A Study on the Applicability of a Mechanical Supercharger to a Diesel Engine for the Commercial Vehicle

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ABSTRACT

A supercharging system comprising of a mechanical supercharger and a turbo charger was investigated with experimental and numerical analysis in a diesel engine for use in a commercial vehicle. The frictional losses of the mechanical supercharger were reduced by putting it together with a turbo charger. Furthermore, it was found that the fuel consumption of the vehicle could be improved by optimizing power train specifications. This was accomplished due to the increased low-end torque which was enabled by a mechanical supercharger.

INTRODUCTION

Diesel engine technology has advanced remarkably in recent years with improvements in emissions, noise and vibration while keeping advantage of low fuel consumption. Supercharging and exhaust gas recirculation (EGR) are important technologies for realizing both good fuel efficiency and low emission performance, therefore most diesel engines are combined with a turbocharger and EGR [1, 2]. Turbochargers utilize the exhaust energy, so there is almost no increase in boost pressure while the engine is operating at low speed or under low loads due to the small exhaust flow volume. Acceleration from this condition requires time for the turbine speed to rise, which causes an increased delay in boost pressure called “turbo lag”. The boost pressure under the acceleration condition will be lower than the boost pressure of the steady state. Improving the supercharge properties during transient mode such as above is an important issue for improving emissions and fuel consumption performance. Previously there have been reports of combining a mechanical supercharger with a diesel engine to improve the transient torque [3]. However this system had problems due to the difficulty of EGR control and an increase in mechanical loss. There have not been many research reports on the use of a mechanical supercharger for emissions control after exhaust emission regulation levels became stricter.

Engine performance and emissions performance of a 3.0L diesel engine combined with a mechanical supercharger were investigated in a previous paper [4], and obtained the following findings. Using a mechanical supercharger improved the supercharging properties during low exhaust volumetric flow and achieved good low end torque comparable to a 4.8L naturally aspirated engine. Further, improving the transient supercharging properties reduced NOx emissions by 50 % at the same soot level in the Japanese emission test cycle (JE05) compared to turbocharger systems calibrated for the

Japanese new long-term regulation. However, an increase in frictional loss of the mechanical supercharger increased the fuel consumption rate by 5.5 %.

To improve the fuel consumption, the technologies of mechanical supercharged gasoline engines were referenced and the following three methods were proposed on the previous paper [4].

(a) Multi-stage supercharging using a combination of mechanical supercharger and the turbocharger: improving the weak points as well as keeping advantages of the mechanical supercharger.

(b) Down speeding of engine operation points: each gear ratio of the power train is modified to reduce engine friction to improve fuel economy of a vehicle.

(c) Downsizing of engine: reducing an engine friction by using smaller displacement engine, and using mechanical supercharger to compensate for low-speed torque.

Recently there have been some studies on diesel engines and reports of similar concepts [5, 6]. However, there is not much information because the research just began in this research area. In this study, above items of (a) and (b) were investigated and the effect of these methods were confirmed to improve the reduction in fuel consumption caused by the frictional loss that is an issue for using mechanical superchargers in diesel engines for commercial vehicles.

EQUIPMENT AND METHOD

The specifications of the tested engine are given in Table 1, and the test engine's supercharging system schematic diagram is shown in Fig.1. An in-line 4 cylinder 3.0L diesel engine for a heavy duty vehicle was used. The original engine had been calibrated for the Japanese new long-term emissions regulation and was equipped with a common rail fuel injection system, turbocharger with an intercooler, water-cooled EGR

