

The Development of a New Compact CVT for the Fit

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ABSTRACT

Honda has developed a new continuously variable transmission (CVT) for use with new-generation power plants, and this compact, lightweight CVT, known as the Multimatic S, has been installed on the Fit. The redesigned flywheel, oil pump, clutch, and transmission case used in this CVT reduce the number of parts by 15% while simultaneously lowering friction and improving transmission efficiency by 5% over the 1999 model Logo. The new electronic control also provides creep control optimized for cruising conditions and automated switching of the shift schedule. These have yielded both improved fuel economy and a feeling of responsiveness to a wide range of driver demands.

1. Introduction

Year by year there has been increasing demand for better automotive fuel economy, with the focus being protection of the global environment and conservation of resources. The Honda Multimatic, which has excellent fuel efficiency, was used in the '96 Civic and the '98 Logo. Then, for the '99 Hybrid car "Insight", an even more highly efficient, compact CVT Multimatic was used.⁽¹⁾

Now, a new CVT has been developed for the new generation, a New Small "Fit", which replaces the Logo. The technology for making the CVT compact is based on that adopted for the Insight, and further modified.

This paper introduces the new technology.

2. Aims of Development

In order to actualize the Fit's concept, the following performance enhancements and ease of manufacturing improvements were pursued.

(1) A compact design that responds to a short nose package

layout, to achieve both "a spacious interior" and "best-in-class collision safety performance."

- (2) Making the CVT low friction and highly efficient, and modifying CVT controls, all to contribute to "best fuel economy in the world for its class."
- (3) Start-off and shift controls that give "smooth driving pleasure," matching the driver's senses.
- (4) Improve ease of manufacture by reducing the number of parts.

3. Basic Layout and Principal Specifications

Fig. 1 and Fig. 2 show the main section of the CVT and an outside view, while Table 1 gives the major specifications.

The main changes from the '99 Logo (referred to as previous model, hereafter) include a shorter pulley axis length, elimination of the oil pan (a structure without an oil pan), changes to control system layout, adoption of a trochoid pump and elimination of the drive chain, and adoption of a single mass flywheel.

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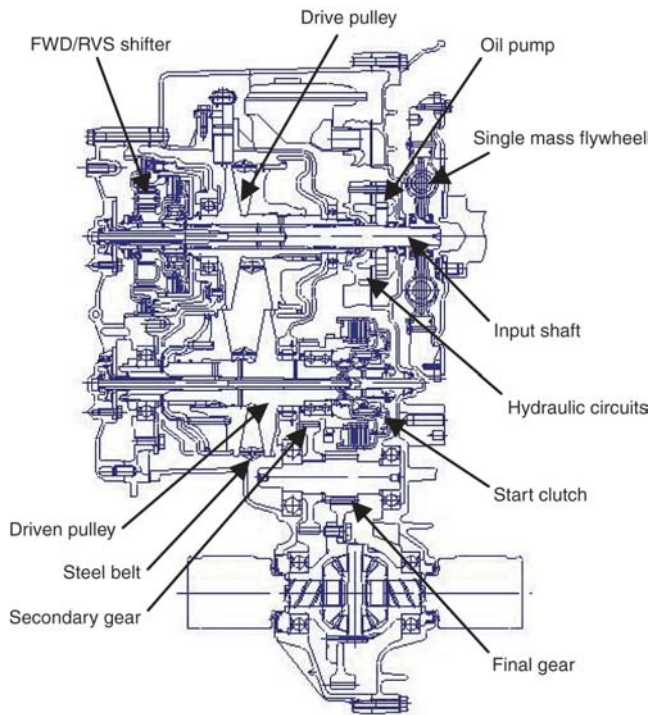


Fig. 1 Main section

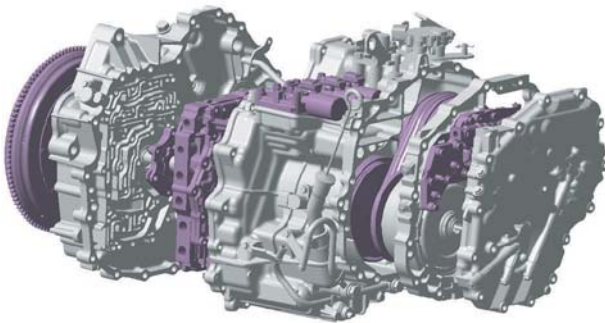


Fig. 2 Outside View

Table 1 Major Specification

		2002 model Fit	1999 model Logo
Engine Specification	Type	1.3L 2VLV	1.3L 2VLV
	Max. power (kW/rpm)	63/5 700	49/5 500
	Max. torque (N·m/rpm)	119/2800	103/2500
Ratio	LOW	2.367	2.466
	2	2	2
	OD	0.407	0.449
	Secondary	1.333 (52/39)	1.333 (52/39)
	Final (2WD)	4.333 (65/15)	4.357 (61/14)
	Final (4WD)	4.714 (66/14)	
Dimension	Reverse	4.226~2.367	2.466
	Overall length (mm)	349	372
	Pulley axis length (mm)	143	155
Weight (Wet)	2WD (kg)	74.5	80.4

With these changes, in comparison with the previous model, a 6.2% reduction in total width, and a 7.3% reduction in weight were achieved. The number of parts was reduced by 15%, from 770 to 655. Fig. 3 shows parts and weight comparison to the previous model.

