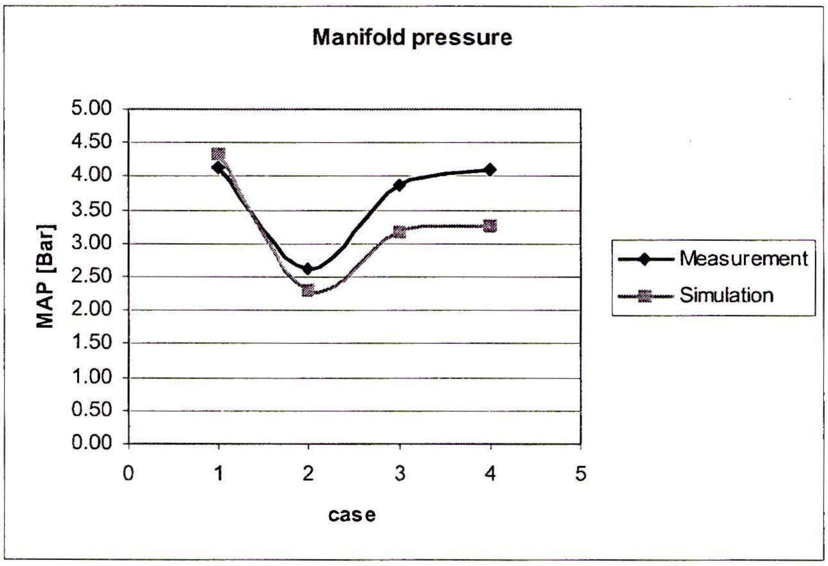



*Figure E3.3.7. comparison Manifold pressure for measurement and simulation*

| **Manifold temperature** | | | |
|---|---|---|---|
| *case* | Measurement [°C] | Simulation [°C] | Difference [%] |
| 1 | 60.35 | 67.41 | 11.71 |
| 2 | 88.00 | 78.89 | -10.36 |
| 3 | 73.56 | 67.61 | -8.09 |
| 4 | 95.35 | 72.10 | -24.38 |

*Table T3.3.9. comparison Manifold temperature for measurement and simulation*

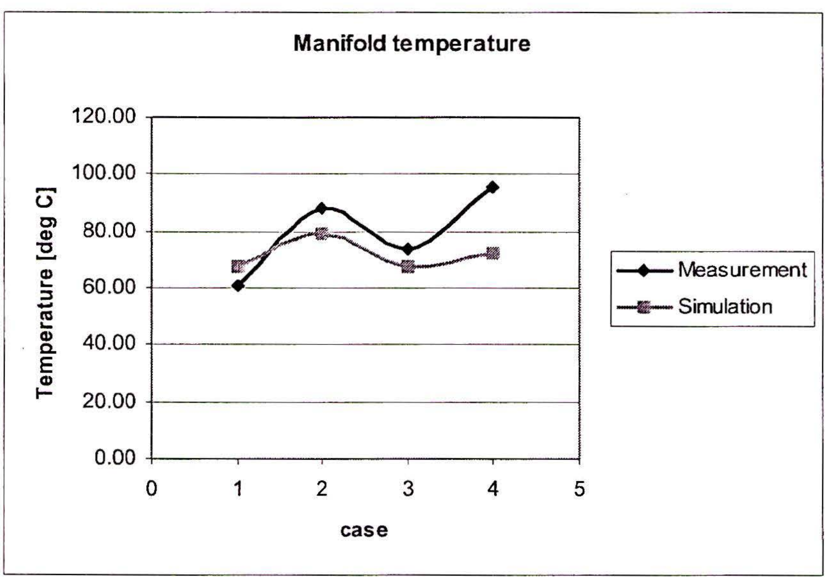



*Figure E3.3.8. comparison Manifold temperature for measurement and simulation*

From table T3.3.8 and figure E3.3.7 it can be seen that there are big differences between simulated and measured manifold pressure for the four cases. In analogy with this it can be seen from table T3.3.9 and figure E3.3.8 that big differences are present between simulated and measured manifold temperature. A cause for these big differences is likely to be the fixed wall temperature of the EGR-cooler and Intercooler in the model (simulations) for all cases, which can be found in the case-setup (tableT3.3.1.) under 'Tegr_wall' and 'Tnik_wall'. It may be concluded that in the model the wall-temperatures are set too high for case 1 resulting in a too high manifold temperature and subsequently a too high required manifold pressure. For the case 2, 3 and 4 the wall temperatures of the EGR-cooler and intercooler are set too low, resulting in too low manifold temperature and subsequently a too low required manifold pressure. Although a large part of the manifold pressure error may be as a consequence of the error in manifold temperature, some part of the manifold pressure error may have other causes.

The difference in manifold pressure and temperature from the simulations in comparison to those from measurements, may be of influence on the efficiency assumed by the model en subsequently BSFC. Particularly at case 3 and 4 where the differences in simulated and measured manifold temperature and pressure are the biggest, the difference between simulated and measured BSFC is also the biggest, suggesting a correlation.

| **EGR temperature pre-mixer** | | | |
|---|---|---|---|
| *case* | Measurement [°C] | Simulation [°C] | Difference [%] |
| 1 | 143.05 | 118.53 | -17.14 |
| 2 | 184.90 | 147.33 | -20.32 |
| 3 | 190.64 | 148.49 | -22.11 |
| 4 | 244.25 | 155.32 | -36.41 |

*Table T3.3.10. comparison EGR pre-mixer temperature for measurement and simulation*

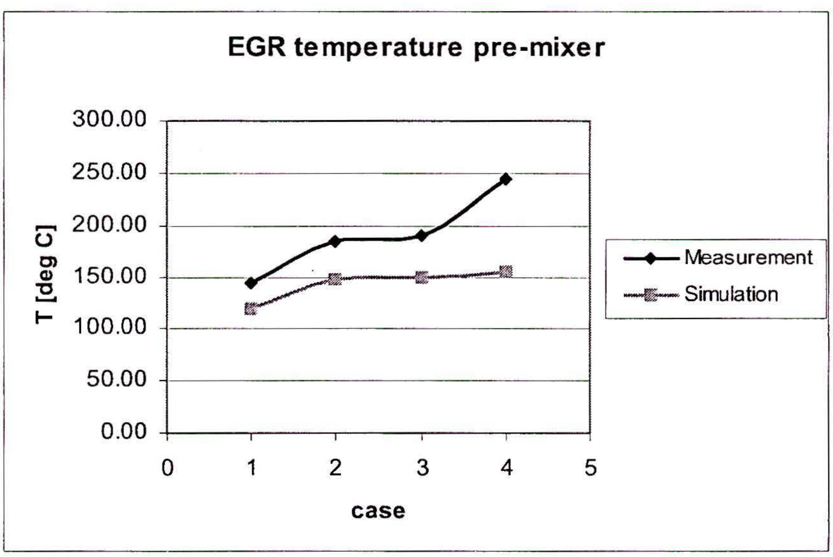



*Figure E3.3.9. comparison EGR pre-mixer temperature for measurement and simulation*

As can be seen from table T3.3.10 and figure E3.3.9 the EGR-temperature during simulation is lower than from measurement, indicating that the EGR-cooler wall temperature (Tegr) is set at too low value.

From additional simulations with altered EGR-cooler wall temperature the manifold temperature as simulated can be brought in good agreement with the measured manifold temperature, table T3.3.11. As a consequence of this the simulated manifold pressure is also in slightly better coherence with the measured manifold better. However the simulated manifold pressure is still less than the measured manifold pressure, table T3.3.12, and the simulated EGR temperature is still less than the measured EGR temperature, table T3.3.13.

| **Manifold temperature** | | | |
|---|---|---|---|
| *case* | Measurement [°C] | Simulation [°C] | Difference [%] |
| 1 | 60.35 | 61.262 | 1.51 |
| 2 | 88 | 85.955 | -2.32 |
| 3 | 73.56 | 73.217 | -0.47 |
| 4 | 95.35 | 94.751 | -0.63 |

*Table T3.3.11. comparison Manifold temperature for measurement and additional simulation*

| **Manifold pressure** | | | |
|---|---|---|---|
| *case* | Measurement [Bar] | Simulation [Bar] | Difference [%] |
| 1 | 4.13 | 4.29 | 3.79 |
| 2 | 2.63 | 2.32 | -11.62 |
| 3 | 3.86 | 3.23 | -16.28 |
| 4 | 4.09 | 3.44 | -15.86 |

*Table T3.3.12. comparison Manifold pressure for measurement and additional simulation*

| **EGR temperature pre-mixer** | | | |
|---|---|---|---|
| *case* | Measurement [°C] | Simulation [°C] | Difference [%] |
| 1 | 143.05 | 99.1 | -30.72 |
| 2 | 184.9 | 165.811 | -10.32 |
| 3 | 190.64 | 169.439 | -11.12 |
| 4 | 244.25 | 234.649 | -3.93 |

*Table T3.3.13. comparison EGR pre-mixer temperature for measurement and additional simulation*

| **BSFC** | | | |
|---|---|---|---|
| *case* | Measurement [g/kWh] | Simulation [g/kWh] | Difference [%] |
| 1 | 201.00 | 200.10 | -0.45 |
| 2 | 208.40 | 206.64 | -2.28 |
| 3 | 212.06 | 200.39 | -5.50 |
| 4 | 209.35 | 199.80 | -4.56 |

*Table T2.3.14 comparison BSFC (Brake Specific Fuel Consumption) for measurement and additional simulation*

As can be seen table 2.3.14 the overall difference in BSFC between measurement and simulation has decreases as a consequence of the more coherent manifold temperature. Indicating that the previous difference in BSFC (table T2.3.7) is to some extent related to inaccurate settings of EGR-cooler and intercooler wall temperature. However a significant difference in BSFC remains for the cases 3 and 4.

Unfortunately the available measurement data is limited. No in-cylinder pressure trace is available and the real VTG-rack position is also not available. Furthermore the measurement inaccuracy is unknown. Therefore it is difficult to determine exactly what causes the differences, particularly in BSFC, between simulation and measurement. Possible causes may be turbo-map inaccuracy, combustion model inaccuracy, differences in pumping loss possibly caused by differences in temperatures of pipe work and coolers. Also measurement inaccuracy could be the cause.

Although it is uncertain what exactly causes the model inaccuracy, it is known that the use of the Wiebe-combustion model can cause inaccuracies with the use of EGR (§4.5). As substantial levels of EGR are used during validation this is a likely cause.
During the validation-simulations a conventional combustion model is used (the same as described in appendix 3.1) which does not take into account the effects of EGR on the combustion process. The use of EGR may have caused a delay in the combustion process during the real-life engines tests. This may have caused combustion deterioration and thus a higher BSFC. In addition to this a delay in combustion may have caused a higher exhaust gas temperature. A higher exhaust temperature may lead to higher EGR-gas temperatures even after the EGR-cooler and higher turbine speeds, causing higher intake manifold pressures. All of which are not taken into account by the conventional Wiebe combustion model.
In §4.5 it is already indicated that the use of a conventional combustion model with high levels of EGR may lead to:

- Too low predicted exhaust-gas temperature
- Too low predicted exhaust-gas pressure
- Too low predicted BSFC
- Too low predicted turbo-power and boost-pressure

The effects described above are also witnessed here. This can be proven by comparing the differences between simulation results and the measurement results for case 3, which showed the largest difference in BSFC, suggesting a large effect of the Wiebe combustion model inaccuracy. See table T2.3.14 and T2.3.7.

| **Case 3** | Measurement | Simulation | Difference [%] |
|---|---|---|---|
| HP turbine entrance temperature [°C] | 598 | 548 | -8.36 |
| HP turbine entrance pressure | 4.4 | 3.27 | -25 |
| HP turbine speed [rpm] | 103125 | 87078 | -15.6 |
| Intake manifold pressure [bar] | 3.86 | 3.23 | -16.28 |
| BSFC [g/kWh] | 212.06 | 200.39 | -5.5 |

*Table T2.3.14. Comparison simulation and measurement results for case 3.*

In conclusion it is believed that the inaccuracy of the Wiebe combustion model with the use of EGR is the main cause for the found model inaccuracy.

Please note that engine speed and load and start of injection are most likely of influence on the degree to which EGR affects diesel combustion. As such no unambiguous relationship exists between the level of EGR and the simulation model inaccuracy with the use of a conventional Wiebe combustion model, which does not take into account the effects of EGR on diesel combustion.

Conclusion

Some differences are apparent between data from simulation and data from measurement for the same case (steady state operation point). In some cases the difference can be explained as caused by controller inaccuracy and different pre-set reference values. Particularly the wall temperatures of the EGR-cooler and intercooler seem badly chosen, but can be improved. The NOx prediction from the model seems unreliable. The largest differences are found in the manifold pressure and manifold temperature and EGR temperature. To some extent the differences in manifold pressure are as a consequence of the difference in manifold temperatures, which may be caused by inaccurate settings of EGR-cooler and intercooler wall temperature (can be improved).

The simulation model underestimates BSFC. To some extend this is caused by inaccurate setting of EGR-cooler and intercooler wall temperature. The exact cause of the underestimation in BSFC is difficult to determine due to a lack of measurement data and accuracy. However it is believed that the main cause for model inaccuracy is due to the inaccuracy of the Wiebe-combustion model with the use of EGR.

Although attempted in §4.5 it was not possible to improve the combustion model for high-EGR purposes due to insufficient measurement data. Therefore a certain level of inaccuracy in BSFC is inevitable with simulations at high levels of EGR due to inaccuracies in the combustion model.

