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Research Article

Development of a Three-Phase Sequential Turbocharging System with Two Unequal-Size Turbochargers

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A three-phase sequential turbocharging system with two unequal-size turbochargers is developed to improve the fuel economy performance and reduce the smoke emission of the automotive diesel engine, and it has wider range of application than the current two-phase sequential turbocharging system. The steady matching method of the turbochargers and engine and the steady switching boundary are presented. The experimental results show that this system is effective to improve the engine performance especially at the low speed and high load. The maximum reductions of BSFC and smoke opacity are 7.1% and 70.9%. The optimized switching strategies of the valves are investigated, and the surge of the compressor in the switching process is avoided. The switching strategies in the accelerating process are optimized, and the acceleration time from 900 r/min and 2100 r/min at constant torque is reduced by approximately 20%.

1. Introduction

With the continuous development of high power density and wide speed range diesel engines, the requirements of the turbocharging systems matching with the diesel engines are increasing. Sequential turbocharging (ST) system is an effective measure to improve the fuel economy performance and the transient responsive performance and to reduce the smoke emission at low speed.

The ST system consists of two or more turbochargers in parallel, and these turbochargers are put into or out of operation in terms of diesel engine operation points. This system can improve the turbochargers matching with the engine, so the efficiency of the turbocharger and boost pressure are both improved. The equivalent turbine flow area is capable of more than 50% variation in the ST system, so it significantly improves the performance of the engine with high brake mean effective pressure (BMEP) at low speed. Compared to other twin turbo system such as two-stage serial turbocharging system, the parallel sequential system has advantages of packaging and costs due to smaller turbocharger size. It gets a wide range of applications in ships, locomotives, and military armored vehicles, and it is also applied in sports car gasoline engine, diesel trucks, and cars.

The concept of the ST system was presented at first by Brown Boveri [1] at 1946, but the research did not begin to rise until the late 1970s due to technical restrictions. The first published research was from Germany KHD Company [2], but this technique was not applied in any actual product. In the early 1980s, the first commercial application of the ST system was from Germany MTU Company [3]. The first ST system application in car engine was from Borila Sweden Volvo Company [4–6]. This research was based on a two-phase sequential turbocharging with two unequal-size turbochargers which was equipped with Volvo TD121FD six-cylinder diesel engine. The experimental results show that the fuel consumption rate is decreased by 3%, mostly at low speed, the smoke and exhaust temperature before the turbine are also decreased. Esch and Zickhwolf [7] and Hiereth and Prenninger [8] from Germany Porsche Company used two equal turbochargers ST system on the 2.85 L, six-cylinder gasoline engine of Porsche 959 sports car, and it showed that the engine acceleration time from the idle speed to 7000 r/min is decreased by 25%.

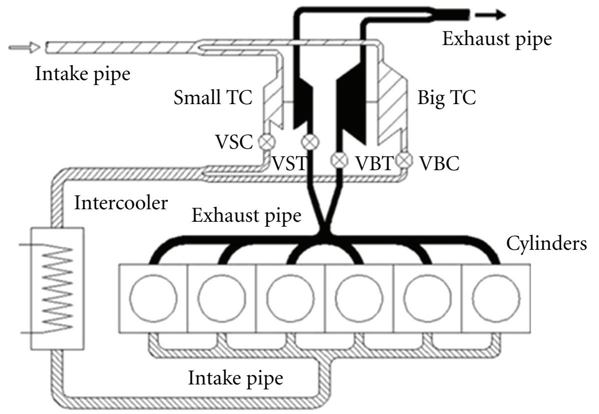


Figure 1: Schematic diagram of the ST system with two unequal-size turbochargers.

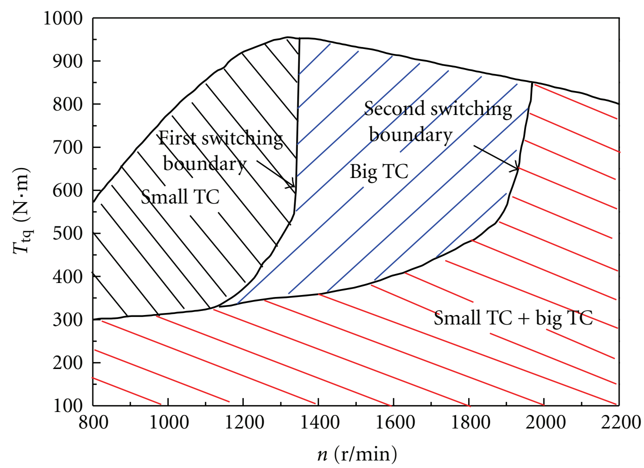


Figure 2: Switching boundaries of the 3-phase ST system.

