

Development of CVT for 2004 Model Odyssey

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ABSTRACT

A 2.4L class CVT has been newly developed for the 2004 Odyssey to provide superior automotive fuel economy and a high level of driving pleasure. In addition to a torque converter, a first for a Honda CVT, the new CVT includes among its features lock-up control operating from immediately after start-off, a high-efficiency oil pump with a delivery switching mechanism, technologies enabling overall length to be reduced, a simple hydraulic circuit capable of optimizing pulley pressure, an ECU to realize excellent driving performance, and a three-part case. With a maximum transmitted torque of 218 Nm at 385 mm in length and weighing 95.3 kg the new CVT records 7.5-11% higher transmission efficiency than competitors' CVTs in the same class. The new CVT has enabled optimization of shift control, balancing fuel economy with sporty driving performance.

1. Introduction

Automotive fuel economy performance has become an extremely important issue as global concern over energy resources and the environment has increased. At the same time, manufacturers are required to increase the level of driving pleasure of their vehicles, which remains the basis of commercial appeal. As part of its response to these widespread social demands, Honda has employed CVTs offering superb fuel economy in its vehicle range since 1996. These CVTs have been consistently modified through the Civic, Logo and Insight ranges; significant modifications were made to enable the development of a new 1.5L class CVT able to satisfy demands for fuel economy and driving pleasure for use in the 2002 Fit.

This paper will discuss the technologies employed in the development of a new 2.4L class CVT offering both superb fuel economy and a high level of driving pleasure as part of the full model change of the Odyssey.

2. Development Goals

The following goals were established to enable the development of a 2.4L class CVT offering top-of-class fuel economy and running performance:

- (1) Use of torque converter to provide smooth and powerful start-off
- (2) Operation of lock-up clutch control from directly after start-off to minimize slip loss
- (3) Significantly increased transmission efficiency
- (4) Optimum shift control balancing fuel economy with sporty driving performance

3. Basic Layout and Main Specifications

Figures 1 and 2, respectively, show a cross-section and an external view of the CVT developed for the new Odyssey. Table 1 shows the major specifications of the CVT.

The basic layout of the CVT features a torque converter

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as part of the startoff mechanism (a first for a Honda CVT), a vane pump including a delivery switching mechanism to increase the efficiency of the drive system, a stator shaft with built-in oil channels to enable overall length to be reduced, a hydraulic pressure sensor which constantly monitors driven pulley pressure to increase efficiency, sharing of linear solenoid functions to make the structure of the control circuits more efficient, and the simplest possible three-part structure for the case.

These technologies have enabled the required specifications to be realized, with a maximum transmitted torque of 218 Nm at an overall length of 385 mm and a total weight of 95.3 kg.

