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Düşük-Yüklü Dizel Motor Operasyonlarında İç Egzoz Gazı Devridaimi Yoluyla Egzoz Sıcaklığı Yönetiminin İyileştirilmesi

Improving Exhaust Temperature Management At Low-loaded Diesel Engine Operations Via Internal Exhaust Gas Recirculation

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Öz

Modern karayolu otomotiv araçları sıkı emisyon yönetmeliklerini karşılamak için genellikle egzoz arıtım sistemleri (EAS) kullanmaktadırlar. Bu sistemler emisyon oranlarını azaltmada çoğu zaman etkili olsa da, düşük-yüklerde düşük egzoz sıcaklıkları (250^{o}C'nin altında) yüzünden etkisiz olmaktadırlar. Bu çalışma, bir dizel motor modeli üzerinde, egzoz sıcaklıklarının düşük yüklerde iç egzoz gazı devridaimi (İEGD) metoduyla 250^{o}C'nin üzerine çıkarılabileceğini göstermektedir. Motor sistemi 1700 devir/dakika hızda ve 2,5-4,5 bar fren ortalama efektif basınç motor yükleri arasında çalışmaktadır. İEGD silindir-içi sıcak artık egzoz gazı miktarını arttırmakta ve bu yüzden önemli bir egzoz sıcaklığı artışına (70^{o} C'ye kadar) sebep olmaktadır. Daha sıcak egzoz sistemi, EAS emisyon dönüşüm verimini çoğunlukla % 90'ın üstünde tutar ve arttırılan ısı transfer oranlarıyla (% 142'ye kadar) EAS katalizör yatağının daha hızlı ısınmasını sağlar. İEGD geleneksel EAS ısıtma teknikleri kadar yakıt tüketici değildir ve yakıt tüketimi artışını % 5'in altında tutabilmektedir.

Anahtar Kelimeler: *Dizel Motorlar, Egzoz Gazı Sıcaklığı, Egzoz Arıtımı Yönetimi, İç Egzoz Gazı Devridaimi*

Abstract

Modern on-road automotive vehicles mostly utilize exhaust after-treatment (EAT) systems to meet the stringent emission regulations. Although those systems are generally effective to reduce emission rates, they are ineffectual at low loads due to low exhaust temperatures (below 250^{o}C). This study demonstrates on a diesel engine model that exhaust temperatures can be increased above 250^{o}C at light loads through internal exhaust gas recirculation (IEGR) method. Engine system operates at 1700 RPM engine speed and within 2.5-4.5 bar brake mean effective pressure (BMEP) engine loads. IEGR increases the amount of in-cylinder hot residual exhaust gas and thus causes a considerable exhaust temperature rise (up to 70^{o}C). Warmer exhaust system keeps EAT emission conversion efficiency mostly above 90 % and accelerates EAT catalyst bed warm-up through increased (up to 142 %) heat transfer rates. IEGR is not as fuel-consuming as conventional EAT warming techniques and can keep the fuel consumption rise below 5 %.

Keywords: *Diesel Engines, Exhaust Gas Temperature, Exhaust After-treatment Management, Internal Exhaust Gas Recirculation*

1. Introduction

Diesel engines are widely used on automotive vehicles due to their fuel efficient and cost effective performance. However, current diesel tailpipe emission limits are highly strict and pose a significant challenge for their widespread usage on land transportation [1]. Engine producers constantly search new on-engine techniques (renewable energy, advanced combustion methods) to minimize engine-out emission rates [2-4]. Although those methods have a significant potential to reduce emission rates, they either cause fuel inefficiency or cannot satisfactorily curb harmful pollutants at some loads. Therefore, diesel engines are generally equipped with EAT systems to meet the strict emission regulations.

Typical EAT systems are Selective Catalytic Reduction (SCR), Diesel Particulate Filter (DPF) and Diesel Oxidation Catalyst (DOC). While DOC is used to abate tailpipe unburned hydrocarbons (HCs) and carbon monoxide (CO), DPF and SCR are utilized to decrease engine-out particulate matter (PM) and nitrogen oxide (NO_x), respectively. Although those systems are powerful tools to minimize emission rates, they have a considerable shortcoming: they highly depend on temperature. They generally need to be kept above a certain temperature (normally 250^{o}C) in order to perform effectively [5,6]. At high loads, this can be easily overcome since exhaust gas temperatures remain generally above 250^{o}C. However, at low loads, exhaust temperatures decrease well below 250^{o}C and thus EAT effectiveness reduces dramatically. There is a need to rise exhaust temperatures at those loads to improve exhaust thermal management on diesel engine systems [7-8].

On previous studies, early exhaust valve opening (EEVO) was examined by many researchers as a solution for low exhaust temperatures at low loads [9-11]. In EEVO mode, exhaust discharge starts while piston is relatively close to top dead center (TDC) where in-cylinder pressure and temperature are still high. Thus, there is a considerable improvement on exhaust temperature. However, EEVO also decreases the total work produced due to earlier blow-down of the pressure during the expansion stroke. Therefore, it requires fuel consumption penalty (above 5 %) to keep engine load constant [12]. In addition to EEVO, decreasing in-cylinder air induction can also increase exhaust temperatures on diesel engine systems [13-16]. Those air-flow reducing methods are highly effective, however, they cause a dramatic reduction on volumetric efficiency which can be critical during transient-state operations. Moreover, reduced exhaust flow rates decrease the total heat transferred to the EAT systems and result in late EAT warm-up.

