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Ranking of four dual loop EGR modes

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Abstract
Modern Diesel engines contain sophisticated control systems to keep their environmental impact low for sustainable road transport. One of these systems is the dual loop exhaust gas recirculation, which can change combustion properties in several ways. This article presents an engine dyno measurement based on analyzing the duel loop EGR with a medium-duty Diesel engine. Intake throttles and exhaust brakes for the highest freedom in the air-fuel ratio setting support the EGR systems. The EGR modes are compared from fuel consumption, NO_x emission, and exhaust gas opacity in steady and transient operations. There are expected results, for instance, in the HP EGR's faster reaction time or the LP EGR's boost pressure holding property. Unexpected results are also presented. Contrary to theory, LP EGR generally provides a fuel consumption advantage in many operation points due to its higher boost pressure. Typically, HP EGR can provide lower fuel consumption at higher engine power. In the emission results, LP EGR is favourable both in NO_x emission and exhaust gas opacity, whereas the intake throttle-supported HP EGR usually shows the highest emission. Besides, with high EGR rates at lower engine loads, LP EGR can realize low-temperature combustion, where both emissions decrease. No difference can be detected between the LP EGR's support valves' results. HP EGR was faster in the transient operation, with about half the reaction time. The results can be utilized for dual-loop EGR layout and control design.
Keywords
Diesel engines, dual loop exhaust gas recirculation, exhaust brakes, nitrogen-oxide emission, exhaust gas opacity.

1. Introduction

Today's Diesel engines are one of the most complex engineering products ever made (Guzella and Onder, 2010). Besides the automated driving functions, internal combustion engine control is probably the most sophisticated system in road vehicles. They can work reliably when several science disciplines' achievements and experiments are designed into their complex systems. However, their operation principle cannot be changed: imperfect combustion processes produce undesirable, harmful emissions. The changes for sustainable road transport impose new requirements, and new technologies are applied. Due to the electrification of road transport, research in the field of internal combustion engines is in decline. The future of these engines depends on emission regulation and green or synthetic fuels *(Rajkumar and Thangaraja, 2019, Virt and Arnold, 2022*), but even with these, engines cannot reach zero harmful emissions (*Zöldy et al., 2022*). The change will be more difficult with commercial vehicles. Probably, Diesel engines will remain in this segment for a longer term (Sinay et al., 2018).

Before the alternative fuel research, Diesel engines were developed with more precise control systems (*Wang, 2008; Bárdos et al., 2016; Castillo Buenaventura et al., 2015*). There can be several control aims: low fuel consumption, low emission, better performance, etc. One of them is to minimize nitrogen oxide (NO_x) and particulate matter (PM) emissions altogether. Their formation during combustion is counterproductive (*Abián et al., 2018*). However, optimums can be found with the control of the combustion (*Divekar et al., 2015; Luján et al., 2015*). Furthermore, cooperation with the exhaust after-treatment system can also determine the rate aims of forming these harmful emissions (*Mao, 2015*).

Exhaust gas recirculation (EGR) is a commonly used solution to reduce NO_x emissions. EGR can be realized with the valvetrain (internal EGR) and an additional pipe system between the exhaust and the intake side (external EGR). In a turbocharged engine, the latter can be realized in two ways: on the low and high-pressure sides of the turbocharger (Grondin et al., 2009). Based on this, low-pressure exhaust gas recirculation systems (LP EGR) and high-pressure exhaust gas recirculation systems (HP EGR) can operate in a given engine. Their common name is the dual-loop EGR system. Several articles discuss these systems' properties (*Millo et al., 2012; Reifarth, 2014*). A summary presents them in Table 1:

Table 1. Advantages and disadvantages of HP and LP EGR *(Nyerges and Németh, 2014)*

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HP EGR	LP EGR
fast response, favourable in transient cycles	slower response, higher boost pressure because all exhaust gas runs through the turbine, favourable in stationary cycles
the time for gas mixing is shorter	longer time for mixing, but the condensed water can damage the compressor
it increases fuel consumption, but in high power demand operations, it can provide lower BSFC	it increases the BSFC less than the HP EGR because of the higher boost pressure
Using both systems on the same engine has the advantage of optimization	

In external EGR systems, the EGR valve controls the mass flow rate of the recirculated exhaust gas. The pressure difference between the exhaust and intake sides determines this configuration's maximum EGR mass flow rate. Diesel engines sometimes require more EGR than this limit in low-load operation. Support valves can be utilized to increase further EGR mass flow rate: an intake throttle (THR) can reduce fresh air intake, or an exhaust brake (EB) can reduce the exhaust gases that leave the air-path system (*Nyerges and Zöldy, 2020a*). These valves give intake oxygen rate control a significant degree of freedom (*Bárdos and Németh, 2017*). Even rich combustion can be set too. These valves can realize four EGR modes: HP or LP EGR with intake throttle or exhaust brake support.

Traditionally, engine performance and emission can be influenced in two ways: changing the combustion or changing the charge exchange processes. The former can be achieved by combustion timing or with different injection strategies (Vass and Zöldy, 2020). The latter method also changes the combustion indirectly by the pressure, temperature, or composition of the intake charge (Nyerges and Zöldy, 2020b). The EGR plays a role in the second method. In summary, the effects of EGR on engine operation are composed of the following.

- The oxygen concentration of the intake charge and the air-fuel ratio of the combustion decreases.
- The higher ratio of burnt gases slows down the combustion processes.
- The higher intake charge temperature decreases volumetric efficiency.
- With HP EGR, the turbine mass flow rate decreases, i.e., the engine-turbocharger cooperation changes.
- With HP EGR, the engine's pumping loss may decrease, i.e., the volumetric efficiency can be improved.
- Due to these, brake efficiency and fuel consumption change generally increase. Higher fuel consumption causes higher exhaust gas enthalpy, which may improve the engine-turbocharger cooperation.
- Consequently, the emission map changes (generally, NO_x emission decreases, and PM emission increases).
- Due to the slower combustion, the in-cylinder pressure characteristics are also changed, and the engine's run becomes smoother and more silent.

Several articles compare the HP and LP EGR systems (*Zamboni and Capobianco, 2012; Cornolti et al., 2013; Mao et al., 2016*), and some analyze the effect of high EGR rates (*Nyerges and Zöldy, 2020c*). However, no article exists where these four EGR modes are analyzed and compared. This paper aims to analyze the properties of the different EGR modes. The measurements were done using a medium-duty Diesel engine (MDD) dyno. The paper will compare the EGR modes from the following aspects: NO_x and soot emission, fuel consumption and efficiency, air path system's O_2 concentrations, and pressures, both in steady state and transient operation. The paper aims to offer advice on optimal air path system layout.

2. Measurement considerations

2.1. Engine dyno

Our research group has an engine dyno with an MDD engine. It has been used in several research projects in the last decades, primarily for engine modelling (*Nyerges and Zöldy, 2020a*), control (*Bárdos et al., 2016*), and injection and combustion analysis (*Virt et al., 2021*). The general parameters of the engine are shown in Table 2. The low speed and high torque range with relatively high boost pressure is important properties. The boost pressure control is conventional, wastegate type. Initially, it did not have any EGR systems. They have been designed and mounted onto the engine in the last few years.

Table 2. Engine parameters (*Bárdos and Németh, 2013*)

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Type	turbocharged Diesel, in-line, four-cylinder
Maximum power	125 kW (2500 1/min)
Maximum torque	600 Nm (1200-1600 1/min)
Displacement	3.9 l
Stroke/bore ratio	1.176
Compression ratio	17.3
Injection system	direct injection, common rail
Maximum boost pressure	2.5 bar

The air path system of the engine is depicted in Figure 1. It has two EGR systems and four support valves. The LP intake throttle is mounted upstream of the LP EGR mixer; therefore, it can increase the LP EGR rate. Similarly, the HP intake throttle can also increase the HP EGR rate. It is mounted between the compressor and the HP EGR mixer.

The placement of the exhaust brakes is slightly different. The short air path of the exhaust manifold is important for efficient turbocharger operation, i.e. the first exhaust brake (EB1) cannot be mounted upstream of the turbine. Instead, it supports the HP EGR system downstream of the turbine at low pressure. The second exhaust brake (EB2) is placed at the end of the air path system. All of the six valves are controlled electronically.

For comparison, several measurement opportunities are available. Besides the usual operation properties (speed, torque, fuel consumption), the O_2 and NO_x concentrations in the air-path system can be measured by Lambda and UniNOx sensors (*Nyerges and Zöldy, 2020c*), an opacimeter can measure the opacity of the exhaust gas, and the boost pressure and the intake fresh air mass flow rate are also measurable.

The engine operation, basically the injection strategy, is controlled by the engine's own ECU:

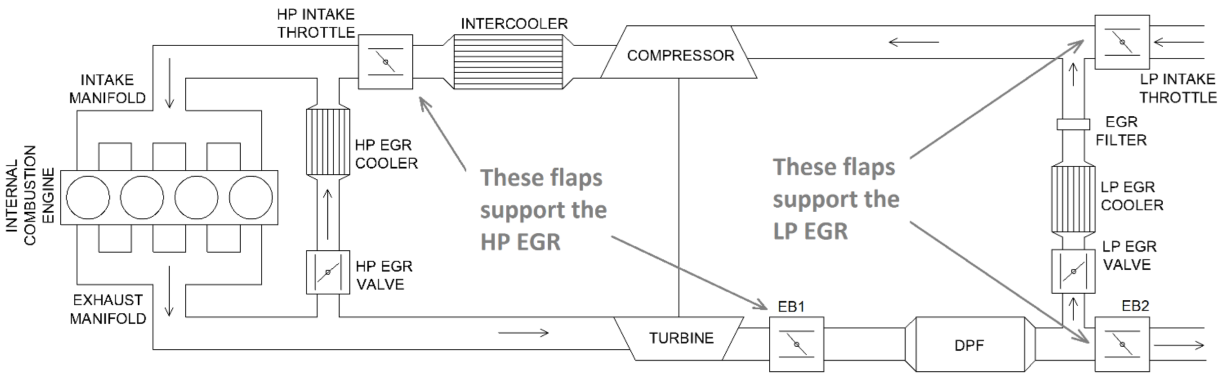

Figure 1. The air path system of the engine dyno (*Nyerges and Németh, 2014 p3.*)

2.2. Measured operation points and experiences

Conventional Diesel engines always run with a lean air-fuel ratio (AFR). Lower AFR results in progressively increasing PM emissions. Therefore, the PM emission classically determines the lowest AFR limit and, indirectly, the engine's maximum torque (*Huang et al., 2014*). As the load is reduced, the AFR also gets leaner. Since the EGR decreases the AFR, EGR can be applied only at lower loads. The steady-state operation point selection considered these for the EGR mode comparison. The selected operation points are shown in Figure 2:

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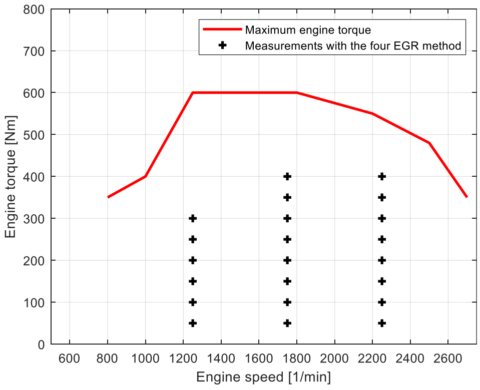

Figure 2. The measured and analyzed operation points

Three-speed levels are selected evenly from the engine's speed range. There are two main differences between these speed levels. Due to the engine's original emission standard, the injection strategy (by the ECU) on lower speeds does not have pre-injection: this causes high NOx and low PM emissions. Due to the significant pre-injection, the emission map changes to lower NO_x and higher PM emission at higher speeds. The other difference between the speed levels is the turbocharger operation. At 1250 1/min, the boost pressure is not significant. At 1750 1/min, the boost pressure changes show its effect from low loads to higher ones.