For the following simulations the focus is on the EGR flow, the AF ratio and the FS level. The simulation model can predict these three values with good accuracy. Therefore, despite the differences between simulation- and measured-data it is decided to continue with the current model.

The predicted BSFC in the following simulations must be regarded as indicative, due to model inaccuracy.

## Appendix 3.4 VTG position test

GT-power-File; EGR_HS_2stage14
Engine; DAF MX 12.9 L, 2 stage HOLSET HE431 VTG & GT50,
EGR; External routed internally driven High pressure
Simulation Case setup;

| | Min AF | VTG rack position | RPM | Load [Nm] | Tegr_wall [K] | Tnik_wall [K[ | SOI ° |
|---|---|---|---|---|---|---|---|
| 1) A100 | 14.5 | 0.1 | 1212 | 2557 | 398.15 | 313.15 | -5 |
| 2) A100 | 14.5 | 0.2 | 1212 | 2557 | 398.15 | 313.15 | -5 |
| 3) A100 | 14.5 | 0.3 | 1212 | 2557 | 398.15 | 313.15 | -5 |
| 4) A100 | 14.5 | 0.4 | 1212 | 2557 | 398.15 | 313.15 | -5 |
| 5) A100 | 14.5 | 0.5 | 1212 | 2557 | 398.15 | 313.15 | -5 |
| 6) A100 | 14.5 | 0.6 | 1212 | 2557 | 398.15 | 313.15 | -5 |
| 7) A100 | 14.5 | 0.7 | 1212 | 2557 | 398.15 | 313.15 | -5 |
| 8) A100 | 14.5 | 0.344056 | 1212 | 2557 | 398.15 | 313.15 | -5 |

*Table T3.4.1; simulation case-setup*

Note; no use is made of the EGR-valves
Torque is controlled by additional PI-controller on the fuel injecting, which can be overruled by the smoke limiter (hence 'Min AF')

The VTG-position test is performed in order to gain insight on the effects of VTG rack-position, particularly on EGR, AF and FS. Several fixed VTG positions are simulated see case set-up.

Results

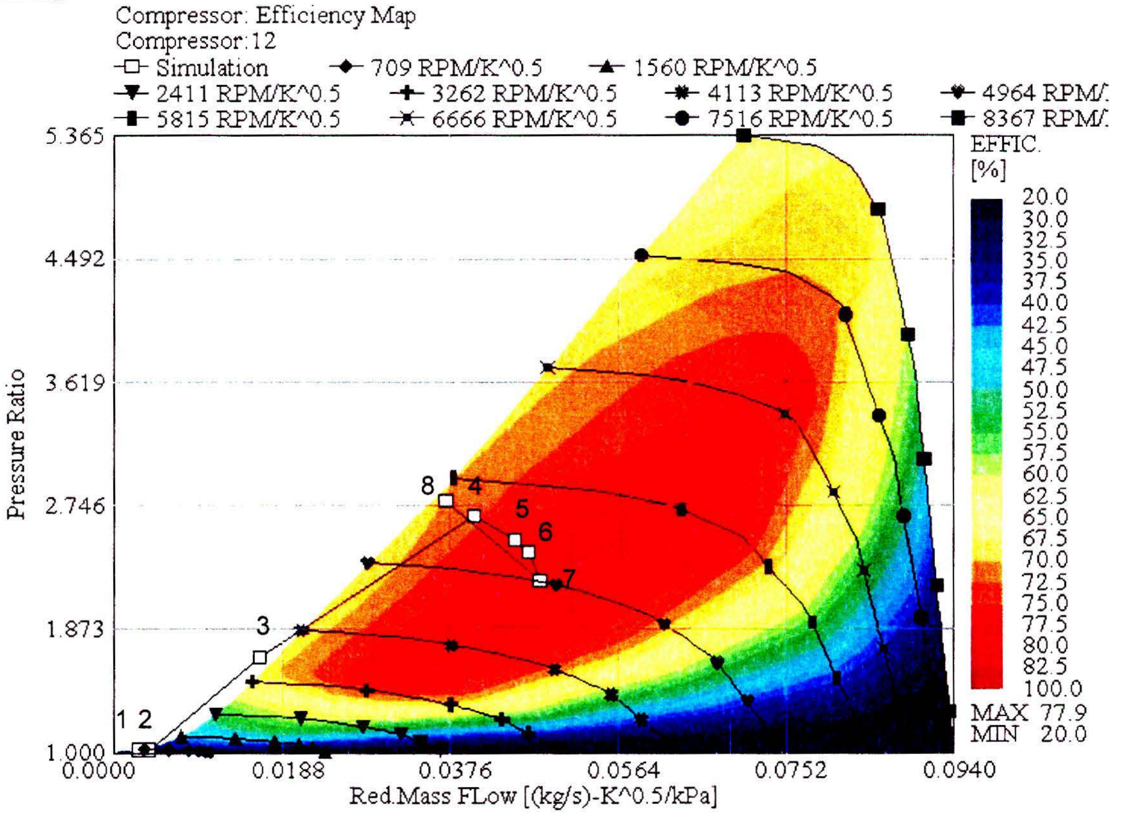



*Figure E3.4.1; VTG-Compressor (HP) efficiency map, with simulation results*

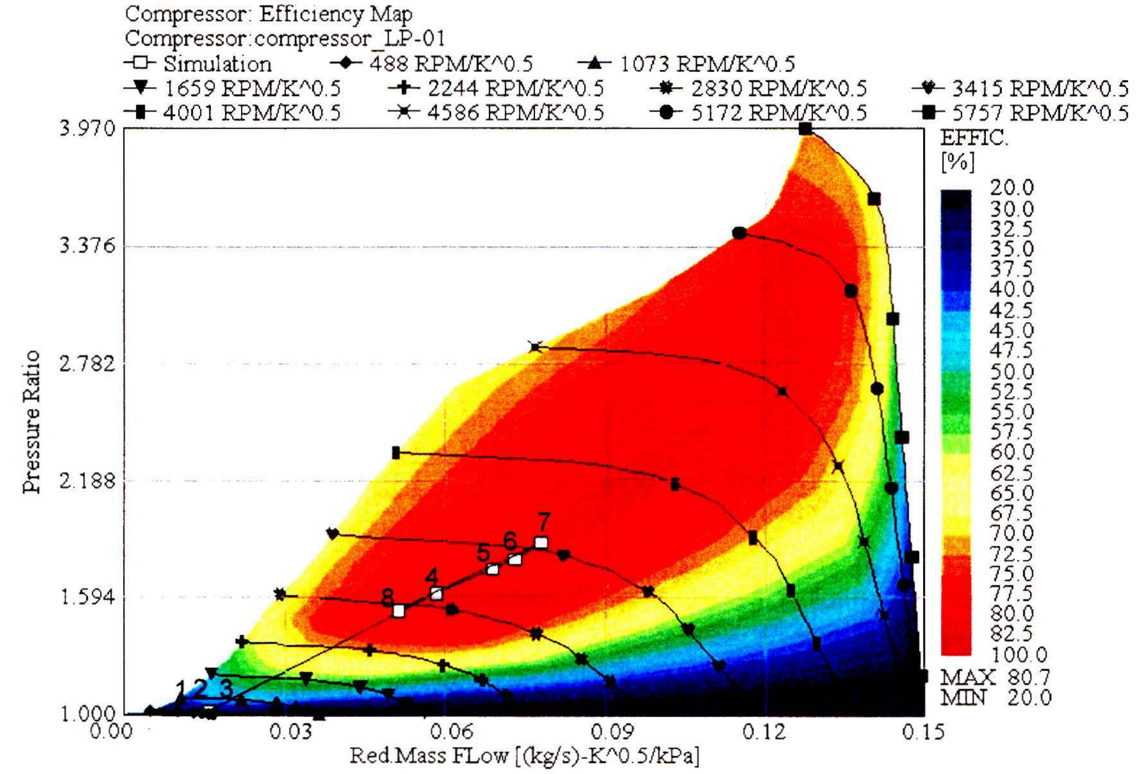



*Figure E3.4.2; GT-50 Compressor (LP) efficiency map, with simulation results. (points 1,2 & 3 are overlapping)*

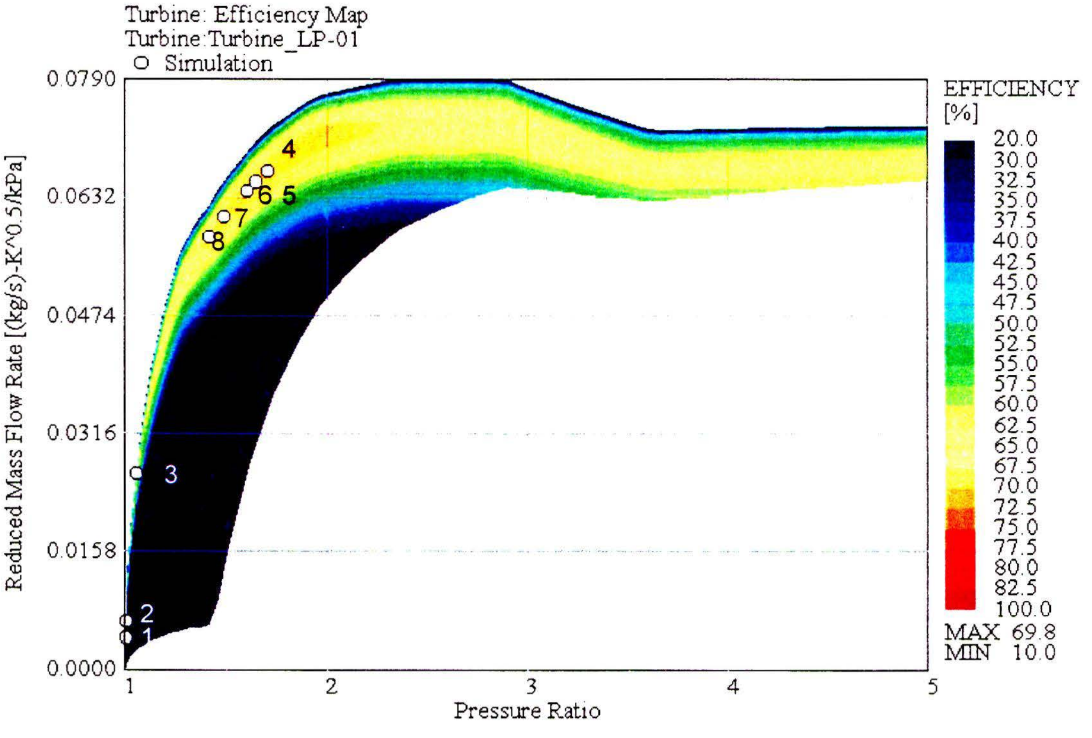



*Figure E3.4.3; GT-50 (LP) Turbine efficiency map, with simulation results*

As can been seen from figure E3.4.1 till E3.4.3 in case 1, 2 and 3 where the VTG is nearly closed, the engine's turbo-systems stalls and the High-Pressure compressor operating points are outside the compressor-map. GT-power continues calculating by extrapolating the turbo-maps to the point outside the turbo-map.

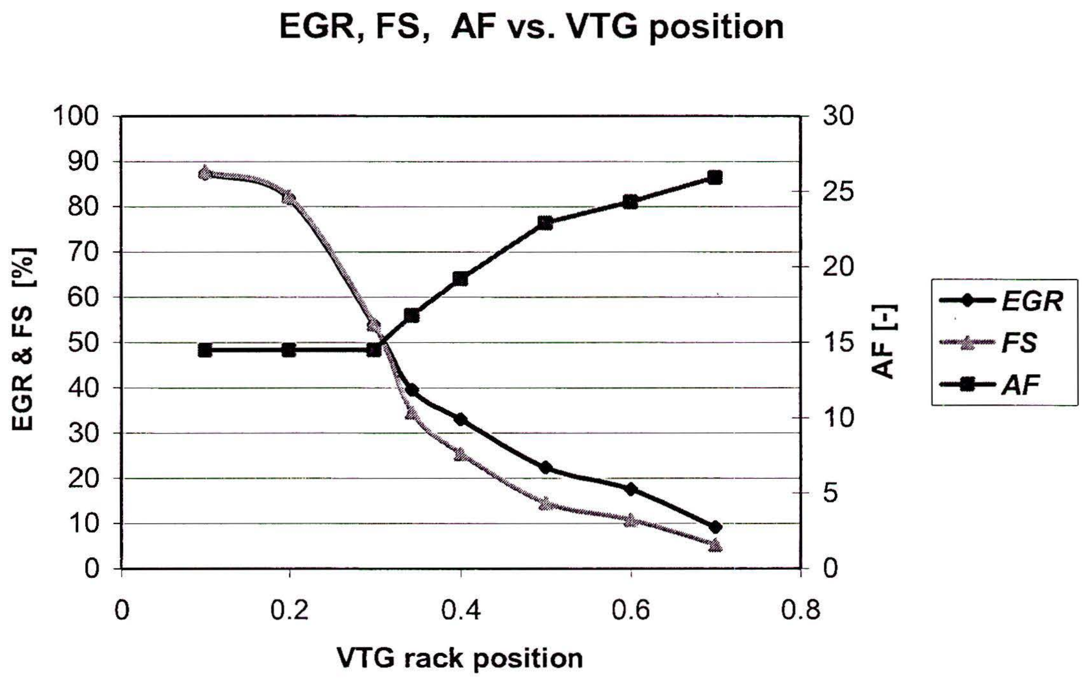



*Figure E3.4.4; EGR and AF level vs. case numbers*

It can be seen from figure E3.4.4) that large EGR levels are achieved at nearly closed VTG-positions (case 1 & 2). The further the VTG is closed the more EGR is obtained. Also the further the VTG is closed the lower the AF ratio becomes. The constant AF level of 14.5 at the lowest VTG positions is maintained by cutting the fuel-supply (smoke-limiter). Cutting the fuel-supply at the lowest VTG-positions reduces power and causes the turbo-system to stall as can be seen in the figures E3.4.1 till E3.4.3. The FS value is a function of EGR and AF and increases with lower VTG-rack positions.