cooler, and an air intake throttle. As shown in Fig.1, the supercharging system used for this research is a 2-stage supercharging system (Turbo-Super system) in that the air is supercharged by a turbocharger first and then further supercharged by a mechanical supercharger. The specifications of the mechanical supercharger are the same as a previous study [4] and it is driven by the engine crankshaft via a ribbed V-belt. A mechanical supercharger was selected from the maker's line-up by a simulation analysis that predicted the best fuel consumption rate. The mechanical supercharger has a bypass circuit that connects the inlet and outlet, and the boost pressure is controlled by adjusting the bypass valve position in the bypass circuit. The turbocharger is a variable geometry turbocharger (VGT) equipped with a variable nozzle inside the turbine housing and is the same as the previous paper [4]. In this study, the variable nozzle vane position used fixed angle to use the turbocharger as a conventional turbocharger. For the EGR system, a low-pressure EGR loop that circulates the exhaust gas in the compressor low-pressure side was used. The EGR gas first has the particulate matter (PM) in the exhaust removed by a catalyzed DPF (C-DPF) and then is cooled in the EGR cooler before being circulated in the intake air system. The injection control calibration

parameters (fuel injection timing, pilot injection, etc) were used to the Japanese new long-term regulation values except where they interfered with engine operation, and the air intake system conditions (boost pressure, EGR rate) were optimized for each system. The emissions measurement system is the same as the previous study [4]. As test fuel, category No.2 diesel fuel as defined in Japanese Industrial Standards (JIS) was used.

RESULTS AND DISCUSSION

2-stage supercharging concept using a turbocharger and a mechanical supercharger

Fig.2 shows the Turbo-Super system concept used for this research. Assuming conditions that increase the load in steps under the operating conditions of an engine, a turbocharger will require some time until the boost pressure rises. On the other hand, a mechanical supercharger is always discharging a constant flow that corresponds to the rotational speed of the crankshaft, and the boost pressure adjusted with the bypass circuit valve position only. A time lag until the target boost pressure is reached does not exist. In this concept, a mechanical supercharger is used for supercharging mainly during the period when the boost pressure from the turbocharger has not yet reached the target. As the turbocharger boost pressure increases, the workload of the mechanical supercharger is reduced and also the frictional loss caused by the mechanical supercharger is reduced. When the engine is in a steady operation state, the pumping work of the mechanical supercharger is reduced, and this correspondingly reduces the increase in fuel consumption by approaching frictional losses only. And this concept can achieve the required EGR rates because of realizing boost pressure almost the same as a steady engine operation condition even during transient conditions. As the results, it is able to reduce the NOx emission in the transient cycle.

Table 1 Test engine specifications

Engine type	Inline 4 cylinders DI diesel
Valve train	DOHC 4 valves (2 inlet / 2 exhaust)
Bore x Stroke	95.4 x 104.9 [mm]
Displacement	2,999 [cm ³]
Compression ratio	17.5
Injection system	Common Rail System
Air-charging system	Inter cooled turbocharger + supercharger

Fig.1 Schematic layout of examined supercharging system (Turbo-Super system)

Fig.2 Concept diagram of Turbo-Super system

Examination of a Turbo-Super system with a computational simulation

A computational study was performed to estimate the fuel efficiency improvement effect of this system. The computational 1D-CFD code used "WAVE" which is developed by Ricardo. The WAVE model was made for the system shown in Fig.1 and the computational results are shown in Fig.3. The computation was made by changing the fuel mass flow rate at an engine speed of 1,600 rpm. The figure shows the simulation results of the standalone turbocharger (Turbocharger), the standalone mechanical supercharger (Supercharger), and 2-stage supercharging system which combines the turbocharger and the mechanical supercharger (Turbo-Super). The engine and pipe model in the WAVE simulation were improved and verified based on the results of the standalone turbocharger and standalone mechanical supercharger in a previous study [4]. The boost pressure, pumping mean effective pressure (PMEP), mechanical supercharger mean effective pressure of friction loss ((FMEP)sc), and the relative brake specific fuel consumption (Relative BSFC) are shown in Fig.3. The

