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FUEL-SAVING EXHAUST AFTER-TREATMENT MANAGEMENT ON A SPARK-IGNITION ENGINE SYSTEM VIA CYLINDER DEACTIVATION METHOD

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Abstract: Current on-road automotive engines generally require exhaust after-treatment systems to meet the stringent emission regulations. One major drawback of those systems is their inefficient performance at low loads due to low exhaust temperatures ($T_{exhaust}$ < 250°C). Typical on-engine techniques such as late fuel injection, exhaust gas recirculation and early exhaust valve opening require fuel consumption rise for exhaust temperature improvement. This study demonstrates that exhaust temperatures at light loads can be increased in a more fuel-saving manner via cylinder deactivation (CDA) method on a spark-ignition (SI) engine. Lotus Engine Simulation (LES) software is used to build the engine model and predict performance at 1500 RPM engine speed and over a range of constant engine torques (5 Nm to 25 Nm). In CDA mode, increased equivalence ratio of active cylinders results in up to 110°C exhaust temperature rise ($T_{exhaust}$ > 250°C) which maintains effective after-treatment at low loads. Also, after-treatment catalyst bed warm-up is improved through increased heat transfer rates. CDA technique causes a significant reduction on engine pumping loss through decreased air induction and hence is highly fuel-efficient. However, lower air induction also causes reduced exhaust flow rate which affects after-treatment warm-up negatively at some loads.
Anahtar Kelimler: Cylinder deactivation, spark-ignition engines, fuel efficiency, exhaust gas temperature, exhaust after-treatment management

SİLİNDİR DEVRE DIŞI BIRAKMA METODU İLE KIVILCIM-ATEŞLEMELİ BİR MOTOR SİSTEMİNDE YAKIT-VERİMLİ EGZOZ ARITMA YÖNETİMİ

Özet: Günümüz karayolu otomotiv motorları genellikle sıkı emisyon yönetmeliklerini karşılamak için egzoz arıtma sistemlerine gereksinim duyarlar. Bu sistemlerin önemli bir dezavantajı, düşük yüklerde düşük egzoz sıcaklıkları (T_{gaz} < 250°C) nedeniyle verimsiz performans göstermeleridir. Geç yakıt enjeksiyonu, egzoz gazı devridaimi ve erken egzoz valfi açıklığı gibi tipik motor teknikleri egzoz sıcaklığı iyileştirmesi için yakıt tüketimi artışı gerektirmektedir. Bu çalışma, egzoz sıcaklıklarının düşük yüklerde silindir devre dışı bırakılması (CDA) metodu ile yakıt tasarrufu sağlayan bir şekilde arttırılabileceğini bir kıvılcımlı-ateşlemeli motor üzerinde göstermektedir. Lotus Motor Simülasyonu (LES) yazılımı, motor modelinin oluşturulması ve 1500 RPM motor hızı ve bir dizi sabit motor torku üzerinde (5 Nm'den 25 Nm'ye) motor performans hesabı için kullanılmıştır. CDA modunda, aktif silindirlerin artan yakıt-hava oranı 110°C'a kadar egzoz sıcaklık artışına neden olmakta (T_{egzoz} > 250°C) ki bu da düşük yüklerde verimli egzoz arıtımını sağlamaktadır. Ayrıca, artan ısı transfer oranları ile egzoz arıtım katalizör yatağının ısınması hızlandırılmaktadır. CDA tekniği, azaltılmış hava endüksiyonu yoluyla motor pompalama kaybında önemli bir azalmaya sebep olmakta ve bu nedenle yakıt tasarrufu sağlamaktadır. Bununla birlikte, daha düşük hava endüksiyonu düşük egzoz akış oranına neden olmakta ve bu da bazı yüklerde arıtma sisteminin ısınmasını olumsuz etkilemektedir.
Keywords: Silindir devre dışı bırakma, kıvılcım-ateşlemeli motorlar, yakıt verimliliği, egzoz gazı sıcaklığı, egzoz arıtma yönetimi

SYMBOLS AND NOMENCLATURE

$\dot{m}$	Mass flow rate, kg/h
N	Engine speed, RPM
p	Pressure, bar
P_e	Brake power, kW
T	Temperature, °C
V_d	Cylinder displacement, m^3
Z	Cylinder number
η	Efficiency, %
θ	Burn angle, degree
θ_b	Total burn angle, degree
τ_e	Engine torque, Nm
ω	Angular speed, rad/s
Re	Reynolds number
ABDC	After bottom dead center
ATDC	After top dead center
BBDC	Before bottom dead center
BTDC	Before top dead center
BMEP	Brake mean effective pressure, bar

BSFC	Brake specific fuel consumption, g/kWh
BDC	Bottom dead center
CA	Crank angle, degree
CDA	Cylinder deactivation
CO	Carbon monoxide
CR	Compression ratio
EVC	Exhaust valve closure, degree
EVO	Exhaust valve opening, degree
FMEP	Friction mean effective pressure, bar
HCCI	Homogeneous charge compression ignition
IMEP	Indicated mean effective pressure, bar
IVC	Intake valve closure, degree
IVO	Intake valve opening, degree
LES	Lotus engine simulation
NO_x	Nitrogen oxide
PM	Particulate matter
PMEP	Pumping mean effective pressure, bar
PPCI	Partially premixed compression ignition
RCCI	Reactivity controlled compression ignition
RPM	Revolution per minute
SI	Spark-ignition
TDC	Top dead center
TWC	Three-way catalytic convertor
UHCs	Unburned hydrocarbons

INTRODUCTION

Modern on-road automotive vehicles generally utilize diesel or gasoline engines in transportation. Those engine systems have high thermal efficiency and are cost-effective in fuel economy. However, they also emit high rates of harmful criteria pollutants including nitrogen oxides (NO_x) and particulate matter (PM) into the atmosphere. Therefore, environmental authorities have issued some strict regulations to limit the amount of those emissions that automotive engines are allowed to release into the environment. For example, in the European Union (EU), Euro 6 emissions standards for passenger cars demand that emission rates of NO_x for diesel and gasoline engines should remain below 0.08 g/km and 0.06 g/km, respectively. Euro 6 also requires that PM emission rates on passenger cars using either diesel or gasoline engine should not exceed 0.005 g/km (Dieselnet, 2018). Those strict emission regulations force engine producers to explore new strategies to meet tailpipe emission limits. Recently, some innovative combustion methods such as homogeneous charge compression ignition (HCCI), reactivity controlled compression ignition (RCCI) and partially premixed compression ignition (PPCI) have been developed (Benajes et al., 2014; Benajes et al., 2015; Pipitone & Genchi, 2016) and renewable fuels have been found to be effective at reducing emission rates (Magno et al., 2015; Dubey & Gupta, 2017). Although those methods have a significant potential to curb emission rates, they cannot improve emission rates satisfactorily at some loads. Therefore, engine manufacturers not only use those aforementioned emission-decreasing methods, but also they generally outfit automotive engines with modern exhaust thermal management systems in order to meet stringent emission regulations.

