

Control Parameter Optimization for Reduction of 2nd Order Vibration in Hybrid Drive Mode of SPORT HYBRID i-MMD

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ABSTRACT

During development of the 2014 model-year Accord Hybrid, for which the SPORT HYBRID i-MMD new model hybrid system was adopted, 2nd-order vibration transmitted to the steering became an issue. This occurred because the vehicle has higher engine net torque than a gasoline engine vehicle, so that there is larger crankshaft revolution 2nd-order fluctuation, which is the source of the vibration. There is concern that using only reduction of engine net torque as the means to lessen crankshaft revolution 2nd-order fluctuation could cause an increase in the brake-specific fuel consumption. Therefore the combustion control parameters, such as ignition timing and valve operation timing, were changed as a way of optimizing crankshaft revolution 2nd-order fluctuation, engine net torque, and brake-specific fuel consumption. These measures achieved the steering vibration target while also determining the optimal point where the amount of engine net torque reduction could be lessened by 5 Nm and 1% of the increase in brake-specific fuel consumption could be recovered.

1. Introduction

As hybrid electric vehicles (HEV) are now being sold by automobile manufacturers, it has become important to enhance the appeal of HEV qualities other than fuel economy performance. In this context, noise and vibration (NV) performance has become an issue because the cabin noise level in a moving HEV differs greatly between when the engine is running and when it is not, and this difference is easy to recognize⁽¹⁾.

In 2012, Honda announced the SPORT HYBRID Intelligent Multi-Mode Drive (SPORT HYBRID i-MMD) as a new HEV system comprising a 2.0-L gasoline engine and two motors in combination^{(2), (3)}. Integrated Motor Assist (IMA)⁽⁴⁾, which is a conventional one motor-type HEV system, is a parallel hybrid mechanism, but the SPORT HYBRID i-MMD is a series hybrid system, and has achieved the low fuel consumption of 30.0 km/L in JC08 mode.

In hybrid drive mode, which involves series hybrid operation, the engine power is used only to generate power from a generator. The actual driving of the vehicle is performed only by the motors. Since the engine power is not directly linked to the drivetrain when this happens, the engine operating conditions are not dependent on

the vehicle acceleration or the vehicle speed demanded. Consequently, engine control can set operating conditions that minimize brake-specific fuel consumption (BSFC).

In gasoline engine vehicles, the engine and the drivetrain are directly linked. When driving under moderate acceleration, the required engine net torque is 30 Nm lower than when operating at minimum BSFC, so these vehicles cannot operate in the minimum BSFC region. In the hybrid drive mode of the SPORT HYBRID i-MMD, however, the engine can be operated at high engine net torque in which BSFC is always at a minimum. As a result, the increase in crankshaft revolution 2nd-order fluctuation causes increased engine 2nd-order vibration that is transmitted to the vehicle body. It is the resulting steering wheel 2nd-order vibration that has become an issue.

When using reduction of engine net torque alone to lessen crankshaft revolution 2nd-order fluctuation, increased BSFC becomes a concern. A method was therefore employed to limit engine net torque by means of control that changes the ignition timing, valve operation timing, and other such combustion control parameters so as to optimize the crankshaft revolution 2nd-order fluctuation, engine net torque, and BSFC. This paper reports on the content of this series of control events.

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2. SPORT HYBRID i-MMD Characteristics

Figure 1 shows the SPORT HYBRID i-MMD system configuration⁽⁵⁾. The engine is the source of power, and the generator uses that power to produce electricity. The generated electricity is stored in batteries and used as the power to drive the motors. The solid lines in the diagram show the route of mechanical transmission of power, and the dotted lines show the route by which electricity is transmitted.