Fig. 4 shows an external shape comparison with the previous model (contour profile).

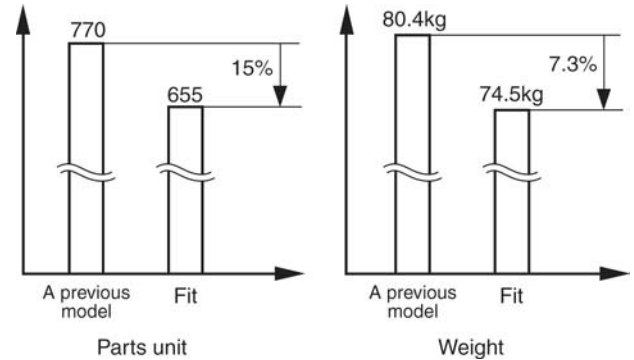


Fig. 3 Comparison of complete parts unit and weight

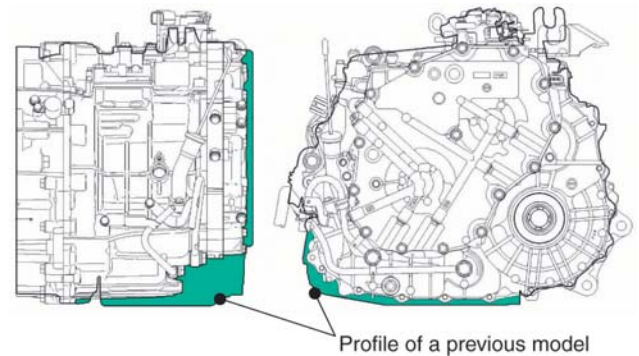


Fig. 4 Comparison of external shape

4. Technologies Adopted

4.1. Single Mass Flywheel

4.1.1. Background to application

The previous model used a high damping performance dual mass type flywheel, in order to prevent gear rattling noise during reverse idling, generated by the planetary gear

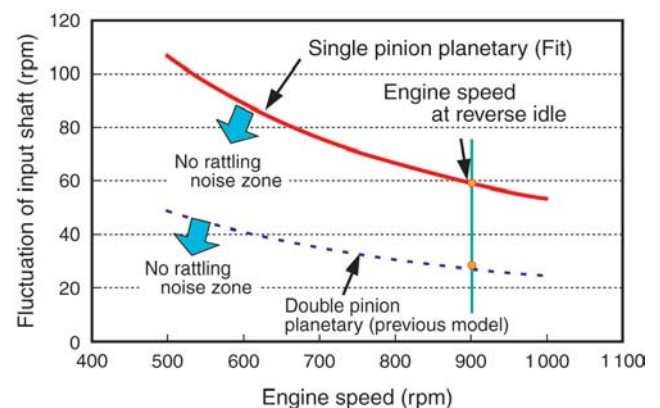


Fig. 5 Rattling noise zone chart

for reverse mechanism. However, with the change in planetary gear from a double pinion type to a single pinion type, and with a reduction in pulley inertia by shortening the pulley axis length, the generation limit for gear rattling noise due to rev fluctuation is more than twice what it has been conventionally, so it became possible to change to a single mass flywheel. Fig. 5 shows the reversing gear rattling noise generation limits with single and with double pinion planetary gears.

4.1.2. Basic structure

Because the inertial mass is not divided in a single mass flywheel, it is a simpler structure than the dual mass flywheel. Comparing the number of parts, a dual mass flywheel has 39 parts, while a single mass type has 29 parts. Fig. 6 shows the structure of the single mass flywheel.

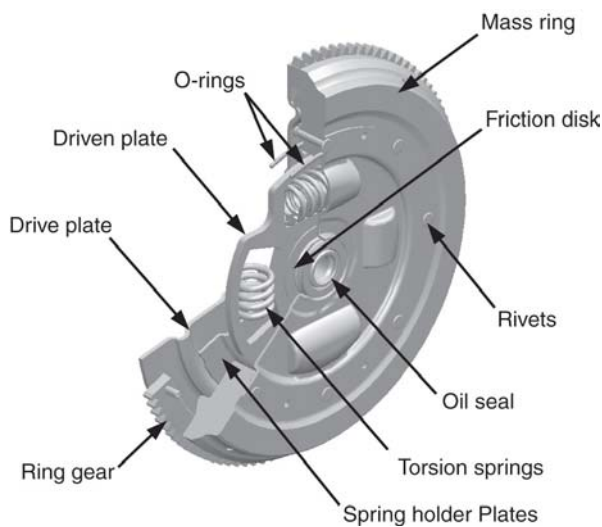


Fig. 6 Structure of single mass flywheel

Regarding torsion characteristic, as shown in Fig. 7, with 3-stage characteristic using two of each 3 types of springs, maximum torsion angle $\pm 25^\circ$, and torque capacity is a 207 Nm long travel type.

In addition, in order to generate hysteresis torque, plastic friction disks were added to each space between spring holder plates and driven plates, and utilizing spring holder plate's elastic deformation, pressing load on friction disks is generated.

