

Development of New CVT for K-car

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ABSTRACT

A continuously variable transmission (CVT) adopting a parallel-shaft, helical gear primary reduction mechanism for greater transmission efficiency was developed for K-car use. The distance between the input shaft and output shaft in the front-to-back direction was reduced and the hydraulic control system was positioned all together below the transmission. This supported the reduction of engine compartment size and, by extension, contributed to the extended length of the K-car cabin interior.

Two types of 0.66-L engine, naturally aspirated (NA) and turbocharged (TC), are supported, as well as two-wheel drive (2WD) and four-wheel drive (4WD) systems, with a minimum configuration of special parts. Idle stop was also supported with just the addition of an electric oil pump.

1. Introduction

K-car fuel-saving performance has become a major focus of attention in recent years. An enhanced level of fuel consumption is also being demanded of Honda K-cars, and a new model CVT (hereafter, the present CVT) was developed for the K-car.

This CVT is to replace the lightweight, reasonably priced four-speed automatic transmission (AT)⁽¹⁾ being used in Honda K-cars, and so a lightweight, compact structure was adopted. The previous CVT had the drive pulley positioned coaxially with the input shaft, so the separation between the input shaft and the output shaft where the differential assembly is located could not be reduced. It was reasoned that the adoption of a parallel-shaft, helical gear primary reduction mechanism would allow the drive pulley to be located above the input shaft, in which case the distance between the input shaft and output shaft could be reduced.

The adoption of a primary reduction mechanism lowers the rotational speed of the CVT's metal V-belt and increases the transmitting torque. It was reasoned that setting an optimal primary reduction ratio would enhance the transmission efficiency of the CVT.

The previous CVT had a high gear-meshing frequency, and therefore required high machining accuracy for the gears, careful attention to the shaft supports, and other such considerations. However, the present CVT has adopted a primary reduction mechanism with parallel shafts and helical gears. This allows a lower gear-meshing frequency than other mechanisms, and this was therefore considered an advantageous structure for the quietness of the cabin.

The NA and TC engine types could be supported by

differentiating specifications for torque converter and primary reduction gear, the 2WD and 4WD drive system types by differentiating specifications for the torque converter case (T/C case) and the differential case, and the idle stop specification just by the addition of an electric oil pump, with a minimal configuration of spatial parts.

2. Development Goals

The development of a new CVT for use with K-cars was carried out with the following items as the development goals:

- (1) Light weight and compactness at the top of its class;
- (2) Transmission efficiency at the top of its class;
- (3) Lower gear-meshing frequency in order to enhance quietness of the cabin; and
- (4) Commonization of parts
 - Support NA and TC with the same construction
 - Support 4WD with minimum parts change
 - Support idle stop with just the addition of an electric oil pump.

3. Basic Layout and Main Specifications

A main section view and a perspective external view of the present CVT are shown in Fig. 1 and Fig. 2. The major specifications are shown in Table 1.

The basic layout of the present CVT has the following features:

- (1) A primary reduction mechanism and forward-reverse switching mechanism using helical gears between parallel shafts were adopted and set upstream of the main transmission mechanism (continuously variable

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transmission mechanism using a belt and pulley). These together eliminated any secondary reduction mechanism. Torque flows from the input shaft, to the drive pulley shaft, then to the driven pulley shaft, and finally to the differential shaft, in a four-shaft structure.

- (2) Parts for the NA and TC engine types downstream from the main transmission mechanism were commonized by adjusting the fluid characteristics of the torque converter and the primary reduction ratio.
- (3) A shaft arrangement that supports reduction in the front-to-back length of the engine compartment was adopted.

