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A study on the Engine Downsizing Using Mechanical Supercharger

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One means of fulfilling CO_2 emission legislation is to downsize engines by boosting their power using turbochargers or mechanical superchargers. This reduces fuel consumption by decreasing the engine displacement. When a turbocharger, which is preferable to a mechanical supercharger in terms of fuel efficiency, is used, there is insufficient availability of exhaust gas energy at low engine speeds, resulting in an unfavorable engine response. Therefore, mechanically driven superchargers have increased in popularity due to their quick response to changing speeds in the transient phase. However, since a mechanical supercharger obtains its driving power from the engine, it is difficult to decrease its fuel consumption. This remains a large negative factor for superchargers, despite their excellent dynamic performance. This study aims to develop a power control concept to improve the fuel economy of a mechanical screw supercharger, which could then be used for engine downsizing.

Key Words : Screw Supercharger, Inlet throttle Body, External bypass Valve, Internal Bypass Valve

Nomenclature

EBV : External bypass valve
IBV : Internal bypass valve
TB : Inlet throttle body
n_M : Engine speed (rpm)
n_{SC} : Supercharger speed (rpm)
p_i : Inlet pressure (kPa)
p_o : Outlet pressure (kPa)
p_d : Discharge pressure (kPa)

1. Introduction

The current popularity of internal combustion engines, such as gasoline and diesel engines, originates from the combination of their attractive driving performance due to high torque levels and low fuel consumption. The decisive role of these characteristics has led to the development of charger technology, resulting in a remarkable increase in power density and mean effective pressure (Gollochand Mercker, 2005 ; von Langen et al., 1993 ; Stan, 2002). Turbocharging technology has been widely used in engines for different applications, ranging from small vehicles to large marine vessels, thanks to its technical advantages such as no demand for driving power from the engine and excellent charging effects during static operation at mid to high engine speeds. However, there is a response time delay, especially for passenger car turbochargers, which can be problematic since most driving is done under downtown conditions at mid to low engine speeds resulting in a "turbolag" when there is sudden acceleration. Recently, variable geometry turbocharger, electrically supported turbocharger and two-stage controlled turbocharger have been

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adopted, resulting in good acceleration performance. But the operation principle of the turbocharger itself hinders its performance as well as resulting in unfavorable price factors (Furukawa and Ikeya, 1988 ; Balis et al., 2002 ; Christmann et al., 2005).

Volkswagen developed its own G-Lader supercharger that is mechanically driven to provide a quick response (Kuck et al., 1986). For the same purpose, Daimler-Benz uses a 3-lobe roots Eaton supercharger in its medium-sized cars (Huettebraeucker et al., 1995). Mazda uses a Lysholm screw supercharger for their Miller cycle engine (Goto et al., 1994; Shigeru et al., 1994). Therefore, many car manufactures use superchargers together with turbochargers.

Although superchargers can be found in an ever-widening range of applications, from engines to secondary air pumps in gasoline engines to next generation technology fuel cell vehicles, they suffer from unfavorable fuel consumption compared to turbochargers. One means of overcoming this problem is to decrease the driving power delivered directly from the engine when a fully charged engine is not required, such as when idling, decelerating, or driving slowly. This will be at the expense of reducing the compression work at the supercharger (Lehmann et al., 2000 ; Kemmler et al., 2000; Wiedemann and Kuck, 1987).

Various technical possibilities for reducing the compression work have been investigated and evaluated. The results have demonstrated that this is an effective technique for reducing fuel consumption. This study aims to develop a power control concept to minimize the driving power required from an engine and thereby improve its fuel consumption.