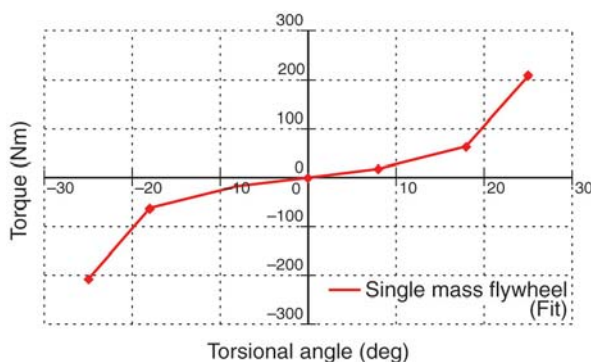


Fig. 7 Torsion characteristics

4.1.3. NV improvements

With a dual mass flywheel, inertial mass is divided into primary and secondary, while with a single mass flywheel, the inertial mass is one piece, so it became possible to increase engine side inertial mass. As a result, engine rev fluctuations are reduced, and there is an improvement in NV performances, such as surge and booming noise.

Fig. 8 shows engine rev fluctuation with both flywheels.

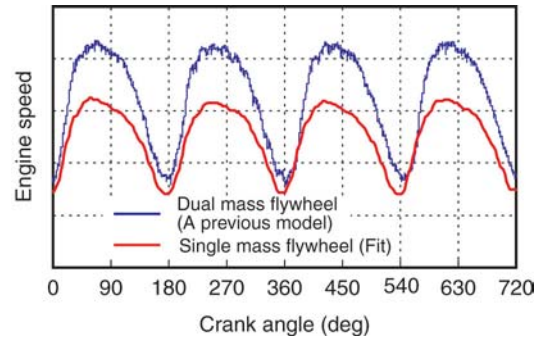


Fig. 8 Cycle irregularity

4.2. Direct Driven Trochoid Oil Pump

4.2.1. Reduction in number of parts (rational design)

With the previous model, the input shaft drove the pump shaft with a chain. With this CVT the input shaft directly drives the pump, so parts such as the chain, sprockets, etc., were eliminated, reducing the number of parts.

The pump itself is a trochoid pump using a sintering rotor, with the inner rotor driven by input shaft spline.

A comparison with the previous model is shown in Fig. 9.

4.2.2. High pressure measures

As this CVT uses a discharge pressure of 3.5 MPa or more, there is a drop in discharge performance due to deformation of the pump body. FEM analysis was used for carrying out optimization of each pump part's shapes, at the same time, pump body was changed to cast iron to reduce deformation. By these effects, discharge performance in a high pressure state can be maintained.

Fig. 10 shows the results of the FEM analysis.

In addition, because the pump rotor's and body's coefficients of thermal expansion are roughly equal, there is little side clearance changes due to temperature changes, and stable discharge performance is obtained.

4.2.3. Discharge performance

The oil pump discharge performance is shown in Fig. 11. With a theoretical discharge amount of 9 cc/rev, at high oil temperature, low rev speed, the pump discharge amount can be maintained at a level equivalent to that with previous model, or better.

4.2.4. Friction reduction

Many trochoid pumps have a hub on the inner rotor, and the structure is supported by a bush. With that particular structure, hydraulically there is a load on the inner rotor, and momentum load is generated by the offset of rotor and hub.