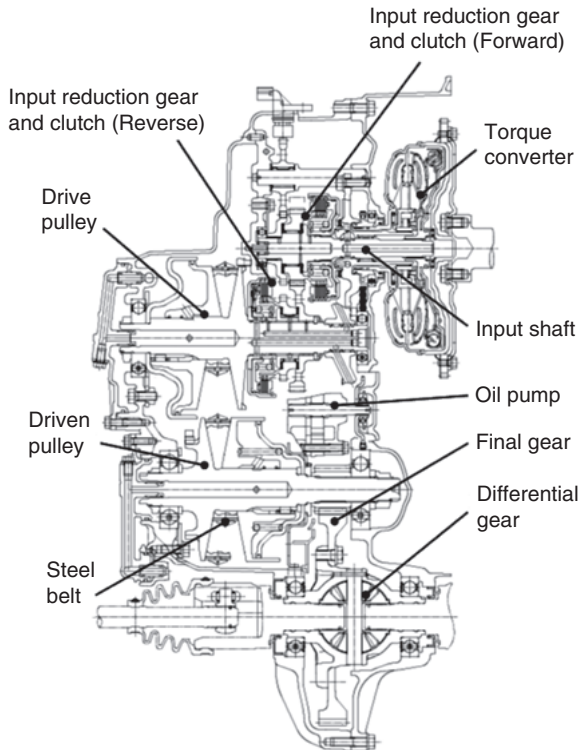


Fig. 1 Main section of developed CVT

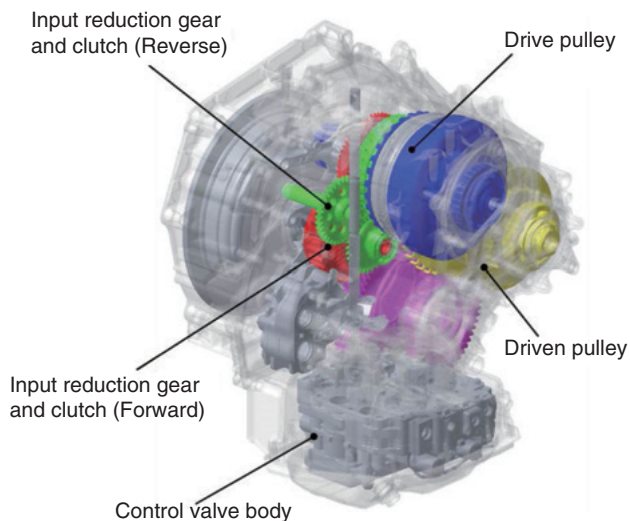


Fig. 2 Perspective view of developed CVT

Table 1 Specifications of developed CVT

Engine specifications	Type		0.66 L NA	0.66 L TC
	Max. power [kW/rpm]		43 /7300	47 /6000
	Max. torque [Nm/rpm]		65 /3500	104 /2600
Ratio	Primary reduction	Forward	1.605	1.375
		Reverse	1.187	
	Pulley (low-overdrive)		2.293-0.420	
	Final		5.176	
Dimensions	Overall length [mm]		348	
	Shaft axis distance [mm]	Input /Drive	87	
		Drive /Driven	147	
		Driven /Final	123.5	
Weight (wet) {Idle stop}	[kg]		58.5 {59.6}	58.6

- (4) The case was made of a simple two-piece construction.
- (5) 4WD was supported by adopting different T/C case and differential specifications for 4WD.
- (6) The hydraulic control system was placed all together under the transmission, and hydraulic circuits capable of supporting 4WD were adopted.
- (7) Idle stop capability was supported merely by adding an electric oil pump.

4. Main Technologies

4.1. Parallel-shaft, Helical Gear Primary Reduction Mechanism

For the present development of a new CVT for K-car use, the development of a new metal V-belt could also be considered. This would have involved elimination of primary reduction and development of a metal V-belt matched to the maximum power of the K-car with the width reduced and the circumferential length adjusted to the necessary ratio range. On the other hand, common use of the previous metal V-belt with a 24-mm width would provide an extra margin of allowable input torque that would enable use with primary reduction and with increased input torque to the metal V-belt. The choice was made to take the previous 24-mm wide metal V-belt for use in the present CVT.

A primary reduction mechanism using helical gears for location between parallel shafts was placed upstream of the continuously variable transmission mechanism operated using a metal V-belt. The forward gear set and clutch were placed inversely with the reverse gear set and clutch for a compact construction. The reverse clutch was positioned on the driven side of the parallel shafts so as to halve the differential rotation within the clutch and reduce friction. That structure is shown in Fig. 3.

In general, the metal V-belt has the greatest transmission efficiency when operated close to the belt's allowable

input torque. As the input speed increases, the transmission efficiency tends to diminish.

The rate of fuel consumption enhancement in the NA engine due to the difference in primary reduction ratio was simulated by taking fuel consumption in 10-15 mode and without primary reduction (primary reduction ratio of 1.0) as the base. Figure 4 shows the results. In the NA engine, a 1% or better enhancement in fuel consumption was obtained with primary reduction ratios from 1.45 to 2.0.

