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RESEARCH ARTICLE / ARAŞTIRMA MAKALESI

Combining Early Intake Valve Closure and Exhaust Throttling to Achieve Rapid Exhaust After-treatment Warm up in Diesel Engine Systems

Dizel Motor Sistemlerinde Hızlı Egzoz Son-işlem Isınması Sağlamak için Erken Emme Valfi Kapatma ve Egzoz Kısma İşleminin Birleştirilmesi

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Abstract

Nowadays, the thermal management of exhaust after-treatment (EAT) units is a paramount concern for diesel automotive vehicles to meet the stringent emission regulations. In general, EAT temperatures above 250^{o}C are favorable for effective emission conversion efficiency. At low-loaded operations, it is difficult to achieve that since exhaust temperature remains much below 250^{o}C. Therefore, this numerical work aims to elevate exhaust temperature at a light-loaded diesel engine model through adopting two different engine-base techniques, namely early intake valve closure (EIVC) and exhaust throttling (ET). Both individual and combined modes of EIVC and ET are examined for high exhaust temperatures in the system. ET enhances exhaust temperature over 250^{o}C with high exhaust flow rate, which is desirable for rapid EAT warm up. However, it causes up to % 15 fuel penalty, which highly impairs its practicality. Unlike ET, EIVC is thermally efficient and can raise exhaust temperature above 250^{o}C. Yet, it has the disadvantage of significantly lowered exhaust flow rates, which is inconsistent with fast EAT warm up. Simultaneous application of ET and EIVC, as EIVC+ET, can still keep exhaust temperature above 250^{o}C with reduced fuel penalty (down to % 8.8). It also has the benefit of increased exhaust flow rates compared to EIVC mode, which substantially heightens heat transfer rates to the EAT unit (up to % 101). Thus, it can sustain accelerated EAT warm up in the system. EIVC+ET method is also seen to be effective to improve EAT stay-warm performance (delaying EAT cool off) as it enables high exhaust temperature and high exhaust rates, which is not possible with other methods examined in the analysis.

Keywords: *Diesel Engines, Early Intake Valce Closure, Exhaust Throttling, Exhaust Temperature, After-treatment Thermal Management*

Öz

Günümüzde, egzoz son işlem (ESİ) ünitelerinin ısıl yönetimi, dizel otomotiv araçlarının sıkı emisyon düzenlemelerini karşılaması açısından büyük önem taşımaktadır. Genel olarak, etkili emisyon dönüşüm verimliliği için 250^{o}C'nin üzerindeki ESİ sıcaklıkları uygun olmaktadır. Düşük yüklü operasyonlarda egzoz sıcaklığı 250^{o}C'nin çok altında kaldığı için bunu başarmak güçleşmektedir. Bu nedenle, bu sayısal çalışma, erken emme valfi kapatma (EEVK) ve egzoz kısılması (EK) olmak üzere iki farklı motora bağlı tekniği kullanarak düşük yüklü bir dizel motor modelinde egzoz sıcaklığını yükseltmeyi amaçlamaktadır. Sistemde yüksek egzoz sıcaklığı elde etmek için EEVK ve EK'nin hem tekil hem de birleşik modları incelenmiştir. EK, hızlı ESİ ısınması için ihtiyaç duyulan yüksek egzoz akış hızıyla egzoz sıcaklığını 250^{o}C'nin üzerine çıkarmaktadır. Ancak bu metot % 15'e varan yakıt tüketimi artışına neden olmakta ve bu da pratikte uygulanabilmesini oldukça zorlaştırmaktadır. EK'den farklı olarak, EEVK termal verimliliği iyileştirmekte ve egzoz sıcaklığını 250^{o}C'nin üzerine çıkarabilmektedir. Ancak ESİ ünitesinin hızlı ısınmasını aksatan, ciddi ölçüde düşürülmüş egzoz akışı hızı gibi bir dezavantajı bulunmaktadır. EK ve EEVK'nin EK+EEVK olarak eş zamanlı uygulanması, azaltılmış yakıt tüketimi artışı (% 8.8'e kadar) ile egzoz sıcaklığını hala 250^{o}C'nin üzerinde tutabilmektedir. Ayrıca, EEVK moduna kıyasla daha yüksek egzoz akış hızı avantajına da sahiptir ki, bu da ESİ ünitesine olan ısı transfer oranlarını önemli ölçüde (% 101'e kadar) arttırmaktadır. Bu nedenle, ESİ ünitesinin motor sisteminde çok daha hızlı ısınmasına sağlayabilmektedir. EEVK+EK yönteminin, analizde incelenen diğer yöntemlerle mümkün olmayan, yüksek egzoz sıcaklığına ve yüksek egzoz akış hızlarına olanak verdiği için ESİ ünitesi sıcak kalma performansını (ESİ ünitesi soğumasının geciktirilmesi) iyileştirmede de etkili olduğu görülmüştür.

Anahtar Kelimeler: *Dizel Motorlar, Erken Emme Valfi Kapanması, Egzoz Kısılması, Egzoz Sıcaklığı, Son-işlem Isıl İdaresi*

1. Introduction

Strict emission legislations and the demand for enhanced fuel efficiency require diesel engine producers to continually develop new and advanced technologies [1,2]. One effective strategy is to maintain low temperature combustion through advanced combustion techniques [3,4]. Alternative, non-diesel fuels are also highly examined to curb emission rates in diesel vehicles [5-7]. Another reliable solution is to place EAT units (SCR, DOC and DPF) on the downstream of engine exhaust systems [8]. Since aforementioned advanced approaches cannot always satisfy the required emission reduction, manufacturers prefer to utilize both those techniques and EAT units to sustain tailpipe emission rates below certain limits [9].

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One drawback with EAT units is that they mostly need a temperature window to complete the emission-cleaning process. For instance, SCR in most cases requires temperatures above 250°C for an effective NOx conversion [10,11]. Also, the light-off temperature of DOC remains above 200°C [12], which cannot be achieved at low-loaded cruise operations of diesel vehicles. Thus, thermal management of EAT units is a significant concern for researchers to maintain the reduced emission rates in diesel engine systems [13].

