DEVELOPMENT OF HIGH-EFFICIENCY CO-GENERATION SYSTEM -HIGH EFFCIENCY GAS ENGINE FOR CO-GENERATION-

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1 BACKGROUND

Co-generation systems, which can supply electricity and heat on site, are getting popularity and the market is expected to grow in the future, since it is finding favor for their economic merit, energy-saving features, and high environmental performance(Fig.1). In Japan, the capacity increases rapidly in the recent decade, reaching 800 MW (power generation) in the service area of Tokyo Gas. Pushed by growing awareness of environmental concerns, further increase is projected [1].

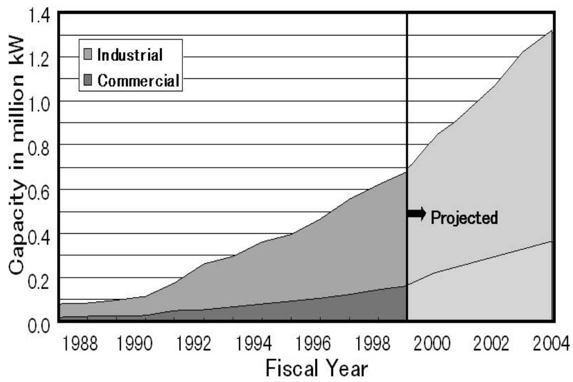

Fig.1 Utilization of co-generation in the Tokyo area

In Fig. 2, co-generation systems are classified on a power generation capacity basis. Gas turbines are often applied for large-scale classes over 1MW power generation capacity. On the other hand, gas engines fall into relatively below a few MW classes suitable for commercial facilities in urban areas. Furthermore, gas engines have electrical efficiency superior to gas turbine. Co-generation systems with small power generation capacity, such as micro gas engines and turbines, are also in wider use recently. However, these small equipments suffer from lower electrical efficiency. To compensate this drawback, it is necessary to investigate optimized ways of energy usage through, for instance, effective use of heat for small co-generation systems.

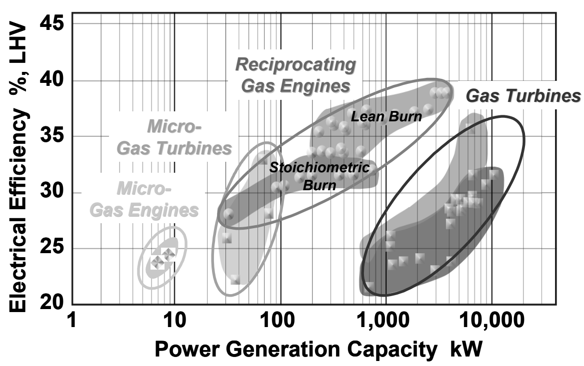

Fig.2 Classification of co-generation power units

In Japan, heat demand is much less than electricity needs in cities and buildings and, therefore, high power generation efficiency is desired. In view of increasing installation in urban commercial areas, 300-500kW class co-generation systems with high generation efficiency are much demanded. For achieving higher efficiency, it is essential to improve efficiency of the co-generation engine itself. As shown in Fig. 2, use of gas engines is suitable in increasing efficiency of 300～500kW-class systems. Hence, aiming toward higher power generation efficiency, development of a highly efficient lean burn gas engine was started. As a first step, experimental studies were conducted on component technologies to improve efficiency using a test engine. This was followed by collaborative development project for a commercial system between Tokyo Gas Co., Ltd. and Yanmar Co., Ltd. The recent outcome was achievement of thermal efficiency of 43 %, the highest among the class of engines.

There are largely two combustion methods in existing gas engines: stoichiometric combustion, in which in-cylinder mixture is burned with the theoretical mixture ratio, and lean burn for which lean mixture is employed. In stoichiometric combustion, exhaust gas can be cleaned well using three way catalyst. Emission of nitric oxides (NOx), in particular, can be suppressed below 40 ppm near the exhaust pipe exit. So it is a clean combustion method. Since combustion temperature is very high, much heat may be recovered and high general efficiency as a co-generation system is achieved. But it is prone to abnormal combustion, or knocking, due to very high temperature reached, which lowers efficiency.

While combustion temperature is low in lean burn, low NOx emission may be maintained even without after-treatment. It is also less susceptible to knocking and, therefore, is possible to yield higher power output from the engine compared with stoichiometric combustion. As a result, power generation efficiency could be increased.

In the present development, thermal efficiency of 43 % has been achieved through combination of an improved lean burn method, adoption of Miller cycle, knock suppression techniques, increased power output, and IT technologies for comprehensible control of these features. The achieved efficiency is the highest among 350kW (rated output) class engines. The present paper describes effort in achieving higher engine efficiency. It also presents specification of a commercial co-generation package and future development for commercialization.

2 ABBREVIATIONS AND SYMBOLS

BDC, aBDC	: Bottom Dead Center, after BDC	BMEP	: Brake Mean Effective Pressure
FMEP	: Friction Mean Effective Pressure	IVC, LIVC	: Intake Valve Closure, Late IVC
Ploss	: Pumping Loss	TDC, bTDC	: Top Dead Center, before TDC
TNA	: Turbine Nozzle Area	T/V, θ T/V	: Throttle-Valve, T/V Open Angle
π c	: Pressure Ratio of the Compressor	π t	: Pressure Ratio of a Turbine
λ	: Air Excess Ratio	WOT	: Wide Open Throttle

3 STUDIES OF COMPOMENT TECHNOLOGIES TOWARD HIGHER EFFICIENCY

3.1 Key Technologies toward Higher Efficiency

For achieving higher efficiency, a pre-chamber gas engine has been focused on, for which ignition and combustion of lean mixture is assured. As a first step toward higher efficiency of a pre-chamber gas engine, comparison is made between a conventional gas engine and a latest model high-efficient diesel engine for truck. Aim of comparison is to find the proper guideline for improving gas engine performance. Various experimental works and a cycle simulation are performed to examine underlying factors attributable to the difference in engine efficiency.

