ISUZU 6WA1TC Turbocharged Diesel Engine

ABSTRACT

The GWA1TC engine was recently designed to meet the demands of higher output power, lower fuel consumption and longer useful life as well as meeting future tighter exhaust emission standards. This new turbocharged diesel engine is equipped with an intercooler, which was developed for 20-ton gross vehicle weight cargo trucks. The 6WA1TC diesel engine is the successor to the current 6RB1TC diesel engine, which went into production in July 1992.

In September 1994, Japan's stricter exhaust emission regulation became effective. In order to meet the stricter requirements, the 6WA1TC engine was improved for emission performance and installed to the brand new model truck series called "GIGA". 6WA1TC was developed for the worldwide market, but mainly will be sold in Asia and other markets.

The basic fundamental of the engine structure consists of an OHC 4-valve type cylinder head and a ladder frame type cylinder block. These improvements were made to satisfy the requirements for higher output power, lower fuel consumption and reduce engine weight. A high pressure fuel injection system was installed along with various variable systems, such as a variable geometry turbocharging system or a combined intake charging system, and a new combustion chamber was incorporated to meet the exhaust emission requirements and meet the demand of lower fuel consumption.

INTRODUCTION

In Japan market, the higher output power is continuously required because the gross vehicle weight limit for cargo trucks was raised up from 20-ton to 25-ton in November 1993, and customer's demand for tractors which equipped with a turbocharged engines with an intercooler increased (Fig 1). Also in accordance to implementation of Japan's stricter exhaust emission regulation in September 1994, the limits for NOx and black smoke were decreased and the limit for Particulate Matter was newly added (Table 1).

In order to meet this stricter requirements, 6WA1TC diesel engine was improved for emission performance without a change in fuel consumption, by means of adopting with the new combustion chamber and the combined charging intake system and so on. Table 2 shows the main specification and equipment of 6WA1TC which meet '94 Japan's emission regulations.

The target of useful life for this engine was set at 1,300,000 Km according to the demand for longer useful life in Japan market. Fig 2 shows the progress of useful life trends in Japan market and the target for this engine is plotted.

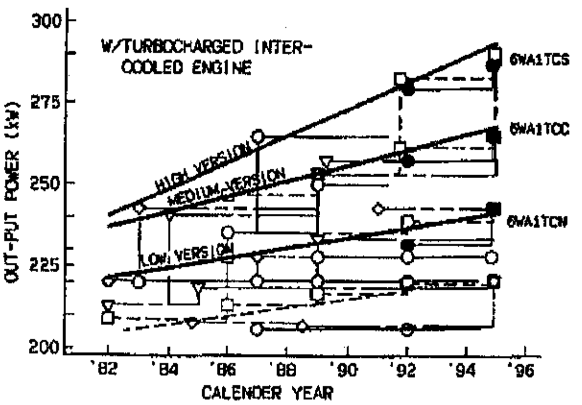


Figure 1: Progress in output power of cargo vehicle in Japan

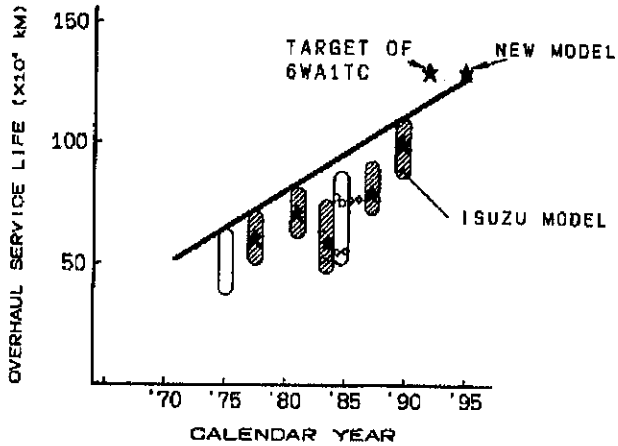


Figure 2: Progress of overhaul service life

	'90 Japan	'94 Japan	'94 US	'92 Euro 1	'95 Euro 2
NOx	400 ppm	6.0 g/kW h	5.0g /HP h	8.0 g/kW h	7.0g /kW h
PM	—	0.7 g/kW h	0.1 g/HP h	0.36 g/kW h	0.15 g/kW h
Smoke	50%	40%			

Table 1: Emission standards for diesel engine used in commercial vehicles exceeding 3.5 ton GVW

	6WA1TCN	6WA1TCC	6WA1TCS
Bore x Stoke (mm)	132.9 x 145		
Displacement (litre)	L6-12.068		
Output Power (kW/rpm)	243 / 2000	265 / 2000	287 / 2000
Torque (Nm/rpm)	1324 / 1200	1422 / 1200	1569 / 1200
Max. Rev. (rpm)	2580		
Inj. Pump	Bosch S3S	HD-TICS	HD-TICS
Governor type	Elec.		
Turbocharger	STD turbo	STD turbo	V.G.S. Turbo
VSS	with	with	with
CCS	with	with	with

Table 2: Specifications of Japan model

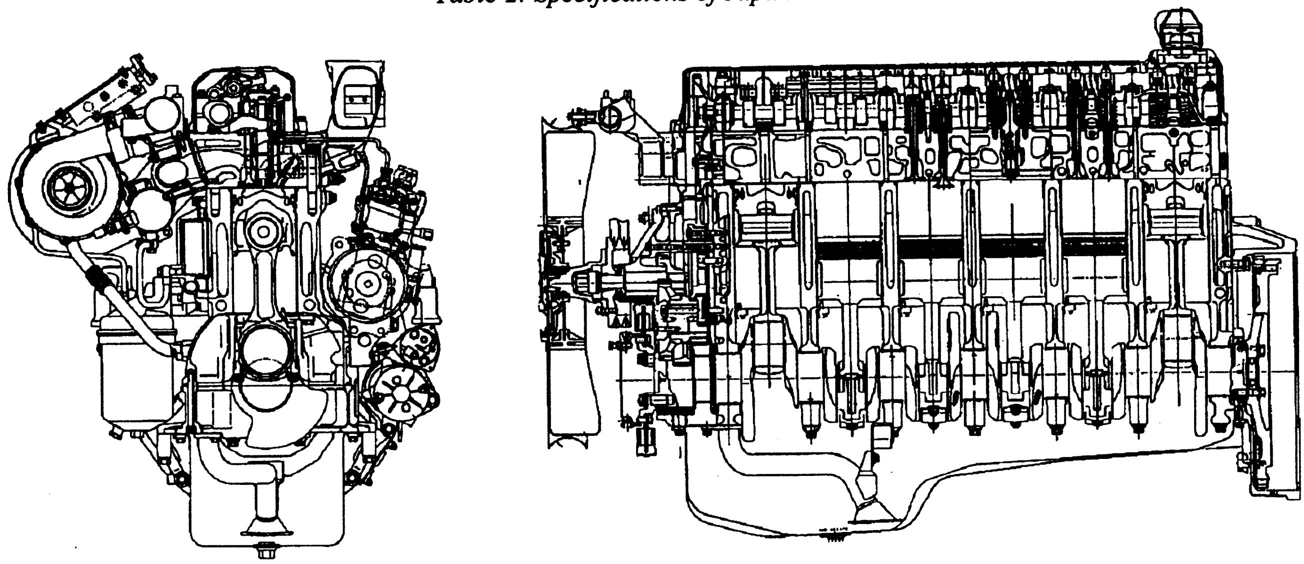


Figure 3: Crosswise and endwise section of 6WA1TC engine

DESIGN CRITERIA

Design criteria of the 6WA1TC engine are shown below;