Selective catalytic reduction (SCR), diesel oxidation catalyst (DOC) and diesel particulate filter (DPF) are typical exhaust after-treatment systems utilized on automotive engines for reducing tailpipe NO_x, unburned hydrocarbons (UHCs), carbon monoxide (CO) and PM emissions, respectively. Although those exhaust thermal management systems have a considerable potential to decrease emission rates and perform efficiently at many operating cases, they cannot operate effectively at low loaded conditions when exhaust gas temperatures remain below 250°C. Exhaust temperatures should be kept between 250°C and 400°C if those systems are desired to be operated with maximum efficiency (Stadlbauer et al., 2013; Song et al., 2013; Girard et al., 2009; Stanton, 2013). Engines utilized for inner-city transportation (stop and go) generally perform at light loads. Therefore, they spend a considerable amount of time with exhaust temperatures much lower than 250°C (Boriboonsomsin et al., 2018). Exhaust temperatures on those cases need to be improved for effective after-treatment.

There is a constant search for improving the exhaust thermal management systems (Joshi et al., 2006; Parks et al., 2007; Buono et al., 2012; Benaqqa et al., 2014; Palma et al., 2015; E et al., 2016). Some conventional methods such as late fuel injection, exhaust gas throttling and negative valve overlap focus on exhaust temperature rise for more effective after-treatment (Honardar et al., 2011; Gehrke et al., 2013). One recent strategy for increased exhaust temperatures is to apply early exhaust valve opening (EEVO). Roberts et al. (2015) explored the effect of EEVO on exhaust temperatures on a six-cylinder diesel engine at different engine speeds and loads. In the study, 30°C to 100°C exhaust temperature rise is obtained through EEVO which enables the after-treatment to be more effective in a greater part of the engine operation map in comparison to nominal case. Bharath et al. (2014) also examined EEVO on a multi-cylinder light duty engine. It is stated in the analysis that advancing exhaust valve opening timing 32° crank angle (CA) from the original condition results in sufficient exhaust temperature elevation for a DOC system with greater than 90 % conversion efficiency. Gosala et al. (2017) tried to improve after-treatment warm-up via EEVO and internal exhaust gas recirculation on a diesel engine at idle condition. Exhaust gas temperatures reached above 400°C, almost 100°C greater than conventional engine condition. The method also increases the heat transfer to the after-treatment catalyst bed and is more effective at catalyst warm-up in comparison to nominal engine case. Those aforementioned EEVO studies have a significant potential to improve exhaust after-treatment, however, they generally cause fuel consumption penalty (Piano et al., 2017). One other current method to rise exhaust temperatures is to reduce in-cylinder air-induction through early and late intake valve closure (IVC) (Garg et al., 2016; Guan et al., 2017). Basaran & Ozsoysal (2017) demonstrated on a diesel engine model that exhaust temperatures at a low-loading engine case can be elevated up to 60^{o}C via using variable IVC. The technique is found to be fuel-efficient due to reduced pumping losses as well. However, the study also states that variable IVC decreases exhaust flow rate which affects exhaust heat capacity negatively.

In this study, CDA technique - an alternative for variable IVC method - is applied on a SI engine to improve exhaust temperatures at low loads. Similar to variable IVC, CDA is an air-flow reducing technique as well. It decreases the total air inducted into the cylinders via cutting off the air flow on deactivated cylinders. However, unlike variable IVC, not all cylinders are active in CDA mode. For instance, in a four-cylinder gasoline engine, either internal (2-3) or external (1-4) cylinders are deactivated. There is no fuel injection, spark plug or combustion inside the deactivated cylinders (Millo et al., 2016). Intake and exhaust valves are all closed in inactive cylinders. Only active cylinders continue to produce power. Equivalence ratio of active cylinders is increased in CDA mode so as to keep engine load constant. Therefore, gasoline engine in this mode can still perform as powerful as it is in conventional mode.

CDA has been broadly investigated on gasoline engines and it is a proven method to reduce fuel consumption (Leone & Pozar, 2001; Douglas et al., 2005; Kuruppu et al., 2014; Millo et al., 2016). While those studies have considered the application and fuel-saving benefit of CDA, very few have examined the utilization of CDA as a way of rising exhaust gas temperatures. However, CDA has been implemented on diesel engines as a means of increasing exhaust temperatures and found to be highly successful (Magee, 2014). Some current studies demonstrate that it is a relatively new, practical and fuel-efficient method for after-treatment management (Zammit et al., 2014; Zammit et al., 2015; Lu et al., 2015; Gosala et al., 2017; Ramesh et al., 2017; Joshi et al., 2017). Therefore, t objective of this study demonstrate that CDA can also be applied on spark-ignition engines to improve exhaust temperatures and hence after-treatment management at low loads. Moreover, while aforementioned CDA-applications on diesel engines are all experimental, this study focuses on numerical analysis of CDA on gasoline engines via LES which can be beneficial considering the high costs of experiments.

In the study, gasoline engine operates at constant engine loads (taken as engine torque) during both conventional (all cylinders active) and CDA (only external cylinders active) modes. Engine load range is shown on the following Table 1. Exhaust temperatures at those engine loads mostly remain much lower than 250^{o}C. The fuel-saving CDA method is used to improve exhaust temperatures above 250^{o}C. At first, the properties of the gasoline engine and its built-in model are explicitly explained. Then, the model is validated in conventional mode at different engine speeds. On results & discussion section, effects of CDA on exhaust temperature, fuel consumption and after-treatment catalyst warm-up are evaluated. In conclusions part, the study is summarized and some future work for further analysis is suggested.

Table 1. Gasoline engine operation conditions in CDA mode.

