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A Simulation Based Study to Improve Active Diesel Particulate Filter Regeneration through Waste-gate Valve Opening Modulation

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Abstract

Nowadays, automotive vehicles are generally equipped with diesel particulate filter (DPF) systems in order to meet the strict particulate matter (PM) emission legislations. DPFs are highly practical systems; however, they need periodic active regeneration to clean the collected PM on their filters. Those regeneration processes are mostly effective when engine exhaust gas temperature is above 500ºC. At medium load diesel engine operations, exhaust temperatures generally remain below 500ºC which is insufficient to maintain active DPF regeneration. Therefore, the aim of this study is to elevate exhaust temperatures above 500ºC at those engine loads via modulating waste-gate valve opening (WGVO).

In the analysis, a medium-duty diesel engine is modeled via using Lotus Engine Simulation (LES) software. It is set to operate at 1700 RPM engine speed and within 5.75-7.75 bar brake mean effective pressure (BMEP) engine load. WGVO modulation can control the mass flow rate of hot exhaust gas which bypasses the turbine and moves directly to the DPF. Lower exhaust expansion on turbine due to open waste-gate decreases compressor effectiveness and thus reduces volumetric efficiency. Reduced airflow causes an increase in exhaust temperature from 35ºC to 100ºC in the load range. While exhaust system can be warmed up above 500ºC at 6.75 bar BMEP in half open waste-gate mode, this can only be achieved at 7.5 bar BMEP in waste-gate closed mode. The method is highly effective; however, it results in rise (up to % 9.2) on brake specific fuel consumption (BSFC) due to increased cylinder heat loss which needs to be considered.

Keywords: Diesel Particulate Filter; Active Regeneration; Thermal Management; Exhaust Temperature; Waste-gate Valve Opening

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1. Introduction

Diesel engines have a major role in the automotive vehicle market due to their high efficiency, low operating costs, high durability and reliability. However, they are also responsible for releasing high levels of PM into the environment which present a serious threat to human health [1]. Therefore, some stringent regulations on PM emission rates of diesel engines have been issued by environmental authorities. For instance, in the United States, Environmental Protection Agency (EPA) requires that PM emission rates for heavy-duty diesel engines should remain below 0.013 g/kWh. Also, in the European Union, Euro 6 emission legislation demands that PM emission rates for passenger cars should not exceed 0.005 g/km [2]. Engine producers generally utilize DPF technology on highway vehicles in order to meet those strict PM emission regulations. DPF system can store the engine-out PM within its filter during vehicle operation. For effective performance, those collected PM within the filter should be cleaned periodically [3]. One way to restore the filter is to rise DPF wall temperature until the trapped PM can be oxidized which is called active regeneration [4]. That process generally needs engine outlet temperatures maintained between 500ºC and 600ºC for a complete DPF regeneration [5-7]. Diesel exhaust temperatures mostly remain below 500ºC at medium load operations which is inadequate for active regeneration. Exhaust temperatures at those loads need to be increased to improve active DPF regeneration so that DPF system can renew its filter capacity and operate in an effective manner.

On previous studies, one solution to increase exhaust temperatures at DPF inlet is to inject diesel fuel at the inlet of diesel oxidation catalyst (DOC) [8, 9]. Using burner at DPF inlet is also another method to rise exhaust temperatures [10, 11]. Some researchers utilize electric heaters to improve DPF regeneration [12-15]. Those engine-independent methods not only require extra energy (other than operating the vehicle) for DPF regeneration, but also need some additional components to be mounted on the engine system which is cost-ineffective.

Engine-dependent thermal management techniques are also examined to elevate exhaust temperatures. Airflow reducing techniques such as intake throttling [16-18], early and late intake valve closure [19-21] and cylinder deactivation [22-25] achieve to obtain high exhaust temperature rise through decreasing air-fuel ratio (AFR) on diesel and gasoline engine systems. Early exhaust valve opening (EEVO) is found to be highly effective to increase exhaust temperatures as well [26-28]. However, it results in fuel inefficiency due to the early blow-down of cylinder pressure [29]. Another engine-dependent method is to use late fuel injection (LFI) [30, 31]. LFI postpones combustion and lowers the cool-off time of in-cylinder gas. Thus, exhaust temperature at after-treatment inlet is increased. However, similar to EEVO, it requires high fuel consumption due to its negative effect on combustion efficiency. Those aforementioned engine-parameter based techniques are generally combined with engine-independent methods (fuel-dosing, utilizing burner, electrical heating) in order to minimize BSFC rise during operation. Therefore, there is a constant search for alternative engine-specific thermal management strategies for effective after-treatment management [32].

The objective of this study is to demonstrate on a diesel engine model that WGVO can be modulated at medium load operations to rise exhaust temperature above 500ºC and thus improve active DPF regeneration. The method can rise the exhaust temperature up to 100ºC which is sufficient to sustain active DPF regeneration without using a burner or hydrocarbon dosing of a DOC. Therefore, it can be used as an alternative engine-dependent thermal management technique to enhance active DPF regeneration.

