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Chien-Cheng Chen
Department of Mechanical and Mechatronic Engineering, National Taiwan Ocean University, Taiwan, ROC, 20972001@mail.ntou.edu.tw

Yuan-Liang Jeng
Department of Mechanical and Mechatronic Engineering, National Taiwan Ocean University, Taiwan, ROC

Shun-Chang Yen
Department of Mechanical and Mechatronic Engineering, National Taiwan Ocean University, Taiwan, ROC

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RESEARCH ARTICLE

Applicability of Reducing Valve Timing Overlap for Diesel Engines Under High Exhaust Back Pressure

Chien-Cheng Chen*, Yuan-Liang Jeng, Shun-Chang Yen

Department of Mechanical and Mechatronic Engineering, National Taiwan Ocean University, Taiwan, ROC

Abstract

The exhaust back pressure of diesel engines becomes increasing higher nowadays. As an example, the De-NO_x system and DE-SO_x system necessitated by the increasingly stringent emission standards, would result in increased exhaust back pressure for those diesel engines adopting such systems. Some ships adopt underwater exhaust system to save space on the working deck and to reduce noise and air pollution, while the hydrostatic pressure under water level has made the exhaust back pressure of diesel engines getting much higher. Under high exhaust back pressure, to keep discharging exhaust unhindered and operating smoothly for diesel engine, it often results in large extent of reduction in engine maximum brake output, increased fuel consumption, and lower combustion efficiency with heavy exhaust smokes.

To address the above-mentioned disadvantages for diesel engines under high exhaust back pressure, one solution is proposed in this study by reducing valve timing overlap (the overlap crank angle between exhaust valve closing and intake valve opening. In this study, valve timing overlap is varied to study the engine performance characteristics and applicability, especially the improvement of brake output and brake specific fuel consumption of engine.

Via engine simulation, 25 kPa, 45 kPa and 65 kPa gauge of exhaust back pressure are studied for a turbocharged diesel engine. The results of engine parameters, including brake output, brake specific fuel consumption, turbine inlet temperature, intake air mass flow rate and exhaust mass flow rate are analyzed to establish empirical equations for the relations with valve timing overlap. The results indicate that the brake output and brake specific fuel consumption are improved by reduced valve timing overlap.

An optimal condition of full load operation has been found that a reduced valve timing overlap of 12 °CA would lead to highest brake output. Thus, reducing valve timing overlap is proved to be a feasible solution for the diesel engines under high exhaust back pressure, partially resolve the problems of reduced maximum brake output and increased fuel consumption.

Keywords: Diesel engine, Exhaust back pressure, Valve timing overlap, Engine performance, Brake specific fuel consumption

1. Introduction

In shipbuilding industry, diesel engine is an important source of power. Its structure, design, and applicability are all key aspects of shipbuilding, and all are worthy of their own in-depth research. Among the various aspects of the diesel engine, the exhaust pressure and load characteristics of specific diesel engines are of particularly importance for propulsion configuration, due to the concern of main power source of an entire ship, or the output source of an entire power plant (in the cases of diesel-generator propulsion).

Nowadays, the exhaust back pressure of diesel engines becomes increasing higher. As an example, the increasingly stringent emission standards have necessitated diesel engines to install denitrification (De-NO_x) system and desulfurization (DE-SO_x) system, which both would result in increased exhaust back pressure for diesel engines.

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* Corresponding author.
E-mail address: 20972001@mail.ntou.edu.tw (C.-C. Chen).

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High-speed watercrafts include ferryboats, sightseeing boats, yachts, patrol boats, and catamarans. To reduce ship resistance, the draft of the hull of high-speed ships has to be rather shallow. And due to the passenger-carrying nature, the diesel engine of those ships commonly adopt "wet exhaust" and underwater exhaust system [1,2]. The benefits of underwater exhaust system include saving space on the working deck and reducing noise and air pollution. The exhaust pipe emitting exhaust gases extends into seawater to reduce smoke emissions into air, while turning air pollutants into water soluble inorganic salt, thus reducing air pollution. On the other hand, the exhaust is cooled by seawater, thus reducing the risk of fire caused by high exhaust temperature. However, the exhaust back pressure of diesel engine becomes much higher due to the hydrostatic pressure difference between water level and exhaust pipe exit.

Under high exhaust back pressure, to keep the discharge exhaust unhindered and engine operation smooth for diesel engine, some problems arose such as reduced engine maximum brake output, increased fuel consumption, and lower combustion efficiency with heavy exhaust smokes.

Currently, there are only limited studies on exhaust back pressure of diesel engines, yet it is believed to be worthy of more academic attention.

1.1. Scope and objectives

In this study, to resolve the problems of diesel engines operating under heightened back pressure, including reduced volumetric efficiency, reduced brake power, and increased fuel consumption, a solution is proposed by reducing valve timing overlap. The valve timing overlap of a diesel engine is defined as the overlap crank angle between the exhaust valve closing (EVC) and the intake valve opening (IVO).

Via engine simulation of this study, valve timing overlap is varied. 25 kPa, 45 kPa and 65 kPa gauge of exhaust back pressure are studied for a turbocharged diesel engine.

1.2. Overview of dissertation

The simulation tool used in this study is GT-Power engine simulation software, which is a sub-module of GT-Suite software. The engine model is established according to CAT C32 diesel engine, which is a 12-cylinder, twin-turbocharger diesel engine. The first step of engine model establishment is to input CAT C32 engine design parameters. In the second step, the simulation results of engine model are validated with the factory acceptance test (FAT) results of C32 diesel engine until the results of brake power under 50 %, 75 %, and 100 % load operations are accurate within the range of 1 % error.

The valve timing overlap is varied by the combinations of 8 different IVOs (intake valve opening angle) and 7 different EVCs (exhaust valve closing angle). The 8 different IVOs are 350 °CA, 351.5 °CA, 353 °CA, 354.5 °CA, 356 °CA, 357.5 °CA, 359 °CA, 360 °CA. The 7 EVCs are 395 °CA, 391.5 °CA, 388 °CA, 384.5 °CA, 378.5 °CA, 372.5 °CA, 366.5 °CA. Excluding the repetitive combinations, there are 45 distinct valve timing overlaps. Three high exhaust back pressure conditions are studied, including 25 kPa, 45 kPa, and 65 kPa gauge, while the results of 0 kPa back pressure are served for comparison purpose.

The results of full load characteristics under the conditions of 45 valve timing overlaps and 4 exhaust back pressure conditions are studied.