Moreover, finally, at 2250 1/min, the boost pressure is significant even at a low load. The torque demand steps are selected at 50 Nm intervals. Altogether 22 operation points were analyzed. On the highest applied torques, the AFR is low even without EGR.

It can be stated that the support valves can only be applied on very low loads. The comparison between the intake and exhaust support valves cannot be predicted. The exhaust brakes can provide better fuel consumption (*Nyerges and Zöldy, 2018*), but this research was based only on simulation results. The pressure drop through the HP EGR system is more significant (due to the turbocharging) than in the LP EGR systems. The LP EGR system's pressure levels are always close to ambient pressure. This is why LP EGR is more sensitive to the EGR valve position (*Zamboni et al., 2017*).

2.3. Measurement challenges

The measurements presented some unexpected challenges. To ensure the reproducibility of the measured characteristics, the four EGR modes were measured one after another on the same speed and torque level but always in changing order. The aim was to avoid the EGR modes' consistent effect on each other.

The EGR mode's measurement always started without EGR. Then in the first phase, the EGR valve (HP or LP) was opened step by step. When the EGR valve was fully opened, the selected support valve (intake throttle or exhaust brake) was closed step by step again in the second phase. The support valve closes until the combustion becomes unstable – typically in the rich AFR zone. Conventionally, the stoichiometric and rich AFR in road vehicles' Diesel engines is not utilized due to the high PM emission (*Zeng and Wang, 2014*). However, the present paper also aims to analyze the effects of the unusually high EGR rates where even alternative Diesel combustion can be realized. Typically an EGR mode measurement resulted in 10-15 steady-state operation points on the same speed and torque level.

It was important to consider the transient effects on the results. There were fast transient changes after the valve position changes: the pressures, mass flow rates and charge compositions. The decay of these transients typically requires 5–15 seconds.

Eliminating the slower transients from the results was the more difficult challenge. These are typically temperature changes. During the measurements, the temperature of the engine dyno's cooling system slightly changes, as well as the temperature of fresh air intake. As shown above, this effect is eliminated by order of the operation points and the EGR modes. Due to the EGR mass flow rate changes, the wall temperatures of the air path system also changed with slightly slow transients. These changes were slower than those mentioned above, 5–15 seconds, but their effect was about 5–20 ppm NO_x, which was finally neglected. In summary, the effects of the temperature changes were difficult to eliminate, but the extent of the effect is significantly slighter than the EGR modes' and changes' effects.

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The third property that can cause inaccuracy is the positioning of the valves. There is always some leakage with closed valves, and the realized valve position is not always the same as the position demand. The presentation of the results eliminates the effect of this issue: the different characteristics are always depicted in the function of the O_2 concentrations, the air-fuel equivalence ratio, or the EGR rate.

3. Measurement results

3.1. Experiences

The first challenge in creating the measurement strategy was how the different EGR methods could be compared. The aim is to select a parameter that can be the same in the different EGR mode operations. The following important parameters change due to EGR:

- Intake parameters: oxygen concentration, boost pressure, temperature.
- Exhaust parameters: oxygen concentration, turbine backpressure.
- In-cylinder: air-fuel ratio, combustion speed.
- The mass flow rate through the cylinders, EGR mass flow rates.

The selected parameter should characterize the EGR operation. It should provide an objective comparison of the different modes. Advantageous if the available engine sensors can measure it.

Usually, the first approach for the comparison is the EGR rate, which gives the ratio between the EGR's and the engine's mass flow rate. Due to the lean AFR operation, the exhaust gas of Diesel engines can contain a large amount of fresh air (i.e. oxygen). Therefore, the EGR rate does not reveal much important information about the engine's operation. Moreover, with the same EGR rate, all engine parameters can be different (*Park and Bae, 2014*).

EGR rate (x_{EGR}) can be estimated from the O_2 concentrations by (1), where AMB refers to the ambient conditions, IM refers to the intake manifold, and EM refers to the exhaust manifold (*Guzella and Onder, 2010*).

(1) $$x_{EGR} = \frac{x_{AMB} - x_{IM}}{x_{AMB} - x_{EM}}$$

Without EGR, the intake charge has the ambient oxygen concentration. With the increasing EGR rate, the intake O_2 concentration decreases. The lowest level depends on the engine's O_2 consumption. On the exhaust side, the change is similar to an "O_2 concentration offset". Without EGR, the exhaust side O_2 concentration depends on the engine load, but it is equal to the difference between the ambient O_2 concentration and the O_2 consumption. With the increasing EGR rate, the exhaust side O_2 concentration also decreases until the combustion process becomes unstable, typically with rich AFR.

The air-fuel equivalence ratio can be estimated from the measurements by (2) from the fresh air and fuel mass flow rates (σ_{air} and σ_{fuel}) or by the O_2 concentrations of the intake and exhaust manifold ($x_{O2,IM}$ and $x_{O2,EM}$). Due to accuracy issues, the mass flow rate-based equation was used (Jung et al., 2014). K_{L0} is the stoichiometric air-fuel ratio.

(2) $$\lambda = \frac{\sigma_{air}}{\sigma_{fuel} K_{L0}} \approx \frac{x_{O_2,IM}}{x_{O_2,IM} - x_{O_2,EM}}$$

In summary, the advantages of the different aspects:

- The intake O_2 concentration shows the EGR's operation.
- The exhaust O_2 concentration shows the EGR potential.
- Moreover, the AFR shows an important parameter of combustion properties.

And the disadvantages:

- None of them contains the effects of boost pressure changes.
- The intake O_2 concentration does not show the EGR potential.
- The exhaust O_2 concentration does not show the intake properties.
- Moreover, while the O_2 concentrations are directly measurable, the AFR is just estimated, i.e. it can be inaccurate.

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Finally, intake O_2 concentration was selected for the basis of the comparison. This choice has several arguments: O_2 concentration provides understandable information about EGR and is easily measurable and controllable. Since it does not contain the mass flow rate changes, it highlights the turbocharger operation advantages of LP EGR. Consequently, in this paper, the EGR modes are comparable when they produce the same intake oxygen concentration.

The comparison of the different EGR presentation diagrams is shown in Figure 3 and Figure 4. The analyzed effect is the fuel consumption change with the four EGR modes. The explained properties significantly appear: the different presentations show different rankings for the EGR modes. In function of the intake O_2 concentration LP EGR provides lower fuel consumption characteristics in these operation points. Meanwhile, its advantage disappears in the function of the exhaust O2 concentration, air-fuel equivalence ratio, or even ranking turns.

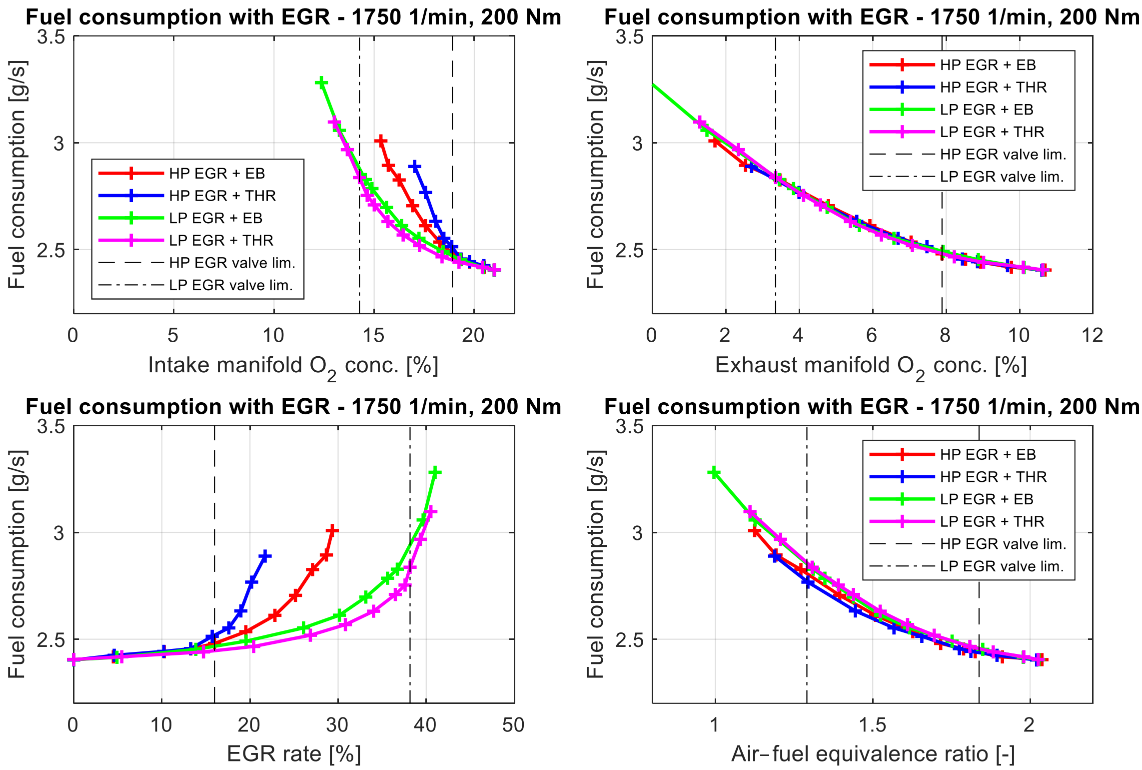

Figure 3. An example of the different EGR effect presentations at the same operation point (1750 1/min, 200 Nm)

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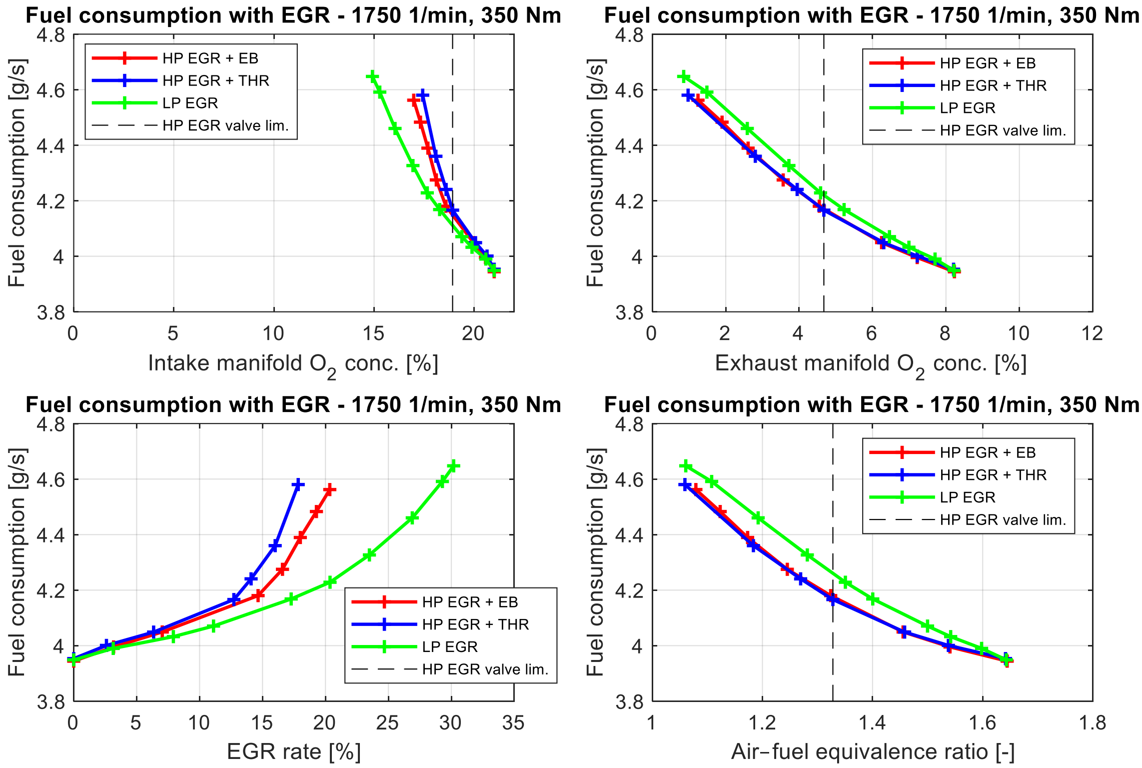

Figure 4. An example of the different EGR effect presentations at the same operation point (1750 1/min, 350 Nm)

As for the curve characteristics, the EGR modes' curves should start from the same point as in the case without EGR. If only the EGR valves set the mass flow rates, there should be only two curves (HP and LP EGR) because the support valves do not change in this phase. The figures in the results section will also show the limits of the EGR valves (without using the support valves). In this example, the LP EGR valve has a much higher EGR rate potential than the HP EGR – this property will be typical for all results. Finally, with high EGR rates, there can be four different curves due to the intake throttles and exhaust brakes – if there is any difference in the operation of these valves.