Tashima et al. [9, 10] from Mazda applied twin sequential turbo system for rotary engine. The twin turbochargers are equal size and the compressors and turbines are modified. The results show that the time that the boost pressure increases to the objective one is 43% less than the time with ordinary turbocharged system. Benvenuto and Campora [11] develop a dynamic model for a high-performance marine diesel engine in order to simulate the dynamic behavior of ST system and related control apparatus. Galindo et al. [12–14] develop transient ST system simulation and experimental research. The sequential turbocharged diesel engine transient response is significantly improved by the switching valves strategies optimization.

The current ST system is generally the 2-phase ST with two equal-or unequal-size turbochargers. When diesel engines work with 2-phase ST, the regulating capacity is still limited and it cannot meet the whole engine operation range [15]. So the measures such as the intake and exhaust bypass, the waste gate, or the bypass and complementary combustion are adopted in order to take into account the whole operation range performance. The more number of the phases of ST system, the performances of the turbocharging system are more suitable to the demands of diesel engines. However, the existing 3-phase ST systems must have three turbochargers at least [16, 17]. The system becomes more complicated and expensive.

In this paper, a novel 3-phase ST system with two parallel unequal-size turbochargers has been presented and studied by experiments. It has wider regulating capacity than the current 2-phase ST system, and it is simpler and more economic than the existing 3-phase ST system.

2. Steady-State Matching Method and Switching Boundary

The working principle of the 3-phase sequential turbocharging system with two parallel unequal-size turbochargers is the small TC works at low-speed operation range of the diesel engine; the big TC works at medium-speed operation range and the two TCs work in parallel at high-speed operation range. In this paper, this ST system is designed for D6114 diesel engine, and the structure of the ST system is shown in Figure 1, and the main specifications of this diesel engine are shown in Table 1.

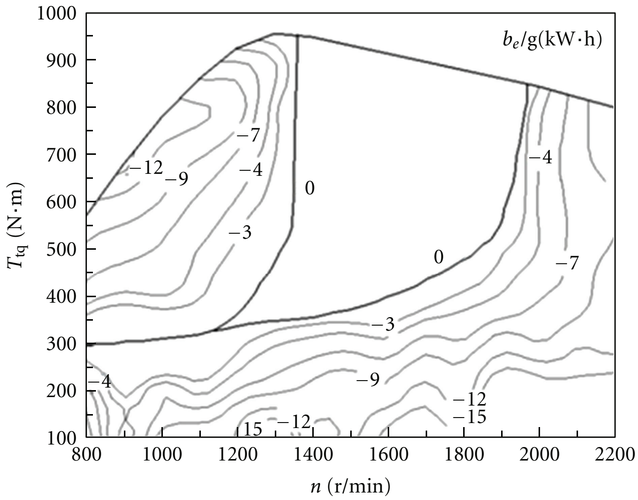
Table 1: Specifications of the test engine.

Type	In-line, 6-cylinder, turbocharged intercooler
Bore × stroke	114 mm × 135 mm
Cylinder volume	8.26 L
Compression ratio	17.7 : 1
Rated power	184 kW (2200 r/min)
Max. torque	955 N·m (1300 r/min)

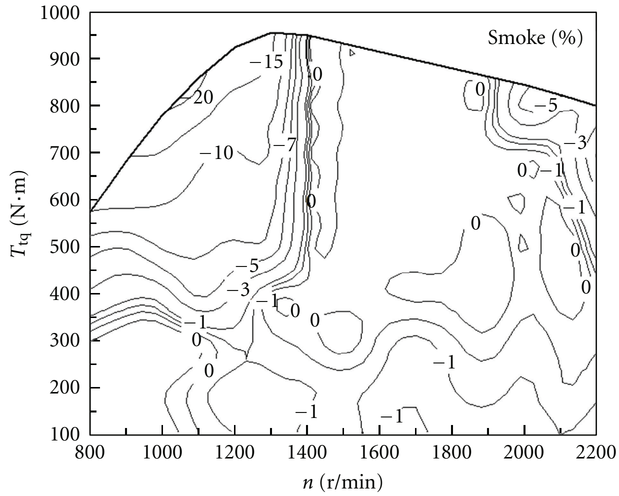
First of all, the process to choose the suitable size turbochargers which match with the engine is as follow: the flow area of the small turbocharger is calculated according to the air-fuel ratio which meets the requirement at minimum speed and full load of the engine; the gross flow area which is equal to the flow area of the big and small turbochargers in parallel is also calculated according to the air-fuel ratio which meets the requirement at rated speed and full load. In this paper, the requirements of the air-fuel ratio at the minimum and rated speed are 22.2 and 26.5. According to the above process to choose turbochargers, the matching result is that the small turbocharger adopts IHI RHF5 (maximum flow capacity is 0.25 kg/s) and the big turbocharger adopts Honeywell TBP4 (maximum flow capacity is 0.38 kg/s).

In the feasible range of the diesel engine and turbochargers, three cases (only the small turbocharger works; only the big turbocharger works; both the two turbochargers in parallel work) are tested at every speed and torque of the diesel engine. The brake-specific fuel consumptions (BSFCs) are compared between these three cases at the same power.