The SPORT HYBRID i-MMD switches the driving mode according to the vehicle speed and the battery level at the time. Figure 2 shows the route by which power

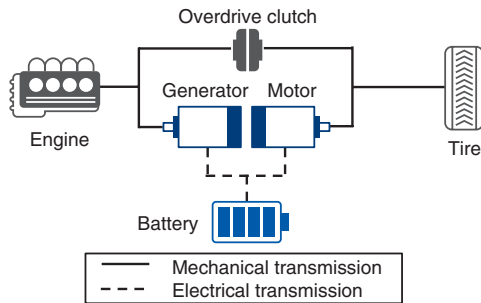


Fig. 1 SPORT HYBRID i-MMD system configuration

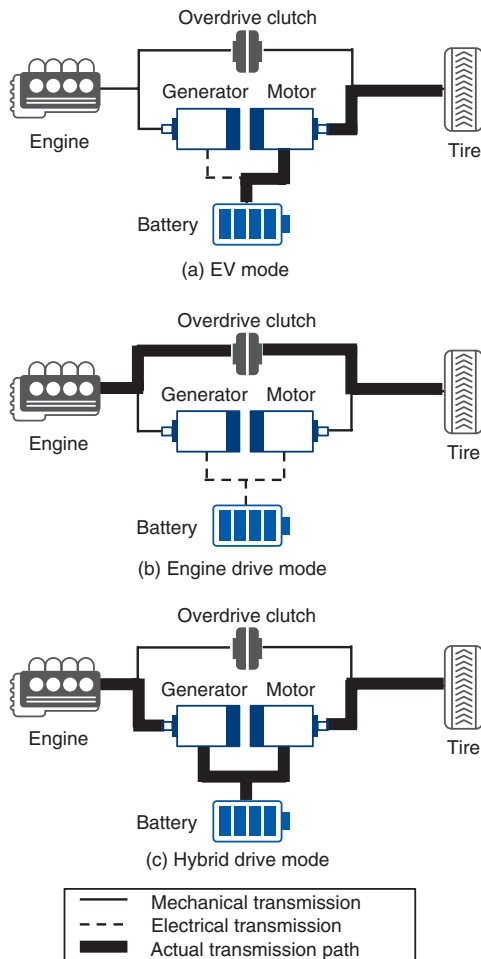


Fig. 2 Three drive modes in SPORT HYBRID i-MMD

is transmitted in each mode. The heavy solid lines in the figure indicate the routes by which power and electricity are actually transmitted in each mode.

In the electric vehicle (EV) mode shown in Fig. 2(a), the engine is stopped and the motors drive the tires using electricity stored in a battery.

In the engine drive mode shown in Fig. 2(b), the overdrive clutch is engaged and engine power goes directly to the tires via mechanical transmission. This is used when cruising at a vehicle speed of 70 km/h or more, where the engine is more efficient than the motors.

In the hybrid drive mode shown in Fig. 2(c), engine power does not go directly to the tires via mechanical transmission, but instead is used only to generate electric power with the generator. Power transmission to the tires occurs entirely from the motors. This is used when cruising at a vehicle speed of 70 km/h or less, where the motors are more efficient than the engine. This hybrid drive mode is not found in the conventional IMA, which is a parallel hybrid mechanism, and this is the greatest distinguishing characteristic of the SPORT HYBRID i-MMD.

3. Steering Vibration Issue and Reason for Overrunning Vibration Target

3.1. Steering Vibration Level in the Prototype

Figure 3 shows the level of steering wheel 2nd-order vibration occurring in the prototype 2014 model year Accord Hybrid (new model Accord Hybrid) when driving under moderate acceleration. The prototype overran the target line by 4 dB at approximately 1800 rpm to 2000 rpm, and this overrun had to be reduced.

3.2. Reason for Overrunning Steering Wheel 2nd-order Vibration Target

The mechanism by which the steering wheel 2nd-order vibration overran the target was analyzed in terms of the vibration exciting force.

