技術論文

Development of CO2 emission reduction technology for sport motorcycles

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要旨

気候変動に対する世界的な関心が高まる中、二輪車（MC）においてもカーボンニュートラル（CN）を実現することが課題であり、CN を実現するためには様々なアプローチが必要である。内燃機関（ICE）を用いたパワートレインでは、e-fuel やバイオ燃料などの CN 燃料を採用することで CN を達成できるが、コストや供給面を考慮すると、ICE の CO2削減技術の開発が重要である。MC は四輪自動車と比較して、高出力、軽量、コンパクト、長距離走行が要求されるが、採用できる CO2削減技術は動力性能と CO2削減のトレードオフになりがちであり、両方の要求を高いレベルで実現することが課題である。そこで、特に高い動力性能が要求されるミドルクラスのスポーツ MC に着目し、CO2削減技術を開発することにした。技術開発目標としては、スポーツ MC に求められる動力性能を維持しつつ、CO2排出量を worldwide-harmonized motorcycle test cycle (WMTC) クラス3-2で65g/km とした。目標達成に必要な技術の組み合わせをシミュレーションし、技術実証のコンセプトを選定した。その結果、電動アシストターボ（E-Turbo）付ダウンサイジングコンセプトを選定し、試作エンジンを用いて CO2排出量と動力性能をダイナモで検証した。本稿では、これらのシミュレーションに基づく技術選定の考え方と、試作エンジンを用いた実証結果について述べる。

Abstract

With growing global concern about climate change, the challenge is to achieve carbon neutrality (CN) in motorcycles (MCs) as well, and various approaches are needed to achieve CN. For powertrains using internal combustion engines (ICEs), CN can be achieved by adopting CN fuels such as e-fuel and biofuel, but considering cost and supply, it is important to develop CO2 reduction technologies for ICEs. Compared with 4-wheel vehicles, MCs are required to be powerful, lightweight, compact and capable of travelling long distances, the CO2 reduction technologies that can be adopted tend to be a trade-off between dynamic performance and CO2 reduction, and a challenge is to achieve a high level of both requirements. We decided to focus on middle-class sports MCs, which require particularly high dynamic performance, and to develop CO2 reduction technologies. As a technology development target, CO2 emissions were set at 65 g/km in the worldwide-harmonized motorcycle test cycle (WMTC) class 3-2, while maintaining the dynamic performance required for sports MCs. The combination of technologies required to achieve the target was simulated and a concept was selected for technology demonstration. As a result, the downsizing concept with electrically assisted turbocharger (E-Turbo) was selected and CO2 emissions and dynamic performance were verified on a dynamometer using a prototype engine. This paper describes our approach to selecting the technologies based on these simulations and the demonstration results using the prototype engine.

1 INTRODUCTION

In recent years, global concern about climate change has increased and efforts are being made in various fields to reduce CO2 emissions to achieve CN. In the transport equipment industry, CN compliance is also a major challenge. For vehicles using ICEs, CN can be achieved by adopting CN fuels such as e-fuel and biofuel, but there are also issues in terms of cost and supply[1][2], so not only replacing fuel types but also improving fuel consumption is important as a value provided to customers. To maintain and improve the value provided to customers while meeting social demands for reducing environmental impact, it is necessary to develop ICEs technologies to reduce CO2 emissions. Various CO2 reduction technologies are also being researched and developed for MCs, with the use of electric components[3][4] and the adoption of turbochargers to reduce CO2 emissions[5]. On the other hand, sports MCs with medium to large displacement engines tend to require dynamic performance such as high power, light weight and compactness. Figure 1 shows the relationship between power weight ratio (PWR) and CO2 emissions for MCs, four-wheel vehicles with ICEs and four-wheel hybrid electric vehicles (HEVs), showing that MCs are a product group with a smaller PWR than four-wheel vehicles. However, CO2 reduction technologies tend to require various additional devices for existing conventional MCs, which do not directly lead to improved power performance or impair this. We have recognized that the compatibility between low CO2 emissions and competitive dynamic performance is a major issue in sports MCs and have conducted technical verification of CO2 reduction technologies suitable for sports MCs, which can achieve a top-class CO2 emission of 65 g/km for four-wheel vehicles.

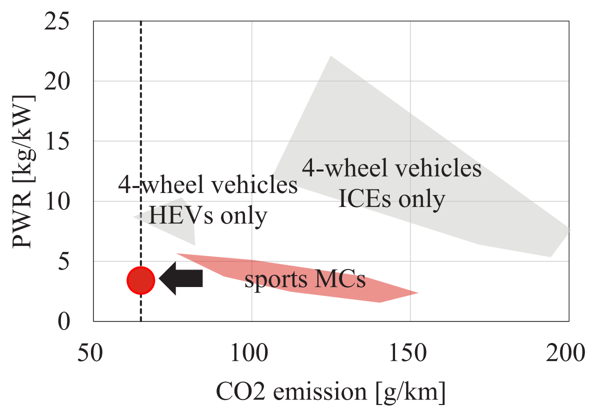

Fig. 1 PWR as a function of CO2 emission

2 CONCEPT RESEARCH

To narrow down the technologies required to achieve a CO2 emission to 65 g/km, a driving simulation was used to calculate the CO2 emission level. The driving simulation used was a one-dimensional (1D) simulation by Gamma Technologies' GT-SUITE. An overview of the calculation method is shown in figure 2. The main input parameters are driving pattern, gear position schedule, gear ratio, vehicle weight, displacement, brake specific fuel consumption (BSFC) map, friction mean effective pressure (FMEP) map, cooling loss map, FMEP increase factor at cold-start, and fuel injection increase factor at cold-start. The engine operating points are calculated from the driving pattern and gear position schedule to obtain fuel consumption, which is converted into CO2 emissions during WMTC mode driving. The oil and water temperature are calculated from the cooling loss map to determine the amount of heat received and is reflected in the increase in oil and water temperature. The FMEP increase factor at cold-start and fuel injection increase factor at cold-start are used to calculate the fuel consumption deterioration at low engine temperatures.

For the input parameters, a virtual BSFC map created with reference to data from previously developed engines and high thermal efficiency engines for four-wheel vehicles[6][7][8] was used. The specification table for each calculation model is shown in table 1 and the BSFC maps created are shown in figure 3.