1.3. Achievements

The results of engine parameters, including brake power output, brake specific fuel consumption (BSFC), turbine inlet temperature, intake air mass flow rate and exhaust mass flow rate, are analyzed. The results indicate that the brake output and brake specific fuel consumption are improved under full load operation by reduced valve timing overlap.

Empirical equations are established for the relations for the above mentioned engine parameters with valve timing overlap. The results indicate that the brake output and brake specific fuel consumption are improved by reduced valve timing overlap.

An optimal condition of has been found that a reduced valve timing overlap of 12 °CA would lead to highest brake output. Thus, reducing valve timing overlap is proved to be a feasible solution for the diesel engines under high exhaust back pressure, to partially resolve the problems of reduced maximum brake output and increased fuel consumption.

2. Overview of simulation

2.1. Literature review

In the study by Basaran H. and Ozsoysal O. (2016) [3], the effects on engine exhaust by changing intake valve closing (IVC) timing is studied by simulation. The results of this study on diesel engine performance show the turbine exit temperature (TET) increases as the IVC timing advances to 65 °CA from the nominal valve timing. Besides, the accompanying effects of the advancing IVC timing are reduced volumetric efficiency, lessened pumping loss, and less fuel consumption.

In another research, Dalla Nora et al. (2016) studied the effects of various intake and exhaust valve timings on the performance and gas exchanging of a two-stroke GDI engine with overhead valves [4]. The research team studied the impacts of exhaust throttling at low and maximum exhaust valve lifts in order to evaluate potential effects of turbocharger and after-treatment technologies。Additionally, the effects of engine performance and gas exchanging are analyzed with varied intake valve lifts.

The results of study indicate the effects of longer intake and exhaust valve opening durations. The effective compression ratio, the effective expansion ratio, and air trapping efficiency are decreased, and leads to lower indicated output and increased turbocharger power consumption. Meanwhile, the charging efficiency and the scavenging ratio are decreased by shorter valve opening durations.

In one of the research scenarios with 130 °CA IVO, 240 °CA IVC (intake valve closing), 120 °CA EVO (exhaust valve opening), and 230 °CA EVC, the engine's performance indicators were all ideal at two test speeds. Reducing the valve timing between EVO and IVO to 10 °CA leads to smooth exhaust without negative effects on scavenging. Reducing the valve timing between EVC and IVC to 10 °CA also increase boost effect.

Sapra H. et al. (2017) conducted experiments on marine diesel engine operating under high back pressure using a simulation model [5]. The research team test on pulse turbocharged and constant pressure mid-speed turbocharged engines under different static back pressures (but only up to 0.05 bar gauge), and for two different values of valve overlap. The results show that the adaption of pulse turbocharged engine under high back pressures is improved at smaller valve overlaps.

The results of 100 °CA valve overlap by Sapra H. et al. also shows the results on distributions of charge-air ratio, charge-air pressure, exhaust valve temperature, and exhaust gas temperature. Besides, such mapped data as the charge-air ratio and the exhaust valve temperature are also investigated with various back pressure conditions.

Increased exhaust back pressure would bring down the efficiency of exhaust turbine, lower air-excess ratio (even to the point causing smokes). It also even causes the overload of exhaust valve, and high exhaust temperatures.

Furthermore, under the operation of high exhaust back pressure, a large valve timing overlap (such as 100 °CA) would result in exhaust flow in reverse direction, lower fresh air-mass in the cylinder, which hinders combustion efficiency.

At the beginning of compression, trapped burned gases within cylinder would raise both in−cylinder temperature and exhaust temperatures. Additionally, negative scavenging creates a counter pressure at compressor outlet, which may cause compressor surge, raising the risk of compressor damage.

On the other hand, a smaller valve timing overlap under high exhaust back pressure would reduce the negative scavenging and keep the air-excess ratio from dropping too low. When the valve timing overlap is 30 °CA, the air-excess ratio increases and the exhaust valve temperature decreases, both helping the normal operation of engine.

This proves that a smaller valve timing overlap would causing the increase of amount of total trapped mass within cylinder (under the same air-excess ratio). And for an earlier close of exhaust valve, it would cause more air flowing into cylinder due to larger valve timing overlap during scavenging [3,6].

2.2. Model description

The research process in this study follows the steps below (summarized in Fig. 1):

- Collect and study related academic literature on reduced diesel engine valve timing overlaps; compile relevant discussions and performance analyses.
- Use GT Power software for diesel engine simulation; define its applicability scope, simulation setting, and theoretical information [7,8]. Draw up a simulation plan in which the parametric variations, intended results, asymptotes, and intended observations are clearly defined.
- CAT C32 diesel engine is selected as the research platform. Based on the design data of CAT C32 engine [9−11], an engine simulation model is built [12]. Based on the factory acceptance test (FAT) results of C32 engine [13], the simulation results of engine model are validated until the results of brake power under 50 %, 75 %, and 100 % load operations are accurate within the range of 1 % error.
- Using 8 different IVOs and 7 different EVCs to obtain different valve timing overlap angles. Simulations are done for 4 different exhaust back pressure conditions (0 kPa, 25 kPa, 45 kPa,

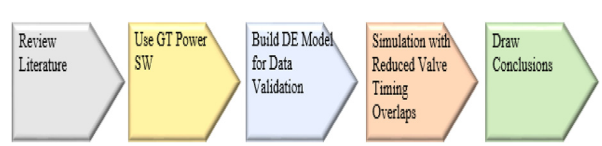

Fig. 1. Key steps of research processes.

and 65 kPa gauge) to study the effects of reducing valve timing overlap angle on the load characteristics and power output of C32 diesel engine.

3. Simulation method

3.1. Caterpillar C32 diesel engine (CAT C32)

CAT C32 diesel engine, manufactured by Caterpillar Inc., is selected as the research platform. CAT C32 diesel engine has different versions : main engine for marine propulsion, diesel-generator set propulsion, and generator engine. The version of diesel-generator set C32 engine is used. The engine speed is kept constant at 1800 rpm. This engine is 4 stroke, direct injected, turbocharged and aftercooled. The engine configuration is V12. Two turbochargers are placed in parallel. Each side of 6 cylinders share one turbocharger and one aftercooler. The specifications of C32 engine is shown in Table 1. The major FAT results of C32 Engine Performance is shown in Table 2.

Table 1. Specifications of C32 diesel engine.

