<Original>

Development of an Integrated Charging System for 4 Stroke Turbocharged Automotive Diesel Engine

Sae-Zong Oh*, Tae Shik Oh*, Sun Kook Chung and Dong In Lee*****

(Received October 19, 1983)

4행정 터어보과급 자동차용 디이젤 엔진의 통합과급방식의 개발

오 세 종*·오 태 식*·정 선 국**·이 동 인***

초 록

터어보과급 디이젤 엔진의 저속 및 급가속영역에서 발생하는 매연의 배출을 억제하기 위하여 흡입공기량을 증가시키는 방안으로서 흡기관의 동적효과를 이용하기 위한 통합과급 시스템을 개발하였다.

동조회전수에 있어서 음향임피던스 방법에 의하여 공명흡기관의 칫수를 결정하였고 흡입공기 냉각기를 부착하여 전 회전영역에서의 흡입공기 밀도비를 증가시켰다.

기존 엔진을 변형한 두 가지 시스템을 설계하여 성능측정을 하였으며, 이들에 대한 비교 및 실용성에 관해 자세히 언급하였다.

1. Introduction

The gas in the intake and exhaust pipe of an internal combustion engine experiences pressure vibration due to repeated induction and exhaust cycles. Many researchers have tried to increase volumetric efficiency, utilizing dynamic effect by intake pipe tuning, since Voissel[1]. In the earlier studies it was possible to obtain volumetric efficiency greater than 1 by tuning intake or exhaust pipes for naturally aspirated engine[2], but the system volume increased and the unbalance between tuned and untuned conditons was so serious that practical application did not occur in automotive use.

The appearance of turbocharger succeeded in changing the design concept of heavy duty diesel engine by maintaining increased performance without showing particular problems concerning the interaction between engine and turbocharger, at least under steady-state conditions. However, slow response of the turbocharger caused another problem of reduced torque and increased black smoke emission at accelerating and low engine speeds. Many techniques have been tried to improve boost pressure provided by the turbocharger at low engine speeds, without overboosting at high speeds. These include

* Member, Division of Mechanical Engineering, Korea Advanced Institute of Science & Technology

** Technical Center, Daewoo Heavy Industries Ltd.

***Member, Technical Center, Daewoo Heavy Industries Ltd.

pulse turbocharging, twin entry turbines, nozzleless turbines, matching for low speed operation, exhaust wastegases, tailoring fuel delivery, reduced engine speed, etc[3]. All of these measures are successful in improvement of low speed characteristics but are not sufficient in achieving optimum conditions over the whole speed range.

A novel technique of achieving improvement of performances over whole engine speed was proposed by Cser[4], with the idea of utilizing resonant intake system at low engine speeds and exhaust turbocharging at high speeds-both supported by charge air cooling. Charge air cooling as a mean of increasing the specific output of an engine has long been recognized[5~8]. The main features of the charge air cooling are higher air density and lower turbine inlet temperature that are known to have significant favorable effects on engine fuel economy and NOx emission.

The purpose of this paper is to present improvement of low speed characteristics of a turbocharged diesel engine using an integrated charging system which is composed of exhaust turbocharger, charge air cooler, and resonant-intake system. Instead of Cser's system using a combination of resonant volume and pipes (Helmholtz type resonator), resonant-intake system with resonance pipe only is applied, designed by acoustic impedance method. Two different systems from baseline engine were constructed for comparison of performances. Test results show mutual relations between the three ways of charging clearly.

2. Design Criteria of Resonant System

Dynamic effect of intake pipe has been classified into intertia(ram) and pulsation (standing wave) effect. However, Eberman[9] and Taylor[10] showed that the latter could be neglected comparing to the former. Therefore, in this paper, design of resonant system is based on the utilization of the inertia effect only. An extreme value of volumetric efficiency can be achieved by selecting proper length of intake pipe by which n times of pressure vibration are produced during an intake period. When n becomes greater than 2, amplitude of the pressure vibraiton decreases and these are not considered here.

The optimum frequency ν_{opt} can be expressed, for given engine speed N(rev/min), as

$$\nu_{opt}=\frac{6N}{\phi_{eff}} \tag{1}$$

where ϕ_{eff} is the effective intake valve opening period(deg. crankangle) and equal to the valve opening period minus valve overlap. If the natural frequency ν_{nat} of the whole intake system satisfies

$$\nu_{nat}=\nu_{opt} \tag{2}$$

one can expect one time pressure vibration during an intake period for engine speed N.