1. 1,300,000 km useful life (overhaul service life)
2. The same external dimension as predecessor engine, 6RB1TC
3. Reduction of total number of parts by approximately 20% from predecessor
4. Maximum bore pressure; 14.7 MPa or more
5. Reduce engine weight and noise compare to predecessor
6. Capabilities of displacement increase for possible future higher output power requirements
7. Capabilities of power take-out from engine crank front for industrial applications

In order to satisfy those requirements, an OHC 4-valve type six-cylinder-one-piece cylinder head and a ladder frame type bearing support had been adapted as the basic fundamental of engine structure. Considering future displacement increase, the bore pitch was determined on 170 mm. Fig 3 shows crosswise and endwise section of 6WA1TC engine and Fig 4 shows its appearance.

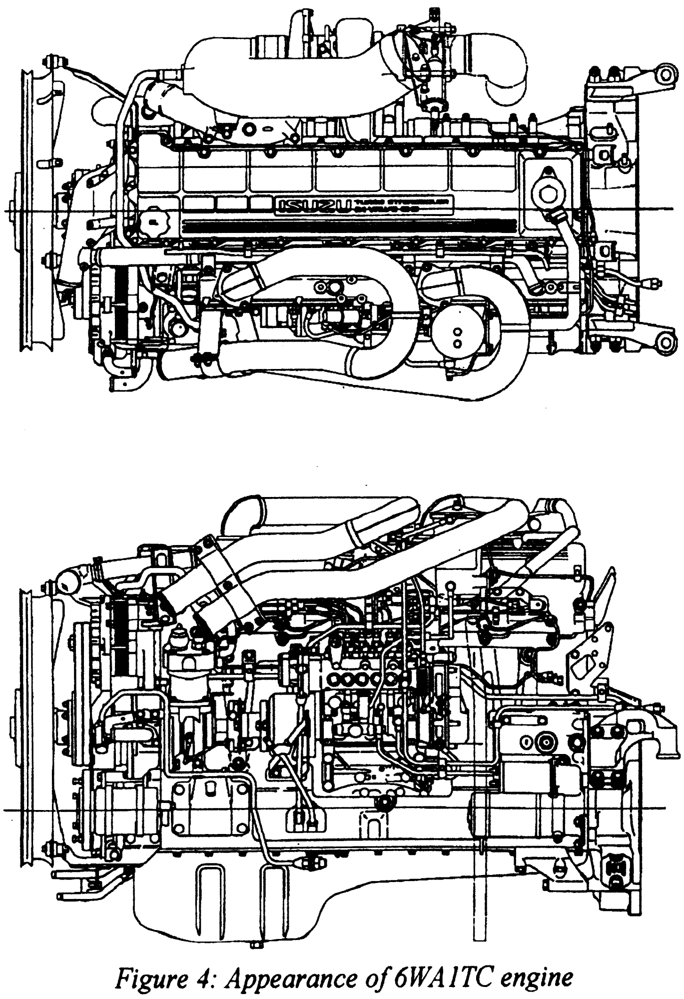


Figure 4: Appearance of 6WA1TC engine

DESIGN FEATURES

CYLINDER HEAD

A six cylinder one-piece cylinder head is made of ferroalloy and an aluminium cover is installed as the idle gear housing on front side. Valve layout can be also suitable for installation of unit injectors. Fig 5 shows the status of unit injector installation.

Inlet ports consists of a helical port equipped with a drilled sub-port for variable swirl and a tangential port. Sub-port is opened and closed by the air actuator mounted to the inlet manifold. A glow plug can be installed as an auxiliary starter under cold condition.

Fig 6 shows appearance of cylinder head and Fig 7 shows the cooling channel between each valve seat. Fig 8 and Fig 9 indicate the shape of variable swirl system and the port's performance.