The presentation of all the measurement results would exceed the limits of this paper. The most representative ones are always selected for each type of test. Above average 200 Nm, the support valves are unnecessary to reach the stoichiometric AFR. Therefore, the lower load operation is preferred.

3.2. Steady state operation

In this section, the stationary measurement results will be presented. First, how the oxygen concentration changes in the air path system will be shown. These characteristics allow the EGR systems' operation to be understood more thoroughly. Next, the engine performance-related results will be analyzed: fuel consumption, engine efficiency, and boost pressure. Finally, the emission-related properties will be discussed: NO_x and exhaust gas opacity.

3.2.1. Oxygen concentrations

This section aims to analyze the operation of the EGR systems through the oxygen concentration changes in the air path system. Two operation points will be used: 1250 1/min with 100 Nm (small power, slight boost pressure), and 1750 1/min, 150 Nm (medium power, medium boost pressure). Figure 5 and Figure 6 depict the changes from several aspects.

As explained above, the intake side oxygen concentration shows the amount of the recirculated burnt gases. On the other "side", the exhaust gases' oxygen concentration indicates the combustion properties within the AFR. I.e., the exhaust side oxygen concentration is an EGR reserve: when it runs out, the combustion reaches the rich AFR zone.

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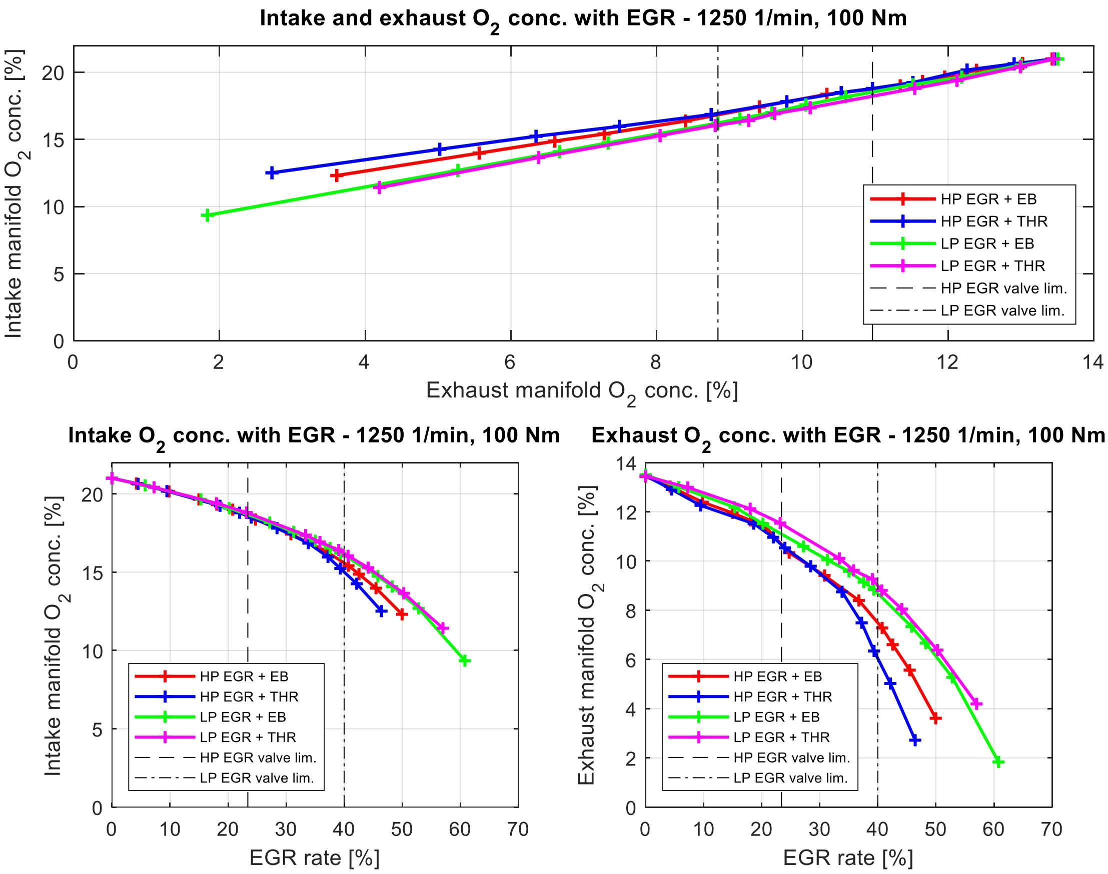

Figure 5. Intake and exhaust oxygen concentration changes (1250 1/min, 100 Nm)

Significant differences are shown between the HP and LP EGR; smaller differences are between the support valves if we compare the curves. The higher O_2 concentration change through the engine can signal either higher fuel consumption or lower mass flow rate through the engine (or generally both). Since the cylinders' mass flow rate significantly decreases with HP EGR, HP EGR's oxygen concentration change is usually higher.

The EGR rate-based diagrams also prove the engine mass flow rate changes: much higher EGR rates can be reached with LP EGR. The exhaust side O2 concentration is also higher, with a higher engine mass flow rate. Therefore, more exhaust gases can be mixed into the intake charge.

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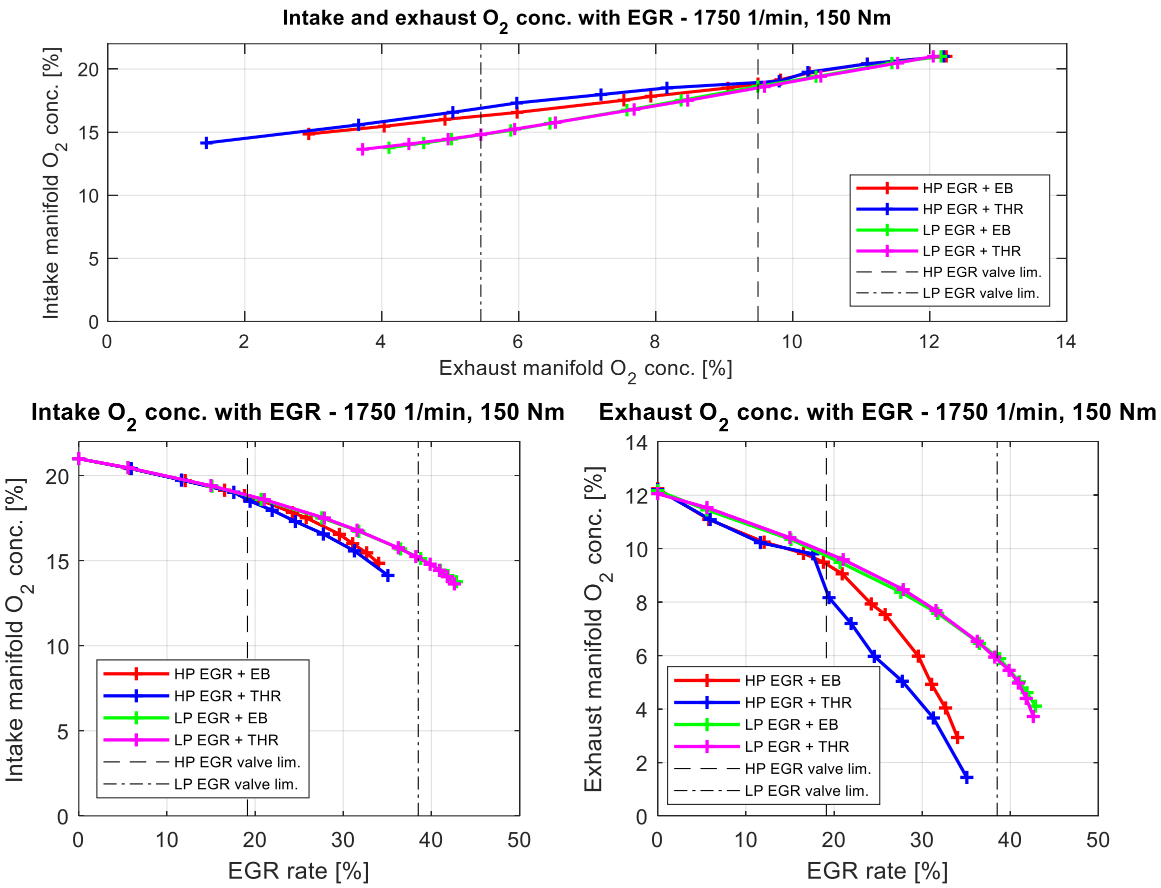

Figure 6. Intake and exhaust oxygen concentration changes (1750 1/min, 150 Nm)

Both in Figure 5 and Figure 6, there is a significant difference between the HP EGR's support valves: the exhaust brake provides higher O_2 concentrations. When the support valves take over the role, there is a break in the curves, which is more significant with higher boost pressures. This refers to engine-turbocharger cooperation changes.

With LP EGR, no definite difference can be discovered between the support valves, but in most cases, the intake throttle has slightly lower oxygen consumption. This difference is difficult to detect because the measurement inaccuracies have nearly the same scale. However, the intake throttle-supported HP EGR increases oxygen consumption in both cases.

3.2.2. Performance

For the evaluation of results on fuel consumption, three operation points were selected: a small power one (1250 1/min, 150 Nm), a medium power one (1750 1/min, 200 Nm), and a higher power one (2250 1/min, 200 Nm). In these operation points, all four support valves can be utilized. Below, two other operation points will also be presented where the LP EGR does not need support valves.

Figure 7 and Figure 8 show the tendencies at the medium power operation point. In Figure 6, the first diagram depicts the fundamental changes, while the second diagram's y-axis shows the critical interval for detecting the relative changes. In this operation point, fuel consumption increases with increasing EGR. The tendency is progressive. Higher EGR amounts cause higher fuel consumption increases. The overall fuel consumption increase is about 25%, but conventionally the highest EGR rates are not used in road vehicle engines. The fuel consumption differences between the different EGR modes are only significant when the HP EGR support valves take over the role. LP EGR provides lower fuel consumption (the difference can reach 7–10%), and the intake support valve has favourable results. In the case of HP EGR, the exhaust brake is the better choice, with around 3%. Brake-specific fuel consumption (BSFC) and brake efficiency are proportional to the fuel consumption, i.e., they provide similar information. These characteristics can be seen in Figure 8.