It is found that a difference of almost 7 points in thermal efficiency is found between the diesel and gas engines and that the difference may be well explained by the factors cited in Table 1. Based on the findings, four major tasks are set to realize higher efficiency of gas engines as follows:

(1) Increase the expansion ratio
(2) Reduction of unburned hydro carbon
(3) Reduction of the intake and exhaust losses
(4) Improvement of mechanical efficiency

Field tests are conducted with view toward higher efficiency based on the above results.

	Gas Engine	Diesel Engine	Difference in Thermal Efficiency
Compression Ratio	13.1	15.5	***1.1%***
Combustion Efficiency	Low	High	***1.9%***
PMEP kPa	60	0	***1.5%***
Fuel	$C_{1.2}H_{4.4}$	$C_{13.5}H_{23.6}$	***0.3%***
FMEP kPa (Mech. Efficiency %)	230 (82.6)	160 (88.1)	***2.2%***
Thermal Efficiency	36.8	43.8	***7.0%***

Table1 Comparison between Gas Engine and Diesel Engine

3.2 Test Engine

A conventional pre-chamber lean burn combustion engine is used for fundamental tests toward higher efficiency. Table 2 summarizes specifications of the test engine.

Experimental research work was conducted about the effect of the four tasks on engine efficiency.

Type	Spark-Ignited 4 Stroke Cycle Pre-Chamber Lean Burn
Cylinders × Bore × Stroke (mm)	6 × 165 × 185
Displacement Volume (littre)	23.7
Rated Power (kW / rpm)	324 / 1500
BMEP (MPa)	1.09 MPa
Supercharging	Turbo-Charger / Inter-cooler
Fuel	Natural Gas:LHV=41.6MJ/Nm^3 (88.5%-CH_4, 4.6%-C_2H_6, 5.4%-C_3H_8, 1.5%-C_4H_{10})
Expansion Ratio (Geometric Compression Ratio)	13.1

Table 2 Engine Specifications

3.3 Increasing in the Expansion Ratio and Mechanical Efficiency (Task1 and Task4)

There are two methods for increasing mechanical efficiency: design an engine for low friction loss, or increase power density, thereby decreasing relative friction loss. In the present study, the latter approach is chosen to examine effects on efficiency improvement.

A means of increased power density (BMEP) is accompanied by larger thermal load in the cylinder, which can cause knocking. Based on the past studies, the most effective anti-knocking method is to convert the engine to Miller cycle. This approach does not affect mechanical efficiency caused by increased expansion ratio associated with Miller cycle. As a result, combination of a larger expansion ratio and power density contributes much to improvement in mechanical efficiency. Hence, Miller cycle technology is adopted and effects are examined.

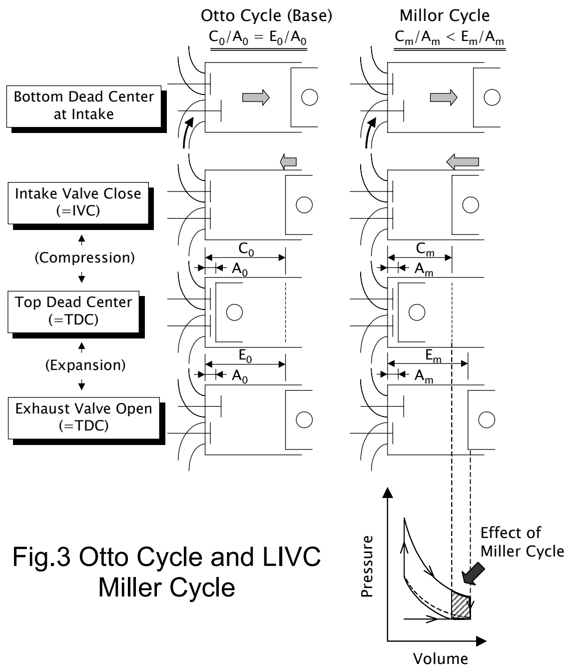

Fig.3 Otto Cycle and LIVC Miller Cycle

3.3.1 Effects of Miller cycle [2, 3, 4]

In Miller cycle, the expansion ratio of engine cycle is larger during combustion than that during the compression stage. It allows yielding more work from expansion, which is positive work, than from compression (negative work), thereby increasing mechanical efficiency. Compression and expansion ratios are adjusted by closure timing of the intake valve. There are EIVC (Early Intake Valve Closing) and LIVC (Late IVC) in Miller cycle, both of which aim to gain the compression ratio < the expansion ratio. A schematic diagram of the conventional cycle in which the compression ratio = the expansion ratio and LIVC is presented in Fig. 3.

Effects of intake valve closure (IVC) timing which is a representative index of Miller cycle on effective expansion ratio, intake manifold pressure and intake port temperature are shown in Fig. 4. In this example, power output, the expansion ratio, and inter cooler exit temperature remain constant. The effective compression ratio, which differs from the compression ratio determined from the geometrical configuration of engine, refers to an effective compression ratio defined under the assumption that mixture undergoes polytrope compression in the cylinder. As shown in Fig. 3, the more retarded intake valve closure timing (IVC hereafter), the smaller "Cm". Hence, effective compression ratio reduces by 2.35, from 13.24 to 10.89, as IVC changes from IVC=46degCA aBDC to 100degCA aBDC (a). On the other hand, LIVC shorten the actual compression stage. Consequently, in order to maintain power output with LIVC, combustible mixture must be pushed into a smaller volume with a higher pressure. Therefore, conversion to Miller cycle through LIVC increases intake manifold pressure (b). Furthermore, in LIVC Miller cycle, mixture once compressed into the cylinder is pushed back to the intake manifold, which raises intake port temperature.