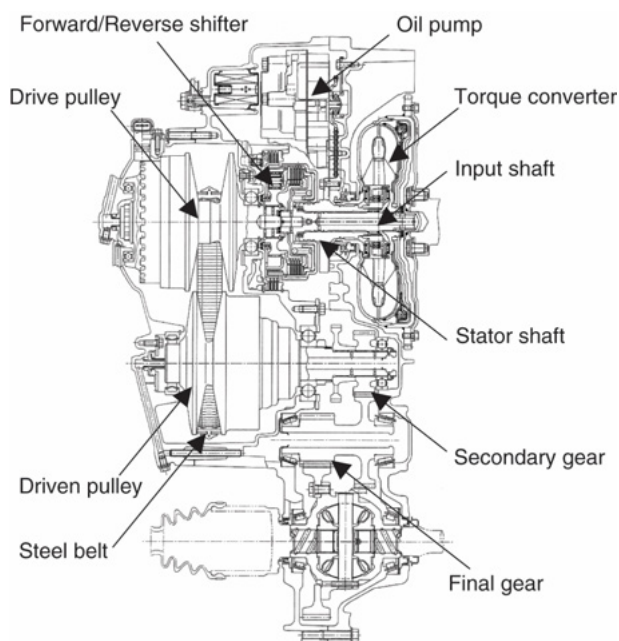


Fig. 1 Main section of CVT

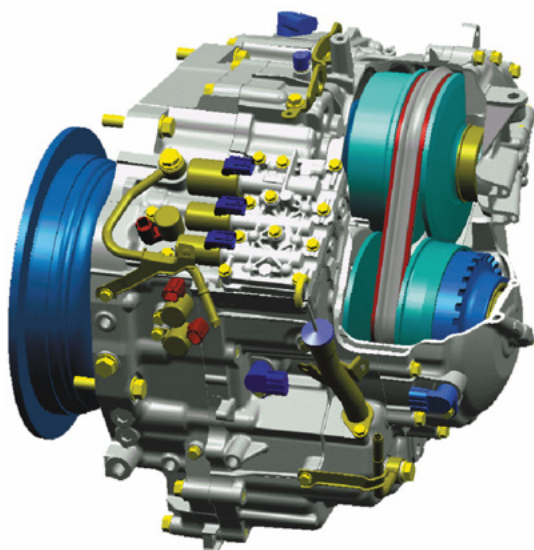


Fig. 2 View of CVT for Odyssey

Table 1 CVT specifications

		2004 Model Odyssey
Engine specifications	Type	2.4L 4VLV
	Max. power (kW/rpm)	118/5500
	Max. torque (Nm/rpm)	218/4500
Ratio	LOW	2.470
	OD	0.450
	Secondary	1.514
	Final	3.350
	Reverse	1.735
Dimension	Overall length (mm)	385
	Pulley axis length (mm)	170
	Weight (kg)	95.3 (Wet)

4. Main Technologies

4.1. Torque Converter

The employment of a torque converter in the startoff system enables the system to provide a smooth acceleration feeling at start-off, combining peak acceleration and continuous acceleration, as shown in Fig. 3.

4.2. Vane Oil Pump with Delivery Switching Mechanism

A high-efficiency oil pump has been developed to reduce the required oil pump drive torque. Inclusion of a delivery switching mechanism with the pump enabled a reduction in pump driving torque while guaranteeing the high oil pressure required for 2.4L engine torque loads and ensuring the required oil supply at low engine speeds.

4.2.1. Basic configuration

The former gear-driven oil pump has been replaced with a pressure balance-type vane oil pump to ensure stable high oil pressure and sufficient oil supply from low to high temperatures.

In addition, the oil pressure control valves have been built into the body and base making up the oil pump to make the unit more compact.

Figure 4 shows the basic configuration of the vane pump and the position of the valves.

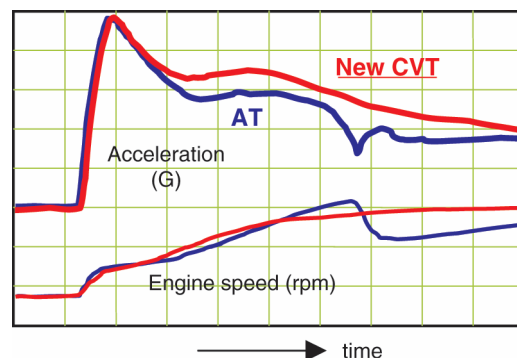


Fig. 3 Start-off acceleration characteristic

4.2.2. Delivery switching mechanism

The oil pump employs a delivery switching mechanism to effectively exploit the characteristics of a pressure balance-type vane pump in which two chambers take in and deliver oil with every revolution.

In driving modes in which the oil pump would supply excess oil, the pump switches from two-chamber delivery (full delivery) to one chamber delivery (half delivery). Signal pressure controlled by a separate ON/OFF solenoid operates a pump shift valve to switch delivery mode.

Figure 5 shows the oil delivery paths in full delivery and half delivery modes.

This system has enabled the drive torque for the pump to be reduced, thus increasing the transmission efficiency of the CVT.

Figure 6 shows the difference in pump drive torque for full delivery and half delivery modes.

4.2.3. Vane guide mechanism

The oil pump used in a CVT is required to produce the necessary oil pressure for control as soon as the engine is started.

In a vane pump, the resistance of the vane to movement is normally increased by the viscosity of the oil between the rotor and the vane when the engine is started at low temperatures, and oil pressure response therefore declines. A

vane guide, as shown in Fig. 7, has been employed in the new pump to prevent this from occurring, enabling oil pressure response to satisfy CVT performance requirements even when the engine is started at low temperatures.

4.3. High Torque Response

4.3.1. Metal pushing belt

A 30 mm metal pushing belt manufactured by Van Doorne's Transmissie b.v. has been employed in the CVT to ensure a maximum transmitted torque of 218 Nm. The shift ratio range used for the belt is approximately 5.5. Figure 8 shows a comparison of the dimensions of the 24 mm belt employed in the Civic and the 30 mm belt employed in the new Odyssey.

4.3.2. Responses to increased hydraulic pressure

To achieve high torque output in the new CVT, the maximum pulley pressure has been increased to 5.4 MPa from the maximum pressure of 3.5 MPa in the former CVT. Ensuring strength and resistance to deformation in the control valve bodies became a concern with this increased pressure, and FEM analysis was therefore employed to optimize the shape of the high-pressure oil channels and the bolt pitch in the valve bodies.