The optimal turbocharger matching strategy is determined in terms of BSFC differences between the three cases and the original TC case (the original TC case indicates a single stage strategy with the big TC), shown in Figure 2. The boundaries are the steady-state optimal switching boundaries.



(a) BSFC



(b) Smoke emission

Figure 3: Comparison of BSFC and smoke emissions between the 3-phase ST and original TC.

Table 2: Valves actions in switching processes.

Switching boundary	Switching process	Valves actions			
		VBC	VBT	VSC	VST
First switching boundary	Small TC⟶big TC	Open	Open	Close	Close
	Big TC⟶small TC	Close	Close	Open	Open
Second switching boundary	Big TC⟶2TC	/	/	Open	Open
	2TC⟶big TC	/	/	Close	Close

The 3-phase ST system is tested in terms of the switching boundary which is shown in Figure 2. The comparisons of BSFC and smoke emissions between the 3-phase ST case and original TC case are shown in Figure 3.

Figure 3 shows that 3-phase ST significantly improves the engine performance, and especially the BSFC and smoke emission are reduced at low speed and high load. The BSFC of 3-phase ST at low speed and high load, at high speed and high load, at all speed and low load are lower than those of original TC. The BSFC of 3-phase ST is 15.9 g/(kW·h) less than that with the original TC at the engine speed of n = 900 r/min and torque T_{tq} = 680 N·m, reduced by 7.1%. Furthermore, at high speed and high load (n = 2100 r/min, T_{tq} = 820 N·m), it is also reduced by 9 g/(kW·h). The smoke emission with 3-phase ST is lower than that with original TC at low speed high load and at high speed high load. The maximum smoke decrease is at n = 1100 r/min and T_{tq} = 860 N·m. The smoke opacity is 20.9%, reduced by 70.2%, compared with the original TC.

3. Valves Strategy in Switching Process

Two switching processes occur at both the first and the second switching boundaries (shown in Figure 2), and the valves actions in these switching processes are shown in Table 2. The four valves are all butterfly valves activated by boost air and their opening and closing process is very quickly (about 0.1 s).

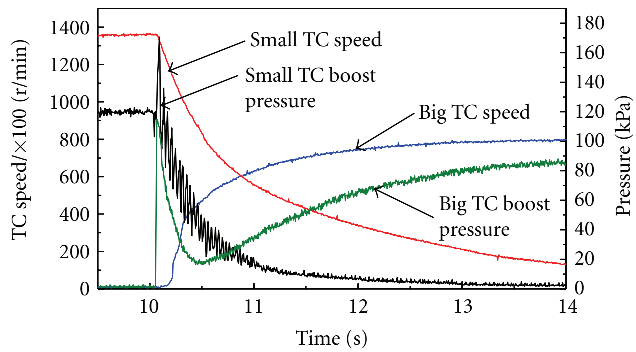


Figure 4: Process from the small TC to the big TC with the four valves activated synchronously.

At speed of 1350 r/min and 100% load on the first switching boundary, the process from the small TC to the big TC is tested. The four valves are activated synchronously, and rotation speed and boost pressure of the two turbochargers are shown in Figure 4.

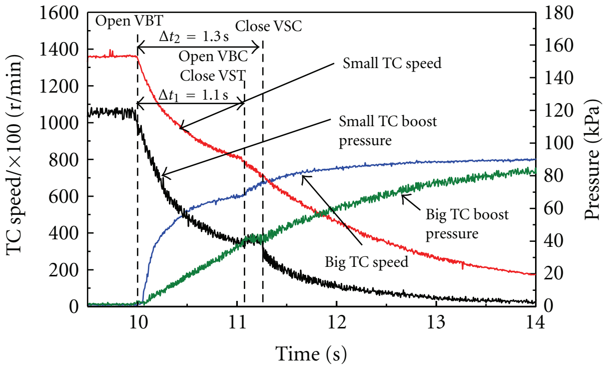


FIGURE 5: Process from the small TC to the big TC with $\Delta t_1 = 1.1$ s and $\Delta t_2 = 1.3$ s.

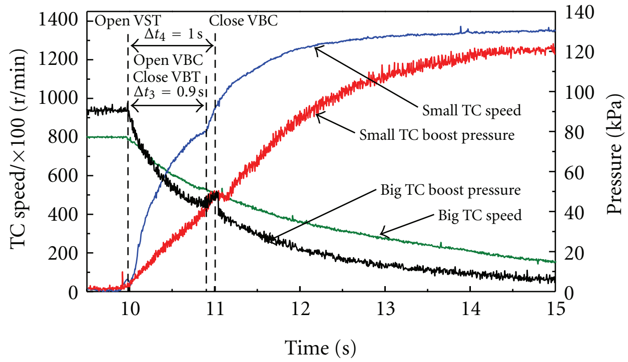


FIGURE 7: Process from the big TC to the small TC at $\Delta t_3 = 0.9$ s and $\Delta t_4 = 1$ s.