ETM-improving methods are normally classified into two main groups as air-based and fuel-based techniques [14]. An effective air-based method is the actuation of the closure timing of intake valve [15,16]. Early intake valve closure (EIVC) is seen to lead to a noticeable rise in exhaust temperature, which enables more efficient SCR at low loads [17]. EIVC achieves enhanced ETM with lowered pumping loss and increased fuel efficiency, which is preferable for ETM improvement. Similar to EIVC, intake throttling elevates engine-out temperature as well [18,19]. However, unlike EIVC, pumping loss is increased and fuel consumption is deteriorated via intake throttling method, which makes it a less favorable inner-engine technique [20]. In addition to intake throttling, exhaust throttling (ET) is examined to boost exhaust temperature [21]. ET leads to higher pumping loss and fuel consumption, but it rises exhaust temperature more than intake throttling method [22]. In contrast to air-based approaches, fuel-based methods consider fuel injection timing, fuel injection amount, post-fuel injection and timing in diesel engines [23,24]. Noticeable improvement can be obtained via particularly close and late post fuel injection [25-27]. However, modulating the injection characteristics of fuel generally results in undesirable fuel inefficiency [28,29]. Thus, currently, researchers try to couple air-based and fuel-based techniques to particularly improve the fuel efficiency while EAT heat up is accelerated through increased exhaust temperature [30-32].

As the literature review demonstrates, there is the tendency of application of multiple engine-dependent methods to improve the trade-off between fuel consumption and EAT warm up process [33]. Therefore, this numerical study attempts to combine two effective inner-engine techniques (EIVC and ET) to improve EAT heat up in a diesel engine system. Coupled application of ET and EIVC has the advantage of achieving exhaust temperature above 250°C with low fuel penalty and high exhaust flow rate, which is helpful to improve EAT warm up without causing a significant fuel inefficiency in the system.

2. Materials and Methods

In the analysis, a six-cylinder heavy-duty (HD) diesel engine model is used to observe the impact of ET, EIVC and EIVC+ET combined techniques o0n EAT heat up improvement.

2.1. Engine model

Figure 1 displays the detailed engine model used in this work. Lotus Engine Simulation (LES) program is utilized to build the model [34,35]. The system works with 6 consecutive cylinders, with a firing order of 1-5-3-6-2-4. Those types of HD compression-ignition engines are commonly preferred in commercial trucks or public buses due to the potential to produce high power in a reliable and fuel efficient manner.

A throttle valve is placed at the downstream of exhaust ports with the aim to alter the performance characteristics of the system, in particular exhaust temperature. Two sensors (EFR and TDT sensors) are inserted into the model to quantify the variation of exhaust flow rate (EFR) and temperature at downstream of turbine (TDT). Unless an external device such as an electrical heater or a burner is equipped, exhaust heat is the main source to warm up the EAT unit. Thus, the change of EFR and TDT has an utmost role as fast EAT warm up is desired in an engine system.

The properties for the diesel engine shown in Figure 1 are given in Table 1.

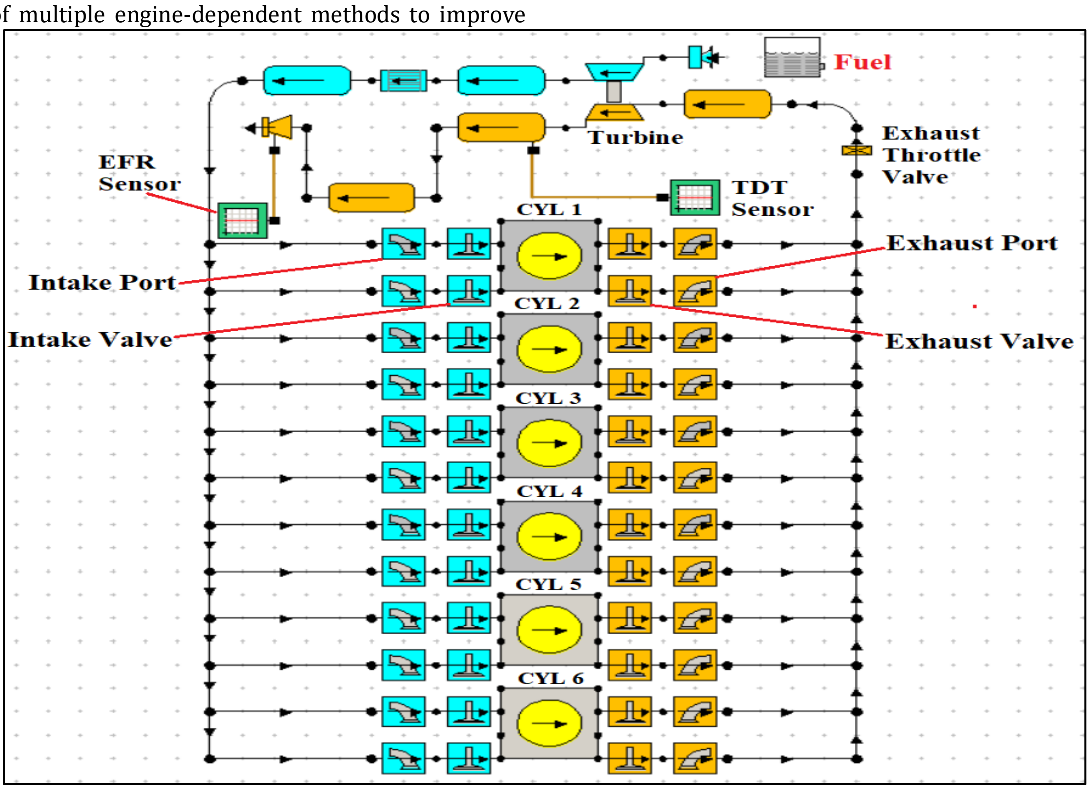

Figure 1. Engine model.

Table 1. The engine specifications.