2. Concept of Engine Downsizing

Engines are "downsized" when they possess high power levels, such as a high power density and/or high full-load mean pressure. There are two possible ways to increase the power density, P_e/V_h : increase the engine speed n or the engine effective mean pressure p_{me} via the equation

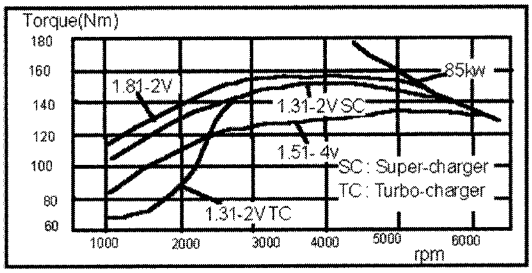


Fig. 1 Performance characteristics of different engine concepts (Kuck et al., 1986)

$$P_e/V_h = i \times n \times p_{me} = 2 \times n \times T/V_h \qquad (1)$$

where P_e is the effective power output (kW), V_h is the piston displacement, i is the number of cycles per crankshaft rotation, and T is the torque (Nm). However, the main objective of downsizing is to obtain a significant reduction in the fuel consumption. Figure 1, which shows an example of engine downsizing, indicates that a 1.3-liter boosted engine can have the same power output (85 kW) as a 1.8-liter standard engine, contributing to economical driving without sacrificing engine power.

3. Concept of Partial Load Control for a Screw Supercharger

3.1 Necessity of partial load control

Fig. 2. shows the relationship between the air quantity and charging pressure for a screw supercharger and a turbocharger at unthrottled conditions. The turbocharger has a slow response to acceleration due to its low charging pressure at low speeds whereas the supercharger has an excellent response because of its relatively stable charging pressure regardless of speed, as shown in Fig. 2 and Fig.3 (Kuck et.al.,1986).

However, when idling, decelerating, or driving slowly, which do not require full charging, the high charging pressure of the supercharger equivalent to the high inlet air quantity causes poor fuel economy in gasoline engines. In these engines, the fuel consumption and emissions are controlled by electronic fuel injection, which regulates the supply of fuel in proportion to the inlet air quantity to obtain perfect combustion

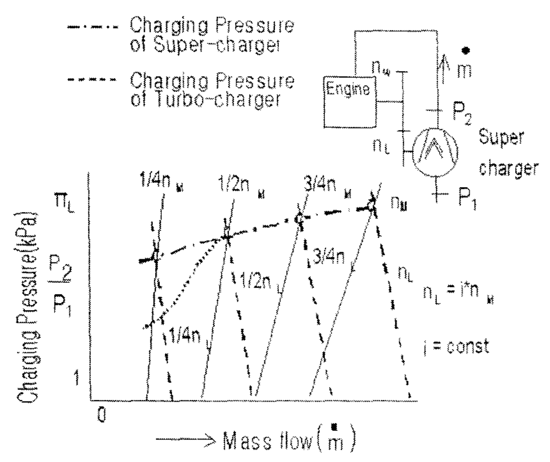


Fig. 2 Relationship of air quantity and charging pressure for supercharged and turbocharged engines at WOT conditions

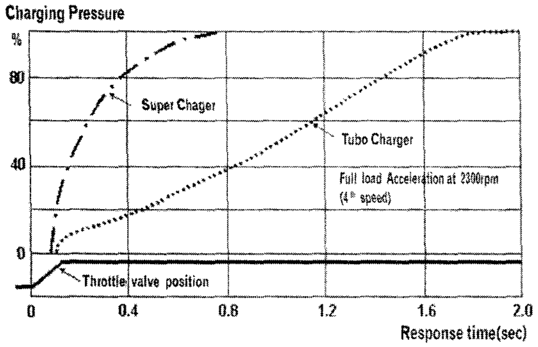


Fig. 3 Response time of different charging system (Kuck et al., 1986)

($\lambda=1$). Therefore, the excessive fuel demand encountered when no charging is required can be improved by decreasing the charging pressure and driving power of the supercharger.