The steering wheel 2nd-order vibration in a gasoline engine vehicle occurs via two routes. In one, crankshaft revolution 2nd-order fluctuation is transmitted from

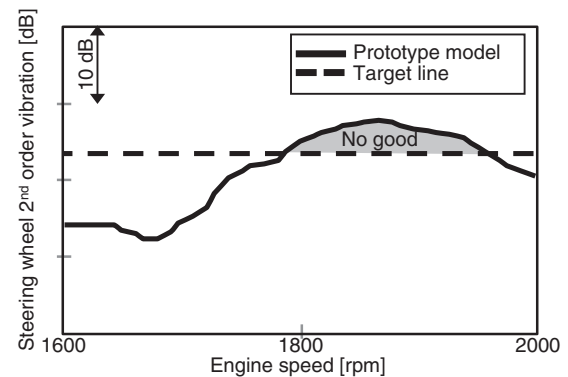


Fig. 3 Prototype model steering wheel 2nd order vibration and target line

drivetrain parts through suspension parts to the steering wheel. In the other, engine 2nd-order vibration that originates in crankshaft revolution 2nd-order fluctuation is transmitted through the engine mounts to the steering wheel.

When the new model Accord Hybrid is in hybrid drive mode, the engine power is used only to generate electricity with the generator, and there is no transmission route from the engine to the drivetrain. Therefore, the steering wheel 2nd-order vibration occurs through the transmission route by means of the engine mounts.

Figure 4 is a BSFC contour diagram that shows control configurations minimizing BSFC for each engine net torque in the 2.0-L engine used with the SPORT HYBRID i-MMD. It indicates that areas in the diagram where BSFC values are smaller have better fuel economy.

The outlined area marked with diagonal lines shows a gasoline engine vehicle traveling under moderate acceleration. The engine net torque varies greatly according to how the driver utilizes the accelerator, even at the same engine speed. Since the engine and the drivetrain are directly linked together, if the driver attempts to drive in the area of minimum BSFC, the resulting high engine net torque ends up causing the vehicle to accelerate suddenly. When driving under moderate acceleration, therefore, the high torque area where BSFC is minimal cannot be used readily.

When the new model Accord Hybrid is in hybrid drive mode, however, the engine power is not directly linked to the drivetrain. Consequently, the engine is not dependent on how the driver utilizes the accelerator, or on the vehicle acceleration or the vehicle speed demanded, and control can be implemented to keep the engine tracing the solid line in the diagram where the BSFC is always at a minimum. As a result, the engine net torque generated under moderate acceleration is greater than in a gasoline engine vehicle.

Figure 5 shows the relationship between engine net torque and steering wheel 2nd-order vibration, engine 2nd-order vibration, and crankshaft revolution 2nd-order fluctuation at 2000 rpm. The round symbols (○) in the figure mark the positions where engine net torque changed when the new model Accord Hybrid was operating in hybrid drive mode under moderate acceleration. The square

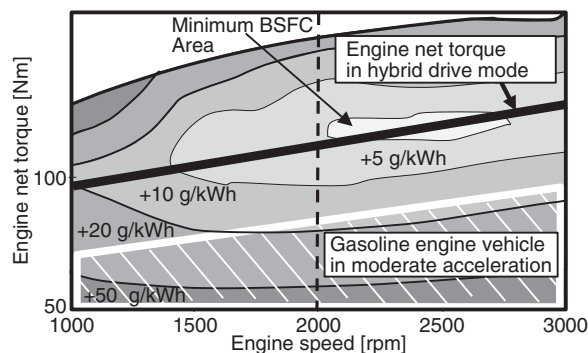
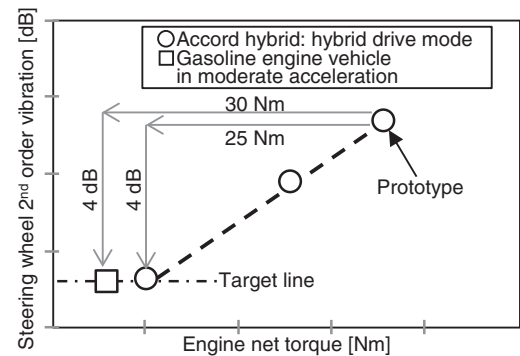


