Investigations on the Potential of Miller Cycle for Performance Improvement of Gas Engine

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Abstract

To further improve thermal efficiency, Miller Cycle was applied to a turbo-charged 338 kW gas engine. Different methods of Miller Cycle were analyzed, including three Early Intake- Valve Closing (EIVC) methods and three Late Intake-Valve Closing (LIVC) methods. After the relatively suitable methods were chosen, the combination of the Miller Cycle and higher compression ratio was extensively investigated. The experimental results demonstrated that the combination of intake valves closing 40°CA earlier(EIVC40) and the compression ratio increasing to 13, the maximum thermal efficiency reached 47

Index terms— miller cycle, compression ratio, thermal efficiency, performance.

1 Investigations on the Potential of Miller Cycle for Performance Improvement of Gas Engine

Abstract-To further improve thermal efficiency, Miller Cycle was applied to a turbo-charged 338 kW gas engine. Different methods of Miller Cycle were analyzed, including three Early Intake-Valve Closing (EIVC) methods and three Late Intake-Valve Closing (LIVC) methods. After the relatively suitable methods were chosen, the combination of the Miller Cycle and higher compression ratio was extensively investigated. The experimental results demonstrated that the combination of intake valves closing 40°CA earlier(EIVC40) and the compression ratio increasing to 13, the maximum thermal efficiency reached 47% and it is about 5~7% higher than the original cycle.

2 Nomenclature

CA : crank angle EIVC : early intake-valve closing LIVC : late intake-valve closing BDC : bottom dead center ABDC : after bottom dead center I. Introduction atural gas is a relatively clean alternative energy, but the higher ignition temperature, the slower flame propagation speed and the smaller coefficient of molecular after the combustion all result in lower thermal efficiency. One effective way to improve the thermal efficiency of the engine is to increase the compression ratio. But for natural gas engines, the combustion temperature will increase when the compression ratio is higher, which leads to the increase of knocking tendency, reducing the reliability and service life of the engine. To avoid knocking, the general measure is to delay the ignition timing that will cancel out the improving of thermal efficiency caused by the increase of the compression ratio [1, 2].

Changing the intake valve closing time so the mixture in cylinder goes through an expansion process before the compression stroke, Miller cycle can decrease the maximum combustion temperature to some extent, combined with a higher compression ratio; it can improve the thermal efficiency. In this paper, the engine performance of applying the Miller cycle on a nature gas engine was studied by simulation, and optimized by different levels of compression ratio. Simulation results indicated that the combination of intake valves closing 40°CA earlier(EIVC40) and the compression ratio increasing to 13 was the most potential method to further improve thermal efficiency, and experimental results proved that the method made engine fuel consumption rate got worse at high engine speeds but improved at media and low speeds, the engine knocking tendency increased at the same time.

3 II. Concept of Miller Cycle

In comparison with the standard cycles, miller cycle is shown in Figure 1 (a). Standard cycle of a gas engines is 1-5-2-3-4-6-7-1, 1-5 is the intake stroke, after the piston moves down to the bottom dead center (point 5) the intake valves are closed; 5-2 is the compression stroke; 2-3-4 is the expansion stroke; 4-6-7 is the exhaust stroke.

EIVC Miller cycle is 1-la-5a-2a-3a-4a-6-7-1, 1-1a is the intake stroke, the intake valves are closed at point 1a, the piston continues to move down to the bottom dead center(BDC) which is denoted by point 5a, from 1a to 5a the in-cylinder mixture expanses, which makes the in-cylinder temperature drops, so when the piston reach BDC, the temperature of the charge is lower in the Miller cycle compared to the standard cycle; the compression stroke is 5a-2a, the temperature of the charge at the point of 2a is also lower, so does the pressure; the expansion stroke is 2a-3a-4a, burst pressure and maximum combustion temperature are relatively low in miller cycle; the exhaust stroke is 4a-6-7.

Another form of Miller cycle is late intake-valve closing (LIVC), as Figure 1 (b) shows, the intake valves which should be closed at point 5 are postponed to point 1, LIVC makes the expansion ratio greater than the compression ratio, and extract heat from fuel as much as possible, which will improve the thermal efficiency.

4 III. Calculation of Miller Cycle

From the thermodynamic point of view, Miller cycle reduces the combustion temperature, which helps to decrease knock tendency and make it possible to raise the compression ratio. In addition, Miller cycle changes the intake quantity which also has impact on the combustion process. In order to optimize the performance, one-dimensional and multi-dimensional numerical modes are set up to study the in-cylinder gas flow and heat release in a gas engine, together with experimental results. The parameters of the gas engine are shown in Table 1. Generally speaking, in order to optimize the intake and exhaust process, the intake valves are usually closed after bottom dead center(ABDC), EIVC or LIVC of Miller cycle discussed here is based on the intake valve close timing of original cycle which is 22°CA ABDC, not the top dead center. Six kinds of Miller cycle are proposed on the first ground, including three EIVC and three LIVC, the valve lifting curves are shown in Figure 2. EIVC20, intake valve close timing is at 2°CA ABDC; EIVC40, intake valve close timing is at 18°CA ABDC; EIVC60, intake valve close timing is at 38°CA ABDC; LIVC20, intake valve close timing is at 42°CA ABDC; LIVC40, intake valve close timing is at 62°CA ABDC; LIVC60, intake valve close timing is at 82°CA ABDC.

The one-dimensional model was built with the software of GT-POWER, and the heat release rate model was Weber function, the heat transfer model was Woschni 1978.

