EGR System in a Turbocharged and Intercooled Heavy-Duty Diesel Engine – Expansion of EGR Area with Venturi EGR System –

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Abstract

The EGR system is a very effective technique for reducing NOx emission from a diesel engine, particularly at the high load of engine operation condition where the engine emits more NOx than at other conditions.

In a turbocharged engine, however it is difficult to introduce EGR at the high load because of the high boost pressure in the air intake.

In this study, effective technologies to introduce EGR in a turbocharged engine were evaluated, and the Venturi type EGR system was found to be promising, without penalty of fuel economy caused by the increase of pumping loss.

Key words: *Exhaust Gas Recirculation (EGR), Diesel Engine/Venturi EGR System*

1. Introduction

Diesel emission regulations continue to be tightened in many countries, necessitating diesel engines with the least possible emissions. Exhaust gas recirculation (EGR) is one of the most effective methods for reducing the emissions of nitrogen oxides (NOx) of diesel engines. EGR system has already been used to mass-produced diesel engines, in which EGR is used at the low and medium load of engine operating condition, resulting in effective NOx reduction. In order to meet future emission standards, EGR must be done over wider range of engine operation, and heavier EGR rate will be needed. It is especially important for EGR to be done in a high engine load range since the amount of NOx is larger than the other engine operation conditions.

EGR systems adapted to the diesel engines of trucks usually recirculate exhaust gas utilizing the pressure difference between upstream part of the turbocharger turbine and downstream part of the compressor. This method of EGR needs that the pressure upstream of the turbine (hereinafter referred to as "turbine pressure") is higher than the pressure downstream of the compressor (hereinafter referred to as "boost pressure"), i.e.,

$$\Delta P = (\text{turbine pressure} - \text{boost pressure}) > 0$$

In a turbocharged diesel engine for truck, however, $\Delta P > 0$ is limited to the low load region, while in the high load region ΔP is smaller than zero and EGR is impossible. Several methods have been proposed to achive $\Delta P > 0$ condition even under a high-load condition. For example, some of these systems increase the turbine pressure to obtain a $\Delta P > 0$ condition by means of an exhaust choke valve or a variable geometry (VG) turbocharger, whereas other systems decrease the boost pressure to obtain a $\Delta P > 0$ condition by means of an intake throttle valve or a venturi[(1) – (4)].

This paper discusses the effects of EGR for NOx reduction and evaluates some methods proposed for creating a $\Delta P > 0$ condition. The paper then discusses the characteristics of the venturi EGR system that is the most promising among the proposed and shows that the system is rather effective for expanding the EGR range up to high engine load conditions.

Table 1 Engine specifications

Type	Turbocharged and intercooled direct injection diesel engine
Number of cylinders	6
Bore x stroke	135 x 140 mm
Displacement	12.023 liters
Maximum output	287 kW/2200 rpm

2. Test equipment

2.1 Engine

The engine used for the test was a turbocharged, intercooled direct injection 12-liter diesel engine for heavy-duty trucks. The major specifications of the engine are shown in **Table 1**.

2.2 EGR system

The EGR system used with the test engine was the type that exhaust gas was recirculated from the upstream part of the turbocharger turbine to the downstream part of the compressor. **Fig. 1** shows the configuration of the EGR system. The EGR passage is taken

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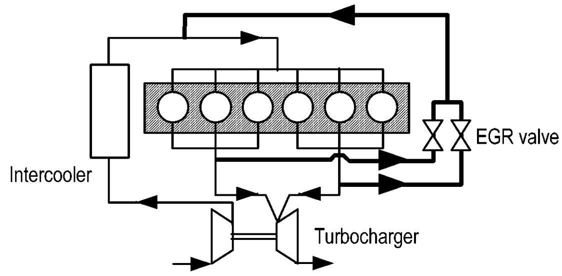