Manufacturer	CAT
Configuration	V12, 4 stroke cycle, direct injection diesel.
Emissions	US EPA Tier 3/IMO II emissions certified
Rated engine speed	1800 rpm
Bore x stroke	145 mm × 162 mm
Displacement	32.1 L
Aspiration	tubocharged aftercooled aspiration
Governor	Electronic (A4 ECM)
Refill capacity	lube oil system w/oil filter change; 146L
Oil change interval	750 h
Cooling	heat exchanger or keel cooled
Fly wheel housing	SAE No. 0 with SAE No. 18 fly wheel (136 teeth)
Rotation	counterclockwise from flywheel end
Sales model	C32
Engine power (bkW)	994
Compression ratio	15

3.2. Model validation with FAT results

According to the official specifications and technical data, an engine simulation model is built on GT-Power software, to simulate the performance of C32 engine at differing loads. Below are the details:

- An engine simulation model is built on GT-Power software, as shown in Fig. 2.
- Based on the V12 configuration of C32 engine, 2 separate turbochargers and 2 after-coolers, only one side of 6 cylinders of the V-type engines are modeled for the simulation. Some of the simulation results (such as mass flow rates and brake power output) must be multiplied by 2 to obtain the results of the V12 engine.
- The design parameters, such as (but not limited to) engine bore, stroke, compression ratio, valve size, valve lash, the sizes of intake and exhaust manifolds are used in the model according to the engine spec. and all possible sources such as maintenance and service guide.
- The rated engine speed is set constant at 1800 rpm. Different load settings of 50 %, 75 %, 100 % and 110 % are used in the model validation process described later. The air fuel ratios used for different load settings are estimated based on the technical data of dry intake air mass and wet exhaust mass found in the manufacture's FAT results of C32 engine.
- The crank angle is based on 0 °CA crank angle which corresponds to TDC at the end of compression. Figure 3 show valve lift vs crank angle curve for intake and exhaust valves. The valve lift for mechanical lift cams equals to the result of subtracting valve lash from the product of cam robe lift and rocker ratio.
- The valve timing angles used for model validation are:
  IVO (intake valve open) 350 °CA, or 10 °CA bTDC.
  IVC (intake valve close) 586 °CA, or 46 °CA aBDC.
  EVO (exhaust valve open) 135 °CA, or 45 °CA bBDC.
  EVC (exhaust valve close) 395 °CA, or 35 °CA aTDC.

Table 2. Major FAT results of C32 engine performance.

Load	Engine Power	Brake mean eff. Pre. (BMEP)	Brake spec. consumption (BSFC)
%	bkW	kPa	g/(bkW-h)
110	10,930	22,710	204.7
100	994.0	2064.0	207.2
75	746.0	1548.0	216.3
50	497.0	1032.0	220.1

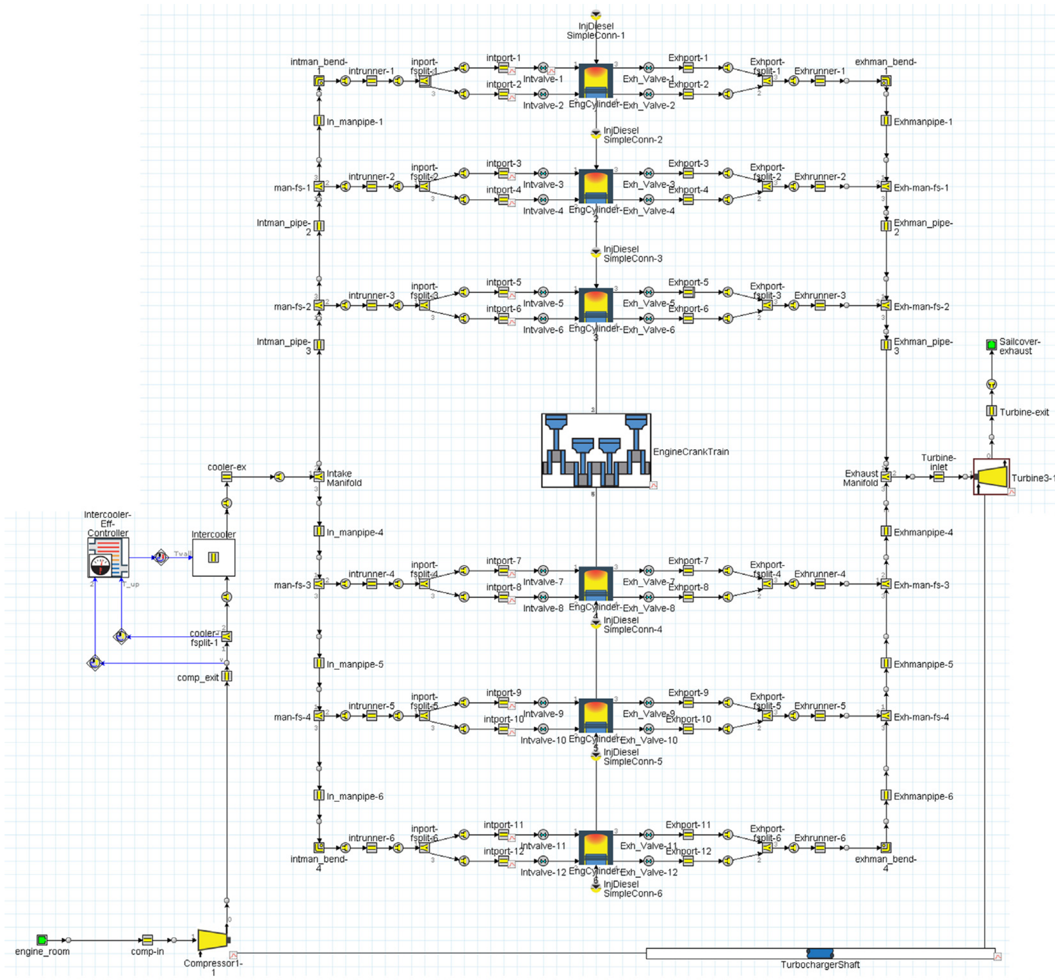

Fig. 2. GT-Power simulation model for data validation.

The valve timing overlap between EVC and IVO is therefore 45 °CA, which is also shown in Fig. 3.

- The performance maps of compressor and turbine are established from the mass flow rates and pressure ratios compressor at different load settings are estimated based on the technical data in the manufacture's FAT results of C32 engine. Fixed-geometry turbocharger is used throughout the simulation of this study.