This study demonstrates on a diesel engine model that IEGR can be utilized to elevate exhaust temperatures above 250^{o}C at low loads. IEGR boosts exhaust temperatures through increasing the amount of residual hot exhaust gas inside the cylinders. It can keep the fuel penalty relatively low (below 5 %) compared to other fuel-consuming methods. It can also improve EAT warm-up at low loads, unlike aforementioned airflow reducing techniques.

2. Material and Method

2.1. Engine properties and model

In the study, IEGR is applied on a four-cylinder turbocharged diesel engine to improve exhaust temperature management. Main engine properties are given on Table 1. Those engine specifications are utilized to build the engine model in Lotus Engine Simulation (LES) program [17, 18].

The model is demonstrated explicitly below on Figure 1. The simulation starts with an intake system including compressor, intercooler, inlet pipes, ports and valves. On the discharging part, as exhaust valves open, exhaust gas begins to flow through the exhaust ports and the turbine.

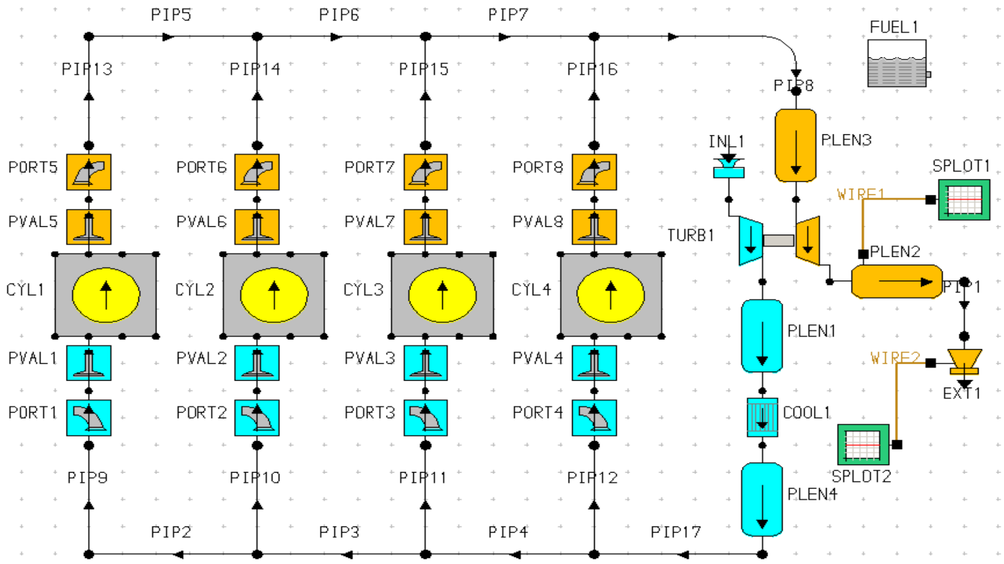

Figure 1. Engine model

After turbine, exhaust gas moves directly to the EAT system where some chemical processes minimize harmful emission rates. EAT system is not built in the simulation. The model mainly focuses on the exhaust temperature at turbine exit which directly affects the EAT system efficiency. SPLOT1, seen on Figure 1, is connected to the turbine outlet and predicts the exhaust temperature. Exhaust mass flow rate of the system is also important as it directly affects the total heat transferred to the EAT system. Therefore, SPLOT2 is used to calculate the exhaust mass flow rate.

Table 1. Engine specifications.

Model	Four-cylinder diesel engine
Air intake	Turbocharged
Bore (mm)	84.2
Stroke (mm)	102.2
Connecting rod length (mm)	171.65
Compression ratio	21:0
IVO	10° CA BTDC
IVC	30° CA ABDC
EVO	47° CA BBDC
EVC	10° CA ATDC
Cylinder firing order	1-3-4-2
Fuel injection timing	14° CA BTDC

For the air flow on intake and exhaust tailpipe system, LES uses one-dimensional model of pipe gas dynamics. Two-step Lax-Wendroff technique is used to solve mass, momentum and energy equations inside the pipe components.

The performance predictions are generally dependent on Heywood's engine performance equations [20]. Engine brake power is found with:

$$P_e = \left[BMEP V_d NZ \Big/ n_r 60 \right] \qquad (1)$$

where V_d, N, Z and n_r are displaced volume, engine speed, number of cylinders and revolutions per cycle respectively.

Engine volumetric efficiency is calculated with the following equation:

$$\eta_{vol} = \left[2\dot{m}_{ai} 10^3 \Big/ 30 \rho_{ai} V_d N \right] \qquad (2)$$

where $\dot{m}_{ai}$ and ρ_{ai} denote the mass flow rate and density of the air inducted into the cylinders. The brake thermal efficiency is found with:

$$\eta_{th} = \left[3600 P_e / Q_{LHV} \dot{m}_f \right] \quad (3)$$

where Q_{LHV} and $\dot{m}_f$ are fuel heating value (taken as 42600 kJ/kg) and fuel mass flow rate (kg/h), respectively.