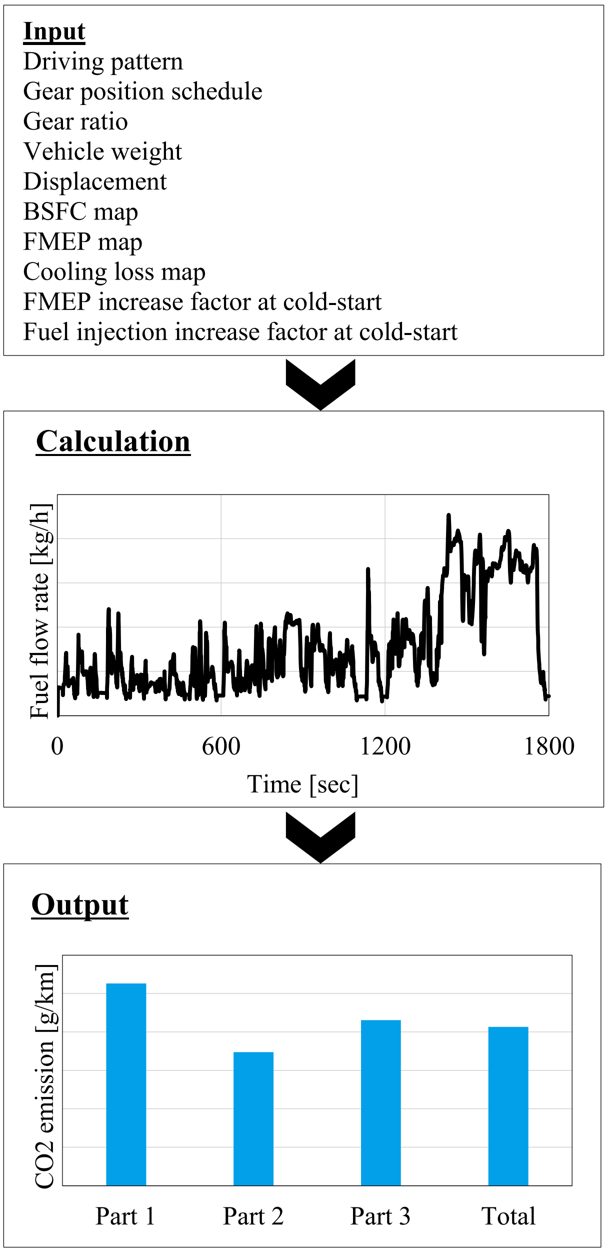

Fig. 2 Outline of calculation logic

Table 1 Specifications of calculated engine model

	Base engine	Engine A	Engine B	Engine C
Displacement	689 cc	689 cc	412 cc	689 cc
CR	11.5	13.6	13.6	13.6
High tumble port		✔	✔	✔
VVT		✔	✔	✔
Cooled EGR		✔	✔	✔
Lean burn		✔		
DI-Turbo			✔	
Cylinder deactivation				✔

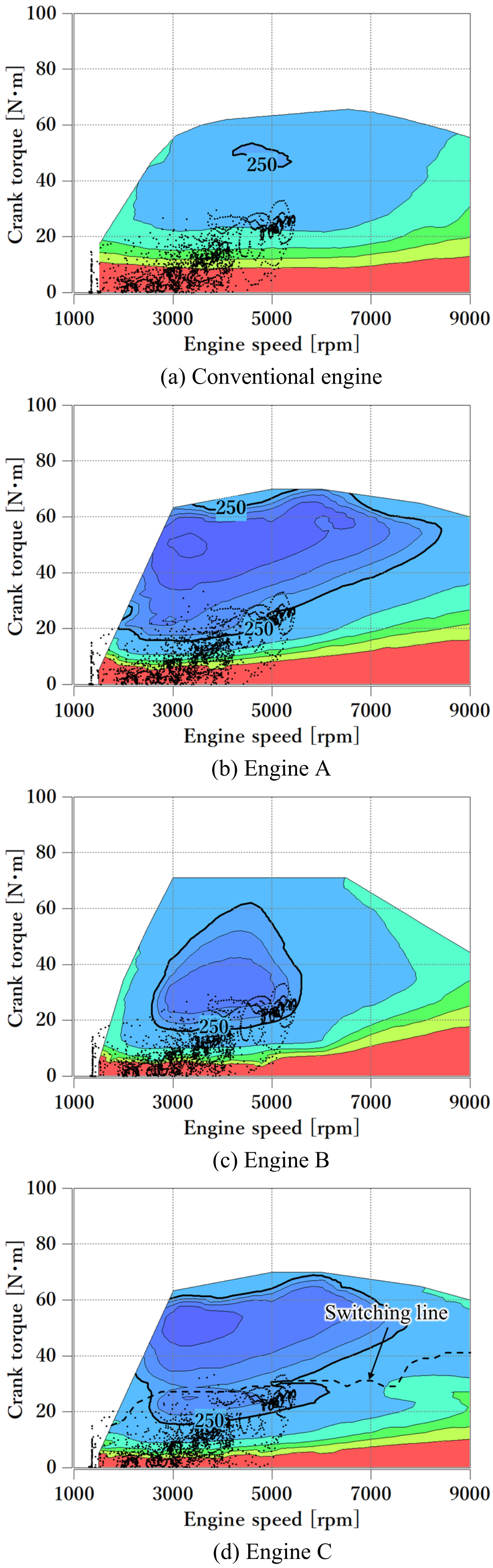

Fig. 3 BSFC map of calculation model

The Base engine is a middle class general sports MC naturally aspirated (NA) engine; the BSFC best point indicates 250 g/kWh around an engine speed 5000 rpm and a crank torque 50 N•m. The engine operating points during WMTC mode driving uses engine speeds up to 5500 rpm and crank torque up to 33 N•m. The engine operating points are same for Engine A, Engine B and Engine C as they have the same gear ratios.

Engine A is a NA engine aiming for high thermal efficiency as combustion enhancement technologies such as high compression ratio (CR), high tumble port, variable valve timing (VVT), cooled exhaust gas recirculation (EGR), and lean burn. The BSFC best point is around an engine speed 3500 rpm and a crank torque 50 N•m, which is on the lower engine speed than the base engine. The main concept is to increase the required throttle opening at low loads and reduce pumping losses through lean or dilute combustion, and the engine is characterized by a wide low BSFC range centered on the BSFC best point.