In the following sections, firstly, engine model is introduced and mathematical relations used are given. Secondly, the WGVO modulation on the engine system is explained. Then, effects of WGVO control on engine performance are demonstrated on several figures. Finally, the study is summarized and some future work is recommended.

2. Methodology

2.1 Engine Specifications and Engine Model

Specifications of the diesel engine used in this theoretical analysis are given on Table 1. It is a four-stroke four-cylinder diesel engine which uses a turbocharger for air-intake. Those kinds of diesel engines are mostly used on highway vehicles which need periodic DPF regeneration during operation.

Engine model is shown on Figure 1. It is built on a 1-D engine simulation program, Lotus Engine Simulation (LES) [33, 34], via using the properties on Table 1. Engine valves, ports, cylinders, inlet & exhaust pipe systems, turbocharger, waste-gate valve, sensors for exhaust temperature and exhaust gas flow rate are explicitly demonstrated on the figure.

Table 1. Diesel engine properties.

Model	Four-stroke diesel engine
Stroke (mm)	102.2
Bore (mm)	84.2
Connecting rod length (mm)	171.65
Compression ratio	21:0
Air-intake	Turbocharged
Cylinder firing order	1-3-4-2

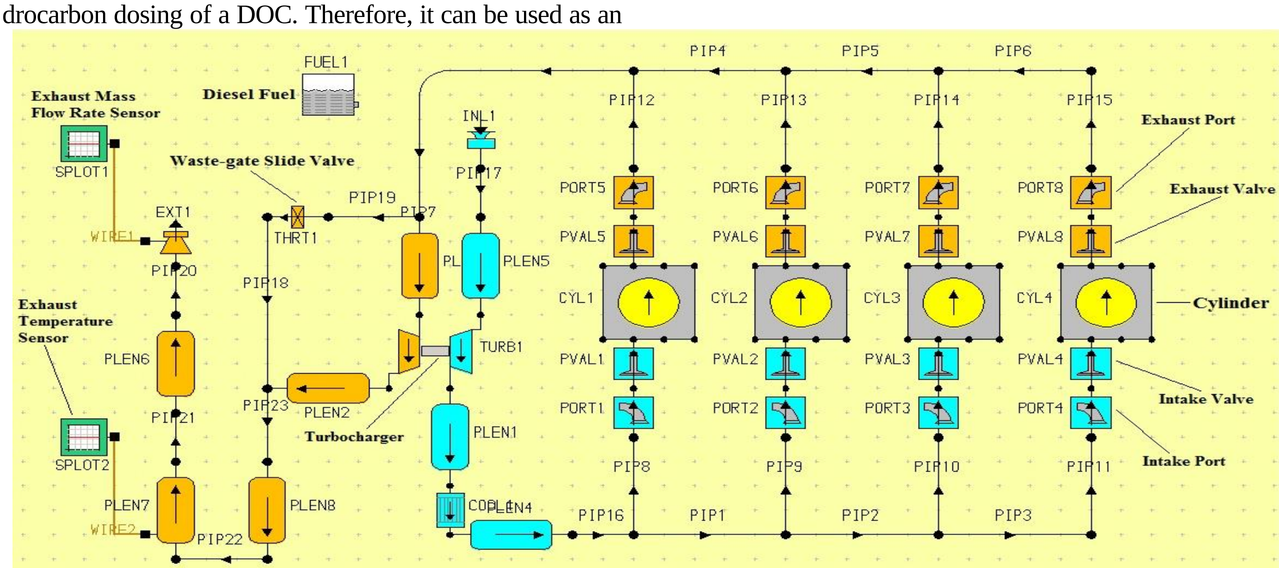

Fig. 1. Diesel engine model

As illustrated on Figure 1, there are 3 plenums (PLEN6, PLEN7 and PLEN8) before exhaust gas leaves the engine system. An after-treatment system is not modeled in the simulation. It is assumed the model uses a three-way catalytic convertor (TWC) which is widely used on highway vehicles. On a TWC system, exhaust gas flows sequentially through DOC, DPF and finally SCR before discharging into the environment. Therefore, considering the aforementioned flow direction on a TWC system, PLEN7 is assumed as a DPF system (PLEN8 as DOC and PLEN6 as SCR) and exhaust temperature at this plenum is predicted via SPLOT2 which is exhaust temperature sensor. Temperature at PLEN7 should be above 500^{o}C for a complete DPF system regeneration. In the model, waste-gate slide valve opening is modulated to achieve this goal. Exhaust flow rate is also affected by WGVO modulation. SPLOT1 shown on Figure 1 measures the mass flow rate of exhaust gas while valve opening is widened or contracted.