<u>Final result</u>

VTG-position → 0 → EGR↑ → FS↑
→ AF↓ →

It is likely that for a stable engine operation at specific engine speed and load a specific FS value can only be found at a specific VTG position at a specific combination of EGR and AF. Meaning that changing the VTG position while maintaining engine speed and load constant will change the FS value as illustrated above.
The VTG-control strategy can be based on EGR, AF or FS level, in order to achieve the desired level for this parameter.

# Appendix 4 Simulation results

## Appendix 4.1 Simulation results; Internally driven- High pressure- single stage VTG turbo

| | Simulation number | | 1; vtg12 | 2; vtg12_c |
|---|---|---|---|---|
| | HP-compr | [-] | 1 | 0.7 |
| | HP-turb | [-] | 1 | 0.7 |
| | LP-compr | [-] | - | - |
| | LP-turb | [-] | - | - |
| | 2e interc | [K] | - | - |
| | Tegr | [K] | 398.15 | 398.15 |
| | inj.press | [bar] | profile | profile |
| Case | init VTGpos | [-] | 0.30 | 0.30 |
| 1 | EGR | [%] | 38.164 | 41.493 |
| ESC; A100 | AF | [-] | 15.6835 | 17.1061 |
| 2557 Nm | FS | [%] | 35.62 | 35.68 |
| 1212 rpm | BSFC | [g/kWh] | 200.644 | 198.095 |
| | BMEP | [bar] | 24.9177 | 25.0034 |
| | Power | [kW] | 324.693 | 325.809 |
| | turbo speed HP | [rpm] | [redacted] | [redacted] |
| | turbo speed LP | [rpm] | - | - |
| | Pmax | [bar] | 184.385 | 207.628 |
| 2 | EGR | [%] | 40.5002 | 37.3861 |
| ESC; B100 | AF | [-] | 16.6839 | 15.3203 |
| 2442 Nm | FS | [%] | 35.66 | 35.67 |
| 1530 rpm | BSFC | [g/kWh] | 201.101 | 203.555 |
| | BMEP | [bar] | 24.0969 | 23.8328 |
| | Power | [kW] | 396.382 | 395.053 |
| | turbo speed HP | [rpm] | [redacted] | [redacted] |
| | turbo speed LP | [rpm] | - | - |
| | Pmax | [bar] | 195.227 | 172.771 |
| 3 | EGR | [%] | 43.6574 | 41.2555 |
| ESC; C100 | AF | [-] | 18.0617 | 14.5996 |
| 2015 Nm | FS | [%] | 35.66 | 41.18 |
| 1830 rpm | BSFC | [g/kWh] | 206.664 | 210.755 |
| | BMEP | [bar] | 19.9396 | 20.079 |
| | Power | [kW] | 392.309 | 392.038 |
| | turbo speed HP | [rpm] | [redacted] | [redacted] |
| | turbo speed LP | [rpm] | - | - |
| | Pmax | [bar] | 189.465 | 152.649 |
| 4 | EGR | [%] | 43.7053 | 48.6616 |
| ESC; A50 | AF | [-] | 18.1135 | 20.2648 |
| 1279 Nm | FS | [%] | 35.60 | 35.63 |
| 1212 rpm | BSFC | [g/kWh] | 203.293 | 201.697 |
| | BMEP | [bar] | 12.4636 | 12.4455 |
| | Power | [kW] | 162.409 | 162.172 |
| | turbo speed HP | [rpm] | 88711.5 | [redacted] |
| | turbo speed LP | [rpm] | - | - |
| | Pmax | [bar] | 118.057 | 140.963 |
| 5 | EGR | [%] | 50.9222 | 54.7824 |
| ESC; C50 | AF | [-] | 21.2449 | 22.8785 |
| 1008 Nm | FS | [%] | 35.64 | 35.72 |
| 1830 rpm | BSFC | [g/kWh] | 220.9 | 227.532 |
| | BMEP | [bar] | 10.1143 | 9.712 |
| | Power | [kW] | 198.997 | 191.083 |
| | turbo speed HP | [rpm] | [redacted] | [redacted] |
| | turbo speed LP | [rpm] | - | - |
| | Pmax | [bar] | 138.923 | 159.984 |
| 6 | EGR | [%] | 49.3588 | 54.0248 |
| ESC; Rosi | AF | [-] | 20.6065 | 22.6225 |
| 707.4 Nm | FS | [%] | 35.57 | 35.61 |
| 1350 rpm | BSFC | [g/kWh] | 213.631 | 214.773 |
| | BMEP | [bar] | 6.90191 | 6.90361 |
| | Power | [kW] | 100.176 | 100.201 |
| | turbo speed HP | [rpm] | 69295.6 | 83482.5 |
| | turbo speed LP | [rpm] | - | - |
| | Pmax | [bar] | 86.3079 | 103.446 |
| Averages | Avrg. FS | [%] | 35.63567482 | 36.7715977 |
| | Avrg.IBSFC | [g/kWh] | 207.1869817 | 209.2362157 |
| | Avrg.AF | [%] | 18.14505956 | 18.04528416 |
| | max EGR | [%] | 50.9222 | 54.7824 |
| | min EGR | [%] | 38.16 | 37.39 |
| | min AF | [%] | 15.68 | 14.60 |
| | max. Cyl.Pmax | [bar] | 195.23 | 207.63 |
| | Turbo maps exceeded? | [case] | 6 | 4 |

* turbo over-speed

## Appendix 4.2 Simulation results; Internally driven- High pressure- two stage VTG turbo & fixed geometry turbo

- *2nd intercooler*

| | Simulation number | | 3 | 4 |
|---|---|---|---|---|
| | HP-compr | [-] | 0.7 | 0.7 |
| | HP-turb | [-] | 0.7 | 0.7 |
| | LP-compr | [-] | 0.7 | 0.7 |
| | LP-turb | [-] | 0.7 | 0.7 |
| | 2e interc | [K] | - | 300 |
| | Tegr | [K] | 398.15 | 398.15 |
| | inj.press | [bar] | profile | 1500 |
| **Case** | EGR-cooler size | [-] | single | single |
| 1 | EGR | [%] | 39.53 | 42.6423 |
| ESC; A100 | AF | [-] | 20.35 | 20.3246 |
| 2557 Nm | FS | [%] | 28.84 | 31.13 |
| 1212 rpm | BSFC | [g/kWh] | 194.493 | 193.961 |
| | BMEP | [bar] | 24.8519 | 24.8523 |
| | Power | [kW] | 323.835 | 323.84 |
| | turbo speed HP | [rpm] | [redacted] | [redacted] |
| | turbo speed LP | [rpm] | 64156.2 | 63059 |
| | Pmax | [bar] | 230.874 | 244.063 |
| 2 | EGR | [%] | 33.32 | 41.7832 |
| ESC; B100 | AF | [-] | 20.05 | 20.339 |
| 2442 Nm | FS | [%] | 24.65 | 30.49 |
| 1530 rpm | BSFC | [g/kWh] | 196.884 | 196.525 |
| | BMEP | [bar] | 23.8631 | 23.9214 |
| | Power | [kW] | 392.537 | 393.496 |
| | turbo speed HP | [rpm] | 90289.6 | 97499.6 |
| | turbo speed LP | [rpm] | 74173.5 | 71515 |
| | Pmax | [bar] | 202.658 | 237.184 |
| 3 | EGR | [%] | 40.37 | 44.3276 |
| ESC; C100 | AF | [-] | 19.64 | 20.3149 |
| 2015 Nm | FS | [%] | 30.45 | 32.38 |
| 1830 rpm | BSFC | [g/kWh] | 204.178 | 203.955 |
| | BMEP | [bar] | 20.0719 | 20.3573 |
| | Power | [kW] | 394.913 | 400.528 |
| | turbo speed HP | [rpm] | 89896.2 | 93599.8 |
| | turbo speed LP | [rpm] | 74394 | 73992.1 |
| | Pmax | [bar] | 193.444 | 220.579 |
| 4 | EGR | [%] | 45.7485 | 45.9064 |
| ESC; A50 | AF | [-] | 20.3222 | 20.2728 |
| 1279 Nm | FS | [%] | 33.41 | 33.60 |
| 1212 rpm | BSFC | [g/kWh] | 201.408 | 201.545 |
| | BMEP | [bar] | 12.4278 | 12.4332 |
| | Power | [kW] | 161.941 | 162.012 |
| | turbo speed HP | [rpm] | 88922.6 | 88076.7 |
| | turbo speed LP | [rpm] | 36807.7 | 36419.8 |
| | Pmax | [bar] | 133.842 | 131.996 |
| 5 | EGR | [%] | 60.8675 | 61.5466 |
| ESC; C50 | AF | [-] | 20.3183 | 20.3122 |
| 1008 Nm | FS | [%] | 44.46 | 44.96 |
| 1830 rpm | BSFC | [g/kWh] | 237.311 | 238.73 |
| | BMEP | [bar] | 9.54846 | 9.52381 |
| | Power | [kW] | 187.865 | 187.38 |
| | turbo speed HP | [rpm] | 97669.9 | 97772.4 |
| | turbo speed LP | [rpm] | 48593.8 | 48621.2 |
| | Pmax | [bar] | 168.489 | 171.56 |
| 6 | EGR | [%] | 51.8918 | 51.9386 |
| ESC; Rosi | AF | [-] | 20.2496 | 20.2485 |
| 707.4 Nm | FS | [%] | 38.02 | 38.06 |
| 1350 rpm | BSFC | [g/kWh] | 214.526 | 215.491 |
| | BMEP | [bar] | 6.87311 | 6.89567 |
| | Power | [kW] | 99.7582 | 100.086 |
| | turbo speed HP | [rpm] | 71240 | 71503.2 |
| | turbo speed LP | [rpm] | 19947 | 20144.4 |
| | Pmax | [bar] | 88.8978 | 86.7897 |
| **Averages** | Avrg. FS | [%] | 32.2843 | 34.61625 |
| | Avrg.IBSFC | [g/kWh] | 207.3954 | 207.496 |
| | Avrg.AF | [%] | 20.11354 | 20.31201 |
| | max EGR | [%] | 60.8675 | 61.5466 |
| | min EGR | [%] | 33.32 | 41.78 |
| | min AF | [%] | 19.64 | 20.25 |
| | max. Cyl.Pmax | [bar] | 230.87 | 244.06 |
| | Turbo maps exceeded? | [case] | 1 | 1 |