The driving work of a screw supercharger consists of compression and discharge work, as shown in Fig. 4. This driving work W (or adiabatic compression work) can be written as Equation (2) and its power requirement (W) can be reduced if the power demand to operate the two work processes is minimized.

$$W=\int_{p_i}^{p_o} Vdp = \frac{k}{k-1}p_iV_i\left[\left(\frac{p_o}{p_i}\right)^{\frac{k-1}{k}}-1\right]+(p_d-p_i)\,V_o \tag{2}$$

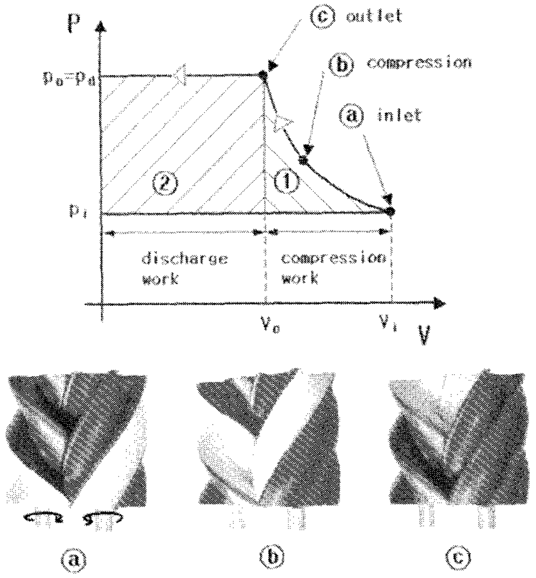


Fig. 4 Pressure and volume curves of a screw supercharger

Here, the first term on the right-hand side of the equation represents compression work (area 1 in Fig. 4) and the second term represents discharge work (area 2 in Fig. 4). The driving work in equation (1) is expressed as a function of the inlet and outlet pressures (p_i, p_o) and the inlet and outlet volumes (V_i, V_o). However, V_i and V_o can be regarded as constants because they are determined by the design of the supercharger in terms of dimensional factors such as rotor diameter, housing diameter, number of rotors, etc. The driving work accordingly will be influenced by only the compression and discharge pressures (p_i, p_o, p_d), which have an impact on the fuel consumption.

3.2 Control method

Figure 5 shows a schematic diagram illustrating how one can vary the compression and discharge pressures to reduce the driving work at the supercharger. In order to vary the inlet (p_i), outlet (p_o), and discharging (p_d) pressures, an inlet throttle body, internal and external bypass valves were installed on our test engine.

This section describes four control methods (a, b, c, and d) that can be used to vary the pressure, as indicated in Fig. 6, and evaluates the

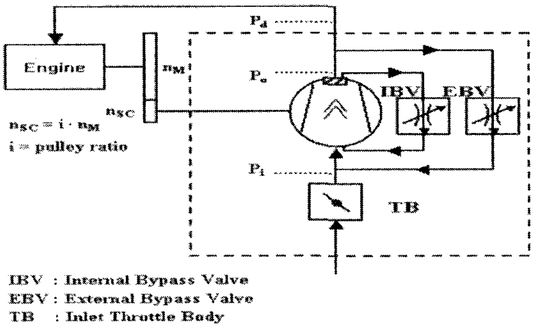
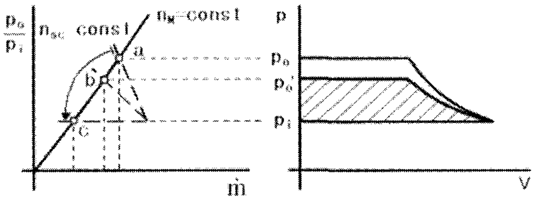
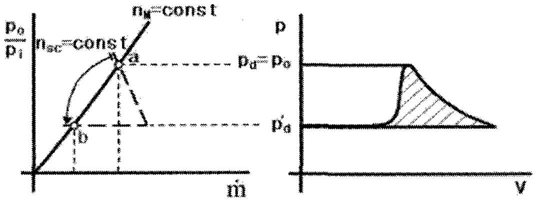