In-cylinder combustion is determined with two-part Wiebe function including premixed and diffusion periods [21]. The fraction of the burned mass for each period is calculated with the following equations:

$$m_{premixed} = 1 - \left[1 - \left(\theta / \theta_b \right)^{C_1} \right]^{C_2} \quad (4)$$

$$m_{diffusion} = 1 - e^{-A\left(\frac{\theta}{\theta_b}\right)^{M+1}} \quad (5)$$

where θ denotes the burn angle (degree). C_1, C_2, A and M on equations (4) and (5) are Wiebe coefficients. These parameters are generally adjusted appropriately on the model in order to obtain a reliable simulation which shows good correlation with experimental results.

2.2. Validation of the model

In order to obtain a reliable model, it is intended to compare the performance of the simulation in nominal mode (using specifications on Table 1) with the experimental test results of a similar engine. It is found from literature review that Ahmad & Babu analyzed a similar diesel engine (same engine specifications and valve timings) in both natural aspirated and turbocharged modes in a previous study [22]. In their analysis, the engine was experimentally tested in turbocharged mode when engine speed is 1700 RPM, fuel injection timing is 14^{o} CA BTDC and cylinder firing order is 1-3-4-2 which are all similar to the properties of the LES model stated on Table 1. They experimentally measured the in-cylinder peak pressure (bar) values at several points along the engine cycle while the engine produces 19 kW power [22]. The same pressure values at 1700 RPM engine speed and constant engine power (19 kW) are calculated with LES model and compared with the experimental data of Ref. [22] in order to improve the accuracy of the model.

The comparison of in-cylinder pressure variation along piston crank angle between experiment and simulation is shown on the following Figure 2. The predictions of the model seem generally compatible with the experimental results. The difference remains below 5 % along piston crank angle which can be assumed as a good correlation. Thus, the model can be assumed as reliable and used to examine the performance of the diesel engine.

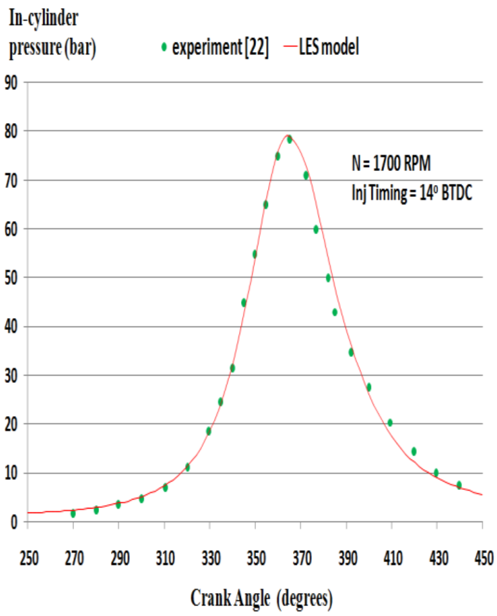

Figure 2. Validation of the simulation

2.3. The implementation of IEGR

In this study, as mentioned previously, the intention is to improve exhaust temperatures at low loads through IEGR. In fact, exhaust gas can be circulated via an external EGR (EEGR) system on diesel engine systems [23]. However, it is also possible to circulate it through modulating engine valve timings (IVO and EVC) as seen on Figures 3 & 4. While EEGR can cause exhaust gases to lose heat during recirculation, IEGR secures total exhaust heat and thus can be more advantageous to enhance exhaust temperature management.