boost pressure of the Turbo-Super system was calculated the same as the boost pressure of the turbocharger only by adjusting the bypass valve position. In the case of the mechanical supercharger, the PMEP is reduced and became positive work because of no existing exhaust turbine, but mechanical loss was generated which was not present in a turbocharger. Using the Turbo-Super system reduces the mechanical loss of the mechanical supercharger because the turbocharger performs the supercharging work, but the positive pumping work is reduced because the turbocharger increases the exhaust port back pressure. The combination of the pumping work and mechanical supercharging work created a negative value for all systems. Systems using a mechanical supercharger increased the negative work compared to a turbocharger, but using a Turbo-Super system slightly reduced the loss. The difference in this loss decreases as fuel flow increases, and will become nearly the same as a turbocharger at high load. Looking at the relative BSFC, the mechanical supercharger system increases by close to 10 % at low loads, but as the load increases, the loss difference decreases and the Turbo-Super system becomes asymptotic with the turbocharger system. In the case of standalone mechanical supercharger, the fuel consumption rate increased about 4 % even at high loads. It is considered that is due to a decline in the excess air ratio of the combustion area caused by the boost pressure not rising sufficiently in the case of a mechanical supercharger. With the Turbo-Super system, it is possible for the boost pressure to make the same value as a turbocharger, and the fuel consumption rate at high loads became equivalent to that of a turbocharger.

The simulation results of the Turbo-Super system predicted improved boost pressure properties and fuel consumption rate. In the next step, engine tests were performed which verify this system.

Full load performance on the Turbo-Super system

Fig.4 shows the full load performance test results using a Turbo-Super system. The figure shows the relative BSFC, full load torque, and boost pressure, and the results for a turbocharger and a mechanical supercharger from the previous study [4] are also shown. The maximum torque was limited with 375 Nm which is same as the maximum torque of a turbocharger that had been considered within the limit of the vehicle's power train. The figure shows that in the Turbo-Super system it is possible to increase the full load torque in the engine speed range of 1,000 to 1,400 rpm compared with the mechanical supercharger. The fuel consumption rate was improved from a mechanical supercharger only, and the difference of fuel consumption is less than 3 % from the turbocharger at engine speeds of below 2,000 rpm. Also, the fuel consumption rate at an engine speed of 1,600 rpm was nearly the same as the predicted results from simulation. The area of slanted lines in Fig.4 shows the range in which a performance test was not conducted due to concerns with the durability of the supercharger due to the high air temperatures generated at the outlet of the

mechanical supercharger. It is thought that the maximum output power of a turbocharger can be generated if the air to the mechanical supercharger can be cooled by placing an intermediate cooler between the turbocharger and mechanical supercharger, or if the mechanical supercharger can be disengaged by installing a clutch mechanism in the drive circuit of the mechanical supercharger.

Turbo-Super system performance evaluation under the emission test cycle (JE05)

The test results of the JE05 cycle conducted on the Turbo-Super system are shown in Fig.5. The EGR control of the Turbo-Super system was done using the mass air flow (MAF) as the target value, and MAF feedback control was used to adjust the EGR rate. Boost

Fig.3 Simulation results of each supercharging system

Fig.4 Full-load performance of Turbo-Super system

pressure control used feedback control of bypass valve position to adjust the target boost pressure. The target boost pressure and target MAF were adjusted to prevent excessive smoke emission before test. The figure shows the relative soot emission amount and the relative BSFC based on a turbocharger. The results of previous study [4] were also shown in this figure. A micro soot sensor [7] was used to measure the mass flow rate of soot during transient mode and the total amount of soot was calculated cumulatively during the test cycle. The lines in the figure are the trade-off lines which are obtained by changes in the target MAF with each system. Fig.5 shows it is possible for the Turbo-Super system to reduce by half the NO_x at the same soot level compared to a turbocharger system. However, the fuel consumption rate and soot emission amount trade-off lines overlap the points of a mechanical supercharger system reported in previous paper [4]. Therefore even if the mechanical supercharger combines with a turbocharger, the fuel consumption and the emissions in the JE05 were not improved.