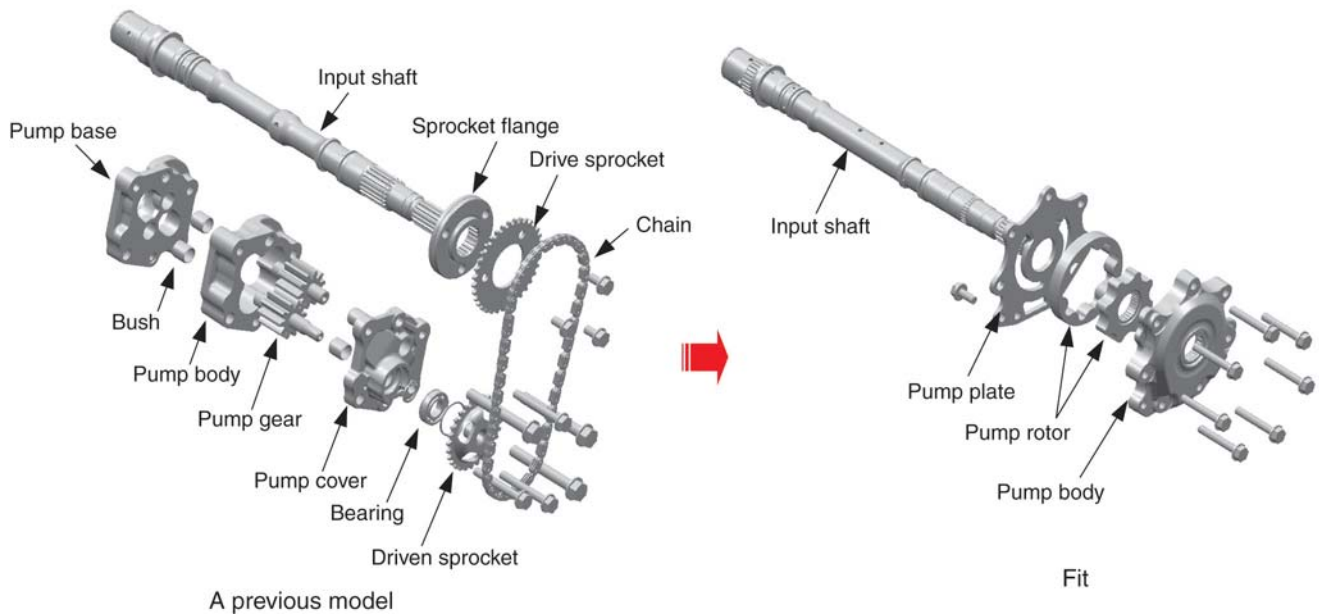


Fig. 9 Oil pump structure

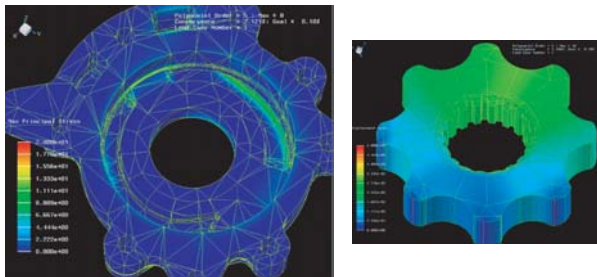


Fig. 10 Pump body stress analysis with FEM

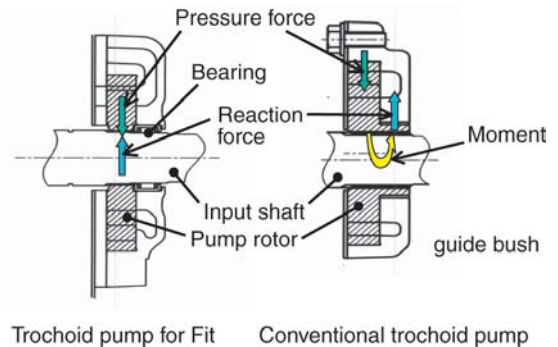


Fig. 12 Reduction of oil pump rotor friction

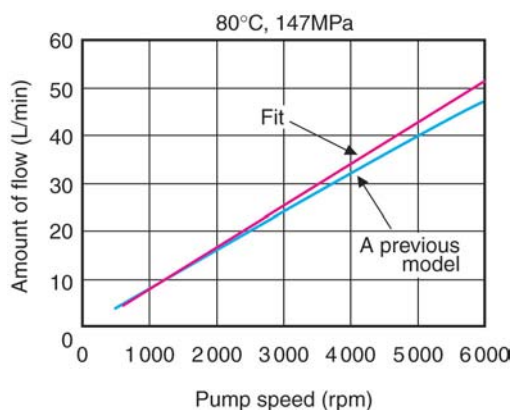


Fig. 11 Oil pump delivery performance

With this oil pump, in order to eliminate the friction caused by the offset load, the structure is such that the inner rotor is supported by the input shaft, and there is a needle bearing on the pump body that determined the position of the input shaft.

Fig. 12 shows a comparison with the conventional structure.

In addition, by eliminating the chain to drive the pump, both drive friction and the friction caused by oil agitation has been reduced.

4.2.5. Reduction in pump noise

To improve pump noise, it is necessary to reduce radiation noise generated by the pump itself, and reduce the transmission of pump vibration to the vehicle body.

As a modification for pump unit radiation noise, the intake port timing advance angle is established to balance manifold vacuum and discharge performance. Discharge noise countermeasures were established by setting the pressurization compression amount between intake port and discharge notch and by setting the timing, width, and depth of the discharge notch 1st stage flat part. Furthermore, by setting the 2nd stage cone shape, pressure change during discharge is made gentler, and generation of cavitation sound is prevented. Fig. 13 shows the discharge notch shape.

As a transmission noise countermeasure, a rib was added to the rear mounting part of the flywheel case to reduce the pump vibration transmission to the body, and as a result, body cabin noise was reduced.

4.2.6. Improved ease of manufacture

(1) Ease of Machining

Because the inner rotor does not have a hub, it is possible to grind both sides of the rotor at the same time, using a double head grinding machine, thereby improving dimensional accuracy and machining speed.

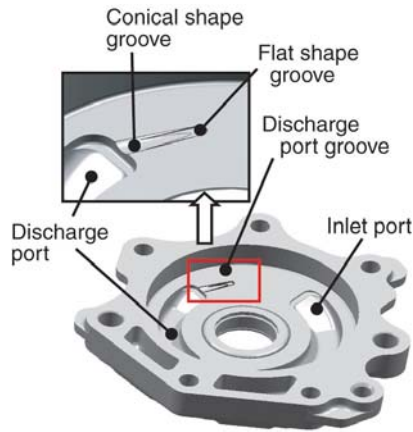


Fig. 13 Shape of discharge port notch

(2) Ease of Assembly

With this CVT, the valve body that was fitted to the oil pump is directly placed on the flywheel case, and is directly driven from the input shaft.

Thanks to this, ease of assembly is improved, as there is no assembly of the drive chain or sprockets, etc. Also, by making the pump body, rotors, and plate a sub-assembly, it is possible to guaranty function with the pump unit parts.