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5 . Calculation Results

Changes of the amount of intake charge are shown in figure 3, ”-60” on the abscissa indicates E60, ”60” indicate L60. Keeping the opening of turbocharger bypass constant, miller cycles change the amount of intake charge. For 1900r/min, due to the longer time for intake, the later the valves close, the more charge the engine gets. Compared to the original cycle, the intake of E20?E40 and E60 decrease by 6.1%, 18% and 31.5% respectively, but intake charge doesn't change a lot from L20 to L60. For 1000r/min, EIVC's influence is not as much as 1900r/min. So in the six Miller cycles, EIVC has more influence than LIVC on the intake amount, and EIVC has more influences at the high speed working condition than the low speed working condition. Figure 4 shows the change of residual gas of miller cycles. For high speed condition, the later the intake valves close, the less the amount of the residual gas, due to the longer time for scavenging. For low speed condition, the residual gas decreases when the intake valves close too early or too late. Generally speaking, the cam profile of the exhaust valve has a greater effect to the residual gas than the intake valve, Figure 5 shows the change of the residual gas according to the exhaust phase, it can be seen that, when the exhaust phase is postponed 10 ~20°CA, the mass fraction of residual gas reaches the smallest value. Theoretically, minimizing the residual gas is an effective measure to broaden the knock limit. The in-cylinder pressure and temperature at the time of intake valves closing both increase while the intake valves close later, as Figure 6 shows. In the case of LIVC, some of the intake charge is pushed reversely into the induct, and the compression work converted into internal energy of the fresh charge in the intake ports, which makes the intake temperature of next cycle increases, as shown in Figure 7. On the contrary, intake temperature drops obviously for EIVC because of the expansion before compression. The purpose of this paper is to reduce the compression temperature by Miller cycle, Figure 7 shows that LVIC doesn't hit the mark, so in the subsequent analysis only EVIC is taken into account. Through the above one-dimensional simulations, we can see that EVIC changes the intake process and decreases the in-cylinder pressure and temperature when the intake valves close, which help to reduce the maximum combustion temperature and detonation tendency, so higher compression ratio is possibly to be employed to improve the thermal efficiency. On the other hand, EVIC changes the state of air-fuel mixture, that will also changes the combustion efficiency, especially when the temperature of the mixture drops, the combustion speed will decrease, and the thermal efficiency will decrease also, so it is necessary to do further analysis for in-cylinder processes by multidimensional numerical simulation.

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7 b) Multidimensional numerical simulation of in-cycle cylinder flow and combustion of EVIC

Large eddy simulation (LES) and G equation flamelet model are employed to investigate the in-cycle cylinder flow and combustion of different EVICs.

i. Large Eddy Simulations When LES is used, the flow variables are decomposed into two parts, the large-scale motions that are supported by the mesh size and the small, sub-grid scale (SGS) motions that are less than the grid size. Spatial filtering is applied to both the variables and the governing equations, leading to governing equations for the resolved large-scale motions. The governing equations can be written as the following [3,4] :( ) ( ) j j j j j j j j i j i j j ij i j j i i i i i hu u h x x h D x t p x u h t h u u u u x x x p x u u x u x u t ? ? ? ? ? ? ? ? ? ? ? ? ? ? ? + ? ? ? ? ? ? ? ? ? ? ? ? ? + ? ? = ? ? + ? ? ? ? ? + ? ? + ? ? ? = ? ? + ? ? = ? ? + ? ? 0 ~

Where ? is the density of the mixture; i u is the velocity component in the i ; , ,i i i i i i i i i i i x t F x x x t d x h x t F x x x t h x t d x ? ? ? ? ? ? ? ?? ?? ?? ? ? ? ?? ?? ?? ? ? ? ? = ? ? ? ? ? ? ? = ? ? ? ? ? ? ? ? ?

Where F is a filter function and ? is the filter width (here taken as the cell size). The last term in Eq.(**??**) is the SGS stress whereas the last term in Eq.(3) is the turbulent transport flux that accounts for the effect of the un-resolved sub-grid turbulence. The SGS stresses are modeled here using the scale-similarity model [5,6] , whereas the SGS scalar transport fluxes are modeled by Smagorinski model. An equation of state is used to couple the pressure, temperature and density in the cylinder. The calorific equation of state is used to compute the temperature of the fluid from the enthalpy. In the present engine test, the in-cylinder flow speed is less than 120 m/s and the Mach number is lower than 0.3. For simplicity, in Eq. (3) the 'low Mach number approximation' has been used. With this approximation, the pressure is split into two parts, a hydrodynamic pressure and a thermodynamic pressure. The former is responsible for the pressure gradient that drives the flow, in Eq. (**??**). The latter depends on time (or the crank

8 ii. Ignition Model

To spark ignition engines, the current numerical grid scale is too large to capture the small structure of the initial phase of the fire kernel, so fire kernel growth in this paper is described by the discrete particle ignition kernel model(DPIK) [7] .

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The Simulation is performed on the software of KIVA. An engine mesh of medium grid density was built with the same geometry as the real engine, as figure 8 shows. The calculation point is 1900r/min-1800Nm, the excess air ratio is 1.5, a spark plug located at the center of the combustion chamber, ignition timing is 20°CA before top dead center. Figure 9 shows the change of turbulence parameters at the time of ignition around the spark plug. When the intake valve closed early, the instantaneous speed increases, which is in favor of the initial flame propagation; the integral length scale increases slightly, it means that EIVC makes the flame be more inclined to propagate at a laminar speed at the first moment; on the other hand, the fluctuation velocity and the turbulent kinetic energy decrease, Which go against the rapid spread of flame.

where k r is the radium of the fire kernel; u ? is the density of the burned fire kernel; plasma S is the Plasma growth rate; Figure 10 shows the heat release, in-cylinder pressure and temperature during combustion. With the intake valves closing early, the heat release lags behind and becomes slow, so the burst pressure and the maximum temperature decrease. E20 doesn't make much difference and the in-cylinder pressure decreases too much for E60. According to the above simulation, E40 has a lower combustion temperature and a relatively rapid combustion process compared with the original cycle, so it is chosen as a scheme to improve the gas engine's performance. Miller cycle decreases the combustion temperature, so the geometry compression ratio can be improved to optimize the engine's performances. And the next step is to optimize the compression ratio.

9 d) Optimization of compression ratio

The original compression ratio is 11, and the chamber profile is shown in Figure 11. Modify the main structural parameters D and H to increase the geometric compression ratio to 12 and 13, as shown in table 2. Table 3 compares the calculated main turbulence parameters with different compression ratios at the ignition time under the condition of 1900r/min-1800N, it shows that the locate turbulence parameters don't change a lot when the compress ratio changes. Figure 12 shows the in-cylinder parameters under different compression ratios at 1900r/min-1800Nm. The higher the compression ratio, the faster the combustion heat release, the higher the boost pressure, and the maximum temperature in the cylinder. Compared to 11, the maximum instantaneous heat release rate, the highest pressure and the maximum temperature raise by 3.2%, 38.1% and 25.9% respectively when the compress ratio raises to 13. In this paper, we focus on the potential of improving the thermal efficiency, so the compression ratio 13 is chosen to used in E40 in the following experiment.

10 IV. Experimental Results

In the test research, in order to optimize the performance, the boost pressure, excess air coefficient, ignition advance angle are carefully chosen for each working condition, the comparisons of the experimental results between original cycle and miller cycle are given below.