* turbo over-speed

- *turbo-scale*

| | Simulation number | | 5 | 6 | 4 | 7 | 8 |
|---|---|---|---|---|---|---|---|
| | HP-compr | [-] | 0.5 | 0.6 | 0.7 | 0.7 | 0.8 |
| | HP-turb | [-] | 0.5 | 0.6 | 0.7 | 0.7 | 0.8 |
| | LP-compr | [-] | 0.5 | 0.6 | 0.7 | 0.8 | 0.8 |
| | LP-turb | [-] | 0.5 | 0.6 | 0.7 | 0.8 | 0.8 |
| | 2e interc | [K] | 300 | 300 | 300 | 300 | 300 |
| | Tegr | [K] | 398.15 | 398.15 | 398.15 | 398.15 | 398.15 |
| | inj.press | [bar] | 1500 | 1500 | 1500 | 1500 | 1500 |
| **Case** | EGR-cooler size | [-] | single | single | single | single | single |
| 1 | EGR | [%] | 37.0565 | 45.6785 | 42.6423 | 40.6817 | 38.0404 |
| ESC; A100 | AF | [-] | 21.0504 | 20.3654 | 20.3246 | 20.2852 | 20.2875 |
| 2557 Nm | FS | [%] | 26.17 | 33.29 | 31.13 | 29.76 | 27.82 |
| 1212 rpm | BSFC | [g/kWh] | 193.034 | 194.147 | 193.961 | 193.849 | 193.814 |
| | BMEP | [bar] | 25.0551 | 24.8509 | 24.8523 | 24.8498 | 24.85 |
| | Power | [kW] | 326.483 | 323.822 | 323.84 | 323.808 | 323.81 |
| | turbo speed HP | [rpm] | 85352.7 | 102255 | [redacted] | [redacted] | [redacted] |
| | turbo speed LP | [rpm] | 80054.7 | 68371.3 | 63059 | 57863.7 | 58829.8 |
| | Pmax | [bar] | 232.272 | 257.705 | 244.063 | 236.497 | 227.034 |
| 2 | EGR | [%] | 48.0687 | 39.9671 | 41.7832 | 41.6712 | 40.7923 |
| ESC; B100 | AF | [-] | 17.6073 | 19.9202 | 20.339 | 20.3305 | 20.426 |
| 2442 Nm | FS | [%] | 40.22 | 29.75 | 30.49 | 30.42 | 29.64 |
| 1530 rpm | BSFC | [g/kWh] | 199.696 | 196.383 | 196.525 | 196.518 | 196.312 |
| | BMEP | [bar] | 24.272 | 24.0968 | 23.9214 | 23.9344 | 23.9575 |
| | Power | [kW] | 399.262 | 396.38 | 393.496 | 393.71 | 394.089 |
| | turbo speed HP | [rpm] | 86646.6 | 86215.5 | 97499.6 | [redacted] | 101841 |
| | turbo speed LP | [rpm] | 84250.9 | 79353.3 | 71515 | 66294.2 | 66726.1 |
| | Pmax | [bar] | 237.338 | 227.967 | 237.184 | 237.039 | 234.524 |
| 3 | EGR | [%] | 52.8331 | 45.8593 | 44.3276 | 44.3593 | 44.8231 |
| ESC; C100 | AF | [-] | 16.7563 | 19.3146 | 20.3149 | 20.383 | 20.3931 |
| 2015 Nm | FS | [%] | 46.33 | 35.15 | 32.38 | 32.30 | 32.62 |
| 1830 rpm | BSFC | [g/kWh] | 210.144 | 205.101 | 203.955 | 203.99 | 204.288 |
| | BMEP | [bar] | 20.6884 | 20.2422 | 20.3573 | 20.39 | 20.2794 |
| | Power | [kW] | 407.043 | 398.264 | 400.528 | 401.171 | 398.995 |
| | turbo speed HP | [rpm] | 87436.6 | 86343.5 | 93599.8 | 102117 | 100574 |
| | turbo speed LP | [rpm] | 86132 | 79733.5 | 73992.1 | 68602.3 | 68195.2 |
| | Pmax | [bar] | 224.7 | 215.732 | 220.579 | 221.81 | 222.86 |
| 4 | EGR | [%] | 58.411 | 53.8677 | 45.9064 | 42.2225 | 38.7927 |
| ESC; A50 | AF | [-] | 20.3558 | 20.3711 | 20.2728 | 20.2538 | 20.2687 |
| 1279 Nm | FS | [%] | 42.59 | 39.25 | 33.60 | 30.93 | 28.40 |
| 1212 rpm | BSFC | [g/kWh] | 204.331 | 202.784 | 201.545 | 201.425 | 201.437 |
| | BMEP | [bar] | 12.404 | 12.4214 | 12.4332 | 12.4333 | 12.4311 |
| | Power | [kW] | 161.631 | 161.859 | 162.012 | 162.014 | 161.985 |
| | turbo speed HP | [rpm] | 92396.5 | 90641.5 | 88076.7 | 90016 | 84982 |
| | turbo speed LP | [rpm] | 50561.9 | 45226.4 | 36419.8 | 29735.1 | 30788.3 |
| | Pmax | [bar] | 172.937 | 155.36 | 131.996 | 123.853 | 117.501 |
| 5 | EGR | [%] | 67.5553 | 64.6676 | 61.5466 | 59.4148 | 57.3504 |
| ESC; C50 | AF | [-] | 20.3079 | 20.3123 | 20.3122 | 20.3258 | 20.3161 |
| 1008 Nm | FS | [%] | 49.36 | 47.24 | 44.96 | 43.38 | 41.89 |
| 1830 rpm | BSFC | [g/kWh] | 262.529 | 249.233 | 238.73 | 233.427 | 229.589 |
| | BMEP | [bar] | 9.50674 | 9.52705 | 9.52381 | 9.52538 | 9.54104 |
| | Power | [kW] | 187.044 | 187.444 | 187.38 | 187.411 | 187.719 |
| | turbo speed HP | [rpm] | [redacted] | [redacted] | 97772.4 | [redacted] | 94196.2 |
| | turbo speed LP | [rpm] | 61877.5 | 54589.4 | 48621.2 | 42539.2 | 42806.8 |
| | Pmax | [bar] | 224.26 | 195.576 | 171.56 | 159.038 | 148.829 |
| 6 | EGR | [%] | 61.0627 | 56.0404 | 51.9386 | 51.9435 | 48.476 |
| ESC; Rosi | AF | [-] | 20.2426 | 20.2482 | 20.2485 | 20.2493 | 20.2499 |
| 707.4 Nm | FS | [%] | 44.76 | 41.06 | 38.06 | 38.06 | 35.52 |
| 1350 rpm | BSFC | [g/kWh] | 219.432 | 216.866 | 215.491 | 215.472 | 214.772 |
| | BMEP | [bar] | 6.87916 | 6.88107 | 6.89567 | 6.9039 | 6.90399 |
| | Power | [kW] | 99.8461 | 99.8738 | 100.086 | 100.205 | 100.206 |
| | turbo speed HP | [rpm] | 81769.9 | 76459.9 | 71503.2 | 73326 | 67503.6 |
| | turbo speed LP | [rpm] | 30469.7 | 24152.1 | 20144.4 | 17041.5 | 17310 |
| | Pmax | [bar] | 108.542 | 95.1705 | 86.7897 | 86.8705 | 80.9672 |
| **Averages** | Avrg. FS | [%] | 41.46078465 | 36.67655849 | 34.61625306 | 33.73342115 | 32.51721975 |
| | Avrg.IBSFC | [g/kWh] | 214.3567196 | 209.9214375 | 207.4960416 | 206.476512 | 205.7213401 |
| | Avrg.AF | [%] | 19.02332292 | 20.02273145 | 20.31200754 | 20.31759893 | 20.34379747 |
| | max EGR | [%] | 67.5553 | 64.6676 | 61.5466 | 59.4148 | 57.3504 |
| | min EGR | [%] | 37.06 | 39.97 | 41.78 | 40.68 | 38.04 |
| | min AF | [%] | 16.76 | 19.31 | 20.25 | 20.25 | 20.25 |
| | max. Cyl.Pmax | [bar] | 237.34 | 257.71 | 244.06 | 237.04 | 234.52 |
| | Turbo maps exceeded? | [case] | 1 | 2 | 1 | 3 | 1 |

* turbo over-speed

## - *EGR-cooler temperature*

| | Simulation number | | 9 | 10 | 11 | 12 | 4 |
|---|---|---|---|---|---|---|---|
| | HP-compr | [-] | 0.7 | 0.7 | 0.7 | 0.7 | 0.7 |
| | HP-turb | [-] | 0.7 | 0.7 | 0.7 | 0.7 | 0.7 |
| | LP-compr | [-] | 0.7 | 0.7 | 0.7 | 0.7 | 0.7 |
| | LP-turb | [-] | 0.7 | 0.7 | 0.7 | 0.7 | 0.7 |
| | 2e interc | [K] | 300 | 300 | 300 | 300 | 300 |
| | Tegr | [K] | 313 | 323 | 333 | 373 | 398.15 |
| | inj.press | [bar] | 1500 | 1500 | 1500 | 1500 | 1500 |
| **Case** | EGR-cooler size | [-] | single | single | single | single | single |
| **1** | EGR | [%] | 43.0018 | 43.4346 | 42.7494 | 42.7562 | 42.6423 |
| ESC; A100 | AF | [-] | 20.3331 | 20.2871 | 20.2981 | 20.2821 | 20.3246 |
| 2557 Nm | FS | [%] | 31.38 | 31.77 | 31.25 | 31.28 | 31.13 |
| 1212 rpm | BSFC | [g/kWh] | 191.567 | 191.666 | 192.109 | 193.151 | 193.961 |
| | BMEP | [bar] | 24.8621 | 24.8559 | 24.3959 | 24.8504 | 24.8523 |
| | Power | [kW] | 323.968 | 323.888 | 317.893 | 323.816 | 323.84 |
| | turbo speed HP | [rpm] | [redacted] | | | | |
| | turbo speed LP | [rpm] | 62134.6 | 62110.4 | 61330.3 | 62602.4 | 63059 |
| | Pmax | [bar] | 232.726 | 234.853 | 229.498 | 239.041 | 244.063 |
| **2** | EGR | [%] | 39.8643 | 40.7118 | 39.4704 | 41.0614 | 41.7832 |
| ESC; B100 | AF | [-] | 20.3518 | 20.3506 | 20.2518 | 20.3421 | 20.339 |
| 2442 Nm | FS | [%] | 29.07 | 29.69 | 28.92 | 29.96 | 30.49 |
| 1530 rpm | BSFC | [g/kWh] | 193.998 | 194.136 | 194.416 | 195.584 | 196.525 |
| | BMEP | [bar] | 24.0491 | 23.9785 | 23.7325 | 23.9384 | 23.9214 |
| | Power | [kW] | 395.596 | 394.435 | 390.387 | 393.775 | 393.496 |
| | turbo speed HP | [rpm] | 90700.9 | 92121.5 | 91733.9 | 94949.3 | 97499.6 |
| | turbo speed LP | [rpm] | 71860 | 71552.4 | 70631.8 | 71442.4 | 71515 |
| | Pmax | [bar] | 220.305 | 223.332 | 217.076 | 229.91 | 237.184 |
| **3** | EGR | [%] | 42.0102 | 43.623 | 41.5137 | 43.8031 | 44.3276 |
| ESC; C100 | AF | [-] | 20.0862 | 20.186 | 20.0469 | 20.3421 | 20.3149 |
| 2015 Nm | FS | [%] | 31.02 | 32.06 | 30.71 | 31.96 | 32.38 |
| 1830 rpm | BSFC | [g/kWh] | 201.067 | 201.402 | 201.332 | 203.031 | 203.955 |
| | BMEP | [bar] | 19.7149 | 19.9173 | 19.6185 | 20.0695 | 20.3573 |
| | Power | [kW] | 387.889 | 391.871 | 385.993 | 394.867 | 400.528 |
| | turbo speed HP | [rpm] | 85175.6 | 88174.5 | 85683.5 | 91209.7 | 93599.8 |
| | turbo speed LP | [rpm] | 72029.2 | 72394.3 | 71567 | 73009.9 | 73992.1 |
| | Pmax | [bar] | 192.138 | 201.509 | 191.251 | 210.597 | 220.579 |
| **4** | EGR | [%] | 45.8172 | 46.1856 | 45.2292 | 45.6973 | 45.9064 |
| ESC; A50 | AF | [-] | 20.2736 | 20.2749 | 20.3165 | 20.272 | 20.2728 |
| 1279 Nm | FS | [%] | 33.53 | 33.80 | 33.04 | 33.45 | 33.60 |
| 1212 rpm | BSFC | [g/kWh] | 198.807 | 198.881 | 199.491 | 200.568 | 201.545 |
| | BMEP | [bar] | 12.4355 | 12.4364 | 12.2213 | 12.4332 | 12.4332 |
| | Power | [kW] | 162.042 | 162.054 | 159.251 | 162.012 | 162.012 |
| | turbo speed HP | [rpm] | 84803.1 | 85403.3 | 84665.9 | 86677.4 | 88076.7 |
| | turbo speed LP | [rpm] | 35006.7 | 35178.7 | 34501.3 | 35790.1 | 36419.8 |
| | Pmax | [bar] | 124.067 | 125.517 | 122.502 | 128.539 | 131.996 |
| **5** | EGR | [%] | 62.6381 | 62.59 | 62.3974 | 61.8935 | 61.5466 |
| ESC; C50 | AF | [-] | 20.3132 | 20.313 | 20.298 | 20.3106 | 20.3122 |
| 1008 Nm | FS | [%] | 45.76 | 45.72 | 45.62 | 45.22 | 44.96 |
| 1830 rpm | BSFC | [g/kWh] | 232.944 | 233.823 | 234.577 | 236.977 | 238.73 |
| | BMEP | [bar] | 9.57855 | 9.56188 | 9.47111 | 9.54948 | 9.52381 |
| | Power | [kW] | 188.457 | 188.129 | 186.343 | 187.885 | 187.38 |
| | turbo speed HP | [rpm] | 94915.3 | 95332.8 | 95361 | 96907.6 | 97772.4 |
| | turbo speed LP | [rpm] | 47400.5 | 47545.4 | 47269.9 | 48168.5 | 48621.2 |
| | Pmax | [bar] | 163.621 | 164.679 | 163.739 | 168.88 | 171.56 |
| **6** | EGR | [%] | 53.3345 | 53.0565 | 52.929 | 51.9843 | 51.9386 |
| ESC; Rosi | AF | [-] | 20.1952 | 20.2473 | 20.2494 | 20.2497 | 20.2485 |
| 707.4 Nm | FS | [%] | 39.18 | 38.88 | 38.78 | 38.09 | 38.06 |
| 1350 rpm | BSFC | [g/kWh] | 212.114 | 212.228 | 212.612 | 214.302 | 215.491 |
| | BMEP | [bar] | 6.90376 | 6.90451 | 6.90376 | 6.89878 | 6.89567 |
| | Power | [kW] | 100.203 | 100.203 | 100.197 | 100.131 | 100.086 |
| | turbo speed HP | [rpm] | 67924.6 | 68264.9 | 68672 | 69664.3 | 71503.2 |
| | turbo speed LP | [rpm] | 19455.2 | 19546.6 | 19612.2 | 19842.3 | 20144.4 |
| | Pmax | [bar] | 83.1288 | 83.4413 | 83.8775 | 84.6951 | 86.7897 |
| **Averages** | Avrg. FS | [%] | 34.27593 | 34.70078 | 34.04376 | 34.46536 | 34.61625 |
| | Avrg.IBSFC | [g/kWh] | 204.2585 | 204.5514 | 204.9614 | 206.4298 | 207.496 |
| | Avrg.AF | [%] | 20.26757 | 20.28315 | 20.23485 | 20.31092 | 20.31201 |
| | max EGR | [%] | 62.6381 | 62.59 | 62.3974 | 61.8935 | 61.5466 |
| | min EGR | [%] | 39.86 | 40.71 | 39.47 | 41.06 | 41.78 |
| | min AF | [%] | 20.09 | 20.19 | 20.05 | 20.25 | 20.25 |
| | max. Cyl.Pmax | [bar] | 232.73 | 234.85 | 229.50 | 239.04 | 244.06 |
| | Turbo maps exceeded? | [case] | 1 | 1 | 1 | 1 | 1 |