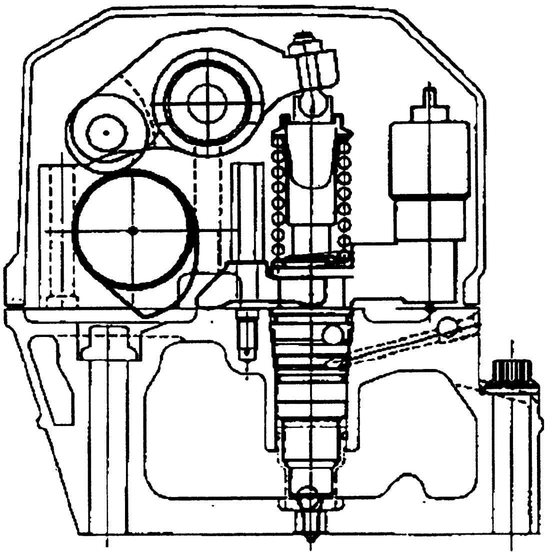


Figure 5: Status of unit injector installation

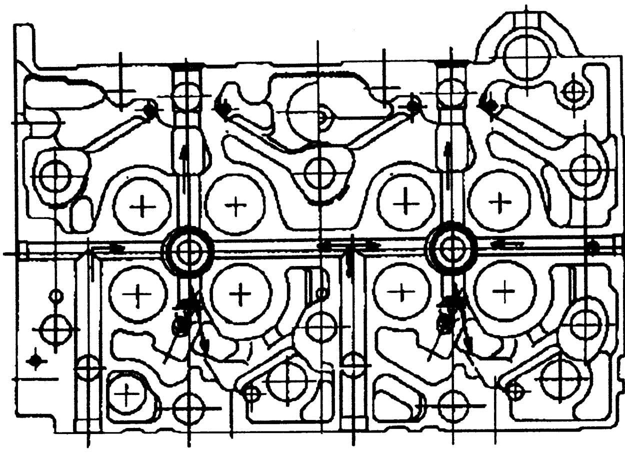


Figure 6: Cooling channel in the cylinder head

Figure 7: Appearance of cylinder head

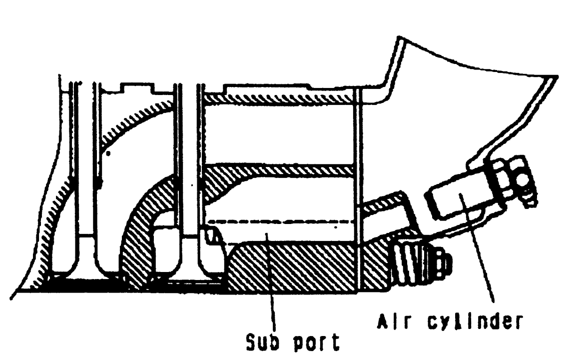


Figure 8: Variable swirl system

Figure 9: Inlet port performance

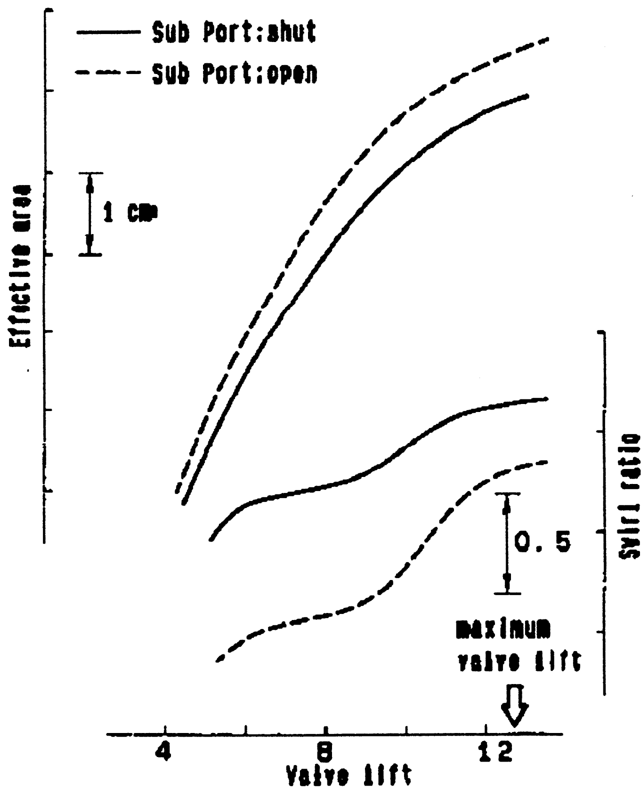


Figure 9: Inlet port performance

CYLINDER BLOCK

The cylinder block is of the ladder-frame type bearing support and the dry sleeve type is employed. The head bolt layout is of a 6-point arrangement of approximately equal pitch and coupled with the selection of side wall shape, which contributes to lessen bore deformation. The cylinder liner is made of special cast ferroalloy and the clearance fit method is adopted for mounting ease. Fig 10 shows cylinder block vibration characteristics compared the ladder frame type to the other methods. This method greatly contributes to the improvement in skirt vibration, and also to stress reduction of crankshaft.(1)

Side wall and rib shapes of cylinder block is so designed to accept the load of combustion force straight between head bolt and bearing support, and the side wall is of wave-shape which is apt to cause less distortion. Fig 11 shows the appearance of the cylinder block and Fig 12 indicates comparison in the bore distortion between the wave-shaped and straight-shaped wall.