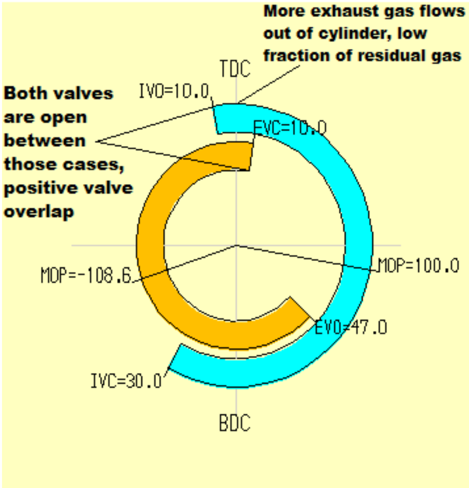

Figure 3. Nominal valve timings

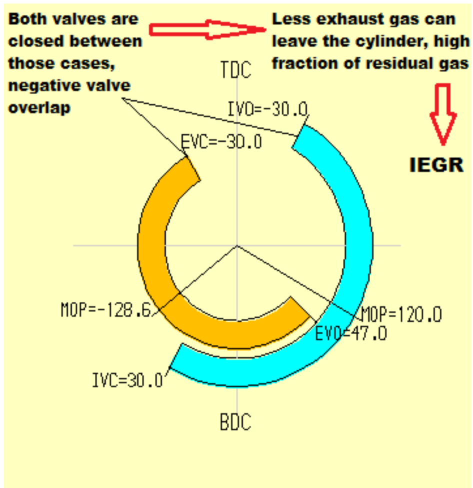

Figure 4. Negative valve overlap for IEGR

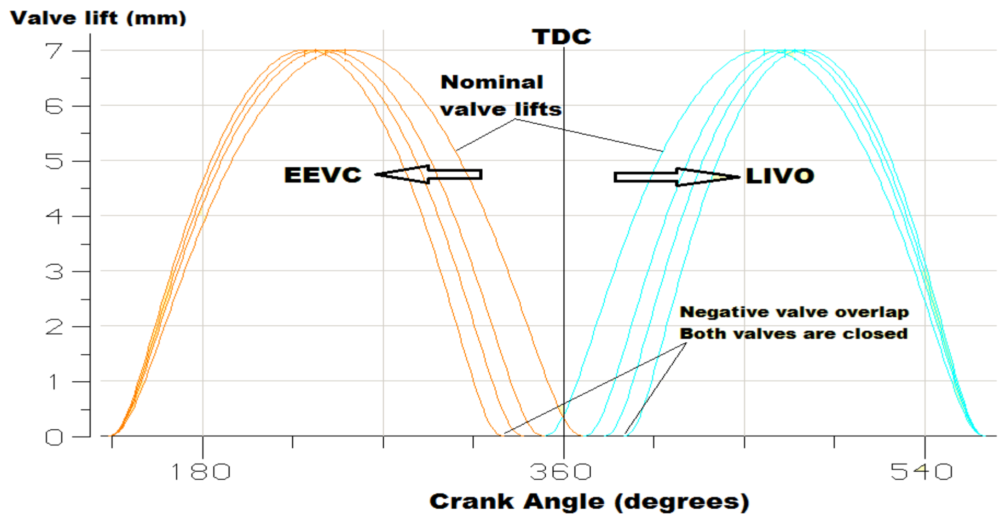

Figure 5. Change of valve lifts with early EVC (EEVC) and late IVO (LIVO)

Nominal intake & exhaust valve timings are shown explicitly on Figure 3. IEGR is not related with EVO and IVC, those valve timings are kept constant. It is rather achieved through appropriate adjustment of IVO and EVC as illustrated on Figure 4. In order to re-induct exhaust gases, valve overlap (crank angle (CA) duration both intake and exhaust valves remain open) is decreased. In nominal condition on Figure 3, a high percentage of in-cylinder exhaust gas can flow out of the cylinder (mostly through exhaust ports, tiny fraction through intake ports) since both valves are open close to TDC (total 20° CA opening, positive valve overlap). In this mode, exhaust gases have both time and a high valve opening area to leave the cylinder. Only a small amount of residual gas can remain inside the cylinders and thus, exhaust temperatures are not affected significantly. However, as shown on Figures 4 & 5, in negative valve overlap (NVO) mode, exhaust gases have limited time and a relatively low valve opening area to discharge out of the cylinder since both valves remain closed for a certain CA duration before and after TDC (EVC is advanced and IVO is retarded). Compared to nominal mode, a higher amount of residual hot exhaust gas remains inside the cylinders (IEGR occurs) and is mixed with the air inducted on the next cycle. Therefore, exhaust temperature flowing through the EAT system increases considerably. In the study, NVO is not allowed to exceed 60° CA (EVC is adjusted as early as 30° CA BTDC and IVO is modified as late as 30° CA ATDC) and effects of IEGR are examined within that limit.

3. Results

At first, the effect of NVO on exhaust flow rate is demonstrated for nominal & IEGR modes on Figure 6. Engine load is managed constant at 3.0 bar BMEP in both modes. As seen, exhaust gas release is cut off in IEGR mode before TDC due to early EVC. Late IVO avoids any exhaust flow into the intake port as well. As EVO is constant in both modes, there is less time for discharge in IEGR mode. Exhaust flow rate in IEGR mode is higher at the beginning of the discharge due to faster rise of exhaust valve lift seen on Figure 5. However, exhaust valve lift also decreases and goes back to zero earlier in IEGR mode. This early closure causes a sudden reduction on exhaust flow rate. Therefore, as shown on Figure 7, there is a steady rise on the amount of in-cylinder hot residual exhaust gas as NVO is increased. While residual exhaust gas percentage is below 5 % in nominal mode, it rises up to 10 % as NVO is adjusted to 20° CA and even exceeds 20 % as NVO is set to 60° CA.