Fig. 1 Original EGR system

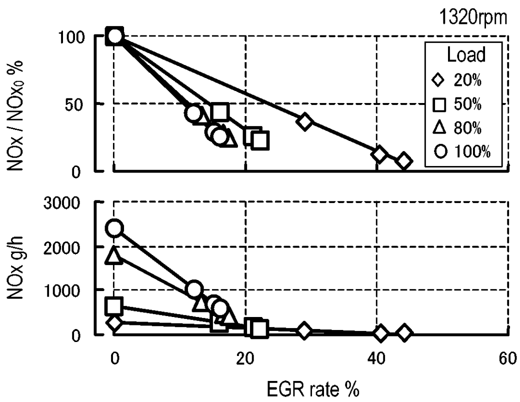

Fig. 2 Relationship between EGR rate and NOx

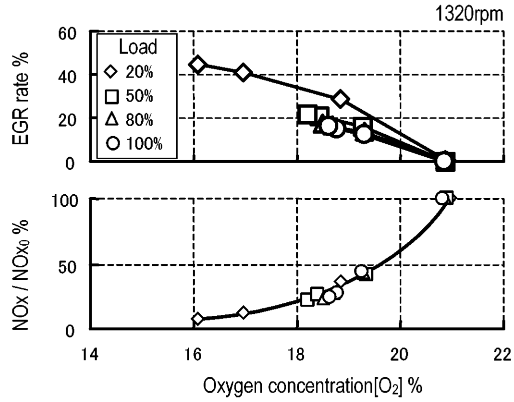

Fig. 3 Relationship between oxygen concentration and NOx reduction

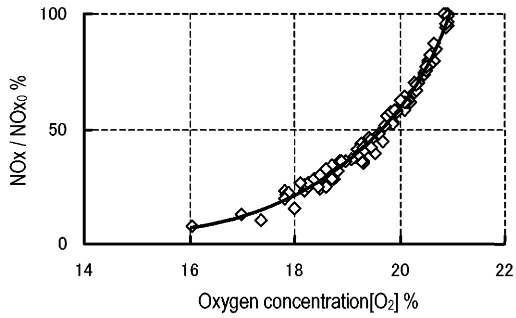

Fig. 4 Relationship between oxygen concentration and NOx reduction

out each manifold which separate front bank cylinder and rear cylinders, and connects to the intake pipe after two pipes connet to a single pipe. On each of the two pipes, an EGR valve is installed as a means of regulating the EGR rate. Each of the two valves has a capacity of 3500 liters/min (at pressure difference of 6.7 kPa).

Experiments were conducted by applying some of the proposed systems to this original EGR system to cause a positive ΔP condition.

3. Experiment results and evaluation

3.1 NOx reduction effect of EGR

Fig. 2 shows the typical NOx reduction effect of EGR at the mid-speed range of the test engine. Under all load conditions, the amount of NOx decreases as the EGR rate increases. The graph also shows that the NOx reduction curves with the 0 % EGR point as the origin slope downward at different angles according to the load; the higher the load, the steeper the angle. In other words, the NOx reduction effect at the same EGR rate increases as the engine load becomes higher.

It is generally known that there are two reasons to reduce NOx by EGR. The first of them is the reduction of combustion temperature. The addition of exhaust gases to the intake air increases the amount of combustion-accompanying gases (mainly CO_2), which in turn increases the heat capacity and lowers the combustion temperature. The second effect is the reduction of oxygen concentration in the intake air, which restrains the generation of NOx. **Fig. 3** shows the NOx emission test results as a function of the concentration of oxygen in the intake air/EGR gas mixture. This graph shows that the NOx reduction rate depends mostly on oxygen concentration, and not on the engine load or EGR rate. **Fig. 4** shows the results of NOx emission tests conducted while varying both the engine operating conditions and EGR rate, in which the test results shown in **Fig. 3** are merged. As in **Fig. 3**, almost all the data are on or in a single curve, indicating that there is a strong correlation between the oxygen concentration and NOx reduction rate. The reason for this is thought to be as follows: In **Fig. 2**, the NOx reduction rate under a certain load is different from that under another load even when the EGR rate remains the same because the difference in load causes a difference in the amount of combustion-accompanying gases and oxygen concentration in EGR gas, which in turn changes the oxygen concentration in the intake gas (mixture of intake air and EGR gas).