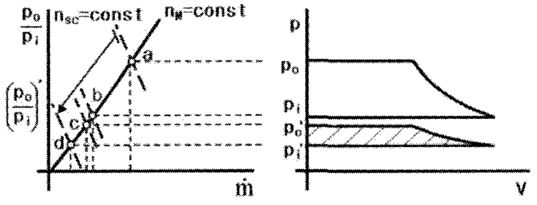
Fig. 5 System concept for partial load control



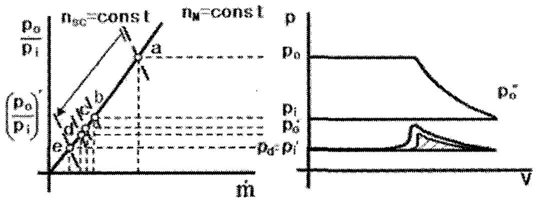
(a) IBV=Open, EBV=Close, TB=Open



(b) IBV=Open, EBV=Open, TB=Open



(c) IBV=Open, EBV=Open, TB=Close



(d) Combination of (a), (b) and (c)

Fig. 6 Schematic investigation for the partial load control concept of screw type

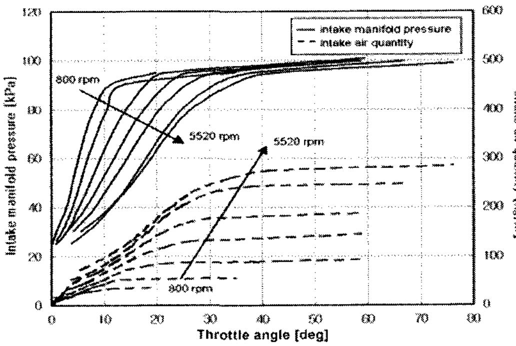


Fig. 7 Relationship between intake manifold pressure and intake air quentity depend on throttle angle

effect of each control idea.

(a) In the compression stroke, air is vented partially through the internal bypass valve (see Fig. 5 and 8(a), (b)) toward the air inlet side so that the outlet pressure p_o after the compression stroke decreases to p_o', which results in a decrease in the compression work.

(b) Air in the process of discharging after the compression stroke is vented through the external bypass valve (see Fig. 5) toward the air inlet side so that the discharge pressure p_d decreases to p_d', which results in a decrease in the discharge work.

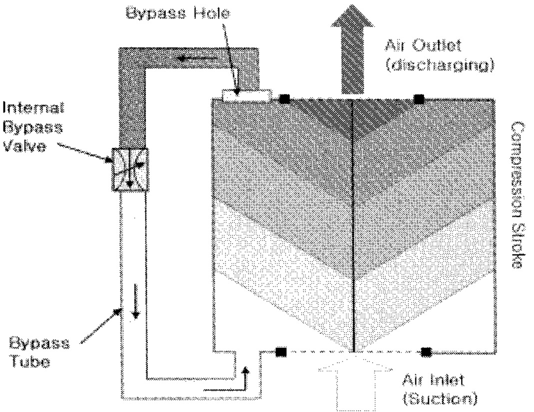
(c) The inlet pressure of air p_i is decreased to p_i' in the range of 25-50 kPa during idling or at low engine speeds, as shown in Fig. 7, by the throttling effect caused by the installation of a supercharger behind the throttle body. This leads to a decrease the in the outlet pressure from p_o to p_o' according to the adiabatic compression equation (3).

$$p_o = p_i (V_i / V_o)^{\kappa} \qquad (3)$$

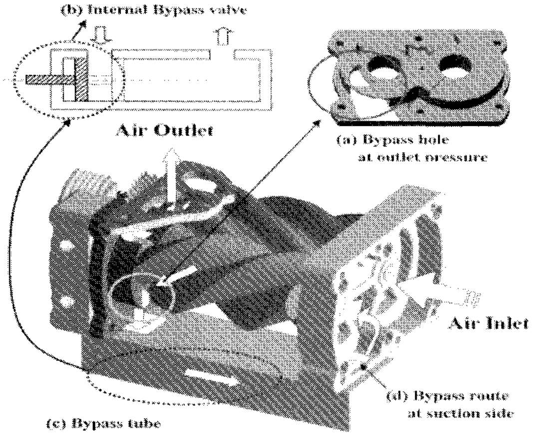
(d) The most effective way to reduce the compression work is to apply all three methods : (a), (b), and (c).