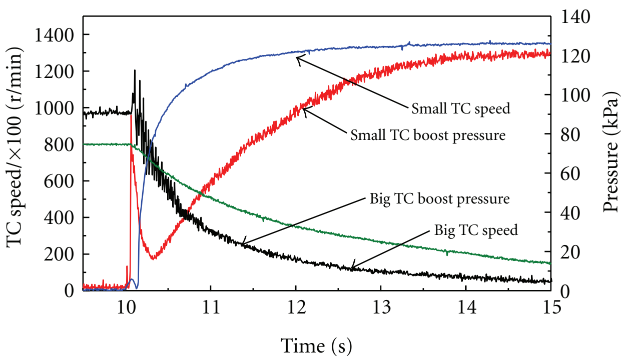


FIGURE 6: Process from the big TC to the small TC with the four valves activated synchronously.

From the switching time on, the compressor of the small TC surges. The surge frequency is approximately 24 Hz and the surge lasts about 1 s. The surge noise is heard at the switching timing. In addition, the pressure after the compressor of the big TC first increases to that of the small TC and then drops quickly after the switching. These are both harmful to the engine operation stability. So the strategy that the four valves are activated synchronously is not reasonable in the process from the small TC to the big TC.

After analysis of the result, the reasonable valves activated sequence is as follow: in the process from the small TC to the big TC, valve VBT (the names of the valves are shown in Figure 1) is opened firstly. It results that the speed and boost pressure of the big TC increase, meanwhile the speed and boost pressure of the small TC decrease. At the timing when the boost pressures after the big TC and small TC are equal, valve VBC is opened and VST is closed. Then VSC is closed at the timing that the small compressor does not surge when VSC is closed. This strategy will make the boost pressure stability and avoid small compressor surge. But if the big compressor begins to surge before VBC is opened, VBC should be opened at the timing when big compressor begins to surge. In the process from the small TC to the big TC, VBT open timing is defined as the reference timing, VBC open and VST closed timing is Δt_1 s later, and VSC closed timing is Δt_2 s later. The optimized Δt_1 and Δt_2 at this diesel engine operation point are determined by the above analysis, and the experimental results of the optimized strategy are shown in Figure 5.

At first VBT is opened at the timing $t = 10$ s, and then VBC is opened and VST is closed when the boost pressure of the big TC and the small TC is equal at $t = 11.1$ s ($\Delta t_1 = 1.1$ s), after that VSC is closed at $t = 11.3$ s ($\Delta t_2 = 1.3$ s). It is shown in Figure 5 that the small compressor does not surge with the optimized strategy ($\Delta t_1 = 1.1$ s, $\Delta t_2 = 1.3$ s) and the boost pressure decrease is 20 kPa less than the strategy that four valves are activated synchronously ($\Delta t_1 = 0$ s, $\Delta t_2 = 0$ s).

Then another switching process at the first switching boundary is from the big TC to the small TC. This switching process is also tested at $n = 1350$ r/min and 100% load. The four valves are activated synchronously, and the result is similar to the process from the small TC to the big TC. The difference is that the surge duration time and pressure oscillation amplitude of the switch-off turbocharger (big TC) are less than the switch-off turbocharger (small TC) in the process from the small TC to the big TC. The big compressor surge lasts 0.5 s from the beginning of the switching process. The results are shown in Figure 6.

So the reasonable valves activated strategy is also similar to the above one: VST is opened firstly and this timing is defined as the reference timing, and then VSC is opened and VBT is closed Δt_3 s later, at last VBC is closed Δt_4 s later. The optimized timing is $\Delta t_3 = 0.9$ s and $\Delta t_4 = 1$ s in terms of the experimental results (shown in Figure 7). The big compressor does not surge and the boost pressure decrease is 25 kPa less than the one with the strategy which four valves are activated synchronously ($\Delta t_3 = 0$ s, $\Delta t_4 = 0$ s).

In the switching process at the second switching boundary, VBC and VBT are always open, so only the strategies of VSC and VST are analyzed. Firstly, the two valves are opened synchronously in the process from the big TC to the two TC at $n = 1960$ r/min and 100% load. The experimental result is shown in Figure 8.

In Figure 8, VSC and VST are opened synchronously at $t = 10$ s, the minimum boost pressure is approximately 50 kPa less than the steady one and the recovery process takes about 3 s. These are harmful to the diesel engine

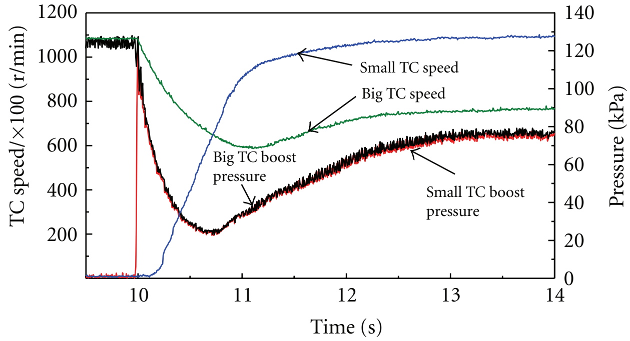


Figure 8: Process from the big TC to the 2TC with the two valves activated synchronously.