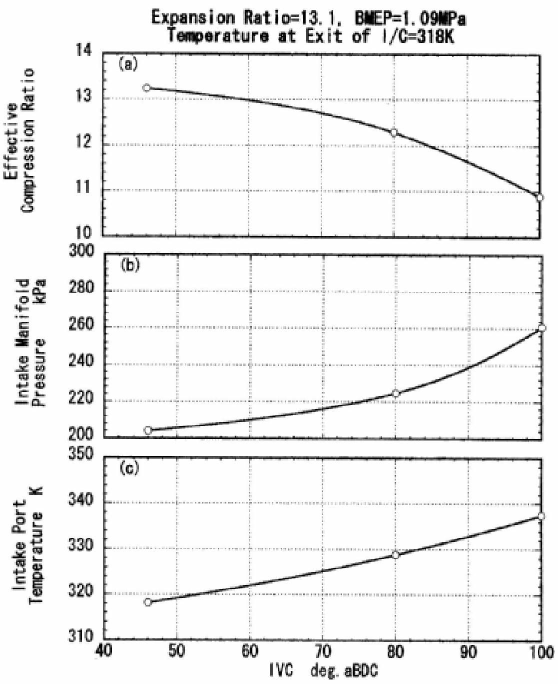

Fig.4 Effects of Miller Cycle

Effects of IVC on engine performance under a constant expansion ratio of 13.1, BMEP=1.09, and IT=9degCA bTDC are shown in Fig. 5. Even at the same air excess ratio λ, LIVC (i.e. enhanced

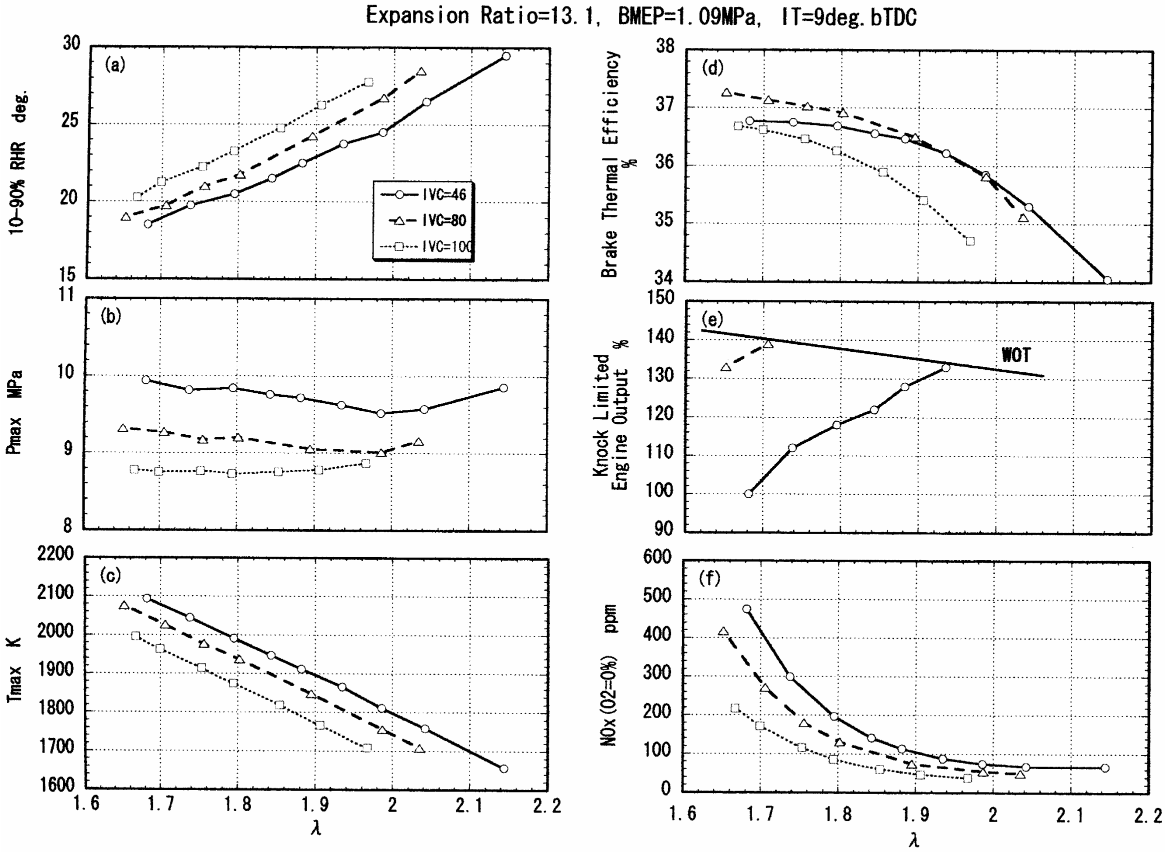

Fig.5 Effects of IVC Timing on Engine Performance

Miller cycle) lowers combustion velocity; see (a). This is due to a lower compression TDC temperature through a reduced effective compression ratio under LIVC. Lower effective compression ratio and combustion velocity decrease in-cylinder maximum pressure and temperature (Pmax and Tmax, respectively), as seen in (b) and (c). This lowers NOx emission (f). It is seen in (d) that thermal efficiency is highest at IVC=80degCA aBDC, which may mainly be attributable to matching with the turbo-charger. The knock limit improves significantly owing to lowered temperature and pressure levels through Miller cycle. In this experiments, knocking does not occur even with WOT (Wide Open Throttle) under IVC=100degCA aBDC; see (e).

Figure 6 represents the relationship between NOx and thermal efficiency under the same experimental condition as Fig. 5. LIVC (promoted Miller cycle) realizes engine operations under a lower NOx range without harming thermal efficiency.