Engine B is a downsized turbocharged engine with displacement downsized to 412 cc, high CR, high tumble port, VVT, cooled EGR and turbocharger. The fuel injection system adopted direct injection (DI) to counter knock caused by boosting and to suppress fuel blow-off during intake valve and exhaust valve overlap. The BSFC below 10 N•m is improved compared to Engine A. The main concept is to reduce pumping losses like Engine A. Turbocharger adopted in engine B is able to supply a larger mass of air than the same displacement NA engines by compressed air and that enables output targets to be achieved with a relatively small displacement compared to NA engines. In the low-load range, where a large driving force is not required, the required throttle opening can be increased, and the pumping loss can be reduced compared to a large-displacement NA engine.

Engine C combines a high CR, high tumble port, VVT, cooled EGR and cylinder deactivation with the base engine and operates with a displacement of 344.5 cc during cylinder deactivation. The dashed line indicates the cylinder deactivation on and off boundary. In the analysis, cylinder deactivation is mechanically switched on and off with reference to the boundary line, irrespective of the operating conditions. The cylinder deactivation used in Engine C is a mechanism that operates at a cylinder number with high engine efficiency according to the required driving force. In general, cylinder deactivation is used in the low-load range to reduce pumping losses by increasing the required throttle opening by operating with a smaller displacement than the actual total displacement.

The CO2 emissions in the calculation results are shown in figure 4. According to the calculation results, engine A showed CO2 emissions of 78.8 g/km, falling short of the target value of 65 g/km. Engine B achieved the target value of CO2 emissions of 64.3 g/km. Engine C showed CO2 emissions of 66.1 g/km, which was short of the target value, but confirmed its potential to achieve the target value.

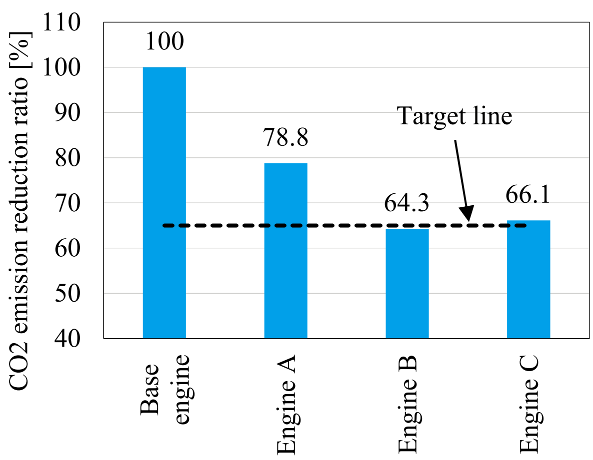

Fig. 4 Calculated CO2 emission

All concepts aim to improve BSFC overall, and reduce pumping losses especially at low loads, but the analysis results show that mechanisms such as Engine B and Engine C, which run at low displacement at low loads, are effective in significantly reducing CO2 emissions. These are concepts that can be called displacement-on-demand, and it was found that a reduction in displacement at low loads is necessary to reduce CO2 emissions to the 65 g/km in sports MCs.

3 ENGINE DESIGN

Based on the results of the concept research and our previous development experience[5], we decided to select downsized turbocharging as one of the technology options to achieve CN for verification. In conducting a detailed study of the downsizing turbocharger, the target values were set in table 2. In addition to the CO_2 emission targets, these target values were set to ensure the dynamic performance of the sports MCs. The CO_2 emissions in the demonstration were only measured in a hot-start due to the workloads required engine calibration. The difference in CO_2 emissions between for a cold-start and a hot-start was calculated and 61 g/km was set as the target value for the demonstration.

Table 2 Target value

CO2 emission in WMTC class3-2		65 g/km (simulation at cold-start) 61 g/km (demonstration at hot- start)
Maximum crank power		62 kW 7000 rpm
Maximum crank torque		93 N･m 3000 to 6400 rpm
Vehicle weight		210 kg PWR = 3.39
Throttle response		Less than 1000 msec at 3rd gear Road load
Acceleration performance Time to travel 200 m	From 80 km/h	6.6 sec at 6th gear Road load
	From 100 km/h	5.8 sec at 6th gear Road load
	From 120 km/h	5.2 sec at 6th gear Road load

3-1. Selection of displacement

To select the displacement, the CO_2 emissions for the combination of displacement and gear ratio were calculated. The BSFC map of the engine for the study is shown in figure 5, which is the previously prototyped NA engine with high CR, high tumble port, VVT and cooled EGR. The fuel supply system for this engine is port fuel injection, but the demonstration engine is adopted DI based on the concept of the engine B in Table 1. The BSFC best point indicates 220 g/kWh around an engine speed 3500 rpm and crank BMEP 9 bar, and a wide low BSFC area centered on the best point is characteristic. Driving simulations were conducted by changing the combination of displacement and gear ratio based on this map.

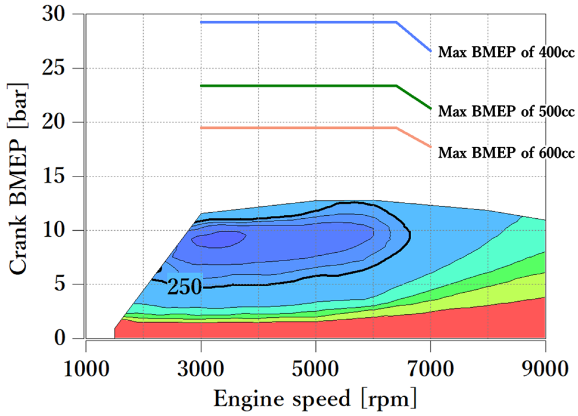

Fig. 5 BSFC map for the study

Figure 6 shows the CO_2 emissions in the analysis results. The analysis results show that there is a CO_2 emission target limit between 470 cc and 560 cc depending on the combination with the overall gear ratio at 6th gear. Further downsizing is expected to further reduce CO_2 emissions, but excessive downsizing increases the required higher brake mean effective pressure (BMEP) and there are concerns about pressure and reduced thermal efficiency due to the reduction of compression ratio as a countermeasure will increase[9].