Engine Parameter	Value
Stroke (mm)	124
Bore (mm)	107
Connecting rod length (mm)	192
Compression ratio	17.3:1
Start of injection (SOI)	3°CA BTDC
Cylinder firing order	1-5-3-6-2-4
Maximum Intake Valve Lift (mm)	8.5
Maximum E Valve Lift (mm)	10.0
Intake Valve Closure	25°CA ABDC
Intake Valve Opening	20°CA BTDC
Exhaust Valve Closure	20°CA ATDC
Exhaust Valve Opening	20°CA BBDC

Different modes result in different performance characteristics of the engine system. The change of those characteristics needs to be examined to observe whether the desirable condition (low fuel penalty, high volumetric efficiency, high exhaust temperature etc.) is achieved in the system. An important engine characteristic is the engine brake power (Pe). It is calculated in the model with [36]:

$$P_e = \left(BMEPV_dNZ/_{n_r60}\right) \tag{1}$$

where Z shows the cylinder number, N represents the engine speed, and V_d is the displaced cylinder volume. The number of revolutions per cycle, n_r, is taken as 2.0 in equation (1). Brake mean effective pressure, BMEP, represents the loading of the engine. It is held constant at 2.5 bar at all modes in the study. BMEP is found with [36]:

$$\mathrm{BMEP} = (IMEP_{nominal} - FMEP) \tag{2}$$

where FMEP, friction mean effective pressure, is the friction loss in the system [37]. $IMEP_{nominal}$ in equation (2), as nominal indicated mean effective pressure, is calculated with [38]:

$$IMEP_{nominal} = IMEP_{gross} + PMEP \tag{3}$$

where $IMEP_{gross}$, gross indicated mean effecture pressure, shows the power generating capacity of the system considering only the closed cycle of the engine. Unlike $IMEP_{gross}$, PMEP is related to the open cycle of the engine and shows the pumping loss of the system. Those two characteristics are determined with the following equations [38], respectively:

$$IMEP_{gross} = \int \left(PdV/_{V_d}\right)_{closed\ cycle} \tag{4}$$

$$PMEP = \int \left(PdV/_{V_d}\right)_{open\ cycle} \tag{5}$$

Another significant parameter is the brake specific fuel consumption (BSFC) of the engine. It is calculated with [36]:

$$BSFC = \dot{m}_f/_{P_e} \tag{6}$$

where P_e is the brake power (kW) calculated in equation (1) and $\dot{m}_f$ denotes the flow rate of the fuel (g/h) into the cylinders. Other than BSFC, volumetric efficiency, η_{vol}, is found with [36]:

$$\eta_{vol} = \left(2\dot{m}_{ia}10^3/_{30V_dN\rho_{ia}}\right) \tag{7}$$

where $\dot{m}_{ia}$ denotes the fresh air flow rate (g/h) and ρ_{ia} is the density of the air. The comparison of the η_{vol} at different modes is significant in order to evaluate the change of exhaust flow rate and thus, the potential for fast EAT heat up.

2.2. Methodology

This numerical work is motivated by a previous work, which indicates the benefit of increased exhaust temperature by reasonable control of IVC timing [17]. In that former study, the engine model was explicitly explained and validated with the experimental results (exhaust temperatures at different IVC timings) of a research in the literature [15]. Similar to Ref. [15] & [17], the system in this present work is assumed to work at 1200 RPM engine speed and 2.5 bar BMEP, where exhaust temperature is too low (<< 250°C) for effective EAT operation. However, unlike that former study, the analysis at hand considers both temperature and flow rate at turbine outlet (with EFR and TDT sensors) to quantify the capacity of the system to warm up the EAT unit. Therefore, heat transfer rates to the EAT component are calculated with the following equation to achieve that evaluation [39]:

$$\dot{Q} = \dot{m}_{exhaust}^{4/5} C(T_{turbine\ out} - T_{EAT}) \tag{8}$$

T_{EAT} denotes the temperature of the after-treatment system in equation (8). Although an EAT unit is not placed in Figure 1, exhaust gas leaving the turbine is known to flow instantly through the after-treatment mechanism. Thus, it can heat EAT unit above 250°C as long as the exhaust flow continues in the system. Temperature at turbine exit is represented with $T_{turbine\ out}$ and $\dot{m}_{exhaust}$ shows the exhaust mass flow rate at EAT inlet. C is a parameter for the type of the after-treatment device. Since the transfer rates are normalized considering the nominal mode in the analysis, C cancels out and does not play a role for the evaluation of normalized $\dot{Q}$ values.

In the analysis, elevated exhaust temperature is aimed to achieve through the control of nominal IVC timing (25°CA ABDC) stated in Table 1 and also the actuation of the opening (%) of exhaust throttle valve illustrated in Figure 1. EIVC, which was experimentally demonstrated to be applicable in diesel engines in previous works [40-42], is preferred as a realistic technique in the model at hand. Those earlier works examined the EIVC mechanically through modulating the shape and positioning of the camshaft lobes in the system [43]. The proper modification of the camshaft lobes facilitates the actuation of the lift and the closure timing of intake valves [44].

Figure 2 demonstrates the change of lifts for intake valves in different EIVC modes. Intake valves, in those modes, still open at 20°CA BTDC, which is exactly the same with the nominal mode. However, they begin to close earlier than nominal mode, which actually shortens the total opening duration in EIVC modes. Since the maximum valve lift is held constant at 8.5 mm in all modes, it is seen in Figure 2 that valve lift profile increases and decreases in a sharper manner in EIVC operations. As IVC timing is advanced, the rising and the reducing parts of the valve lift form can be adjusted in LES program to fit the whole profile with the same maximum lift (8.5 mm) in this new mode. The valve lift points are advanced proportionately based on the advancement of the IVC timing. Overall, the profile as a whole tends to approach top dead center (TDC) in EIVC modes. This is in contrast to the nominal mode, which tends to lean towards bottom dead center (BDC). As the analysis concentrates on IVC control, exhaust valve lift form is not altered in Figure 2.

In nominal mode, exhaust throttle valve, which is taken as a butterfly valve, is positioned in Figure 1 at the downstream of the exhaust ports and is assumed to be completely open. In other words, it is % 100 open. As in Table 2, the opening of the valve is reduced to lower levels in ET mode. The throttle valve is no more fully open, as it is in nominal mode, but is maintained as partially open in ET mode. Exhaust throttle opening (ETO), defined as the main parameter in ET mode in Table 2, explicitly demonstrates the reduction in valve opening. The valve area for the exhaust flow is restricted to smaller ETO values as further steps are applied in the system (% 60 at step 4 and % 30 at step 9). In the last step (step 12), where TDT is ensured to exceed 250°C, the valve opening in ET mode is reduced to as low as % 26.5. With 12 steps of reduction, exhaust throttle valve is closed by approximately three quarters in the system.

A similar contraction is seen in EIVC mode for the intake valves in Table 2. IVC timing, defined as the main parameter in EIVC mode, is modified as earlier than the base timing with different steps. In line with Table 2, nominal IVC timing, which explicitly extends beyond BDC in Figure 2, starts to move backwards in EIVC modes in the same plot. IVC timing in Table 2 is swept backwards by 70°CA with nine steps in EIVC mode. It is closed at 5°CA before BDC at step 3 and at 30°CA before BDC at step 6. The higher the step is kept, the earlier the IVC timing is controlled in the system. In extreme EIVC mode, IVC timing is controlled at 45°CA BBDC, which considerably shortens the duration for fresh inlet charge for the cylinders.