Engine model used in this work is actually used on a previous study by the author [35] and validated with the experimental results of a similar diesel engine [36]. The comparison between test and model results in Ref.[35] is based on in-cylinder pressure variation at engine speed and brake power of 1700 RPM and 19 kW, respectively. The comparison is valid for nominal condition where waste-gate is closed. Similar to Ref.[35], this study aims to rise exhaust temperatures as well. However, instead of applying internal exhaust gas recirculation at low loads to increase exhaust temperature above 250^{o}C [35], current study implements WGVO modulation at medium loads to improve exhaust temperature above 500^{o}C.

2.2 Mathematical Formulations

LES utilizes 1-D model of pipe gas dynamics for the flow of fluid through the inlet and exhaust pipe systems. Mass, momentum and energy equations are solved in the program via using two-step Lax-Wendroff method [37].

Volumetric efficiency of the diesel engine model is found with [38]:

$$\eta_{vol} = (2\dot{m}_a 10^3/30NV_d\rho_a) \tag{1}$$

where N is the engine speed (RPM), V_d is the displaced volume during operation, ρ_a is the inlet air density and $\dot{m}_a$ is the mass flow rate of air charging into the cylinders.

Diesel engine brake power is calculated with the following formulation [38]:

$$P_B = (NZV_dBMEP/60n_r) \tag{2}$$

where Z is the number of cylinders and n_r represents the number of revolutions per cycle which is taken as 2 for four-stroke engines. As BMEP is kept constant in both nominal and waste-gate open modes, P_B (kW) remains constant too.

Two-part Wiebe function is utilized to model the combustion inside the cylinders. That function has two consecutive combustion periods: premixed and diffusion. LES uses the following two formulations for predicting the burned mass fraction in each combustion period [39]:

$$m_{premix} = 1 - \left[1 - \left[{}^{\theta}/_{\theta_b}\right]^{C_1}\right]^{C_2} \tag{3}$$

$$m_{diff} = 1 - e^{-A\left({}^{\theta}/_{\theta_b}\right)^{M+1}} \tag{4}$$

where C_1 and C_2 in equation (3) and A and M in equation (4) are Wiebe coefficients. θ shows the burn angle and θ_b is the total burn angle.

Sandoval & Heywood friction model is used on the model [40]. Friction losses due to reciprocating & rotating parts, valves and also auxiliaries are all considered in this model.

Cylinder heat losses are calculated via Annand heat transfer model. The model is defined as [41]:

$$ARe^B = {}^{hB_{cyl}}/_{k} \tag{5}$$

where B_{cyl} is the cylinder bore, k is the thermal conductivity of gas in the cylinder, h is the heat transfer coefficient, A and B are Annand coefficients for open and closed cycles. Heat transfer per unit area is determined with the following equation [41]:

$${}^{dQ}/_{A} = h(T_g - T_w) + C(T_g^4 - T_w^4) \tag{6}$$

where T_g and T_w denote temperatures of gas and cylinder wall. Also, C is another Annand coefficient which is used only for closed cycle.

BSFC (g/kWh) on the diesel engine system is found with [38]:

$$BSFC = \dot{m}_f/P_B \tag{7}$$

where $\dot{m}_f$ (g/h) is the mass flow rate of the diesel fuel.

Engine brake thermal efficiency (BTE) is predicted with equation (8) below [38]:

$$\eta_{BTE} = 3600\left[P_B/\dot{m}_f Q_{LHV}\right] \tag{8}$$

where Q_{LHV} shows the heating value of the diesel fuel which is defined as 42600 kJ/kg on the model.

2.3 Waste-gate Valve Opening Modulation

On engine model, waste-gate valve throttle is chosen as a slide valve. Slide valves can control the fluid flow by modulating the lift distance as shown on Figure 2. When valve is closed (there is no lift, h = 0), flow through the waste-gate is not allowed. As the lift (h) is increased, some exhaust gas bypasses the turbine and starts to flow through the waste-gate valve. Rate of exhaust gas flow can be controlled via modulating the valve lift. In the study, waste-gate opening is demonstrated as percentage (%) on Figures in Results and Discussion section. Valve lift is divided to the valve diameter (D) and thus waste-gate opening is stated as 50 % when it is half open (h = D/2) and 0 % when it is closed (h =0).

In the analysis, waste-gate opening is increased with five percent increments until it is half open. As the valve opening widens, more exhaust gas begins to bypass the turbine and flows through the waste-gate valve. In this mode, not all exhaust gas loses heat on turbine. Thus, some exhaust gas in the system can move directly to the after-treatment system with warmer temperatures and rise temperature at DPF inlet.