When the primary reduction ratio is higher, the torque passing through the belt increases. When the torque passing through the belt is greater, an axial thrust matched to the torque becomes necessary, and a larger hydraulic pressure is required by the system. The relationship between the primary reduction ratio and system-required pressure is shown by the red lines in Fig. 5 and Fig. 6. Increasing the pressure generated by the oil pump also allows the primary reduction ratio to be higher. However, this also means the various parts need greater capability to resist pressure. Considering such factors and the capacity of the oil pump, it was decided that the maximum pressure generated in the present CVT should be 5 MPa.

The elimination of a secondary reduction mechanism also means that the overall gear ratio is determined by the product of the stepless gear ratio of the metal V-belt and

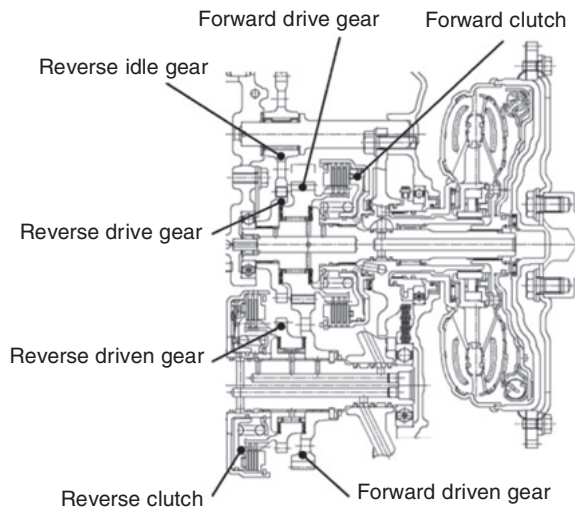


Fig. 3 Structure of helical gear input reduction

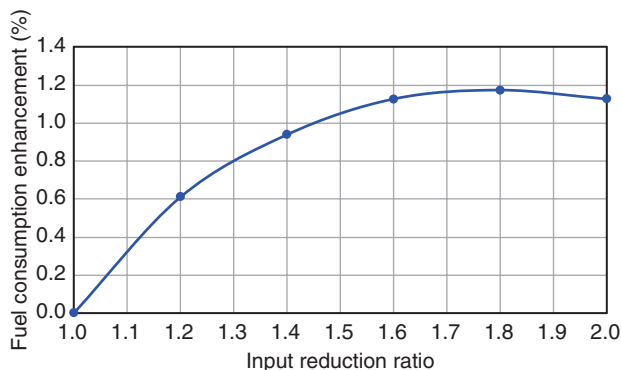


Fig. 4 Input reduction effect for fuel consumption of NA

the primary reduction ratio and the final reduction ratio. When optimal overall gear ratios are respectively set for the NA and TC, the final reduction ratio is determined by the primary reduction ratio. The relationship between the primary reduction ratio and final reduction ratio is shown by the blue lines in Fig. 5 and Fig. 6. Limitations on the front-to-back length of the transmission place constraints on the maximum value that can be obtained for the final reduction ratio (the maximum diameter of the final gear), and as a result there are constraints on the primary reduction ratio that can be obtained. In the present CVT, the upper limit on the final reduction ratio is 5.5.

Consequently, the primary reduction ratios within the shaded regions of Fig. 5 and Fig. 6 can be set. Since the gear ratio for a parallel-shaft, helical gear primary reduction mechanism can be readily set, separate primary reduction figures of 1.605 for the NA and 1.375 for the TC were adopted. The product of this primary reduction ratio, the torque generated during a stall in the NA and TC, and the torque ratio of the respective torque converters is the maximum torque that is input to the belt transmission mechanism. Setting the maximum input torque to the belt transmission mechanism more or less equal for the NA and TC specifications enabled their use of common parts downstream from the belt transmission mechanism.