3.3 Design modification for a screw supercharger

An OA 1040 supercharger manufactured by Opcon Auto-Rotor in Sweden was modified as shown in Fig. 8(a), (b) to apply the partial load control concepts described in Section 3.2a). A bypass tube, including a bypass valve, was installed on the supercharger to vent air toward the air inlet side through a bypass hole on the outlet side.



(a) Schematic Diagram of a design modification for partial load control



(b) Design modifications and installation of an internal bypass valve and tube

Fig. 8 Design modification of supercharger

4. Experimental Method and Apparatus

An experimental study was undertaken to evaluate the effectiveness of the various supercharger partial load control concepts at minimizing the driving power demand and thereby reducing fuel consumption in a gasoline engine.

4.1 Technical data and description of the screw supercharger

Table 1 Techical data for the OA1040 supercharger

OA 1040	Specificationn
Rotor Combination	3 (male) +6 (female)
Built-in Press Ratio	1.4
Discharging Pressure	1.6 (bar)
Suction Pressure	1.0 (bar)
Displacement	0.4 (Liter/Rev.)
Suction Temp.	20℃

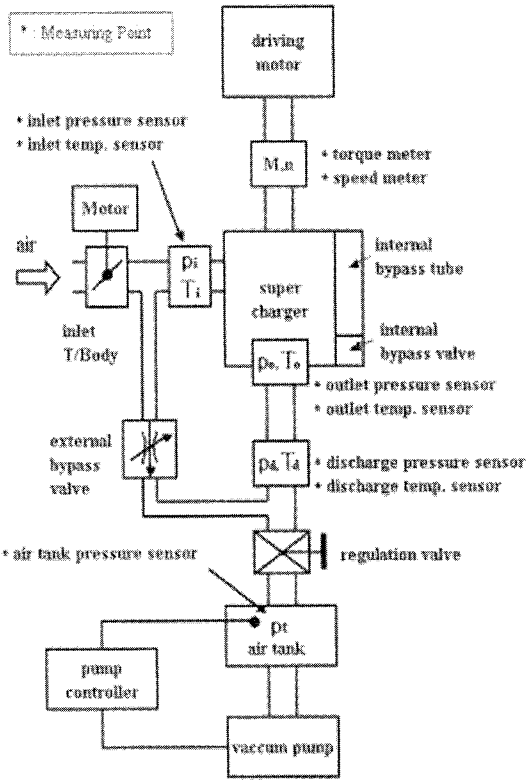


Fig. 9 Test bench setup for the screw supercharger with partial load control

The specified technical data for the screw supercharger are listed in Table 1. The test bench concept illustrated in Fig. 9 consists of measurement, control, and driving parts. The torque (T_{SC}) and speed (n_{SC}) were measured to calculate the driving power of the supercharger.

In addition, several thermodynamic factors, such as inlet and outlet pressures (p_i, p_o), inlet and outlet temperatures (T_i, T_o), and air mass ($\dot{m}$) were measured to investigate their influence on the driving power demand. To examine the partial load control, the inlet, outlet, and discharge pressures were varied by controlling the air inlet throttle body, external bypass valve, and internal bypass valve. An electric motor drove the supercharger, a regulation valve regulated the pressure in the air tank, and a vacuum pump created the same vacuum condition in the surge tank during partial load driving.

4.2 Technical data and description of the boosted engine

A description of a 1.5-liter gasoline engine manufactured by the Hyundai Motor Company is given in Table 2. The torque (T_M) and engine speed (n_M) were measured to calculate the power of the engine when boosted by a supercharger. Pressure sensors (p_1, p_2, p_3, p_4, p_5, and p_ξ) were installed on the engine to the pressure variation during the compression and discharge processes, as shown in Fig. 10. The supercharger was driven directly by the engine through a pulley connection with a ratio of 3 : 1.