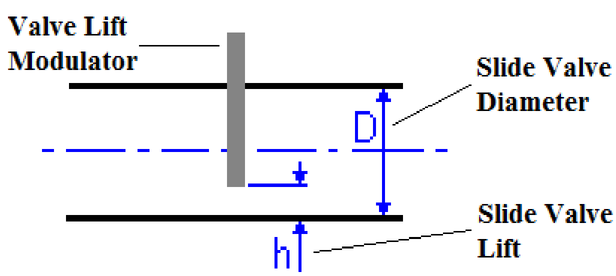

Fig. 2. Schematic of the waste-gate slide valve parameters

3. Results and Discussion

Engine model illustrated on Figure 1 is set to operate at 1700 RPM engine speed and 6.75 bar BMEP engine load. At this middle load condition, exhaust temperatures generally remain below 500ºC which is inadequate for DPF regeneration. WGVO modulation is presented as a thermal management method for active DPF regeneration in this study. As waste-gate opening is altered, fuel injection rate (FIR) on the system is adjusted to keep engine load constant at 6.75 bar.

Effect of WGVO on exhaust temperature at DPF inlet is demonstrated on Figure 3. When waste-gate is closed, exhaust temperature is much below 500ºC. At this nominal condition, active DPF regeneration is highly ineffective. Higher than 50ºC temperature rise is required to reach effective DPF regeneration line. Therefore, an additional device is needed to rise temperature and complete regeneration. However, as WGVO increases, exhaust temperature starts to rise too. It is too close to 500ºC at 40 % opening rate and it even exceeds 500ºC as waste-gate is half open.

WGVO modulation on the engine system rises exhaust temperatures due to the reduction on volumetric efficiency (η_{vol}) and air-fuel ratio (AFR) as shown on Figures 4 & 5, respectively.

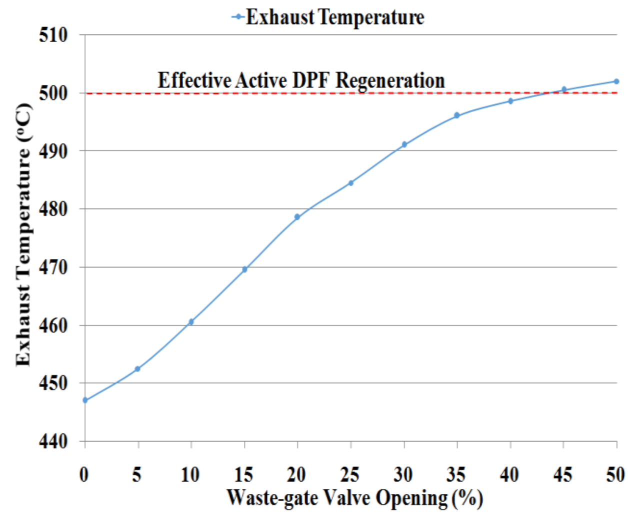

Fig. 3. Effect of WGVO on exhaust temperature

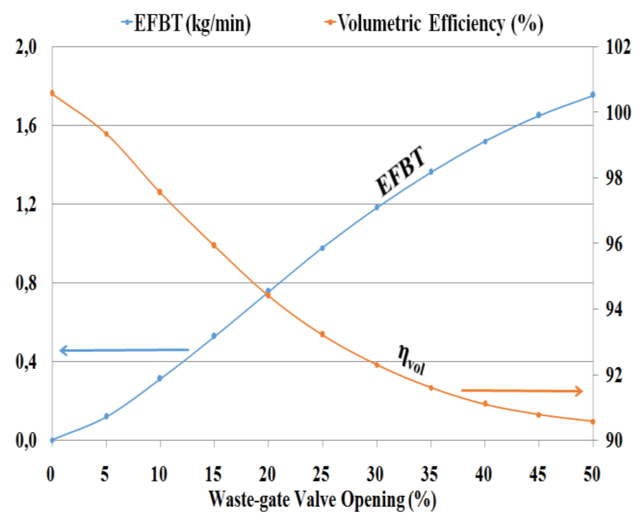

Fig. 4. Effect of WGVO on η_{vol} and EFBT

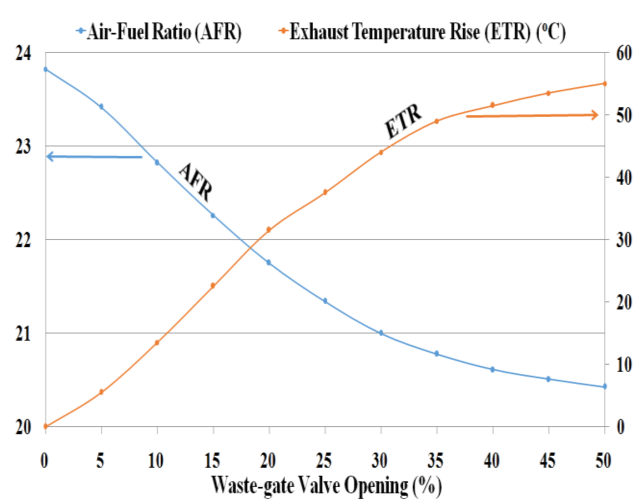

Fig. 5. Effect of WGVO on AFR and exhaust temperature rise

As valve opening is widened, exhaust flow bypassing the turbine (EFBT) rises and more exhaust gas flows through the waste-gate valve as illustrated on Figure 4. It is close to 1.0 kg/min when waste-gate opening rate is 25 % and it exceeds 1.7 kg/min at half open waste-gate mode. Compared to nominal case, lower exhaust gas flows through the turbine which results in lower turbine expansion and thus lower compressor work. As seen on Figure 4, volumetric efficiency decreases as waste-gate opening rate is increased due to the reduction on compressor work. While η_{vol} exceeds 100 % at nominal case, it reduces close to 90 % at half open waste-gate mode.