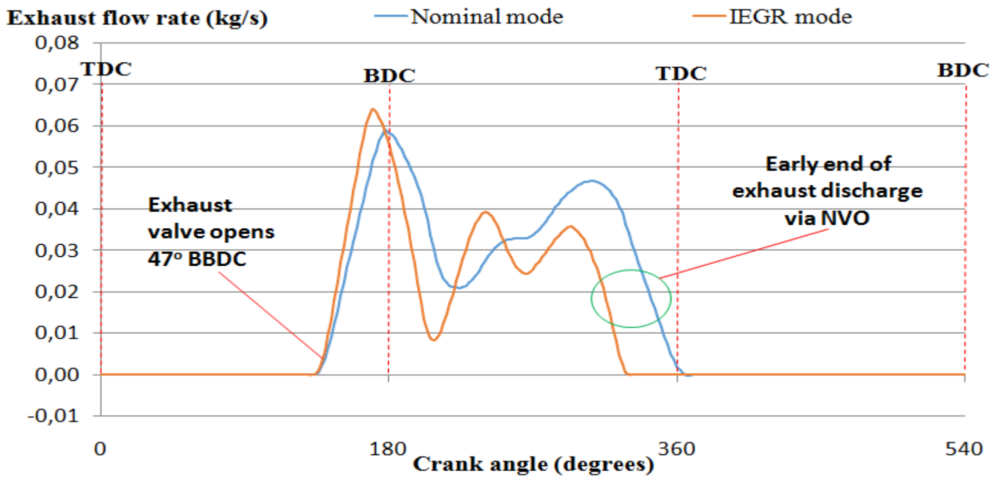

Figure 6. Exhaust flow rate during nominal and IEGR modes

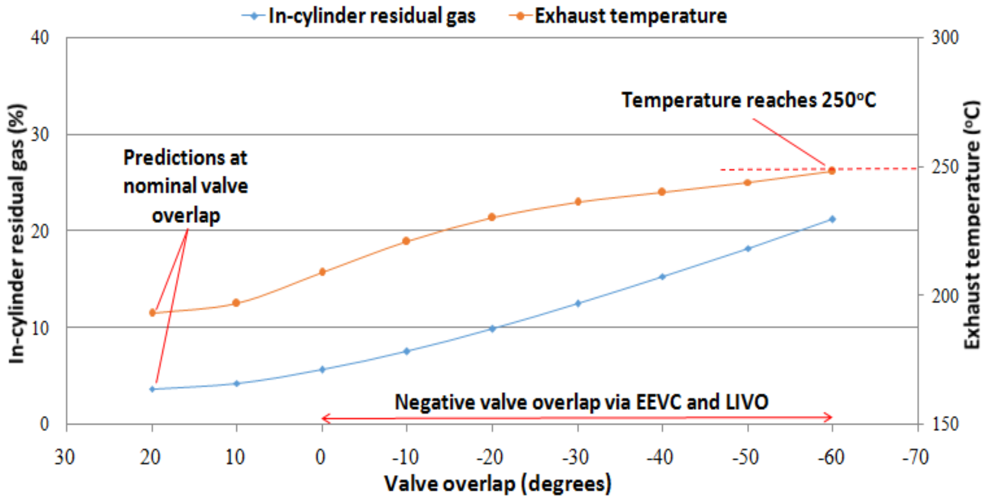

Figure 7. Change of exhaust temperature and residual gas via NVO

Rise on residual gas directly affects the exhaust temperature as illustrated on Figure 7. The higher percentage the residual gas remains inside the cylinders, the higher temperature the exhaust gases flow into the EAT system. While the nominal exhaust temperature is below 200°C (too cold for effective EAT), keeping NVO at 60° CA results in higher than 50°C exhaust temperature rise, enabling engine system to manage temperature at 250°C (appropriate for effective EAT). IEGR proves that it can keep the EAT system at an effective level. It keeps total exhaust heat and instantly improves exhaust temperature. In fact, exhaust temperature is hardly expected to exceed 250°C via re-inducting the same amount of exhaust gas with an EEGR system considering the heat loss it causes during recirculation.

IEGR boosts exhaust temperatures, however, it also rises fuel consumption as seen on Figure 8. Keeping more exhaust gas inside the cylinders affects combustion negatively and more fuel (up to 4.5 %) is required to keep engine load constant at 3.0 bar BMEP. Therefore, brake thermal efficiency is reduced up to 1.2 %. However, the trade-off engine has to suffer to improve exhaust temperatures is relatively low in comparison to conventional fuel-consuming methods [9-12].

As illustrated on Figure 9 below, IEGR causes reduction on volumetric efficiency as well. However, the reduction is relatively low compared to aforementioned conventional air-flow reduction techniques [13-16]. Thus, it can be predicted that IEGR can be more successful during transient operations compared to those methods.