Fig. 5 shows the oxygen concentration calculated on the assumption that the EGR rate is increased while keeping the excess air ratio of the intake air and EGR gas mixture constant. The EGR rates and oxygen concentrations are plotted on the graph for each of the excess air ratios $\lambda = 1.5$ to 5 that correspond to the high-to-low engine load range. This graph indicates that, given the same oxygen concentration, the required EGR rate at $\lambda = 1.5$ is less than half of that at $\lambda = 5$. Considering this finding together with **Fig. 4**, we con-

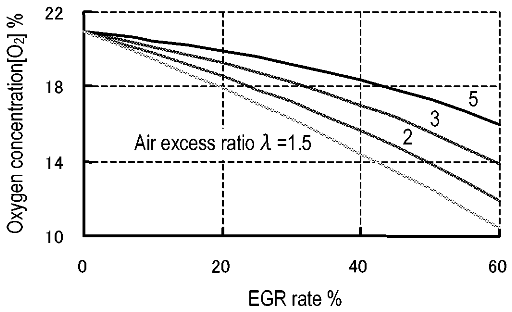

Fig. 5 Relationship between EGR rate and oxygen concentration

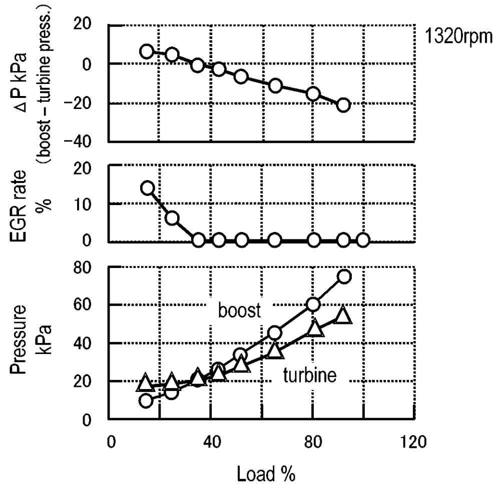

Fig. 6 EGR rate and pressure difference between boost and turbine pressure

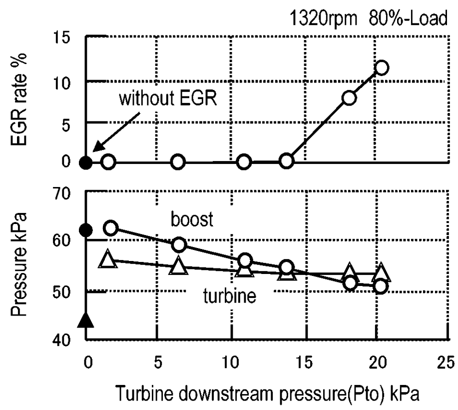

Fig. 7 Turbine and boost pressures and EGR rate in increasing Pto

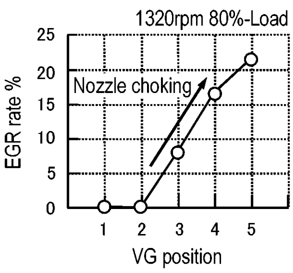

Fig. 8 Relationship between VG position and EGR rate

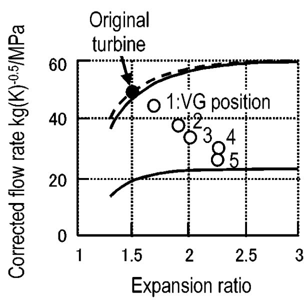

Fig. 9 Characteristics of VG turbocharger

clude that EGR operating under a high-load condition will bring about a NOx reduction rate equivalent to that attained under a low-load operation even with an EGR rate lower than that under the low-load condition. Since high-load operation involves a large amount of emissions, even a low rate of NOx reduction will have a very large effect on the overall NOx reduction.

It is therefore essential to extend the EGR-feasible load range to encompass high-load operation in order to reduce NOx emissions from turbocharged diesel engines.