4.3. Structure without an Oil Pan

By creating a layout where the starter motor, conventionally on the CVT side, is moved to the engine side, and moving the control body that had been in the center of the oil pan into the space created by the starter motor shift, the oil pan was eliminated, and the number of case parts was reduced.

Fig. 14 is a comparison with the conventional CVT with an oil pan. As shown in the diagram, integrating the 3 linear solenoid valve bodies that control the drive and driven pulleys and the start clutch control, they are placed on the top of the CVT. The reverse inhibitor solenoid valve, too, is

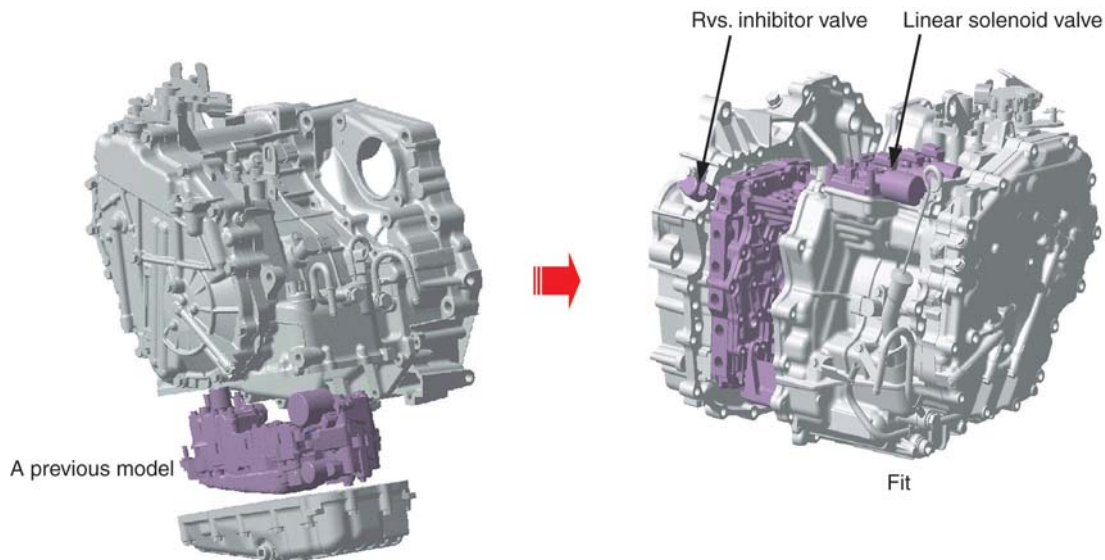


Fig. 14 Casing without an oil pan and valve body layout

placed on the outside, improving the ease of maintenance of electronic control parts. Other hydraulic control parts, in combination with an oil pump, are placed so that they are directly mounted to the flywheel case. As for the space between the flywheel case and the pulley chamber to its left, conventionally the oil pump and drive parts are located there, but the drive chain and sprockets have been eliminated. By combining the pump with the control body, it was possible for the hydraulic system to be placed in the space conventionally occupied by the oil pump.

From these changes, oil-filling capacity is increased, and oil level can be set at a lower level, thereby reducing agitation resistance.

4.4. Start Clutch Modifications

4.4.1. Bonded seal

Conventional clutch pistons are inserted into separate oil seals, but with this clutch the pistons and the oil seals are integrated using a bonded seal piston.

As a result, the start clutch is more compact, and the number of parts is reduced. Fig. 15 shows as comparison with the previous model's start clutch pistons.

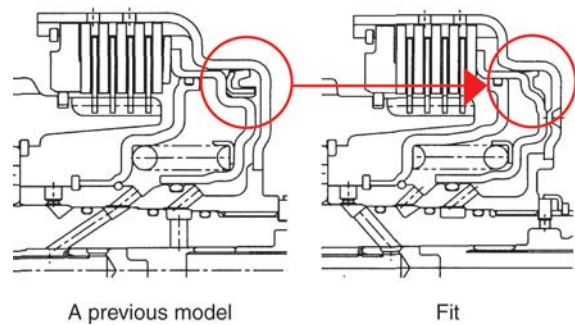


Fig. 15 Structure of start clutch

4.4.2. Modifications to friction material

The friction material adopted at this time has highly heat resistant fibers added in comparison with the conventional friction material, and other component proportions were revised. Through these measures, durability was enhanced and at the same time good friction characteristics were maintained.

By adopting this material an even smoother start-off performance was achieved. Fig. 16 is an enlargement of the friction material surfaces, and Fig. 17 compares friction characteristics. There is an improvement in friction characteristics throughout the whole range.

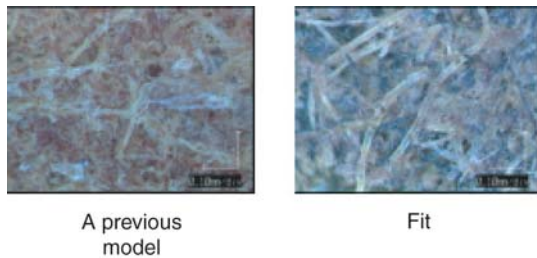


Fig. 16 Magnified disc surface

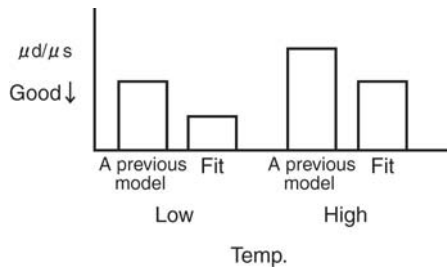


Fig. 17 Disc characteristic

4.5. Creep Control

Creep is an essential function of the automatic transmission vehicles (including CVT), from the point of view of start-off smoothness and ease. On the other hand, because traction force is generated when stopping with brakes, it affects fuel consumption and body vibration. With the new CVT the start clutch creep control is revised, maintaining the creep function only when it is necessary to reduce fuel consumption and body vibration during idling.