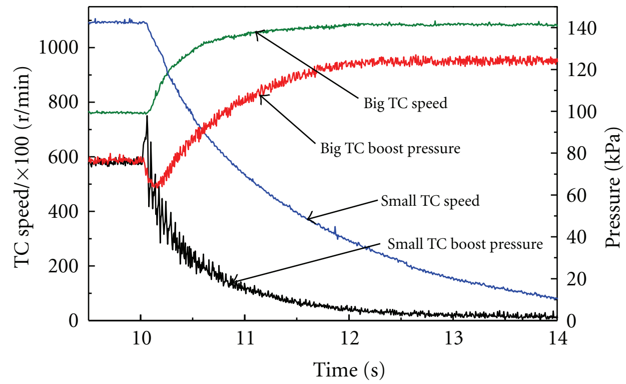


Figure 10: Process from the 2TC to the big TC with the two valves closed synchronously.

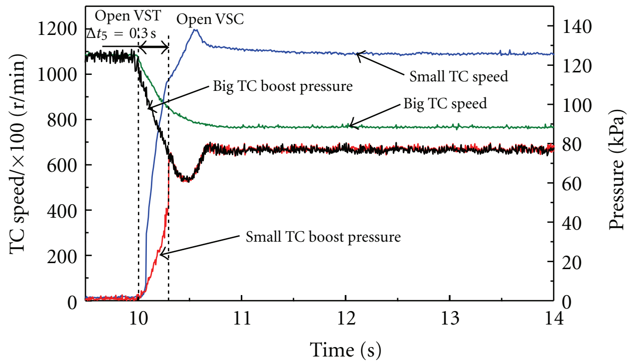


Figure 9: Process from the big TC to the 2TC with $\Delta t_5 = 0.3$ s.

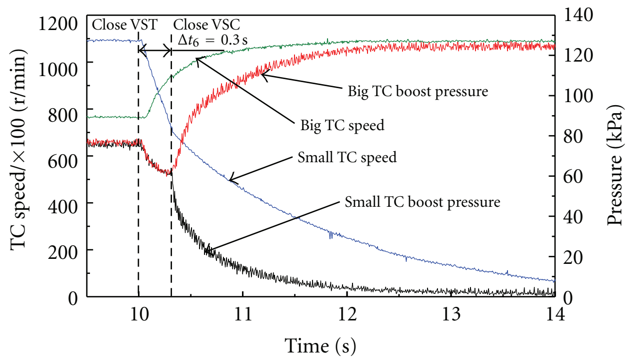


Figure 11: Process from the 2TC to the big TC with $\Delta t_6 = 0.3$ s.

operation stability and it is necessary to reduce the boost pressure decrease in the switching process by reasonable valves activation strategy. The reasonable strategy is: VST is opened at first, so the boost pressure of the small TC increases and the boost pressure of the big TC decreases. When the two boost pressures are equal, VSC is opened. If the small compressor surges before the two pressures are equal, VSC should be opened before the surge occurs. The VST open timing is defined as the reference timing, and then VSC is opened Δt_5 s later. When $\Delta t_5 > 0.3$ s, the small compressor begins to surge according to the experimental result.

So set $\Delta t_5 = 0.3$ s, the result is shown in Figure 9.

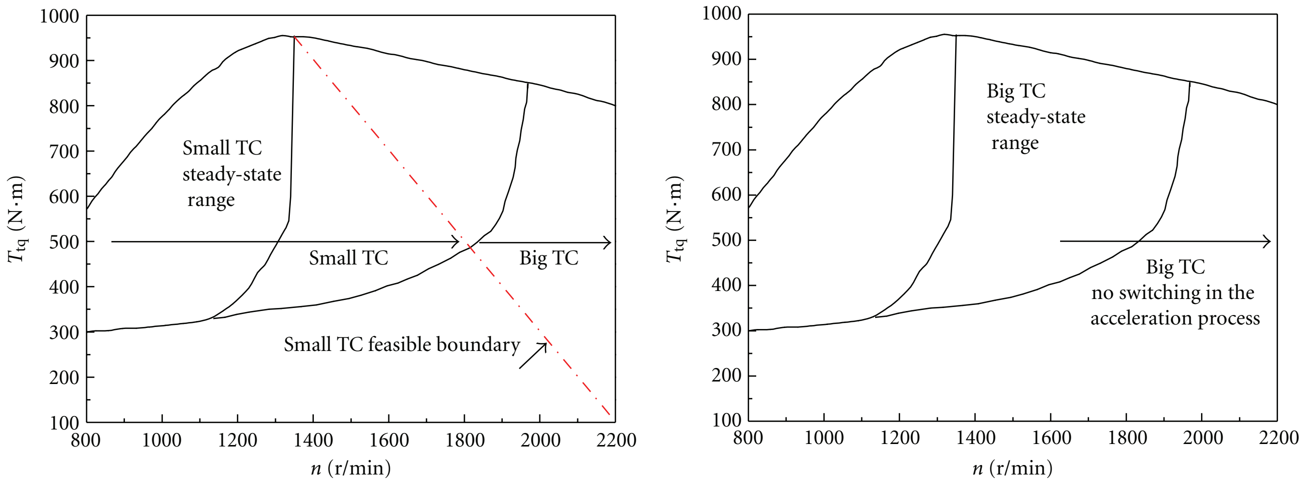
In Figure 9, when VSC is opened 0.3 s later after VST is opened, the difference between the minimum and steady boost pressure is only 16 kPa and the boost pressure recovers to the steady one in less than 1 s. These results are both improved compared with the strategy that two valves are opened synchronously.