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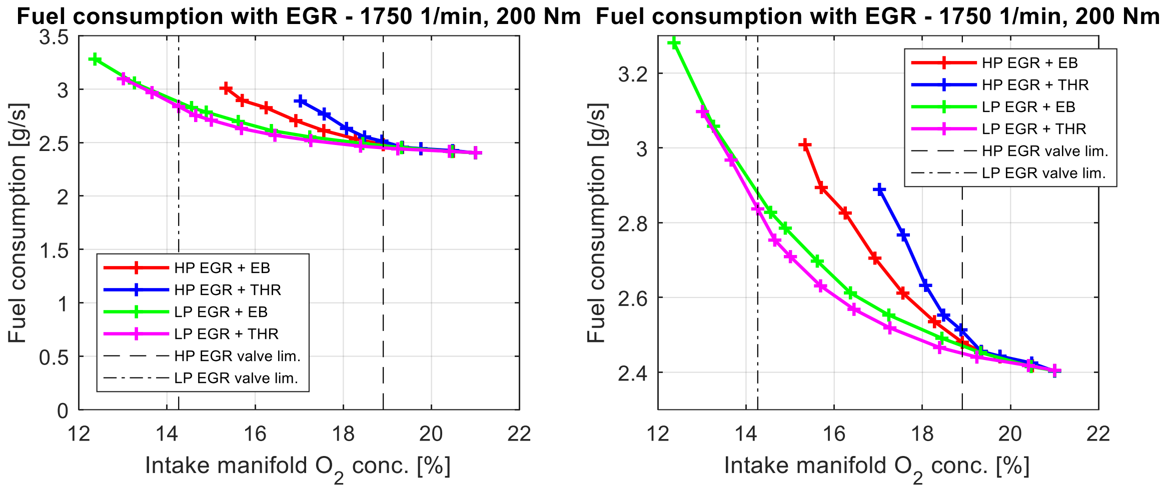

Figure 7. Fuel consumption changes with different EGR modes (1750 1/min, 200 Nm)

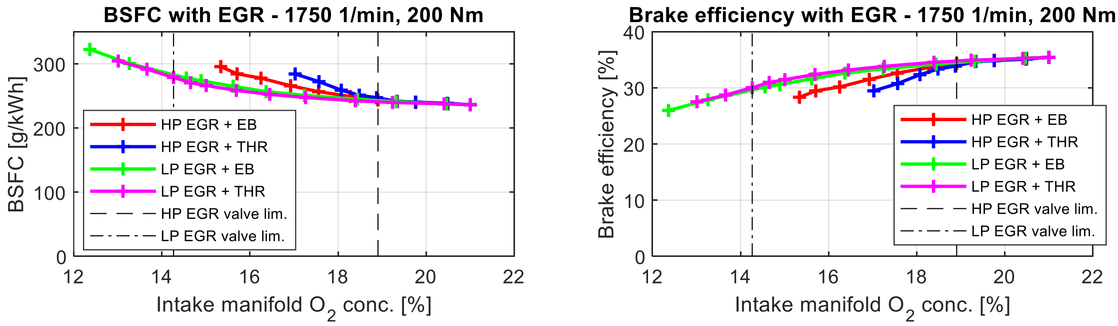

Figure 8. BSFC and brake efficiency changes with different EGR modes (1750 1/min, 200 Nm)

According to measurement experiences, the turbocharger operation highly influences the results. The ranking of the EGR modes can differ when the boost pressure is small (at the increasing phase) or when the boost pressure is high and the wastegate controls it. Considering the basic properties of dual loop EGR, the boost pressure change with LP EGR will be smaller because the turbine mass flow rate can remain similar—instead, the wastegate mass flow rate changes.

Figure 9 shows an example of low-power engine operation with small boost pressure. This operation's overall fuel consumption changes and the differences are smaller. At this operation point, the LP EGR provides lower fuel consumption. Within this example, the intake throttle support mode provides the lowest consumption. The differences are only significant with higher EGR amounts.

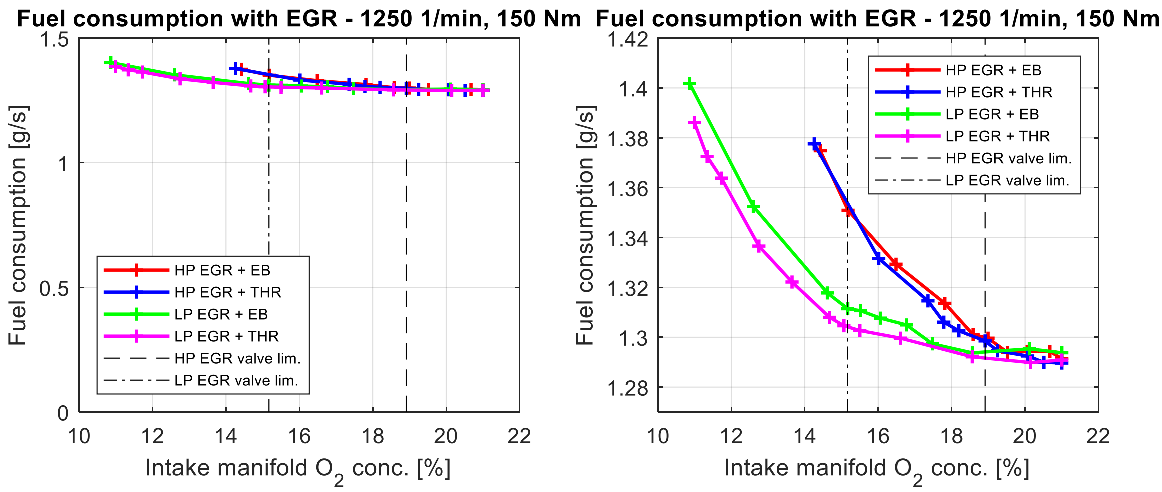

Figure 9. Fuel consumption changes with different EGR modes (1250 1/min, 150 Nm)

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Theoretically, the HP EGR may be able to decrease fuel consumption. If the turbine's backpressure is much higher than the boost pressure, the HP EGR can reduce this difference, i.e., the pumping loss of the engine. The operation with a higher power is represented in Figure 10. At this operation point, the theory is confirmed: the HP EGR system with lower EGR rates improves fuel consumption and increases with LP EGR. HP EGR's maximum fuel consumption advantage is about 2-3%. The ranking turns when the HP EGR support valves take over the role. After that, the LP EGR provides lower fuel consumption again.

The fuel consumption reduction is not notable with HP EGR, it is less than 1%, but it exists. The differences in the support valves are only significant with HP EGR: the exhaust brake lowers fuel consumption by around 7 %.

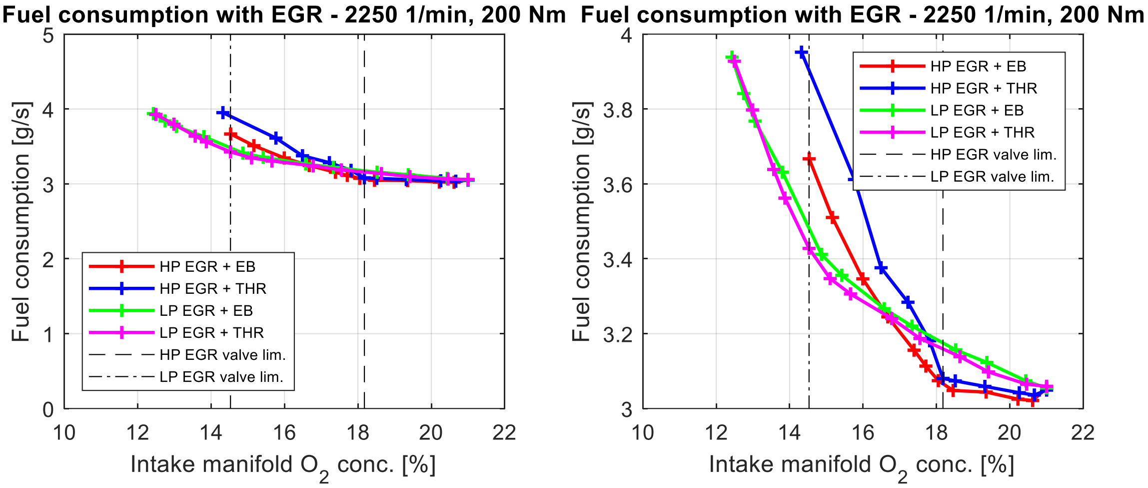

Figure 10. Fuel consumption changes with different EGR modes (2250 1/min, 200 Nm)

Figure 11 and Figure 12 show fuel consumption results with higher loads, where the LP EGR valve can reach the EGR limit, i.e., no support valve is necessary in these cases. The tendencies are similar. LP EGR provides lower fuel consumption at the lower engine speed (1250 1/min, 250 Nm). The difference is slight, barely 1-2% with low EGR amounts, but it can reach 6-7% with higher EGR amounts.

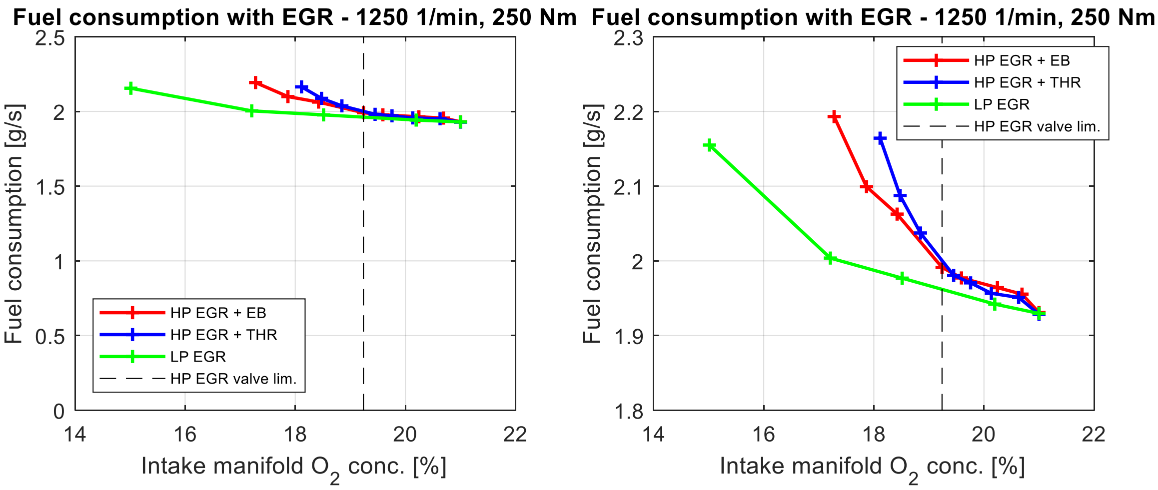

Figure 11. Fuel consumption changes with different EGR modes (1250 1/min, 250 Nm)

In Figure 12, with a higher engine speed (2250 1/min, 400Nm), the tendency turns again: HP EGR has more favourable fuel consumption in the full EGR range. In this case, the difference reaches 1-5%. Both in Figure 11 and Figure 12, the exhaust brake-supported HP EGR provides lower fuel consumption than the intake throttle. The difference is not significant.