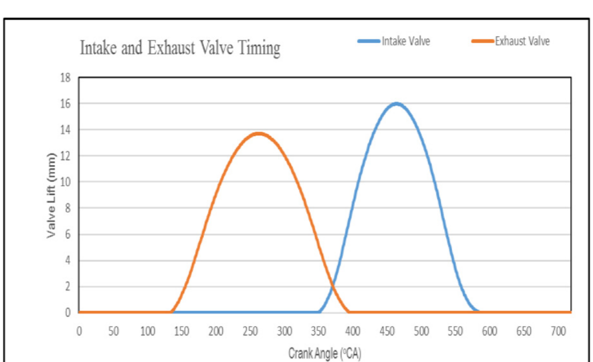

Fig. 3. Valve lift curve for intake and exhaust valves.

- The intake boundary conditions are set at 1 atm (or 101.3 kPa) and 25 °Cof dry air. The exhaust boundary condition for model validation process is set at 101.3 kPa (i.e., 0 kPa gauge of exhaust back pressure).
- The simulation results are compared with the manufacture's FAT results of C32 engine. The results of model validation are shown in Table 3. The simulation result of brake power for the 4 load settings are all within the range of 1 % error. Only the errors of BMEP and BSFC at 75 % load are higher, but are still within acceptable 3 % range.
- The simulation model has been successfully validated. Simulations may now proceed to study the impact of different valve timing overlaps and under different exhaust back pressures on the performance parameters of diesel engine.

Table 3. Results of model Validation.

Load		Brake Power	BMEP	BSFC
		bkW	bar	g/(bkW-h)
110%	DE FAT data	1093.00	22.71	204.70
	GT result	1094.60	22.70	204.50
	percentage error	0.1%	0.0%	-0.1%
100%	DE FAT data	994.00	20.64	207.20
	GT result	1001.80	20.80	205.50
	percentage error	0.8%	0.8%	-0.8%
75%	DE FAT data	746.00	15.48	216.30
	GT result	741.00	15.90	210.70
	percentage error	-0.7%	2.7%	-2.6%
50%	DE FAT data	497.00	10.32	220.10
	GT result	492.60	10.20	219.70
	percentage error	-0.9%	-1.2%	-0.2%

3.3. Valve timing overlap variations for simulation

To study the effects of reducing valve timing overlap to improve engine performance and fuel consumption for diesel engine under high exhaust back pressure, the valve timing overlap is reduced from initial setting of 45 °CA.

The valve timing overlap is varied by the combinations of 8 different IVOs (intake valve opening angle) and 7 different EVCs (exhaust valve closing angle).

- 8 IVO parametric variations : 350 °CA, 351.5 °CA, 353 °CA, 354.5 °CA, 356 °CA, 357.5 °CA, 359 °CA, 360 °CA.
- 7 EVC parametric variations : 395 °CA, 391.5 °CA, 388 °CA, 384.5 °CA, 378.5 °CA, 372.5 °CA, 366.5 °CA.

Among the variations of 56 IVO and EVC combinations, 11 of which are repetitive. Excluding the repetitive combinations, 45 distinct valve timing overlaps and four exhaust back pressure conditions (0 kPa, 25 kPa, 45 kPa, and 65 kPa gauge) are used as independent parameters, to study the full load engine performance and fuel consumption.

Table 4 shows the 45 distinct valve timing overlaps and 4 exhaust back pressure conditions.

4. Results and discussions

4.1. Analysis of brake output

Under 0 kPa exhaust back pressure and 100 % load, the brake output of diesel engine decreases slightly as the valve timing overlap is being reduced, as shown in Fig. 4.

As the valve timing overlap is reduced, the brake output remains rather constant in the 0 kPa exhaust back pressure scenario.

Fig. 5 shows the analysis of brake output under 25 kPa, 45 kPa, 65 kPa back pressures and 100 % engine load. Comparing the brake outputs of 25 kPa, 45 kPa, 65 kPa exhaust back pressure scenarios: the brake output is the lowest under 65 kPa exhaust back pressure and is the highest under 25 kPa exhaust back pressure, which indicates that brake output is indeed negatively affected when the diesel engine operates under high back pressure.

As the valve timing overlap is reduced, the brake output remains rather constant in the 0 kPa exhaust back pressure scenario (Fig. 4).

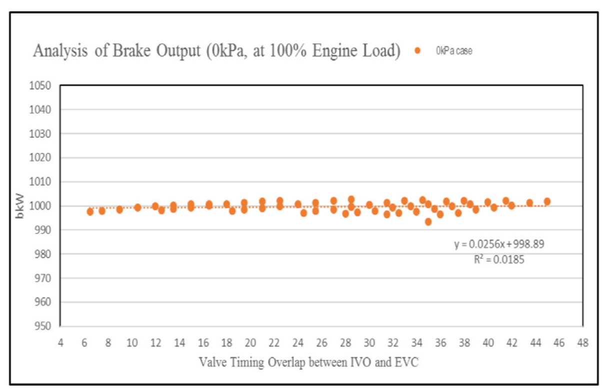

Fig. 4. Analysis of brake output under 0 kPa back pressure and 100 % engine load.

Table 4. Variations for valve overlap and back pressure.

Parameter 1 : Valve Overlap 45 lots Variations	8 IVO Variations	45 Distinct Valve Timing Overlap Values	7 EVC Variations
	350.0 °CA	45.0 °, 43.5 °, 42.0 °, 41.5 °, 40.5 °, 40.0 °, 39.0 °, 38.5 °, 38.0 °,	395.0 °CA
	351.5 °CA	37.5 °, 37.0 °, 36.5 °, 36.0 °, 35.5 °, 35.0 °, 34.5 °, 34.0 °, 33.5 °,	391.5 °CA
	353.0 °CA	33.0 °, 32.5 °, 32.0 °, 31.5 °, 30.5 °, 30.0 °, 29.0 °, 28.0 °, 28.5 °,	388.0 °CA
	354.5 °CA	27.0 °, 25.5 °, 24.5 °, 24.0 °, 22.5 °, 21.0 °, 19.5 °, 18.5 °, 18.0 °,	384.5 °CA
	356.0 °CA	16.5 °, 15.0 °, 13.5 °, 12.5 °, 12.0 °, 10.5 °, 9.0 °, 7.5 °, 6.5 °.	378.5 °CA
	357.5 °CA		372.5 °CA
	359.0 °CA		366.5 °CA
	360.0 °CA		
Parameter 2 : Backpressure 4 lots Variations	0 kPa, 25 kPa, 45 kPa, 65 kPa		

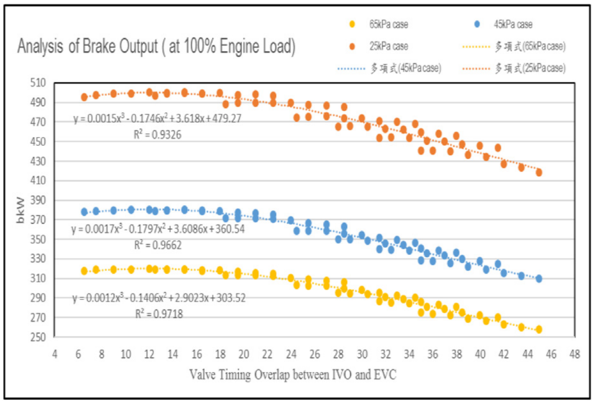

Fig. 5. Analysis of brake output under 25 kPa, 45 kPa, 65 kPa back pressures and 100 % engine load.