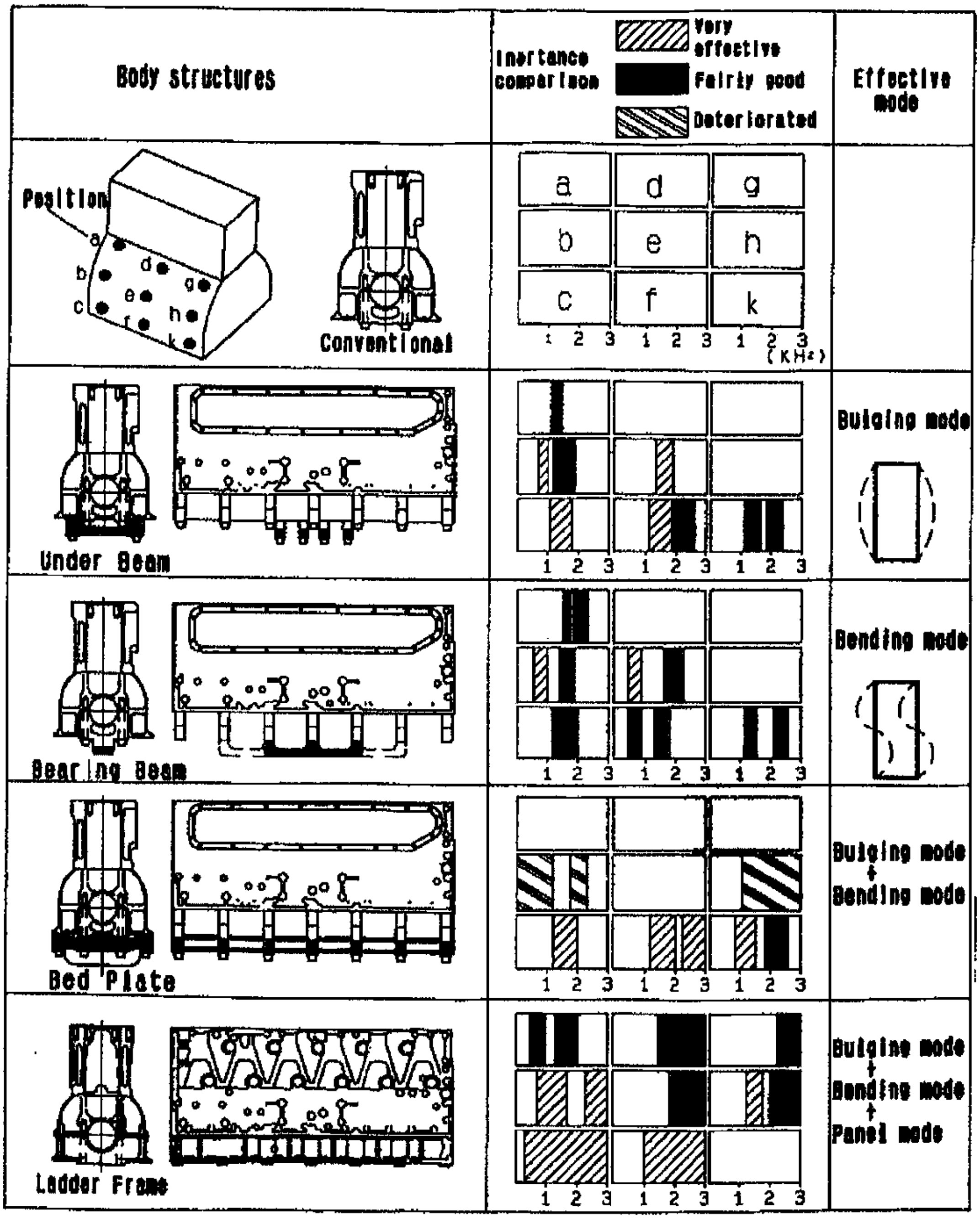


Figure 10: Bearing support method and block vibration characteristics

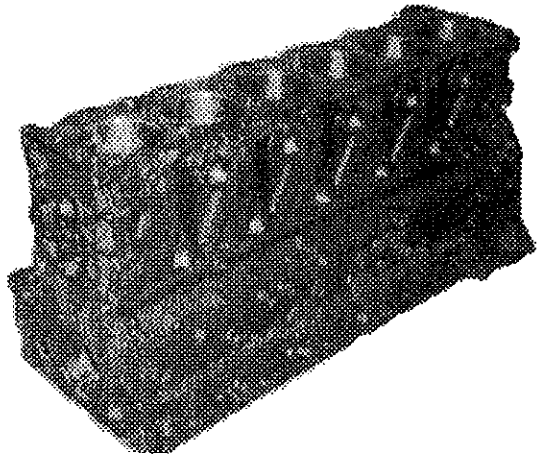


Figure 11: Appearance of cylinder block

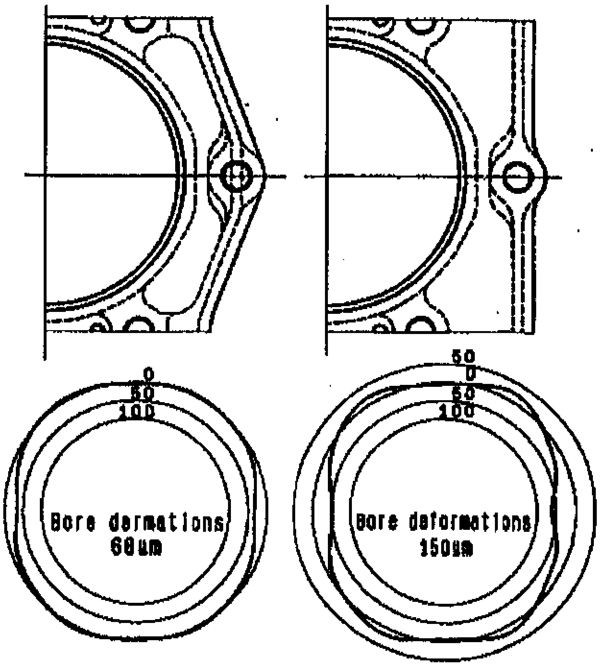


Figure 12: Comparison of block bore deformation between two different side wall shapes

CYLINDER HEAD COVER

The cylinder head cover is made of reinforced resin, the design of which make it possible to reduce weight and noise. Fig 13 shows comparison of noise level between die-cast aluminium head cover and synthetic resin head cover.

A useful life evaluation of the long one-piece head cover was carried out by preparing a limit distortion chart. Low cycle fatigue is especially important and shown below are the evaluation methods;

1. Detecting the weakest point by the photoelasticity method
2. Preparing for a limit distortion and fatigue chart using test pieces cut out of head cover (Fig 14-a)
3. Measuring the distortion caused by temperature changes during the on-board cruising test (Fig 14-b)
4. Calculating the overall stress on the Minor Rule
5. Evaluating the useful life using the limit stress chart and measured stress

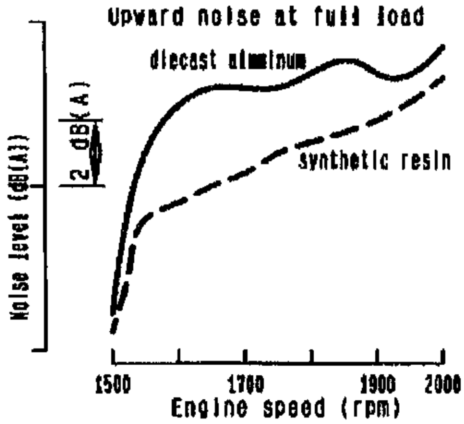


Figure 13: Comparison of noise level between die-cast aluminium and synthetic resin head cover

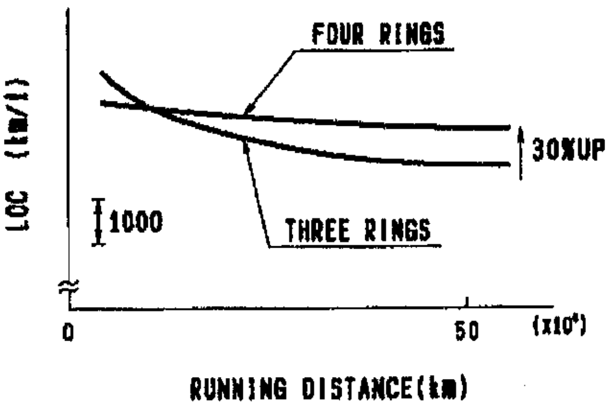


Figure 15 Comparison of lubricant oil consumption

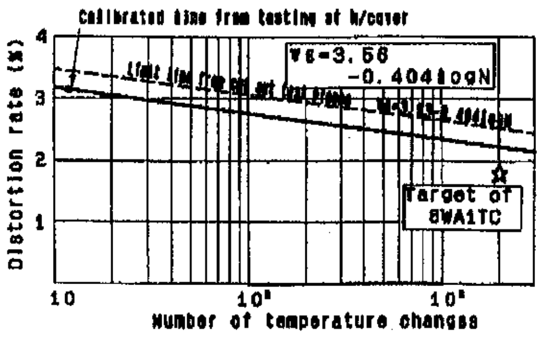


Figure 14a: Limit distortion chart of a synthetic resin head cover

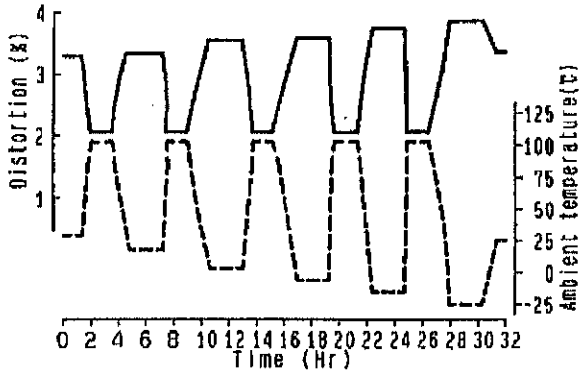


Figure 14b: Distortion characteristic of a synthetic head cover

PISTON AND RINGS

Pistons made of aluminium are employed and an integral Ni-resisting ring carrier are adopted as the 1st and 2nd ring-groove. Ring groove's temperature is maintained at 240°C or lower due to oil cooling channel between the combustion chamber and the ring groove. The rings are of a quadruple construction, which saves on oil consumption during useful life (Fig 15), and the key-stone type ring is adopted as the 1st ring to improve carbon resistance (Fig 16).