* turbo over-speed

## EGR-cooler size

| | Simulation number | | 13 | 14 | 15 | 16 | 17 |
|---|---|---|---|---|---|---|---|
| | HP-compr | [-] | 0.7 | 0.7 | 0.7 | 0.7 | 0.7 |
| | HP-turb | [-] | 0.7 | 0.7 | 0.7 | 0.7 | 0.7 |
| | LP-compr | [-] | 0.7 | 0.7 | 0.7 | 0.7 | 0.7 |
| | LP-turb | [-] | 0.7 | 0.7 | 0.7 | 0.7 | 0.7 |
| | 2e interc | [K] | 300 | 300 | 300 | 300 | 300 |
| | Tegr | [K] | 313 | 323 | 333 | 373 | 398.15 |
| | inj.press | [bar] | 1500 | 1500 | 1500 | 1500 | 1500 |
| **Case** | EGR-cooler size | [-] | double | double | double | double | double |
| **1** | EGR | [%] | 42.7266 | 42.786 | 42.7495 | 42.4602 | 42.2623 |
| ESC; A100 | AF | [-] | 20.3023 | 20.3344 | 20.334 | 20.2938 | 20.2949 |
| 2557 Nm | FS | [%] | 31.23 | 31.23 | 31.20 | 31.05 | 30.90 |
| 1212 rpm | BSFC | [g/kWh] | 191.068 | 191.313 | 191.566 | 192.676 | 193.338 |
| | BMEP | [bar] | 24.4329 | 24.8737 | 24.8628 | 24.3728 | 24.3506 |
| | Power | [kW] | 318.376 | 324.12 | 323.977 | 317.592 | 317.303 |
| | turbo speed HP | [rpm] | 99911.8 | 100629 | 100955 | [redacted] | [redacted] |
| | turbo speed LP | [rpm] | 61147 | 62104.1 | 62158.1 | 61509.1 | 61687.2 |
| | Pmax | [bar] | 225.785 | 231.607 | 232.834 | 231.906 | 234.323 |
| **2** | EGR | [%] | 43.2288 | 39.778 | 39.9946 | 43.7946 | 43.9266 |
| ESC; B100 | AF | [-] | 20.2216 | 20.3508 | 20.3489 | 20.2231 | 20.2238 |
| 2442 Nm | FS | [%] | 31.72 | 29.01 | 29.17 | 32.13 | 32.23 |
| 1530 rpm | BSFC | [g/kWh] | 194.279 | 194.002 | 194.22 | 195.935 | 196.644 |
| | BMEP | [bar] | 23.0497 | 23.9903 | 23.9649 | 23.0725 | 23.063 |
| | Power | [kW] | 379.157 | 394.629 | 394.211 | 379.532 | 379.375 |
| | turbo speed HP | [rpm] | 93751.7 | 90686 | 91400.8 | 97326.2 | 98657 |
| | turbo speed LP | [rpm] | 68984.6 | 71671.7 | 71567.1 | 69130.2 | 69208.8 |
| | Pmax | [bar] | 219.817 | 219.439 | 221.185 | 229.574 | 233.26 |
| **3** | EGR | [%] | 42.5409 | 42.2087 | 41.8927 | 42.8909 | 42.4627 |
| ESC; C100 | AF | [-] | 19.9326 | 20.0885 | 20.1076 | 20.1489 | 20.2233 |
| 2015 Nm | FS | [%] | 31.64 | 31.16 | 30.90 | 31.58 | 31.15 |
| 1830 rpm | BSFC | [g/kWh] | 200.807 | 200.976 | 201.212 | 202.469 | 203.039 |
| | BMEP | [bar] | 19.5982 | 19.7551 | 19.7019 | 19.6118 | 19.6276 |
| | Power | [kW] | 385.593 | 388.68 | 387.632 | 385.86 | 386.172 |
| | turbo speed HP | [rpm] | 85314.6 | 85205.6 | 85330.1 | 89359.5 | 90154.9 |
| | turbo speed LP | [rpm] | 71084.4 | 72089 | 72026.5 | 71354.3 | 71621.4 |
| | Pmax | [bar] | 189.491 | 192.704 | 192.304 | 199.36 | 201.685 |
| **4** | EGR | [%] | 45.5715 | 45.527 | 45.511 | 44.8001 | 44.5511 |
| ESC; A50 | AF | [-] | 20.2643 | 20.2702 | 20.2706 | 20.3137 | 20.325 |
| 1279 Nm | FS | [%] | 33.37 | 33.33 | 33.31 | 32.73 | 32.53 |
| 1212 rpm | BSFC | [g/kWh] | 198.637 | 198.666 | 198.94 | 200.187 | 201.003 |
| | BMEP | [bar] | 12.0672 | 12.4368 | 12.4371 | 12.2587 | 12.2142 |
| | Power | [kW] | 157.243 | 162.059 | 162.063 | 159.738 | 159.158 |
| | turbo speed HP | [rpm] | 83511.3 | 84645.1 | 84964.2 | 85481.2 | 86094.5 |
| | turbo speed LP | [rpm] | 33474.9 | 34869.6 | 35004.9 | 34885.9 | 35007.5 |
| | Pmax | [bar] | 119.332 | 123.584 | 124.37 | 124.398 | 125.418 |
| **5** | EGR | [%] | 62.9507 | 62.7985 | 62.6612 | 62.1933 | 61.8941 |
| ESC; C50 | AF | [-] | 20.306 | 20.3198 | 20.3197 | 20.3037 | 20.2965 |
| 1008 Nm | FS | [%] | 46.00 | 45.86 | 45.76 | 45.45 | 45.25 |
| 1830 rpm | BSFC | [g/kWh] | 233.129 | 233.778 | 234.518 | 237.633 | 239.401 |
| | BMEP | [bar] | 9.48152 | 9.58452 | 9.57626 | 9.4722 | 9.46848 |
| | Power | [kW] | 186.548 | 188.575 | 188.412 | 186.365 | 186.292 |
| | turbo speed HP | [rpm] | 94597.4 | 95248.2 | 95583.7 | 96856.9 | [redacted] |
| | turbo speed LP | [rpm] | 46843.8 | 47428.6 | 47561.6 | 47771.8 | 48144.8 |
| | Pmax | [bar] | 161.446 | 164.313 | 165.176 | 167.867 | 170.56 |
| **6** | EGR | [%] | 53.6269 | 53.1831 | 52.8167 | 52.423 | 52.0999 |
| ESC; Rosi | AF | [-] | 20.2042 | 20.1939 | 20.2087 | 20.248 | 20.2467 |
| 707.4 Nm | FS | [%] | 39.38 | 39.07 | 38.77 | 38.41 | 38.18 |
| 1350 rpm | BSFC | [g/kWh] | 211.542 | 212.083 | 212.433 | 213.761 | 214.718 |
| | BMEP | [bar] | 6.90332 | 6.9031 | 6.90308 | 6.90331 | 6.90332 |
| | Power | [kW] | 100.214 | 100.193 | 100.193 | 100.197 | 100.197 |
| | turbo speed HP | [rpm] | 67814.8 | 67826.3 | 67983.1 | 69844.2 | 70814.3 |
| | turbo speed LP | [rpm] | 19384.7 | 19418.9 | 19478.7 | 19796.1 | 19982 |
| | Pmax | [bar] | 83.168 | 83.1235 | 83.184 | 85.1242 | 86.1419 |
| **Averages** | Avrg. FS | [%] | 35.11439 | 34.25158 | 34.19474 | 34.91197 | 34.74548 |
| | Avrg.IBSFC | [g/kWh] | 204.2782 | 204.3442 | 204.6873 | 206.4835 | 207.3925 |
| | Avrg.AF | [%] | 20.19672 | 20.26861 | 20.27305 | 20.24819 | 20.26342 |
| | max EGR | [%] | 62.9507 | 62.7985 | 62.6612 | 62.1933 | 61.8941 |
| | min EGR | [%] | 42.54 | 39.78 | 39.99 | 42.46 | 42.26 |
| | min AF | [%] | 19.93 | 20.09 | 20.11 | 20.15 | 20.22 |
| | max. Cyl.Pmax | [bar] | 225.79 | 231.61 | 232.83 | 231.91 | 234.32 |
| | Turbo maps exceeded? | [case] | 0 | 0 | 0 | 1 | 2 |

* turbo over-speed

## Appendix 4.3 Simulation results; Internally driven- Low pressure- two stage VTG turbo & fixed geometry turbo

- *turbo scale*

| | Simulation number | | 18 | 18 | 18 | 18 | 18 | 18 |
|---|---|---|---|---|---|---|---|---|
| | HP-compr | [-] | 0.7 | 0.8 | 0.9 | 1 | 1.25 | 1.5 |
| | HP-turb | [-] | 0.7 | 0.8 | 0.9 | 1 | 1.25 | 1.5 |
| | LP-compr | [-] | 0.7 | 0.8 | 0.9 | 1 | 1.25 | 1.5 |
| | LP-turb | [-] | 0.7 | 0.8 | 0.9 | 1 | 1.25 | 1.5 |
| | 2e interc | [K] | 300 | 300 | 300 | 300 | 300 | 300 |
| | Tegr | [K] | 293 | 293 | 293 | 293 | 293 | 293 |
| | inj.press | [bar] | 1500 | 1500 | 1500 | 1500 | 1500 | 1500 |
| | VTG-rack pos. | [-] | 1.00 | 1.00 | 1.00 | 1.00 | 1.00 | 1.00 |
| **Case** | EGR-cooler size | [-] | 5 | 5 | 5 | 5 | 5 | 5 |
| 1 | EGR | [%] | | | | | | |
| ESC; A100 | AF | [-] | | | | | | |
| 2557 Nm | FS | [%] | | | | | | |
| 1212 rpm | BSFC | [g/kWh] | | | | | | |
| | BMEP | [bar] | | | | | | |
| | Power | [kW] | | | | | | |
| | turbo speed HP | [rpm] | | | | | | |
| | turbo speed LP | [rpm] | | | | | | |
| | Pmax | [bar] | | | | | | |
| 2 | EGR | [%] | 44.536 | 44.4218 | 44.3946 | 44.4656 | 44.6215 | 45.891 |
| ESC; B100 | AF | [-] | 14.4778 | 16.1401 | 17.0977 | 16.7722 | 14.5066 | 14.5026 |
| 2442 Nm | FS | [%] | 44.80 | 40.35 | 38.19 | 38.96 | 44.80 | 46.09 |
| 1530 rpm | BSFC | [g/kWh] | 222.804 | 210.957 | 203.002 | 199.251 | 198.591 | 213.952 |
| | BMEP | [bar] | 22.4754 | 23.7232 | 23.9288 | 24.0556 | 22.1437 | 9.70065 |
| | Power | [kW] | 369.71 | 390.234 | 393.617 | 395.366 | 364.254 | 159.571 |
| | turbo speed HP | [rpm] | 93468.9 | 90942.4 | 90317 | 89076.7 | 84713.7 | 67676.5 |
| | turbo speed LP | [rpm] | 95877.6 | 89556.6 | 81775.6 | 76536.7 | 62820.2 | 24338.8 |
| | Pmax | [bar] | 186.95 | 204.655 | 207.735 | 200.183 | 159.362 | 75.579 |
| 3 | EGR | [%] | | | | | | |
| ESC; C100 | AF | [-] | | | | | | |
| 2015 Nm | FS | [%] | | | | | | |
| 1830 rpm | BSFC | [g/kWh] | | | | | | |
| | BMEP | [bar] | | | | | | |
| | Power | [kW] | | | | | | |
| | turbo speed HP | [rpm] | | | | | | |
| | turbo speed LP | [rpm] | | | | | | |
| | Pmax | [bar] | | | | | | |
| 4 | EGR | [%] | | | | | | |
| ESC; A50 | AF | [-] | | | | | | |
| 1279 Nm | FS | [%] | | | | | | |
| 1212 rpm | BSFC | [g/kWh] | | | | | | |
| | BMEP | [bar] | | | | | | |
| | Power | [kW] | | | | | | |
| | turbo speed HP | [rpm] | | | | | | |
| | turbo speed LP | [rpm] | | | | | | |
| | Pmax | [bar] | | | | | | |
| 5 | EGR | [%] | | | | | | |
| ESC; C50 | AF | [-] | | | | | | |
| 1008 Nm | FS | [%] | | | | | | |
| 1830 rpm | BSFC | [g/kWh] | | | | | | |
| | BMEP | [bar] | | | | | | |
| | Power | [kW] | | | | | | |
| | turbo speed HP | [rpm] | | | | | | |
| | turbo speed LP | [rpm] | | | | | | |
| | Pmax | [bar] | | | | | | |
| 6 | EGR | [%] | | | | | | |
| ESC; Rosi | AF | [-] | | | | | | |
| 707.4 Nm | FS | [%] | | | | | | |
| 1350 rpm | BSFC | [g/kWh] | | | | | | |
| | BMEP | [bar] | | | | | | |
| | Power | [kW] | | | | | | |
| | turbo speed HP | [rpm] | | | | | | |
| | turbo speed LP | [rpm] | | | | | | |
| | Pmax | [bar] | | | | | | |
| **Averages** | Avrg. FS | [%] | 44.80 | 40.35 | 38.19 | 38.96 | 44.80 | 46.09 |
| | Avrg.IBSFC | [g/kWh] | 222.80 | 210.96 | 203.00 | 199.25 | 198.59 | 213.95 |
| | Avrg.AF | [%] | 14.48 | 16.14 | 17.10 | 16.77 | 14.51 | 14.50 |
| | max EGR | [%] | 44.54 | 44.42 | 44.39 | 44.47 | 44.62 | 45.89 |
| | min EGR | [%] | 44.54 | 44.42 | 44.39 | 44.47 | 44.62 | 45.89 |
| | min AF | [%] | 14.48 | 16.14 | 17.10 | 16.77 | 14.51 | 14.50 |
| | max. Cyl.Pmax | [bar] | 186.95 | 204.66 | 207.74 | 200.18 | 159.36 | 75.58 |