When modifying the control it was established that the following driving states require creep when starting off.

- Starting off partway up a hill
- Driving slowly, or starting and stopping repeatedly along in a traffic jam.
- Inching forward in a garage or when parallel parking.

In order to be able to distinguish these situations, the following were established.

- (1) Grade judgment
- (2) Speed history judgment
- (3) Shift position judgment

For conventional inclined road detection methods, a certain amount of speed is necessary (start-off clutch completely tightened), so at low speeds judgment is not possible. With this new control, transmitted torque is calculated from start-off clutch control pressure, and by assuming the clutch slip range drive power it is possible to make a grade judgment even in the low speed range.

Fig. 18 is a flow chart of grade calculation.

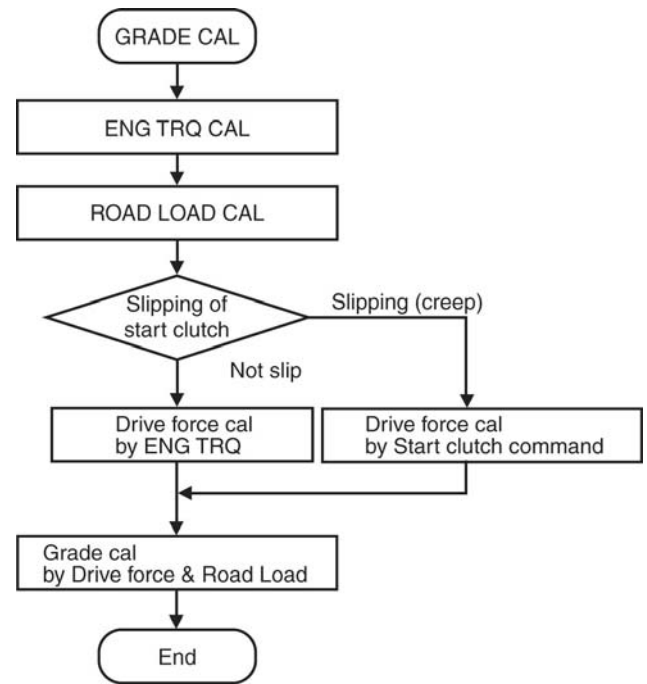


Fig. 18 Flow chart of grade calculation

As outlined above, by appropriately distinguishing the condition that needs creep, it became possible to reduce fuel consumption and body vibration during idling, while maintaining creep performance.

4.6. Shift Control

4.6.1. Enhancements to shift responsiveness

The Fit uses the same pulley pressure direct control method⁽¹⁾ as that in the Insight. Drive and driven pulley pressures are controlled separately, and shift speeds can be set freely. Due to this, rapid kick-down response is possible, and from gentle acceleration to sudden acceleration a linear acceleration feeling is actualized, matching the driver's senses. (Fig. 19)

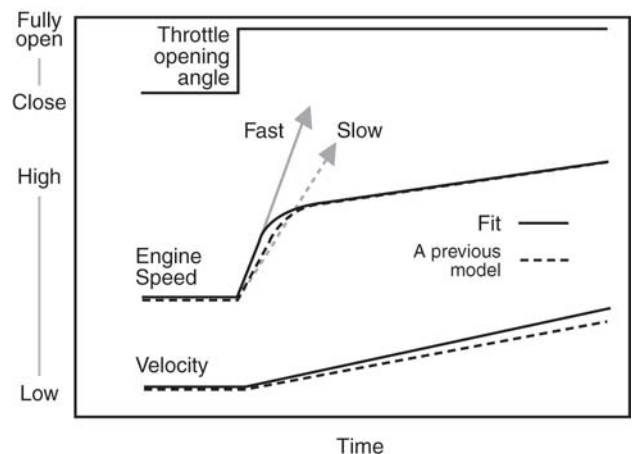


Fig. 19 Kick down response

4.6.2. Shift schedule

The shift schedule in the drive range is established to provide comfortable driving, whether on ordinary roads or joining traffic flow on a highway. However, when driving in the suburbs or on mountain roads, the driver wants to drive with more positive acceleration (aggressive driving), and in this state, it is desired that shift schedule with engine revs is set higher (low ratio side) than the normal drive range shift schedule. Therefore, with the Fit, two shift schedules have been established in the drive range — D1 and D2. Controls have been adopted that automatically switch between them, according to the driver's driving indication.

(1) D1 Schedule

With a shift schedule that focuses on fuel consumption, the throttle opening is in the low to medium range, engine revs are restricted, and the vehicle is running mainly on engine torque.

In the medium range of throttle opening and higher, responsiveness is maintained using the kick-down mode where shifting is quick.