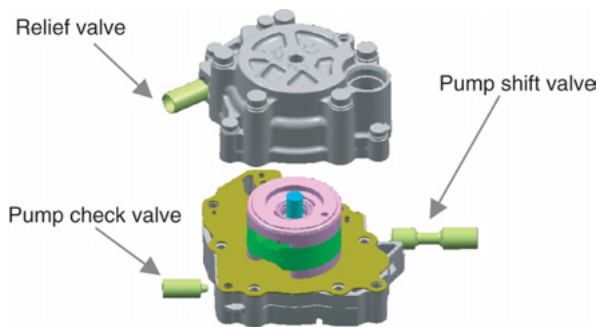


Fig. 4 Layout of pump and valves

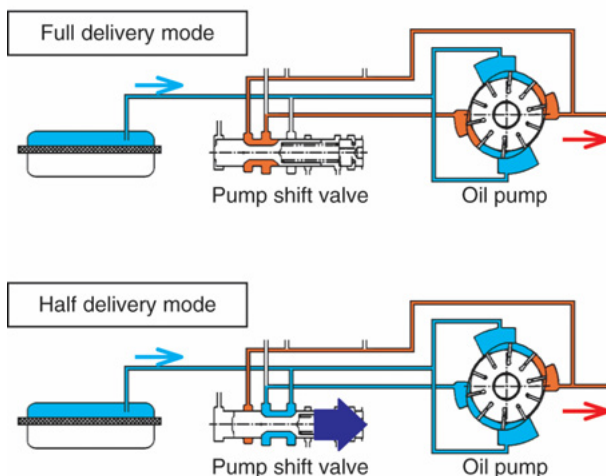


Fig. 5 Delivery modes of pump

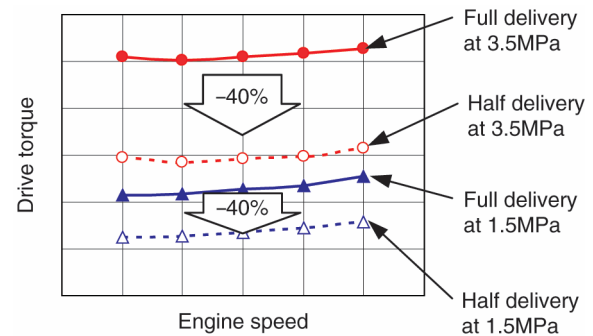


Fig. 6 Drive torque of pump

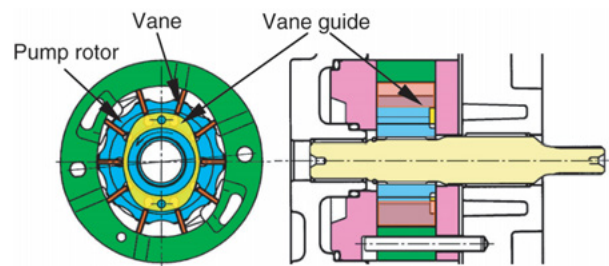


Fig. 7 Vane guide

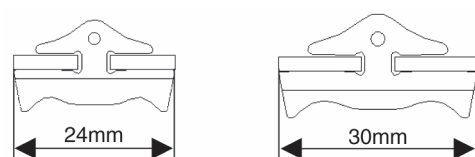


Fig. 8 Comparison of CVT belt size

The results of FEM analysis are shown in Fig. 9.

4.3.3. Case strength

FEM was also used in stress analyses and analyses of pressure distribution on the contact faces of the case in order to optimize the shape of each section as a response to the high transmitted torque of the CVT.

Examples of the results of stress analysis are shown in Fig. 10 and of analysis of contact face pressure distribution in Fig. 11.

Particular considerations in the optimization of the shapes employed in the case were ensuring the strength of the supports for the bearings around the final gear system and guaranteeing contact pressure between mating faces in sections in which it was impossible to use linear shapes between the bolts when coupling with the engine.