## - *VTG rack-sweep & EGR-cooler temperature*

| | Simulation number | | 19 | 19 | 19 | 19 | selection 19 |
|---|---|---|---|---|---|---|---|
| | HP-compr | [-] | 1 | 1 | 1 | 1 | 1 |
| | HP-turb | [-] | 1 | 1 | 1 | 1 | 1 |
| | LP-compr | [-] | 1 | 1 | 1 | 1 | 1 |
| | LP-turb | [-] | 1 | 1 | 1 | 1 | 1 |
| | 2e interc | [K] | 300 | 300 | 300 | 300 | 300 |
| | Tegr | [K] | 313 | 313 | 313 | 313 | **313** |
| | inj.press | [bar] | 1500 | 1500 | 1500 | 1500 | 1500 |
| | VTG-rack pos. | [-] | 0.25 | 0.50 | 0.75 | 1.00 | 0.5 & 0.75 |
| **Case** | EGR-cooler size | [-] | 5 | 5 | 5 | 5 | 5 |
| **1** | EGR | [%] | 39.8488 | ~~41.1346~~ | 40.404 | 32.462 | 40.404 |
| ESC; A100 | AF | [-] | 20.2876 | ~~21.2623~~ | 20.316 | 20.281 | 20.316 |
| 2557 Nm | FS | [%] | 29.15 | ~~28.77~~ | 29.51 | 23.75 | 29.51 |
| 1212 rpm | BSFC | [g/kWh] | 255.991 | ~~199.694~~ | 189.48 | 188.03 | 189.48 |
| | BMEP | [bar] | 24.1848 | ~~24.8632~~ | 24.918 | 24.942 | 24.918 |
| | Power | [kW] | 315.143 | ~~323.983~~ | 324.69 | 325.02 | 324.69 |
| | turbo speed HP | [rpm] | [redacted] | [redacted] | 91210 | 82890 | 91210 |
| | turbo speed LP | [rpm] | 77785.5 | ~~68157.1~~ | 67269 | 65708 | 67269 |
| | Pmax | [bar] | 306.84 | ~~246.85~~ | 219.39 | 196.23 | 219.39 |
| **2** | EGR | [%] | 21.6286 | 39.2814 | 40.166 | 36.688 | 40.166 |
| ESC; B100 | AF | [-] | 20.2852 | 20.3027 | 20.315 | 20.311 | 20.315 |
| 2442 Nm | FS | [%] | 15.82 | 28.71 | 29.34 | 26.80 | 29.34 |
| 1530 rpm | BSFC | [g/kWh] | 289.485 | 218.118 | 200.64 | 195.6 | 200.64 |
| | BMEP | [bar] | 23.1341 | 23.5479 | 23.744 | 23.835 | 23.744 |
| | Power | [kW] | 380.545 | 387.351 | 390.57 | 392.07 | 390.57 |
| | turbo speed HP | [rpm] | [redacted] | [redacted] | 98376 | 87707 | 98376 |
| | turbo speed LP | [rpm] | 87190.2 | 78806.5 | 76072 | 75270 | 76072 |
| | Pmax | [bar] | 277.482 | 242.986 | 224.9 | 208.96 | 224.9 |
| **3** | EGR | [%] | 5.66182 | 35.7584 | 39.922 | 38.737 | 39.922 |
| ESC; C100 | AF | [-] | 20.2804 | 20.2974 | 20.319 | 20.315 | 20.319 |
| 2015 Nm | FS | [%] | 4.14 | 26.14 | 29.16 | 28.30 | 29.16 |
| 1830 rpm | BSFC | [g/kWh] | 333.431 | 241.683 | 217.52 | 209.24 | 217.52 |
| | BMEP | [bar] | 19.5736 | 19.2072 | 19.615 | 19.676 | 19.615 |
| | Power | [kW] | 385.108 | 377.899 | 385.92 | 387.13 | 385.92 |
| | turbo speed HP | [rpm] | [redacted] | [redacted] | 101940 | 90339 | 101940 |
| | turbo speed LP | [rpm] | 94902.6 | 81710.6 | 79789 | 78379 | 79789 |
| | Pmax | [bar] | 242.139 | 213.67 | 203.79 | 192.02 | 203.79 |
| **4** | EGR | [%] | 56.1623 | 59.517 | 52.0135 | 42.368 | 59.517 |
| ESC; A50 | AF | [-] | 20.2779 | 20.306 | 20.2694 | 20.286 | 20.306 |
| 1279 Nm | FS | [%] | 41.10 | 43.49 | 38.08 | 30.99 | 43.49 |
| 1212 rpm | BSFC | [g/kWh] | 286.902 | 208.52 | 195.689 | 194.48 | 208.52 |
| | BMEP | [bar] | 12.3332 | 12.464 | 12.4794 | 12.475 | 12.464 |
| | Power | [kW] | 160.709 | 162.42 | 162.614 | 162.56 | 162.42 |
| | turbo speed HP | [rpm] | [redacted] | 95056 | 79289 | 68285 | 95056 |
| | turbo speed LP | [rpm] | 64307.9 | 53642 | 50646.1 | 46665 | 53642 |
| | Pmax | [bar] | 228.06 | 170.13 | 134.679 | 114.03 | 170.13 |
| **5** | EGR | [%] | 39.3414 | 58.3453 | 59.668 | 57.589 | 59.668 |
| ESC; C50 | AF | [-] | 20.2634 | 20.2578 | 20.281 | 20.271 | 20.281 |
| 1008 Nm | FS | [%] | 28.81 | 42.73 | 43.66 | 42.15 | 43.66 |
| 1830 rpm | BSFC | [g/kWh] | 460.514 | 281.86 | 239.37 | 225.77 | 239.37 |
| | BMEP | [bar] | 9.72933 | 9.58393 | 9.8376 | 9.8984 | 9.8376 |
| | Power | [kW] | 191.424 | 188.563 | 193.55 | 194.75 | 193.55 |
| | turbo speed HP | [rpm] | [redacted] | [redacted] | 91402 | 81840 | 91402 |
| | turbo speed LP | [rpm] | 83878.8 | 70796.2 | 67160 | 65178 | 67160 |
| | Pmax | [bar] | 229.8 | 178.435 | 157.75 | 141.79 | 157.75 |
| **6** | EGR | [%] | 62.7817 | ~~58.849~~ | 59.254 | 51.472 | 59.254 |
| ESC; Rosi | AF | [-] | 20.2671 | ~~24.7567~~ | 20.287 | 20.283 | 20.287 |
| 707.4 Nm | FS | [%] | 45.96 | ~~35.57~~ | 43.34 | 37.66 | 43.34 |
| 1350 rpm | BSFC | [g/kWh] | 373.128 | ~~234.912~~ | 214.27 | 211.49 | 214.27 |
| | BMEP | [bar] | 6.88709 | ~~6.89842~~ | 6.8994 | 6.8877 | 6.8994 |
| | Power | [kW] | 99.9612 | ~~100.126~~ | 100.14 | 99.969 | 100.14 |
| | turbo speed HP | [rpm] | [redacted] | ~~89198.7~~ | 72119 | 62298 | 72119 |
| | turbo speed LP | [rpm] | 60478.5 | ~~45085.9~~ | 35578 | 30903 | 35578 |
| | Pmax | [bar] | 193.337 | ~~125.92~~ | 94.883 | 79.48 | 94.883 |
| **Averages** | Avrg. FS | [%] | 23.54 | 33.23 | 34.20 | 30.87 | 34.86 |
| | Avrg.IBSFC | [g/kWh] | 331.88 | 231.83 | 209.99 | 203.91 | 211.57 |
| | Avrg.AF | [%] | 20.28 | 20.78 | 20.30 | 20.29 | 20.31 |
| | max EGR | [%] | 62.78 | 59.52 | 59.67 | 57.59 | 59.67 |
| | min EGR | [%] | 5.66 | 35.76 | 39.92 | 32.46 | 39.92 |
| | min AF | [%] | 20.26 | 20.26 | 20.27 | 20.27 | 20.28 |
| | max. Cyl.Pmax | [bar] | 306.84 | 246.85 | 224.90 | 208.96 | 224.90 |
| | Turbo maps exceeded? | [case] | 6 | 5 | 0 | 0 | 0 |

* turbo over-speed

## - *VTG rack-sweep & EGR-cooler temperature*

| Case | Simulation number | | 20 | 20 | 20 | 20 | selection 20 |
|---|---|---|---|---|---|---|---|
| | HP-compr | [-] | 1 | 1 | 1 | 1 | 1 |
| | HP-turb | [-] | 1 | 1 | 1 | 1 | 1 |
| | LP-compr | [-] | 1 | 1 | 1 | 1 | 1 |
| | LP-turb | [-] | 1 | 1 | 1 | 1 | 1 |
| | 2e interc | [K] | 300 | 300 | 300 | 300 | 300 |
| | Tegr | [K] | 323 | 323 | 323 | 323 | **323** |
| | inj.press | [bar] | 1500 | 1500 | 1500 | 1500 | 1500 |
| | VTG-rack pos. | [-] | 0.25 | 0.50 | 0.75 | 1.00 | 0.5 & 0.75 |
| Case | EGR-cooler size | [-] | 5 | 5 | 5 | 5 | 5 |
| 1 | EGR | [%] | 39.364 | 40.98 | 40.231 | 32.056 | 40.231 |
| ESC; A100 | AF | [-] | 20.294 | 21.134 | 20.311 | 20.283 | 20.311 |
| 2557 Nm | FS | [%] | 28.78 | 28.83 | 29.39 | 23.45 | 29.39 |
| 1212 rpm | BSFC | [g/kWh] | 255.63 | 199.82 | 189.59 | 188.16 | 189.59 |
| | BMEP | [bar] | 24.136 | 24.86 | 24.918 | 24.944 | 24.918 |
| | Power | [kW] | 314.51 | 323.94 | 324.7 | 325.03 | 324.7 |
| | turbo speed HP | [rpm] | [redacted] | [redacted] | 91233 | 82952 | 91233 |
| | turbo speed LP | [rpm] | 77751 | 68273 | 67684 | 65772 | 67684 |
| | Pmax | [bar] | 303.99 | 245.21 | 218.93 | 195.5 | 218.93 |
| 2 | EGR | [%] | 21.289 | 38.865 | 39.967 | 36.2 | 39.967 |
| ESC; B100 | AF | [-] | 20.286 | 20.303 | 20.313 | 20.308 | 20.313 |
| 2442 Nm | FS | [%] | 15.57 | 28.41 | 29.20 | 26.45 | 29.20 |
| 1530 rpm | BSFC | [g/kWh] | 289.21 | 218.1 | 200.76 | 195.72 | 200.76 |
| | BMEP | [bar] | 23.156 | 23.542 | 23.741 | 23.835 | 23.741 |
| | Power | [kW] | 380.91 | 387.26 | 390.53 | 392.07 | 390.53 |
| | turbo speed HP | [rpm] | [redacted] | [redacted] | 98305 | 87827 | 98305 |
| | turbo speed LP | [rpm] | 87246 | 78971 | 76663 | 75382 | 76663 |
| | Pmax | [bar] | 276.45 | 241.46 | 224.47 | 207.62 | 224.47 |
| 3 | EGR | [%] | 5.5953 | 35.465 | 39.657 | 38.231 | 39.657 |
| ESC; C100 | AF | [-] | 20.282 | 20.311 | 20.318 | 20.317 | 20.318 |
| 2015 Nm | FS | [%] | 4.09 | 25.91 | 28.96 | 27.92 | 28.96 |
| 1830 rpm | BSFC | [g/kWh] | 333.31 | 241.49 | 217.53 | 209.25 | 217.53 |
| | BMEP | [bar] | 19.574 | 19.161 | 19.614 | 19.678 | 19.614 |
| | Power | [kW] | 385.12 | 377 | 385.91 | 387.17 | 385.91 |
| | turbo speed HP | [rpm] | [redacted] | [redacted] | 101800 | 90480 | 101800 |
| | turbo speed LP | [rpm] | 94926 | 81811 | 80347 | 78503 | 80347 |
| | Pmax | [bar] | 242.13 | 212.31 | 203.41 | 190.75 | 203.41 |
| 4 | EGR | [%] | 55.898 | 59.175 | 51.576 | 42.092 | 59.175 |
| ESC; A50 | AF | [-] | 20.294 | 20.309 | 20.276 | 20.285 | 20.309 |
| 1279 Nm | FS | [%] | 40.87 | 43.24 | 37.74 | 30.79 | 43.24 |
| 1212 rpm | BSFC | [g/kWh] | 286.69 | 208.58 | 195.84 | 194.59 | 208.58 |
| | BMEP | [bar] | 12.255 | 12.46 | 12.481 | 12.475 | 12.46 |
| | Power | [kW] | 159.69 | 162.36 | 162.63 | 162.56 | 162.36 |
| | turbo speed HP | [rpm] | [redacted] | 95148 | 79311 | 68301 | 95148 |
| | turbo speed LP | [rpm] | 64408 | 53815 | 50714 | 46744 | 53815 |
| | Pmax | [bar] | 225.45 | 168.79 | 133.76 | 113.75 | 168.79 |
| 5 | EGR | [%] | ~~39.098~~ | 57.875 | 59.139 | 57.047 | 59.139 |
| ESC; C50 | AF | [-] | ~~20.2~~ | 20.279 | 20.274 | 20.281 | 20.274 |
| 1008 Nm | FS | [%] | ~~28.71~~ | 42.35 | 43.28 | 41.74 | 43.28 |
| 1830 rpm | BSFC | [g/kWh] | ~~459.14~~ | 281.08 | 238.98 | 225.65 | 238.98 |
| | BMEP | [bar] | ~~9.7722~~ | 9.5823 | 9.8414 | 9.9006 | 9.8414 |
| | Power | [kW] | ~~192.27~~ | 188.53 | 193.63 | 194.79 | 193.63 |
| | turbo speed HP | [rpm] | [redacted] | [redacted] | 91541 | 81909 | 91541 |
| | turbo speed LP | [rpm] | ~~84276~~ | 70911 | 67136 | 65201 | 67136 |
| | Pmax | [bar] | ~~228.84~~ | 176.31 | 155.61 | 140.15 | 155.61 |
| 6 | EGR | [%] | 62.465 | ~~62.274~~ | 59.051 | 51.237 | 59.051 |
| ESC; Rosi | AF | [-] | 20.265 | ~~22.497~~ | 20.283 | 20.283 | 20.283 |
| 707.4 Nm | FS | [%] | 45.74 | ~~41.27~~ | 43.20 | 37.48 | 43.20 |
| 1350 rpm | BSFC | [g/kWh] | 371.25 | ~~235.06~~ | 214.38 | 211.62 | 214.38 |
| | BMEP | [bar] | 6.8814 | ~~6.8986~~ | 6.8999 | 6.888 | 6.8999 |
| | Power | [kW] | 99.878 | ~~100.13~~ | 100.15 | 99.975 | 100.15 |
| | turbo speed HP | [rpm] | [redacted] | ~~89278~~ | 72168 | 62298 | 72168 |
| | turbo speed LP | [rpm] | 60480 | ~~45230~~ | 35719 | 30904 | 35719 |
| | Pmax | [bar] | 190.68 | ~~124.85~~ | 94.514 | 79.177 | 94.514 |
| **Averages** | Avrg. FS | [%] | 23.33 | 33.44 | 33.98 | 30.54 | 34.65 |
| | Avrg.IBSFC | [g/kWh] | 331.46 | 231.68 | 210.00 | 203.96 | 211.56 |
| | Avrg.AF | [%] | 20.27 | 20.60 | 20.30 | 20.30 | 20.30 |
| | max EGR | [%] | 62.47 | 62.27 | 59.14 | 57.05 | 59.18 |
| | min EGR | [%] | 5.60 | 35.47 | 39.66 | 32.06 | 39.66 |
| | min AF | [%] | 20.20 | 20.28 | 20.27 | 20.28 | 20.27 |
| | max. Cyl.Pmax | [bar] | 303.99 | 245.21 | 224.47 | 207.62 | 224.47 |
| | Turbo maps exceeded? | [case] | 0 | 0 | 0 | 0 | 0 |