Engine speed (RPM)	Engine torque (Nm) interval
1200	5 - 25

METHODOLOGY

Engine Model

In the study, a naturally-aspirated four-cylinder spark-ignition engine is used for CDA application. The main specifications of the engine are shown on Table 2 below. This gasoline engine has been examined experimentally in an earlier study (Geok et al., 2009). These actual engine properties are utilized on engine model and the experimental results are utilized for the model's validation.

Table 2. Gasoline engine specifications (Geok et al., 2009).

Model	Four-cylinder SI engine
Air intake	Naturally-aspirated
Bore (mm)	82
Stroke (mm)	75.5
Connecting rod length (mm)	131
Compression ratio	9.2:1
Engine rated maximum power (kW/RPM)	66/6000
Engine rated maximum torque (kW/RPM)	126/3000
EVO	57^o CA BBDC
EVC	15^o CA ATDC
IVO	15^o CA BTDC
IVC	53^o CA ABDC
Cylinder firing order	1-3-4-2

The simulation engine model is shown on Figure 1 below. It is based on one dimensional analysis by utilizing LES software (Lotus Engineering Software, 2013; Lotus Engine Simulation Manual, 2018). In order to represent an actual engine condition, the model begins with an intake system and ends up with an exhaust tailpipe system. Pipes, ports, valves and cylinders form the rest of the model operating between intake and exhaust systems. Gasoline engines generally utilize Three-Way Catalytic Convertor (TWC) systems to remove exhaust pollutants of NO_x, CO and UHCs. However, an after-treatment system similar to TWC is not modelled on the simulation. It is assumed that exhaust gases leaving "exit" on the model move directly to the exhaust after-treatment system (such as TWC) placed at the end of the exhaust tailpipe. This study primarily considers the temperature and flow rates of exhaust gases at after-treatment inlet since these two parameters directly affect both the efficiency and warm-up characteristics of after-treatment systems.

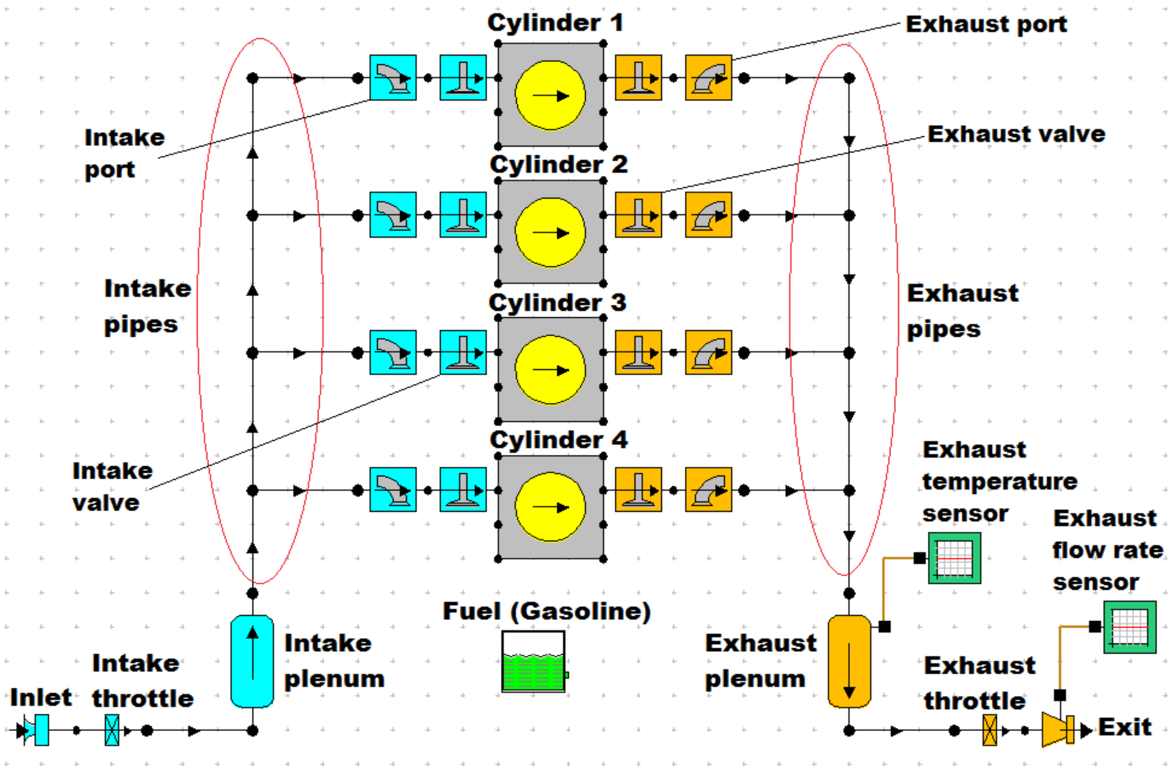

Figure 1. Engine model.

Two sensors, illustrated on Figure 1, are used to predict the exhaust temperature and exhaust flow rate at different loading cases both for conventional and CDA modes. The model also considers the cylinder firing order (1-3-4-2) stated on Table 2 which causes a 180° CA cylinder phase difference during one complete engine cycle.

Theoretical Analysis

LES utilizes one-dimensional model of pipe gas dynamics for the gas flow in pipes. Conservation equations for mass, momentum and energy are solved with two-step Lax-Wendroff (Richtmyer) method for calculating the conditions within the pipe elements (Winterbone & Pearson, 2000).

Engine performance parameters are calculated in the program with some generally known equations. Engine torque, τ_e (Nm), is found with (Heywood, 1988)

$$\tau_e = \left(10^3 \big/ \omega \right) \quad (1)$$

where P_e (kW) shows the engine brake power and ω (rad/s) represents the angular speed of the engine. P_e can be calculated with the following equation (Heywood, 1988)

$$P_e = \left(BMEPV_d NZ \big/ n_r 60 \right) \quad (2)$$

where BMEP (bar) and N (rpm) represent the brake mean effective pressure and engine speed, respectively. As it is widely known, four-stroke engines make two revolutions in a complete engine cycle, this is shown with . When the engine is two-stroke, it is taken as 1. Z in equation (2) demonstrates the total cylinder number. Finally, total displaced volume per cylinder is given with V_d. It is determined with (Heywood, 1988)

$$V_d = S(\pi B^2 / 4) \quad (3)$$

where S and B indicate stroke and cylinder bore of the gasoline engine.