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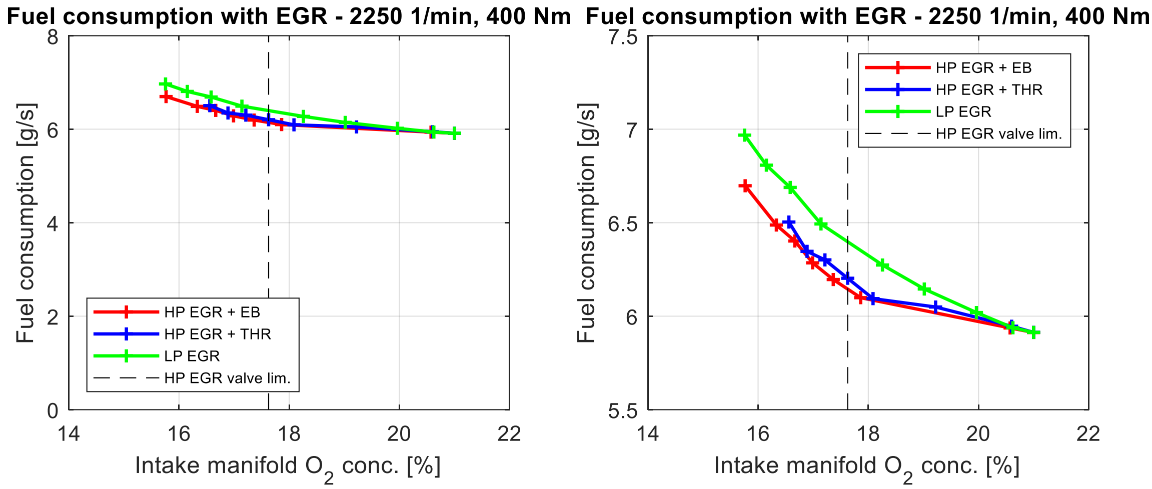

Figure 12. Fuel consumption changes with different EGR modes (2250 1/min, 400 Nm)

As discussed, the main reason for the HP and LP EGR operation differences is the turbocharger operation change. Figure 13 depicts the pressure changes in the intake side in a medium power operation point (1750 1/min, 150 Nm). This operation point has already been presented in Figure 6. Due to the high-pressure intake throttle, the characteristics are different downstream of the compressor and in the intake manifold. The left diagram shows the change in the turbocharger operation: while the HP EGR reduces the boost pressure, with LP EGR, it slightly increases. The boost pressure reduction is expected because the turbine's mass flow rate decreases with the opened HP EGR valve. The LP EGR's increasing boost pressure is caused by the deteriorating efficiency, i.e., the increasing fuel consumption results in higher turbine power.

The pressure in the intake manifold is slightly lower due to pressure losses through the intercooler. Moreover, the high-pressure throttle valve operates here. Therefore, the intake manifold pressure is significantly lower in this EGR mode, as seen in the right diagram in Figure 13. A detailed pumping loss analysis is not the aim of the present paper.

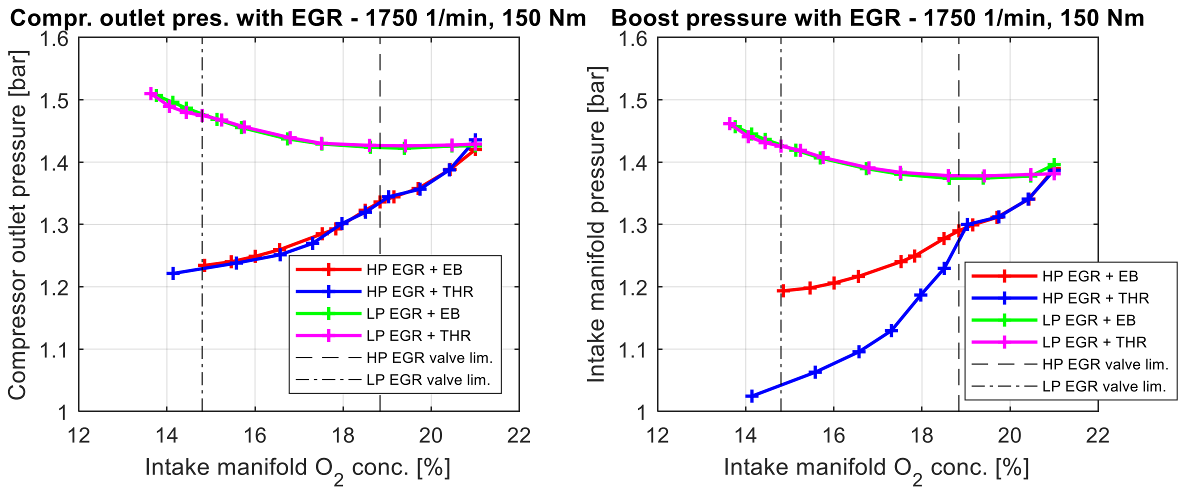

Figure 13. Intake side pressure changes with different EGR modes (1750 1/min, 150 Nm)

Another important aspect would be the detailed analysis of the cooler's effectiveness. The engine has three coolers in the air path system: the intercooler, the HP, and the LP coolers. The left diagram of Figure 14 shows a typical measurement result of the intake charge temperature characteristics. While with HP EGR, the intake temperature keeps its level, with LP EGR, it increases. The engine's LP EGR cooler and intercooler are undersized based on these. Since the compressor multiplies the LP EGR intake temperature, the LP EGR cooler's size would be expedient to redesign.

Figure 14 also presents the intake fresh air mass flow rate characteristics with the different EGR modes. As it is expected, with EGR, it slightly decreases. The comparison basis causes the differences: analyzed as a function of the EGR rate. There would not be such significant differences.

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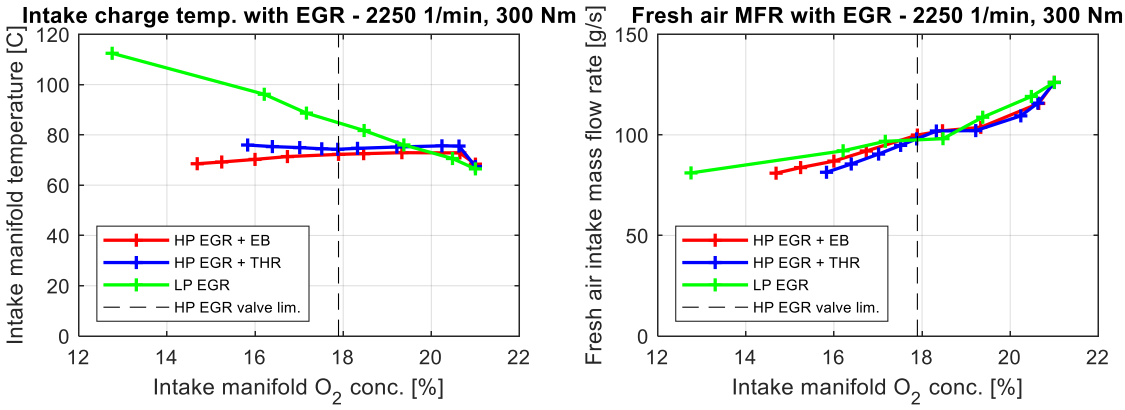

Figure 14. Intake charge temperature and intake fresh air mass flow rate changes with different EGR modes (2250 1/min, 300 Nm)

3.2.3. Emission

Since EGR aims to reduce emissions, the most exciting results are expected from this analysis. Diesel engines emit two crucial harmful materials: NO_x and PM. Their elimination is expensive and complicated because their formation is counterproductive (Grondin et al., 2009). This paper does not consider the CO and HC emissions. They can be handled more effectively with catalysts.

Although the method does not give accurate information on the PM emission amount, the diagrams estimate PM emission based on the opacity of the exhaust gas. An opacimeter was available in our research.

The AFR of the combustion mainly influences the proportions of the emission components. Although the injection strategy has a strong effect, it does not change in this research. The engine's original ECU controls injection. This engine produces high NO_x below 1400 1/min. Above this value, NO_x emission is much lower, as shown in the figures below.

The emission characteristics show several different behaviours according to the engine operation points. Therefore, this section will present several engine speeds, and load (and power) levels, generally in ascending order.

First, the low-speed, small-load operation results are presented. Figure 15 shows the measurement results at 1250 1/min and 50 Nm. The NO_x emission takes shape as expected: it strongly decreases with increasing EGR. According to Zamboni and Capobianco (2012), the decrease is exponential. LP EGR provides a lower amount with a medium EGR rate. The difference sometimes reaches 35%. Between the two HP EGR modes, the exhaust brake is significantly better. The difference is about 20-30%. Below and above, the difference is not so significant. There are not any observable differences between the LP EGR modes again. The shape of the opacity curve in Figure 15 offers exciting information. In the first phase, the opacities grow rapidly. The ranking is the opposite as it was with NO_x. The change occurs with a high EGR rate: the opacity with LP EGR decreases. Low-temperature combustion with low AFR is called low-temperature combustion (LTC) (Divekar et al., 2015). The opacity is lower with LP EGR all over the range.

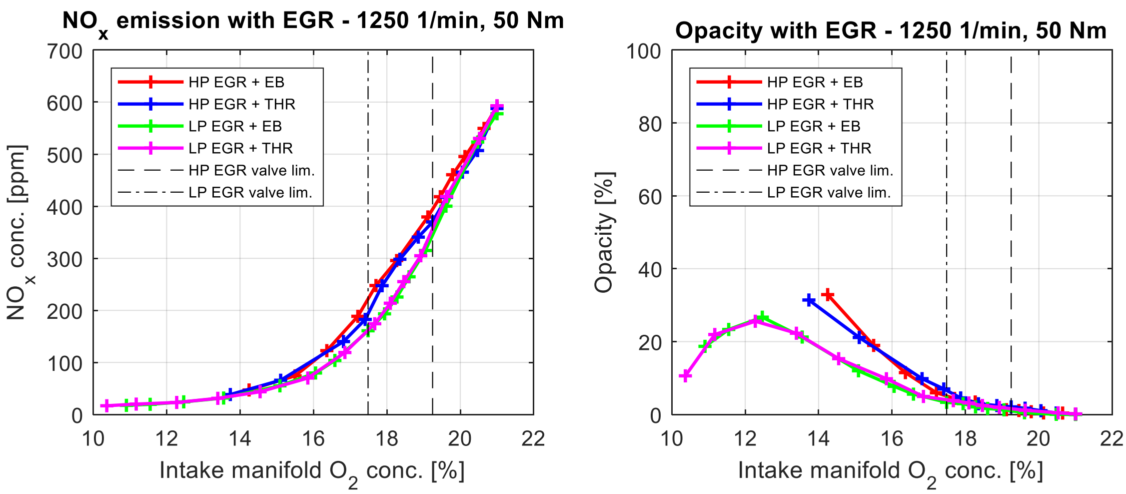

Figure 15. NO_x emission and opacity of the exhaust gas with EGR (1250 1/min, 50 Nm)

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The amounts of harmful emissions are strongly increased with higher load. Figure 16 shows the results at 1250 1/min again, but at 200 Nm. The ranking of the EGR modes is the same, i.e., LP EGR's advantage is low for low NO_x emission but much higher for low soot emission. The HP EGR with intake throttle is a slightly worse mode for NO_x emission again. There is only a tiny difference between the intake throttles and exhaust brakes for better exhaust gas opacity.