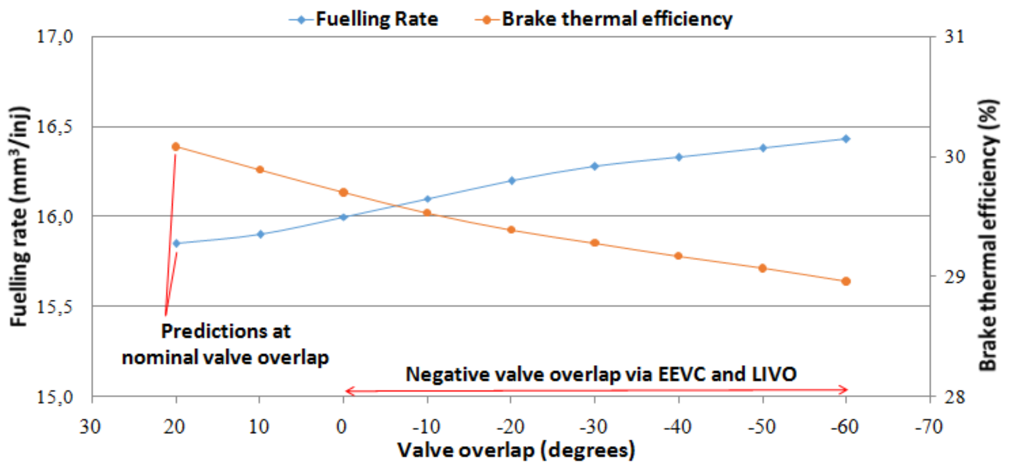

Figure 8. Change of fuelling rate and brake thermal efficiency via NVO

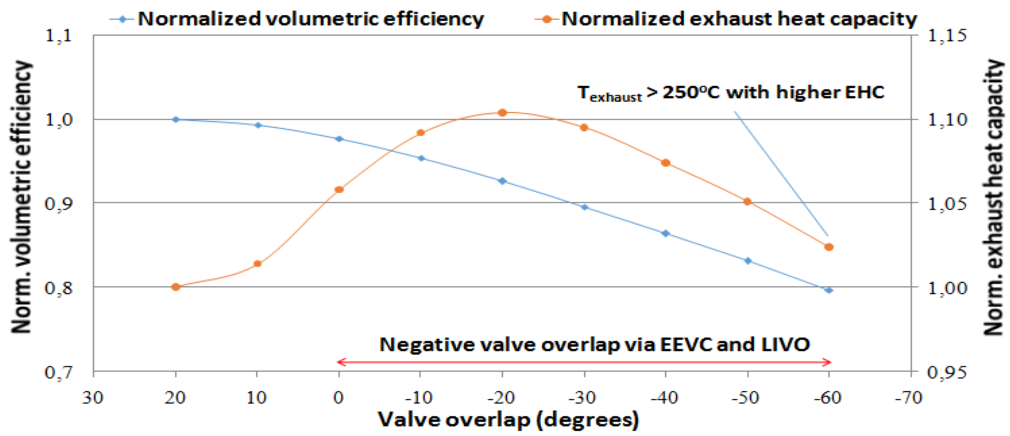

Figure 9. Change of normalized η_{vol} and normalized EHC via NVO

It is also shown on Figure 9 that IEGR can rise exhaust heat capacity (EHC) while managing exhaust temperature above 250°C. Air-decreasing methods generally cause a dramatic reduction on EHC ($m_{exhaust}$ $C_{p,e}$ $T_{exhaust}$) which affects EAT warm-up negatively. EHC decreases after a certain NVO (20° CA) on Figure 9 due to the sharp reduction of exhaust flow rate.

The method is shown to be effective on raising exhaust temperature and exhaust heat capacity on a particular engine load (3.0 bar BMEP). In order to examine its effects on other loads, IEGR is applied on an extended load range (from 2.5 bar to 4.5 bar BMEP) to improve exhaust temperatures. Change of exhaust temperatures at nominal and IEGR modes within this load range is demonstrated on the following Figure 10 below. 250°C is stated on the figure as the threshold temperature for effective EAT management.

As seen, exhaust temperature remains below 250°C almost at all loads in nominal case. It even decreases below 200°C when load is under 3.0 bar which is too cold for efficient EAT management. However, in IEGR mode, temperature is mostly above the threshold temperature. It exceeds 200°C even at lowest load which is hot to keep EAT system effective. In fact, the improvement on EAT management can be better illustrated on Figure 11 below by comparing the SCR efficiency at both modes.