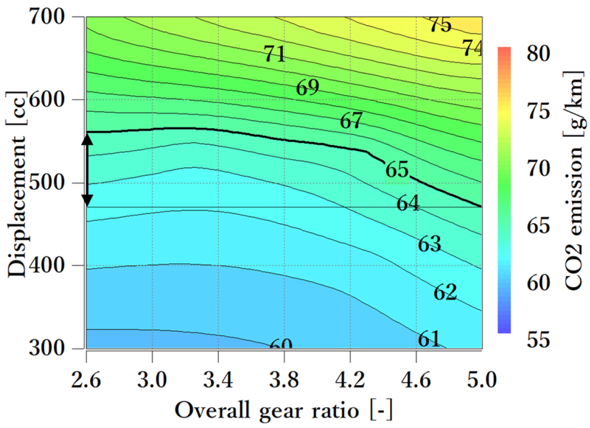

Fig. 6 Calculated CO2 emissions for combinations of displacement and Overall gear ratio at 6th gear

In this study, downsizing is kept to the minimum necessary to minimize the impact on dynamic performance. The bore was set at 70 mm in consideration of fuel adhesion to the cylinder from the DI penetration. The stroke bore ratio should be around 1.0 to ensure turbulence flow in the cylinder, so the stroke was set at 70.2 mm and the total displacement was downsized to 540 cc.

3-2. Turbocharger matching

Selection was conducted by creating an engine performance analysis model using 1D simulation with Gamma Technologies' GT-SUITE. Result of conventional turbocharger analysis showed that it was possible to achieve maximum power output, but there were concerns that low end torque (LET) would not be achieved and acceleration performance would deteriorate due to insufficient exhaust gas volume in the low engine speed range. As a countermeasure, it was decided to adopt an E-Turbo provided by Garrett Motion as shown in figure 7. E-Turbo has a permanent magnet synchronous motor as motor generator (MG) integrated in the center housing between the turbine and compressor stage, and by supplying power to this MG, driving force can be applied to the turbine shaft, which can assist in driving the compressor. As supercharging is possible without waiting for an increase in exhaust energy, an increase in torque and improved throttle response, especially in the low engine speed range, are expected. The E-Turbo also has a harvesting capability, which was not part of these investigations. In line with the adoption of an E-Turbo, the power generation system is based on the 12 V alternating current motor-generator used on existing motorcycles, with modified magnet and winding specifications and a 48 V - 2 kW class MG.

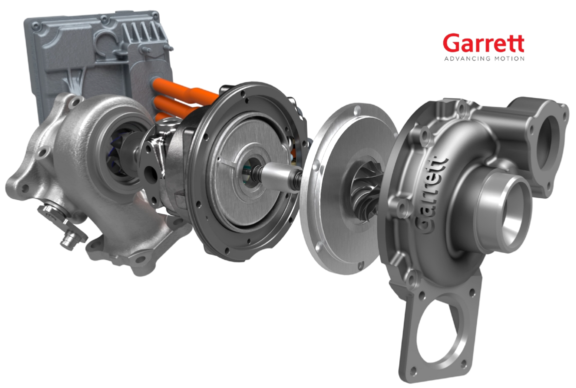

Fig. 7 Structure of the Garrett E-Turbo

The results of the wide-open throttle (WOT) performance analysis are shown in figure 8. It was confirmed that the crank torque at engine speed 3000 rpm did not reach the target value with the conventional turbocharger, but with the addition of 0.88 kW of electric assist (E-assist) by the E-Turbo, the target torque was achieved. The results of the rapid throttle opening response analysis conducted using a same calculation model are shown in figure 9. The calculations were conducted with the engine speed fixed at 3000 rpm and the waste gate actuator (WGA) fully closed. As WGA control is not implemented, the torque continues to increase after the target torque is reached. In this paper, time to torque is defined as the time from the start of acceleration until 90% of the maximum torque is reached, and this is evaluated. By performing E-assist, the torque rises more steeply than without E-assist, and the target time to torque of 1.0 sec can be achieved. As shown in figure 8, w/o E-assist, the target LET is not reached even in the steady state, so the target LET is not reached in the transient analysis.

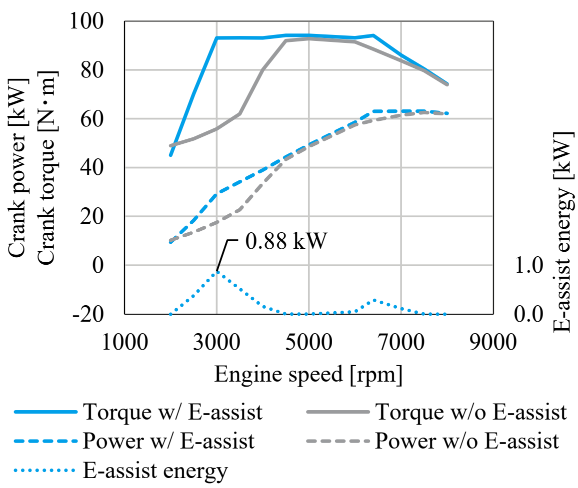

Fig. 8 Calculated crank power and torque as a function of engine speed

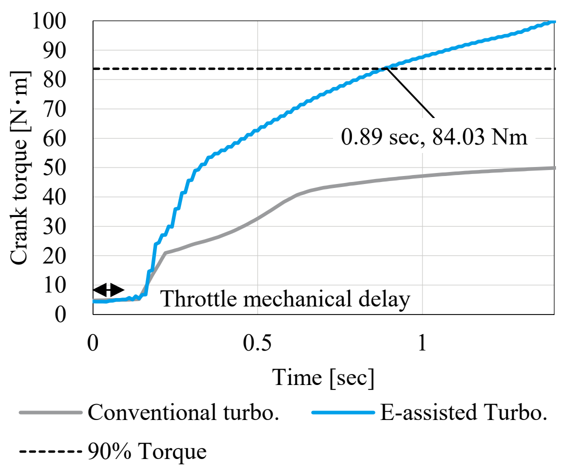

Fig. 9 Calculated crank torque of acceleration

3-3. Selection of gear ratio

The CO2 emission calculation results in figure 6 show that for a displacement of 540 cc, over all gear ratio at 6th gear of 2.993 to 4.240 satisfies the CO2 emission target. From this range, over all gear ratio at 6th gear was selected to 3.972 by balancing the vehicle's acceleration performance. Together with the Automatic transmission (AT) mode, this setting allows the selective use of a low engine speed, high engine load region with good thermal efficiency in WMTC mode. Figure 10 shows the calculated overtaking acceleration performance from the driving simulation. The overtaking acceleration performance is evaluated in terms of the time to travel 200 m after starting acceleration at 80, 100 and 120 km/h. The results of the simulation showed that the target acceleration performance could be achieved with an over-all gear ratio of 3.972 or higher.