11 a) Combustion Duration

Figure 13 shows the comparison of the combustion duration (10%~90%) which varies according to the excess air coefficient and the ignition advance angle. It can be seen that the Miller cycle has a shorter combustion period. At 1900r/min, the combustion duration of the Miller cycle is about 5°CA shorter; at 1000rpm, the combustion duration of the Miller cycle is about 4°CA shorter. Figure 14 shows the comparison of the break specific gas consumption, at 1900r/min, when the load is above 1400Nm, due to the lower volume efficiency and more residual gas, the miller cycle has a higher gas consumption; under 1400Nm, miller cycle has a less gas consumption because of the reduction of cycle charge, At 1000r/min, the specific gas consumption of miller cycle obviously decreases, about 7~8g/kw.h. Figure 15 shows the comparison of the thermal efficiency. At high speed condition, thermal efficiency doesn't have much difference, but at low speed condition, The thermal efficiency of the Miller cycle is about 5~7% higher, the maximum thermal efficiency of the Miller cycle reached 47%, the increase of thermal efficiency is mainly caused by the higher compression ratio, while the residual gas coefficient has great influence on the thermal efficiency of the high speed. The simulation calculation of the Miller system which combines EIVC with higher compression ratio is carried out, and the heat release process is investigated in detail, according to the calculation results, the optimization scheme is selected and put into experimental study. Compared to the original cycle, the optimization scheme has the following characteristics: 1) the pumping loss is slightly lower. 2) the heat release is relatively concentrated and rapid, the combustion center is about 5 °CA earlier, and the time for the combustion duration decreases by 5 °CA, so the thermal efficiency increases.

3) The break specific gas consumption of the middle and low speed improves obviously, but in the condition of high speed and heavy load, it is deteriorated due to the increase of the residual exhaust gas.

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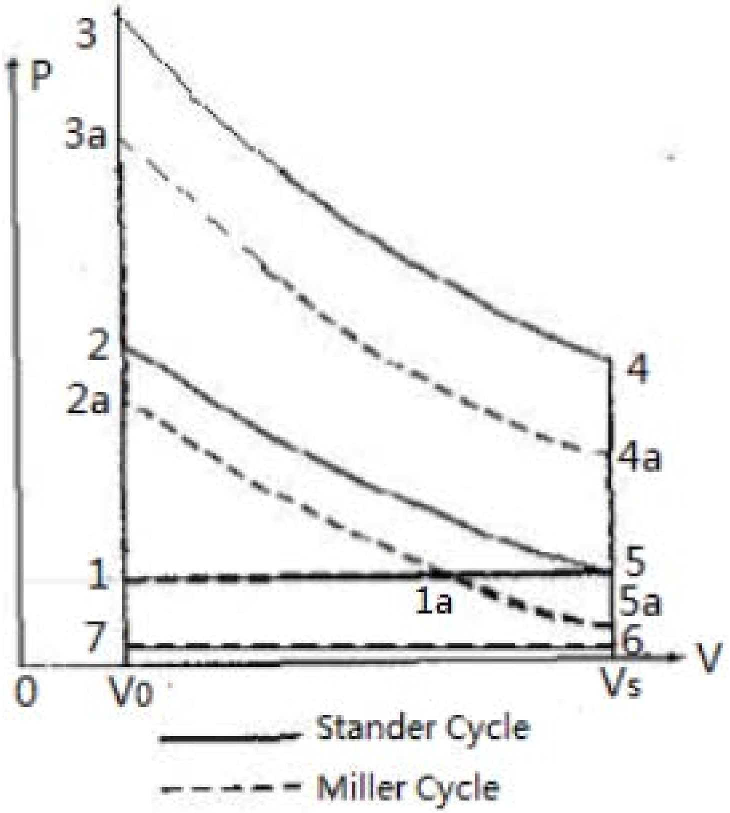

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Figure 1: Figure 1 :

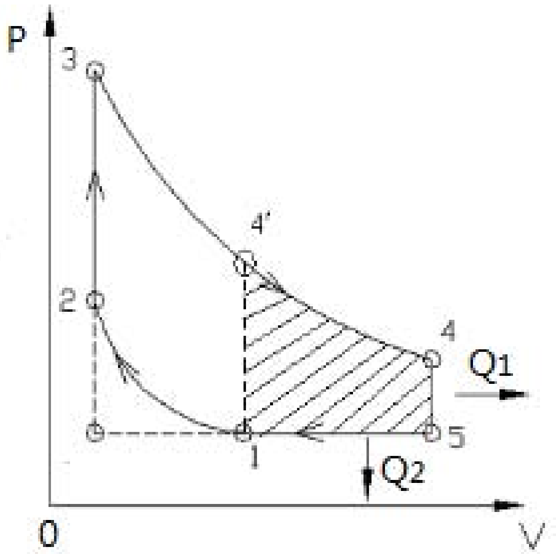

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Figure 2: Figure 2 :