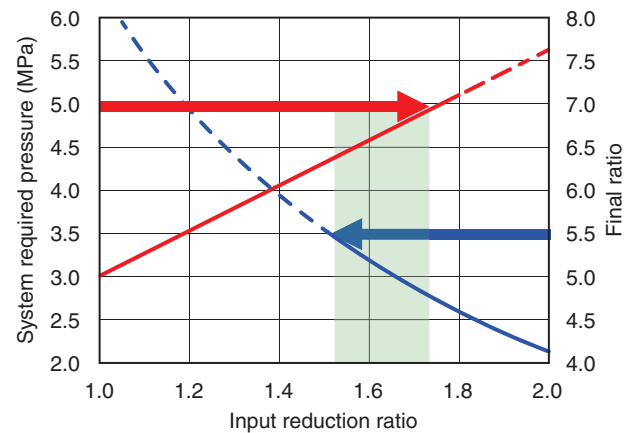


Fig. 5 Input reduction ratio for NA

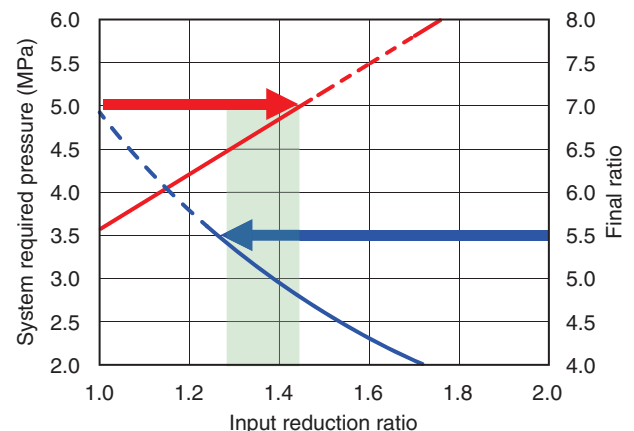


Fig. 6 Input reduction ratio for TC

4.2. Gear-Meshing Frequency Reduction

In general, the noise level of the cabin of a vehicle has greater sound pressure on the low-frequency side, and lower pressure on the high-frequency side of the sound. Moreover, if gear sounds are at or below the sound pressure of the noise level of the cabin, then they will not be audible. When the meshing frequency of the gears is high, then there is a need to lower the level of the gear sound pressure, and it will be necessary to give consideration to the machining accuracy of the gears and to gear support stiffness. In the previous CVT, torque flows from the belt-pulley transmission mechanism, to secondary reduction, and to final reduction. When in overdrive, the belt-pulley transmission mechanism speed rises and then slows down so that the gear-meshing frequency of the secondary reduction gear rises higher, and this means that gears with higher-precision tooth profiles will be required. The present CVT adopts a primary reduction mechanism and eliminates the secondary reduction mechanism, thereby holding the gear-meshing frequency down to a low level. This structure is advantageous for the quietness of the cabin. The previous CVT also adopted a planetary gear mechanism for the forward-reverse switching mechanism, so that the planetary gear mechanism would operate when the vehicle was driven in reverse. At that time, the planetary gear would have a high gear-meshing frequency, so it was also necessary to consider such measures as enhancing the precision of the gear tooth profile in order to lower the gear sound pressure. The present CVT also has helical gears between parallel shafts configured for use in reverse. This structure provides lower gear-meshing frequency and also allows the gear tooth profile to be adjusted readily.

Gear-meshing frequencies were calculated with the ratios for use with a K-car and on the assumption of a CVT (in the conventional arrangement) equipped with a secondary reduction mechanism and of a CVT (planetary gear-type primary reduction) that would perform primary reduction using a planetary gear mechanism. The results are shown in Fig. 7. When cruising in forward gear at 60 km/h, the present CVT can hold the gear-meshing frequency to a level 1000 Hz or more lower than the conventional arrangement CVT and the planetary gear primary reduction CVT.

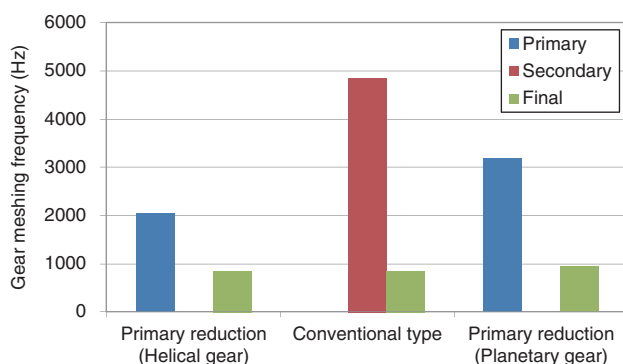


Fig. 7 Gear meshing frequency at 60 km/h cruise

Figure 8 similarly shows the gear-meshing frequency when moving in reverse at 20 km/h. The primary reduction mechanism of the present CVT has a gear-meshing frequency at a level one-half or lower than the CVT that activates a planetary gear mechanism.