3.3.2 Increase in the expansion ratio

Improved the knock limit through LIVC enables larger effective compression ratio and expansion ratio. Hence, an attempt is made to increase the expansion ratio at IVC=80degCA

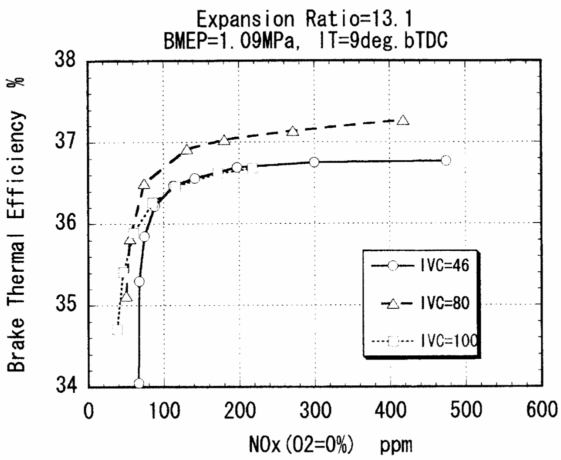

Fig.6 Relationship between NOx and Thermal Efficiency

	Baseline Otto Cycle	LIVC Miller Cycle
Intake Valve Close (deg aBDC)	46	80
Expansion Ratio	13.1	15.0
Effective Compression Ratio	13.24	14.05
Temperature at Exit of I/C K Intake Port Temperature	318 318	318 328

Table3 Specification of Otto and Miller Cycle

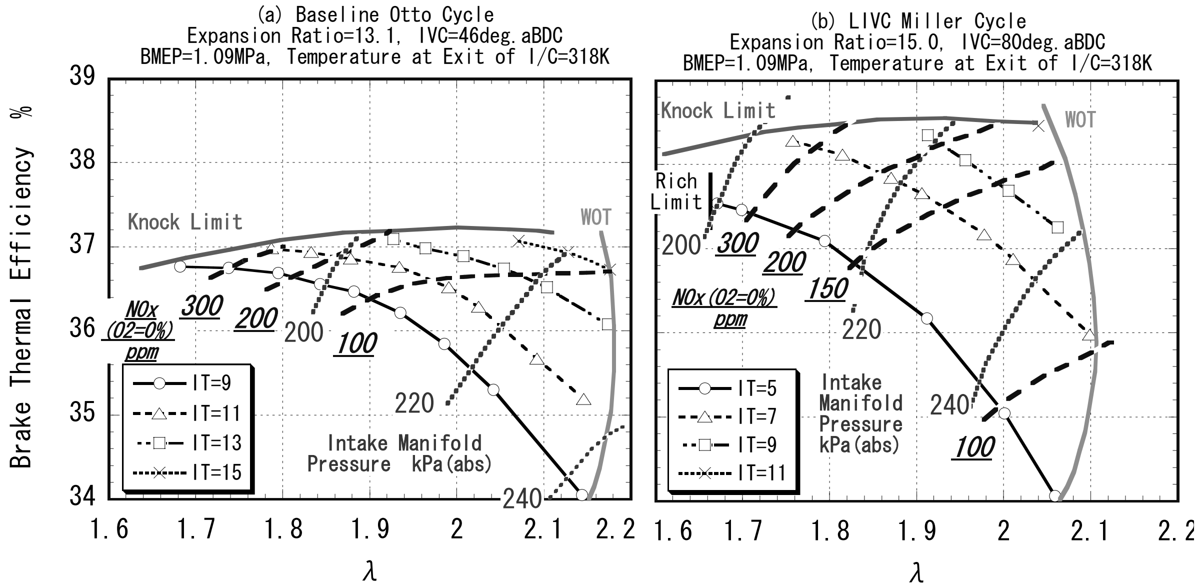

Fig.7 Comparison between Otto and Miller Cycles

aBDC. Table 3 compiles specification of baseline Otto cycle and LIVC Miller cycle.

A map representation of engine performances of (a) Otto cycle and (b) LIVC Miller cycle at a constant BMEP of 1.09 is shown in Fig. 7. By taking ignition timing as a parameter, the experiment is performed over a range extending from λ at knocking limit corresponding to WOT. Iso-NOx contours (O2=0%) and isobars of the intake manifold are also plotted in the figure. From the results, data shift toward higher efficiency through LIVC at IVC=80detCA aBDC and by increasing the expansion ratio to 15.0. When comparison is made at the knock limit, efficiency can improve by about 1.0% in the same NOx level.

3.3.3 Improvement in BMEP [5]

Performance of the engine operated at the knock limit BMEP by varying λ at each ignition timing for the engines having the specification listed in Table 3 is shown in Fig. 8. In addition, Otto cycle efficiency is also displayed for net thermal efficiency results.

It is seen from (b) that even increasing BMEP does not cause significant changes in Indicated Thermal Efficiency. Although the absolute friction loss increases (c), mechanical efficiency improves (d) since FMEP decreases in comparison with BMEP. As a result, net thermal efficiency increases as BMEP increases; see (a).

In the present engine, mechanical and thermal efficiency improves by about 30% and 1.0%, respectively, by increasing power output by about 30 % from the baseline BMEP of 1.09MPa.

To summarize, it is demonstrated that Miller cycle enables low NOx operations without lowering efficiency. Also, Miller cycle is much effective in improving the knock limit and it is possible to increase the expansion ratio and to further improve efficiency with more power output.

3.4 Reduction in Unburned THC (Task2)

Lowering unburned THC cuts waste energy and contributes to higher mechanical efficiency. Presence of unburned mixture during the