BMEP is predicted on the simulation with the following formulation (Heywood, 1988)

$$BMEP = IMEP - FMEP \quad (4)$$

where IMEP (bar) and FMEP (bar) show the indicated mean effective pressure and friction mean effective pressure, respectively. FMEP is determined via using a modified version of H. B. Moss gasoline engine friction model (Barnes-Moss, 1975)

$$FMEP = 0.6 + \left(1.167 * 10^{-4} N\right) + \left(0.06 * \overline{U}_{piston}\right) \quad (5)$$

where $\overline{U}_{piston}$ (m/s) shows the mean piston speed.

Brake specific fuel consumption (BSFC) of the spark-ignition engine is calculated on engine model with the following mathematical relation (Heywood, 1988)

$$BSFC = \dot{m}_f / P_e \quad (6)$$

where $\dot{m}_f$ (g/h) is the total fuel mass flow rate required to keep engine loading constant at specified torque.

The simulation uses Wiebe function to define the mass burn rate inside the cylinders (Watson & Pilley, 1980). In the program, single Wiebe and two-part Wiebe modelling options are available for the definition of the combustion. While single Wiebe model offers a simple combustion model for engine research considering only the diffusion combustion phase, two-part Wiebe model is complicated. Unlike single part model, it also considers the premixed combustion phase and can model the engine heat release characteristics closer to actual engine combustion cases (Ghojel, 2010). Therefore, two-part Wiebe is chosen considering it can better represent a real engine combustion in the simulation. The formulations for premixed & diffusion combustion phases are given below

$$m_{premixed\ phase} = 1.0 - \left[1 - \left(\frac{\theta}{\theta_b}\right)^{Cp_1}\right]^{Cp_2} \tag{7}$$

$$m_{diffusion\ phase} = 1.0 - \exp^{-A\left(\frac{\theta-\Delta}{\theta_b-\Delta}\right)^{M+1}} \tag{8}$$

The parameters in equations (7) & (8) are defined appropriately during validation of the gasoline engine model with experimental results at different engine speeds. While Cp_1 and Cp_2 are taken as 500 and 2.0, A and M are selected as 10.0 and 1.5, respectively.

In the model, cylinder heat transfer is determined with semi-empirical convective heat transfer formulation proposed by Annand (Annand, 1963)

$$\frac{dQ}{A} = h\left(T_g - T_w\right) + c\left(T_g^4 - T_w^4\right) \tag{9}$$

where h is found with

$$h = a\frac{k\,\mathrm{Re}^b}{B_{cyl}} \tag{10}$$

The parameters a & b in equation (10) are defined as 0.1 and 0.7 during engine open cycle and 0.2 and 0.8 during engine closed cycle. The coefficient c in formulation (9) is only valid during closed cycle and taken as $4.29*10^{-9}$.

Validation of the Model

The engine model is built based on actual engine parameters via utilizing LES on the previous section. The simulation should be compared with experimental results (Geok et al., 2009) in order to prove its reliability. The validation of the engine simulation at conventional mode is illustrated on the following Figure 2 below.

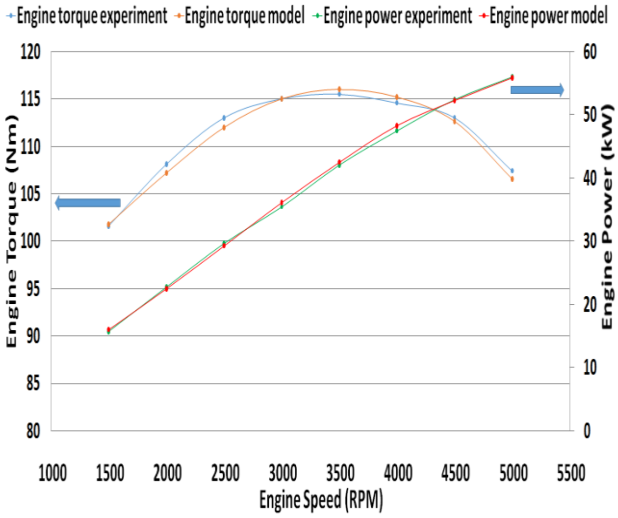

Figure 2. Comparison of model & experimental results in conventional mode.

The experimental and model results are compared between 1500 rpm and 5000 rpm engine speeds at nominal mode (CDA mode is not on). Engine torque and engine brake power predictions of the model are generally compatible with experimental results. The errors between simulation and test results are lower than 3% which can be assumed acceptable for a correlated model. In the study, engine load is kept constant at both conventional & CDA modes. Reduction on engine load is not desired when CDA is implemented to the engine system. Therefore, the compatibility of engine torque calculations with experimental results is particularly considered as significant for the model.

RESULTS AND DISCUSSION

Effects on Exhaust Gas Temperature

At first, the impact of CDA application on exhaust temperature is examined. In fact, this is the primary focus of this study. The comparison of exhaust gas temperatures at conventional & CDA modes is illustrated on Figure 3. Engine torque is maintained constant at both modes during the operation.

The operation conditions given on Table 1 are valid for exhaust temperature calculations on Figure 3. The 250ºC temperature line is particularly stated on the figure to emphasize that after-treatment works inefficiently below this line. It is seen that exhaust temperatures remain below 250ºC at most part of the operation in conventional mode.

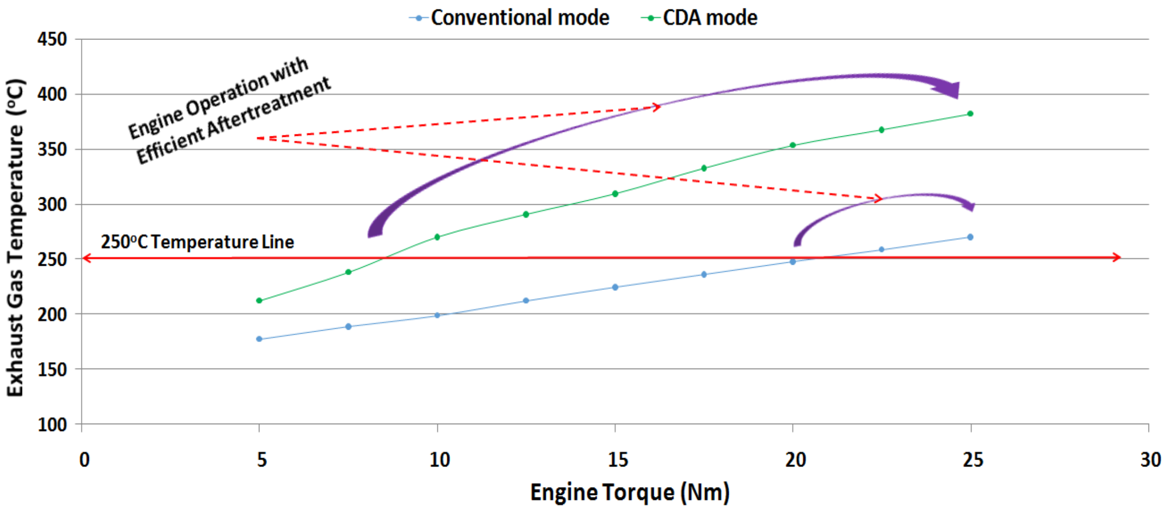

Figure 3. Exhaust gas temperature change at different loads in conventional & CDA modes.