In Fig. 5 of 25 kPa, 45 kPa, and 65 kPa back pressure scenarios, however, when the valve timing overlap is reduced, notable increases of brake output are observed. The brake outputs reach the maximum at a certain valve timing overlap and then go down as the overlaps continue to decrease.

In the 25 kPa, 45 kPa, 65 kPa back pressure scenarios, the maximum brake output occurs at approximately the same valve timing overlap, 12 °CA.

For the above 3 back pressure scenarios, the brake output may be related to valve timing overlap by empirical equations. In the following empirical equations, y stands for brake output in bkW, and x stands for valve timing overlap in °CA.

For 25 kPa gauge back pressure, the empirical relation is:

$$y = 0.0015x^3 - 0.1746x^2 + 3.618x + 479.27 \tag{1}$$

For 45 kPa gauge back pressure, the empirical relation is:

$$y = 0.0017x^3 - 0.1797x^2 + 3.6086x + 360.54 \tag{2}$$

For 65 kPa gauge back pressure, the empirical relation is:

$$y = 0.0012x^3 - 0.1406x^2 + 2.9023x + 303.52 \tag{3}$$

An empirical equation can be derived from the database and the three asymptote equations above—the brake output values in 45 kPa and 65 kPa scenarios can be predicted with the brake output value in 25 kPa scenario at the same valve timing overlap. The calculated (i.e., predicted) values are within 3.4 % errors with the simulation values. Let B_{25kpa}, B_{45kpa}, B_{65kpa} stand for the brake output under 25 kPa, 45 kPa and 65 kPa back pressure, respectively. The empirical equation is:

$$B_i = B_{25kPa} * F^{K_1} \tag{4}$$

Where B_i is the brake output under 45 kPa (or 65 kPa) back pressure condition. B_{25kpa} is the brake output under 25 kPa back pressure condition, at the same valve timing overlap value.

F= (25/45) if B_{45kpa} is to be predicted, or

F= (25/65) if B_{65kpa} is to be predicted.

K_1: empirical value (=0.5)

4.2. Analysis on brake specific fuel consumption

Under 0 kPa exhaust back pressure and 100 % load, the brake specific fuel consumption (BSFC) increases slightly as the valve timing overlap is being reduced, as shown in Fig. 6.

Fig. 7 shows the analysis of Brake Specific Fuel Consumption (BSFC) under 25 kPa, 45 kPa, 65 kPa back pressures and 100 % engine load. Comparing the BSFC's of the 25 kPa, 45 kPa, 65 kPa exhaust back pressure scenarios: the BSFC is the highest under

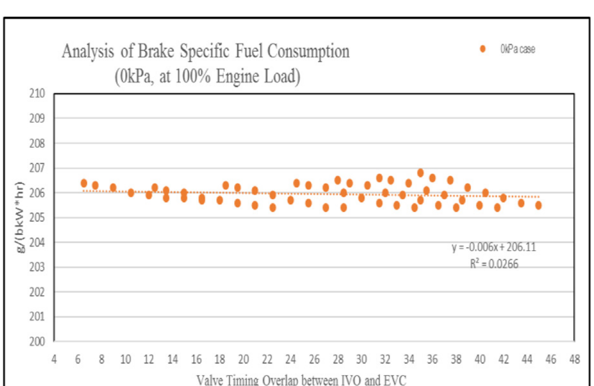

Fig. 6. Analysis of brake specific fuel consumption under 0 kPa back pressure and 100 % engine load.

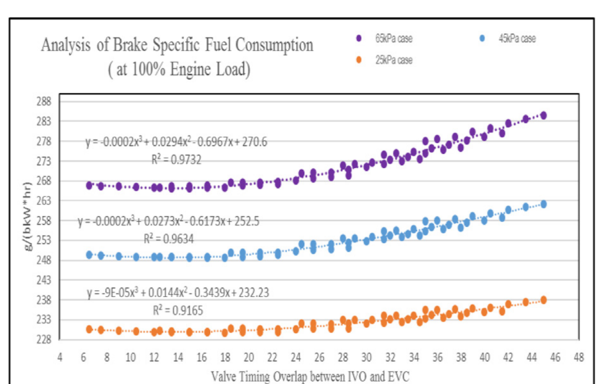

Fig. 7. Analysis of brake specific fuel consumption under 25 kPa, 45 kPa, 65 kPa back pressures and 100 % load.

65 kPa exhaust back pressure and is the lowest under 25 kPa exhaust back pressure, which indicates that BSFC is indeed negatively affected when the diesel engine operates under high back pressure.

While the brake output remains rather constant as the valve timing overlap is being reduced in 0 kPa exhaust back pressure scenario, it shows a more notable decrease in the 25 kPa, 45 kPa, and 65 kPa scenarios. The BSFCs reach the minimum at a certain valve timing overlap and then go up as the overlaps continue to decrease.

In Fig. 7 of 25 kPa, 45 kPa, and 65 kPa back pressure scenarios, notable decreases of BSFC are observed when the valve timing overlap is reduced. The minimum BSFC at different back pressures occurs approximately at 14 °CA valve timing overlap for the 25 kPa and 65 kPa scenarios, while approximately at 13 °CA valve timing overlap for the 45 kPa scenario.

For the 3 back pressure scenarios, the BSFC may be related to valve timing overlap by empirical equations. In the following empirical equations, y stands for BSFC in g/bkW-hr, and x stands for valve timing overlap in °CA.