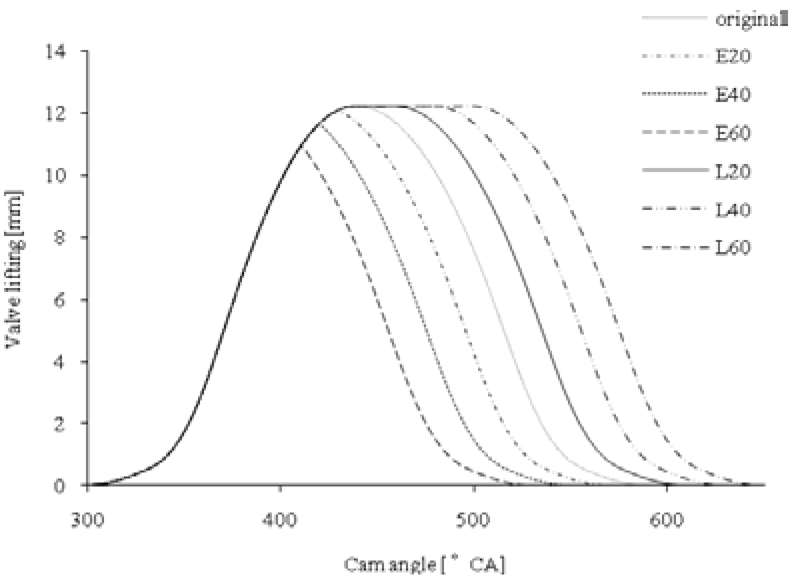

Figure 3: (

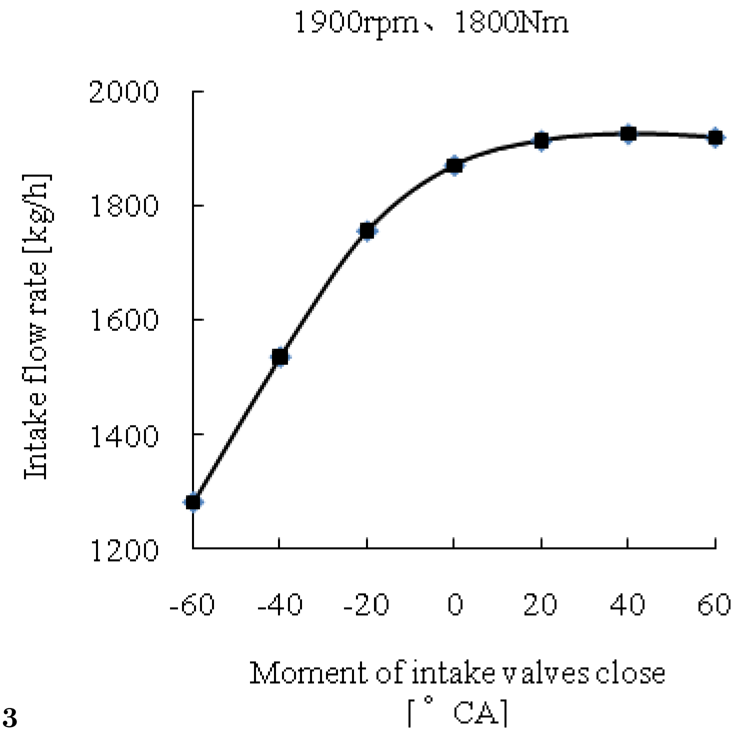

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Figure 4: Figure 3 :

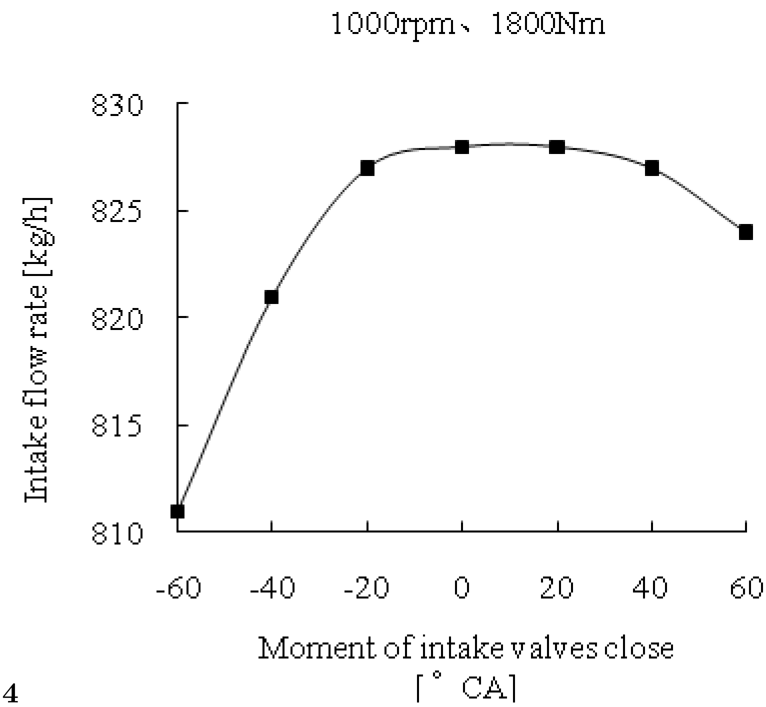

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Figure 5: Figure 4 :

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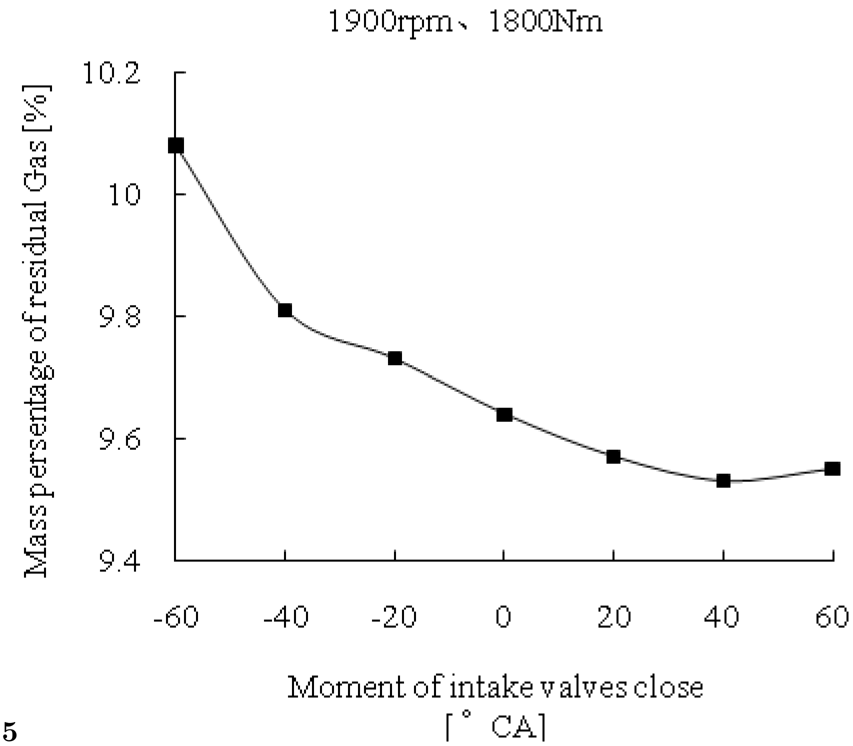

Figure 6: Figure 5 :

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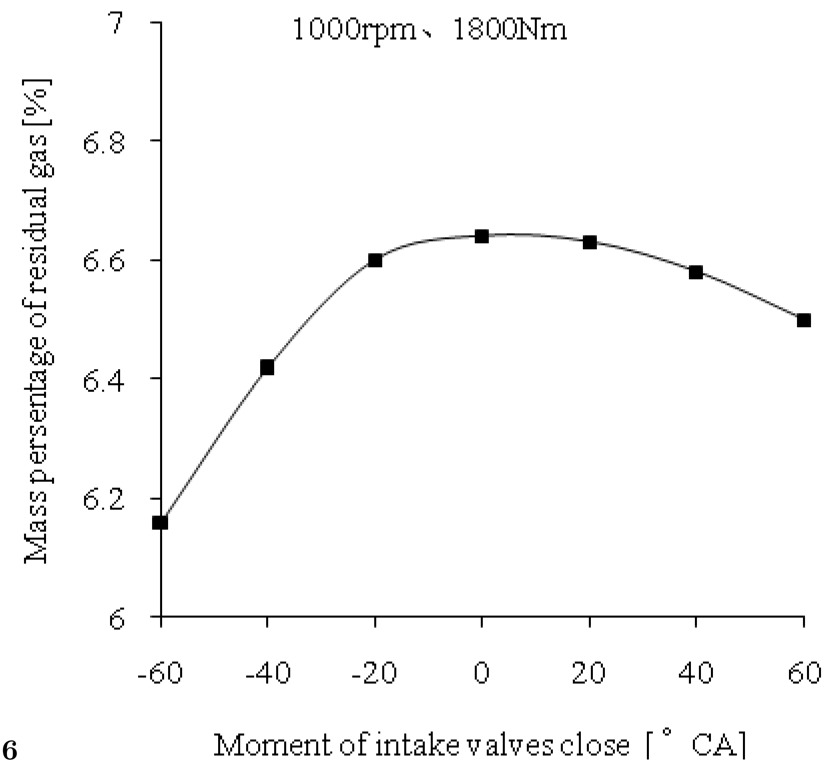