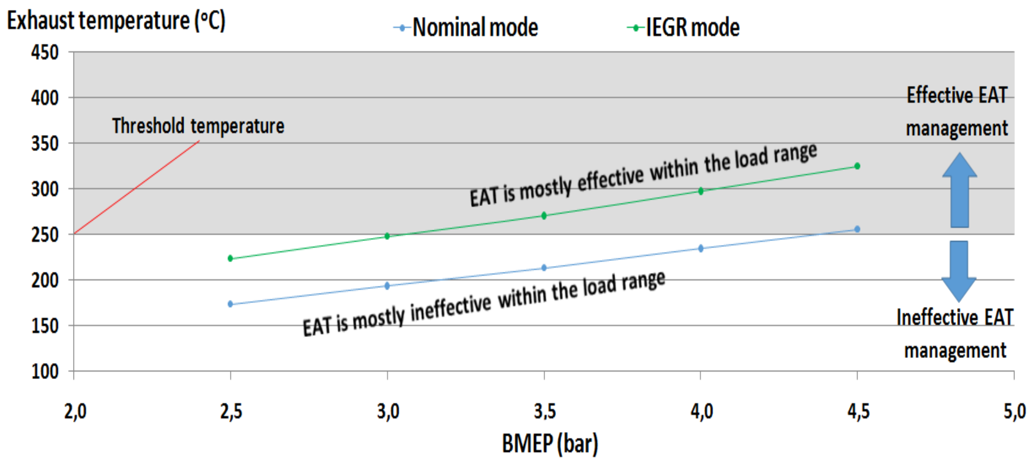

Figure 10. Change of exhaust temperature at different engine loads

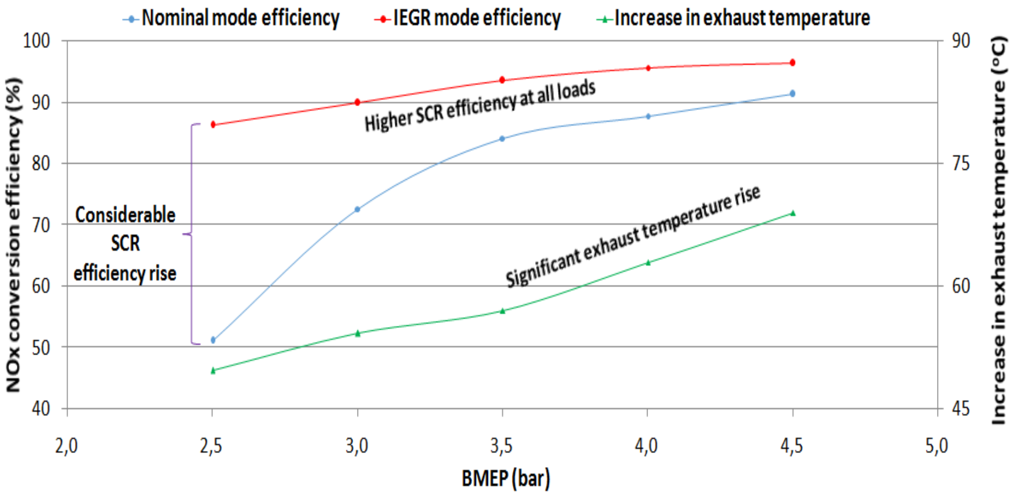

Figure 11. Change of SCR efficiency at different engine loads

Change in NOx conversion efficiency on Figure 11 is obtained at both modes using the diagram of SCR efficiency variation depending on exhaust temperature in Ref. [24]. The predictions based on this diagram demonstrates on Figure 11 that SCR efficiency can be kept above 90 % only at a limited load range in nominal mode. It is highly insufficient at most of the range and even falls close to 50 % at 2.5 bar. However, IEGR holds SCR efficiency generally above 90 % and even above 95 % at relatively higher loads. Moreover, it boosts efficiency above 85 % at 2.5 bar which is a noticeable improvement. The improvement on SCR efficiency at high loads degrades even though IEGR causes higher exhaust temperature rise at those loads. This is because exhaust temperatures in nominal mode are already hot (above 200°C) at high loads and SCR efficiency remains mostly constant between 300°C and 400°C exhaust temperatures [24].

Higher exhaust temperatures via hot residual gases is definitely effective on EAT efficiency at low loads. However, EAT system is not only affected by exhaust temperature, but also by exhaust mass flow rate. Most of the emissions are released during engine warm-up which is directly related to the EAT warm-up. Therefore, heat transfer rates to the EAT system are compared in IEGR and nominal modes to assess the improvement on EAT warm-up.

For each mode, heat transfer rates can be calculated with the following formula [25]:

$$Q = C\left(\dot{m}_{exhaust}^{4/5}\right)\left(T_{exhaust} - T_{catalyst}\right) \qquad (6)$$

where $m_{exhaust}$ denotes exhaust mass flow rate, $T_{exhaust}$ and $T_{catalyst}$ are exhaust and catalyst temperatures, Q represents the heat transfer from the exhaust gases to the EAT catalyst substrate and C is a constant depending on the geometry and material of the EAT system.

At constant load, while $m_{exhaust}$ and $T_{exhaust}$ remain constant, steady exhaust flow increases $T_{catalyst}$ until it is equal to $T_{exhaust}$. When temperatures match, heat transfer reduces to zero. However, during transient conditions (from high loading cases to low loads), $T_{exhaust}$ decreases abruptly and EAT system loses heat (negative heat transfer) until $T_{catalyst}$ is equal to $T_{exhaust}$. Lower negative heat transfer can be useful to keep EAT effectiveness longer on transient operations.

All heat transfer rates at each load (2.0 and 4.0 bar BMEP) are normalized by denoting the prediction obtained for 0°C $T_{catalyst}$ in nominal mode as 1.0. Depending on this criteria, normalized heat transfer rates are illustrated on the following Figures 12 & 13, respectively.