For 25 kPa gauge back pressure, the empirical relation is:

$$y = -0.00009x^3 + 0.0144x^2 - 0.3439x + 232.23 \tag{5}$$

For 45 kPa gauge back pressure, the empirical relation is:

$$y = -0.0002x^3 + 0.0273x^2 - 0.6173x + 252.5 \tag{6}$$

For 65 kPa gauge back pressure, the empirical relation is:

$$y = -0.0002x^3 + 0.0294x^2 - 0.6967x + 270.6 \tag{7}$$

The BSFC values in 45 kPa and 65 kPa scenarios can be predicted with the BSFC in 25 kPa scenario at the same valve timing overlap value. The calculated values are within 2.5 % errors with simulation values. Let C_{25kpa}, C_{45kpa}, C_{65kpa} stand for the BSFC under 25 kPa, 45 kPa and 65 kPa back pressure, respectively. The empirical equation is:

$$C_i = C_{25kPa} * F^{K_2} \tag{8}$$

where C_i is the BSFC under 45 kPa (or 65 kPa) back pressure condition. C_{25kpa} is the BSFC under 25 kPa back pressure condition, at the same valve timing overlap value.

F= (25/45) if C_{45kpa} is to be predicted, or.

F= (25/65) if C_{65kpa} is to be predicted.

K_2: empirical value (=0.16)

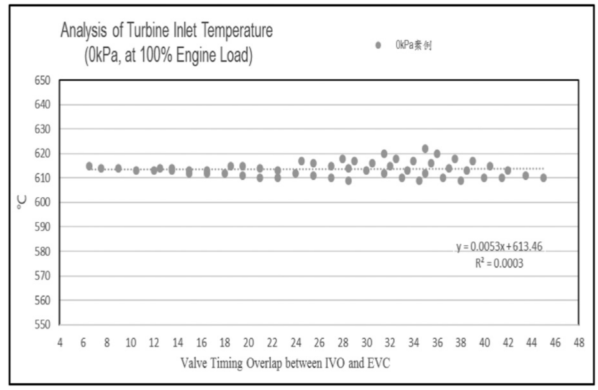

Fig. 8. Analysis of turbine inlet temperature under 0 kPa back pressure and 100 % engine load.

4.3. Analysis of turbine inlet temperature

Fig. 8 shows the inlet temperature of turbocharger under 0 kPa exhaust back pressure and 100 % load. The DE's turbocharger inlet temperature increases slightly as the valve timing overlap is being reduced.

Fig. 9 shows the analysis of turbine inlet temperature under 25 kPa, 45 kPa, 65 kPa back pressures and 100 % engine load. Comparing the results of 25 kPa, 45 kPa, 65 kPa exhaust back pressure scenarios: the turbine inlet temperature is the highest under 65 kPa exhaust back pressure and is the lowest under 25 kPa exhaust back pressure, which indicates that turbine inlet temperature is indeed negatively affected when the diesel engine operates under high back pressure.

Comparing with Fig. 8 the rather constant turbine inlet temperature in 0 kPa scenario, the turbine inlet temperatures notably increase in 25 kPa, 45 kPa, and 65 kPa scenarios when the valve timing overlap is reduced.

For the 3 back pressure scenarios, the turbine inlet temperatures may be related to valve timing overlap

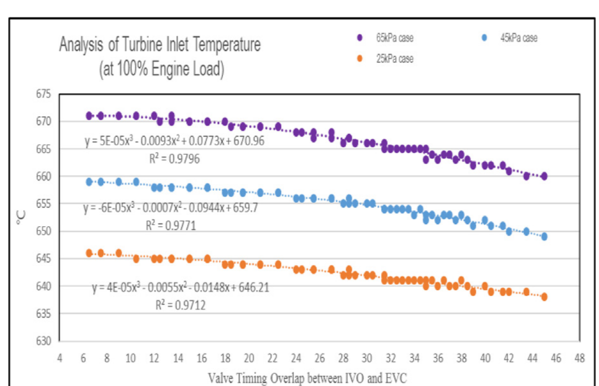

Fig. 9. Analysis of turbine inlet temperature under 25 kPa, 45 kPa, 65 kPa back pressures and 100 % load.

by empirical equations. In the following empirical equations, y stands for turbine inlet temperature in °C, and x stands for valve timing overlap in °CA.

For 25 kPa gauge back pressure, the empirical relation is:

$$y = 0.00004x^3 - 0.0055x^2 - 0.0148x + 646.21 \tag{9}$$

For 45 kPa gauge back pressure, the empirical relation is:

$$y = -0.00006x^3 - 0.00075x^2 - 0.0944x + 659.7 \tag{10}$$

For 65 kPa gauge back pressure, the empirical relation is:

$$y = 0.00005x^3 - 0.0093x^2 - 0.0773x + 670.96 \tag{11}$$

The turbine inlet temperature values in 45 kPa and 65 kPa scenarios can be predicted with the turbine inlet temperature in 25 kPa scenario at the same valve timing overlap value. The calculated values are within 0.7 % errors with simulation values. Let T_{25kpa}, T_{45kpa}, T_{65kpa} stand for the turbine inlet temperature under 25 kPa, 45 kPa and 65 kPa back pressure, respectively. The empirical equation is:

$$T_i = T_{25kPa} * F^{K_3} \tag{12}$$

where T_i is the Turbine inlet temperature under 45 kPa (or 65 kPa) back pressure condition. T_{25kpa} is the BSFC under 25 kPa back under 25 kPa back pressure condition, at the same valve timing overlap value.

$F=$ (25/45) if T_{45kpa} is to be predicted, or

$F=$ (25/65) if T_{65kpa} is to be predicted.

K_3: empirical value ($=-0.04$)

4.4. Analysis of intake air mass flow rate

Under 0 kPa exhaust back pressure and 100 % load, the intake air mass flow rate decreases slightly

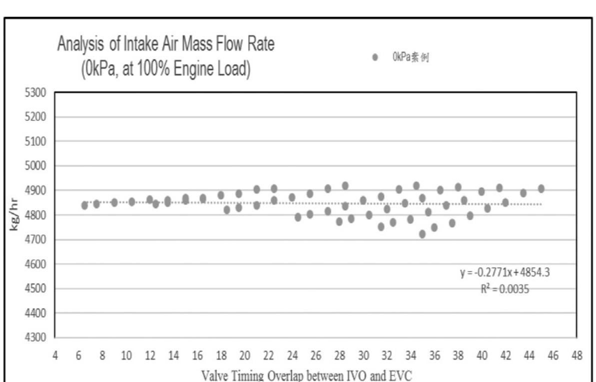

Fig. 10. Analysis of intake air mass flow rate under 0 kPa back pressure and 100 % engine load.