Table 2 Engine sqecifications

Engine	Specification
Type	SOHC, Turbocharging In-line 4 cylinders
Bore×Stroke (mm)	75.5×83.5
Displacement (cc)	1495
Valves per cylinder	3
Compression ratio	7.5
Valve timing	8°BTDC/52°ABDC 52°BBDC/8°ATDC

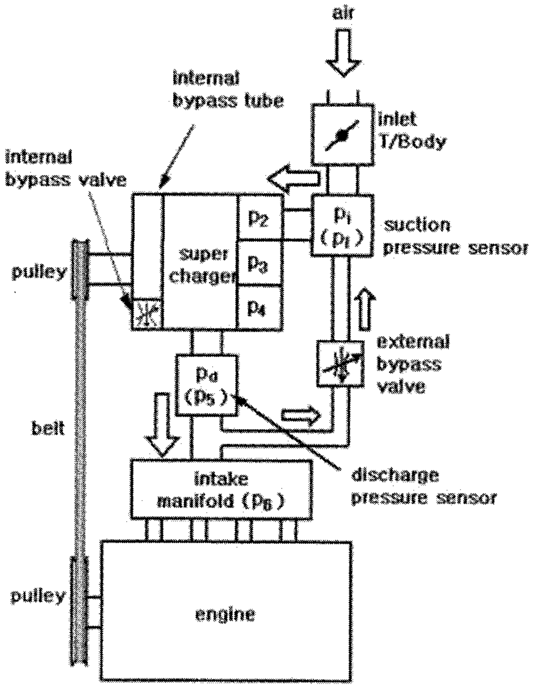


Fig. 10 Schematic diagram of the test engine for partial load control

5. Experimental Result

The p-V diagram in Fig. 11 shows how the pressure changed as work which was performed during the compression and discharge processes with the partial load concepts illustrated in Fig. 6(a)~(d). These concepts resulted in a reduction of the compression and discharge work, i.e., a reduction in the supercharger driving power

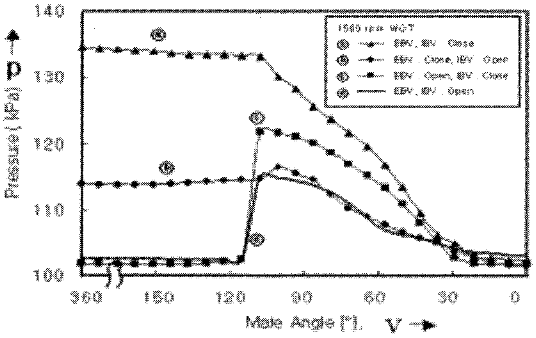


Fig. 11 Measured pressure variation at compression and discharge by different control mothods

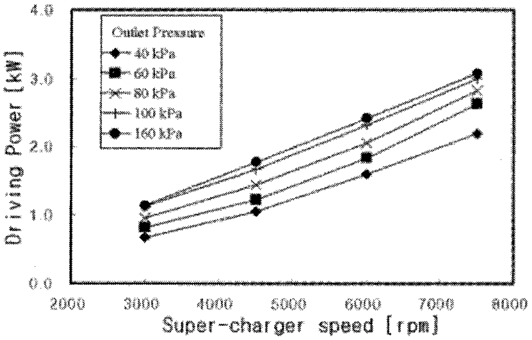


Fig. 12 Comparison of supercharger power consumption by throttling effect at different outlet pressure

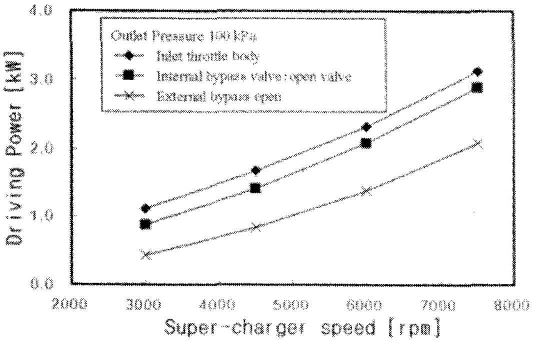


Fig. 13 Effect of different partial load control due to reducing the compression and discharge work at outlet pressure $p_o = 100$ kPa