4.3. Case Construction

4.3.1. Shaft arrangement

In order to obtain as much cabin space as feasible within the limited space of a K-car, the dimensions of the input shaft and output shaft in the front-to-back direction were brought as close together as could be managed. The front-to-back length of the transmission was made 4 mm shorter than that of the previous four-speed AT⁽¹⁾ for K-car use. The hydraulic pressure control solenoid was also moved from in front of the transmission to a position below it, and the profile along the outside diameter was matched to the dashboard profile. This pushed back the amount of engine travel (stroke) during a collision and contributed to a 70-mm increase in the interior length of the cabin. A side view of the present CVT is shown in Fig. 9.

4.3.2. Two-piece case construction

A structure that supports the drive and driven pulley shafts at both ends was adopted, and a two-piece basic case structure was adopted, with a T/C case and a transmission

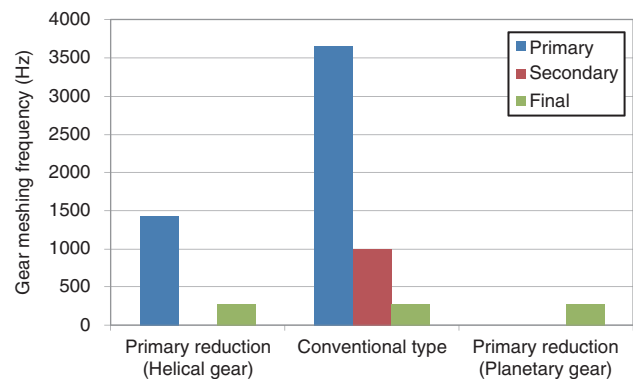


Fig. 8 Gear meshing frequency at reverse 20 km/h

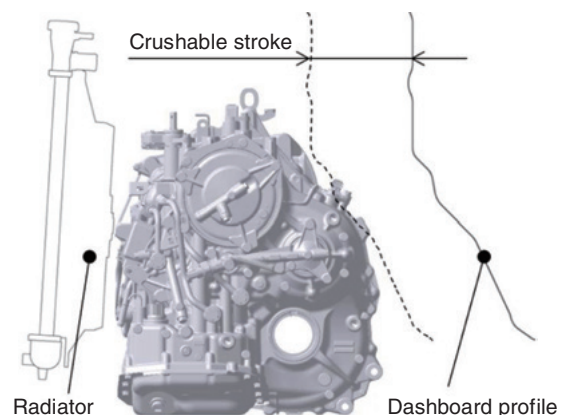


Fig. 9 Side view of developed CVT

case (T/M case). The partition between the cases and the flange mating surfaces could be reduced, realizing a reduction in weight. The case construction is shown in Fig. 10.

4.3.3. 4WD support

Here, a transfer was developed that is coaxial with the differential and that connects the differential case and the transfer drive gear with a spline. The only special 4WD parts in the CVT are the T/C case and the differential case. The differences between the specifications for the 2WD T/C case and the 4WD T/C case were minimized, and the machining and casting plans for the manufacturing process were commonized, making the structure advantageous in terms of manufacturing cost as well as quality. That configuration is shown in Fig. 11.