In combined mode, both intake valves and exhaust throttle valve are simultaneously modulated. Advanced IVC timings in Table 2 are combined with reduced ETO cases in EIVC+ET mode. Similar to EIVC and ET modes, ETO and IVC timing are the main parameters to consider in combined mode as well. Considering the noticeable decrease in exhaust rate due to aggressive EIVC seen in previous works [15,17,45], extreme use of it is avoided in combined mode. Unlike EIVC mode, IVC is moderately advanced (up to 10°CA BBDC) in EIVC+ET mode to limit the reduction in air charge and thus, in exhaust flow rate. It is not as aggressively applied as it is in EIVC mode (up to 45°CA BBDC), which can be technically less challenging. ETO needs to be constrained down to % 28 with 8 steps in EIVC+ET combined mode in order to maintain TDT above 250°C. Overall, all parameter modulations in this work intend to achieve high TDT with different physical means in the system.

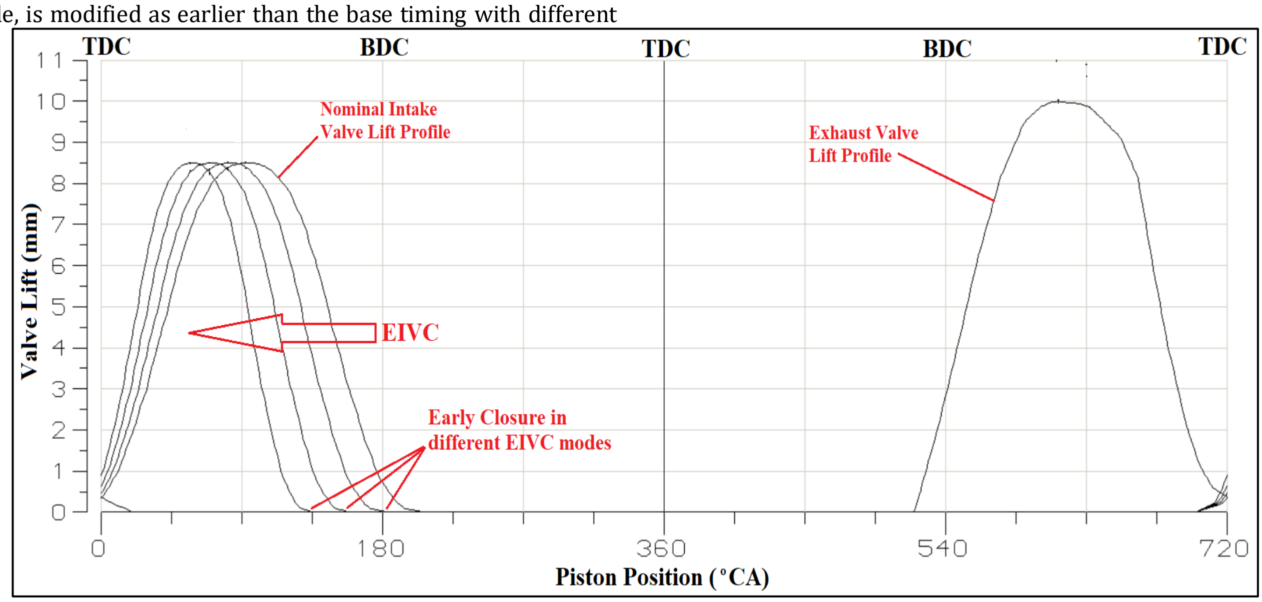

Figure 2. The application of EIVC technique.

Table 2. The modulation of IVC and ETO in EIVC, ET and EIVC+ET modes.

Method applied	Engine parameter	STEPS												
		Nominal	1st	2nd	3rd	4th	5th	6th	7th	8th	9th	10th	11th	12th
EIVC	IVC (°CA ABDC)	25	15	5	-5	-15	-25	-30	-35	-40	-45			
ET	ETO (%)	100	90	80	70	60	50	40	35	32.5	30	28.5	27.5	26.5
EIVC	IVC (°CA ABDC)	(25)	(25)	(20)	(15)	(10)	(5)	(0)	(-5)	(-10)				
+	+	+	+	+	+	+	+	+	+	+				
ET	ETO (%)	(100)	(50)	(45)	(40)	(37.5)	(35)	(32.5)	(30)	(28)				

3. Results and Discussion

In this section, the potential of nominal, ET and EIVC+ET modes on ETM improvement is quantified considering the effects on diesel engine performance characteristics.

The impact of ET on TDT and air-to-fuel ratio (AFR) is illustrated in Figure 3. As mentioned previously, the TDT sensor measures the exhaust temperature at turbine-out position in the system. It is seen that TDT is not altered significantly as ETO is steadily decreased from % 100 (fully open) to % 50 (half open) with % 10 decrements. However, when ETO remains below % 50, TDT has a strong tendency to move up. It is possible to elevate it above 230°C at % 30 opening and even maintain it above 250°C as ETO is actuated at % 26.5 in the system. It is observed that TDT is highly sensitive to ETO as it particularly drops below % 35 in Figure 3. This is mainly due to the sudden reduction in AFR at those low ETO values. AFR in Figure 3 remains more or less the same at high ETO conditions, where TDT is only slightly affected. However, similar to TDT, it is noticeably sensitive to ETO as it goes down below % 40. That dramatically decreased AFR (from 63.5 to as low as 55) seems to be related to the high TDT (above 250°C) in the model. Overall, TDT and AFR are inversely proportional and move in opposite directions as exhaust throttle valve is adequately closed in the system.

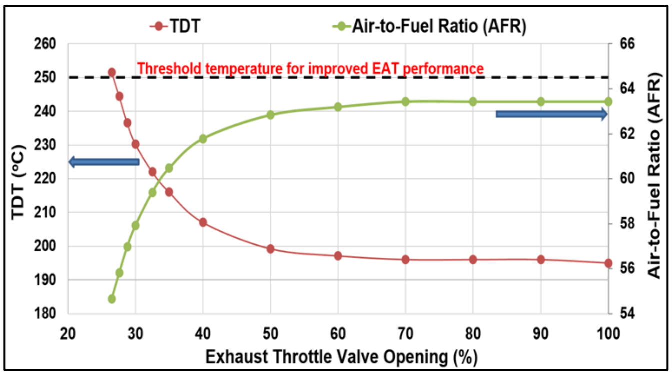

Figure 3. Effect of exhaust throttling in TDT and AFR.