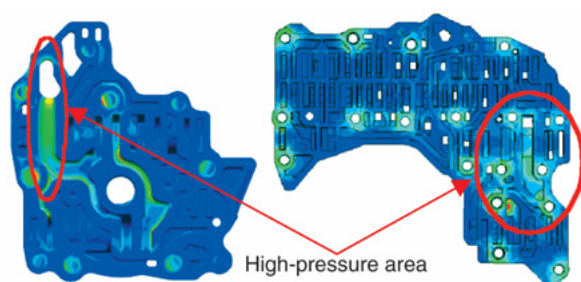


Fig. 9 Valve body stress analysis with FEM

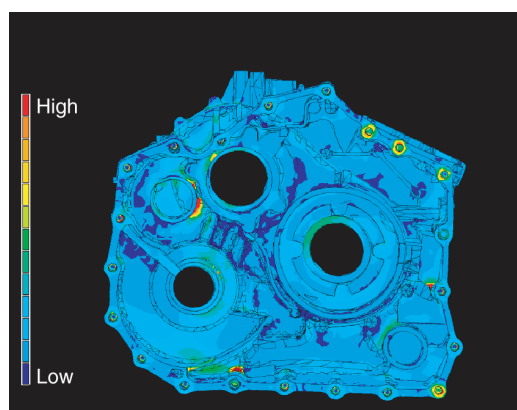


Fig. 10 Stress contour

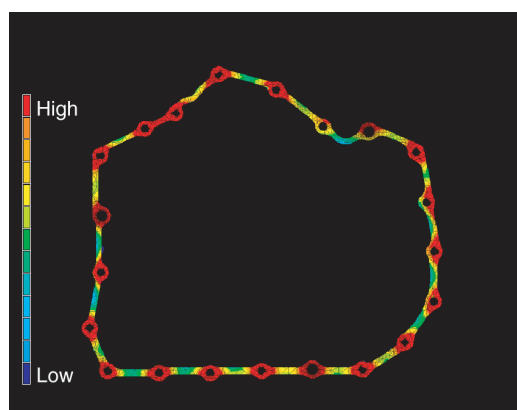


Fig. 11 Contact pressure

4.3.4. Pulley stiffness

FEM stress and deformation analyses of the four V-surfaces on the fixed and movable sides of the drive and driven pulleys were conducted to enable shapes to be optimized with consideration of a balance of stiffness between the four surfaces. The balance of stiffness was taken into consideration to minimize belt misalignment between the drive and driven pulleys and in particular to prevent contact between the metal belt rings and the pulley V-surfaces at high speeds and high loads.

Figure 12 shows the results of FEM analysis of pulley deformation. The amount of deformation shown here was used in analyses of the frequency of contact between the rings and pulley V-surfaces and the rings and elements during operation, with consideration of the trajectory of the belt.

4.4. Technologies to Reduce Overall Length

Development focused on elements in the drive pulley shafts to reduce overall length. This section discusses the main technologies employed and their effect in reducing the length of the CVT.

4.4.1. Stator shaft

Oil channels have been employed in the stator shaft to enable it to guide oil from the oil pump to the input shaft, and to supply oil to other components such as the torque converter. This has reduced overall length by approximately 25 mm against a conventional automatic transmission (in-house comparison).

4.4.2. Bearings

Length was further reduced by not directly clamping the drive pulley shaft support bearings to the drive pulley shaft. As shown in Fig. 13, the nuts connecting the pulley cover to the pulley shaft and the bearings are positioned in the same plane, reducing the overall length of the CVT by approximately 10 mm, which is equivalent to the width of the bearings.

4.4.3. Input shaft

The input shaft and the forward clutch can be coupled by a variety of methods, including spline clamping. In the new

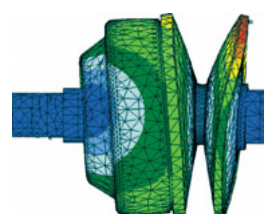


Fig. 12 FEM pulley deformation

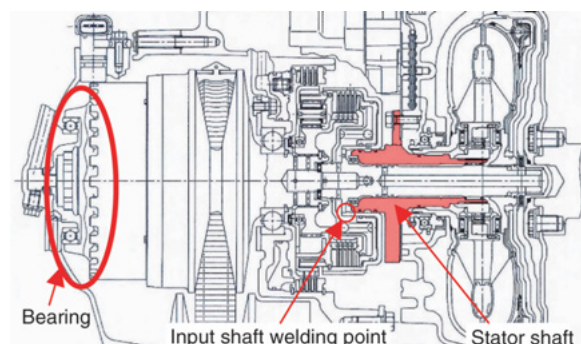


Fig. 13 Layout of drive pulley shaft

CVT they have been welded to reduce overall length. The welding point is shown in Fig. 13. This method of coupling has reduced the length of the unit by approximately 10 mm compared to spline clamping. The integration of the two components has also reduced the number of components employed.

4.5. Increasing Efficiency

4.5.1. Oil pressure sensor

An oil pressure sensor has been employed in the hydraulic pressure control system to provide constant monitoring, enabling self-learning control of hydraulic pressure as determined by the ECU in response to operating conditions and CVT operating hydraulic pressure through feedback of measurements. The use of the sensor has reduced the incidence of excess oil pressure and also enabled oil pump drive torque and belt friction to be reduced against the previous model.

Figure 14 shows the control logic of the oil pressure sensor.

4.5.2. Drive pulley double piston

The former single drive pulley piston has been replaced by a double piston, increasing the area of the piston receiving pressure while the external diameter remains the same. This has reduced the pulley control pressure when the vehicle is cruising, thus reducing the oil pump drive torque during cruise.

Figure 15 shows the area of the single and the double pistons on which pulley pressure acts, and a cutaway view of the piston mechanism.