Fig. 4 BSFC and engine net torque under moderate acceleration in Accord Hybrid

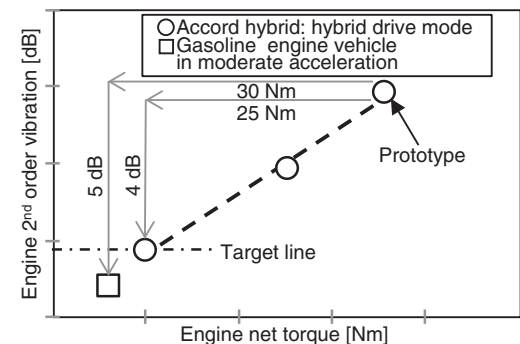
symbols (□) mark the positions where a gasoline engine vehicle was operated under moderate acceleration. These correlations exist because when the engine net torque increases, the engine 2nd-order vibration and crankshaft revolution 2nd-order fluctuation increase, and there is also a tendency for the steering wheel 2nd-order vibration to increase.

The prototype of the new model Accord Hybrid has engine net torque that has increased by approximately 30 Nm relative to gasoline engine vehicles. The engine 2nd-order vibration shows an increase of 5 dB and the crankshaft revolution 2nd-order fluctuation an increase of 6 rpm.

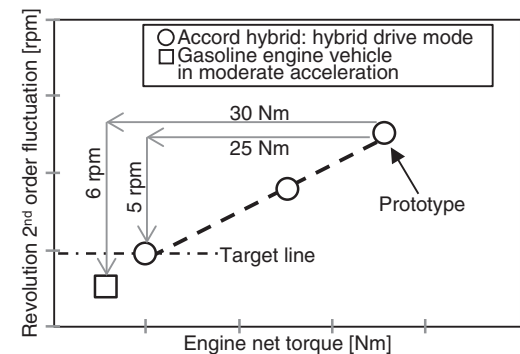
This suggests that the main reason for overrunning the steering wheel 2nd-order vibration target in hybrid drive



(a) Engine net torque and steering wheel 2nd order vibration at 2000 rpm



(b) Engine net torque and engine 2nd order vibration at 2000 rpm



(c) Engine net torque and revolution 2nd order fluctuation at 2000 rpm

Fig. 5 Engine net torque and each 2nd order parameter at 2000 rpm

mode is the increase in engine net torque.

It is also apparent that the engine 2nd-order vibration must be lowered by 4 dB and the crankshaft revolution 2nd-order fluctuation must be reduced by 5 rpm in order to meet the steering wheel 2nd-order vibration target.

4. Reduction of Crankshaft Revolution 2nd-order Fluctuation and Reduction of BSFC as Mutually Exclusive Aims

The extent of change in the BSFC due to steps to reduce crankshaft revolution 2nd-order fluctuation by limiting engine net torque was checked.

Figure 6 is a graph showing the relationship between engine net torque and BSFC. The tendency here is the reverse of what was seen in Fig. 5. When the engine net torque is reduced, the BSFC increases, and it is apparent that crankshaft revolution 2nd-order fluctuation and BSFC are mutually exclusive aims. This is because when engine net torque is reduced, it becomes more distant from the solid line marking minimum BSFC in Fig. 4. When engine net torque is reduced to a level that meets the crankshaft revolution 2nd-order fluctuation target, the BSFC increases by 6% relative to the prototype. It was decided, therefore, to investigate methods for obtaining a better BSFC level.

5. Mitigating the BSFC Increase by Combustion Control Parameter Optimization

The ideal in terms of NV and fuel economy is control that will meet the crankshaft revolution 2nd-order fluctuation target while reducing the amount by which engine net torque is limited so as to mitigate the BSFC increase. When the amount of electricity generated by the generator is also taken into account, it is desirable to have a higher engine net torque.