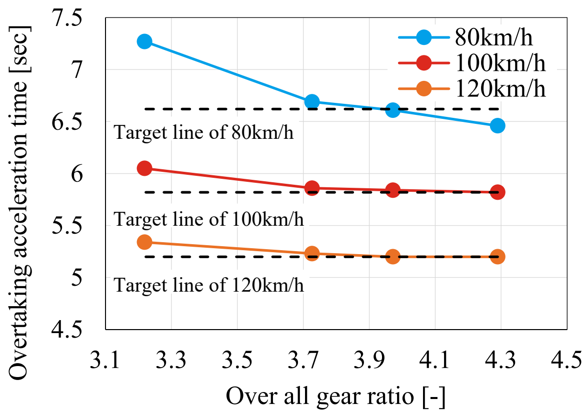

Fig. 10 Calculated overtaking acceleration time

Figure 11 shows the electrical energy used for E-assist during acceleration. The smaller the overall gear ratio, the more electrical energy is required. An overall gear ratio of 3.972 used around 4 kJ of electrical energy for all acceleration conditions. E-assist is assumed to be powered by external power supply such as a battery or capacitor, but it may also be possible to generate the power using an MG mounted on the crankshaft and use it immediately as assist power without charging the power source, which is called "on demand assist". The feasibility of on demand assist will be tested in demonstrations.

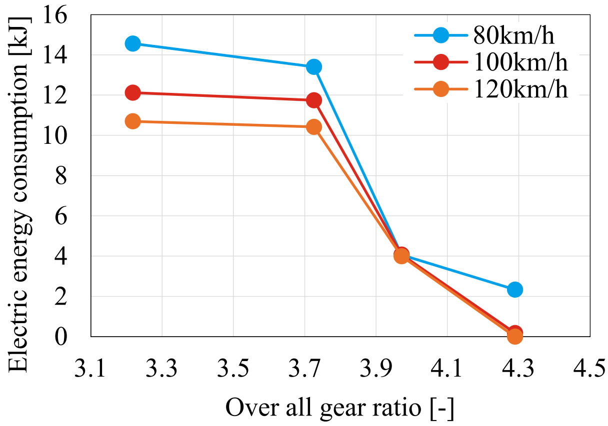

Fig. 11 Calculated electric energy consumption

3-4. Decision specifications

Specifications of the demonstration engine incorporating these design concepts are shown in table 3.

Table 3 Fixed specifications

Displacement [cc]		540
Cylinder number		2
Bore [mm]		70
Stroke [mm]		70.2
Stroke/Bore ratio		1.00
Compression ratio		13.6
Intake system		E-Turbo
Injector type		DI
Maximum power [kw]/[rpm]		62 / 7000
Maximum torque [N•m]/[rpm]		93 / 3000 to 6400
Gear ratio	Primary	1.681
	1st	2.667
	2nd	1.833
	3rd	1.421
	4th	1.200
	5th	1.037
	6th	0.879
	Secondary	2.688
Final gear ratio at 6th gear		3.972
Other items		Tumble port
		VVT
		Cooled EGR
		48V MG
		AT

An example vehicle layout for the demonstration engine is shown in figure 12. As an example, a mass-produced sports MCs frame was used, the demonstration engine is certified to be able to mount on an existing motorcycle body. The vehicle weight was 207.7 kg at the designed value, including the exterior, and the target vehicle weight was achieved.

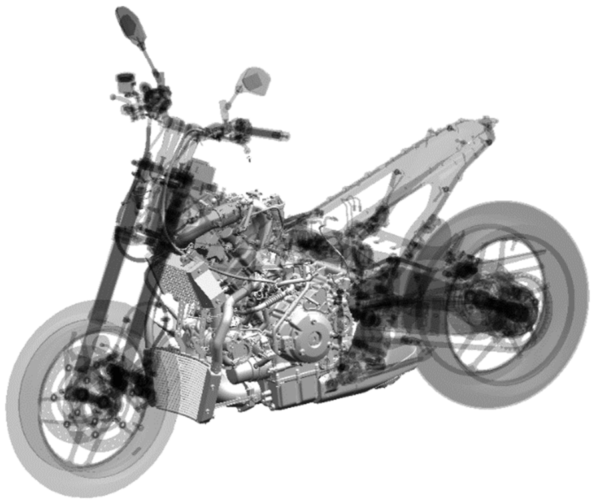

Fig. 12 Example of a vehicle layout

4 DEMONSTRATION

The engine investigated in the section "ENGINE DESIGN" was prototyped and evaluated. The turbocharger was prototyped at Garrett Motion. An overview diagram of the prototype engine and evaluation system is shown in figure 13. The evaluation system used a virtual real simulator (VRS) dyno, which enables the engine to be evaluated as a stand-alone unit simulating vehicle driving. The prototype engine has equivalent to those assumed to be installed in a vehicle except for a cooling system and a power supply system. The cooling system is connected to an external heat exchanger for water temperature control, and the power supply system is connected to an external power supply for testing under various power supply conditions.

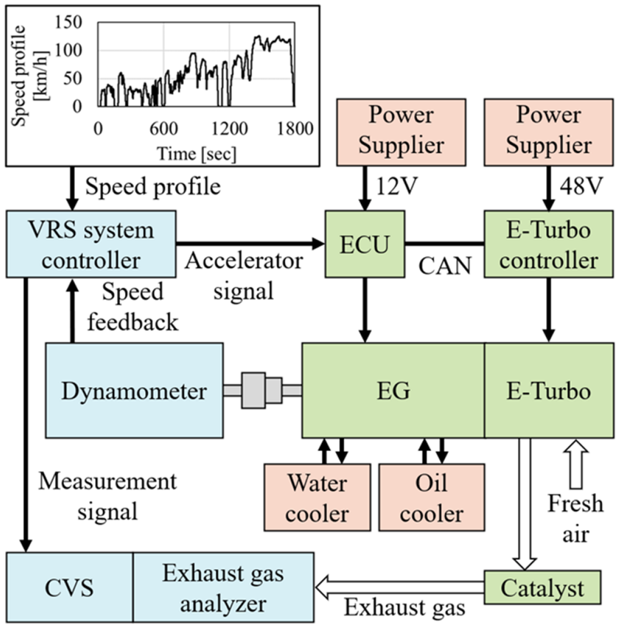

Fig. 13 Outline of dyno system

4-1. Static performance

The BSFC map for the demonstration engine is shown in figure 14. Compared to Figure 3 (a) base engine, the low BSFC region has expanded: the best BSFC point indicated approximately 220 g/kWh around an engine speed 3500 rpm and crank torque 50 N･m, which is equivalent to the best BSFC point in figure 5. Due to the vibration of dyno system, BSFC was not able to be measured below 2000 rpm.