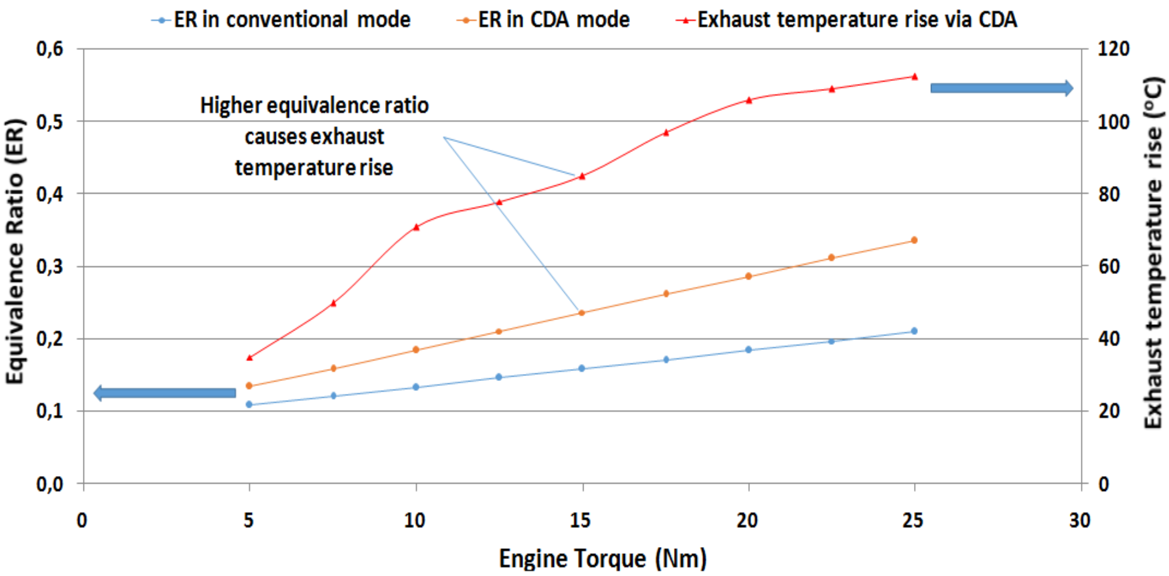

Figure 4. Equivalence ratio comparison at different loads in conventional & CDA modes.

Temperatures decrease even below 200°C when engine torque is lower than 10 Nm. Therefore, after-treatment can only operate effectively when engine torque is above 20 Nm. However, in CDA mode, exhaust temperatures are above 200°C at all loading cases and reach 250°C at almost 8.5 Nm engine torque which is much lower in comparison to conventional mode. Thus, a greater portion of the operating region has the potential to maintain efficient exhaust after-treatment in CDA mode. This positive effect can be very useful particularly for automotive vehicles which spend most of their time at low loads during inner-city transportation.

It is also shown on Figure 3 that exhaust temperature improvement via CDA is low at light loads (almost 35°C at 5 Nm), however, it is significant at higher loads (around 110°C at 25 Nm). The reason behind the temperature rise variation at different loads is directly related to the equivalence ratio variations of conventional & CDA modes shown on Figure 4 above.

At low loads, minor adjustments on equivalence ratio is sufficient to manage engine torque constant. Therefore, low fuel consumption increments on active cylinders in CDA mode result in low exhaust temperature rise. However, at higher loads, active cylinders require more fuel consumption to hold engine torque constant even though total air induction into the cylinders does not rise in a significant manner. Therefore, equivalence ratio at CDA mode rises faster than it does at conventional mode and higher exhaust temperature rise is achieved at high engine loading conditions.

Effects on Fuel Consumption

CDA technique seems to be highly useful for after-treatment efficiency improvement through substantially increased exhaust gas temperatures. However, it is also important to examine the method's effect on engine fuel economy. Therefore, fuel consumption behaviour of CDA mode at different loads is explored in this section.

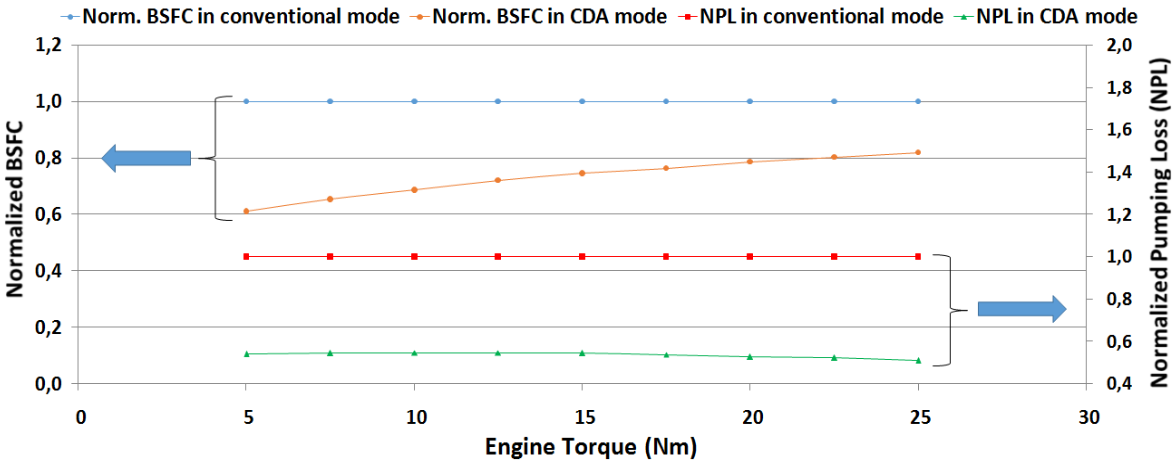

Figure 5. Normalized BSFC and normalized pumping loss at different loads for conventional & CDA modes.