3.2 Methods for expanding EGR-feasible load range

This section discusses two feasible methods for expanding EGR range to the high-load, one by using an exhaust throttle valve and the other by using a VG turbocharger. **Fig. 6** shows the EGR rates by the original EGR system at mid-speed range. The turbine pressure exceeds the boost pressure under low loads of engine operation condition but it goes below the boost pressure as the load increases. This means that the EGR-feasible range is limited only to low-load operations.

Fig. 7 shows the results of the pressure and EGR rate equipped with an exhaust choke in the mid-speed range and under the high-load condition. In this test, the opening of the exhaust choke valve located downstream of the turbine was changed to increase the pressure there. When the pressure downstream of the turbine is made low, EGR does not take place because the boost pressure exceeds the turbine pressure. Instead, narrowing the choke valve opening increases the pressure downstream of the turbine and EGR can take place as soon as the pressure levels reverse. The resulting situation, however, gives rise to a drop of the excess air ratio due to the decrease in boost pressure, which may increase smoke emissions.

The results of tests conducted using the VG turbocharger are described next. The VG turbocharger used in the test was of the same type as that actually used in large-size diesel engines. **Figs. 8** and **9** show the EGR rates achieved in the mid-speed range under a high-load condition for different turbocharger geometry points created by progressively narrowing the noz-

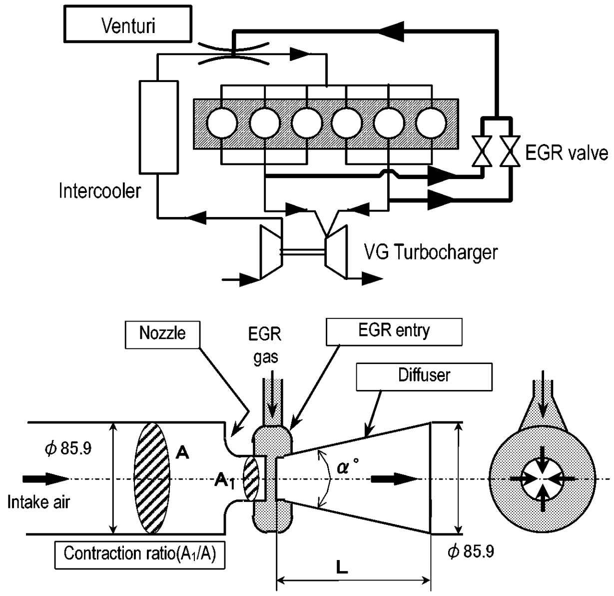

Fig. 10 Schematic of venturi EGR system

zle area of the VG turbocharger. The EGR rate increases by narrowing the nozzle area. A maximum EGR rate over 20 % can be attained.

Both the above-mentioned methods certainly make it possible for EGR to take place in the high-load range, but they inevitably involve a higher pumping loss during intake and exhaust strokes because they work on the turbine pressure to make it higher than the boost pressure.

3.3 Venturi EGR system

The venturi system assures a positive ΔP by decreasing the pressure on the intake side only regionally. With this system, the boost pressure is once reduced ($\Delta P > 0$) in the EGR entry section, but a pressure is re-established to the original boost pressure downstream of that section. Therefore it is theoretically possible for this system to expand the EGR-feasible range without increasing the pumping loss. **Fig. 10** shows the configuration of the venturi EGR system. The venturi is located on the intake passage downstream of the intercooler and consists of three sections: a nozzle, an EGR entry section, and a diffuser. Through the gap formed between the nozzle and diffuser, the EGR gas is circumferentially drawn into the intake air stream and mixes with the air. The contraction ratio of the venturi is defined as the ratio of the sectional area of the intake pipe to the sectional area of the nozzle (A_1/A).