As indicated in Figure 3, TDT and AFR are strongly related and affect one other. To understand why AFR goes through an abrupt reduction at low ETO values, the effect of exhaust throttling in volumetric efficiency and fuel injection rate can be examined in Figure 4.

The volumetric efficiency and the fuel injection rate in Figure 4 are mostly fixed at high ETO values (above % 50), which is in line with the constant AFR at those points, as in Figure 3. However, as ETO especially falls below % 40, fuel injection rate begins to rise dramatically in Figure 4. Volumetric efficiency is only slightly increased at those positions. The decreasing AFR at low ETO cases in Figure 3 can be attributed to the increasing trend of fuel consumption in Figure 4. In other words, the system marginally improves the air charge and is still forced to utilize considerably higher amounts of fuel to secure constant engine load (2.5 bar BMEP) at low ETO cases, thus leading to reduced AFR in Figure 3. The dramatic change in pumping loss, shown as PMEP in Figure 5, accounts for the rapid surge in fuel injection rate in Figure 4.

To a certain ETO point (about % 50), pumping loss is not that altered in Figure 5, which explains the stable fuel injection rate and stable AFR in Figure 4 and Figure 3, respectively. However, when ETO is restricted further, no more a negligible increase is seen in pumping loss, but a steep climb in Figure 5. That evident rise imposes an unavoidable increase in fuel consumption, as in Figure 4. Brake thermal efficiency (BTE) is affected highly negatively due to high fuel demand in Figure 5 as well. BTE is high (% 32) at high ETO values since pumping loss is relatively low at those cases. Yet it is forced to decline below % 30 as ETO is kept below % 35. It decreases below % 27 in extremely closed ETO case, where pumping loss is seen to surpass 1.2 bar (more than twice that obtained in nominal mode). The major rise in pumping loss enables elevated TDT (improved above 250°C), as in Figure 3, with a noticeable fuel inefficiency (close to % 5) in the system. As Figure 6 demonstrates, there is a certain demand in high $IMEP_{gross}$ (the potential of the engine to yield useful power) at low ETO cases, where there is dramatically increased pumping loss.

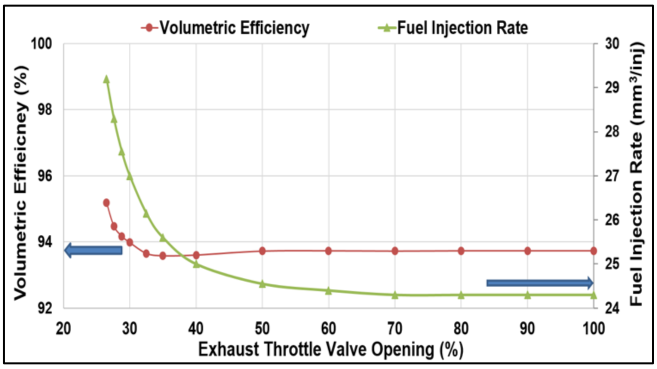

Figure 4. Effect of exhaust throttling in volumetric efficiency and fuel injection rate.

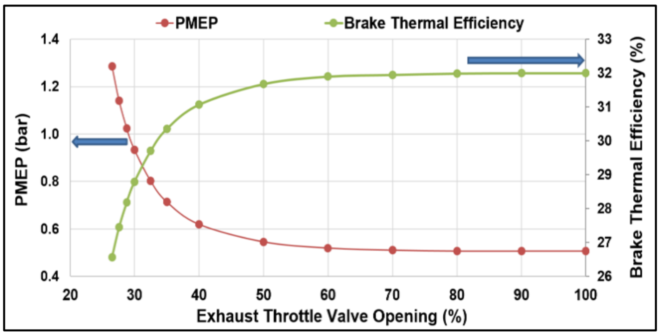

Figure 5. Effect of exhaust throttling in pumping loss and brake thermal efficiency.

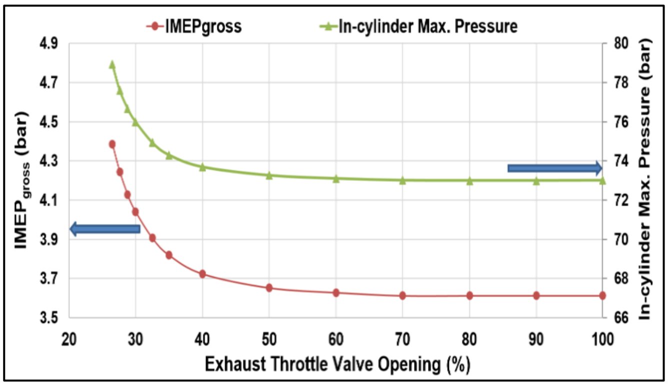

Figure 6. Effect of exhaust throttling in $IMEP_{gross}$ and in-cylinder max. pressure.

The system is not compelled to increase $IMEP_{gross}$ at high ETO cases in Figure 6 since pumping loss is secured at low levels at those cases (as in Figure 5). The requirement of high $IMEP_{gross}$ starts as pumping loss is shifted to rather higher levels compared to nominal mode. More fuel is injected to recover the increased loss (as in Figure 4) and thus, $IMEP_{gross}$ is raised beyond 4.3 bar to keep the steady operation of the engine. Considering $IMEP_{gross}$ is 3.6 bar in nominal mode, it increases by approximately % 20 in extreme exhaust throttling case. Additional fuel consumption at low ETO cases improves in-cylinder maximum pressure in Figure 6 as well. While it exhibits a continuous behavior at high ETO values (remains close to 73 bar), a certain deviation towards higher levels (up to 79 bar) is evident, especially at ETO values below % 35, due to the sudden rise in fuel usage (as in Figure 4) in the system. The elevated in-cylinder maximum pressure due to exhaust throttling can be attributed to the noticeable change in pressure at exhaust-port exit, as illustrated in Figure 7.