Lower volumetric efficiency leads to lower AFR as shown on Figure 5. Compared to nominal condition, AFR decreases closed to 17 %. However, the system also obtains an exhaust temperature rise of 55ºC which is sufficient to maintain DPF wall temperature above 500ºC. It can be derived that exhaust temperature rise is directly proportional with the turbine-by-passing mass flow rate and is inversely proportional with the volumetric efficiency and AFR. The lower the volumetric efficiency and the AFR are, the higher the exhaust gas temperature is kept at DPF inlet.

Lower AFR through WGVO modulation seems similar to LFI technique explained on introduction section. Both methods have richer mixtures compared to nominal condition and thus result in rise on exhaust temperatures. However, LFI does not reduce airflow induction and only rises fuel injection rate due to ineffective combustion of retarded fuel injection. On previous studies, it is seen that LFI requires high BSFC rise and causes HC and CO emissions to increase [32, 42, 43]. WGVO causes rise on BSFC as well. But, it also improves pumping loss through η_{vol} reduction which is expected to limit BSFC rise. In WGVO mode, combustion occurs at higher pressure and higher temperature zone compared to LFI mode. Therefore, a significant rise on unburned HCs and CO emissions is not predicted in WGVO mode.

The method is found to be effective for improving DPF regeneration. However, its effect on other engine performance parameters should also be examined. At first, effects on pumping loss & cylinder heat loss are illustrated on Figure 6.

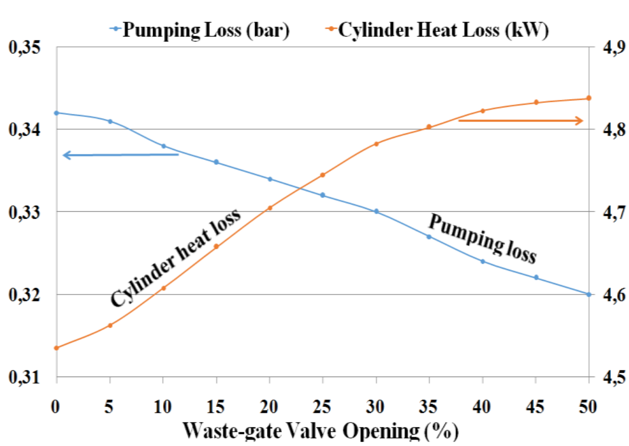

Fig. 6. Effect of WGVO on pumping loss and cylinder heat loss

As predicted, pumping loss is decreased due to the decreased volumetric efficiency. Inducting lower air into the cylinders causes a slight improvement on pumping losses. However, there is a dramatic rise on cylinder heat loss. It is certain that cylinders in half open waste-gate mode suffer higher heat losses. Those extra heat losses are due to the higher in-cylinder temperature (ICT) management in half open waste-gate mode during engine cycle, as shown on Figure 7.

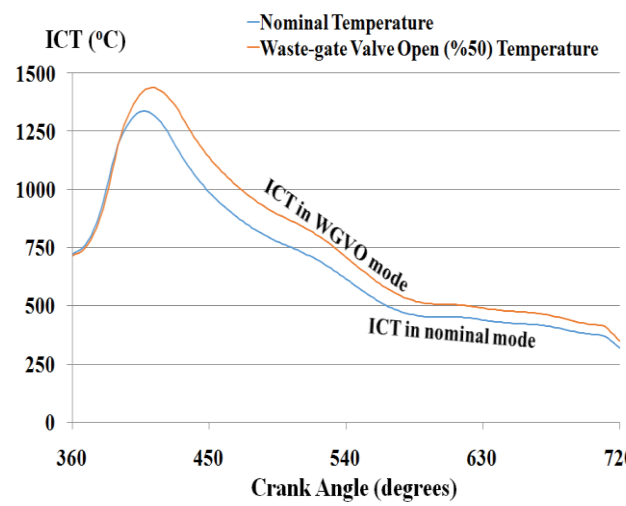

Fig. 7. In-cylinder temperature variation in nominal and half-open waste-gate modes during engine cycle

This additional heat loss due to elevated in-cylinder temperature requires engine system to use higher fuel to keep engine load constant at 6.75 bar BMEP. In fact, at middle load operations, fuel inefficiency can be predicted when volumetric efficiency decreases. Because, unlike low loads, air charge need for effective combustion is relatively high at medium loads. Therefore, more fuel is injected to compensate for ineffective combustion and cylinder heat losses. The brake power (P_B) loss on Figure 8 demonstrates the need of extra fuel injection to keep engine power (also BMEP) constant in the system.