In the process from the two TC to the big TC at n = 1960 r/min and 100% load, the two valves for the small TC are closed synchronously. The experimental results show that the small compressor surges and surge lasts about 0.7 s (shown in Figure 10). So the reasonable sequence is: VST is closed firstly, and then VSC is closed. VST closed timing is defined as the reference timing, and then VSC is closed Δt_6 s later. The experimental results show that when $\Delta t_6 = 0.3$ s, the small compressor does not surge (shown in Figure 11).

4. Switching Strategies in Diesel Engine Acceleration Process

The strategies of 3-phase ST are different in the diesel engine acceleration process from the steady-state ones. The main reason is that the two processes focus on different objects. The steady-state strategy is for the lowest BSFC at the steady-state, but the transient responsive performance of the diesel engine is the main considerable specific in the diesel engine acceleration process. So different turbochargers matching cases are studied in the diesel engine acceleration process at constant torque in order to obtain the optimal switching boundaries of 3-phase ST in the acceleration process.

In the engine operation range where small TC is feasible (this feasible boundary is determined by the experimental small TC overspeed limit), three different turbochargers matching cases which are engine with the small TC, with the big TC, and with the ST (the switching boundary is the steady-state boundary, shown in Figure 2) are tested in the



(a) Small TC works at initial case

(b) Big TC works at initial case

Figure 12: Optimal acceleration switching boundary of the ST.

acceleration process at constant torque. The experimental results show that the engine with the small TC takes the shortest time from the initial speed to the target speed; the engine with the ST is inferior to the engine with the small TC; the engine with the big TC takes the longest time. Moreover the differences between the three increase, as the torque increases. The smoke emission and high smoke emission duration time of the engine with the small TC are also the lowest; that of the ST is the second lowest; that of the big TC is the highest.

It is concluded that the engine with the small TC has the best acceleration performance in the small TC feasible operation range. So the optimal acceleration switching boundaries are: if only the small TC works at the initial case and the target case is in the small TC feasible range, the small TC always works; if the small TC works at the initial case but the target case is out of the small TC feasible range, the small TC works until the small TC feasible boundary and then switches to the big TC (shown in Figure 12(a)); if the big TC works at the initial case and the target case is at the two TC steady-state range, the switching process from the big TC to the two TCs is not executed in the acceleration process (shown in Figure 12(b)).

The engine with the above optimal acceleration switching boundary strategy is tested in the acceleration process at constant torque. The results are compared to the results with the steady-state switching boundary strategy. The comparisons of the results with different strategies in the engine speed acceleration process from 900 r/min to 2100 r/min, and the torque is 400 N·m, 500 N·m, 600 N·m, are shown in Figure 13. It is shown in Figure 13 that the acceleration process with the optimal acceleration switching boundary strategy takes shorter time to reach the target speed, and the time difference increases as the torque increases.

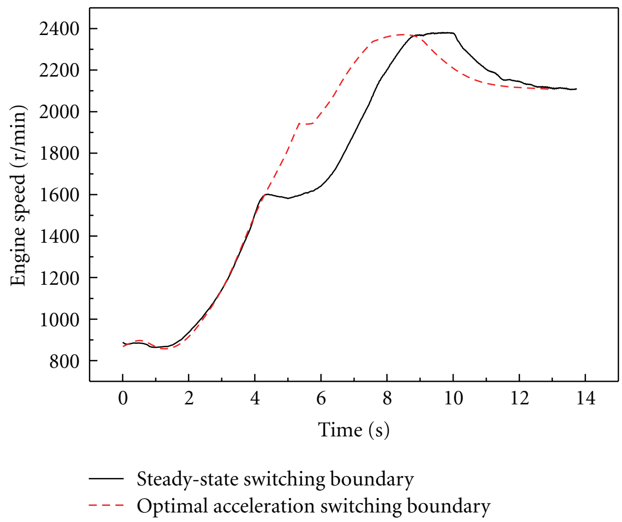
In the acceleration process at $T_{tq} = 400$ N·m, the acceleration time with steady-state switching boundary strategy is 3.23 s, but that with optimal acceleration switching boundary strategy is 2.49 s, so the latter is less than the former by 22.9%. In the acceleration process at $T_{tq} = 600$ N·m, the former is 6.41 s, and the latter is 5.21 s, so the latter is less than the former by 18.7%.