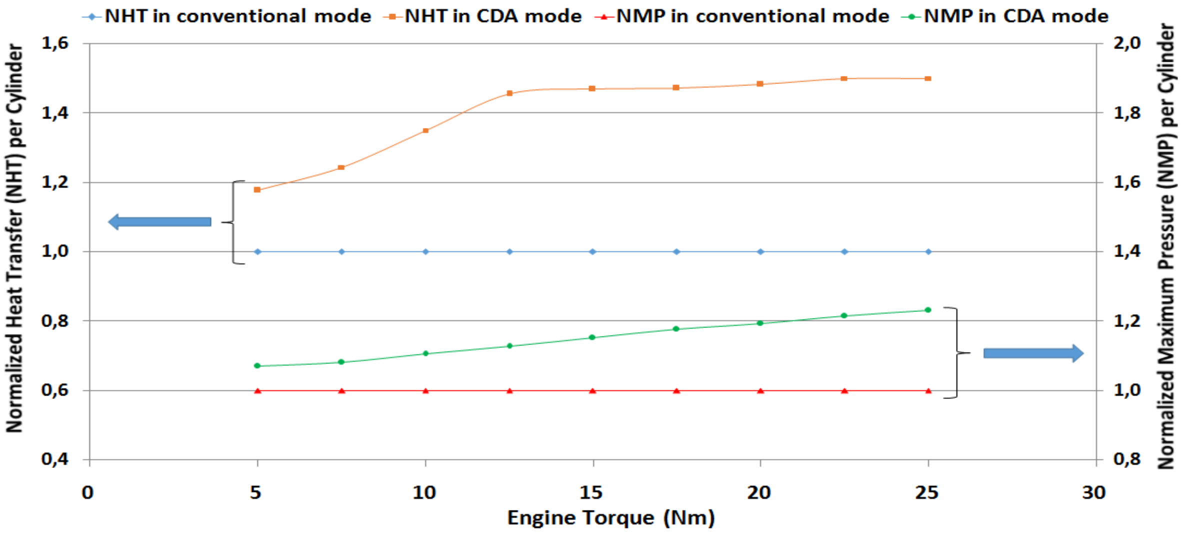

Figure 6. Normalized cylinder heat transfer & maximum pressure at different loads for conventional & CDA modes.

BSFC and pumping loss at CDA & conventional modes at different loads is compared on Figure 5 above. At each load, all predictions for BSFC are divided by the prediction obtained at conventional mode. Therefore, all calculations are normalized. The same normalization is valid for pumping loss predictions too. Figure 5 demonstrates that CDA mode requires lower fuel consumption at all loading cases. In CDA operation, only two cylinders are active. On other cylinders, intake & exhaust valves are switched off and air flow is cut off. Thus, air can only be inducted into operating cylinders. Unlike CDA mode, nominal engine uses all four cylinders during operation and air induction is available for all cylinders. Therefore, pumping losses are much higher in conventional operation in comparison to CDA mode. CDA results in a significant reduction on pumping losses at all loads. In fact, that is why it needs lower fuel injection to keep engine load constant.

While fuel efficiency improvement through CDA is considerably high at low loads, the method's fuel-saving effect starts to fade as engine loading rises. Even though a similar amount of pumping loss reduction is achieved at almost all loads, gasoline engine at high loads is not as fuel-efficient as it is at light loads. This is due to the rise of cylinder heat transfer and cylinder maximum pressure at high loads in CDA mode as shown on Figure 6. Similar to Figure 5, all calculations at each engine torque for cylinder heat transfer are divided by the calculation obtained at conventional mode. Thus, all predictions are normalized. The same normalization is valid for cylinder maximum pressure too.

Active cylinders in CDA mode need to acquire higher equivalence ratios than cylinders in conventional operation as illustrated previously on Figure 4. In standard operation, all four cylinders are available to produce the required torque, whereas gasoline engine has only two operating cylinders to provide the same amount of torque in CDA mode. Therefore, fuel consumption per active cylinder rises sharply as the engine starts to perform at high engine torques. Fuel injection rise in active cylinders results in higher maximum pressures inside the cylinders. Higher pressures lead to higher in-cylinder temperatures and the temperature difference between the cylinder wall and the environment increases. Therefore, as seen on Figure 6, there is much higher cylinder heat transfer loss at high loads in CDA mode which results in low improvement on fuel-efficiency (Figure 5). It can be derived that CDA will be no more fuel-saving after a certain engine load where pumping loss improvement cannot compensate the increased heat transfer loss.

Effects on After-treatment Catalyst Warm-up Characteristics

On the previous two sections, CDA proved to be an effective method to keep after-treatment catalyst at high temperatures at low loads in a fuel-efficient manner. However, efforts on those sections mainly focus on improvement of exhaust temperature and fuel consumption and does not consider how fast CDA technique warms up the after-treatment catalyst bed. This section intends to find out the impact of CDA on catalyst warm-up and compare it with conventional engine condition.

After-treatment catalyst warm-up depends on mainly the heat transfer rate from the exhaust gases to the catalyst substrates. Heat transfer rate to the after-treatment catalyst bed is affected by three fundamental characteristics: exhaust temperature, exhaust flow rate and catalyst temperature. The effect of those characteristics on heat transfer rate can be examined in an approximate manner by utilizing the formula below (Ding et al., 2017)

$$Q = C \times \dot{m}_{exhaust}^{4/5} \times \left(T_{exhaust} - T_{catalyst}\right) \tag{11}$$

In equation (11), Q represents the heat transfer rate, $T_{exhaust}$ is the exhaust gas temperature flowing directly to the after-treatment catalyst bed, $T_{catalyst}$ shows the after-treatment catalyst temperature, $\dot{m}_{exhaust}$ is the exhaust mass flow rate discharged from the gasoline engine and C is a constant which is dependent on the material and geometry of the after-treatment system.

In this model, for any $T_{catalyst}$, heat transfer rate from the exhaust gases to the catalyst bed can be calculated easily by using the formula (11) above. At each loading case, $\dot{m}_{exhaust}$ and $T_{exhaust}$ can be assumed constant as the engine performs in a steady-state manner at constant engine torque. In the study, the heat transfer rates are predicted for both CDA & conventional modes at engine torques of 5 Nm, 10 Nm and 15 Nm, respectively. Those are the loading cases where exhaust temperatures remain much lower than 250°C and high heat transfer rates to the catalyst bed are needed for fast after-treatment warm-up. At every aforementioned engine torque case, all heat transfer calculations in conventional & CDA modes are divided by the heat transfer prediction of conventional mode at 0°C catalyst temperature. Therefore, heat transfer predictions at each load are all normalized.