3. Acoustic Impedance Idealization

In this paper, an intake system is defined as a comined system of pipe and volume elements located between outlet of charge air cooler to engine cylinder. Acoustic impedance Z, which is similar to electric and mechanical impedance, is defined as the ratio of pressure fluctuation $\tilde{p}$ and volume variation δ as follows[11].

$$Z\equiv\frac{\tilde{p}}{\delta} \tag{3}$$

Assuming perfect gas and isentropic change in intake system, relationships between acoustic impedences can be easily obtained for all constituent elements. Several representative formulations about cylidner-pipe combination(Fig. 1) are given in Table 1, using index i as the number of cylidner. Apply-

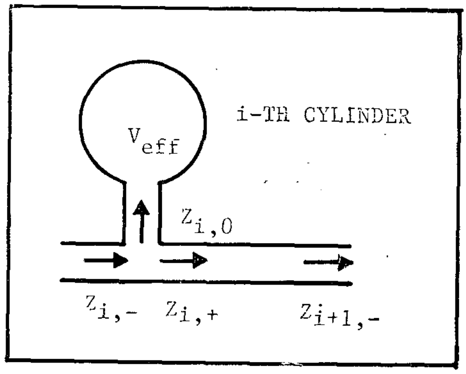


Fig. 1 Acoustic impedance idealization of cylinder-pipe combination

Table 1 Relationships between acoustic impedances for cylinder-pipe combination

$$Z_{i,0}=-4\pi^2\frac{\rho}{C_i}(\nu^2_{\mathrm{nat}}-\nu^2_i)$$

$$Z_{i,-}=\frac{Z_{i,0}Z_{i,+}}{Z_{i,0}+Z_{i,+}}$$

$$Z_{i+1,-}=\frac{\beta}{f_{i+1,-}}\,\frac{Z_{i,+}\{1+\alpha_a\tan kl/kl\}+\beta/f_{i,+}\tan kl}{-Z_{i,+}\{[1+\alpha_a\alpha_b/(kl)^2]\tan kl-\alpha_a\alpha_b/kl\}+\beta/f_{i,+}\{1-\alpha_b\tan kl/kl\}}$$

$Z=0$ for open end

$Z=\infty$ for closed end

Where $\nu_i=\frac{a}{2\pi}\sqrt{\frac{C_i}{V_i}}$: Natural frequency of i-th cylinder

$V_i=\frac{1}{\phi_{\mathrm{eff}}}\int Vd\phi$: Effective cylinder volume of i-th cylinder

$C_i=\frac{f_{i,0}}{l_{i,0}}$: Conductivity of i-th cylinder

$\alpha_a=\sqrt{f_{i+1,-}/f_{i,+}}-1(=0$ for straight pipe)

$\alpha_b=\sqrt{f_{i+1,-}/f_{i,+}}(=1$ for straight pipe)

$a=\sqrt{\gamma gRT}$: Acoustic velocity of air

$\rho=P/RT$: Density of air

$\beta=a^2\rho k$

$k=\omega/a$

$\omega=2\pi\nu_{\mathrm{nat}}$: Angular velocity of vibration

f : Cross-sectional area of pipe

l : Length of pipe

ing these relationships to the whole intake system, one can obtain n simultaneous equations with $(n+2)$ unknowns. This set of equations can be solved by graphical or numerical methods provided that proper boundary condition is prescribed and optimum condition of Eqs. (1) and (2) are given.

4. Application to Real System

The baseline engine is a turbocharged 6 cylinder inline type of which schematic reresentation of intake system is shown in Fig. 2a. Specifications of the baseline engine are given in Table 2. Two modified intake systms are constructed and their arrangements are shown in Fig. 2b and Fig. 2c. Major differences with original one are installation of charge aircooler and adoption of divided manifold to avoid intake valve overlap by matching valve opening period and firing interval of a group of three cylinders. The difference between system B and C is that system B uses a much shorter intake pipe to reduce flow friction while system C uses two longer pipes whose length is optimized to achieve tuning at low engine speed(1200rev/min). System C requires some care in minimizing the degree of