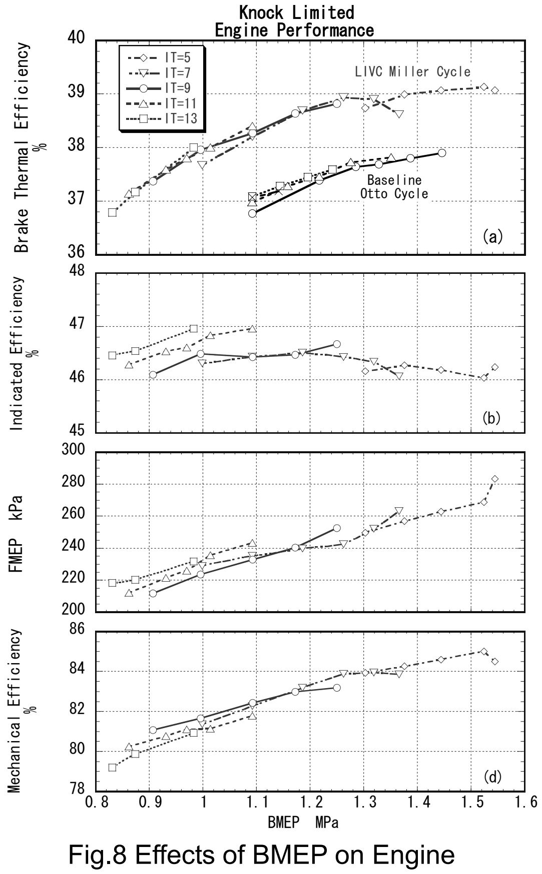

Fig.8 Effects of BMEP on Engine Performance

combustion stage can cause emission of unburned THC. Also, if portion of combustible mixture is pushed into a small space inside the cylinder called crevice, it does not burn at all and is exhausted as unburned THC. Here, the crevice volume is reduced to examine experimentally its effect on lowering unburned THC.

3.4.1 Effects of Crevice Volume Reduction

Effects of the crevice volume on engine performance under the expansion ratio of 14.7, IVC=80deg.aBDC, BMEP=1.09MPa, and ignition timing of 5deg.bTDC are shown in Fig.9. And fig.10 indicates the relationship between crevice volume and engine performance. Since combustion velocity depends largely on the shape of the combustion chamber, performance changes little with the crevice volume as seen in (c). Under this condition, unburned THC contained in the exhaust depends on the crevice volume (b) and, as a result, thermal efficiency increases irrespective of λ values.

As demonstrated above, there are significant effects of crevice volume reduction on cutting unburned THC in the lean burn engine. Also, thermal efficiency is shown to increases by about 1 point.

3.5 Reduction of Intake and Exhaust Losses (Task3)

In general, gas engines suffer from large pressure losses from intake and exhaust systems, which directly leads to large pumping losses (Ploss) and lower engine performance. Especially, since engine speed and output is controlled using throttle, a large pressure loss is always present across the throttle.

3.5.1 Effects of Throttle-less Operation

First, effects of reducing throttle loss as a major factor among intake and exhaust losses are verified.

In the experiment, the throttle open angle is adjusted so that for each turbine nozzle (TNA) constant values of BMEP=1.09MPa, IT=7deg.bTDC, and λ=1.78 are maintained. A dynamometer is used to control engine speed.

Turbo-charger characteristics are shown in Fig. 11. By increasing TNA from 22 to 27, πc is reduced roughly from 2.4 to 2.0; see (a). Note that in (b), turbo-charger efficiency remains about 60 % for any TNA.

Engine performance is presented in Fig. 12. Since increasing TNA from 22 to 27 lowers πc by 0.4point, θT/V increases from 16.5° to 70° (WOT) as seen in (a) and the pressure difference across throttle (ΔPT/V) decreases by 40kPa (b). This leads to reduction in Ploss by about 26～27kPa (c), and

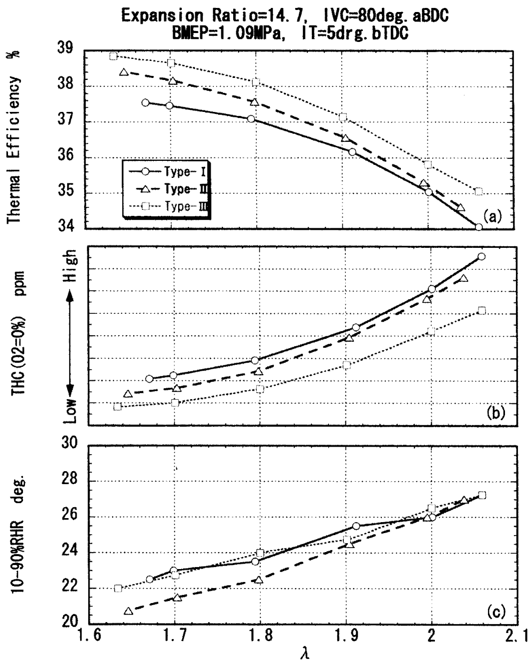

Fig.9 Engine Performance of the Three Combustion Chamber Types

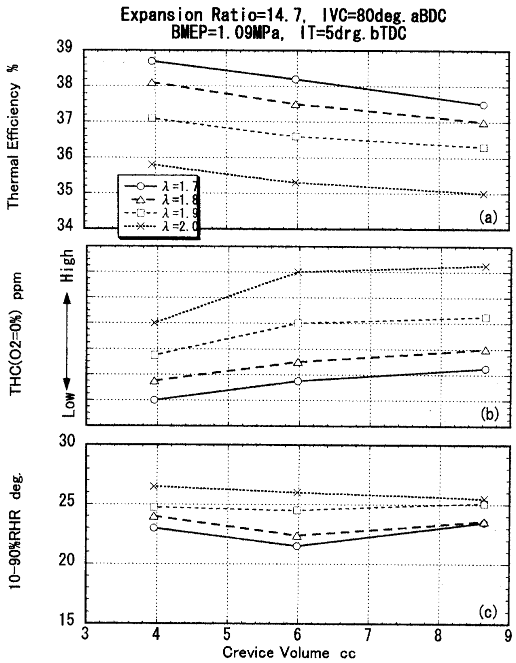

Fig.10 Effects of Crevice Volume on Engine Performance

an increase in thermal efficiency by almost 0.5 point; see (d).

Reduction in the pressure difference across the throttle is thus seen to improve thermal efficiency. Furthermore, since smaller Ploss reduces back pressure of the exhaust side, residual exhaust gases trapped in the cylinder decreases, leading to a better knock limit. As a result, it is expected to increase efficiency by higher power output.