* turbo over-speed

- *VTG rack-sweep & EGR-cooler temperature*

| | Simulation number | | 21 | 21 | 21 | 21 | selection 21 |
|---|---|---|---|---|---|---|---|
| | HP-compr | [-] | 1 | 1 | 1 | 1 | 1 |
| | HP-turb | [-] | 1 | 1 | 1 | 1 | 1 |
| | LP-compr | [-] | 1 | 1 | 1 | 1 | 1 |
| | LP-turb | [-] | 1 | 1 | 1 | 1 | 1 |
| | 2e interc | [K] | 300 | 300 | 300 | 300 | 300 |
| | Tegr | [K] | 333 | 333 | 333 | 333 | 333 |
| | inj.press | [bar] | 1500 | 1500 | 1500 | 1500 | 1500 |
| | VTG-rack pos. | [-] | 0.25 | 0.50 | 0.75 | 1.00 | 0.5 & 0.75 |
| **Case** | EGR-cooler size | [-] | 5 | 5 | 5 | 5 | 5 |
| 1 | EGR | [%] | 38.836 | ~~40.836~~ | 40.062 | 31.721 | 40.062 |
| ESC; A100 | AF | [-] | 20.289 | ~~21.007~~ | 20.309 | 20.282 | 20.309 |
| 2557 Nm | FS | [%] | 28.40 | 28.89 | 29.27 | 23.21 | 29.27 |
| 1212 rpm | BSFC | [g/kWh] | 255.26 | ~~199.98~~ | 189.7 | 188.29 | 189.7 |
| | BMEP | [bar] | 24.141 | ~~24.859~~ | 24.918 | 24.944 | 24.918 |
| | Power | [kW] | 314.57 | ~~323.93~~ | 324.7 | 325.04 | 324.7 |
| | turbo speed HP | [rpm] | [redacted] | [redacted] | 91257 | 82998 | 91257 |
| | turbo speed LP | [rpm] | 77753 | ~~68406~~ | 68104 | 65883 | 68104 |
| | Pmax | [bar] | 301.43 | ~~243.65~~ | 218.43 | 194.92 | 218.43 |
| 2 | EGR | [%] | 20.967 | 38.47 | 39.532 | 35.666 | 39.532 |
| ESC; B100 | AF | [-] | 20.285 | 20.303 | 20.312 | 20.313 | 20.312 |
| 2442 Nm | FS | [%] | 15.34 | 28.12 | 28.88 | 26.06 | 28.88 |
| 1530 rpm | BSFC | [g/kWh] | 288.91 | 218.03 | 200.84 | 195.83 | 200.84 |
| | BMEP | [bar] | 23.18 | 23.539 | 23.739 | 23.835 | 23.739 |
| | Power | [kW] | 381.3 | 387.21 | 390.49 | 392.08 | 390.49 |
| | turbo speed HP | [rpm] | [redacted] | [redacted] | 98424 | 87958 | 98424 |
| | turbo speed LP | [rpm] | 87292 | 79128 | 76918 | 75452 | 76918 |
| | Pmax | [bar] | 275.63 | 240.37 | 223.25 | 206.43 | 223.25 |
| 3 | EGR | [%] | 5.537 | 33.78 | 39.239 | 37.745 | 39.239 |
| ESC; C100 | AF | [-] | 20.281 | 20.282 | 20.316 | 20.317 | 20.316 |
| 2015 Nm | FS | [%] | 4.05 | 24.71 | 28.66 | 27.57 | 28.66 |
| 1830 rpm | BSFC | [g/kWh] | 333.23 | 241.04 | 217.55 | 209.26 | 217.55 |
| | BMEP | [bar] | 19.574 | 19.577 | 19.615 | 19.678 | 19.615 |
| | Power | [kW] | 385.11 | 385.17 | 385.92 | 387.17 | 385.92 |
| | turbo speed HP | [rpm] | [redacted] | [redacted] | 101920 | 90612 | 101920 |
| | turbo speed LP | [rpm] | 94952 | 81980 | 80540 | 78614 | 80540 |
| | Pmax | [bar] | 241.93 | 211.63 | 202 | 189.73 | 202 |
| 4 | EGR | [%] | 55.428 | 58.849 | 49.298 | 41.646 | 58.849 |
| ESC; A50 | AF | [-] | 20.279 | 20.303 | 20.281 | 20.282 | 20.303 |
| 1279 Nm | FS | [%] | 40.56 | 43.01 | 36.07 | 30.47 | 43.01 |
| 1212 rpm | BSFC | [g/kWh] | 285.61 | 208.64 | 196.27 | 194.75 | 208.64 |
| | BMEP | [bar] | 12.264 | 12.458 | 12.474 | 12.475 | 12.458 |
| | Power | [kW] | 159.81 | 162.33 | 162.54 | 162.55 | 162.33 |
| | turbo speed HP | [rpm] | [redacted] | 95241 | 78927 | 68323 | 95241 |
| | turbo speed LP | [rpm] | 64311 | 53984 | 48960 | 46592 | 53984 |
| | Pmax | [bar] | 222.44 | 167.53 | 128.74 | 113.15 | 167.53 |
| 5 | EGR | [%] | 39.004 | 57.438 | 58.774 | 56.485 | 58.774 |
| ESC; C50 | AF | [-] | 20.154 | 20.268 | 20.264 | 20.285 | 20.264 |
| 1008 Nm | FS | [%] | 28.71 | 42.05 | 43.04 | 41.32 | 43.04 |
| 1830 rpm | BSFC | [g/kWh] | 457.7 | 280.21 | 238.87 | 225.52 | 238.87 |
| | BMEP | [bar] | 9.7714 | 9.5872 | 9.839 | 9.9045 | 9.839 |
| | Power | [kW] | 192.25 | 188.63 | 193.58 | 194.87 | 193.58 |
| | turbo speed HP | [rpm] | [redacted] | [redacted] | 91668 | 81979 | 91668 |
| | turbo speed LP | [rpm] | 84394 | 70991 | 67363 | 65206 | 67363 |
| | Pmax | [bar] | 227.29 | 174.19 | 154.19 | 138.5 | 154.19 |
| 6 | EGR | [%] | 62.166 | ~~65.234~~ | 58.705 | 50.938 | 58.705 |
| ESC; Rosi | AF | [-] | 20.267 | ~~20.272~~ | 20.283 | 20.283 | 20.283 |
| 707.4 Nm | FS | [%] | 45.51 | 47.75 | 42.95 | 37.26 | 42.95 |
| 1350 rpm | BSFC | [g/kWh] | 369.58 | ~~234.93~~ | 214.52 | 211.73 | 214.52 |
| | BMEP | [bar] | 6.8743 | ~~6.8966~~ | 6.8999 | 6.8894 | 6.8999 |
| | Power | [kW] | 99.775 | ~~100.1~~ | 100.15 | 99.995 | 100.15 |
| | turbo speed HP | [rpm] | [redacted] | ~~89198~~ | 72182 | 62272 | 72182 |
| | turbo speed LP | [rpm] | 60515 | ~~44210~~ | 35650 | 30825 | 35650 |
| | Pmax | [bar] | 188.33 | ~~121.93~~ | 93.841 | 78.817 | 93.841 |
| **Averages** | Avrg. FS | [%] | 23.14 | 33.50 | 33.55 | 30.20 | 34.40 |
| | Avrg.lBSFC | [g/kWh] | 330.75 | 231.48 | 210.08 | 204.02 | 211.60 |
| | Avrg.AF | [%] | 20.26 | 20.41 | 20.30 | 20.30 | 20.30 |
| | max EGR | [%] | 62.17 | 65.23 | 58.77 | 56.49 | 58.85 |
| | min EGR | [%] | 5.54 | 33.78 | 39.24 | 31.72 | 39.24 |
| | min AF | [%] | 20.15 | 20.27 | 20.26 | 20.28 | 20.26 |
| | max. Cyl.Pmax | [bar] | 301.43 | 243.65 | 223.25 | 206.43 | 223.25 |
| | Turbo maps exceeded? | [case] | 6 | 5 | 0 | 0 | 0 |

* turbo over-speed

- VTG rack-sweep & EGR-cooler temperature

| | Simulation number | | 22 | 23 |
|---|---|---|---|---|
| | HP-compr | [-] | 1 | 1 |
| | HP-turb | [-] | 1 | 1 |
| | LP-compr | [-] | 1 | 1 |
| | LP-turb | [-] | 1 | 1 |
| | 2e interc | [K] | 300 | 300 |
| | Tegr | [K] | 373 | 398 |
| | inj.press | [bar] | 1500 | 1500 |
| | VTG-rack pos. | [-] | 0.75 & 0.5@A50 | 0.75 & 0.5@A50 |
| **Case** | EGR-cooler size | [-] | 5 | 5 |
| 1 | EGR | [%] | 38.702 | 37.617 |
| ESC; A100 | AF | [-] | 20.305 | 20.302 |
| 2557 Nm | FS | [%] | 28.28 | 27.49 |
| 1212 rpm | BSFC | [g/kWh] | 190.15 | 190.44 |
| | BMEP | [bar] | 24.912 | 24.911 |
| | Power | [kW] | 324.62 | 324.61 |
| | turbo speed HP | [rpm] | 91566 | 91819 |
| | turbo speed LP | [rpm] | 68913 | 69028 |
| | Pmax | [bar] | 215.15 | 212.47 |
| 2 | EGR | [%] | 37.469 | 36.289 |
| ESC; B100 | AF | [-] | 20.317 | 20.317 |
| 2442 Nm | FS | [%] | 27.37 | 26.51 |
| 1530 rpm | BSFC | [g/kWh] | 201.12 | 201.31 |
| | BMEP | [bar] | 23.738 | 23.736 |
| | Power | [kW] | 390.47 | 390.45 |
| | turbo speed HP | [rpm] | 99123 | 99504 |
| | turbo speed LP | [rpm] | 77331 | 77597 |
| | Pmax | [bar] | 217.69 | 214.5 |
| 3 | EGR | [%] | 37.637 | 36.64 |
| ESC; C100 | AF | [-] | 20.319 | 20.319 |
| 2015 Nm | FS | [%] | 27.49 | 26.76 |
| 1830 rpm | BSFC | [g/kWh] | 217.39 | 217.38 |
| | BMEP | [bar] | 19.619 | 19.62 |
| | Power | [kW] | 385.99 | 386.02 |
| | turbo speed HP | [rpm] | 102360 | 102640 |
| | turbo speed LP | [rpm] | 81169 | 81498 |
| | Pmax | [bar] | 198.04 | 195.4 |
| 4 | EGR | [%] | 57.561 | 56.737 |
| ESC; A50 | AF | [-] | 20.291 | 20.276 |
| 1279 Nm | FS | [%] | 42.09 | 41.52 |
| 1212 rpm | BSFC | [g/kWh] | 208.87 | 208.97 |
| | BMEP | [bar] | 12.447 | 12.444 |
| | Power | [kW] | 162.19 | 162.16 |
| | turbo speed HP | [rpm] | 95577 | 95744 |
| | turbo speed LP | [rpm] | 54581 | 54841 |
| | Pmax | [bar] | 163.07 | 160.28 |
| 5 | EGR | [%] | 56.982 | 55.82 |
| ESC; C50 | AF | [-] | 20.257 | 20.275 |
| 1008 Nm | FS | [%] | 41.74 | 40.85 |
| 1830 rpm | BSFC | [g/kWh] | 237.91 | 237.21 |
| | BMEP | [bar] | 9.8428 | 9.8464 |
| | Power | [kW] | 193.66 | 193.73 |
| | turbo speed HP | [rpm] | 92114 | 92341 |
| | turbo speed LP | [rpm] | 67594 | 67627 |
| | Pmax | [bar] | 147.65 | 144.03 |
| 6 | EGR | [%] | 57.551 | 57.005 |
| ESC; Rosi | AF | [-] | 20.283 | 20.283 |
| 707.4 Nm | FS | [%] | 42.10 | 41.70 |
| 1350 rpm | BSFC | [g/kWh] | 215.02 | 215.26 |
| | BMEP | [bar] | 6.9 | 6.9004 |
| | Power | [kW] | 100.15 | 100.16 |
| | turbo speed HP | [rpm] | 72226 | 72242 |
| | turbo speed LP | [rpm] | 35594 | 35757 |
| | Pmax | [bar] | 91.742 | 90.793 |
| **Averages** | Avrg. FS | [%] | 33.21 | 32.45 |
| | Avrg.IBSFC | [g/kWh] | 211.59 | 211.58 |
| | Avrg.AF | [%] | 20.30 | 20.30 |
| | max EGR | [%] | 57.56 | 57.01 |
| | min EGR | [%] | 37.47 | 36.29 |
| | min AF | [%] | 20.26 | 20.28 |
| | max. Cyl.Pmax | [bar] | 217.69 | 214.50 |
| | Turbo maps exceeded? | [case] | 0 | 0 |