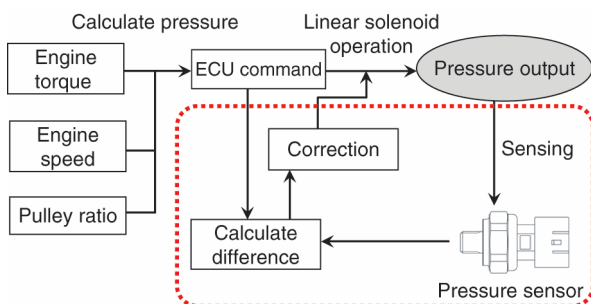


Fig. 14 Control program of pressure sensor

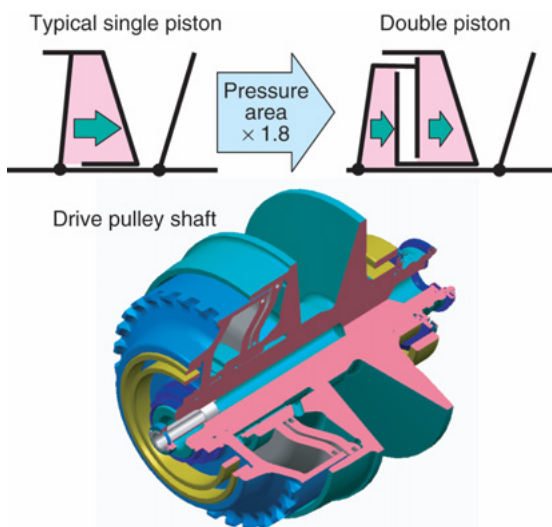


Fig. 15 Double piston mechanism

4.5.3. Reduction of drag resistance of reverse brake

When a vehicle is moving forward, the oil between the reverse brake disc and plate generates constant viscous dragging resistance. This resistance has been reduced in the Odyssey through the employment of a wave disc with a number of wave-like undulations on the circumference.

Figure 16 shows the relationship between drag resistance and engine speed.

4.5.4. Reduction of oil agitation resistance

Because the drive pulley is positioned in the lower section of the CVT, the accumulation of oil in the belt housing would significantly increase agitation resistance. A discharge hole enabling high-efficiency discharge of the oil accumulated in the belt housing after lubrication of the belt and bearings has therefore been employed.

In addition, ribs have been positioned to enable high-efficiency return of the gear and bearing lubricating oil and the oil discharged from the control system to the oil pan, in order to use less oil to maintain the required level to prevent aeration when the vehicle is stopping, starting off, or turning.

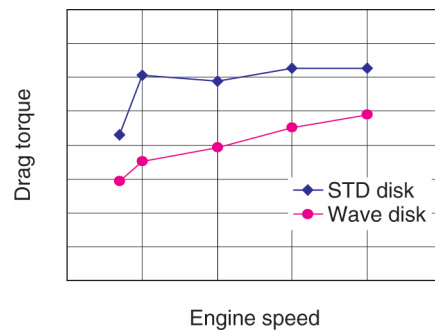


Fig. 16 Drag torque of reverse brake

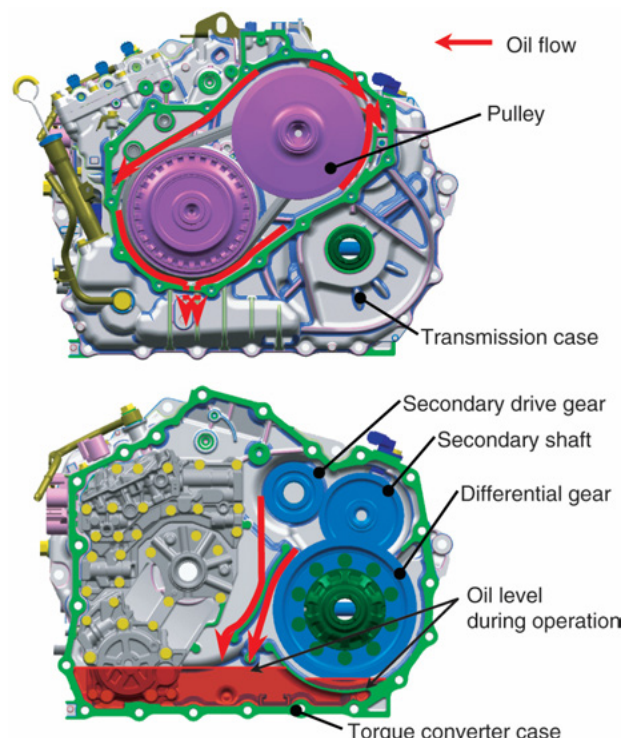


Fig. 17 Oil flow inside case

Figure 17 shows the oil flow inside the transmission.

The employment of the technologies described above has reduced friction torque and increased transmission efficiency by 6%, to a level 7.5 - 11% higher than competitors' CVTs in the same class, as shown in Fig. 18.