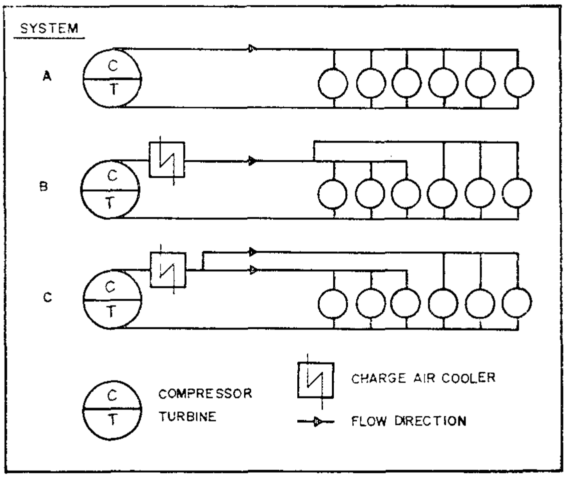
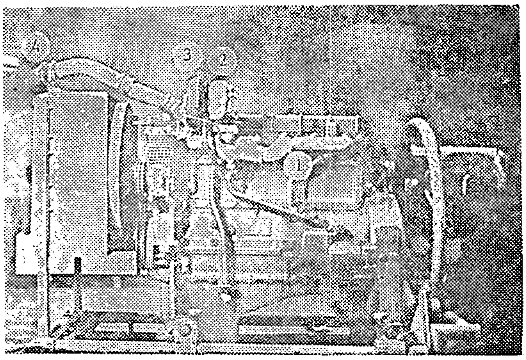
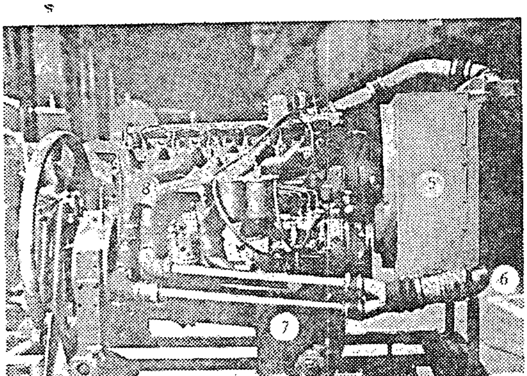


Fig. 2 Schematic representation of intake system from above Fig. 2a : base line system (system A), Fig. 2b : system B, Fig. 2c : system C

Table 2 Specifications of baseline engine

Item	Specication
Type	Turbocharged 4 stroke vertical type
No. of cylinders	6 cylinders in line
Cooling sysem	Water cooling type by forced circulation
Bore X stroke	121×150mm
Displacement volume	10, 350cc
Compression ratio	17 : 1
Firing order	1-5-3-6-2-4
Injection time	23 deg. BTDC
Combustion system	Direct injection M type
Maximum power	256 PS/2200 RPM(DIN)
Maximum torque	91.5kg·m/1600 RPM(DIN)
Valve timings	
In. open	21 deg. BTDC
In. close	35 deg. ABDC
Ex. open	60 deg. BBDC
Ex. close	30 deg. ATDC





① Exhaust manifold, ② Exhaust turbine, ③ Compressor, ④ Intercooler inlet, ⑤ Intercooler assembly, ⑥ Intercooler outlet, ⑦ Resonance pipes, ⑧ Intake manifold

Fig. 3 Layout of integrated charging system (system C)

pipe bendings due to the increased pipe length. The state of installation of system C is shown in Fig. 3.

5. Engine Test Result

Engine tests were conducted over a speed range of 1000 to 2200 rev/min with the fuel quantity set equal at 2150 rev/min for three systems at the Engine Research Laboratory in Daewoo Heavy Industries Ltd.

Substantial volumetric efficiency increase is recorded for whole speed range with system B and even

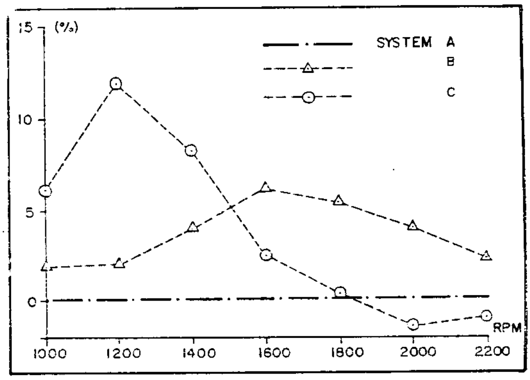


Fig. 4 Gain in volumetric efficiency with integrated charging systems over system A(baseline engine)

higher for tuned speed with system C. Fig. 4 shows gain in volumetric efficiency with systems B and C over the baseline engine. Gain with system C reaches the maximum value of 12% at 1200 rev/min and it is even much higher than that of system B for the range of 1000 to 1400 rev/min, which will let fuelling be increased at low speeds.