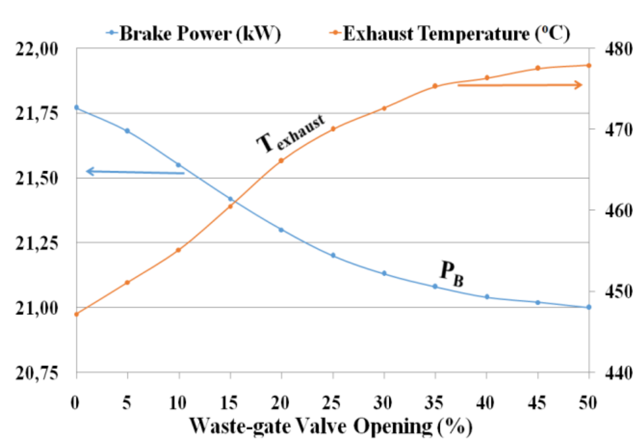

Fig. 8. Effect of WGVO on engine power and exhaust temperature at constant fuel injection rate

It is seen on Figure 8 that P_B is reduced almost 4 % as fuel injection rate is kept constant in the system. Although no additional fuel is used, $T_{exhaust}$ rises in the system. This is attributed to the richer in-cylinder mixture in WGVO mode through reduced η_{vol}. However, $T_{exhaust}$ still remains below 500ºC and power loss on highway vehicles is generally not desired during steady state operation. Therefore, FIR is modulated to maintain constant P_B in the system.

As illustrated on Figure 9, at constant load, BSFC rises steadily as waste-gate opening rate is increased. Thus, there is almost 1.5 % reduction on brake thermal efficiency (BTE) at half open waste-gate mode. However, as previously mentioned on introduction section, all conventional engine-specific methods need extra fuel to secure high exhaust temperatures for active DPF regeneration. On Figure 9, WGVO modulation can keep BSFC rise below 5 % which can be regarded as relatively low compared to those achieved in conventional thermal management methods. Also, management of waste-gate valve is simple. Unlike passive regeneration techniques (burner, reformer, electrical heating etc.), it does not require any extra component to be mounted on the engine system which may be effective to reduce manufacturing costs of engine systems. Some previous works demonstrate practical application of waste gate opening for diesel after-treatment warm-up as well [44-46].

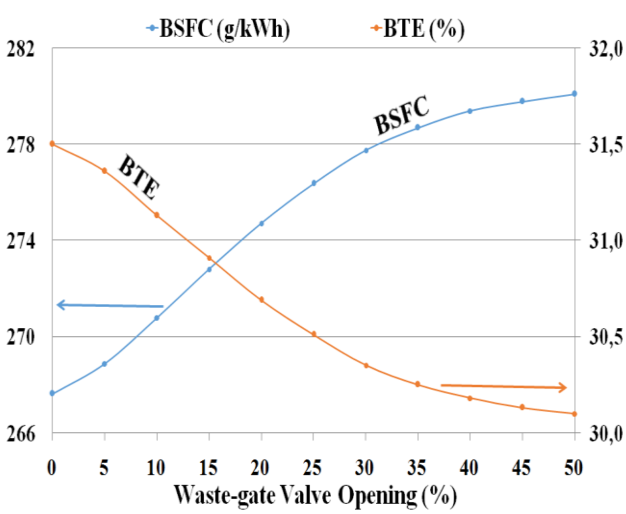

Fig. 9. Effect of WGVO on BSFC and BTE at constant load

Using extra fuel affects combustion duration as well. Midpoint of combustion phasing (CA50) and total combustion duration (0-100 % combustion) are shown on Figure 10.

Both performance parameters on Figure 10 rise steadily until waste-gate is half open. Midpoint of combustion phasing is postponed almost 3 degrees and total combustion duration increased 6 degrees. Those delayed parameters are due to the additional fuel injected into the cylinders. In WGVO mode, more fuel is burnt in extended combustion duration. Therefore, in-cylinder temperature, as illustrated previously on Figure 7, can be kept at higher levels compared to nominal mode. However, it is noted that high temperature management is achieved through rise on BSFC (Figure 9). When combustion duration is extended, greater amount of fuel is burnt in low cylinder pressure medium and thus, combustion is affected negatively.