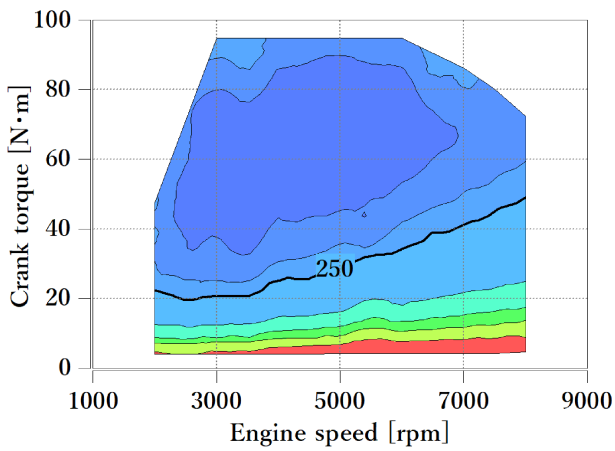

Fig. 14 Measured BSFC map of prototype engine

Figure 15 shows the power performance. The crank power and crank torque achieve the target value, the target torque is not reached in the low engine speed range without E-assist as mentioned in turbo matching.

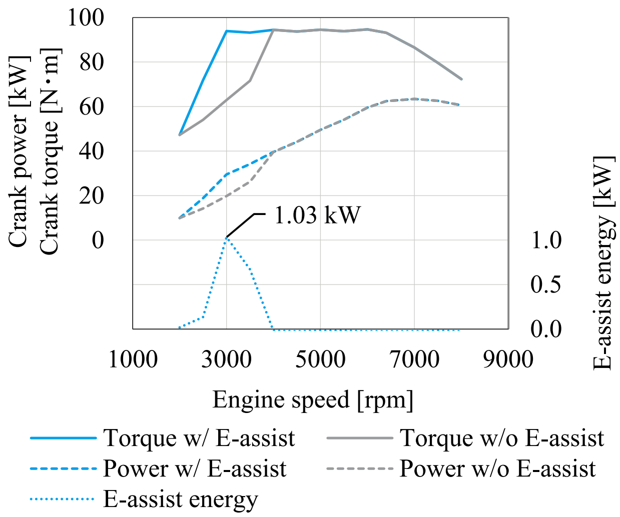

Fig. 15 Measured WOT performance

The knock index and exhaust gas lambda are shown in figure 16. The knock index is within the threshold throughout the all engine speed range. The exhaust gas lambda is measured at the exhaust pipe collective area and is adapted to target stoichiometry for all engine operating points. At engine speed 2500 rpm, the exhaust gas lambda is leaner because the intake and exhaust valve overlap section is enlarged by the VVT and the scavenging effect is used.

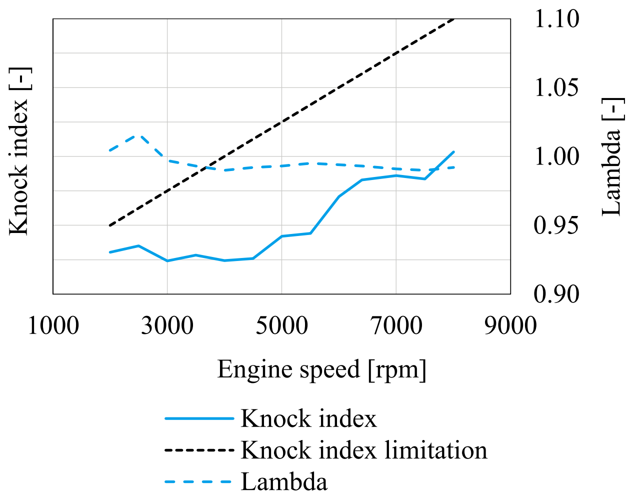

Fig. 16 Measured knock index and exhaust gas lambda

Figure 17 shows the dynamic CR and mass fraction burned 50% (MFB50%). The CR adopted is 13.6, which is a high setting for a supercharged engine, but by adjusting the dynamic CR with VVT, MFB50% could be set to a maximum of about 25 degree crank angle (degCA).

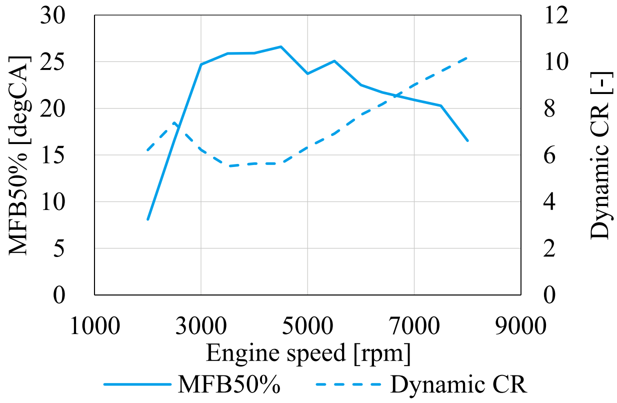

Fig. 17 Measured MFB50% and dynamic CR

4-2. Throttle response

E-assist which uses an external power supply and on demand assist which utilizes the MG were evaluated. On demand assist was assisted by an external power supply due to the number of workloads required for engine calibration, and the generating loss was reproduced by increasing the driving resistance. Figure 18 shows the results of the rapid throttle opening test. Without E-assist, the torque rise was slow and increased with increasing engine speed, with a time to torque of 3.08 sec. External assist, the torque rise was steep above 45 Nm. The time to torque was 0.91 sec, which is close to the analysis results shown in figure 9. In on demand assist, the throttle response improved to 1.12 sec compared with no E-assist, although it did not reach the target value. Figure 19 shows the electric power and energy used by E-assist. External assist gives a maximum power output of more than 2.5 kW and uses 1.81 kJ of energy when the torque is reached at 90% of the maximum torque. On demand assist gives a maximum output of around 1.8 kW and uses 1.38 kJ of energy when the torque is reached at 90% of the maximum torque. The electric power for E-assist of on demand assist is fixed by the MG specification, so a higher capacity MG can increase the electric power for E-assist and shorten the time to torque.