Figure 14 shows the smoke emission comparison of the two ST strategies in the engine speed acceleration process from 900 r/min to 2100 r/min at 600 N·m. The smoke emission with the steady-state switching boundary strategy has three peaks, but that with the optimal acceleration boundary strategy is only two peaks. It is because the former switches twice in the acceleration process (from small TC to big TC and from big TC to two TCs), but the latter switches only once (from small TC to big TC). Although the peak value of the latter is a little higher than that of the former by about 2.1%, the high smoke emission duration time of the latter is 2 s shorter than that of the former. Hence the gross smoke emission of the later is lower than that of the former.

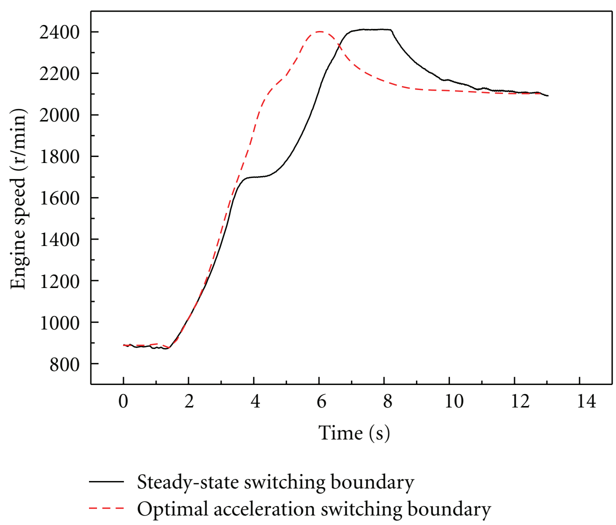
5. Conclusions

(1) A novel 3-phase sequential turbocharging system with two parallel unequal-size turbochargers is presented and the steady-state switching boundary is determined according to lowest BSFC principle. The experimental results show that the engine with this system is improved especially at the low speed and high load. The BSFC is reduced by 15.9 g/(kW·h) at 900 r/min and full load and is 7.1% less than that of the original TC. The smoke opacity is 20.9% at $n = 1100$ r/min and $T_{tq} = 860$ N·m, and it is reduced by 70.2% than that with the original TC.

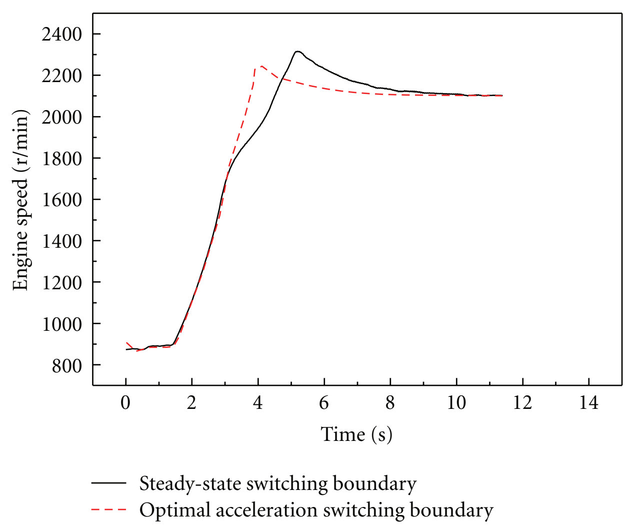
(2) The optimized valves activation strategies in the switching process are presented to avoid the compressor surge and reduce the switching time and boost pressure decrease. In the process from the big TC to the two TC, the optimal valve strategy reduces the switching time from 3 s to 1 s and the boost pressure decrease from 50 kPa to 16 kPa.



(a) 600 N·m



(b) 500 N·m



(c) 400 N·m

FIGURE 13: Comparison of the two ST strategies in the engine speed acceleration process from 900 to 2100 r/min at constant torque.