Fig 17 shows round and re-entrant shaped combustion chamber which contributes to reduce black smoke level and Particulate Matter emissions.

Piston pin has a larger diameter (56mm) so that it can endure a bore pressure of 14.7 MPa or more.

Position	Configuration	Width x Thickness	Material	Coating
1st	Barrel-faced	4.3x5.2	Ductile Cast Iron	Periphery Chromiumcoated
2nd	Taper-faced	3.0x5.2	↑	↑
3rd	Taper-faced	3.0x5.5	Grey Cast Iron	↑
Oil		4.0x2.9	STEEL	Chromiumcoated On Peripheral and Inside Surface

Figure 16: Piston ring structure

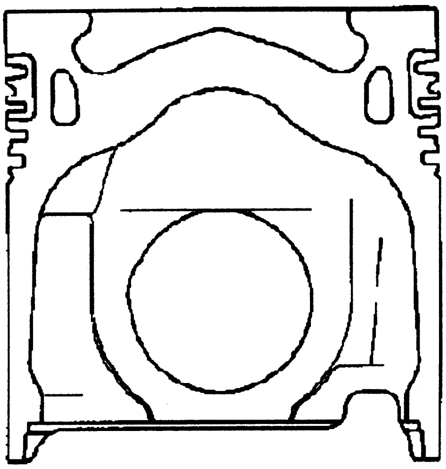


Figure 17: Piston and combustion chamber shape

CRANKSHAFT AND CONNECTING ROD

A four-counter shaped crankshaft made of carbon steel was adopted. For improvement of strength and abrasion resistance a nitriding finish is applied over all surfaces. Journal diameter is 105 mm and pin diameter is 92 mm. To the front end of crankshaft, eight M18 bolts can be set when taking out power for industrial applications. At the rear end, ten M18 bolts are set to secure the transmission of a torque level of 2000 Nm or more.

In the connecting rod, a drilled hole is provided from the large end side through the small end bushing for forced lubrication, thus improving bushing durability.

TIMING GEAR TRAIN

The timing gear train is positioned in the front section of engine for ease of accessories maintenance. The cooling fan and the water pump are changed to a gear-driven type for maintenance free. The overall view of the gear train is shown in Fig 18.

The gears are of a helical type and made of carbon steel with nitriding finished surfaces. A portion of idle gears is supported at both ends to drive the high pressure fuel injection pump.

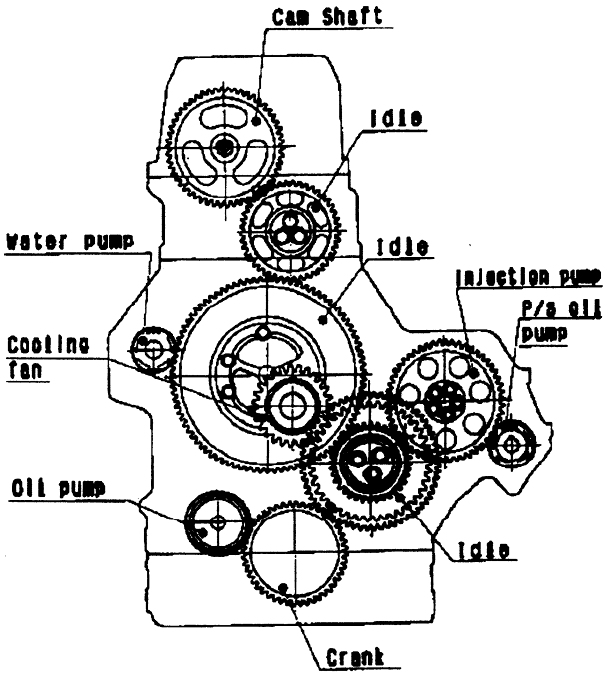


Figure 18: Timing gear train

VALVE MECHANISM

A see-saw type rocker arm system is adopted to reduce the total engine height as well as the roller type for lower friction and reliability increases (Fig 19). An integral rocker shaft which is supported at 7 points is adopted, and mounted to die-cast aluminium cam bearings.

The camshaft is made of induction-hardened carbon steel. In order to actuate two valves at the same time, a valve bridge made of induction-hardened ductile iron is provided. The sliding-cap is adapted to the driving part of the rocker arm to improve the abrasion resistance as well, which is mounted to bridge by means of a holed-plate which prevent its rolling in high speed ranges (Fig 20).

The rocker roller pin is made of special bronze material and larger in diameter with improved surface roughness and suitable oil clearance. A special treatment by shot blasting is applied to the roller outer surface to improve peering resistance.

Double spring is equipped for the exhaust valve to realise a strong exhaust braking power of 206 kW.

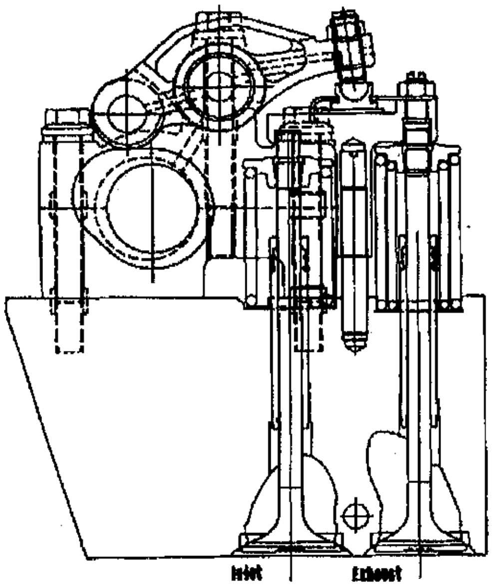


Figure 19 Valve mechanism

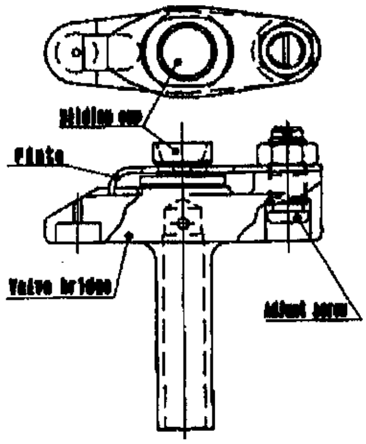


Figure 20 Valve bridge structure

LUBRICATION AND COOLING SYSTEM

A three-gear oil pump is employed to reduce oil pulsation which causes noise under the cold state condition. Also, the oil by-pass valve method is adopted to reduce power loss which controlled the flow rate corresponding to the oil pressure. Fig 21 explains the control mechanism of the oil by-pass method and Fig 22 indicates its effect to driving power. At high speed conditions, the control valve (2) is driven by higher oil pressure of port (A) accordingly oil pressure increases, consequently the port (B) pass through the crank case and certain mount of oil is by-passed. The ball (1) closes the channel by pressure difference between the port (A) and port (B), then oil flow is stopped here. By adopting this method, the driving power can be reduced by 30% at high speed under the same oil pressure condition in engine gallery.