An investigation was made of a procedure to achieve this in which combustion control was configured by standards that would yield the BSFC values in Fig. 4

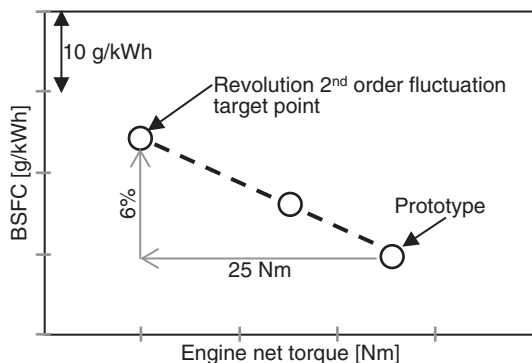


Fig. 6 Engine net torque and BSFC in hybrid drive mode at 2000 rpm

for every engine net torque. Then, from that, the ignition timing, valve operation timing, and other such combustion control parameters were altered to limit the engine net torque. In this way, the crankshaft revolution 2nd-order fluctuation, engine net torque, and BSFC were optimized.

5.1. Altering Combustion Control Parameters to Change Crankshaft Revolution 2nd-order Fluctuation and BSFC

A parametric study test was conducted of combustion control parameters as represented by throttle position (TH), ignition timing (IG), EGR volume, and valve operation timing (VTC) in the variable intake valve system. The test was used to search for a control point at which the 2nd-order fluctuation target is achieved while BSFC is kept to a minimum. This engine speed was set at 2000 rpm.

Figure 7 shows a 3D contour diagram that compares all the data from the test. The VTC 75 deg in the upper row and the VTC 50 deg in the lower row are the results of changes to TH, IG, and EGR volume. Larger VTC angles indicate timing that is farther on the advanced side. The left column shows contour diagrams of crankshaft revolution 2nd-order fluctuation, with darker red colors indicating larger values. Dotted lines demarcate the area in which the crankshaft revolution 2nd-order fluctuation target is achieved. The right column shows contour diagrams indicating BSFC, with darker red colors indicating larger BSFC.

Altering the combustion control parameters was found to significantly change the crankshaft revolution 2nd-order fluctuation and BSFC values.

5.2. Extracting the BSFC Minimum Point from the Crankshaft Revolution 2nd-order Fluctuation Target Achievement Area

Figure 8 shows an organized view of the parametric study test results from Fig. 7 with engine net torque on the horizontal axis and BSFC on the vertical axis. The contour diagram that is completely colored in shades of gray shows the magnitude of the crankshaft revolution 2nd-order fluctuation, with darker gray indicating a larger value. The target value for crankshaft revolution 2nd-order fluctuation was 8 rpm.

The solid line in the figure represents the line of change that occurs when engine net torque in the base settings for combustion control is reduced from the level marked with the round (○) symbol with regard to the prototype. The dotted lines in the figure link together the points marked with the X (×) symbol where BSFC is at a minimum under control conditions with crankshaft revolution 2nd-order fluctuation at 6, 7, 8, 9, and 10 rpm.

Comparison of these two lines shows that engine net torque is larger and BSFC is smaller on the dotted lines than on the solid lines. This is the case even when the crankshaft revolution 2nd-order fluctuation is the same.

The point on the solid line where the crankshaft revolution 2nd-order fluctuation target is achieved is marked

with a blank triangle ($\triangle$) symbol, and the point on the dotted lines where the target is achieved is marked with a black triangle ($\blacktriangle$) symbol. These two points are the same as the control points marked with a blank triangle ($\triangle$) symbol and a black triangle ($\blacktriangle$) symbol plotted on the 3D diagram in Fig. 7. On the black triangle ($\blacktriangle$) symbol, the crankshaft revolution 2nd-order fluctuation is equal to that on the blank triangle ($\triangle$) symbol, but the engine net torque is 5 Nm higher on the former. This is therefore the optimal combustion control point where 1% of the BSFC increase relative to the prototype is also recovered.

In this way, the optimal point for reducing the amount by which engine net torque is limited and mitigating the BSFC increase while achieving the crankshaft revolution 2nd-order fluctuation target was found.

5.3. Characteristics of the Cylinder Pressure Waveform When Optimizing Combustion Control Parameters

Figure 8 made clear that optimization could be sought by means of combustion control parameter conditions. An examination was made of the characteristics of the cylinder pressure waveform when that is done, as well as of the parameter changes that are necessary when seeking optimization by means of combustion parameters.