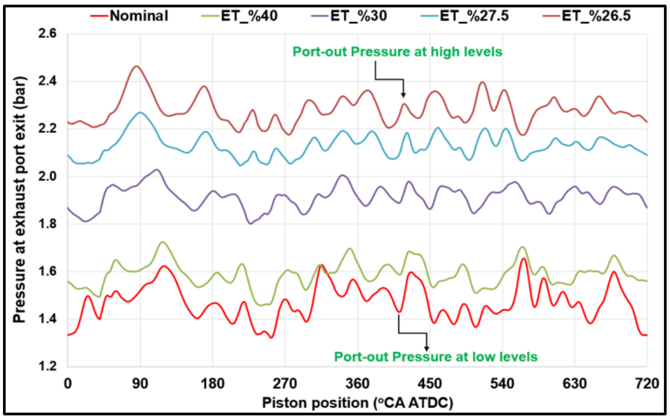

Figure 7. Effect of different exhaust throttling modes in outlet pressure of exhaust ports.

The actuation of exhaust throttle valve positioned at pre-turbine has a significant impact in pressure at exhaust port exit, as indicated in Figure 7. Keeping valve opening percentage to % 40 causes a minor rise in pressure at the outlet of exhaust ports. However, constraining the throttle valve opening percentage to % 30 and below results in a substantial rise in exhaust port-out pressure. That considerable pressure rise at the downstream of exhaust port is the primary reason behind the increased pumping loss and thus, the evident surge in fuel injection rate and in-cylinder maximum pressure, as in Figure 4 and Figure 6, respectively. It is derived that exhaust port-out pressure is extremely sensitive to throttle valve opening area. Particularly as valve opening percentage is held below % 30, even a minor reduction in the valve opening area has the potential to cause a noteworthy rise in exhaust port exit pressure. Therefore, controlling the ETO percentage within a certain range is very important considering it is highly prone to induce high fuel penalty in the system.

As shown in previous plots, exhaust throttling has the capacity to raise TDT above 250°C. However, it also owns a significant disadvantage of up to % 5 decrease in BTE, as in Figure 5. The worsened efficiency not only rises the total fuel expended, but also results in increase in emission rates (particularly CO, CO_2 and PM) until EAT unit is fully warmed up for effective emission conversion efficiency in the system. Therefore, using exhaust throttling alone for rapid EAT warm up is highly inclined to cause undesirable emission rates. That is quite disadvantageous in terms of practical application in highway vehicles. In order to reduce the fuel penalty and a possible hike in emission rates, exhaust throttling is coupled with EIVC technique, which was examined and found to be fuel saving in many former works [15,17,41,46,47]. For a comprehensive analysis, EIVC+ET combined method needs to be compared with EIVC and ET methods. The modulations presented explicitly earlier in Table 2 are used for the comparison of the potential of each technique to achieve threshold exhaust temperature (TDT > 250°C) in the model. Figure 8 below demonstrates the impact of EIVC, ET and EIVC+ET combined techniques on TDT and brake specific fuel consumption (BSFC).

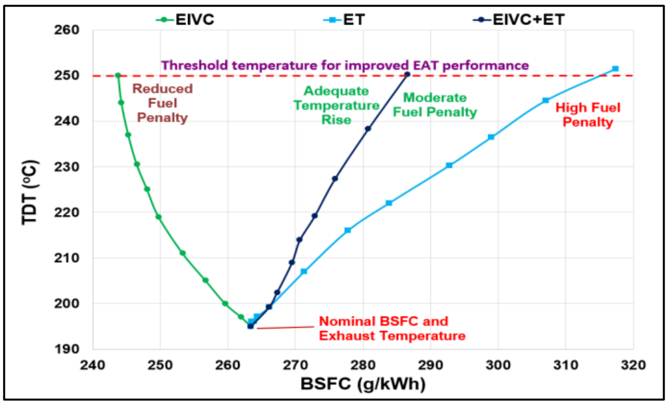

Figure 8. Effect of EIVC, ET, EIVC+ET methods in TDT and BSFC.

ET is seen to be the least efficient technique in Figure 8 as it needs a BSFC of close to 320 g/kWh to reach the threshold exhaust temperature (~250°C). Considering the nominal BSFC remains below 270 g/kWh, utilizing ET alone for enhanced EAT operation is not that advantageous for a practical operation. Unlike ET, EIVC elevates TDT in a fuel saving manner through reduced pumping loss [16], which is beneficial for active post-treatment of exhaust gases. However, EIVC lowers a significant amount of in-cylinder mass to realize TDT above 250°C in the model, which leads to a substantial reduction in exhaust flow rate, as illustrated in Figure 9. Therefore, using EIVC alone can be disadvantageous for fast EAT warm up as exhaust flow rate, similar to exhaust temperature, has a critical role for rapid heat up.

In view of the negative effects of ET and EIVC modes, EIVC+ET combined mode can provide greater than 250°C TDT with improved fuel penalty compared to ET mode in Figure 8. BSFC is decreased below 290 g/kWh via moderate use of EIVC in this mode. The system still suffers from fuel penalty, yet it is more feasible in comparison to ET mode. Use of EIVC allows for less aggressive exhaust throttling (% 28) in combined mode, which is also helpful to reduce BSFC to low, manageable levels.

One other benefit of EIVC+ET mode is to maintain relatively increased exhaust flow rate compared to EIVC mode, as shown in Figure 9. In contrast to the noticeable decrease (from 4.65 kg/min to close to 3.0 kg/min) in EIVC mode, the combined mode can manage the exhaust flow rate above 4.1 kg/min, which is much closer to the nominal mode and thus, is preferable for accelerated EAT warm up. Exhaust flow rate in EIVC+ET mode is less than that in ET mode, however, a similar TDT (~250°C) is sustained in a fuel-efficient manner. Thus, for a similar fuel penalty with ET mode, higher TDT (>> 250°C) can be attained in EIVC+ET mode to further improve the EAT warm up process. EIVC+ET mode can also be more beneficial than ET mode at particularly lower engine loads (BMEP < 2.5 bar), where the requirement in TDT rise is greater than 55°C to control TDT above 250°C. At those cases, unlike EIVC+ET mode, ET mode is highly insufficient to generate a noticeable TDT rise for a similar fuel inefficiency.