The water pump equipped with a synthetic resin-moulded impeller is a gear-driven type and two thermostats are provided on the suction side. The oil cooler is positioned at the exhaust side of the cylinder block, and the order of cooling is like oil cooler, cylinder block and cylinder head. The cylinder liner surface temperature is maintained at 170°C or less and the surface temperature of cylinder head is kept under 300°C

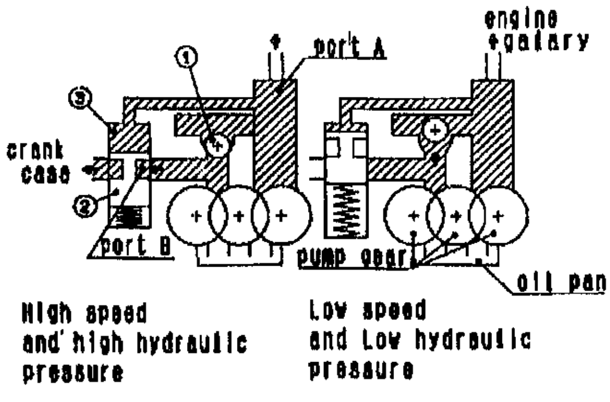


Figure 21: By-pass mechanism of the oil pump

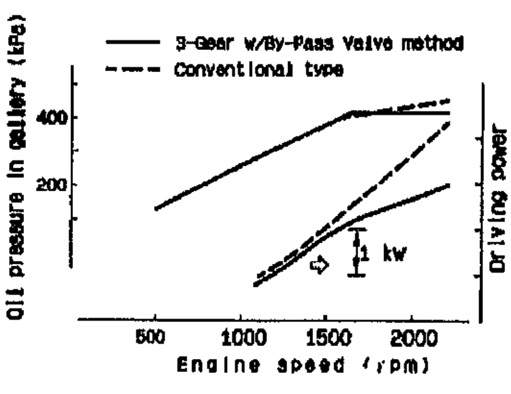


Figure 22: Effect of the oil by-pass method

EXHAUST AND INTAKE SYSTEM

A variable geometry turbocharger system (VGS) controlled by an 8-step air actuator is employed for the highest output power application to improve engine torque performance at low speed. A slide-type exhaust brake valve is installed on the gas inlet side of turbocharger in order to protect the variable nozzle link-mechanism from strong pulsation under the exhaust brake working condition (Fig 23).

The structure of an 8-step air actuator is shown in Fig 24. Three movable pistons are mounted inside the cylinder with its strobe of 3, 6 and 12 mm respectively, so that 3mm stroke intervals can be achieved in-between 21mm full stroke. For low and medium out-put power applications, the conventional turbocharger and a rotary valve type exhaust brake valve are installed. Fig 25 shows the 8-step VGS turbocharger effect to boost pressure performance.

The combined intake charging system (CCS) is newly adopted for all applications. This system contributes to improvement of black smoke level at lower engine speed and fuel consumption.