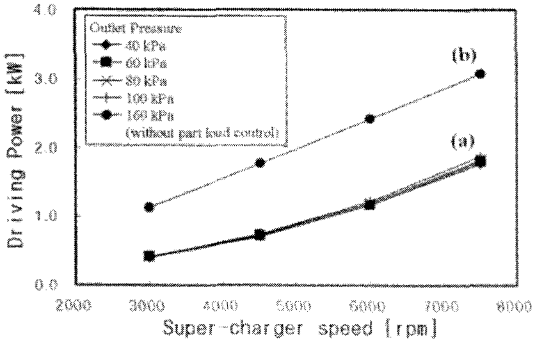


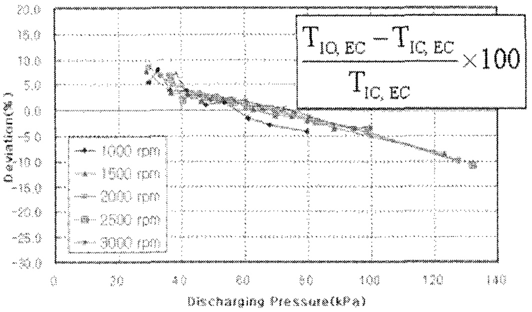
Fig. 14 Power consumption (a) with and (b) without adaption of partial load control

demand, as shown in Figs. 12, 13, and 14. The tests were performed at an engine speed of 1500 rpm and a wide open throttle.

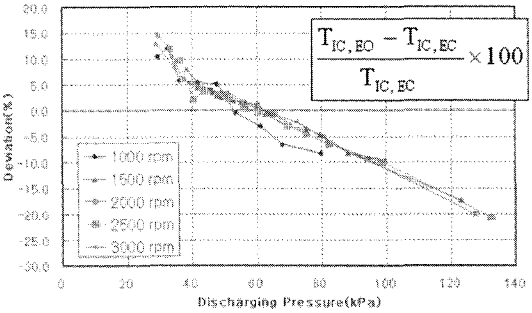
Partial load control concept (c) described in Section 3.2 resulted in a reduced driving work at partial load ($p_o = 40 \sim 100$ kPa) and at full load ($p_o = 160$ kPa), as shown in Fig. 12. Figure 13 illustrates the effectiveness of control methods (a) and (b) under the active partial load condition of method (c). The combination of all partial loading control methods decisively reduced the driving power demand, as plotted in Fig. 14, compared with the power demand obtained without any partial load control.

Figure 15 shows the effect of the partial load control on the engine torque output ($T_{IO,EC}$, $T_{IC,EO}$) with constant 18% throttle. Compared to the case without partial load control ($T_{IC,EC}$), the engine torque increased by 7% in Fig. 15(a), 12-13% in Fig. 15(b), and 15% in Fig. 15(c) in the area where the outlet pressure $p_o \leq 60$ kPa. This corresponds to the region where the engine idles or operates at low speeds, and may be defined as the threshold for supercharger operation during partial load.

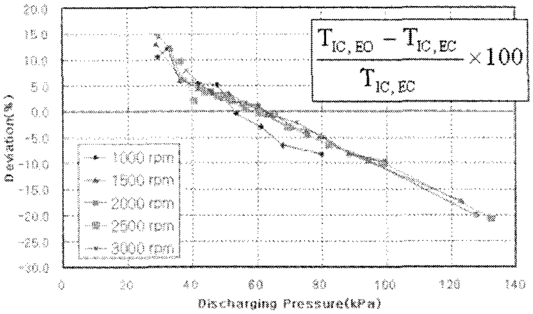
The adoption of partial load control did not provide any benefits to the engine in the mid and high partial load areas ($p_o > 60$ kPa), When the power decreased due to an insufficient supply of air into the cylinder.