Full load-constant speed performance characteristics over the whole speed range for the three systems are given in Fig. 5. Torque and output curves show almost the same levels with the exception of BSFC, since the fuelling is governed by LDA so that increased intake air pressure may cause slight variation. Most striking result is the smoke reduction, with the maximum value of 49% with system C at 1000 rev/min and 61.5% with system B at 1600 rev/min. As in the case of volumetric efficiency, system B shows a good performance over the whole speed range while system C shows an outstanding improvement near the tuned engine speed. Mechanism of formation and combustion of soot in diesel engine is quite complex for the heterogeneous nature, and even in a simple case, a complete quantative picture has not yet been built up. But some qualitative conclusion induced from experimental researches[12,13] can be stated as the contributory factors for combustion system that baseline engine adopts are

1) The controlling influence on the bulk fuel temperature excercised by the chamber wall,
2) That reduced residence time of fuel-rich vapour in the high-temperature zones, due to increased mixing rates, will result from high levels of air swirl[14].

Thus increased air quantity will cause increased swirl strength, which in turn affects the mixing process to some extent. Although this value of air swirl strength is not an optimum, it must be more closely convergent than that of baseline engine.

Fig. 6 shown the result of exhaust gas analysis according to Japan 6 mode. As indicated by Sekar[15], emission of NOx decreases while that of THC increases as inlet temperature is decreased by charge air cooling. System C shows more sharp variations than B and this difference should be checked more

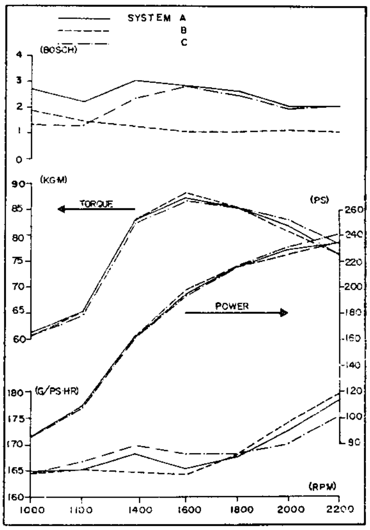


Fig. 5 Full load-constant speed performance curves (according to DIN 70020-engines are equipped with a cooling fan)

quantitatively through repeated measurement.

CO shows similar variation to THC, and reduction of free acceleration smoke emission shows another favorable aspect of charge air cooler and resonance pipe system. 8.9% reduction with system C and 5.1% with B imply possibility of increased fuelling at accelerating speeds and improved torque characteristics at this speed range.

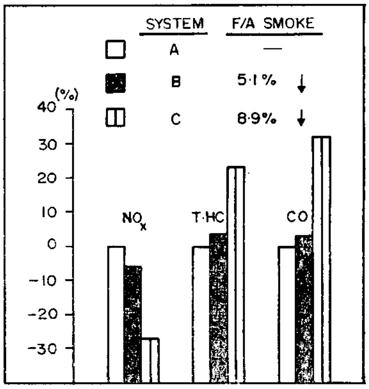


Fig. 6 Exhaust gas analyses by Japanese 6-mode and free acceleration smoke test results

Fig. 7 shows the measured temperature and pressure ratios in compressor-intercooler system and intercooler effectiveness ε calculated from them.

Temperature ratios in system B are higher than those of system C, while the pressure ratios are not, and this reciprocal trend implies that the density ratios in system C would be higher than those of system B as shown in Fig. 4. Some mention should be made about the thermodynamic usefulness of charge air cooler. Installation of charge air cooler causes pressure drop and this value should be less than the temperature drop for density to increase. That is, P_1/P_2 is desired to be less than T_1/T_2 and in either system this condition is shown to be satisfied. Effectiveness ε in system B is shown to be higher than that of system C except 1000 and 2200 rev/min and both have values of ε greater than 75% over the whole engine speed. Future task for design of compact system including radiator and charge air cooler is desirable.