Table 1 presents a comparison of combustion parameters for Control A and Control B where engine net torque and BSFC are the same, but the crankshaft revolution 2nd-order fluctuation is different. Figure 9 shows the combustion pressure waveform for one cylinder at every crank angle for Control A and Control B in the upper half of the graph, and the total value of engine net torque for

Table 1 Comparison of engine control parameters for Control A and Control B

	TH [deg]	IG [deg]	EGR [%]	VTC [deg]	Revolution 2 nd order fluctuation [rpm]	Engine net torque [Nm]	BSFC
Control A	15	15	20	75	11	100	Same
Control B	17	28	10	50	9.2		

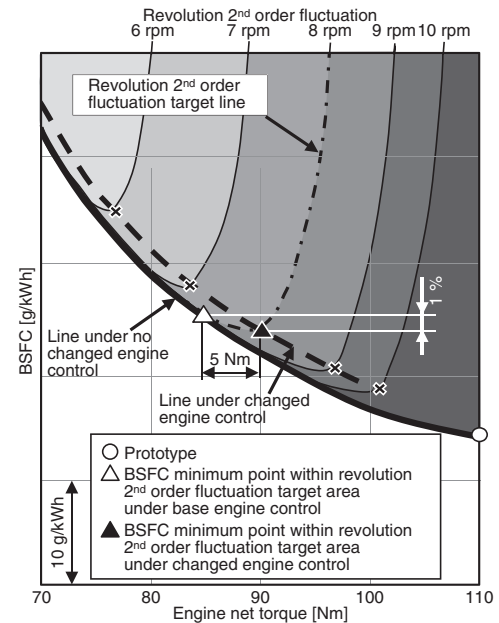


Fig. 8 Comparison of engine net torque, BSFC, and revolution 2nd order fluctuation under different engine combustion controls at 2000 rpm