Figure 7: Figure 6 :

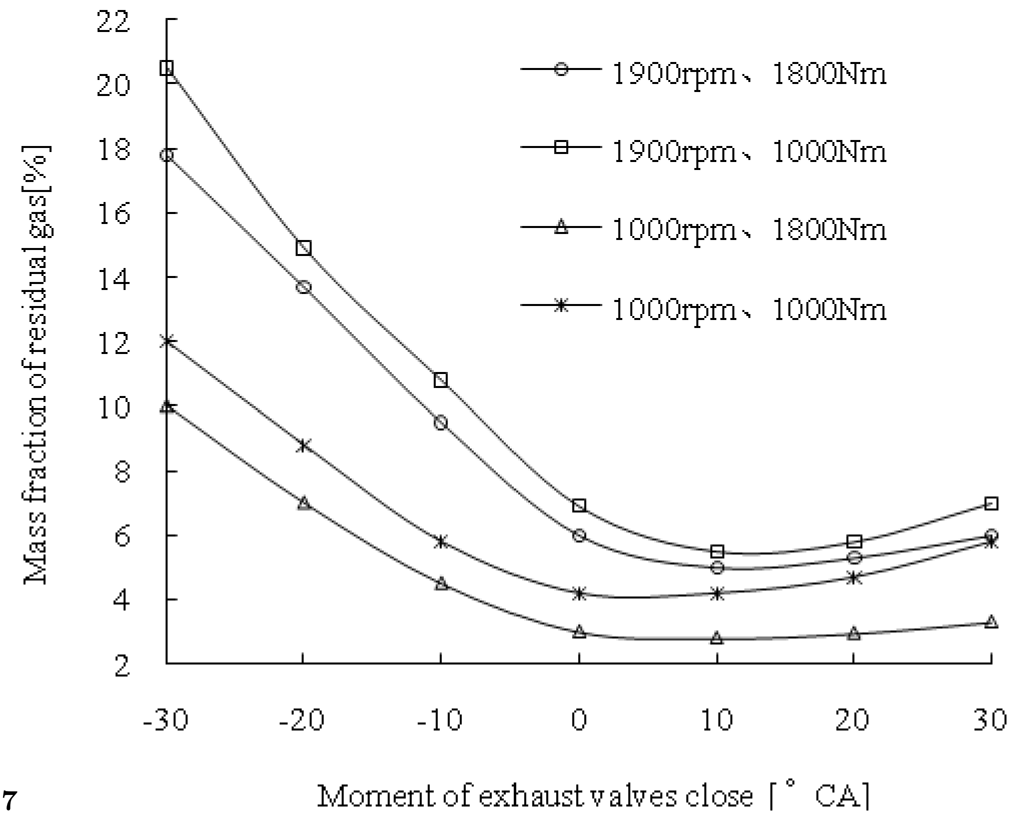

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Figure 8: Figure 7 :

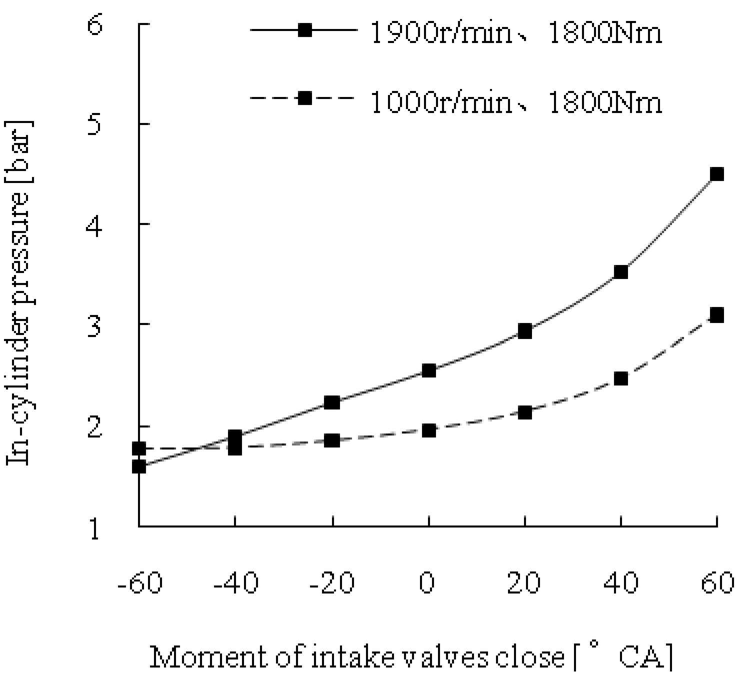


Figure 9: x

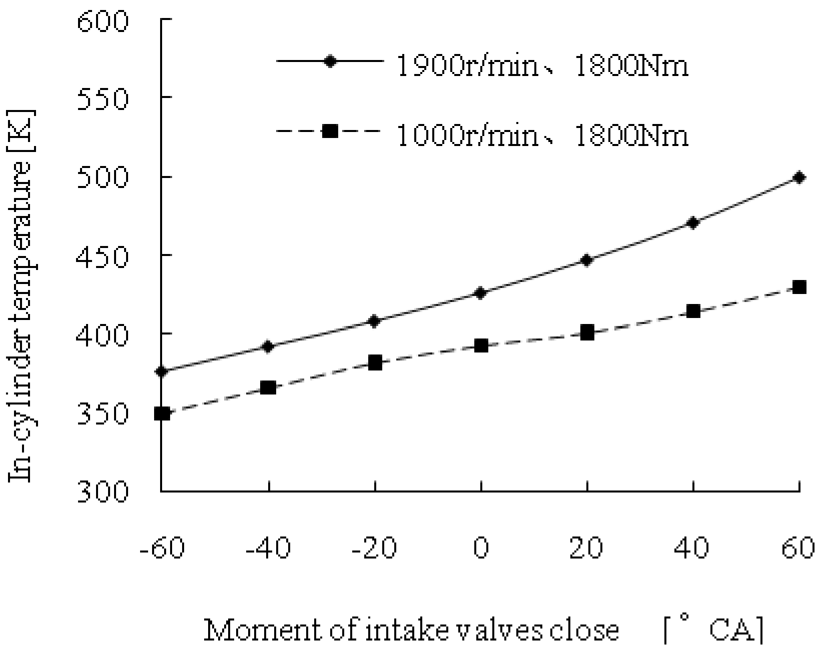


Figure 10:

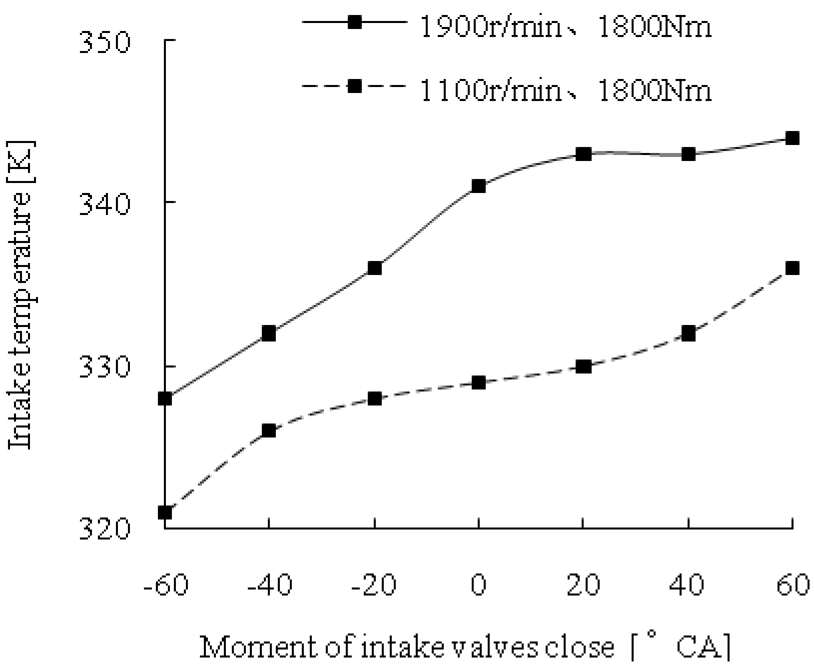


Figure 11:

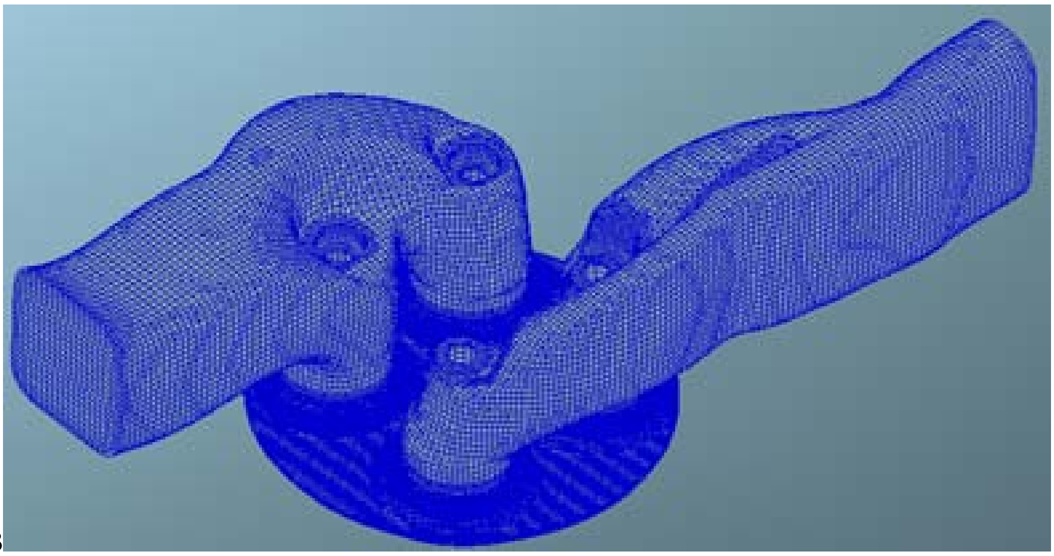
8 

Figure 12: Figure 8 :

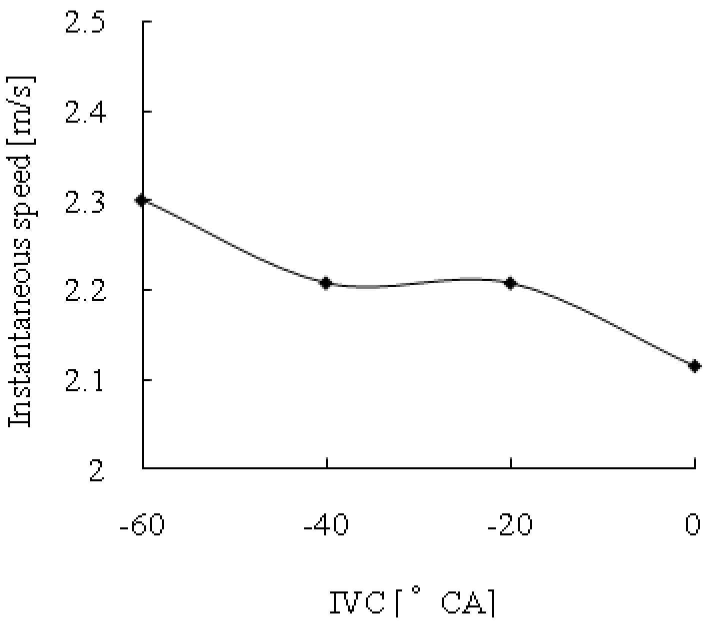


Figure 13:

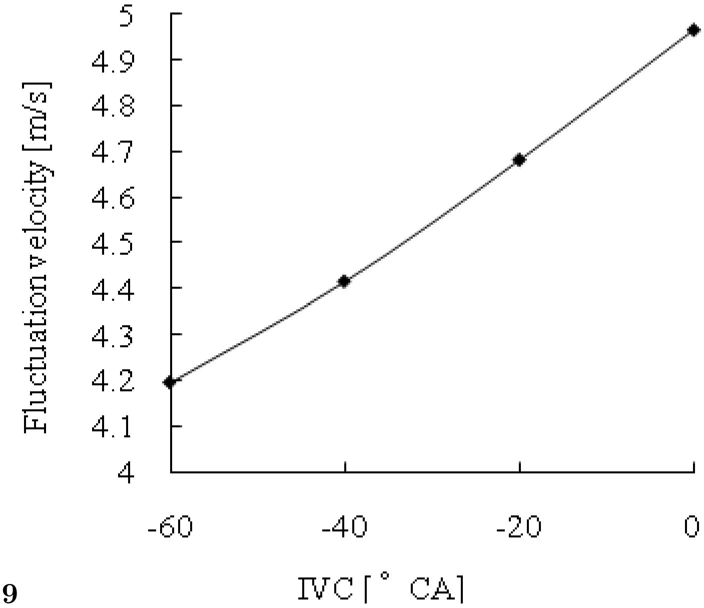


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Figure 14: kTClFigure 9 :

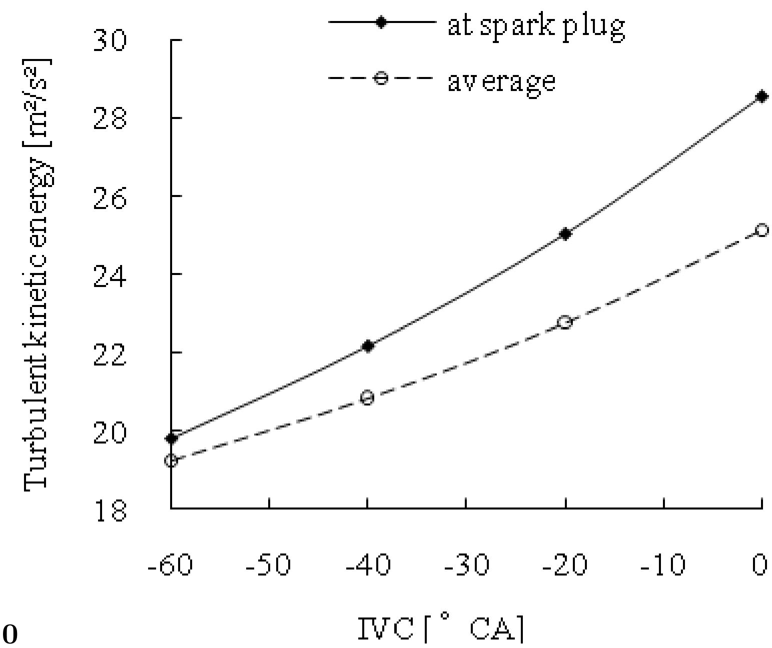

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Figure 15: Figure 10 :

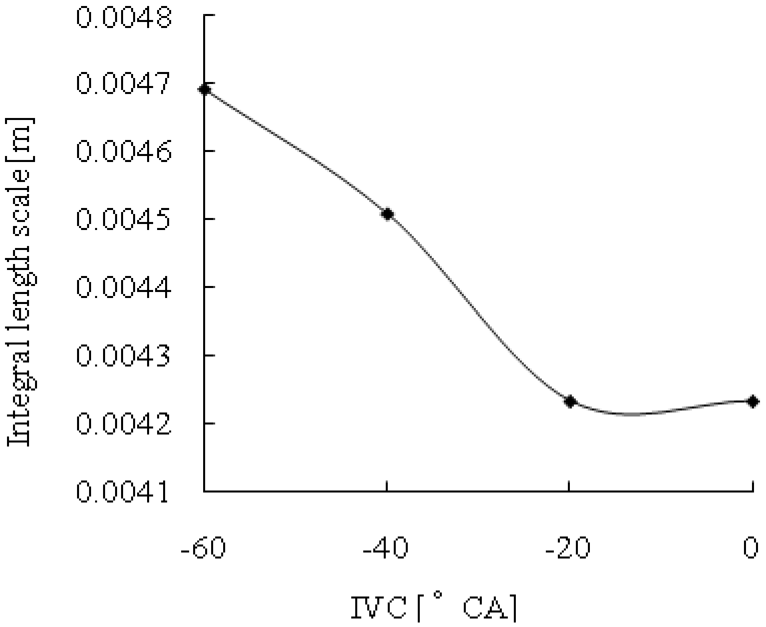

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Figure 16: Figure 11 :

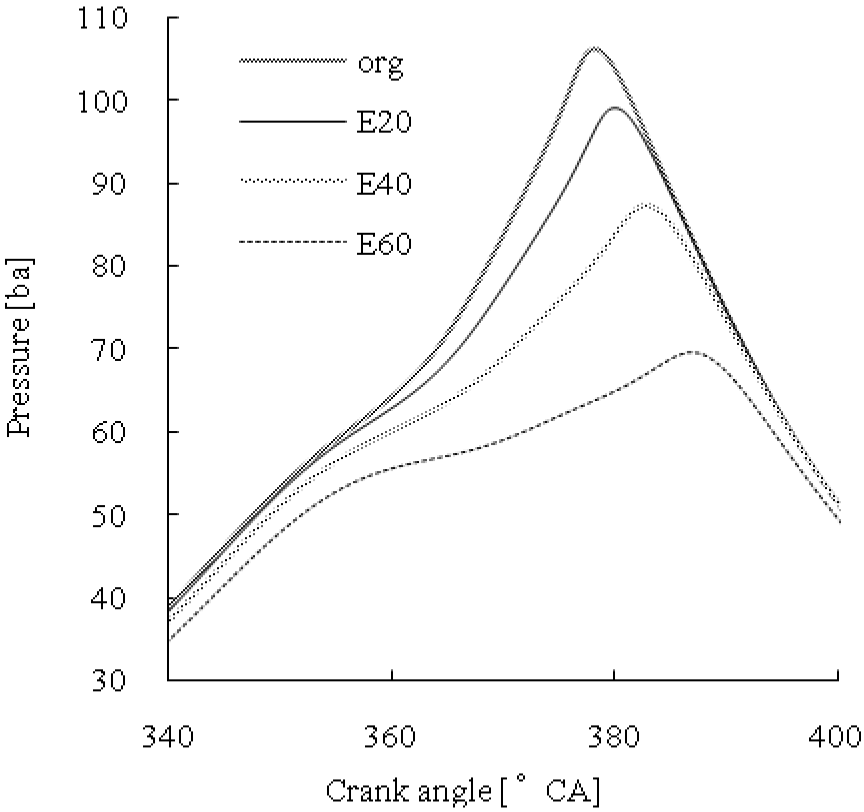

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Figure 17: Figure 12 :