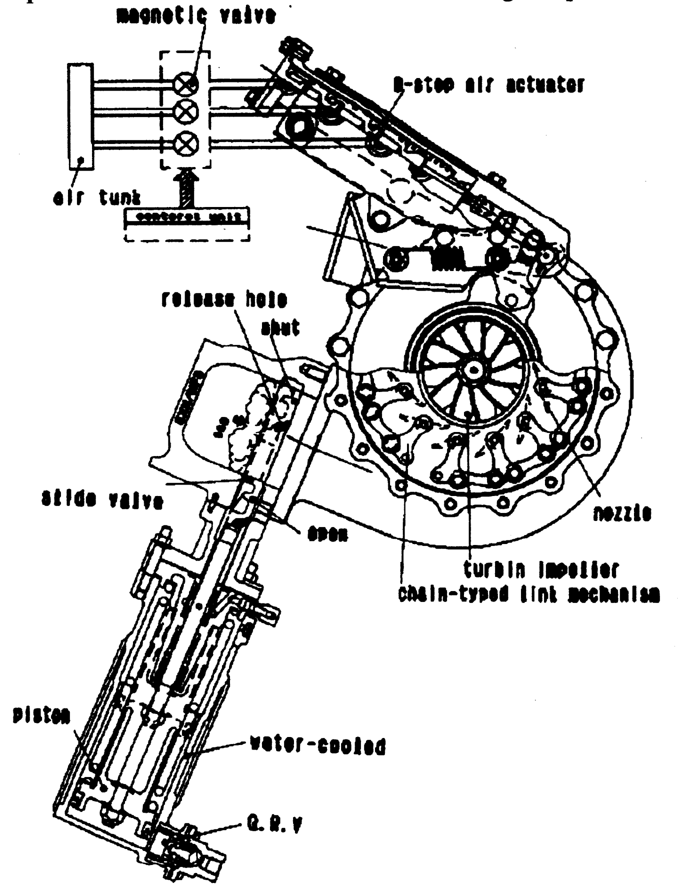


Figure 23: Slide type exhaust brake and V.G. turbocharger

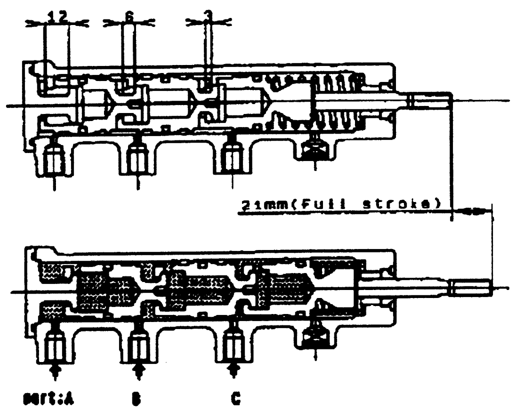


Figure 23: 8-step air actuator

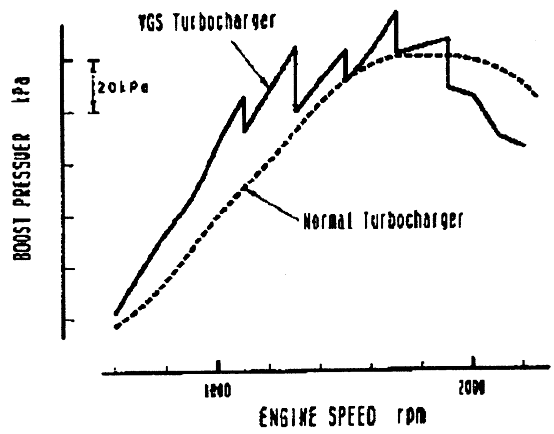


Figure 25: Boost pressure characteristics with VGS

FUEL AND ELECTRONIC CONTROL SYSTEM

A S3S in-line Bosch type pump is adopted for low output power application and HD-TICS (*T*iming and *I*njection *R*ate *C*ontrol *S*ystem) is for medium and high output power applications.(2) Table 3 shows the specifications of fuel system for Japan market. The eight-holes nozzle is selected for all applications to meet the exhaust emission regulation and to improve black smoke level and fuel consumption, in spite of different nozzle hole diameter between each injection pump type.

The following systems are being controlled electronically;

1. Electronic control governor
2. Variable injection timing of fuel injection pump (HD-TICS)
3. Variable swirl system
4. Variable geometry turbocharger system (VGS)
5. Variable combined intake charging system (CCS)

	6WA1TCN	6WA1TCC	6WA1TCS
Output power (kW)	243	265	287
Inj. Pump	Bosch S3S	HD-TICS	HD-TICS
Planger dia.	12	12	12
Planger lift	12	14	14
Governor type	Elec.		
Timer	Elec.	Mech.	Mech.
Nozzle	0.28 x 8	0.24 x 8	0.24 x 8

Table 3: Specifications of fuel injection system for Japan model

For various electronically controlled systems, various sensors are used for each applications. Table 4 shows inter-relation between sensors and systems. The fuel temperature sensor is used for the applications with HD-TICS to correct fuel injection rate, because the fuel temperature in HD-TICS rises due to higher injection pressure than the conventional injection pump.

	Fuel System (Timir-governor TICS)	VG turbo System	Variable Swirl System	Combined Charge System	Shift Advisor System	VG turbo Backup System
(1) Accel Sensor	O		O	O		
(2) Load Sensor	O	O	O	O	O	
(3) Engine Speed & TDC Sensor	O	O	O	O	O	
(4) Timing Sensor	O					
(5) Water Temp Sensor	O		O			
(6) Vehicle Speed Sensor					O	
(7) Boost Sensor	O					O
(8) Ambient Air Temp Sensor						O
(9) Clutch Switch		O	O	O		
(10) Fuel Temp Sensor	●					

Table 4: Inter-relation between sensors and each system

PERFORMANCE

By adoption of high pressure injection pump, combined intake charging system and variable geometry turbocharger, substantially high torque can be obtained even in low engine speed conditions. Especially the vehicle starting ability is improved by adoption of HD-TICS and CCS. Fig 26 shows the starting acceleration torque of the different fuel injection equipment under low engine speed conditions.

New combustion chamber combined with multi-holes and small diameter nozzle contributes to improvement of the exhaust emission, especially black smoke and PM. Fig 27 shows the effects of new combustion chamber to black smoke.

By adapting of the tighten OHC mechanism and strong exhaust valve spring, high exhaust braking force is achieved which contributes to better drivability. Fig 28 shows the comparison of exhaust braking force with other turbocharged engines with an intercooler. Thanks to the increased braking force, frequency of using foot brake is remarkably reduced (Fig 29) and service life of brake-lining can be prolonged.

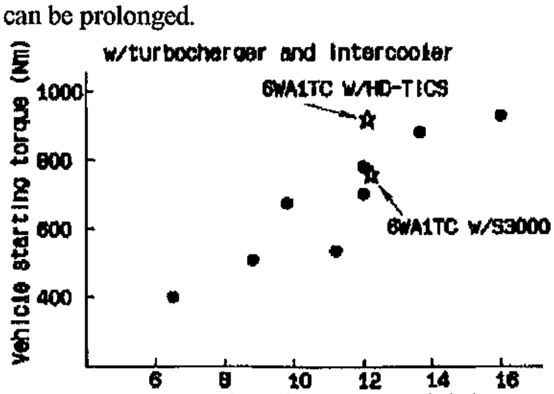


Figure 26: Vehicle starting ability
(at 600 rpm with 0-mmHg of boost pressure)

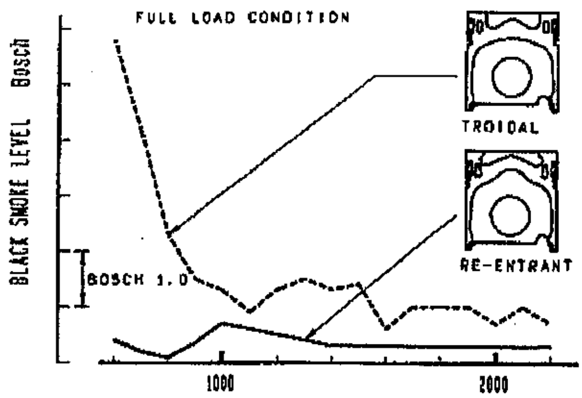


Figure 27: Effect of new combustion chamber to black smoke.

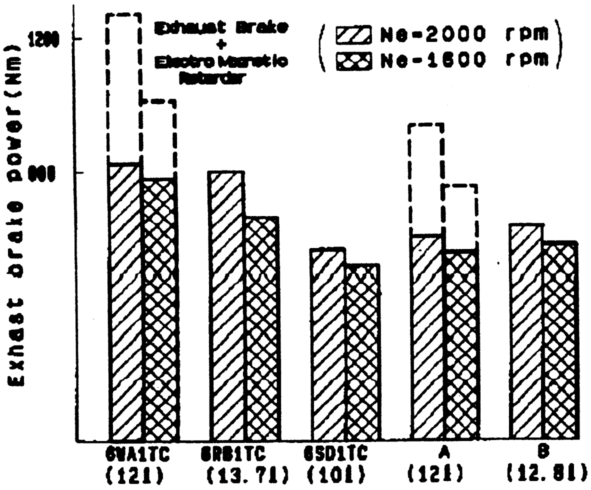


Figure 28: Comparison of exhaust braking force of various turbocharged engines being manufactured in Japan

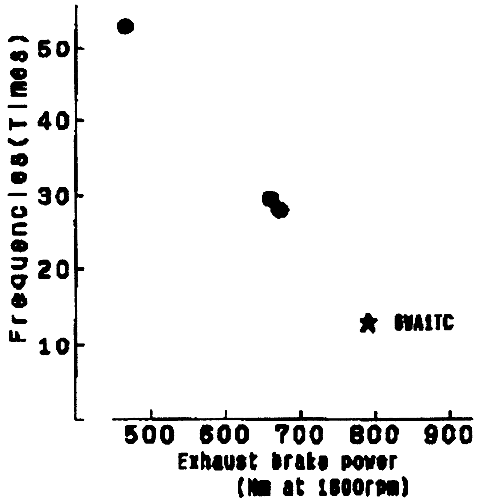


Figure 29: Frequencies of using foot brake

DURABILITY AND RELIABILITY

In order to accomplish the target useful life of 1,300,000 Km in Japan market, on-board measurements, such as engine speed, frequencies of applying load, variation in speed, variation in load, frequencies of exhaust braking action and so on, were carried out before useful life evaluation on the bench and rig stand. In addition, over load tests and abuse tests using actual engines and on-board evaluation tests using the actual vehicles were carried out.