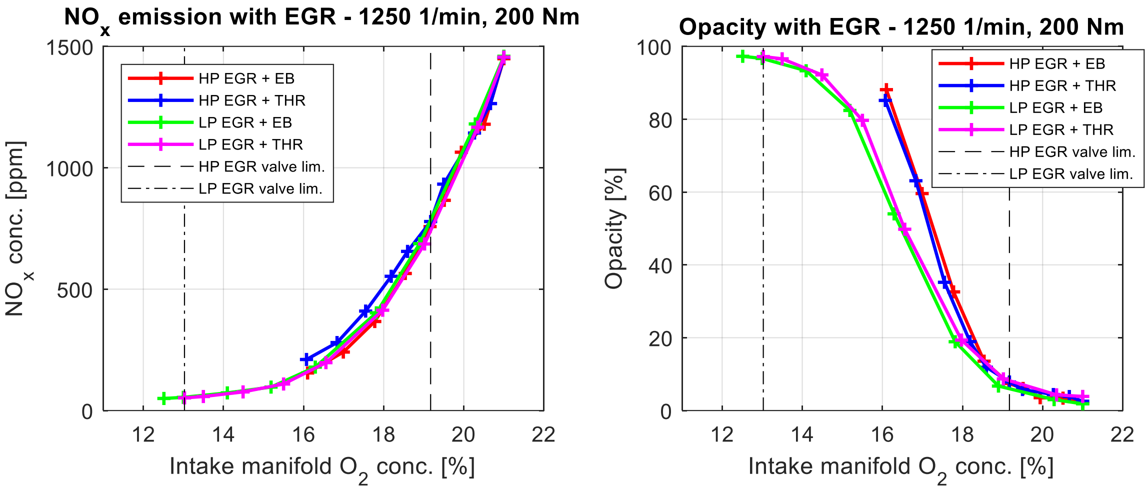

Figure 16. NO_x emission and opacity of the exhaust gas with EGR (1250 1/min, 200 Nm)

In Figure 17 and Figure 18, the emission results can be seen at 1750 1/min. The load level of Figure 17 is 150 Nm. The tendencies are similar to those in Figure 15: with a high LP EGR amount, the opacity decreases again. LTC can be realized at a higher speed than at 1250 1/min. A strong difference can also be seen between the two HP EGR modes: the opacity with intake throttle in the first phase is the best, and then it becomes the worst. From the NO_x aspect, the ranking is the same.

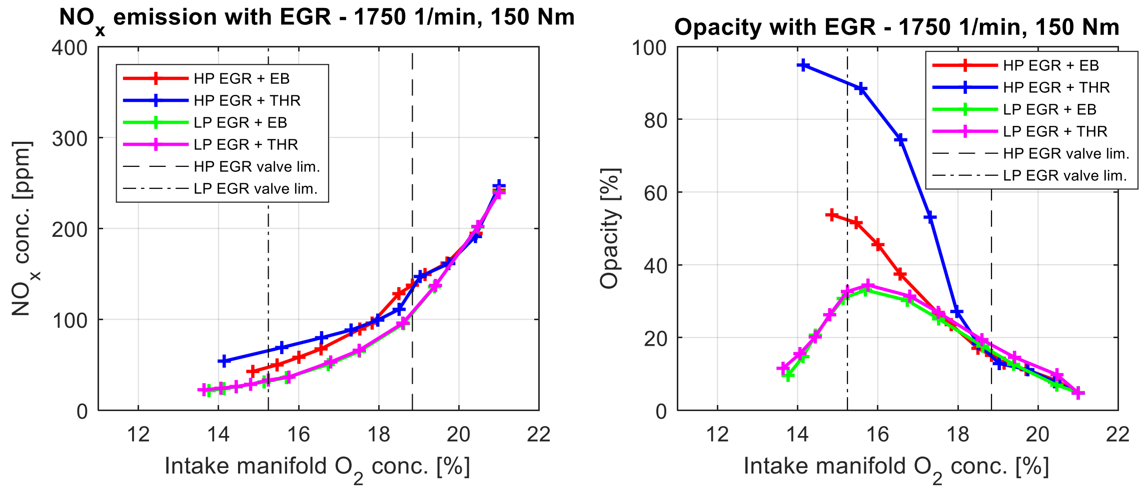

Figure 17. NO_x emission and opacity of the exhaust gas with EGR (1750 1/min, 150 Nm)

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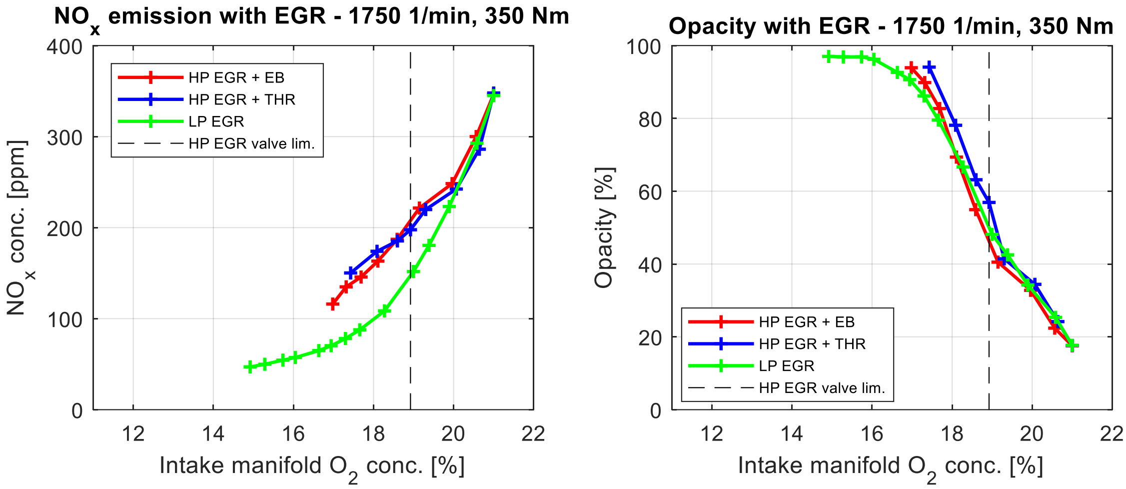

Figure 18. NO_x emission and opacity of the exhaust gas with EGR (1750 1/min, 350 Nm)

In Figure 18, the engine torque is high: 350 Nm. With this load, the LP EGR support valves are not necessary. As in the former results, the differences between the EGR modes are only significant when the HP EGR support valves actuate. After that, the NO_x emission is lower by around 25-30%. LP EGR's opacity advantage is disappeared. For supporting the HP EGR, the exhaust brake is a better choice again. However, the NO_x decrease potential is less effective than with lower loads. The opacity increases very fast for effective NO_x emission reduction.

The following four figures depict the measurement results at 2250 1/min with ascending load levels. The EGR modes show their four diverse faces at different load levels. The first level is 100 Nm in Figure 19. The NO_x characteristics are the same as above: HP EGR with exhaust brake has lower NO_x emission than with intake throttle, and overall, LP EGR provides lower NO_x emission after the HP EGR valve limit (maximum with around 20-30%). The shape of the opacity curves is similar to Figure 17. With small EGR amounts, there is not any difference, but with high LP EGR rates, LTC is realized again. With higher EGR rates, intake throttle-supported HP EGR gets worse again.

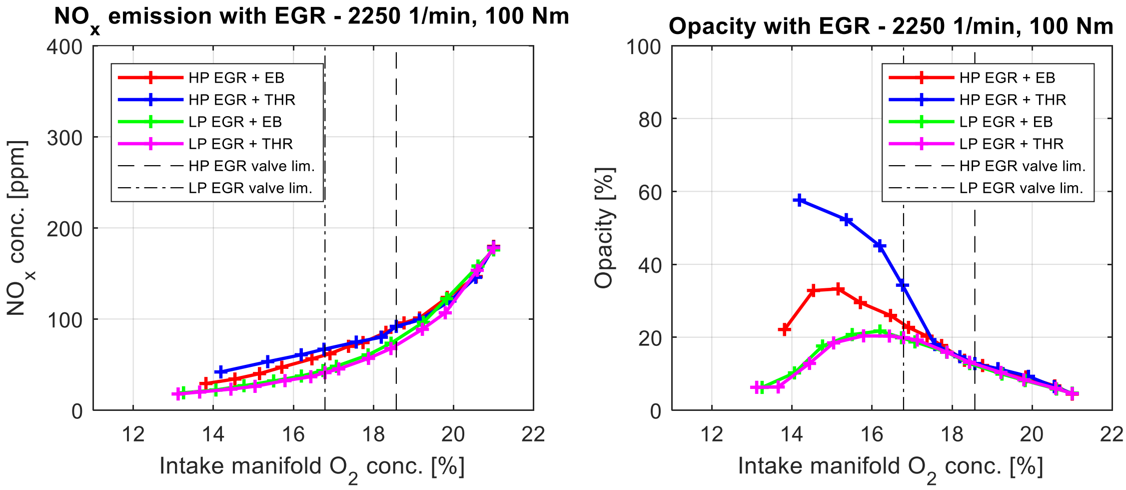

Figure 19. NO_x emission and opacity of the exhaust gas with EGR (2250 1/min, 100 Nm)

The next load level is 150 Nm in Figure 20. Small torque change does not occur in significant changes: the curve shapes and the rankings are nearly identical, and the overall amounts are higher. However, the extremity of the intake throttle supported HP EGR has disappeared. The HP EGR shows a slight NOx advantage (5-10%) with low EGR amounts.

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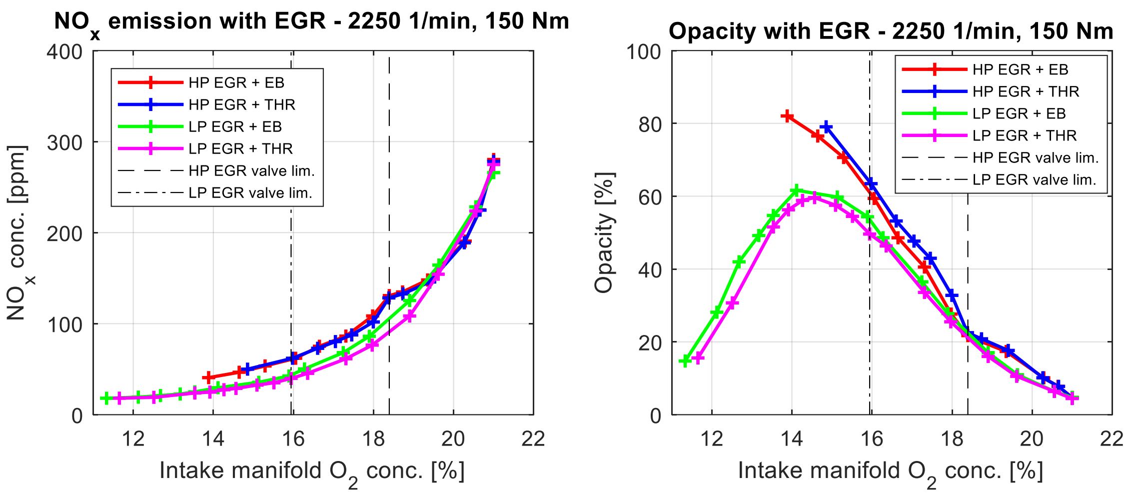

Figure 20. NO_x emission and opacity of the exhaust gas with EGR (2250 1/min, 150 Nm)

The diagrams in Figure 21 show the following torque levels step: 200 Nm. These measurement results have smaller but significant changes again: LTC is no longer feasible. Moreover, HP EGR's NO_x advantage with small EGR amounts slightly increases by around 7-15%. The LP EGR modes are the best, and HP EGR with intake throttle is the worst, as usual.