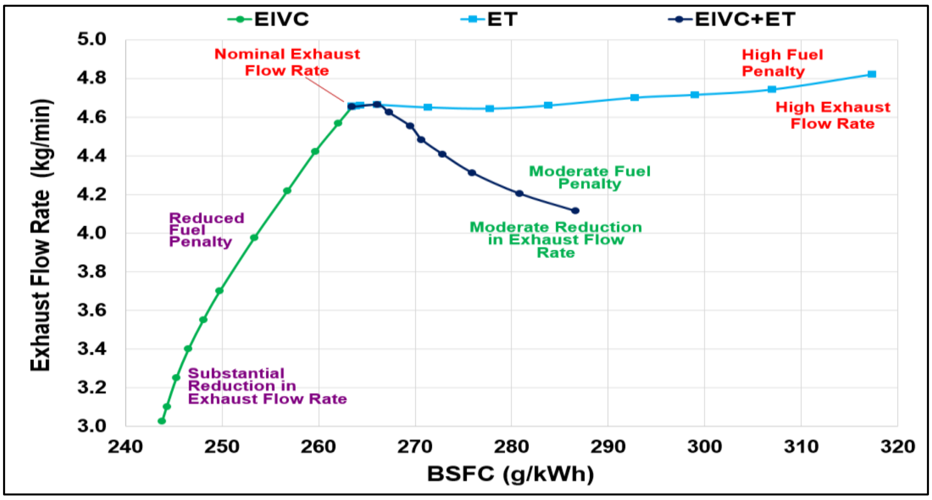

Figure 9. Effect of EIVC, ET, EIVC+ET in exhaust flow rate and BSFC.

After obtaining the influence of EIVC, ET and EIVC+ET strategies on exhaust temperature and exhaust flow rate at turbine exit, their impact on EAT warm up can be evaluated. As expressed in methodology section, EAT heat up mostly depends on the temperature and mass flow rate at EAT inlet. Equation (8) in methodology section considers both of those engine performance parameters. High heat transfer rates $(\dot{Q})$ are needed to speed up the after-treatment heat up mechanism, which necessitates high TDT and high exhaust flow rate.

Elevated TDT can be achieved in each mode, as in Figure 8. However, BSFC is a serious limit as TDT is aimed to boost above 250°C, especially in ET and EIVC+ET modes. Therefore, for the comparison of $(\dot{Q})$ values in those strategies, fuel penalty is restricted to % 8.8. That BSFC constraint allows the application of EIVC+ET technique in its extreme form (EIVC at 10°CA BBDC and an ETO percentage of % 28) and ET technique in its moderate form (an ETO percentage of % 31). EIVC method can be implemented in its extreme form as it is already fuel efficient. The potential of those aforementioned modes on EAT warm up is examined between EAT temperatures of -50°C and 350°C, which considers both cold and warm medium for the EAT unit. The base $(\dot{Q})$ value is taken as the heat transfer rate calculated with equation (8) in nominal mode at EAT temperature of 0°C. Considering that base heat transfer rate as 1.0, all other transfer rates are normalized for the comparison. Normalized heat transfer rates for EIVC_extreme, ET_moderate and EIVC+ET_extreme techniques are indicated in Figure 10.

As shown in Figure 10, there are two main parts for all the heat transfer rates of each mode, which are divided by a "zero heat transfer line". The upper part corresponds to positive transfer rates, which improve the EAT temperature from a very cold temperature (-50°C) to above 200°C in general. Unlike the upper part, lower part corresponds to negative transfer rates, which decrease the EAT temperature from a high temperature (350°C) to below 250°C in general. The points where heat transfer rates and "zero heat transfer line" are crossed represent the highest EAT temperature that can be attained in each mode. Since the $T_{turbine\ out}$ and T_{EAT} are equal at those points, no heat transfer is possible between the exhaust gas and the EAT unit. From an EAT warm up perspective, it is preferable to keep these crossing points at the far right end of the "zero transfer line" whenever possible in Figure 10.

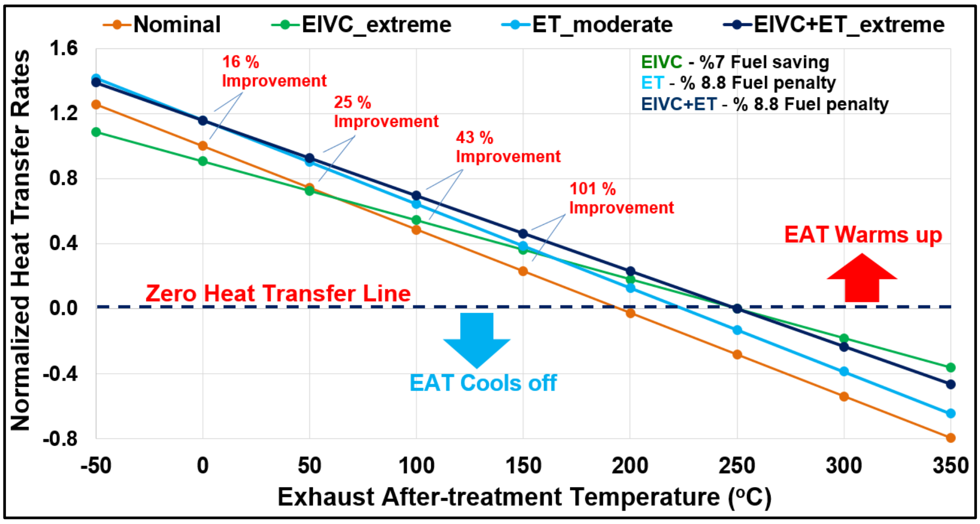

Figure 10. Effect of EIVC_extreme, ET_moderate, EIVC+ET_extreme in EAT warm up

Nominal mode in Figure 10 is seen to be the lowest effective for EAT warm up among all modes. The crossing point remains to the far left of 250°C, which denotes that effective EAT cannot be achieved in this mode. Although the exhaust flow rate is high in nominal mode, as in Figure 9, the heat transfer rates at low T_{EAT} values (between -50°C and 50°C) are worse than both ET_moderate and EIVC+ET_extreme modes in Figure 10. At those temperatures, nominal mode is only better than EIVC_extreme mode, which suffers from a noticeable reduction in exhaust flow rate. Nominal mode is also not preferable for EAT cool off zone since it has the highest negative transfer rates, which accelerate the EAT cool off process.

EIVC_extreme in Figure 10 has the advantage of both fuel efficiency and maintaining the crossing point to the far right of the “zero heat transfer line”. However, as in Figure 9, it induces a considerable decline in exhaust flow rate and thus, particularly at EAT temperatures below 150°C, its warm up performance is mostly ineffective. Similar to nominal mode, its improving effect of EAT warm up is quite limited. Nevertheless, it can be highly beneficial for the system to adopt EIVC_extreme mode as T_{EAT} surpasses 200°C due to % 7 fuel efficient performance. Also, EIVC_extreme is the most successful method in EAT cool off zone. Since it has the lowest negative heat transfer rates, EAT cool off is delayed in that mode more than other modes.