Fig. 11 shows the pressure distribution in the venturi area at the mid-speed range under a high-load condition. The pressure at point A on the upstream side of the venturi is higher than the turbine pressure. At the EGR entry section (point B), the pressure drops below the turbine pressure, and hence $\Delta P > 0$. The pressure is then restored downstream of the EGR entry section (point C) to a level almost equivalent to the pressure at point A. From the obtained pressure distribution data, it was confirmed that the venturi EGR system enabled

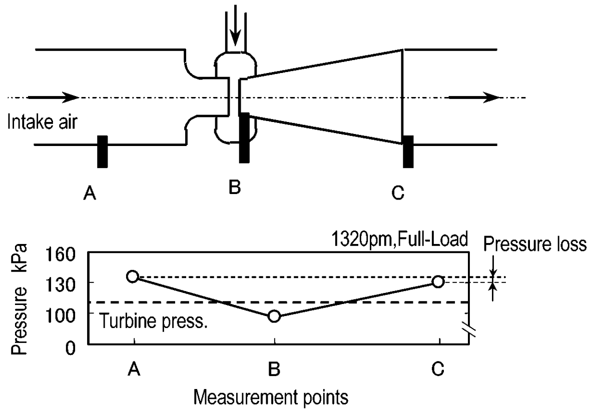

Fig. 11 Pressure distribution of venturi EGR system

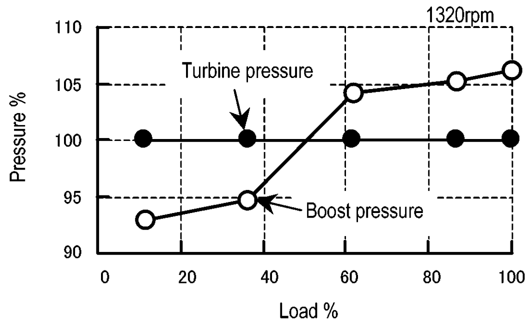

Fig. 12 Relationship between boost pressure and turbine pressure

EGR to take place under a high-load condition without being accompanied by pumping losses resulting from a decrease in the boost pressure.

Next, we conducted an investigation for various shapes of the venturi's component sections in order to find the optimum venturi specifications to increase the EGR rate in the high-load operation range with minimum pressure loss. The following section describes the effects of optimizing the shape of each component section based on the results of the investigation, and compares the venturi method with the EGR rate augmentation method that depends on exhaust choking, and identifies its advantages over other methods.

3.4 Effects of venturi geometry

(1) Contraction ratio

Before investigating the effects of the contraction ratio on the EGR rate through tests, it was necessary to roughly select an appropriate range of the contraction ratio of the venturi. The amount of drop of the boost pressure necessary for EGR to take place was therefore derived from the relationship between the boost and turbine pressures in the original EGR system, and then the contraction ratio necessary for attaining that pressure drop was determined assuming the flow through the venturi to be an adiabatic compression flow.

Fig. 12 shows the relationship between boost and turbine pressures in the original EGR system. The engine was operated in the mid-speed range, in which