4.6. Hydraulic Control System

A direct control system has been employed for the pulley hydraulic control circuit, in which two linear solenoids separately control drive and driven pulley pressure. The employment of a double piston for the drive pulley in the CVT has significantly reduced drive pulley pressure. This eliminates the need for a pulley line pressure control valve, enabling the pulley hydraulic control circuit to be simplified.

In addition, a single solenoid valve is employed for forward and reverse clutch control when the system is in gear and lock-up clutch control. Forward and reverse clutch control and lock-up clutch control are divided into four modes, in gear mode, startoff mode, cruising mode and deceleration mode. Two ON/OFF solenoids enable switching between the modes. This has reduced the shift shock when the vehicle is put in gear, ensured lock-up controllability during startoff and cruising, and enabled control to be applied to protect the belt during deceleration using a single linear solenoid valve, while reducing costs and increasing the efficiency of the space layout.

The hydraulic control system is shown in Fig. 19.

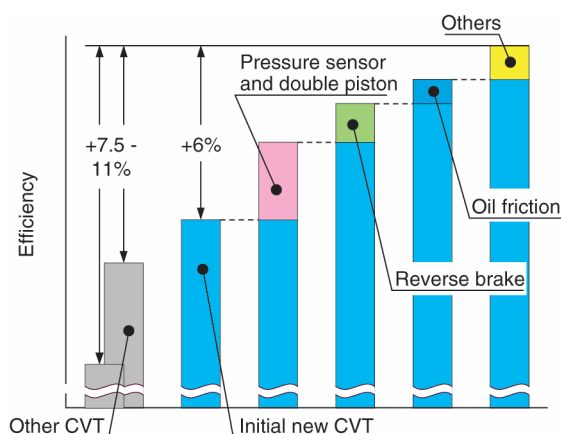


Fig. 18 Transmission efficiency at 60km/h cruise

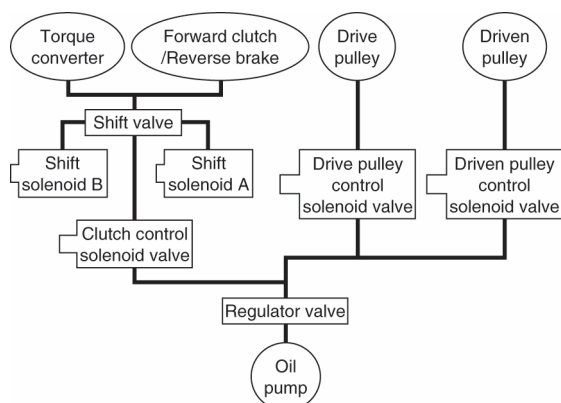


Fig. 19 Hydraulic circuit

4.7. ECU Control System

4.7.1. Lock-up clutch control

Employing a torque converter provided with a lock-up clutch as a startoff device can heighten the appeal of a vehicle by increasing startoff torque against a conventional start clutch, but slip loss at startoff decreases efficiency.

In the new CVT, the range of operation of the lock-up clutch has been extended to vehicle startoff, minimizing slip loss and increasing efficiency.

Figure 20 shows the operating range of the lock-up clutch.

At vehicle speeds close to 0 km/h the lock-up clutch is not engaged, and maximum use has been made of the torque amplification of the torque converter in order to increase the appeal of start-off performance. The moment an operating range is reached in which the torque amplification ratio of the torque converter drops, the lock-up clutch engages and provides control coordinated with the throttle opening angle. At low opening angles, clutch capacity rises rapidly, in order to minimize slip loss, while at high opening angles it rises slowly, in order to prioritize power performance.

Figure 21 shows the slip rate characteristic of the lock-up clutch against vehicle velocity at startoff for low and high throttle opening angles.

As discussed above, the characteristics of the torque converter have been exploited to ensure commercially appealing startoff performance, and optimization of lock-up clutch control has increased efficiency at startoff.

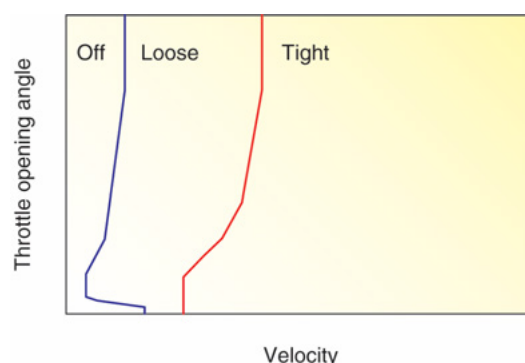


Fig. 20 Lock-up clutch map

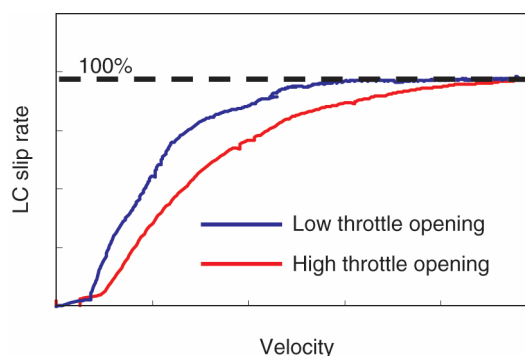


Fig. 21 Lock-up clutch slip rate characteristic

4.7.2. Shift control

The shift control in the new CVT has been optimized for a variety of different driving modes, and represents a new advance in shift control technology.