3.6 Summary of Thermal Efficiency Improvement

So far, assessment has been made of the tasks for improved thermal efficiency one by one. Effects of these technologies applied to the pre-chamber lean burn gas engine on thermal efficiency are now examined.

The process toward higher efficiency is as follows: The baseline Otto cycle has an expansion ratio of 13.1 and IVC=46deg.aBDC. Upon application of Miller cycle, the expansion ratio is changed to 14.7 together with IVC=80deg.aBDC. Furthermore, the Miller cycle engine has a smaller crevice volume and is subjected to throttle-less operations.

Thermal efficiency under the operating condition of the knock limit BMEP through adjusted air excess ratio λ at ignition timing is displayed in Fig. 13. To ease comparison, the data corresponding to the differential pressure across throttle of 20 kPa are selected and engine performance is re-plotted in Fig. 14. Adoption of Miller cycle increases the expansion ratio from 13.1 to 14.7 and BMEP from 1.09 to 1.50MPa. Furthermore, cutting the crevice volume reduces THC by about 50 % (b) and throttle-less operation decreases Ploss by nearly 40 % (c). Compared at the same BMEP, the knock limit improves as back pressure decreases due to lower Ploss, since absence of the throttle makes operations under lower λ possible; see (f).

Applying these technologies, thermal efficiency of the conventional pre-chamber lean burn gas engine improves significantly from the baseline value of 36.8 % to 40.8 %.

4 DEVELOPMENT OF A NEW GAS ENGINE

Based on the research outcome for the major tasks toward higher efficiency discussed so far, a new type of gas engine is developed at Tokyo Gas Co., Ltd. in collaboration with Yanmar Co., Ltd. The newly developed engine belongs to a medium to large class with 300kW power output, which is the mainstream on the Japanese market. The new gas engine is of pre-combustion chamber type for which ignition and combustion of lean mixture is