Normalized heat transfer rates for CDA and nominal engine operation cases are demonstrated at 5 Nm engine torque on Figure 7. As seen, heat transfer predictions go to zero for both methods when $T_{exhaust}$ is equal to $T_{catalyst}$. The effect is positive (warming) on normalized heat transfer lines until that point. However, when the $T_{catalyst}$ is higher than $T_{exhaust}$, there is negative heat transfer (cooling) on the after-treatment catalyst bed. It can be derived from Figure 7 that conventional engine operation is more effective than CDA until $T_{catalyst}$ reaches 125°C. This is because CDA results in a considerable reduction on exhaust mass flow rate which is highly effective at calculating Q in equation (11). Therefore, although CDA increases exhaust temperature above 200°C, its heat transfer potential is lower compared to conventional mode when $T_{catalyst}$ remains below 125°C. However, at higher catalyst temperatures, CDA mode starts to be more effective. There is negative heat transfer in conventional mode as catalyst temperature exceeds 175°C, whereas the same negativity in CDA mode begins at a higher temperature ($T_{catalyst}$ = 210°C). Also, the effect of negative heat transfer on after-treatment catalyst in CDA mode is much lower at higher catalyst temperatures due to decreased exhaust flow rates. At 5 Nm engine torque, it is better to operate the gasoline engine in conventional mode until catalyst bed temperature reaches 125°C. At higher catalyst temperatures, CDA mode can be preferred due to not only its higher heat transfer potential, but also its improved engine fuel-efficiency.

Figure 8 demonstrates the catalyst warm-up characteristics of conventional & CDA modes at 10 Nm engine torque. CDA method seems to be more effective on after-treatment warm-up as the engine load rises from 5 Nm to 10 Nm. This is because it results in higher exhaust temperature rise at higher engine loads and thus, it improves the heat transfer rate to the catalyst bed relatively more than it does at 5 Nm engine torque. However, it is still less effective than conventional mode when catalyst temperature remains below 50°C. It can be derived that although CDA leads to 70°C exhaust temperature rise at this engine load, it is still insufficient to reach the heat transfer potential of conventional mode at low catalyst temperatures (lower than 50°C). That is due to the dramatic reduction on exhaust mass flow rate which affects the heat transfer prediction in equation (11) negatively. Conventional mode can be preferred at low temperatures due to its relatively higher heat transfer rates. At particularly higher than 50°C catalyst temperatures, CDA is definitely more beneficial than conventional engine operation and it can accelerate the after-treatment warm-up much better than conventional mode. Also, it is still fuel-saving at this load and negative heat transfer rates are highly improved.

Finally, effects of both CDA & conventional modes on after-treatment catalyst bed warm-up are compared at 15 Nm engine torque on Figure 9. At this engine load, there is a considerable exhaust temperature rise (almost 85°C) via CDA. Exhaust temperature can be raised above 300°C (high efficient after-treatment) through CDA, whereas it is still below 250°C (inefficient after-treatment) in conventional mode.

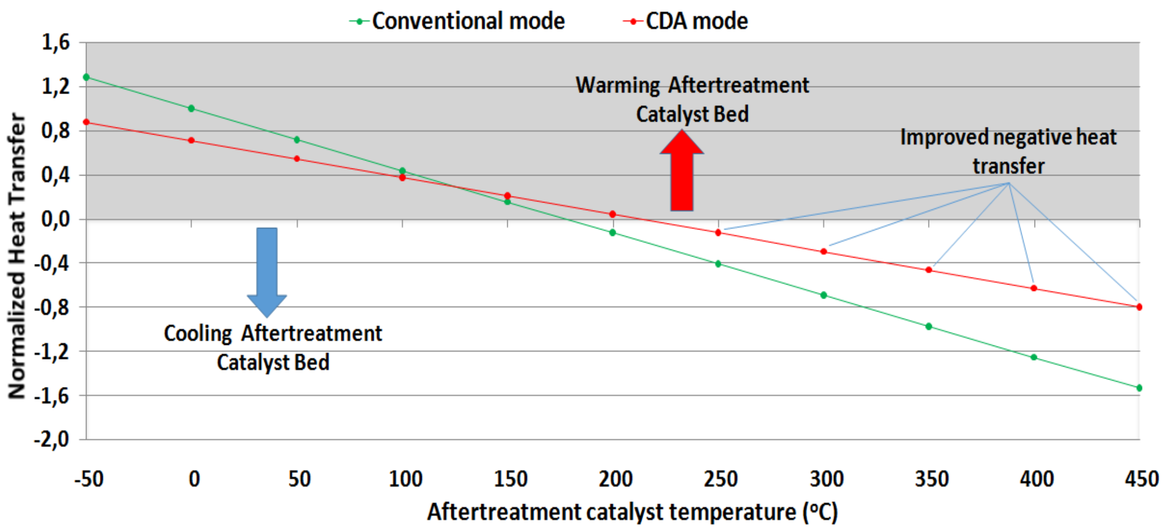

Figure 7. Effects of conventional & CDA mode operation on after-treatment catalyst warm-up at 5 Nm engine torque.