To investigate the reason for the fuel consumption rate not being improved, the energy loss of the pumping work and the frictional loss of the mechanical supercharger were calculated using the measurement data during JE05 cycle operation. The pumping loss was calculated from the P_{MEP} measured concurrently by the combustion analyzer during the emission cycle, and the frictional loss of the mechanical supercharger was calculated using the mechanical supercharger speed and pressure ratio of the boost pressure referenced from the manufacturer's test data. The calculation results are shown in Fig.6. This figure shows that the amount of work lost by pumping is 0.589 kWh in the case of a turbocharger only, and that the pumping work is reduced to 0.217 kWh using the Turbo-Super system, but since the lost work amount from the mechanical supercharger is 0.637 kWh, the difference of total is a 0.265 kWh increase in the work amount. The total work loss is the pumping work reduction and the mechanical supercharger work increase which corresponds to approximately 5 % of the cycle operation fuel consumption amount when the heat efficiency is 40%. It is considered that the main reason for increasing fuel consumption is caused by this work amount increase. The operating points of the JE05 cycle are relatively low load and the work loss of the mechanical supercharger is large at low loads as shown in Fig.3. Therefore the fuel consumption became worse. It is considered that this loss work is reduced by the following method. The nozzle vane position of VGT was fixed in this study. So the pumping loss can be reduced by changing the nozzle valve position depending on the engine speed and load, and also it can be reduced by using a fixed capacity waste gate turbocharger efficient. In addition, the frictional loss of the mechanical supercharger can be reduced by disengaging the mechanical supercharger with a clutch when the boost pressure is increased by only a turbocharger.

Fig.5 Test results of Japanese emission test cycle (JE05)

Fig.6 Comparison of energy losses on JE05 mode

The effects of down speeding to improvement of the fuel economy

The effects of down speeding of engine operating points by revising the vehicle specifications were investigated in this section, which became possible with the increase of the low end torque due to using the mechanical supercharger.

Down speeding involves the problem of lowering the drive force on the vehicle. At first, the generated torque was measured during the transient conditions on the each system. An example of the test results is shown in Fig.7. The figure shows a torque, a boost pressure and a SOOT emission generated by the engine when full acceleration occurs from 10 Nm at the engine speed of 1,200 rpm. The SOOT emission was measured by a micro soot sensor. The fuel injection amount during the transient mode was decreased by smoke limit control to suppress the smoke emission generated before the boost pressure rises, and the fuel injection amount is increased as the boost pressure rises. For these test, the same calibration values were used for the smoke limit control. As shown in the figure, with the Turbo-Super system the boost pressure rises faster and the generated torque increases. However, it is found that the turbocharger requires several seconds to reach the steady state boost pressure due to turbo-lag even if the Turbo-Super system. And the SOOT emission is reduced drastically because the boost pressure of a Turbo-Super system is higher than it of turbocharger system. The engine test was performed as

in Fig.7 between engine speed from 800 rpm to 1,600 rpm. The torque generated after 2 sec from a wide open accelerator were measured. The results are shown in Fig.8. The figure shows that significantly more torque is generated using the Turbo-Super system compared to the turbocharger only system even during the step acceleration condition.

Next, the fuel economy was calculated with the heavy duty vehicle fuel economy calculation program [8] provided by the Ministry of Land, Infrastructure, Transport and Tourism of Japan. The vehicle specifications used for the calculation are given in Table 2, and the vehicle performance diagrams calculated using these vehicle specifications are shown in Fig.9. These specifications, the vehicle category and gear ratio were selected assuming a currently mass-produced vehicle. As the engine maximum torque curves were given the values which are the values of Fig.8 with an engine speed from 800 rpm to 1,600 rpm and a turbocharger system value above 1,800 rpm. The figure shows the vehicle performance from 1st gear to 3rd gear, and for the turbocharger system the final gear ratio is 5.571, and for the Turbo-Super system it is 4.777. Fig.9 shows that the turbocharger system with a final gear ratio of 5.571 and the Turbo-Super system with a final gear ratio of 4.777 have nearly the same maximum driving force at vehicle launching, and so the final gear ratio could be reduced by 14.3 % in the Turbo-Super system.