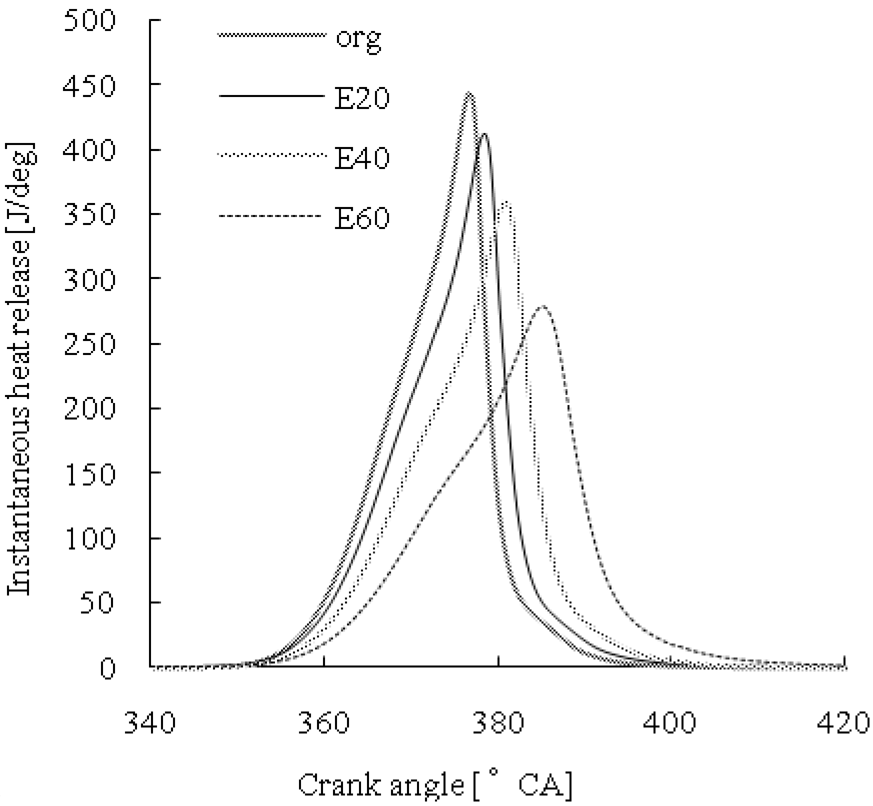

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Figure 18: Figure 13 :

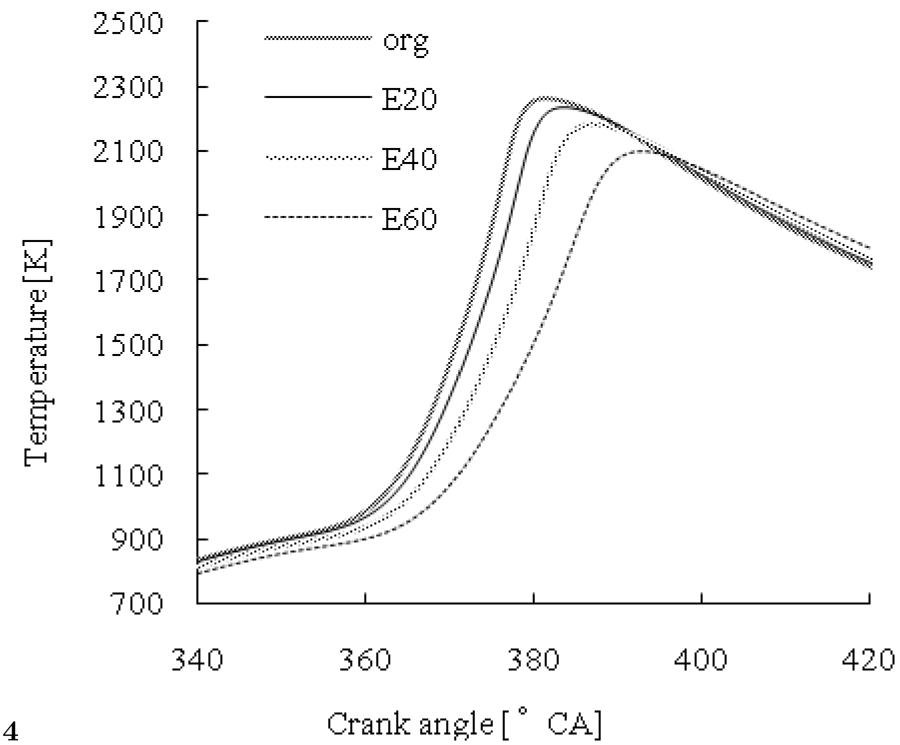

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Figure 19: Figure 14 :

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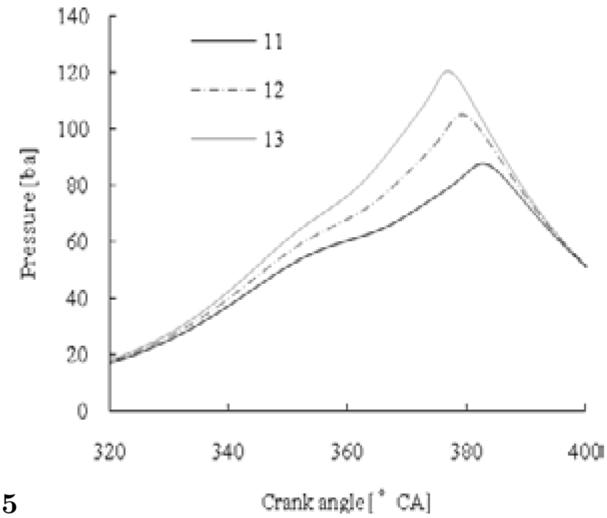

Figure 20: Figure 15 :

1

Number of Cylinders	6
Bore/ mm	129
stroke/ mm	165
displacement/ L	12.939
Compression ratio	11 : 1
Rated power/ kW	338
Rated speed/ r/min	1900
a) One-dimensional Numerical Simulation of Miller Cycle	

Figure 21: Table 1 :

2

	Compression Ratio	
??æ¯?"	D(m)	H(m)
11	0.118	0.024
12	0.114	0.023
13	0.109	0.022

Figure 22: Table 2 :

3

Compression ratio		11	12 13	
Turbulence Kinetic energy(m2/s)		34.7	34. 7	36.3
Flow velocity(m/s)		2.42	2.5 4	2.65
Integral scale(m)	length	0.004 96	0.0 92 04	0.00488

Figure 23: Table 3 :

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