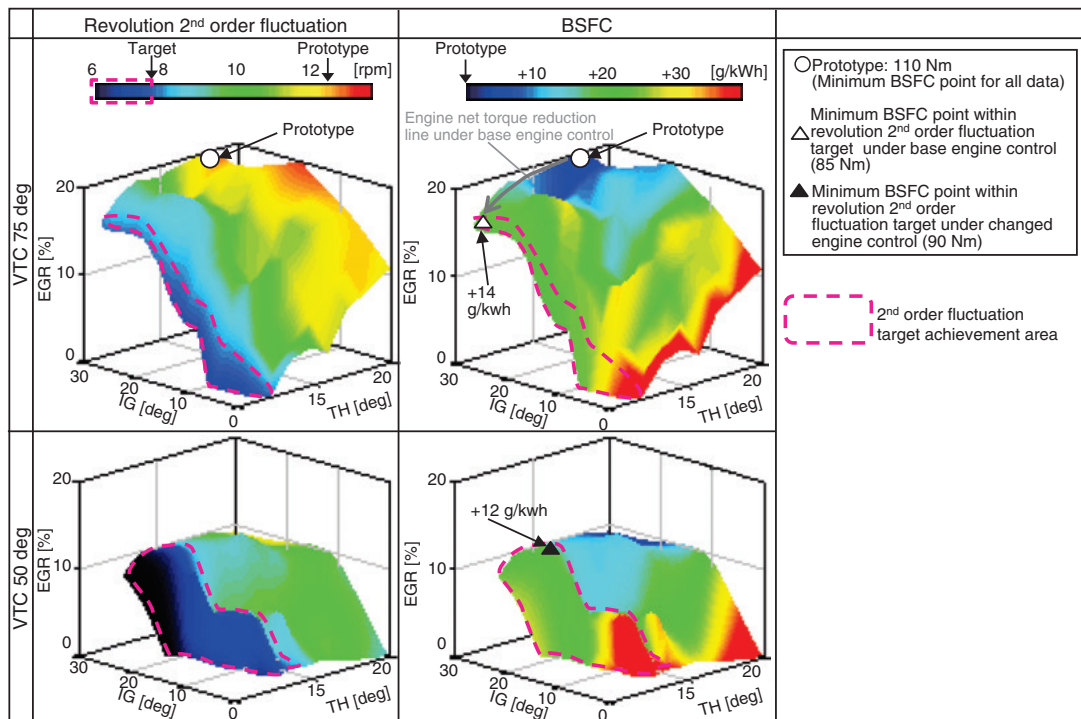


Fig. 7 Relationship of revolution 2nd order fluctuation and BSFC under various TH, IG, EGR, and VTC at 2000 rpm

all cylinders in the lower half of the graph. The crankshaft revolution 2nd-order fluctuation is smaller in Control B than in Control A, so Control B can be considered advantageous in this comparison of the two.

Focusing on the combustion pressure waveform, it is apparent that the Control B compression pressure near crank angle 340 deg, just before compression top dead center, is smaller than in Control A. On the other hand, the cylinder pressure peak that occurs after compression top dead center, near crank angle 375 deg, is higher than the peak in Control A and the position of the occurrence is also further on the advanced side of the ignition timing than in Control A. This indicates that the amount of air inflow in Control B is smaller than in Control A and that the ignition timing is further advanced. The influence of the amount of air inflow results in a smaller absolute value for the torque just before compression top dead center in Control B.

On the other hand, the torque after compression top dead center is the same in Control A and B. This is because, although the Control B cylinder pressure peak is higher because of the advanced ignition, the position where the peak occurs is near compression top dead center, and as a result, the component of cylinder pressure force that is converted into crankshaft torque is smaller.

Figure 10 shows a graph that compares each order of engine net torque fluctuation components for Control A and B. The 2nd-order component that is at issue in this paper is smaller in Control B.

This suggests that the compression pressure before compression top dead center has a significant influence on torque 2nd-order fluctuation. If the position where cylinder pressure immediately after compression top dead center reaches a peak is on the advanced side, at around crank angle 375 deg, it is likely to have little influence on torque 2nd-order fluctuation.

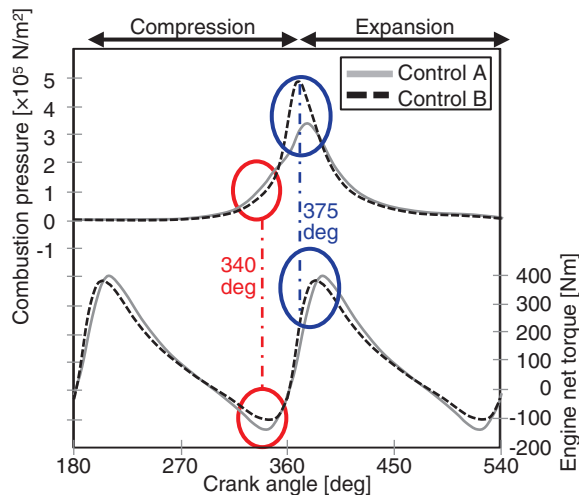


Fig. 9 Crank angle data of combustion pressure and engine net torque for Control A and B at 2000 rpm

These results indicate that combustion control that is desirable from the perspective of reducing crankshaft revolution 2nd-order fluctuation will close down the throttle and reduce the amount of air inflow into the cylinder while advancing the ignition timing to a point just before knocking begins in order to address the engine net torque deficit.

From the perspective of engine net torque and BSFC, on the other hand, it is desirable for combustion control that opens the throttle wide and advances the ignition timing in order to increase the amount of air inflow and reduce pumping loss. This increase in the amount of air inflow, however, results in increased crankshaft revolution 2nd-order fluctuation. Therefore, although advancing the ignition timing is effective in addressing both matters, the two are mutually exclusive aims where the throttle is concerned.

In order to achieve a balance of the two, it is effective to retard the VTC, which means adopting the Atkinson cycle so as to return the air that has flowed into the cylinder back into the intake during the compression cycle. That, together with the combination of a wider open throttle and advanced ignition timing, can be used to bring about optimization.