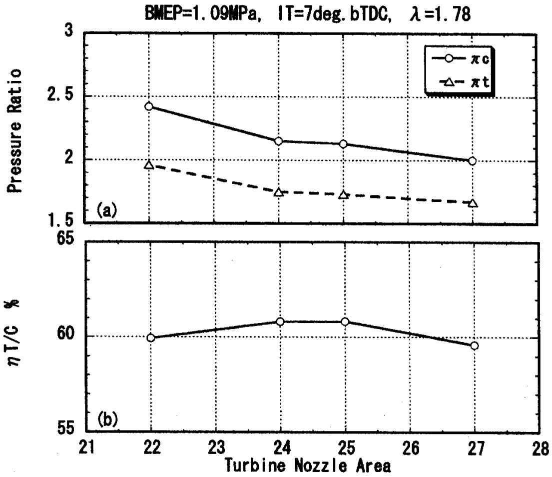

Fig.11 Effects of the Turbine Nozzle Area on Turbo-Charger Performance

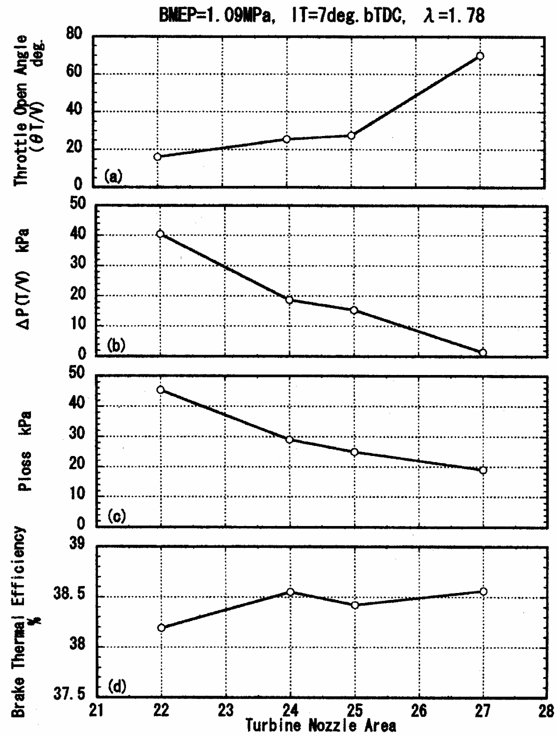

Fig.12 Effects of the Turbine Nozzle Area on Engine Performance

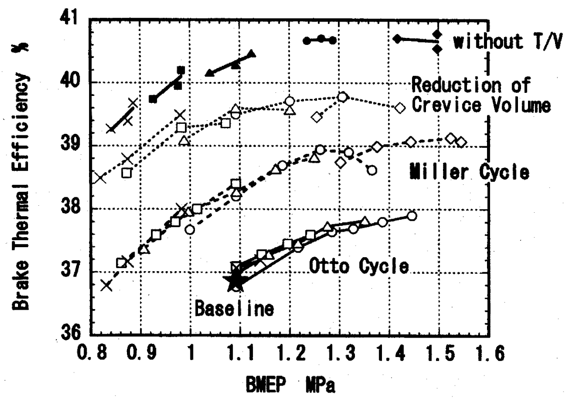

Fig.13 Improvement in Thermal Efficiency

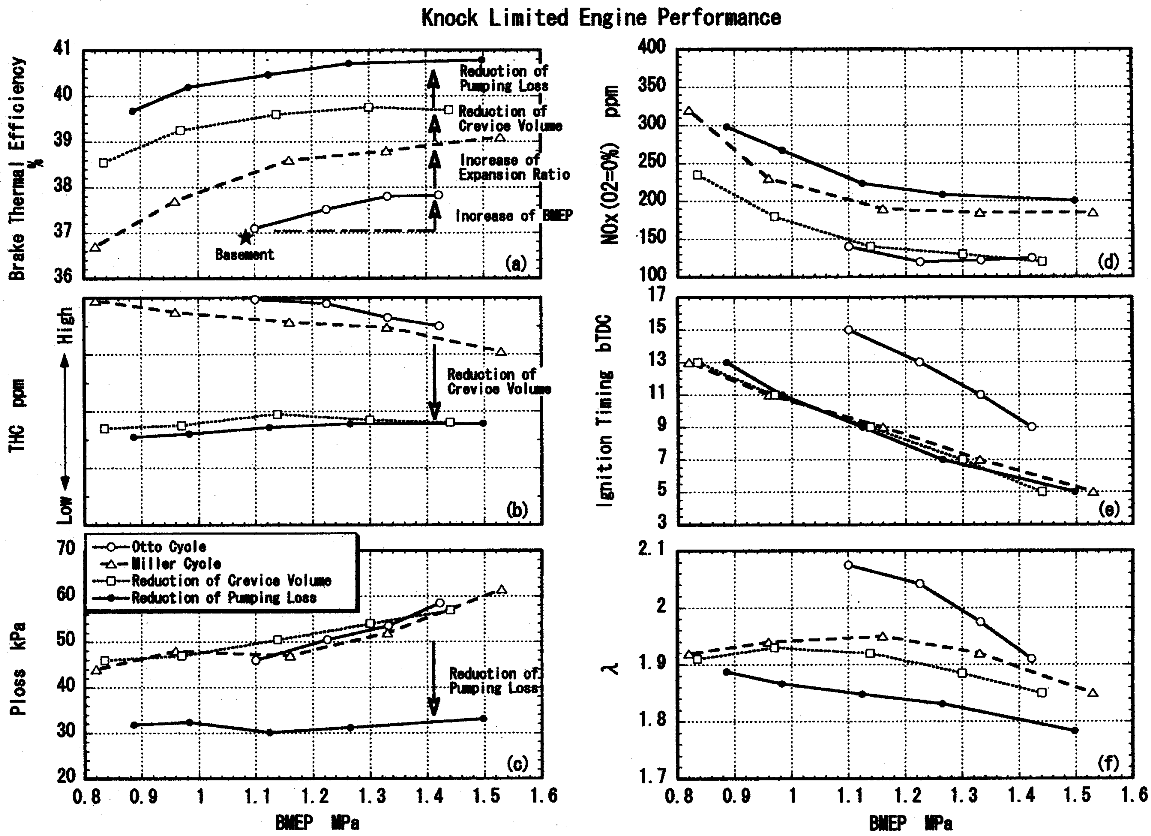

Fig.14 Effect of Expansion Ratio, BMEP, THC and Pumping Loss on Engine Performances

assured. Specification of the new gas engine, whose overview is shown in Fig. 15, are presented in Table 4.

For achieving higher efficiency, the new technologies shown in Fig. 16 are applied to the new gas engine, thereby fulfilling the following four tasks set in the previous study on component technologies.

(1) Increase the engine expansion ratio
- Adoption of EIVC Miller cycle

(2) Reduce unburned THC
- Cutting waste volume in the combustion chamber
- Combustion improvement

(3) Reduce intake and exhaust losses
- Speed/power output control using gas port injection without conventional throttle

(4) Improve mechanical efficiency
- Relative reduction in friction loss by increased power output
- Low friction engine design

More specific description is provided below

4.1 Increasing the Engine Expansion Ratio and Improving Mechanical Efficiency (Tasks 1 and 4)

(1) Miller cycle

As discussed in 3.3, conversion to Miller

Fig.15 Side View of the New Gas Engine

Type	Spark-Ignited 4 Stroke Cycle Pre-Chamber Lean Burn
Cylinders×Bore×Stroke (mm)	6×155×180
Displacement Volume (littre)	20.4
Rated Power (kW / rpm)	382 / 1500
BMEP (MPa)	1.50 MPa
Supercharging	Turbo-Charger / Inter-cooler
Fuel	Natural Gas:LHV=41.6MJ/Nm³ (88.5%-CH4, 4.6%-C2H6 5.4%-C3H8, 1.5%-C4H10)

Table 4 Specification of the New Gas Engine

cycle makes low NOx operations possible without reducing efficiency, leading to an improved the knock limit. As a result, efficiency improves through an increase in the engine expansion ratio and higher power output.

In the present engine, adoption of EIVC Miller cycle increases the expansion ratio, thereby increasing BMEP by as much as roughly 30 %. Hence, improved efficiency is gained.

(2) Optimization of valve overlapping

In conventional engines in which gas and air are premixed inside the mixer before they are fed to the cylinder, the valve-overlapping period is set short in order to prevent unburned mixture slip towards exhaust port.

In the new gas engine, applying gas injection could prevent unburned mixture slip. It allows a longer valve-overlapping period than that of conventional gas engines. This leads to a reduction in residual gas ratio and intake/exhaust efficiency. At the same time, a drop in the cylinder temperature due to the reduction of residual gas ratio improves the knock limit, thereby achieving higher engine output.

(3) Low friction engine design

Low friction of the base engine is crucial for attaining higher engine efficiency. In developing the new gas engine, friction of the slide portion is reduced, thereby achieving significantly low friction loss.

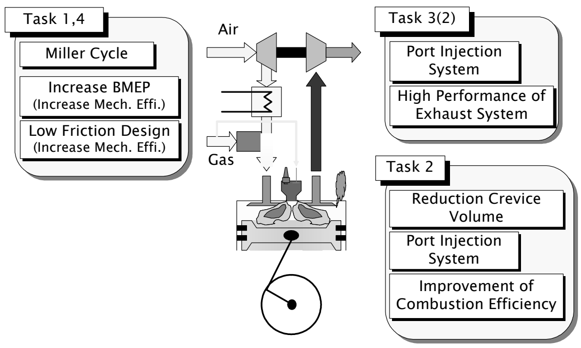

Fig.16 Key Technology toward a Higher Efficiency Gas Engine

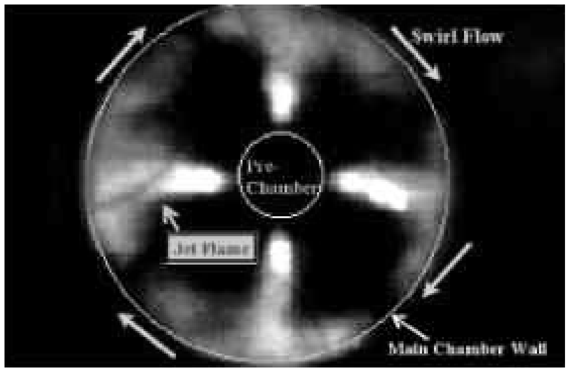

Fig.17 Combustion Photograph

4.2 THC Reduction (Task2)

(1) Crevice volume reduction

As described in 3.4, reduction in THC in the exhaust directly links to improvement of engine efficiency. In new gas engine development, effort is made to reduce the crevice volume in the cylinder. As a result, exhaust THC is cut and efficiency is improved.