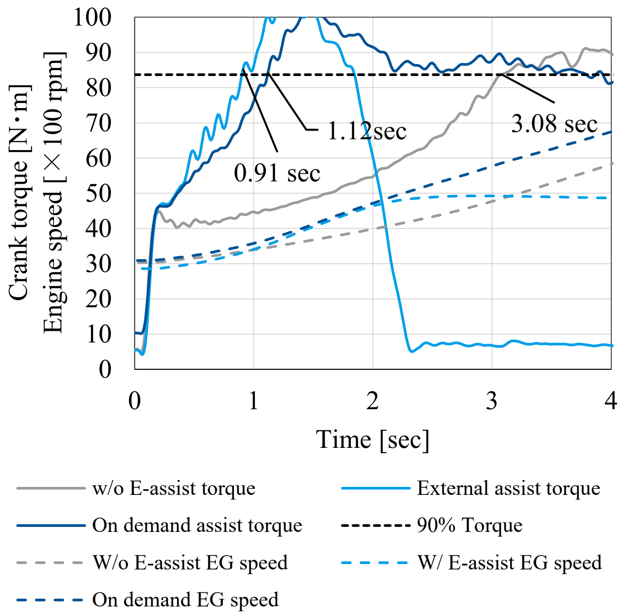

Fig. 18 Measured crank torque of acceleration

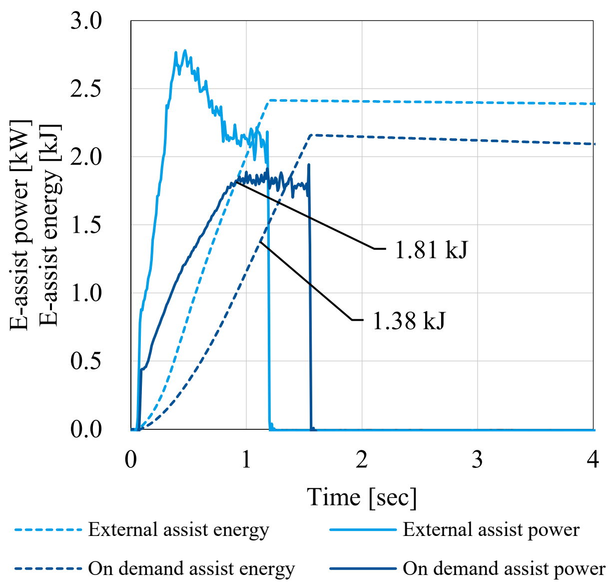

Fig. 19 Measured E-assist power and energy of acceleration

4-3. CO2 emission

The CO2 emissions in WMTC mode are shown in figure 20 and the engine operating points on the BSFC map in figure 21. The CO2 emissions target for demonstration of 61 g/km was achieved. According to the simulation calculations, the CO2 emissions deterioration due to cold-start is 3.8 g/km, which means that the cold-start target of 65 g/km is also expected to be achieved. As in the simulation described in the previous section, the AT is used to actively use the high thermal efficiency area with low engine speed and high engine load, and the shift schedule is programmed to use as high a number of gear steps as possible within the range where engine vibration is tolerated. In this demonstration, regeneration and assist by crank-mounted MG, harvesting capability by E-Turbo and idling stop have not yet been implemented. Further reductions in CO2 emissions can be expected by incorporating these systems.

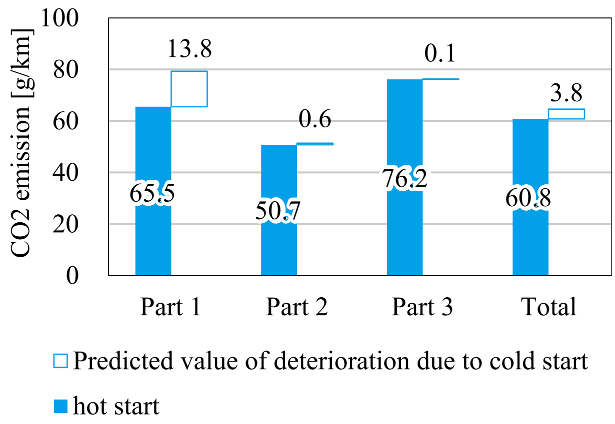

Fig. 20 Measured CO2 emission in WMTC

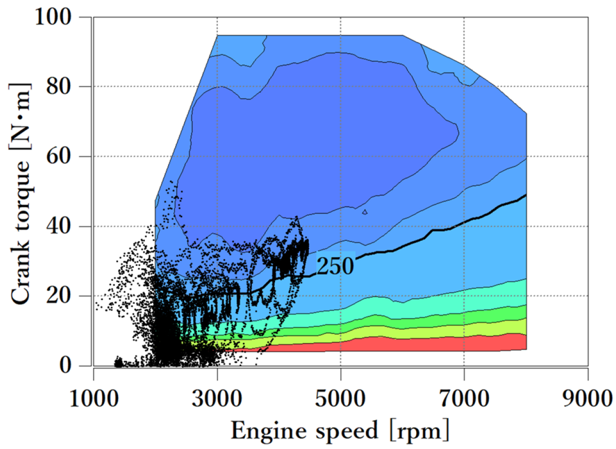

Fig. 21 Measured engine operating points on BSFC map

4-4. Overtaking acceleration

As in the throttle response evaluation, external assist and on demand assist were evaluated. Overtaking acceleration times are shown in figure 22. The time of external assist was achieving the target time, and on demand assist was almost on target; the difference between external assist and on demand assist is greater at lower initial vehicle speeds at the start of acceleration. As a representative example of time continuous data, data at an initial speed of 80 km/h is shown in figure 23. The torque of external assist increases more quickly after 50 N-m than on demand assist. At this time, external assist is assisted by the allowable transient electric power in about 1 sec from the start of acceleration, and then the allowable steady-state electric power is continuously assisted. The crank torque is controlled by opening the WGA, as the setting is such that the assist continues to be applied even after the crank torque reaches the upper limit. On demand assists at a maximum of about 1.8 kW, as the assist power is limited by the MG specifications. After the crank torque is reached to upper limit, the system is set to no E-assist. The electric energy for E-assist used at the end of overtaking acceleration was 13.45 kJ for external assist and 4.79 kJ for on demand assist. Although the overtaking acceleration time can be reduced by continuing to E-assist with external power supply, the fact that the target value can be satisfied almost entirely with on demand assist means that an electric energy around 5 kJ is sufficient to achieve the effect of the E-assist. In addition, because the electric power for E-assist is output from the crankshaft in on demand assist, the electric energy is less than 4.79 kJ when aiming for a time equivalent to the target value with external power supply. Acceleration from 80 km/h is the condition under which the engine speed at the start of acceleration is around LET, and therefore the most benefit from the assist can be obtained. At 100 and 120 km/h, when the engine speed is higher at the start of acceleration, the turbo drive power from the exhaust gases increases, so the effect of the E-assist becomes relatively small.