## Abbreviations and Symbols

| *Abbreviation* | *Description* |
|---|---|
| EGR | Exhaust Gas Recirculation level |
| HD | Heavy Duty |
| CI | Compression Ignition |
| DI | Direct Injection |
| AF or A/F | Air Fuel ratio |
| FS | Fraction Stoichiometric |
| PM | Particle Matters |
| NOx | Nitrogen-Oxides |
| HC | Hydro Carbons |
| HCCI | Homogeneous Charge Compression Ignition |
| PCCI | Premixed Charge Compression Ignition |
| BSFC | Brake Specific Fuel Consumption |
| BMEP | Brake Mean Effective Pressure |
| IMEP | Indicated Mean Effective Pressure |
| ROHR | Rate Of Heat Release |
| CA | Crank Angle |
| TDC | Top Dead Centre (piston) |
| BTDC | Before Top Dead Centre |
| ATDC | After Top Dead Centre |
| SOI | Start Of Injection |
| SOC | Start Of Combustion |
| VVA | Variable Valve Actuation |
| IVO | Inlet Valve Opening |
| IVC | Inlet Valve Closure |
| EVO | Exhaust Valve Opening |
| EVC | Exhaust Valve Closure |
| VTG | Variable Turbo Geometry |
| DPF | Diesel Particulate Filter |
| EPA | (US) Environmental Protection Agency |
| FTP | (US EPA) Federal Test Procedure |
| ESC | European Steady-State Cycle |
| HP | High-pressure |
| LP | Low-pressure |
| HTO | Heat Conduction Object (GT-power) |
| Ign | Ignored (GT-power) |
| Def | Default (GT-power) |

| *Symbol* | *Description* | Unit |
|---|---|---|
| t | time | [s] |
| [NO] | Nitrogen-Oxide concentration | - |
| $[O_2]$ | Oxygen concentration | - |
| $[N_2]$ | Nitrogen concentration | - |
| T | Temperature | [K] |
| m | mass | [kg] |
| f | mass flow rate | [Kg/s] |
| cp | Specific heat value | [KJ/ kgK] |
| Hu or H | Lower combustion value | [MJ/kg] |
| Tb | Temperature after combustion (burnt) | [K] |
| Tu | Temperature before combustion (unburnt) | [K] |
| P | Pressure | [Pa] |
| M | engine torque | [Nm] |
| N | engine speed | [rpm] |
| Vs | Total engine displacement (swept volume) | [L] – [$dm^3$] |
| VN | Theoretic engine volume flow | [$m^3$/s] |
| Q | Specific heat per mass | [kJ/kg] |
| Y | mass fraction | - |
| R | universal gas constant | [J/kgK] |
| j = (1,..,6) | case number | |
| PO | engine power | [kW] |
| WF | Weight Factor | - |
| $\rho$ | Density | [kg/$m^3$] |
| $\lambda$ | Lambda | - |
| $\Phi$ | equivalence ratio | - |
| W | work | [kW] |
| $\tau$ | pressure ratio | [Pa/Pa] |
| $\kappa$ | eatio of specific heat | - |
| $\eta$ | efficiency | [%] |
| dp | pressure difference | [Pa] |
| x | burn rate | [1/deg] |
| $\upsilon$ | crank angle | [deg] |
| $\alpha$ | Wiebe parameter, premix fraction | - |
| $\beta$ | Wiebe parameter, tail fraction | - |
| a | adjustable Wiebe parameter | - |
| $\omega$ | adjustable Wiebe parameter | - |
| $\upsilon_0$ | start of combustion | [deg] |
| $\Delta \upsilon$ | combustion duration | [deg] |
| $\theta$ | angle | [deg] |
| EGR | Exhaust Gas Recirculation level | [%] |
| AF | Air Fuel ratio | [kg/kg] |
| FS | Fraction Stoichiometric level | - or [%] |
| NOx | NOx emission level | [g/kWh] |
| BSFC | Brake Specific Fuel Consumption level | [g/kWh] |
| BMEP | Brake Mean Effective Pressure level | [bar] |
| IMEP | Indicated Mean Effective Pressure level | [Bar] |
| ROHR | Rate Of Heat Release | [1/deg] |
| CA | Crank Angle | [deg] |
| BTDC | Number of degrees CA Before TDC | [deg] CA |
| ATDC | Number of degrees CA After TDC | [deg] CA |
| SOI | Start Of Combustion in relation to TDC | [deg] CA |

| SOC | Start Of Combustion in relation to TDC | [deg] CA |
|---|---|---|
| *GT-power parameters;* | | |
| RPM | set Engine Speed | [rpm] |
| LOAD | set Engine Load | [Nm] |
| DESIRED_AF | set desired Air-Fuel ratio | - |
| FS | set desired Fraction Stoichiometric | [%] |
| MINIMUM_AF | set minimum Air-Fuel ratio (smoke limiter) | - |
| SOI | set Start Of Injection related to TDC | [deg] CA |
| TNIK | set Intercooler wall temperature | [K] |
| TWALL | set initial exhaust manifold + EGR line wall temperature | [K] |
| INITIAL-SPEED | set initial High-pressure- or single turbocharger speed | [rpm] |
| INITIAL-SPEED-LP | set initial Low-pressure turbocharger speed | [rpm] |
| M_INJ | set initial mass of fuel injected | [mg/cycle] |
| IN_RACK | set initial VTG turbo rack position | - |
| SCALE-HP-COMPR | set mass multiplier for High-pressure compressor | - |
| SCALE-HP-TURB | set mass multiplier for High-pressure turbine | - |
| SCALE-LP-COMPR | set mass multiplier for Low-pressure compressor | - |
| SCALE-LP-TURB | set mass multiplier for Low-pressure turbine | - |
| TEGR | set EGR-cooler wall temperature | [K] |

| *Subscript* | *Description* |
|---|---|
| e | equilibrium situation |
| ch & charge | charge |
| f | fuel |
| compressor | compressor |
| turbine | turbine |
| entrance | entrance (compressor) |
| eng | engine |
| man | inlet manifold |
| air | air |
| egr | EGR-gas |
| exh | exhaust-gas |
| ambient | ambient |
| boost | turbo boost |
| stoich | stoichiometric condition/ value |
| real | actual condition/ value |
| new | value at most recent cycle update |
| old | value at previous cycle update |
| max | maximum value |
| b | total combustion process (complete burn) |
| i = 1,2,3 | part of combustion process |
| 1 | premix part of combustion process |
| 2 | main part of combustion process |
| 3 | tail part of combustion process |
| vol | volumetric |
| T | thermal |
| comb | combustion |

## Samenvatting


Dit rapport maakt deel uit van een afstudeerproject binnen de faculteit werktuigbouwkunde aan de Technische Universiteit Eindhoven (TU/e). De opdracht is gedefinieerd door TNO automotive te Delft.

Dit rapport beschrijft een concept studie naar de toepassing van hoge EGR (Uitlaatgas recirculatie) niveaus op Heavy Duty (HD) diesel motoren met zelfontsteking. EGR is een effectieve methode om NOx emissies van diesel motoren te verminderen door een verlaging van de locale vlamtemperatuur tijdens diesel verbranding. Op basis van eerder onderzoeken bij TNO is vastgesteld dat hoge EGR niveaus tussen de 30 en 70%, ook op vollast (hoog EGR), nodig zijn om toekomstige (2010) emissie eisen voor HD diesel motoren te kunnen halen.

Het doel van dit rapport is het bepalen en analyseren van de benodigde aanpassingen aan het motor ontwerp om hoge EGR niveaus te kunnen behalen. Tevens is het de bedoeling na te gaan welke motorconfiguratie het meest geschikt is om hoge EGR niveaus te bereiken met toekomstige productie motoren. De US EPA emissie standaard voor 2010 geeft de laagst toegestane NOx norm aan en wordt daarom in dit rapport gebruikt als streefwaarde voor de NOx emissie van de motor en het hieruit volgende vereiste EGR niveau.

Een literatuurstudie is verricht naar de effecten van EGR op diesel verbranding en naar de benodigde aanpassingen in motorontwerp. Hieruit volgt dat 'additionele EGR' nodig is en de lucht/brandstof verhouding ongewijzigd dient te blijven. Het gevolg hiervan is dat de cilinder vulling verhoogd moet worden in vergelijking met motoren zonder EGR. Uit berekeningen blijkt dat hoge vuldrukken nodig zijn tussen de 5 en 7 bar, voor een 12.9 liter motor en een maximum BMEP van 25 bar. Om dit te kunnen bereiken zou 2-traps turbooplading uitkomst kunnen bieden. Om the voorkomen dat het volumetrisch rendement van de motor achteruit gaat en om smoring van de inlaat the voorkomen is koeling van het EGR-gas nodig. Hiervoor kan gebruikt gemaakt worden van speciale EGR-koelers.

EGR-gas kan via verschillende routes aan de motor worden toegediend. Als gevolg hiervan zijn er verschillende motorconfiguratie mogelijk. Op basis van literatuurstudie is geconcludeerd dat de volgende twee motorconfiguraties het best geschikt zijn om hoge EGR niveaus te bereiken:

- Externe route intern gedreven hoge druk EGR (afgekort: hoge druk EGR)
- Externe route intern gedreven lage druk EGR (afgekort: lage druk EGR)

Simulaties zijn verricht voor zowel de hoge druk als de lage druk variant met GT-power. GT-power is een simulatie pakket op basis van een dimensionale gasdynamica. De simulatiemodellen zijn gebaseerd op de 12.9 liter DAF MX motor met een vermogensafstelling van 390kW. De streefwaarden tijdens de simulaties zijn een algemeen gemiddelde van 50% EGR bij een minimale lucht brandstof verhouding van 20.3. De nadruk ligt tijdens deze simulaties bij het maximaliseren van EGR. Er zijn geen voorspellingen gedaan over uitstoot van deeltje (PM) en NOx omdat het simulatiepakket hiervoor onnauwkeurig voorspellingen geeft.

Uit modeloptimalisatie blijkt dat voor zowel de hoge druk als de lage druk configuratie 2-traps oplading nodig is met gebruikmaking van een standaard VTG turbo en een vaste geometrie turbo. Tussen koeling na iedere compressor trap is eveneens nodig. Voor de hoge druk configuratie zijn turbochargers nodig met een 30% lagere massa stroom bij een gelijke druk verhouding. De EGR koeler moet een factor 2 en 5 groter worden uitgelegd dan de standaard DAF EGR koeler voor respectievelijk de hoge- en lage- druk configuratie. De gemiddelde EGR koeler wand temperatuur moet laag zijn, ordegrote 313-323K. Voor de lage druk configuratie is een extra uitlaatgas tegendruk klep nodig en langer pijpwerk in vergelijking met de hoge druk configuratie.

Op basis van de simulatie resultaten is geconcludeerd dat de lage druk EGR configuratie iets beter in staat is hoge EGR niveaus te bereiken met een minimum lucht/brandstof verhouding vergeleken met de hoge druk EGR configuratie. Desondanks is met de hoge druk configuratie een lager BSFC te realiseren en een significant kleinere inbouw ruimte van de motor benodigd. Daarom is de eindconclusie dat een hoge druk EGR configuratie beter geschikt om hoog-EGR te bereiken.

## Dankwoord

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