(2) D2 Schedule

Rev speed in the medium throttle opening range is set higher than the D1 schedule to increase drive power, while in the low throttle opening range, the difference from D1 is set small to prevent throttle operation busyness and fuel economy drop due to excessive effectiveness of engine brake during cruising.

Fig. 20 shows a comparison between D1 and D2 shift schedules.

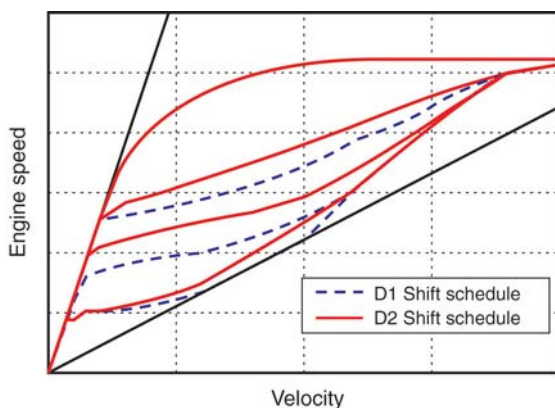


Fig. 20 D1 and D2 shift schedule

4.6.3. Automatic switching between shift schedules

Fig. 21 is a concept diagram showing the relationship between driver driving intent and throttle opening.

When the driver is driving along in a normal flow of traffic, the state is like that in the range at the bottom right of the diagram with low throttle opening. In contrast, when the driver frequently accelerates and decelerates, for example on winding mountain roads, the throttle operation state jumps to the top left range in the diagram.

Fig. 22 shows actual measurements of vehicle speed and throttle opening, as well as the relationship with shift schedule.

The circle on the bottom left is around normal city driving distribution, and the D1 shift schedule is selected.

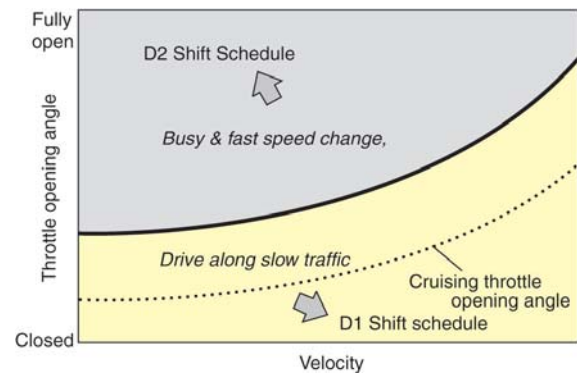


Fig. 21 Condition of shift schedule

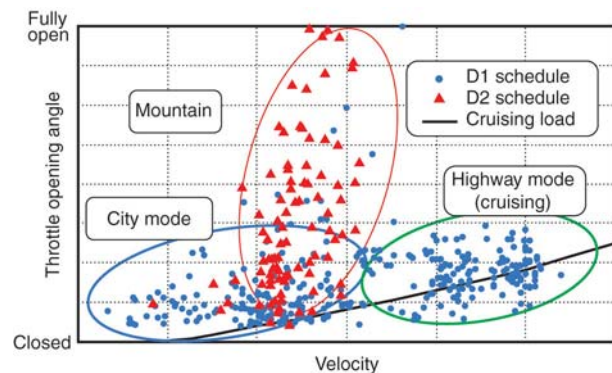


Fig. 22 Throttle distribution chart by shift schedule

The circle on the bottom right is around normal highway driving, as when there is driving with the traffic flow and no overtaking, then the D1 shift schedule is selected.

The circle in the center encloses the distribution of driving on mountain slopes, and because the throttle opening when stepping on the acceleration is greater than on a flat surface, the D2 schedule is selected.

As above, because the shift schedule automatically switches in response to the driver's driving pattern, in comparison with when it doesn't switch, a drive feeling with little uneasiness is obtained.

4.7. Efficiency Improvements⁽²⁾

Loss torque, causing a drop in transmission efficiency, involves resistance of metal belt rotation in drive and driven pulleys, oil pump drive torque, oil agitation resistance, etc., and measures were taken to reduce these factors.

4.7.1. Metal belt rotation resistance reduction

With a metal belt, there is friction between layered rings and elements due to the distance between rocking edges and saddles, and rotation resistance generated due to bend load, while rotating in drive and driven pulleys. As a result of optimum design of transmission capacity in line with engine output characteristics, it became possible to change the ring spec from the conventional 12 rings in 2 rows to 9 rings in 2 rows, thereby reducing belt transmission loss.

4.7.2. Oil pump load reduction due to line pressure reduction

The previous model's hydraulic control involved a linear

solenoid valve creating two pulley side pressures of high and low, and with a four-way valve, the high-pressure side level is adjusted, and shifting control is carried out. However, because this method always generates the maximum pressure necessary to shift normally for the minimum pulley required pressure, when the ratio is stable, such as during cruise driving, there is excess pressure, and the oil pump load is increased. With this new CVT, the drive and driven pulley pressures are separately and directly controlled by linear solenoid valves, making it possible to freely control pulley pressure in response to shift demands, and making it possible to reduce excess pressure during cruise driving. In addition, by adopting this control method, it is possible to have the drive and the driven pulley piston diameters be different, enlarging the piston surface area on the drive pulley side which in most case is the high pressure side when driving (1.3 times that of the driven side), so line pressure can be reduced even further.