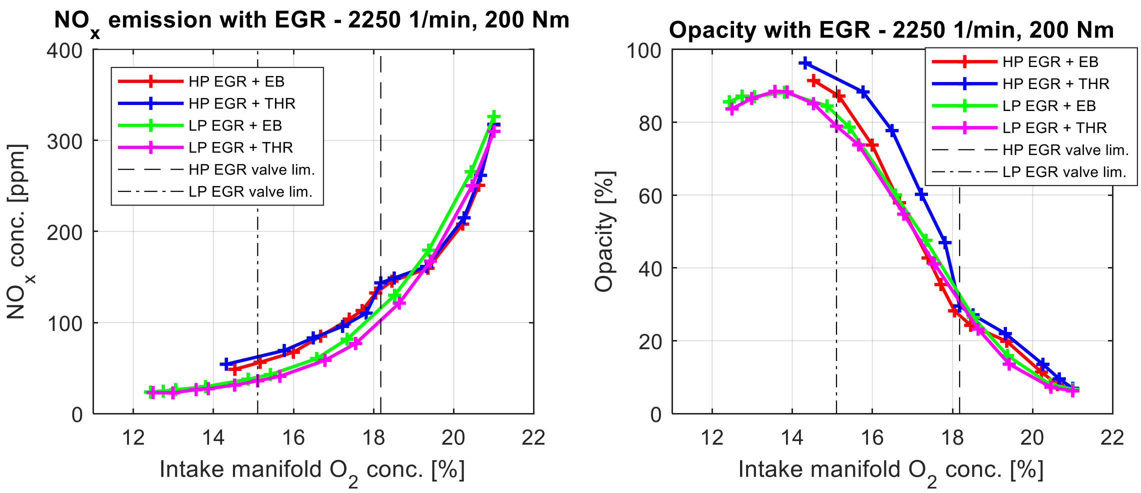

Figure 21. NO_x emission and opacity of the exhaust gas with EGR (2250 1/min, 200 Nm)

The last torque level is 350 Nm, this section's highest power measurement point. Figure 22 depicts the results. Overall the differences become lower. The NO_x advantage of HP EGR is disappeared with low EGR amounts. When the HP EGR support valves take over the role, the LP EGR's NO_x emission decrease is nearly half of HP EGR's. In contrast, the opacity ranking is the opposite. With any intake oxygen rate, the HP EGR has the advantage.

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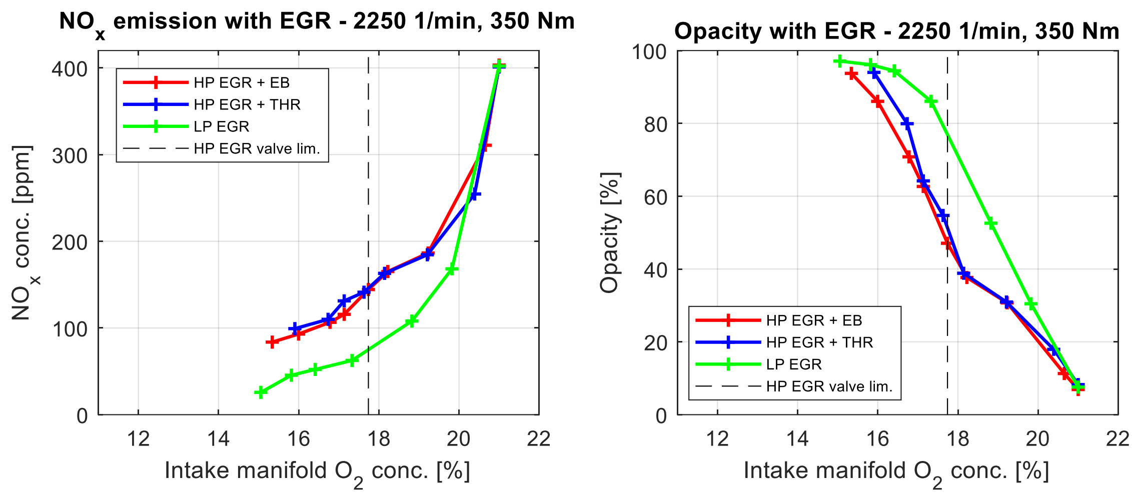

Figure 22. NO$_x$ emission and opacity of the exhaust gas with EGR (2250 1/min, 350 Nm)

3.3. Transient behaviour

As discussed in the introduction, the two EGR systems' behaviours in transient cycles differ. Due to the shorter loop, HP EGR is expected to be faster. In control-oriented dual loop EGR models, the reactions of EGR systems are usually modelled with time constants (*Nyerges and Zöldy, 2020a*). This section may help to understand the transient processes and to model their behaviour in a proper mathematical way.

Three operation points are selected for the transient behaviour representation: 1250 1/min with 200 Nm, 1750 1/min with 300 Nm, and 2250 1/min with 300 Nm. At the first phase of the transients, the engine runs in its steady state operation without EGR. The change starts with the EGR flap position change:

- HP EGR position change with exhaust brake support,
- LP EGR position change.

The target positions were calibrated manually. The aim was to achieve the same intake side O_2 concentration with both EGR systems (like in all the previous sections).

Figure 23 presents the oxygen concentration changes. At the end of the transient's decay, the intake side O_2 concentration is nearly the same – except at 2250 1/min at the late phase of the transient. Following the previous section's results, the exhaust side O2 concentrations are not converged to the same values.

Let us analyze the operation of EGR. Before the EGR valve opening, the intake pipes (volumes) are full of fresh air, while upstream of the EGR valve, the EGR pipes are full of exhaust gases. When the EGR valve opens, the exhaust gas mixing is started in the intake volumes. The mixed intake charge needs a short time to reach the cylinders. In this interval, the AFR in the cylinders does not change; consequently, neither does the exhaust side oxygen concentration. As soon as the recirculated exhaust gas arrives in the cylinders, the combustion process changes lower AFR, slower combustion, higher fuel consumption, and lower exhaust O_2 concentration. The new exhaust gas composition then results in a new intake mix in the later phase of the transient. Overall the reaction delay of the intake side O_2 change should be shorter than the exhaust side O_2 change.

According to the values in Figure 23, at 1250 1/min, the theory is proven. The intake side oxygen concentration change with HP EGR is nearly prompt, and the reaction time on the exhaust side is about 0.5s. The reaction delay of the LP EGR is 0.6 s on the intake side and nearly 1 s on the exhaust side. An important property of this operation point that its boost pressure is small. The total time of the transient decay is about 9 s.

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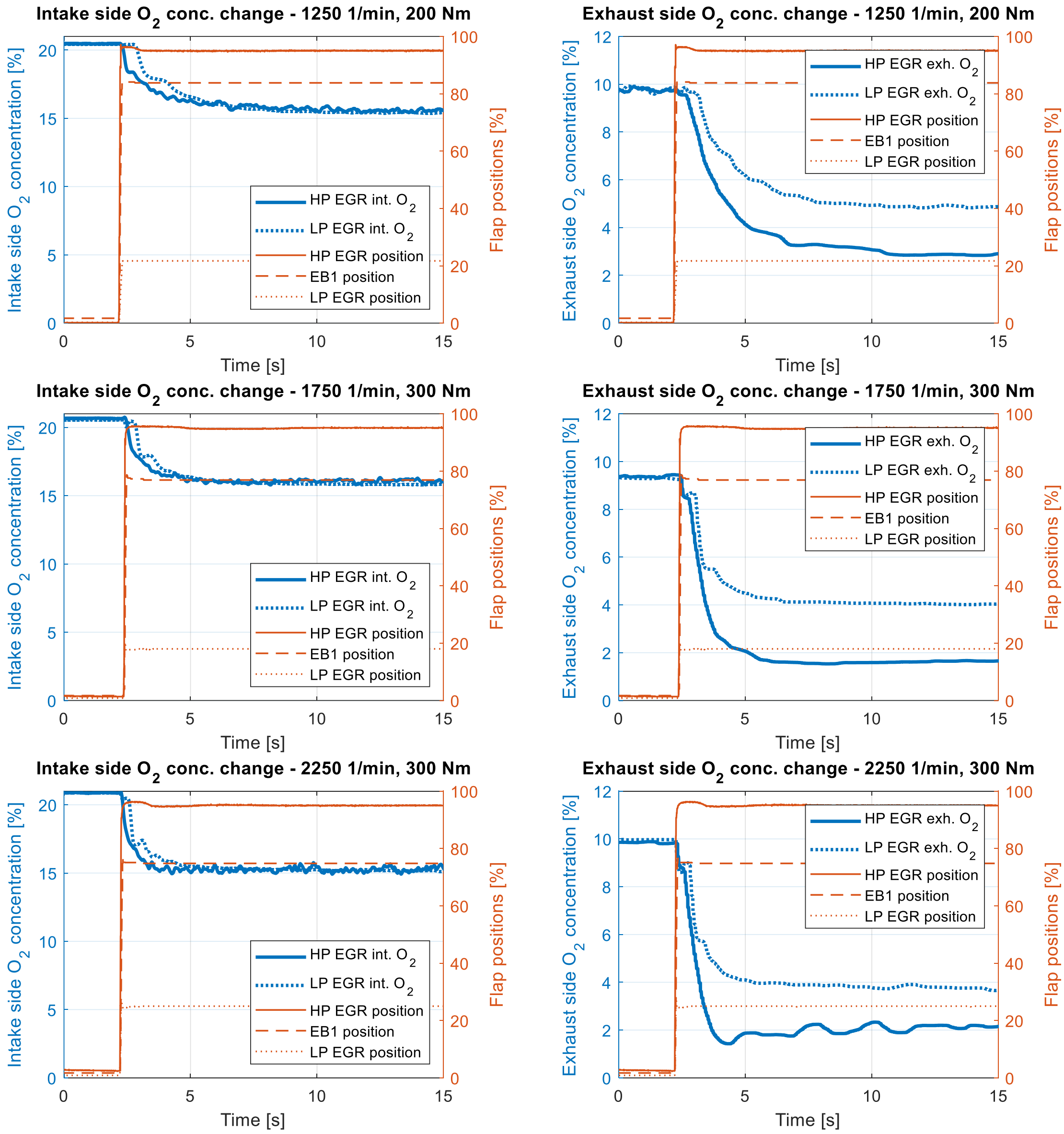

Figure 23. Intake and exhaust side oxygen concentrations in transient operation with HP and LP EGR

The reactions change at 1750 1/min and 2250 1/min due to the higher gas speeds and higher boost pressures. On the intake side, the HP EGR's change is prompt again, while the LP EGR's reaction delay decreases to 0.4 s and 0.3 s, respectively. The total time of the transient decay decreases to 5 s and 3 s, respectively. There is an oxygen concentration step on the exhaust side with both EGR systems, probably due to the turbocharger operation change. Consequently, the reaction delay is difficult to determine. However, the reaction delay difference is decreased between the two EGR systems.

Figure 24 depicts the NO_x emission and the opacity change of the exhaust gas during the transient cycle at the three-speed levels. The reaction time of the emission changes is slower than that of the exhaust gas O_2 concentration (HP EGR: 0.8 s, LP EGR: 1.1 s). The overall transient decay is about 8 s.

The effect of higher engine speeds is similar to the above. The difference between the two EGR systems is greatly decreased again. The reaction times and the overall transient decays decrease at a similar rate.