When the matter of reducing steering wheel 2nd-order vibration in the new model Accord Hybrid first came under investigation, it was thought that crankshaft revolution 2nd-order fluctuation was governed by the influence of combustion pressure after compression top dead center. The focus therefore was on reducing combustion pressure in order to limit engine net torque. At that time, no consideration was yet being given to compression pressure before compression top dead center. However, parametric study test results revealed that it was not only a matter of combustion pressure after compression top dead center; compression pressure before compression top dead center also exerted an influence that could not be ignored. It was found that extending consideration to reducing compression pressure, therefore, could realize more efficient combustion control.

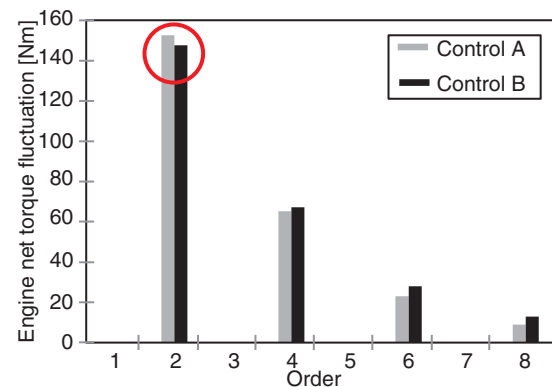


Fig. 10 Orders of engine net torque fluctuation for Control A and B at 2000 rpm

6. Results from Verification of Steering Wheel Vibration Reduction in the Actual Vehicle

Investigation of combustion control parameter optimization was also conducted by the same method for engine speeds other than 2000 rpm. This yielded the prospect of achieving objectives for 2nd-order vibration and fuel economy, and the approach was applied to a mass-production model.

Figure 11 shows the steering wheel 2nd-order vibration reduction effect. The steering wheel 2nd-order vibration in the mass-production vehicle was reduced a maximum of 4 dB relative to the prototype, enabling achievement of the target.

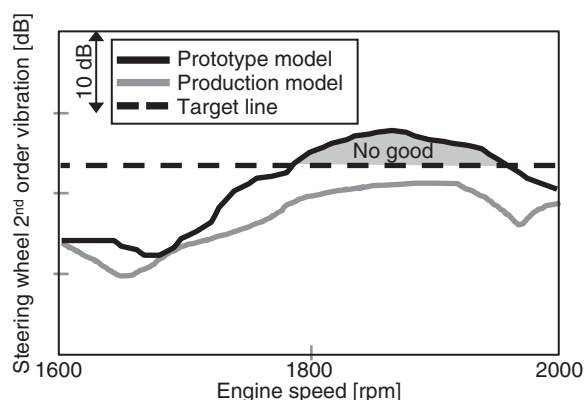


Fig. 11 Steering wheel 2nd order vibration

7. Conclusion

Analysis of the cause of steering wheel 2nd-order vibration and establishment of methods for reducing such vibration are described below. These enabled achievement of the steering wheel 2nd-order vibration target in an actual vehicle.

- (1) When the new model Accord Hybrid is in hybrid drive mode, implementing operation control that holds BSFC to a minimum at each engine speed raises the engine net torque by approximately 30 Nm relative to a conventional gasoline engine vehicle driving under moderate acceleration. That influence causes engine 2nd-order vibration to increase a maximum of 5 dB and crankshaft revolution fluctuation to increase by 6 rpm. This is the main reason that steering wheel 2nd-order vibration overran the target figure in the prototype.
- (2) A parameter study was conducted of combustion control that optimizes crankshaft revolution 2nd-order fluctuation, engine net torque, and BSFC for the purpose of reducing steering wheel 2nd-order vibration. The result was discovery of the optimal point at which combustion control aimed at adopting the Atkinson cycle limited engine net torque and achieved the crankshaft revolution 2nd-order fluctuation reduction target while

also recovering 1% of the BSFC increase relative to combustion control before changes were made.

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