Fig. 23 shows the pulley side pressure direct control system.

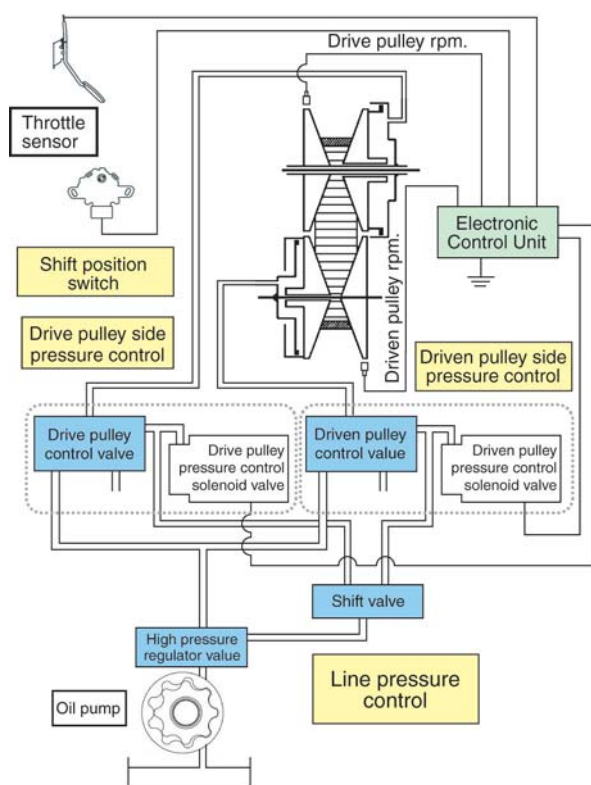


Fig. 23 Pulley side-pressure direct control

4.7.3. Reduction in oil agitation resistance

Agitation resistance is generated by contact between rotating parts and oil. The smaller the oil level, the less the agitation resistance.

As outlined in section 4.3, the oil level is reduced by adopting the new casing structure without an oil pan. In order to reduce it further, low oil temperature aeration measures were taken.

When operating at low oil temperatures, oil viscosity is increased, so there is a delay in return of the oil supplied to each lubricating part and the oil dispersed by agitation. Therefore, the oil level drop is even larger. Because of this, establishing oil levels is carried out according to a low temperature aeration limit, and in the opposite way, at high oil temperatures oil level rises and is a cause of agitation resistance increase. Therefore, a lubrication pressure switching control was added to reduce the lubrication pressure at low temperatures, thereby reducing the amount of oil supplied to each lubricating part, and that directly recirculates any excess to the oil pump. This results in reducing the amount of oil taken in from the strainer at low temperatures, thereby reducing the drop in oil level. Thanks to these measures, it was possible to set the oil level for filling 12% lower (filling amount 8% lower), at the drive pulley shaft center standard.

Agitation resistance is reduced due to this reduction in oil level, and due to the elimination of the oil pump drive-chain.

As a result of reduction of the 3 loss torques as described in sections 4.7.1 to 4.7.3, there is a 5% transmission efficiency improvement at 60 km/h cruising.

Fig. 24 shows a transmission efficiency improvement.

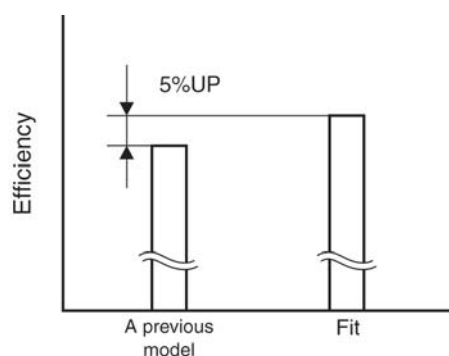


Fig. 24 Cruise efficiency at 60 km/h

5. Summary

A new CVT necessary to actualize the Fit concept was developed, with the following results.

- (1) A design more compact than the previous model, incorporating a structure with no oil pan and a changed control system layout, which resulted in a reduction of 6.2% in overall length.
- (2) By reducing agitation resistance, oil pump drive loss, and operation hydraulic pressure, transmission efficiency at 60 km/h was increased by 5% in comparison with the previous model. In addition, new creep control was developed to contribute to fuel consumption reduction.
- (3) By enhancing shift responsiveness and introducing automatic switching of shift schedules, both fuel economy and linear acceleration performance that can respond to various situations were achieved.
- (4) By adopting a structure without an oil pan, a single mass flywheel, and a direct driven pump, etc., there is a 7.3% reduction in weight, and a 15% reduction in the number of parts in comparison with the previous model.

6. In Conclusion

The new compact CVT developed in this project, the “New Honda Multimatic S”, is the basic design of a transmission that can handle not only 1.3L class vehicles, but also those of even greater capacity. In the future, while expanding its application to more models, research will be continued toward even higher efficiency.

Finally, the authors would like to extend their sincere thanks and appreciation for the cooperation and support they received in the development of this transmission, from members of the development team and all of those at Honda.

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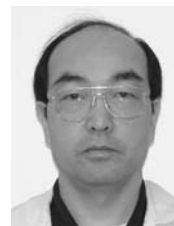
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