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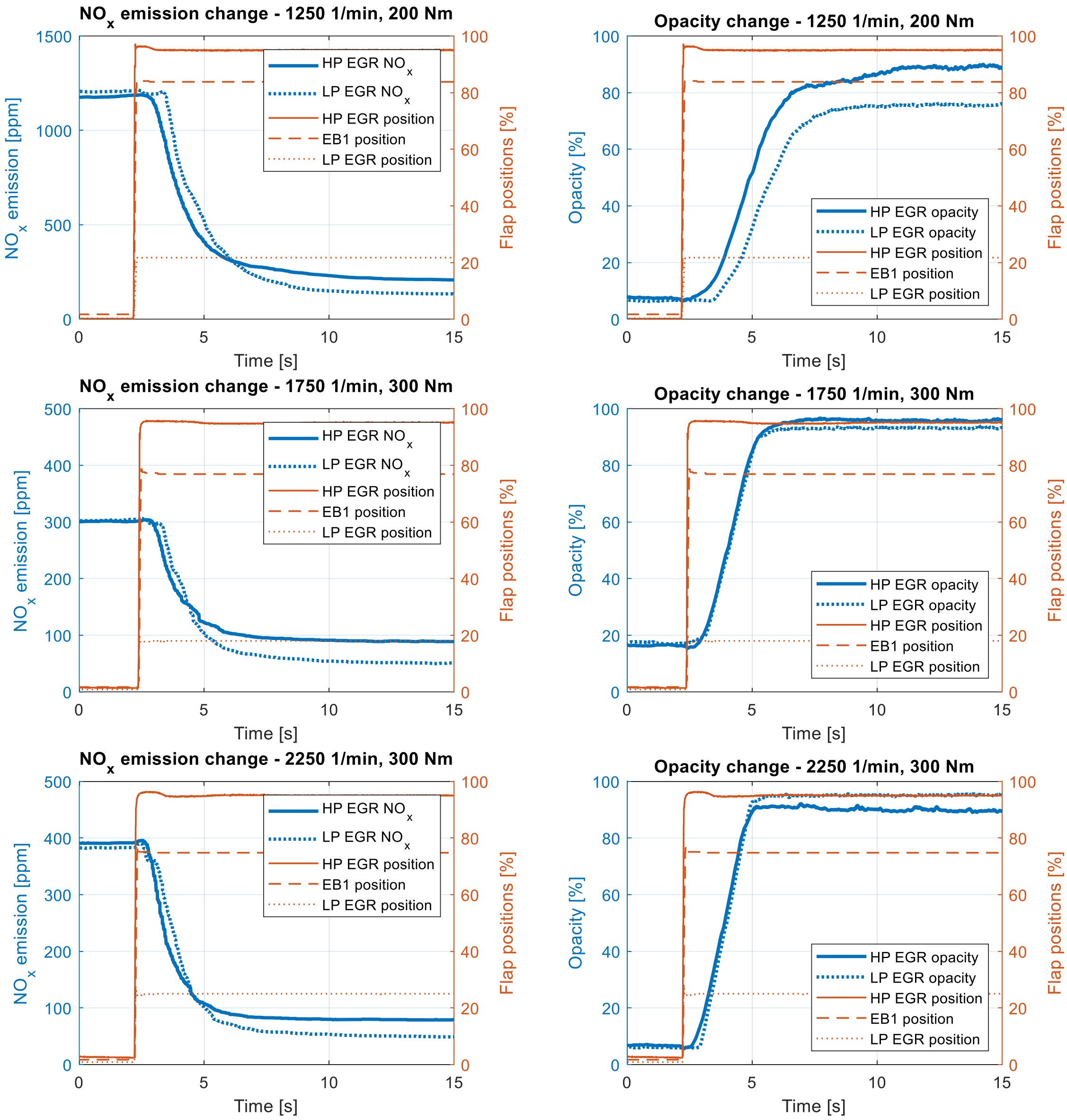

Figure 24. NO$_x$ emission and opacity in transient operation with HP and LP EGR

Finally, Figure 25 presents the boost pressure and fresh air consumption changes at 1250 1/min and 2250 1/min. The boost pressure decreases with HP EGR and slightly increases with LP EGR. While HP EGR simply falls to its convergence value, LP EGR's behaviour differs. At the first phase of the transient, the boost pressure slightly decreases, increasing the convergence value. The overall transient decays are not comparable between the two EGR systems due to their different behaviour. At 1250 1/min, they take about 3-6 s, while at 2250 1/min, they change to 2-3 s.

The fresh air consumption change has a similar duality. It decreases with both EGR systems, but with LP EGR, it has a significant overshot, which increases in function of the engine power.

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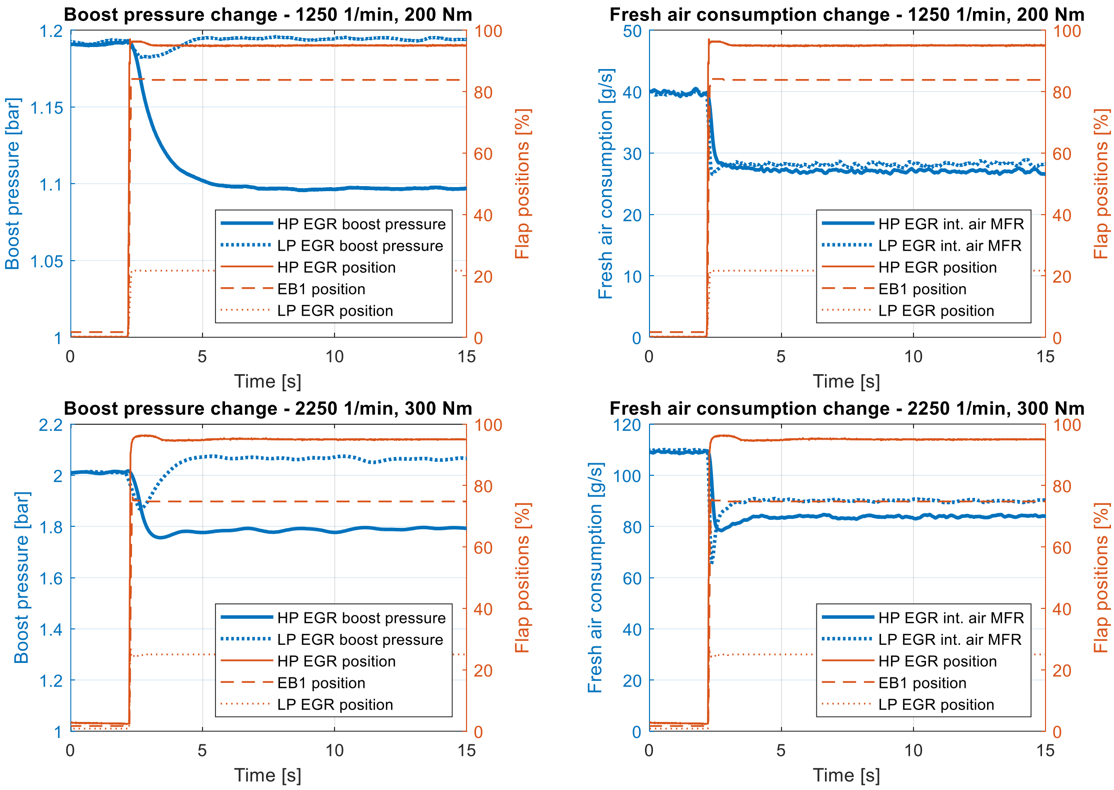

Figure 25. Boost pressure and fresh air consumption change in transient operation with HP and LP EGR

4. Conclusion

Modern Diesel engines have complex, creative control systems for low environmental impact and sustainable mobility. One of these complex elements is the exhaust gas recirculation system. In turbocharged Diesel engines, dual-loop EGR systems are sometimes implemented. However, the pressure differences determine the maximum EGR mass flow rate with only EGR valves. Intake throttles and exhaust brakes can support the EGR systems, providing the highest freedom to set any EGR rate. The present paper showed the differences between the four possible EGR modes from several aspects, for instance, fuel consumption, NO_x, and soot emission or boost pressure. The comparison is made both in steady state and in simplified transient operation. It is based on engine dyno measurements.

The result evaluation also explains the details of the EGR operation. EGR makes several changes in engine operation by changing the intake charge's composition, pressure (through its effect on the turbocharger's operation), and temperature. Indirectly, the combustion process changes, resulting in a different efficiency and fuel consumption map. The performance and emission changes were depicted as a function of the air-fuel equivalence ratio – because both the intake and exhaust oxygen concentrations change with EGR.

Concluding remarks according to the fuel consumption results:

1. EGR increases fuel consumption progressively. The increase is 10-15% with a lower EGR rate. Its maximum can reach 25%.
2. The ranking of the EGR modes strongly depends on the turbocharger's operation changes. Since the boost pressure mainly depends on the engine power, i.e., indirectly, the ranking depends on the engine power.
3. LP EGR provides lower fuel consumption in most operation points than HP EGR. The difference is about 1-10%. It increases with higher EGR rates. The difference is very small during the HP EGR support valve operation.
4. The exhaust brake-supported HP EGR always consumes less fuel than the intake throttle-supported one. The two LP EGR modes do not show significant differences.
5. With high engine powers and with high wastegate openings, the rankings are changed. Firstly the HP EGR provides lower fuel consumption until its support valves operate. With very high engine powers, HP EGR (and within that, the exhaust brake supported one) proved to be better. LP EGR provides up to 1-5% higher fuel consumption.

6. HP EGR can reduce fuel consumption with high boost pressure and a low EGR rate. The decrease is about 1%.

Concluding remarks according to the NO_x and soot emission results:

1. It is well known that NOx and PM emission formation is counterproductive. I.e. generally, the EGR mode that is favourable to one is unfavourable to the other emission.
2. In most of the operation points, there are no emission differences between the two EGR systems until the HP EGR support valves operate. The exception is only with high engine powers.
3. LP EGR provides lower NO_x emission and opacity when HP EGR support valves operate. The NO_x difference is around 20-35%, except with high engine powers.
4. With high engine powers, HP EGR can have a slight NO_x advantage while its support valves operate (7-10%). Moreover, HP EGR also can have a significant opacity advantage with up to 60% engine power.
5. HP EGR with exhaust brake provides lower NO_x emission than with intake throttle. The difference is about 5-20%. The two LP EGR modes do not show significant differences.
6. With higher EGR rates, LP EGR can realize low-temperature combustion, where exhaust gas opacity decreases with increasing EGR. This property gradually disappears in the function of the engine power.

Concluding remarks according to the transient operation results:

1. The reaction times estimated by the oxygen concentration changes. As it was expected, HP EGR reacts faster. The maximum reaction time difference between the two EGR systems is about 0.5 s, gradually decreasing with the EGR rate. At high engine speeds, their difference is almost negligible.
2. The changes on the intake side are faster than on the exhaust side. On the intake side, HP EGR's reaction is prompt.
3. During the transient cycles, the emission changes show similar properties. Both the NO_x and soot emission changes are faster with HP EGR. The overall reaction of the emission changes slightly slower than the O_2 concentration changes.

The present addressed EGR research from a broader perspective: it examined the high EGR rates and the effect of the support valves. At the same time, the article also advised on the dual loop EGR design and control.

The results presented here call for further research in several directions:

- Development of an optimal EGR strategy for low emission and fuel consumption.
- Analysis of the EGR cooling effects.
- Examination of the cooperation of injection strategies and dual loop EGR.
- Dual loop EGR measurements with green fuels.

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