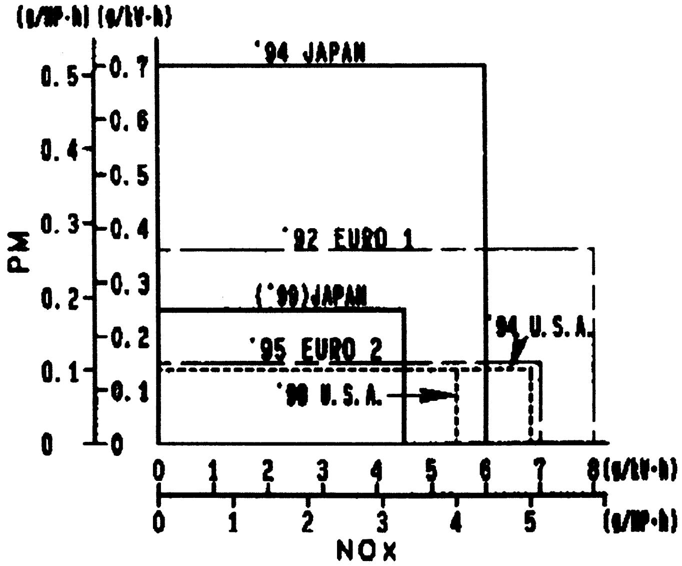


Figure 30: Worldwide exhaust emission regulations

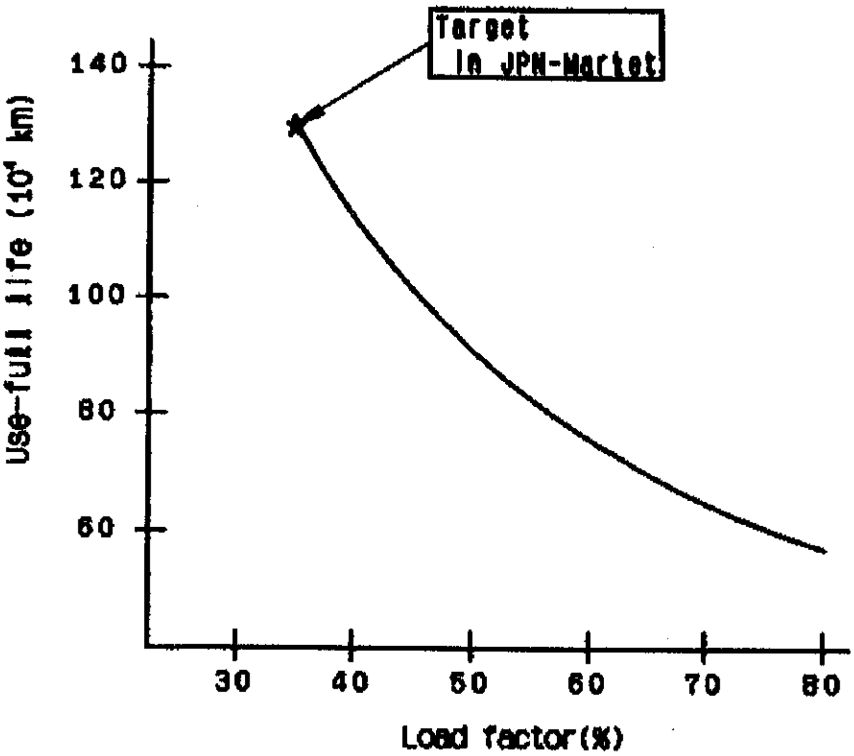
APPLICATION TO WORLDWIDE MARKET

The export applications of 6WA1TC were started production in January 1996 primary for Hong Kong and Taiwan markets. The applications for other markets will be available within a year. In Asia markets, the stricter exhaust emission regulations, such as '90US and Euro-l Standard, were implemented and possible implementation of next stage emission regulation will be anticipated in near future as well (Fig 30). For example, specifications of export applications are shown below;

Market	A	B
Output power (kW)	265	279
Torque (N.m)	1421	1597
Inj pump type	Bosch S3S	Bosch S7S
Governor	Mech.	Mech.
Turbocharger	STD Turbo	STD Turbo
CCS	with (not variable)	
Regulation	'90US	—

Table 5: Specifications of export applications.

Vehicle running condition in Asia and another market is more severe than that of cargo vehicles in Japan market concerning to gross combination weight. Fig 31 shows the relation between useful life and load factor, which means ratio between fuel amount used in running and fuel amount at lated power.



CONCLUSION

This paper introduces the design concept and design features of each component of the newly developed 12-litre diesel engine, 6WA1TC. The most important tasks assigned to engineers were: to render the new engine with a potential for further upgrade in order to meet future requirements for higher out-put power and to cope with future provisions in the exhaust emission and noise regulations: to accomplish the target of longer useful life. All of the original targets had been satisfied and accomplished and the new engines went into production in July 1992, and improved models meeting the new regulations went into production in November 1994. In future this engine should be upgraded and installed on not only cargo series but also tractor series with more power.

REFERENCES

(1) M. Kita, M. Wakabayashi and K. Honjyo, *New 12L 6WA1TC Turbocharged Diesel Engine*, SAE paper 930718

(2) H. Ishiwata and H. Okubo, *A new series of timing and injection rate control systems-Ad-Ties and P-Tics*, SAE paper 880491

(3) T. Hashimoto, K. Okada and T. Oikawa, *ISUZU new 9.8L diesel engine with variable geometry turbocharger*, SAE paper 860460

(4) K. Takenaka, Y. Nitta and K. Kubota, *ISUZU new H-series diesel engines*, SAE paper 910628

(5) T. Kubomiya, T. Kuwahara and K.Araki, *Permanent Magnetic Retarder in Commercial Vehicles*, SAE paper 922455