The following are employed as criteria in determining the driving mode:

- Vehicle velocity
- Throttle opening angle history
- Cornering G (lateral acceleration) history

Switching control between three shift schedules (D1/D2/S shift schedules) and cornering G shift control is applied on the basis of these criteria.

(1) D1 shift schedule

This shift schedule prioritizes fuel economy, and is engaged at low throttle opening angles. In this schedule engine speed is limited and engine torque is the main power source.

(2) D2 shift schedule

This schedule engages when the frequency of medium throttle opening angles increases. Engine speed is higher in D2 than in D1, and the schedule offers superior acceleration performance.

(3) S (Sports) shift schedule

The S shift schedule engages when the frequency of high throttle opening angles increases. Engine speed is higher than D2, and the schedule provides acceleration and engine/brake performance appropriate to sporty driving. Figure 22 shows an image of switching between shift schedules, and Fig. 23 shows a comparison of D1, D2 and S. The schedules offer different shift characteristics at identical throttle opening angles.

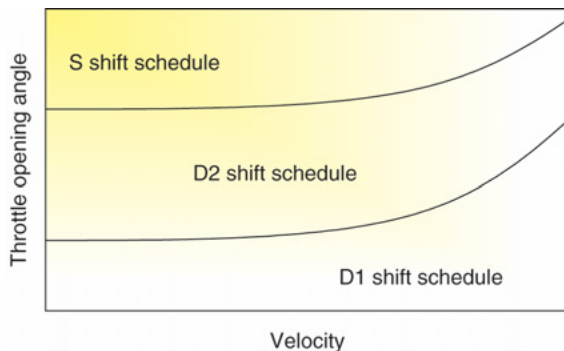


Fig. 22 Switching between shift schedules

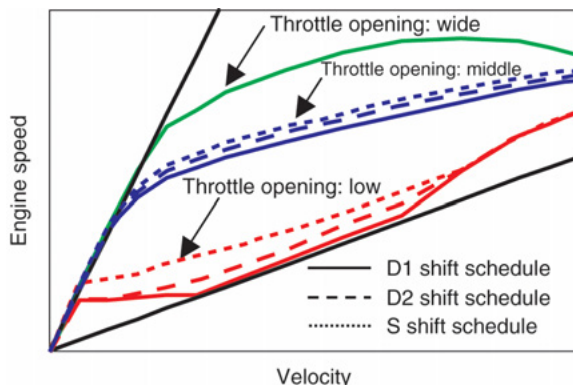


Fig. 23 D1, D2 and S shift schedules

(4) Cornering G shift control

Cornering G shift control uses cornering G history to determine whether the vehicle is on a winding road. When the control goes into operation, it effectively maintains a high engine speed.

(5) Effect of switching between shift schedules

Figures 24 to 26 show the results of measurements of the relationship between vehicle velocity and engine speed in a variety of driving modes.

Figure 24 shows the results of measurements conducted during city and highway driving. These driving modes employ the D1 shift schedule, and the measurements indicate that engine speeds remain low.

Figure 25 shows the results of measurements taken during uphill and downhill driving. In this driving mode, D2 and uphill/downhill shift schedules are engaged. This driving mode uses higher engine speeds than city driving, offering the user a smooth drive feeling despite the fact that the vehicle is proceeding up or downhill.

Figure 26 shows the results of measurements taken during driving on winding roads and circuits. This driving mode uses S and cornering G shift schedules, enabling even higher engine speeds to be maintained. These shift schedules ensure a superb drive feeling even as the vehicle is operating in difficult running conditions.

The automatic switching between shift schedules according to driving mode described above has enabled the achievement of a balance of fuel economy and driving performance in D range driving without the need for the driver to touch the shift lever.