(2) Combustion improvement [6]

In developing the new gas engine, the shape of pre-chamber, flow in the engine, the main combustion chamber shape, and top clearance have been modified in order to improve combustion.

Especially in the pre-chamber type, characteristics of flame jets, which are emitted from the pre-chamber to the main chamber after ignition, much affect succeeding combustion properties. Through fundamental research of direct observation in the combustion chamber, optimization of the number of nozzle holes and their directions in the pre-chamber was conducted to improve combustion (see Fig. 17).

4.3 Intake/Exhaust System Optimization (Task3)

(1) Adoption of gas injection

In conventional gas engines, the amount of mixture supplied to the cylinder is regulated by a throttle, thereby controlling engine load and speed. As discussed in 3.5, however, the throttle causes pumping loss (Ploss), resulting in lower engine efficiency. It is therefore decided to adopt the gas injection method in which fuel gas in supplied directly through the intake port without throttle to control load and engine speed. As a result, pressure loss at throttle, main cause of pumping loss, is removed and then pumping loss could be reduced.

(2) Pulse convert type exhaust system

By adjusting the blanches from each cylinder to the exhaust manifold, suitable flow speed and exhaust pressure pulses are designed to be generated in the exhaust manifold. The exhaust system thus constructed has intermediate properties between static and dynamic pressures, by which improved efficiency of the turbo-charger is realized.

4.4 Summary of Efficiency Improvement in the New Gas Engine

Conventional gas engines attain thermal efficiency of around 37~38 % with BMEP=1.1MPa. Through the effort discussed so far toward the higher efficiency new gas engine, efficiency improved by more than 5.0 points. Thermal efficiency of engine reaches the highest among the present class and is over 43 %. With the engine developed, a co-generation system equipped with superior electrical efficiency, which can fulfill customer needs, becomes feasible.

5 OVERVIEW OF A NEWLY DEVELOPED CO-GENERATION PACKAGE

As presented above, engine thermal efficiency, the highest in-class value, of 43 % has been accomplished through introduction of innovative technologies. By installing the high efficiency gas engine newly developed into a new co-generation package, finalization of product specification and durability tests for the overall package are under way. Electrical efficiency exceeds 40 %, a significant increase from existing co-generation systems. The newly developed system can satisfy environmental and economic evaluations of prospective customers as demonstrated in Fig.18.

Its main features are listed below:

(1) High power generation efficiency

The highest in-class thermal efficiency is achieved through application of pre-chamber lean burn Miller cycle combustion, gas injection per cylinder, and knock detection and ignition timing control technology.

(2) Environmental consideration – low NOx

Through adopting lean burn Miller cycle combustion and the micro pre-combustion chamber effective in combustion improvement of the engine, no after-treatment equipment such as de-nitrate catalyst is required up to NOx concentration of 200ppm (O2=0%). The present system can be introduced into areas where emission is regulated below 200ppm with concurrent use of after-treatment apparatus.

(3) High load input

By adopting the method of gas injection per cylinder, the world's first in below-500kW output class gas engines, comparatively high load input performance is retained among high efficiency lean burn gas engines. This feature allows the present system to be also used for a common disaster prevention system.

(4) High output and compactness

With application of the gas injection method, power output is improved by about 30 % compared to conventional systems. A compact and lightweight system has been constructed which is easy to install.

An oveview of the co-generation system is shown in Fig. 19. Remaining tasks include evaluation of reliability and durability of the present high efficiency gas engine, development of system control by applying IT technologies, and promoting commercialization of a gas engine co-generation package (by spring of 2003) equipped with remote monitor functions. Effort is directed toward wider use of high efficiency co-generation systems, which can contribute to reduction of CO2 as the main agent of global warming.

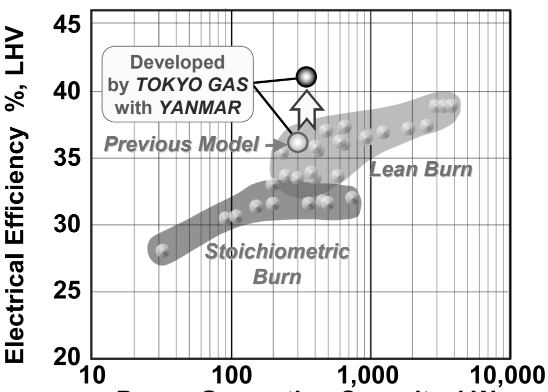

Fig.18 Generation Efficiency of New Co-Generation

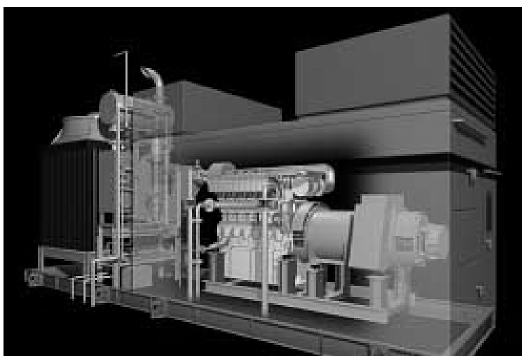

Fig.19 New Co-Generation Package

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