(a) IBV=Open (IO), EBV=Close (EC)

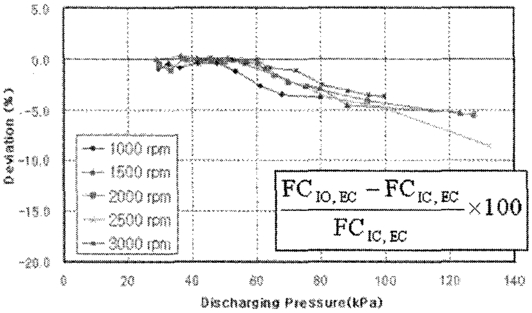


(b) IBV=Close (IC), EBV=Open (EO)

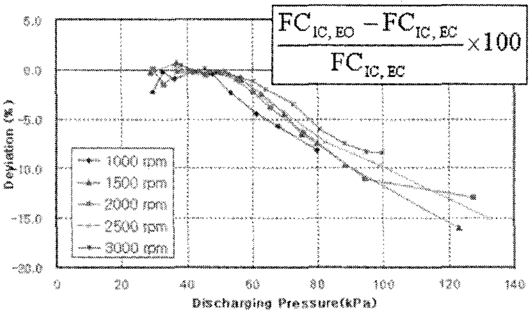


(b) IBV=Open (IO), EBV=Open (EO)

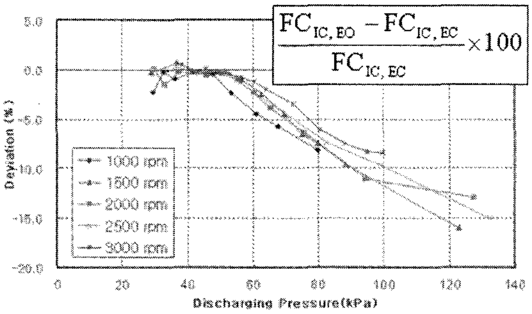
Fig. 15 Effect of different partial load control concepts on torque (T), ($T_{IC,EC}$) is the torque without partial load control



(a) IBV=Open (IO), EBV=Close (EC)



(b) IBV=Close (IC), EBV=Open (EO)



(b) IBV=Open (IO), EBV=Open (EO)

Fig. 16 Effect of different partial load control concepts on fuel consumption (FC), ($FC_{IC,EC}$) is the fuel consumption without partial load control

In the low partial load area ($p_o \leq 60$ kPa), the fuel consumption (FC), shown in Fig. 16, decreased by 1-1.5% in the Fig. 16(a), 2-4.5% in Fig. 16(b) and 2.5-5% in Fig. 16(c) in comparison to the case without partial load control ($FC_{IC,EC}$) due to the recduction of driving power demand of super charger as shown in Figs. 12, 13, 14.

As expected, the concepts of partial load control show a good result for the improvement of fuel consumption as well as for engine power output.

6. Conclusions

A screw supercharger designed to permit engine downsizing was investigated to identify methods that could be used to improve fuel consumption during partial load driving. The following conclusions were drawn.

(1) The driving power demand of the supercharger could be reduced by decreasing the compression work. This could be accomplished through controlling the inlet (p_i) and outlet (p_o) pressures by installing a supercharger behind the throttle body.

(2) Decreasing the compression work by venting the compressed air in the compression stroke toward the air inlet side through an internal bypass contributed to a decrease in the driving power demand.

(3) The driving power demand of the supercharger could be reduced by decreasing the discharge work by venting air during the discharge process toward the inlet side through an external bypass valve.

(4) The driving power demand of the supercharger could be reduced by adopting of all or part of the partial load control concepts.

(5) Applying conclusion 4) to actual downtown driving conditions will improve the fuel economy of an engine, as measured during the tests.

Acknowledgments

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