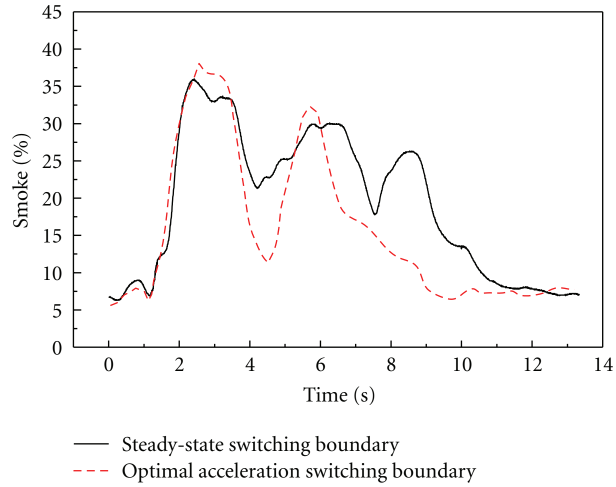


FIGURE 14: Smoke comparison of two ST strategies in the acceleration process from 900 to 2100 r/min at 600 N·m.

(3) The optimal acceleration switching strategies are presented in the acceleration process at constant torque for improvement of the engine transient responsive performance. The optimal acceleration switching boundary is the small TC feasible boundary.

(4) The acceleration experimental results show that the engine transient responsive performance is improved with the optimal acceleration switching boundary strategy. In the engine acceleration process from 900 r/min to 2100 r/min and torque 600 N·m, the acceleration time is reduced by 18.7%.

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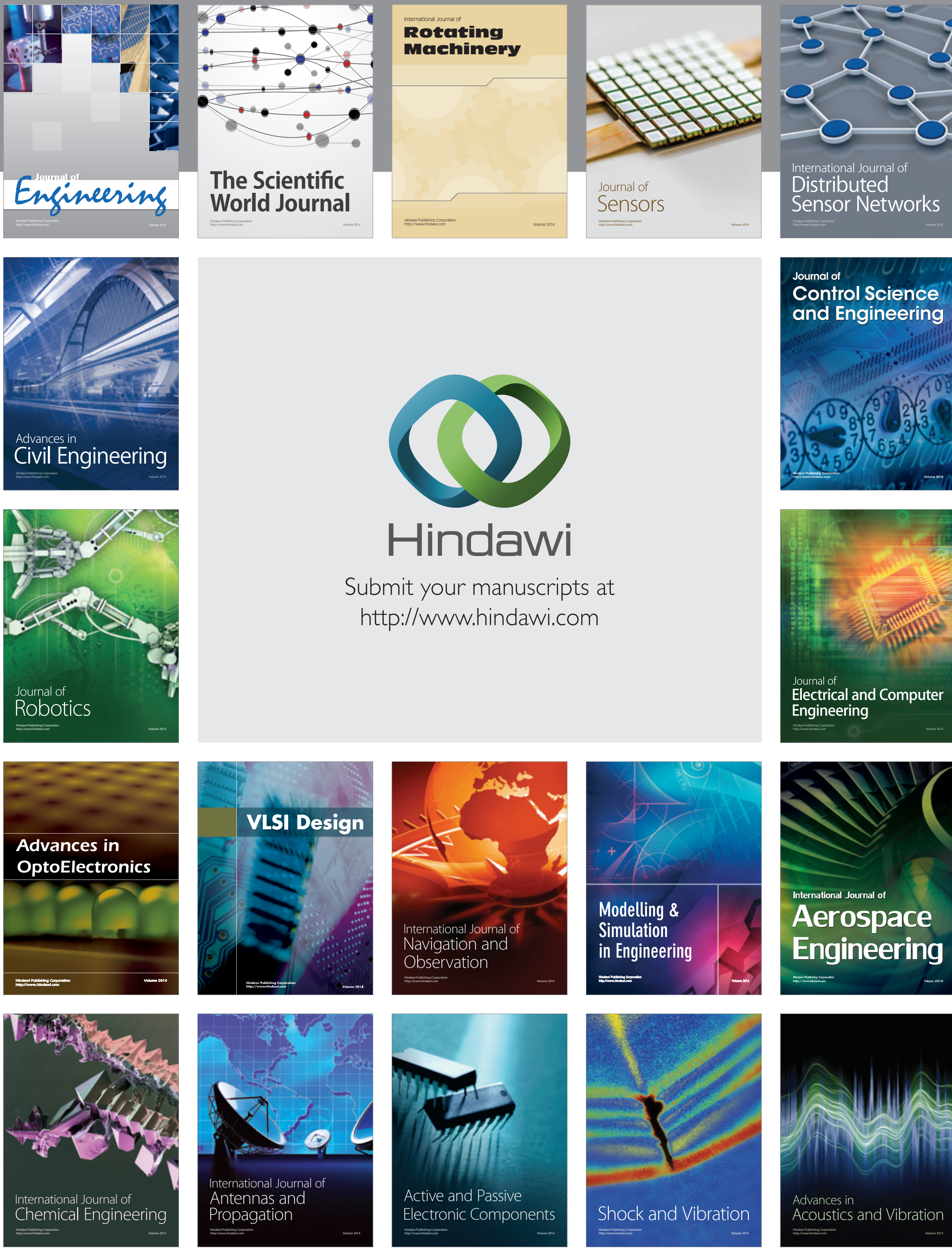
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