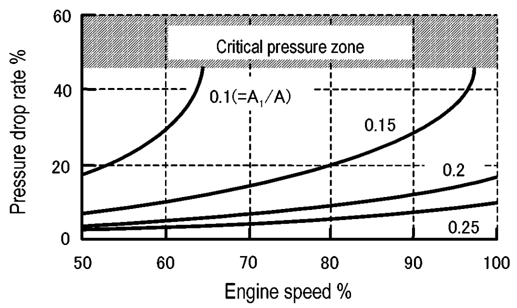

Fig. 13 Estimation of necessary contraction ratio

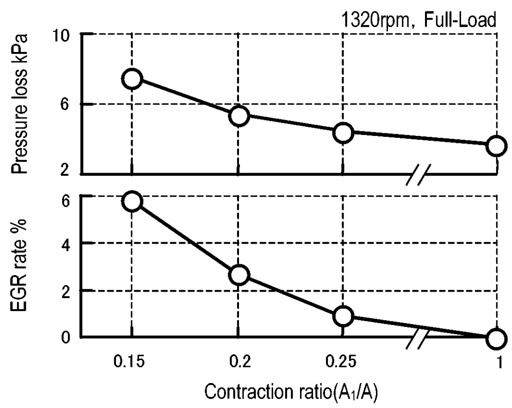

Fig. 14 Effect of contraction ratio

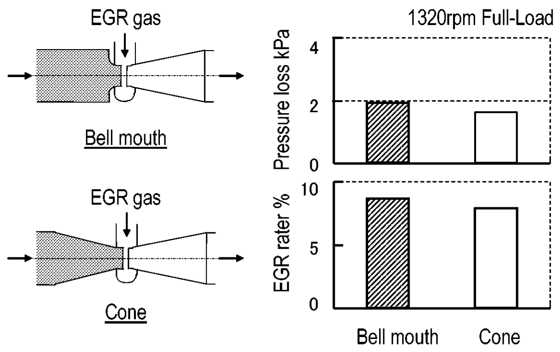

Fig. 15 Effect of nozzle shape

EGR cannot easily occur with conventional systems at the high load. The graph indicates that the boost pressure exceeds the turbine pressure in the mid- and high-load operation ranges, making it difficult to maintain the $\Delta P > 0$ condition. Therefore, EGR can take place when the pressure at the EGR entry section is lowered by using a venturi to less than the turbine pressure. For example, it is necessary to reduce the boost pressure about 5 – 6 % at the full-load point to establish $\Delta P > 0$ condition. **Fig. 13** shows the boost pressure drop rates calculated for different venturi contraction ratios. The indicated pressure drop rates are obtained assuming that the intake gas temperature is 50 ˚C and the contraction ratio is varied from 0.1 to 0.3. The graph indicates that the contraction ratio must be 0.2 or lower to obtain a 5 – 6 % boost pressure drop in the mid-speed range.

In the case of operation at the maximum rated speed with the contraction ratio set at around 0.15, the calculated pressure at the EGR entry section enters the critical pressure zone (a pressure drop rate of 46.5 % was used in the calculation). Based on this estimation, the contraction ratio of 0.2, with which the pressure would not enter the critical zone, was selected as the reference ratio with the next lower ratio 0.15 and next higher ratio 0.25 also selected as contraction ratios for the test of sample venturis.

Fig. 14 shows the test results for the EGR rate and pressure loss with each contraction ratio. The test was conducted at the mid-speed range, full-load condition. In the graph, the contraction ratio of 1 corresponds to the non-venturi condition. The results confirmed that the EGR rate was influenced by the contraction ratios as expected, and that decreasing the contraction ratio to 0.15 could raise the EGR rate to as high as 6 %. With regard to the pressure loss, it increased slightly as the contraction ratio was decreased but the increase in the pressure loss had almost no effect on the air flow rate and fuel consumption under the operating conditions used in the test. It was also confirmed that intake air flow did not reach to sonic speed even with the contraction ratio at 0.15 and the engine speed at the maximum rated speed. Despite the estimation from the calculation, sonic speed was not actually reached presumably because of the pressure loss.

The test results revealed that the optimum contraction ratio was 0.15.

(2) Nozzle

As shapes of the venturi nozzle, an ordinary cone shape and a bell-mouth shape were selected for the study in order to meet the objectives of compactness and minimum pressure loss.

Fig. 15 shows the study results. There was almost no difference in the EGR rate attributable to differences in nozzle shape. Regarding the pressure loss, both the shapes showed almost the same characteristics although the pressure loss of the cone-shaped nozzle was slightly lower than that of the bell-mouth.

The bell-mouth-shaped nozzle with shorter overall length was finally selected rather than the cone-shaped nozzle because it is easier to install on the engine.

(3) Diffuser

The shape of the diffuser was also studied to achieve compactness and minimum pressure loss. It is well known that the pressure loss largely depends on the expansion angle of the diffuser and an expansion angle of 5.5˚ is optimum because it is associated with the minimum pressure loss[(5)]. On the other hand, the diffuser can be made more compact by increasing the expansion angle, as its length is inversely proportional to the expansion angle. In the study, diffusers of angles of 8˚, 11˚, and 17˚ were tested because of their compact size in addition to a diffuser of the ideal 5.5˚ angle. Taking the length of the 5.5˚ diffuser as 1, those of the larger angle diffusers are 0.7, 0.5, and 0.3, respectively. If the expected increase in the pressure loss can be properly controlled, the shorter diffusers will signifi-