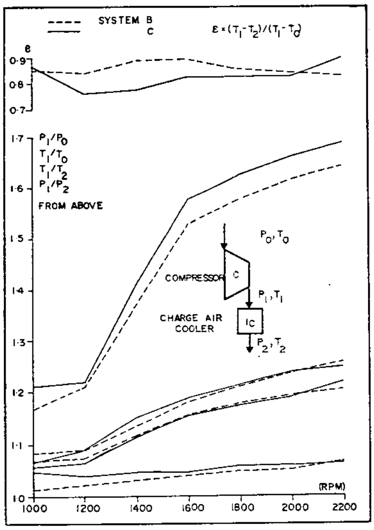


Fig. 7 Performance and effectiveness curves of charge air cooler

6. Conclusion

An intergrated charging system composed of exhaust turbocharger, charge air cooler, and resonant-intake system using resonance pipe only is applied to real engine successfully and the results can be summarized as follows.

1) Considerable increase in volumetric efficiency and decrease in black smoke and NOx emission are achieved over the whole engine speed with system B.
2) More improvements are possible with system C at or near the tuned engine speed.
3) Simple acoustic impedance method can be used to the baseline design of resonant-intake system effectively.
4) Reduction of black smoke emission at accelerating speeds implies possibility of increased fuelling and resulting improved torque characteristics.

If some drawbacks such as space limitation or rematching of other elements can be settled properly, integrated charging system will be used most effectively on heavy duty diesel engines due to its cheapness, simplicity, and reliability with no moving part.

Acknowledgement

The authors would like to thank the Daewoo Heavy Industries Ltd. for the financial support and permission to publish this work. Thanks are also due to all the people connected with the project at the Korea Advanced Institute of Science and Technology. A final note of thanks must be extended to Dr. Y.M. Yoo for his useful advice.

References

(1) P. Voissel; Resonanzerscheinung in der Saugleitung von Kompressoren und Gasmotoren, VDI Forschungsheft. Vol. 106, pp. 27, 1912

(2) 齋藤宗三, 松島富久壽；吸排氣管內壓力脈動により出力增加を得た實用機關, 日本機械學會 33期 通常總會 講演會 前刷 5室

(3) N.Watson; Resonant Intake and Variable Geometry Turbocharging Systems for a V8 Diesel Engine, Instituion of Mechanical Engineers, Paper No. C40/82, pp. 101—113, 1982

(4) G.Cser; "Ein neuartiges Verfahren zur Verbesserung der Abgasturboaufladung," Motortechnische Zeitschrift, Vol. 32, No. 10, pp. 368—373, 1971

(5) J.E. Mitchell; "An Evaluation of Aftercooling

in Turbocharged Diesel Engine Performance", SAE Trans. Vol. 67, pp. 401, 1959

(6) C.J. King; "Turbocharger Aftercooling, Why and How", SAE paper No. 700536

(7) "New High Performance Paxman Has Strong Naval Potential", Shiplog Shipping Rec. Vol. 117, Jan. 15, pp. 26, 1971

(8) H.G. Holler; "The Influence of Induction and Exhaust System Design on Power Producing Characteristics of Diesel Engines", SAE paper No. 700535

(9) L.Eberman; Motorship, Vol. 16, No. 182, pp. 28, 1935

(10) C.F. Taylor, J.C. Livengood & D.H. Tsai; "Dynamics in the Inlet System of a Four-Stroke Single-Cylinder Engine", ASME Trans., Vol. 77, No. 7, pp. 1133, 1955

(11) 嶋本譲；"吸・排氣管効果の利用(上)", 內燃機關, Vol. 10, No. 108, pp. 93—101, 1971

(12) J.S. Meurer; "Multifuel Engine Practice", SAE paper No. 296A, International Congress Detroit, 1961

(13) J.S. Meurer; "シリンタ內で混合氣を作る機關における混合氣の生成と燃燒の進步", 內燃機關, Vol. 6, No. 58, pp. 15—23, 1967

(14) D. Broome and I.M. Khan; "The Mechanisms of Soot Release from Combustion of Hydrocarbon Fuels with Particular Reference to the Diesel Engine", Institution of Mechanical Engineers, paper No. C140/71, pp. 185—197, 1971

(15) R.R. Sekar; "Trends in Diesel Engine Charge Air Cooling", SAE paper No. 820503