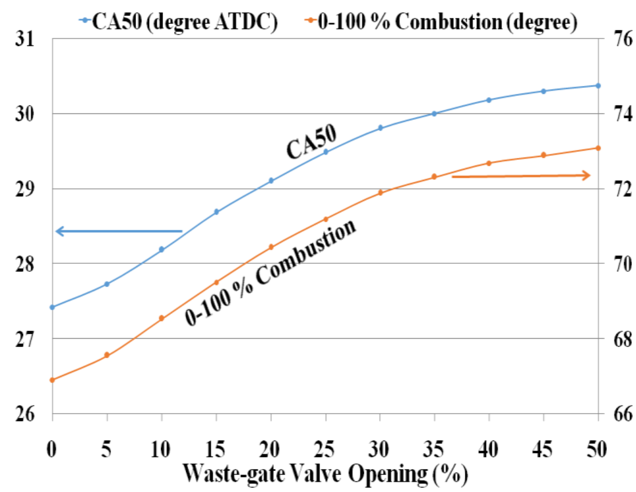

Fig. 10. CA50 and 0-100 % combustion in nominal and WGVO modes

On Figure 11 below, half open waste-gate is applied for different engine load cases (from 5.75 bar to 7.75 bar BMEP) at 1700 RPM engine speed. Similar to previous application (at 6.75 bar BMEP), engine load is kept constant at waste-gate half open (WGHO) mode via modulating fuel injection rate.

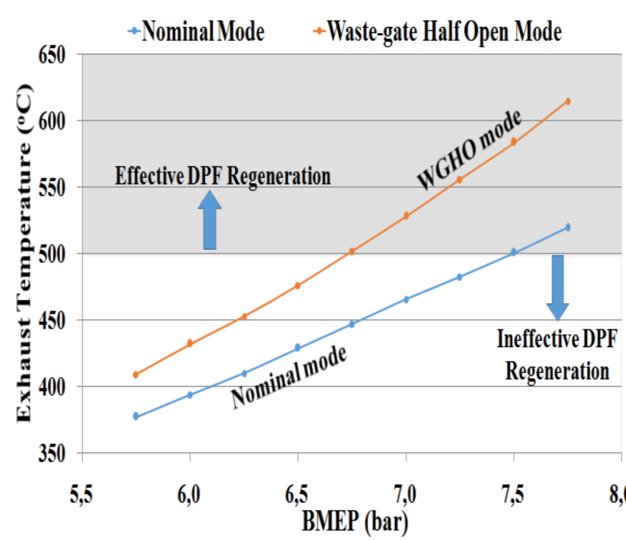

Fig. 11. Exhaust temperature variation in nominal and waste-gate half open modes for different loads

It is seen that exhaust temperatures are mostly below 500ºC in nominal mode. It is close to 375ºC at 5.75 bar which is too cold for active DPF regeneration. It can exceed 500ºC only after engine load rises above 7.50 bar. However, WGHO mode reaches effective DPF regeneration at a lower engine load (6.75

bar) in comparison to nominal mode. It results in close to 100°C exhaust temperature rise at 7.75 bar and thus, exhaust temperature at DPF inlet exceeds 600°C. While nominal mode is barely effective for a limited 0.25 bar engine load range, WGHO mode is highly effective for 1.0 bar engine load period. Although its effectiveness decreases at lower loads, exhaust temperature can still be managed above 400°C even at the lowest engine load case (5.75 bar). In nominal mode, when engine load remains below 6.5 bar, engine system requires a considerable exhaust temperature rise to reach 500°C which is difficult to achieve with conventional engine-base thermal management techniques. Therefore, engine system in this mode requires an additional device such as a burner or a reformer or electrical heating to sustain active DPF regeneration temperature. However, for the same load range, WGHO mode needs relatively lower exhaust temperature rise to exceed 500°C which can be attained via combining it with other engine-specific thermal management methods.

WGHO mode is effective at active DPF regeneration in a greater engine operation range compared to nominal engine mode. However, it also causes fuel inefficiency at all loads as shown on the following Figure 12.

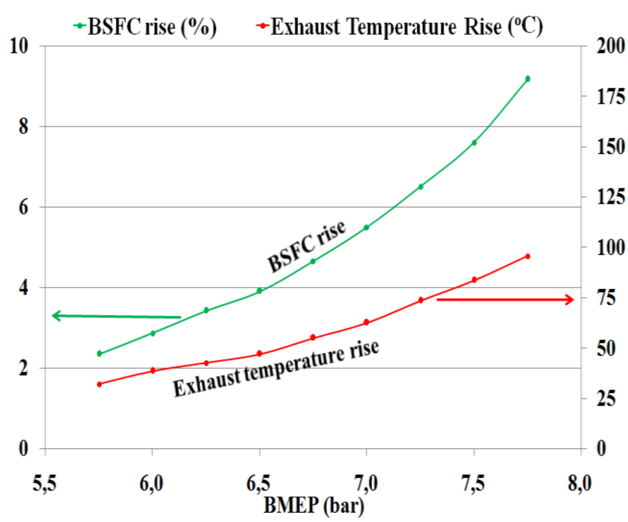

Fig. 12. Variation of BSFC rise and exhaust temperature rise in WGHO mode at different loads

It is seen that BSFC rise due to half open waste-gate decreases at lower loads. Low exhaust temperature rise at those loads probably causes lower cylinder heat losses. Also, at lower loads, reduction on volumetric efficiency is not as effective on combustion efficiency as it is at higher loads. As the load increases, higher BSFC rise is required as well. This can be attributed to high airflow need of cylinders at higher loads in order to secure high BMEP during operation.