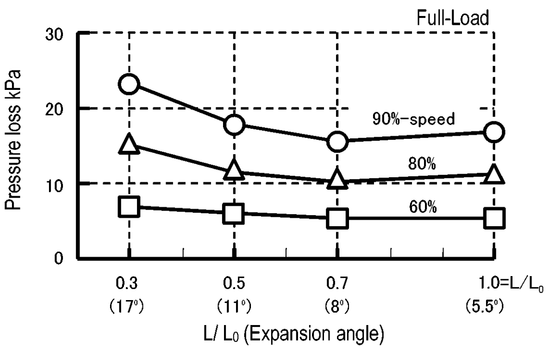

Fig. 16 Effect of diffuser length on pressure loss

Table 2 Optimum venturi geometry

Contraction ratio (A_1/A)	0.15
Nozzle	Bell mouth
EGR entry	Circumferential
Diffuser	Expansion angle α = 11° Overall length L = 273 mm

cantly reduce the size of the venturi.

Fig. 16 shows the pressure loss for each of the expansion angles. Compared with the ideal 5.5° angle, the increase in pressure loss due to an increase in angle is almost negligible up to 11°, but increasing the angle further for shorter length evidently increases pressure loss in the high-speed range of operation. In conclusion, it was determined that the 11° expansion angle was the most appropriate for the diffuser to be adequately compact while being able to control the pressure loss to an appropriately low level.

It was also confirmed that changing the expansion angle from 5.5° to 11° did not cause any substantial change in EGR rate.

(4) Determination of optimum venturi geometry

Based on the findings from the studies conducted with regard to the EGR rate, pressure loss, and compactness, it was found that the geometry specifications shown in **Table 2** corresponded to the optimum characteristics of the venturi.

3.5 EGR rate improvement effect of venturi system

Fig. 17 shows the advantages of the optimized venturi system over the original EGR system in terms of the EGR rate.

With the venturi system, the ΔP > 0 condition is established even in the 50 % or higher load range where ΔP is negative and consequently EGR is impossible with the conventional systems. Furthermore, the EGR-feasible range of the venturi system extends to the full-load engine condition, at which the system attains a 6 % EGR rate. The venturi system also increases the EGR rate by approximately 10 % in the mid-load and low-load ranges regardless of engine speed.

The venturi EGR system involves pressure losses at the nozzle and diffuser sections. The effects of these pressure losses on the fuel consumption of the engine were investigated. **Fig. 18** shows the fuel consumption rates and pressure losses in an engine with the optimized venturi system under the full-load condition. Though the pressure loss increases with increase in airflow as the engine speed becomes higher, the increase in fuel consumption is less than 1 %, indicating that the venturi has almost negligible effect on the fuel consumption.