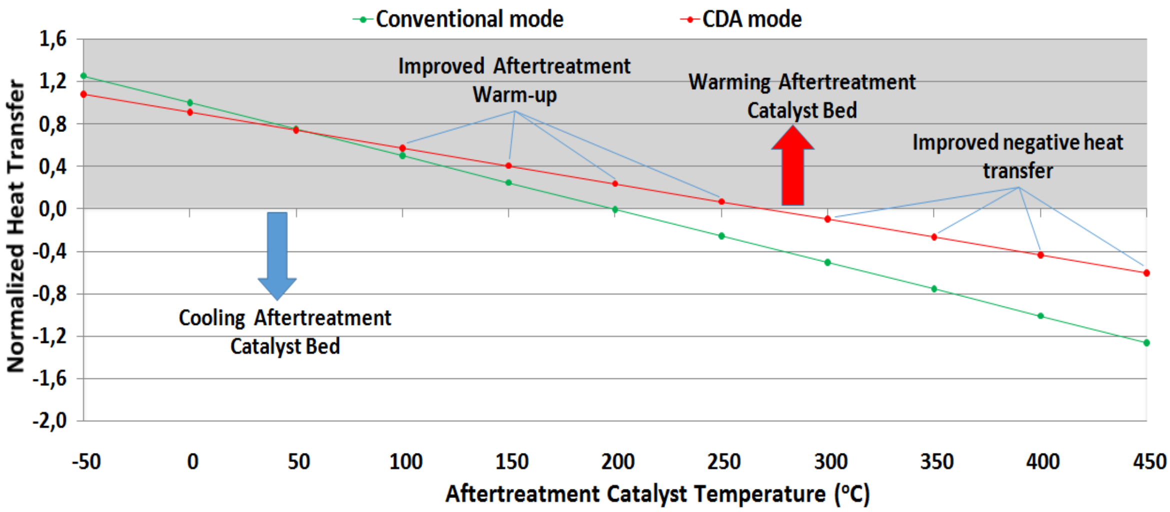

Figure 8. Effects of conventional & CDA mode operation on after-treatment catalyst warm-up at 10 Nm engine torque.

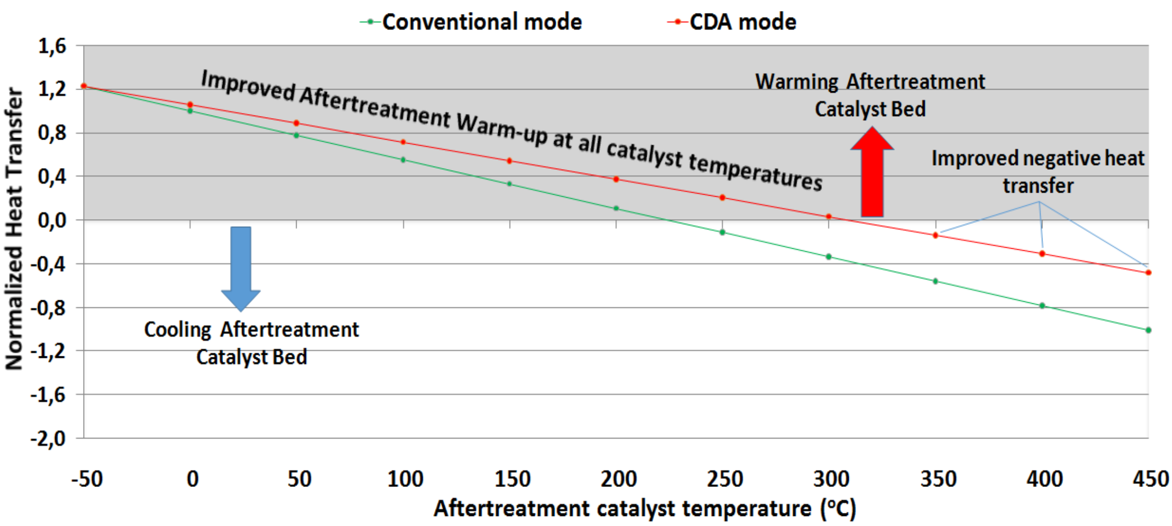

Figure 9. Effects of conventional & CDA mode operation on after-treatment catalyst warm-up at 15 Nm engine torque.

Significant exhaust temperature improvement on Figure 9 enables CDA mode to be more effective than conventional mode at after-treatment warm-up in all catalyst temperatures. In CDA mode, there is still a dramatic reduction on exhaust flow rate. However, exhaust temperature elevates sufficiently to compensate the heat transfer loss due to decreased exhaust flow rate. Therefore, it has more heat transfer potential to after-treatment and can warm up the catalyst bed much faster than conventional mode. The improved fuel economy and enhanced negative heat transfer rates are other advantages of using CDA for after-treatment warm-up at this engine load.

CONCLUSIONS

This study intends to demonstrate that effective exhaust after-treatment management on a spark-ignition engine can be achieved in a fuel-saving manner via utilizing "CDA" technique. It is found in the analysis that the method is highly effective on exhaust temperature rise which is beneficial for after-treatment improvement. Conventional exhaust temperatures are increased up to 110ºC through CDA. Exhaust temperatures at low loads are mostly above 250ºC in CDA mode and thus, after-treatment system can be operated with high efficiency. However, in conventional mode, exhaust temperatures generally remain below 250ºC which results in low-efficient exhaust after-treatment.

CDA is also found to be a fuel-efficient technique. It is seen that decreased air induction due to disabled cylinders results in a significant reduction on engine pumping loss. Therefore, gasoline engine needs lower fuel injection to keep engine load constant. The improvement on fuel consumption is mostly effective at low loads (higher than % 30). At relatively higher loads, CDA needs much higher equivalence ratios in active cylinders to maintain engine torque constant. That results in higher in-cylinder temperatures and hence higher cylinder heat loss to the environment. Therefore, the improvement reduces to lower than % 20 at high loads. It can be predicted from the BSFC trend line of CDA mode that the fuel-saving effect will decrease down to zero as the load is sufficiently increased.

Utilizing CDA at low loads is beneficial for improving after-treatment catalyst warm-up as well. The effect of CDA mode is examined for three different engine loading cases (constant engine torque at 5 Nm, 10 Nm and 15 Nm). It is found to be most effective at 15 Nm. At this load, CDA mode has more heat transfer potential to the catalyst bed in comparison to conventional mode at all catalyst temperatures. Therefore, after-treatment system can be warmed up much faster through CDA at 15 Nm. However, at 5 Nm and 10 Nm cases, CDA technique is not that effective at all catalyst temperatures. At low catalyst temperatures, conventional mode is more effective than CDA mode (below 125ºC at 5 Nm and below 50ºC at 10 Nm). This is because exhaust temperature rise is relatively low at these loads and reduced exhaust flow rates have a dominant role on heat transfer predictions. However, this is only valid when catalyst temperatures are too low. Thus, at higher catalyst temperatures, CDA mode can still be preferred over conventional mode due to not only its higher heat transfer rates, but also its improved fuel economy.

Future studies can examine the combined effect of CDA and other on-engine techniques such as EEVO, variable IVC and late fuel injection on exhaust after-treatment improvement. Those advanced analysis can reduce the after-treatment warm-up time through increased heat transfer rates. Optimum combinations at different loads can be determined considering both exhaust temperature improvement and fuel penalty rise.

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