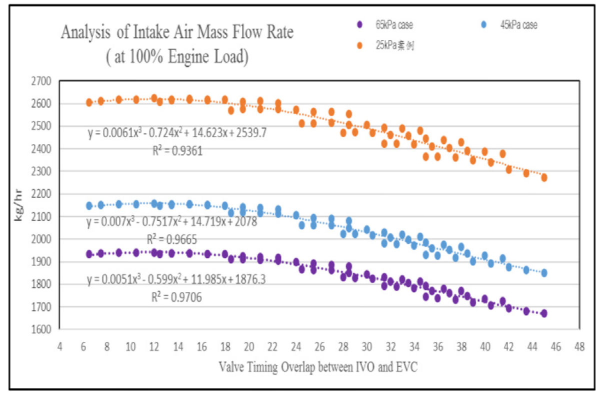

Fig. 11. Analysis of intake air mass flow rate under 25 kPa, 45 kPa, 65 kPa back pressures and 100 % load.

as the valve timing overlap is being reduced, as shown in Fig. 10.

Fig. 11 shows the analysis of intake air mass flow rate under 25 kPa, 45 kPa, 65 kPa back pressures and 100 % load. Comparing the results of 25 kPa, 45 kPa, 65 kPa exhaust back pressure scenarios: the intake air mass flow rate is the lowest under 65 kPa exhaust back pressure and is the highest under 25 kPa exhaust back pressure, which indicates that intake air mass flow rate is indeed negatively affected when the diesel engine operates under high back pressure.

Fig. 11 also shows that intake air mass flow rate notably increases in each of 25 kPa, 45 kPa, and 65 kPa scenarios when the valve timing overlap is reduced. The intake air mass flow rate reaches the maximum at a certain valve timing overlap and then goes down slightly as the overlap continues to decrease.

In the 25 kPa, 45 kPa, 65 kPa back pressure scenarios, the maximum intake air mass flow rate occurs at approximately the same valve timing overlap, 12 °CA.

For the 3 back pressure scenarios, the intake air mass flow rate may be related to valve timing overlap by empirical equations. In the following empirical equations, y stands for intake air mass flow rate in kg/hr, and x stands for valve timing overlap in °CA.

For 25 kPa gauge back pressure, the empirical relation is:

$$y = 0.0061x^3 - 0.724x^2 + 14.623x + 2539.7 \tag{13}$$

For 45 kPa gauge back pressure, the empirical relation is:

$$y = 0.007x^3 - 0.7517x^2 + 14.719x + 2078 \tag{14}$$

For 65 kPa gauge back pressure, the empirical relation is:

$$y = 0.0051x^3 - 0.599x^2 + 11.985x + 1876.3 \tag{15}$$

An empirical equation can be derived from the database. The intake air mass flow rate in 45 kPa and 65 kPa scenarios can be predicted with the intake air mass flow rate value in 25 kPa scenario at the same valve timing overlap. The predicted values are within 2.3 % errors with the simulation values. Let Ma_{25kpa}, Ma_{45kpa}, Ma_{65kpa} stand for the intake air mass flow rate under 25 kPa, 45 kPa and 65 kPa back pressure, respectively. The empirical equation is:

$$Ma_i = Ma_{25kPa} * F^{K_4} \tag{16}$$

Where Ma_i is the intake air mass flow rate under 45 kPa (or 65 kPa) back pressure condition. Ma_{25kpa} is the intake air mass flow rate under 25 kPa back pressure condition, at the same valve timing overlap value.

$F=$ (25/45) if Ma_{45kpa} is to be predicted, or

$F=$ (25/65) if Ma_{65kpa} is to be predicted.

K_4: empirical value (=0.33)

4.5. Analysis of exhaust mass flow rate

During the simulation, the air fuel ratio is kept constant for 100 % full load operation. Therefore, the results of exhaust mass flow rate have the same trend with the results of intake air mass flow rate.

It is shown in Fig. 12, the exhaust mass flow rate decreases slightly as the valve timing overlap is reduced Under 0 kPa exhaust back pressure and 100 % load.

Fig. 13 shows the analysis of the exhaust mass flow rate under 25 kPa, 45 kPa, 65 kPa back pressures and 100 % load. Comparing the results of 25 kPa, 45 kPa, 65 kPa exhaust back pressure scenarios: the exhaust mass flow rate is the lowest under 65 kPa back pressure and is the highest under 25 kPa back pressure, which indicates that exhaust mass flow rate is indeed negatively affected when the diesel engine operates under high back pressure.

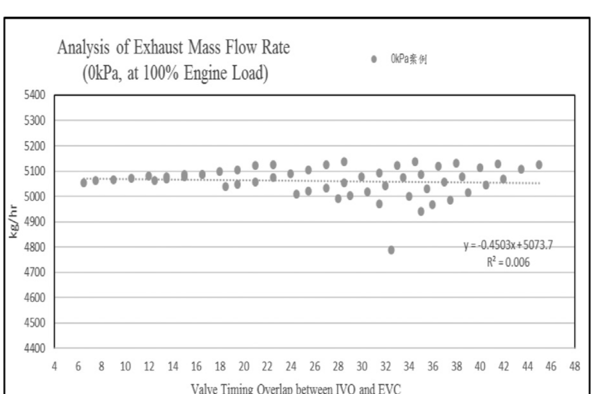

Fig. 12. Analysis of exhaust mass flow rate under 0 kPa back pressure and 100 % load.

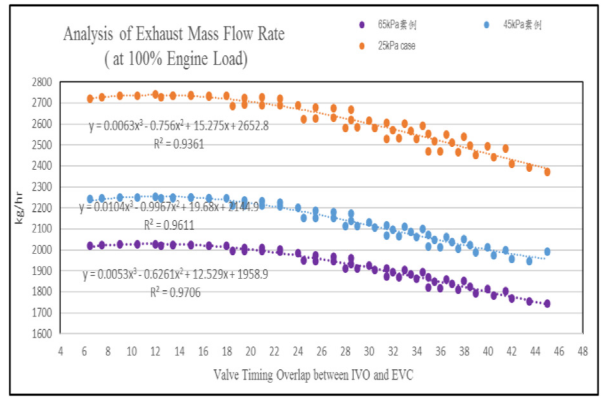

Fig. 13. Analysis of exhaust mass flow rate under 25 kPa, 45 kPa, 65 kPa back pressures and 100 % load.

Fig. 12 also shows that exhaust mass flow rate notably increases in each of 25 kPa, 45 kPa, and 65 kPa scenarios when the valve timing overlap is reduced. The intake air mass flow rate reaches the maximum at a certain valve timing overlap and then goes down slightly as the overlap continues to decrease.