Fig.10 shows the fuel economy calculation results calculated using the heavy duty vehicle fuel economy calculation program. The fuel economy is calculated from fuel consumption map provided by engine test. Using the results for a turbocharger with a final gear ratio of 5.571 as a baseline, Fig.10 showed the comparison of each system which were the standalone mechanical supercharger, the Turbo-Super system and the Turbo-Super system with gear ratio 4.777. The heavy duty vehicle fuel economy calculation program calculate the weighted average fuel economy between the urban mode as a JE05 cycle and the highway mode which is driving at 80 km/h with changing a load. The high-speed driving ratio is 10 % for the heavy duty vehicle fuel consumption category T3. Compared to the turbocharger system in Fig.10, the calculation results for the mechanical supercharger system showed deteriorating fuel consumption. Further, the rate of deterioration of the fuel economy was larger in the highway mode. Using the Turbo-Super system improved the urban mode fuel economy, but worsened it in the highway mode. It is considered that the pumping loss of a turbocharger is increased by higher exhaust gas flow rate at high engine speed region. The case of the Turbo-Super system with a final gear ratio of 4.777 improved the fuel economy. The weighted average fuel economy is improved by 2.4 % which is caused with the down speeding of engine speed. The fuel consumption maps of the Turbocharger system and the Turbo-Super system were shown in Fig.11 and also show the engine operating points at the fuel economy calculation of the urban cycle in this figure.

Fig.7 Transient properties of Turbo-Super system by step acceleration

Fig.8 Torque measurement results after 2 sec from wide open throttle acceleration

Table 2 Vehicle specifications of fuel consumption calculation

Category of vehicle fuel economy		T3
Gear ratio	1st	5.979
	2nd	3.434
	3rd	1.862
	4th	1.297
	5th	1.000
	6th	0.759
Final		5.571, 4.777
Tire radius [m]		0.351

Fig.9 Vehicle performance diagrams of Turbocharger and Turbo-Super system

Fig. 10 Calculated fuel economy of each system using the heavy duty motor vehicle fuel economy calculation program

Fig. 11 Contour maps of the fuel consumption and engine operating points of simulated driving cycle for each system

The engine operation points are shown about the cases of final gear ratio of 5.571 and 4.777. The fuel consumption is deteriorated by the increase of the drive loss of the mechanical supercharger at region of over the engine speed 2000rpm. Fig. 11 (ii) shows the operation points of the engine moved to the low engine speed and high load direction by down speeding. And it is found that a fuel economy was improved by increasing of the low BSFC points of a higher load and a lower engine speed in the case of the Turbo-Super system.

SUMMARY

This study investigated methods to improve the deterioration in fuel consumption caused by the frictional loss that is a problem when using mechanical superchargers in diesel engines for the commercial vehicle. Two methods (2-stage supercharging of a turbocharger and a mechanical supercharger (Turbo-Super system) and the down speeding of the engine speed) were researched and obtained the following findings.

(1) 1D-simulation results predicted that the Turbo-Super system could improve the fuel consumption

rate from a mechanical supercharger system.

(2) A Turbo-Super system increase the low end torque and greatly improved the fuel consumption rate of full load conditions compared to a mechanical supercharger system. The Turbo-Super system restricts the operational range by limiting the mechanical supercharger outlet temperature, so to obtain the same power as a turbocharger an intermediate cooler must be installed between the turbocharger and the supercharger or it must be possible to disengage the mechanical supercharger by using a clutch.

(3) A Turbo-Super system produced NOx emission reductions of 50 % at the same soot level compared with a turbocharger system in the JE05. However, the fuel consumption and emission performance could not be improved due to the same trade-offs as a mechanical supercharger.

(4) With a Turbo-Super system, it is possible to improve the low end torque generated during in the transient condition and to reduce the final gear ratio of the vehicle. The final gear ratio was reduced within a range that would not cause a loss of start-up performance and the heavy duty vehicle fuel economy could be improved by 2.4 % compared with a turbocharger by reducing the engine speed.

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