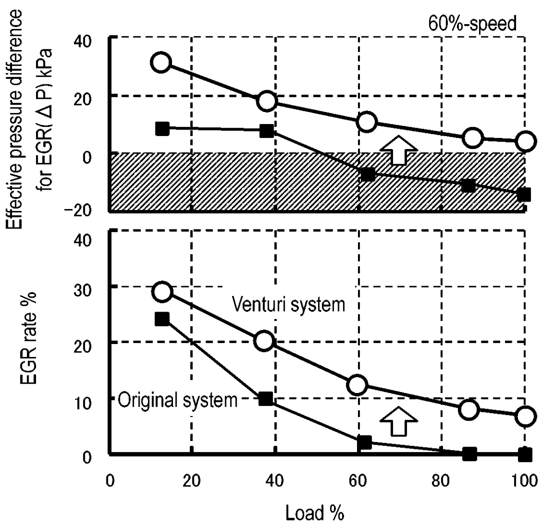

Fig. 17 Advantages of venturi system on EGR

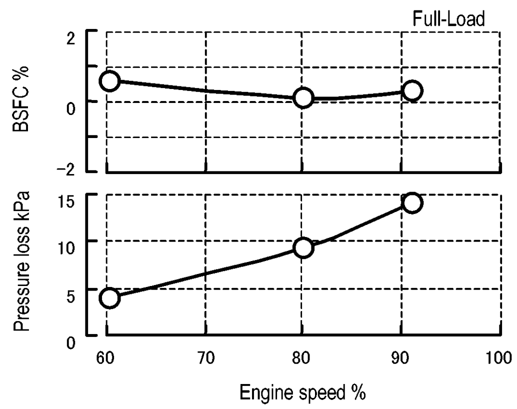

Fig. 18 Effect of pressure loss on fuel consumption

This study thus confirmed that the venturi can expand the range of EGR without increasing the fuel consumption.

3.6 Comparison with other approaches

In order to verify the effectiveness of the venturi EGR system, it was compared with the system that depended on exhaust choking to expand the EGR-feasible range. **Fig. 19** compares the EGR rate and fuel consumption of the two systems. When the fuel consumption rate of the venturi system is compared with that of the exhaust choke system for the same 6.2 % EGR rate, for example, the fuel consumption of the venturi system is about 4 % better than that of the exhaust choke system. **Fig. 20** is a low-pressure indicator diagram,

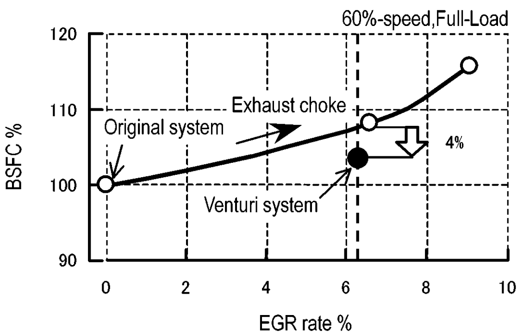

Fig. 19 Comparison of venturi and exhaust choke systems

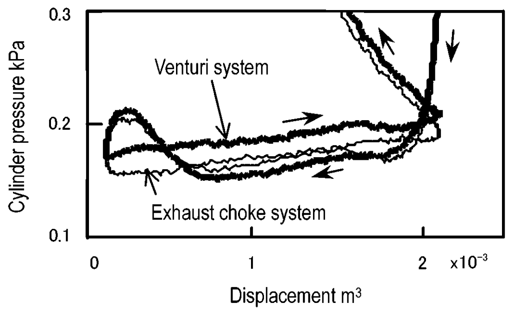

Fig. 20 Analysis of cylinder pressures

which shows why the venturi system is superior to the exhaust choke system in terms of fuel consumption. In the exhaust choke system, creating the $\Delta P > 0$ condition to take place EGR makes the exhaust stroke pressure of the cylinder higher than the intake stroke pressure, which significantly increases the pumping loss. In the venturi system, on the other hand, the intake stroke pressure remains higher than the exhaust stoke pressure throughout the operation, thus making EGR possible, which means that the pumping loss constitutes positive work that enables the unique merits of a turbocharged engine to be achieved. For this reason, the venturi system is superior to the exhaust choke system in terms of fuel consumption.

The pressure indicator diagram shows that the venturi system functions successfully as intended, creating the $\Delta P > 0$ condition in the EGR entry section and allowing the original boost pressure to be restored downstream of that section, so EGR takes place without any increase in pumping loss or fuel consumption.

4. Summary

(1) The application of EGR to diesel engines achieves larger reductions in NOx emissions under a high-load condition than a low-load condition for the same EGR rate. In other words, under the high-load condition, a low EGR rate produces the same NOx emission reduction effect as a high EGR rate under the low-load condition.

(2) It was shown that a method for expanding the EGR-feasible load range to high-load operation is essential when applying EGR to turbocharged diesel engines, and that the venturi EGR system achieves this objective more effectively than the conventional exhaust choke system and VG turbocharger system.

(3) It was shown that the venturi EGR system expands the EGR-feasible range without adversely affecting fuel economy caused by an increase of pumping loss, and it is thus an effective system for turbocharged diesel engines.

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