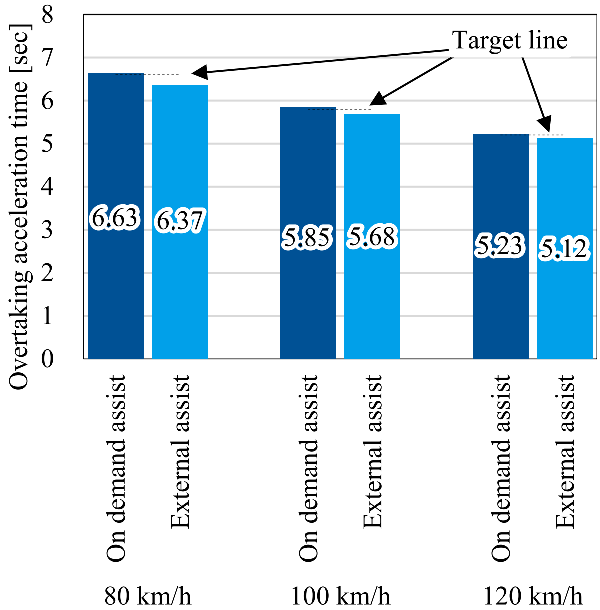

Fig. 22 Measured overtaking acceleration time

5 SUMMARY/CONCLUSIONS

The CO2 emissions of 65 g/km are achieved by downsizing engine with turbocharger or cylinder deactivation, making it clear that displacement-on-demand concepts are effective to 65g/km level of CO2 emissions reduction.

The concept of a highly thermally efficient downsized turbocharged engine combined with E-Turbo has proven that the engine can meet CO2 emission and dynamic performance targets and can be installed in MCs.

An evaluation of the prototyping engine has demonstrated that the selected concept can achieve the target values for CO2 emissions and dynamic performance.

The downsized turbocharged engine with E-Turbo enabled an elevated level of compatibility in terms of both CO2 emissions and dynamic performance and demonstrated one of the directions in the evolution of MCs towards achieving carbon neutrality.

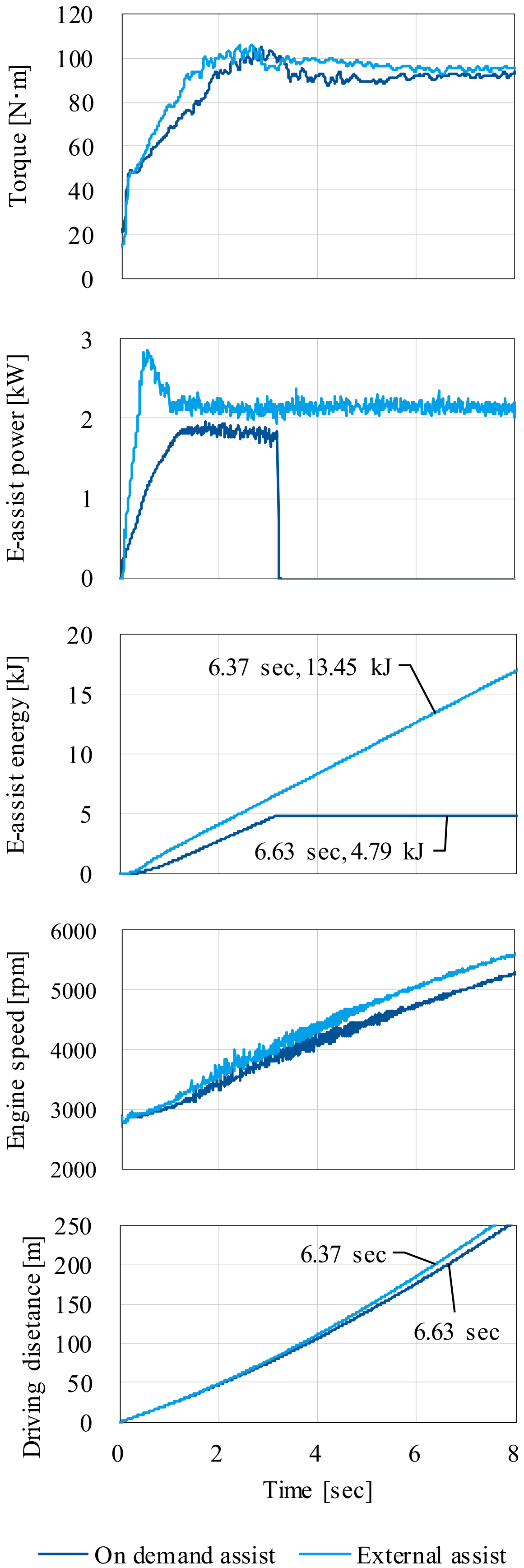

Fig. 23 Measured driving data of overtaking acceleration from 80 km/h

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DEFINITIONS/ABBREVIATIONS

CN	Carbon Neutrality
ICEs	Internal Combustion Engines
WMTC	Worldwide-harmonized Motorcycle Test Cycle
E-Turbo	Electrically assisted Turbocharger
PWR	Power Weight Ratio
HEVs	Hybrid Electric Vehicles
1D	1 Dimensional
BSFC	Brake Specific Fuel Consumption
FMEP	Friction Mean Effective Pressure
NA	Naturally Aspirated
CR	Compression Ratio
VVT	Variable Valve Timing
EGR	Exhaust Gas Recirculation
DI	Direct Injection
BMEP	Brake Mean Effective Pressure
LET	Low End Torque
MG	Motor Generator
WOT	Wide-Open Throttle
E-assist	Electric assist
WGA	Waste Gate Actuator
AT	Automatic Transmission
VRS	Virtual Real simulator
MFB50	Mass Fraction Burned 50%
degCA	degree Crank Angle

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