ET_moderate and EIVC+ET_extreme modes are superior to other modes in EAT heat up zone in Figure 10. Especially at EAT temperatures below 100°C, those two techniques are significantly better than both nominal and EIVC_extreme modes. ET_moderate improves T_{EAT} to 225°C and sustains elevated exhaust flow rate (close to 4.8 kg/min), as in Figure 8 and Figure 9, respectively. Although its crossing point in “no heat transfer line” remains behind the one achieved in EIVC_extreme mode, it raises heat transfer rates more until T_{EAT} exceeds 150°C due to the unimpaired exhaust flow rate. However, its effective performance in EAT warm up area is not valid for EAT cool off area, as in Figure 10. Inadequate temperature rise (compared to EIVC_extreme and EIVC+ET_extreme) and improved exhaust rates result in high negative transfer rates, and thus fast EAT cool down process. Considering it causes % 8.8 fuel penalty, it is not the first option to use particularly below the zero heat transfer line.

EIVC+ET_extreme mode in Figure 10 has two significant benefits for rapid EAT warm up. Firstly, similar to EIVC_extreme mode, its crossing point in “no heat transfer line” is placed as far to the right of the nominal mode as possible, which is better than ET_moderate mode. Secondly, similar to ET_moderate mode, exhaust flow rate in combined mode is not interrupted as much, as in Figure 9, which is better than EIVC_extreme mode. Those two advantages enable it to own comparable heat transfer rates with ET_moderate mode at low EAT temperatures (from -50°C to 50°C) for the same fuel penalty (% 8.8). More importantly, at higher T_{EAT} points (above 50°C), EIVC+ET_extreme mode acquires up to % 101 improvement in heat transfer rates, which none of other strategies can achieve in Figure 10. Apparently, the combined mode is the fastest to reach the threshold temperature (250°C) for active EAT operation among all modes. However, it should be noted that the model at hand focuses mostly on the rise of heat transfer rates and does not consider the dynamic parameters such as valve acceleration and jerk due to EIVC. The warm up duration can take several minutes in some cases (particularly in cold weather conditions). That long heat up process can worsen those dynamic parameters and wear out cam and pushrod mechanisms, which is undesirable. Therefore, in real engine applications, instead of applying a single strategy (such as EIVC+ET combined technique), multiple strategies (first ET technique up to 50°C or 100°C and then continue the process with EIVC+ET combined technique) can be applied during the whole EAT warm up process. So that usage of EIVC can be limited and a possible failure in camshaft mechanism due to variable valve timing (VVT) can be avoided in the system.

Not only is EIVC+ET_extreme mode effective for rapid EAT heat up, but also it is beneficial to enhance EAT cool down as it leads to decreased negative heat transfer rates. The moderately diminished exhaust flow rate is effective to keep negative transfer rates at low levels. However, given the % 8.8 fuel penalty in Figure 10, it is not as attractive as EIVC_extreme mode for the improvement of EAT cool off process. Overall, EAT unit can stay hot (above 250°C) for longer in a more fuel saving manner in EIVC_extreme mode compared to EIVC+ET combined mode.

4. Conclusions

This paper aims to enhance the diesel EAT warm up process with reasonable use of two different engine-specific techniques, EIVC and ET, in an engine model. Both individual and combined application of those techniques are examined in the work.

ET is useful to elevate temperature above 250°C at turbine outlet and thus, has the capacity to achieve enhanced EAT performance. It does not affect volumetric efficiency and exhaust flow rate negatively, which noticeably improves EAT warm up. However, the system is faced with a significant fuel consumption penalty (above % 15) in ET mode, which makes it as an unfeasible technique for rapid EAT warm up.

The application of EIVC on the model results in a substantial rise in $T_{turbine\ out}$ (greater than 55°C) in a fuel efficient manner (up to % 7). However, it also halts the exhaust flow rate at dramatic amounts (more than 30 percent), which is a crucial drawback for quick EAT heat up. Therefore, use of EIVC alone is not favorable for the get-hot period of EAT units in diesel engine systems.

Simultaneous application of EIVC and ET, as EIVC+ET, is found to have advantages compared to EIVC alone and ET alone methods. EIVC+ET combined mode boosts $T_{turbine\ out}$ as much as EIVC alone mode, but with significantly higher exhaust flow rates. Also, for the same fuel penalty (% 8.8), it enables an EAT system of 250°C and above while ET alone mode can barely move the EAT system over 225°C. It does not impair exhaust flow rates considerably as $T_{turbine\ out}$ is enhanced above 250°C. Therefore, it has the most improved heat transfer rates (up to % 101) among all techniques examined. That ascertains the EIVC+ET combined strategy more favorable than other strategies for fast EAT warm up in the system. Combined mode is more preferable than ET alone mode for EAT cool off period as well, due to slightly reduced exhaust flow rates and higher potential to rise exhaust temperature (25°C greater than ET alone mode).

The combined EIVC+ET technique still leads to inefficiency in the system. Thus, not only the proposed method, but also other inner-engine methods such as close and late post fuel injection and engine-independent techniques such as electrical heating and afterburner should be examined to further improve the engine efficiency.

Ethics committee approval and conflict of interest statement

This article has no conflicts of interest with any individual or institution.

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Nomenclature

ABDC After bottom dead center

AFR Air to fuel ratio

ATDC After top dead center

BBDC Before bottom dead center

BDC Bottom dead center

BMEP Brake mean effective pressure, bar

BSFC Brake specific fuel consumption, g/kWh

BTDC Before top dead center

CA Crank angle, degree

DOC Diesel oxidation catalyst

DPF Diesel particulate filter

EAT Exhaust after-treatment

EFR Exhaust flow rate

EIVC Early intake valve closure

ET Exhaust throttling

ETM Exhaust thermal management

ETO Exhaust throttle opening

EVC Exhaust valve closure

IMEP$_{gross}$ Gross indicated mean effective pressure, bar

NOx Nitrogen oxide

PMEP Pumping mean effective pressure, bar

RPM Revolution per minute

SCR Selective catalytic reduction

SOI Start of injection timing, degree

TDC Top dead center

TDT Temperature downstream turbine

VVT Variable valve timing

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