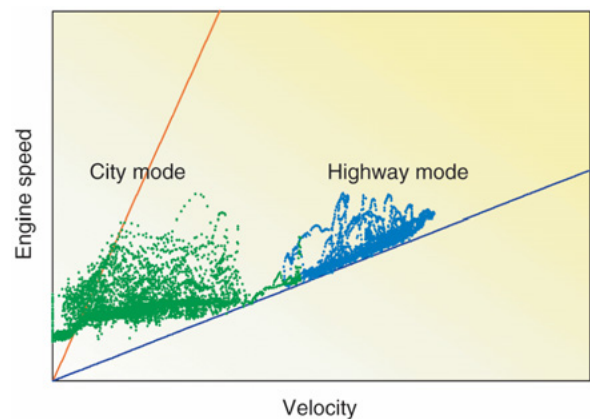


Fig. 24 City and highway mode

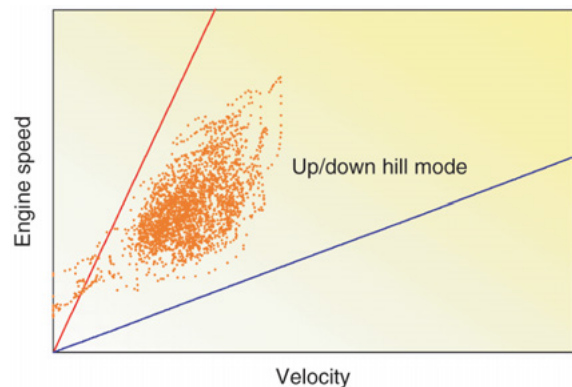


Fig. 25 Uphill and downhill

5. Conclusion

The development of a new 2.4L class CVT employed the following technologies:

- (1) Lock-up clutch control optimization technology to extend the range of operation of lock-up clutch control to directly after start-off in order to minimize slip loss while enabling the characteristics of the torque converter to be maximized to increase commercial appeal
- (2) Technologies to significantly increase the transmission efficiency of the CVT (7.5% - 11% higher than competitors' CVTs)
- (3) Automatic shift schedule switching technology to enable a balance between fuel economy and sporty driving performance

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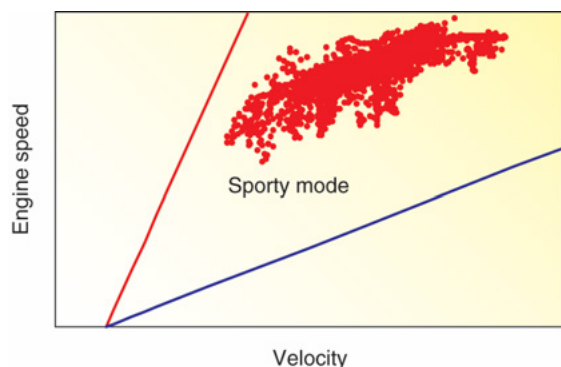


Fig. 26 Sporty mode

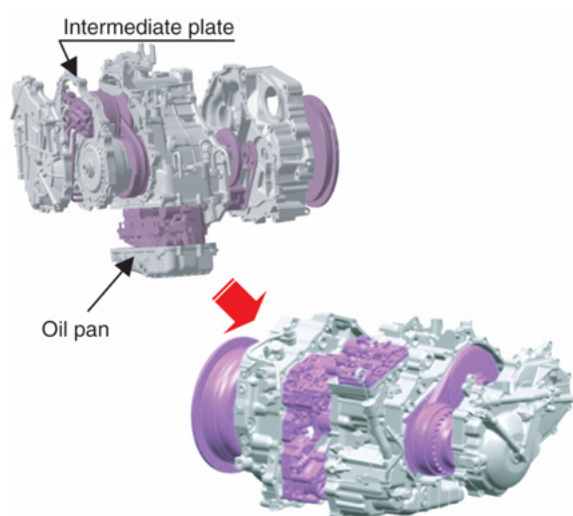


Fig. 27 Transmission consists of three cases

4.8. Three-part Case Structure

The new CVT case has been simplified in comparison to the former case, which was made up of five parts. The oil pan-less structure of the case used in the Fit was adopted and further developed to eliminate the intermediate plate, resulting in a three-part case structure.

Figure 27 shows a comparison of the conventional and the newly developed CVTs. The conventional CVT employed a start clutch and required an intermediate plate to enable positioning of a forward and reverse drive switching mechanism and to support the pulley shaft. The use of a torque converter in the new CVT enabled the positioning of the forward and reverse drive switching mechanism and the pulley oil chamber to be changed, contributing to the realization of a three-part case configuration.

The new configuration has reduced the number of mating faces in the case, increasing the rigidity of the CVT. In addition, it has been possible to both shorten and reduce the number of the bolts employed in the case, which has contributed to weight savings.

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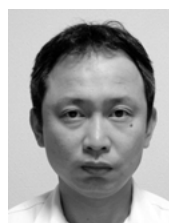
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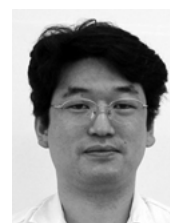
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