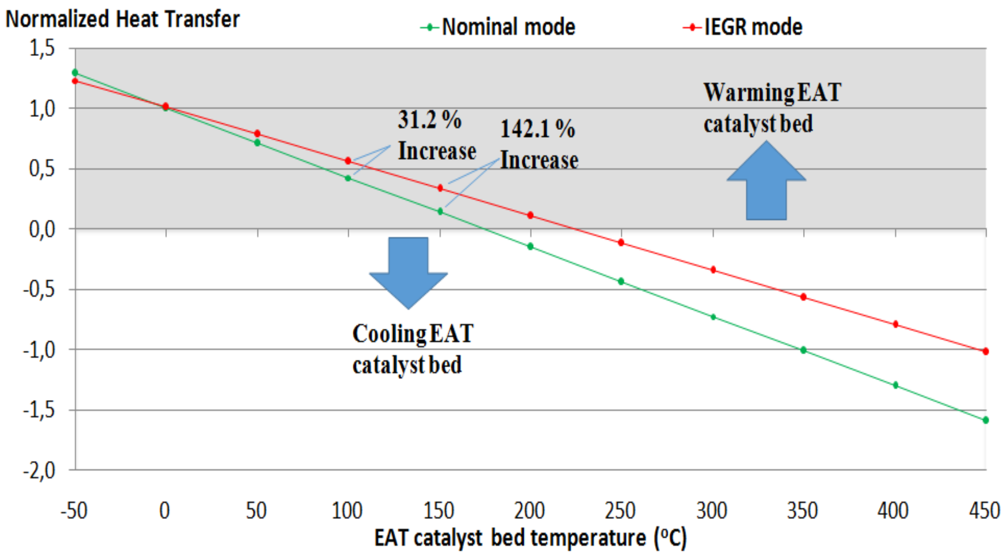

Figure 12. Heat transfer rates in nominal & IEGR modes at 2.0 bar BMEP

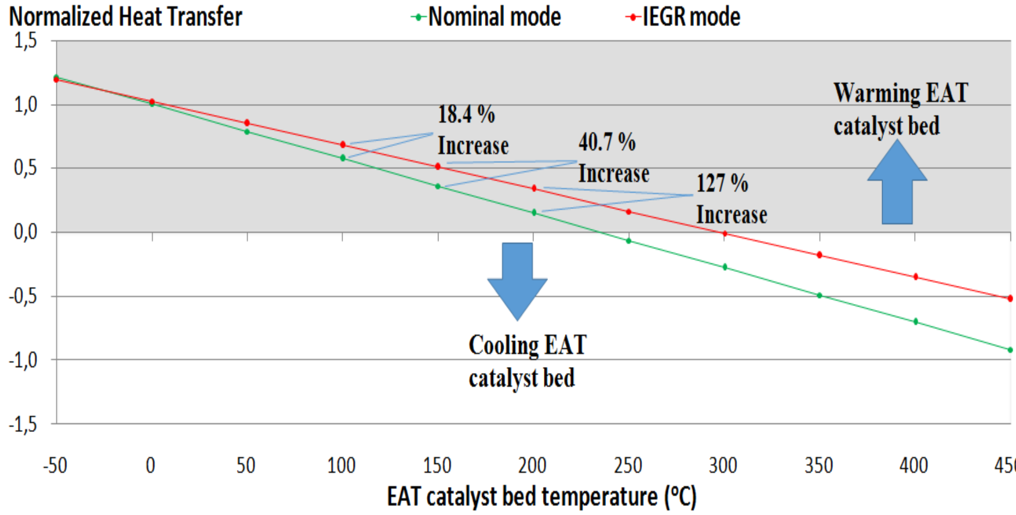

Figure 13. Heat transfer rates in nominal & IEGR modes at 4.0 bar BMEP

As seen on Figure 12, IEGR not only keeps EAT catalyst temperature above 200°C at 2.0 bar BMEP, but also results in faster EAT warm-up through higher heat transfer rates. Rise on heat transfer is relatively negligible when $T_{catalyst}$ remains below 100°C, however, there is up to 142 % improvement on heat transfer rates at warmer catalyst temperatures.

IEGR also improves negative heat transfer rates on Figure 12. In nominal mode, there is a noticeable cooling effect on EAT bed temperature due to low $T_{exhaust}$ and high $m_{exhaust}$. However, decreased $m_{exhaust}$ and increased $T_{exhaust}$ enable EAT system to lose lower heat in IEGR mode. During transient operation, especially from higher to lower engine loads, EAT effectiveness can be endured longer in EGR mode through lower heat losses.

A similar heat transfer improvement is also achieved at 4.0 bar BMEP engine load on Figure 13. At this load, IEGR is effective almost at all catalyst temperatures. Particularly when $T_{catalyst}$ is above 50°C, improvement on heat transfer rate steadily increases up to 127 %. Therefore, there is a faster EAT warm-up in IEGR mode. Negative heat transfer rates are also highly improved due to much warmer $T_{exhaust}$ (exceeding 300°C) and reduced exhaust flow rate.

4. Discussion and Conclusion

In this study, IEGR is implemented on a diesel engine system to improve exhaust temperatures and EAT effectiveness at low loads. IEGR is achieved through NVO which limits the exhaust discharge into the exhaust ports and rises the amount of hot residual exhaust gases inside the cylinders.

IEGR is found to be considerably effective at rising exhaust temperatures (up to 70°C) at light loads (within 2.5 and 4.5 bar BMEP). In comparison to nominal mode, exhaust temperature crosses the critical threshold (250°C) at a lower load (3.1 bar instead of 4.3 bar) and thus EAT effectiveness remains above 90 % in a wider load range.

IEGR improves EAT catalyst bed warm-up as well. Heat transfer rates to the EAT bed are increased up to 142 % at 2.0 bar and up to 127 % at 4.0 bar BMEP engine load cases. It also reduces negative heat transfer rates at both loads which can be beneficial for enduring EAT effectiveness during transient (from high to low load) operations.

In IEGR mode, combustion is affected negatively since fresh air is mixed with higher amounts of residual exhaust gas in comparison to nominal mode. Therefore, brake thermal efficiency is reduced up to 1.2 %. However, the inefficiency is relatively low compared to other conventional exhaust system warming methods.

Unlike EEGR systems, implementing IEGR via NVO neither requires any external material to re-circulate the exhaust gases nor causes any exhaust heat loss during recirculation. It has a significant potential to improve exhaust thermal management at low loads. Therefore, it can be utilized with other on-engine techniques to further enhance EAT management on diesel engine systems.

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