It is also derived that exhaust temperature rise at different loads is directly proportional with the BSFC rise. While the temperature rise is close to 35°C at 5.75 bar BMEP, it gets almost 100°C at 7.75 bar BMEP which is adequate to maintain DPF inlet temperature above 600°C (Figure 11). Highly enhanced exhaust temperatures can accelerate the active DPF regeneration and thus can complete total clean-up of the filter in a faster manner. However, at those higher loads, exhaust temperature is already above 500°C. Unless there is an urgent need to complete regeneration, WGVO modulation may not be preferred at those loads considering the high BSFC rise (up to 9.2 %).

4. Conclusions

This theoretical investigation attempts to demonstrate that proper control of WGVO can be an effective way of improving active DPF regeneration at medium loads through rising exhaust temperatures. The analysis is done on a four-stroke & four-cylinder automotive diesel engine model at 1700 RPM engine speed and within engine load range of 5.75 bar-7.75 bar BMEP. Various engine performance parameters are also calculated on the simulation while WGVO is modulated to elevate exhaust temperature above 500°C.

At first, the method is applied at 6.75 bar BMEP. Nominal exhaust temperature remains below 450°C at this load which is insufficient for a complete DPF regeneration. However, half open waste-gate results in an exhaust temperature rise of 55°C at the same load and thus can sustain exhaust temperature at DPF inlet above 500°C. It is also seen that as waste-gate opening is increased, more amount of exhaust gas bypasses the expansion on turbine which causes lower compressor work and thus lower engine volumetric efficiency. Reduced airflow into the cylinders decreases AFR as well. It is seen that exhaust temperature rise on the system is due to the reduction on both volumetric efficiency and AFR.

Then, effect of WGVO modulation on fuel economy is examined. Engine system in WGVO mode needs extra fuel (up to 4.5 %) to keep engine loading constant at 6.75 bar. Increasing rates of cylinder heat loss due to high in-cylinder temperature seem to be the reason of additional fuel requirement. Using higher fuel extends combustion durations as well. It is seen that those increased combustion durations contribute to keep in-cylinder temperature at high levels.

Finally, half open waste-gate is implemented at the same speed for different engine loads (within 5.75-7.75 bar BMEP). Improvement on active DPF regeneration rises at higher loads (almost 100°C exhaust temperature rise at 7.75 bar) and fades at lower loads (only 35°C exhaust temperature rise at 5.75 bar). The same trend is valid for BSFC rise too. While it remains below 3 % at 5.75 bar, it exceeds 9 % at 7.75 bar. Although the fuel inefficiency is relatively high at high loads, increased exhaust temperatures (up to 615°C) can highly speed up the DPF regeneration process. Also, at those loads, nominal exhaust temperature begins to approach 500°C. Therefore, lower waste-gate opening and thereby lower BSFC rise can be sufficient to reach 500°C at those operating cases. It is noted that the method is applied only during regeneration. Thus, those aforementioned rates on BSFC rise are valid only during DPF regeneration, a limited duration compared to total vehicle operation time.

In future studies, WGVO modulation can be combined with other engine-base thermal management techniques to improve exhaust temperatures (particularly at engine loads below 6.75 bar). Appropriate combinations can eliminate the need of an additional device (burner, reformer or electrical heating) for active DPF regeneration and thus can be cost-effective.

Acknowledgment

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Nomenclature

AFR	:	air-fuel ratio
ATDC	:	after top dead center
B	:	cylinder bore (mm)
BMEP	:	brake mean effective pressure (bar)
BSFC	:	brake specific fuel consumption (g/kWh)
CA	:	crank angle (degree)
CA50	:	midpoint of combustion phasing
D	:	waste-gate valve diameter (mm)
DOC	:	diesel oxidation catalyst
DPF	:	diesel particulate filter
EAT	:	exhaust after-treatment
EFBT	:	exhaust flow bypassing turbine (kg/min)
ETR	:	exhaust temperature rise (^oC)
FIR	:	fuel injection rate (mm^3/injection)
h	:	waste-gate valve lift (mm)
ICT	:	in-cylinder temperature (^oC)
LES	:	lotus engine simulation
LFI	:	late fuel injection
$\dot{m}_a$	:	air flow rate (kg/min)
$\dot{m}_f$	:	fuel flow rate (g/h)
N	:	engine speed (RPM)
P_B	:	brake power (kW)
PM	:	particulate matter
Re	:	reynolds number
RPM	:	revolution per minute
S	:	cylinder stroke (mm)
SCR	:	selective catalytic reduction
T_{exh}	:	exhaust temperature (^oC)
TWC	:	three-way catalytic convertor
V_d	:	displaced volume (mm^3)
WGHO	:	waste-gate half open
WGVO	:	waste-gate valve opening
η_{vol}	:	volumetric efficiency
η_{BTE}	:	brake thermal efficiency
θ	:	burn angle (degree)

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