In the 25 kPa, 45 kPa, 65 kPa back pressure scenarios, the maximum exhaust mass flow rate occurs at approximately the same valve timing overlap, 12 °CA.

For the 3 back pressure scenarios, the exhaust mass flow rate may be related to valve timing overlap by empirical equations. In the following empirical equations, y stands for exhaust mass flow rate in kg/hr, and x stands for valve timing overlap in °CA.

For 25 kPa gauge back pressure, the empirical relation is:

$$y = 0.0063x^3 - 0.756x^2 + 15.275x + 2652.8 \tag{17}$$

For 45 kPa gauge back pressure, the empirical relation is:

$$y = 0.0104x^3 - 0.9967x^2 + 19.68x + 2144.9 \tag{18}$$

For 65 kPa gauge back pressure, the empirical relation is:

$$y = 0.0053x^3 - 0.6261x^2 + 12.529x + 1958.9 \tag{19}$$

An empirical equation can be derived from the database. The exhaust mass flow rate in 45 kPa and

65 kPa scenarios can be predicted with the exhaust mass flow rate value in 25 kPa scenario at the same valve timing overlap. The predicted values are within 2.3 % errors with the simulation values. Let Me_{25kpa}, Me_{45kpa}, Me_{65kpa} stand for the exhaust mass flow rate under 25 kPa, 45 kPa and 65 kPa back pressure, respectively. The empirical equation is:

$$Me_i = Me_{25kPa} * F^{K_4} \tag{20}$$

Where Me_i is the exhaust air mass flow rate under 45 kPa (or 65 kPa) back pressure condition. Me_{25kpa} is the exhaust mass flow rate under 25 kPa back pressure condition, at the same valve timing overlap value.

$F=$ (25/45) if Me_{45kpa} is to be predicted, or

$F=$ (25/65) if Me_{65kpa} is to be predicted.

K_4: empirical value (=0.33)

4.6. Summaries and discussions

Table 5 shows the Summarized simulation results of brake output, BSFC, turbine inlet temperature and intake mass flow rate at 100 % full load, while under 0 kPa, 25 kPa, 45 kPa, 65 kPa gauge of exhaust back pressure. The results are only shown for 2 valve timing overlap angles : the original 45 °CA overlap, and the optimal 12~14 °CA overlap.

Under the original setting of 45 °CA valve overlap, when the exhaust back pressure increases from 0 kPa to 65 kPa, the brake output decreases dramatically from 1001.8bkW to 257.5bkW, while the intake mass flow rate decreases from 4908 kg/h to 2023 kg/h. The extent of increase for turbine inlet temperature (from 610 °C to 660 °C) and BSFC (from 205.5g/bkW-hr to 284.5 °Cg/bkW-hr) are minor compared with brake output and intake mass flow rate.

The major causes of poor engine performance under high exhaust back pressure are the low volumetric efficiency of diesel engine, low compressor pressure ratio and turbine pressure ratio. The poor adaptability of fixed-geometry turbocharger under high exhaust bake pressures has now been recognized in afore studies of our research team.

In this study, reducing valve timing overlap angle is proposed to improved poor engine performance under high exhaust back pressure. The above simulation results show that under 25 kPa, 45 kPa, 65 kPa gauge of back pressure, maximum brake output occurs at 12 °CA valve overlap, while optimal cases of other parameters occurs at 13~14 °CA valve overlap. Actually, the curves of those parameters at the range of 12~14 °CA overlap are rather flat, having nearly constant values, It is shown in Table 5, when the diesel engine operates full load under 65 kPa back pressure, by reducing valve timing overlap angle from original 45 °CA overlap to 12~14 °CA overlap, the brake output increases from 257.5bkW to 319.6bkW (a 24.1 % recovery), while the BSFC decreases from 284.5g/bkW-hr to 266.2g/bkW-hr (a 6.4 % reduction).

The benefits of reducing valve timing overlap angle when the engine operating under high exhaust back pressures are now been clear. This includes the increases of intake mass flow rate (and hence the volumetric efficiency) and the brake output, while the reduction of fuel consumption.

Table 5. Summarized simulation results of 100 % load.

back pressure	kPa gauge	0	0	25	25
valve overlap	°CA	45	12~14	45	12~14
brake output	bkW	1001.8	999.8	418.6	500.6
			−0.20 %		19.60 %
BSFC	g/bkW-hr	205.5	205.9	238.1	230
			0.20 %		−3.40 %
Turbine inlet temp	°C	610	614	638	645
			0.70 %		1.10 %
Intake mass flow rate	kg/hr	4908	4845	2271	2616
			−1.30 %		15.20 %
back pressure	kPa gauge	45	45	65	65
valve overlap	°CA	45	12~14	45	12~14
brake output	bkW	309.8	380.4	257.5	319.6
			22.80 %		24.10 %
BSFC	g/bkW-hr	262.2	249	284.5	266.2
			−5.00 %		−6.40 %
Turbine inlet temp	°C	649	658	660	670
			1.40 %		1.50 %
Intake mass flow rate	kg/hr	1851	2154	1745	2023
			16.40 %		15.90 %

However, the accompanying drawback is the increased turbine inlet temperature. The high exhaust back pressure has already pushed turbine inlet temperature of 610 °C (0 kPa back pressure, 45 °CA overlap) to be increased to 660 °C (65 kPa back pressure, 45 °CA overlap). Now reducing valve timing overlap angle to 12~14°CA would cause turbine inlet temperature even higher to 670 °C. This would impose a challenge for the selection of turbine material in the design and manufacture of turbocharger.

5. Conclusion and future work

- For diesel engines operating under high exhaust back pressures, it has been proven in this study that the poor engine performance of reduced brake output and increased fuel consumption can be improved by reducing the valve timing overlap angle. The optimal valve timing overlap angle 12~14°CA works for 25 kPa, 45 kPa, 65 kPa gauge of exhaust back pressure.
- The side-effect arisen by reduced valve timing overlap is the increase in the turbine inlet temperature, which may cause the complexity in the material selection, design, and manufacturing of turbocharger.
- For diesel engine operating under high exhaust back pressures, the brake output dwindles significantly. This has caused a serious problem in engine applicability. Reducing valve timing overlap can only partially resolve the problem. Other effective way to increase brake output would be a critical topic for future research. A variable geometry turbocharger might be the solution.

Conflict of interest

The GT-power SW used was purchased from a previous research project with CSBC, but the previous project is not related to this study, so this is an independent research report.

Acknowledgements

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