4.3.4. Formed-in-place gasket

The paper gasket used before was replaced with a

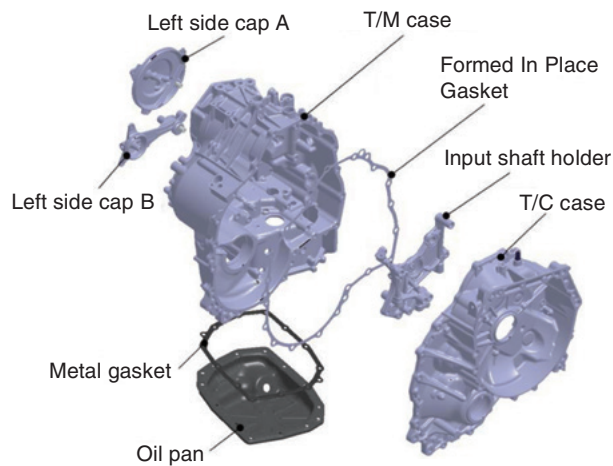


Fig. 10 Case construction

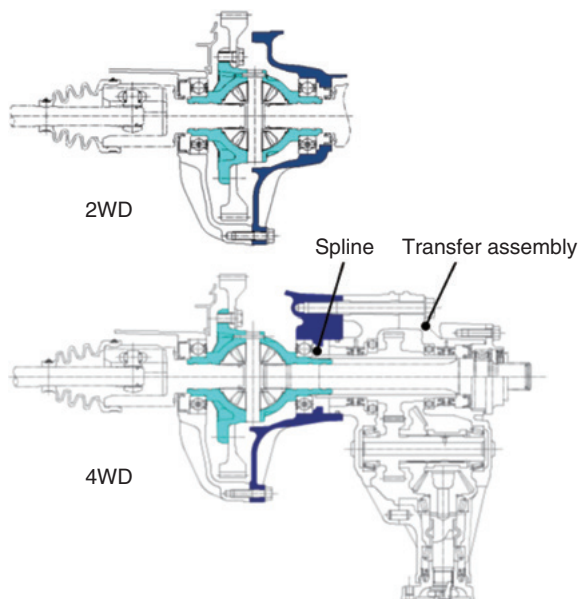


Fig. 11 Configuration of transfer coaxial with differential

formed-in-place gasket (FIPG) for the CVT in order to reduce the case weight and lower its cost. The elution of silicone from the FIPG into the CVT fluid has raised issues of reduced metal friction coefficient μ and diminished defoaming property of the CVT fluid. The FIPG components were therefore reexamined and a new gasket material was developed that would not affect the CVT fluid. Adoption of the FIPG enabled the gasket to follow tensile relative movement and lateral relative movement at the case mating surfaces. This allowed the design to shift from the previous paper gasket, which depended on bolt axial force, to instead optimize tightening stiffness. This enabled the use of shorter bolts as well as reduction in the number of bolts used. This achieved a reduction of 1100 g in weight compared to when the paper gasket was used, and also yielded a reduction in cost.

4.3.5. Metal gasket

In the interest of maintainability and compactness, the hydraulic control system consisting of the valve body and other such elements was positioned together under the transmission. A metal gasket was adopted for use as a seal between the steel-sheet oil pan and the aluminum case. This enabled use of a steel-sheet oil pan, which is lower in cost than an aluminum oil pan. This configuration is shown in Fig. 12.

4.4. Hydraulic Control System

A reasonably priced hydraulic circuit structure with a smaller number of solenoids was adopted for the hydraulic control system. A new control that uses the forward gear clutch to cut off sudden excessive input torque was also adopted. An electric oil pump was additionally configured to support idle stop.

4.4.1. Hydraulic circuit

The previous hydraulic circuit was configured from three linear solenoids and two on-off solenoids. One of those linear solenoids was switched back and forth between use for forward-and-reverse clutch control and use for lock-

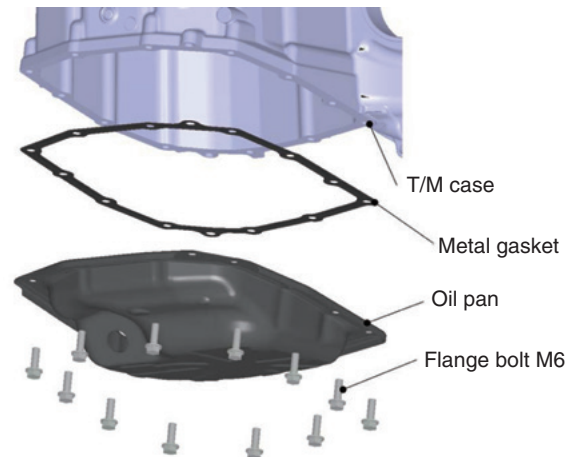


Fig. 12 Metal gasket of oil pan

up clutch control. The circuit was therefore configured so that, while it was being used to control lock-up clutch capacity, it could not control forward clutch capacity. The previous hydraulic circuit is shown in Fig. 13.

The hydraulic circuit for the present CVT controls forward clutch capacity and lock-up clutch capacity independently. A special linear solenoid was therefore set up for each. The linear solenoid for the forward clutch constantly controls forward clutch capacity so it will be at or below belt transmission capacity. The lock-up clutch control is configured for the capability to switch the lock-up clutch on-off circuit and to control the lock-up clutch capacity within the pressure control range of the linear solenoid. The switching solenoids have been eliminated.

A direct control system has been adopted for the pulley pressure control circuit, which follows the configuration of the previous control circuit. It uses two linear solenoid valves to perform separate control of the drive and driven pulley pressure while constantly keeping line pressure at the minimum necessary level. The above configuration achieved a balance between enhancing its functionality and making it more compact and lighter in weight. Figure 14 shows the hydraulic circuit of the present CVT.

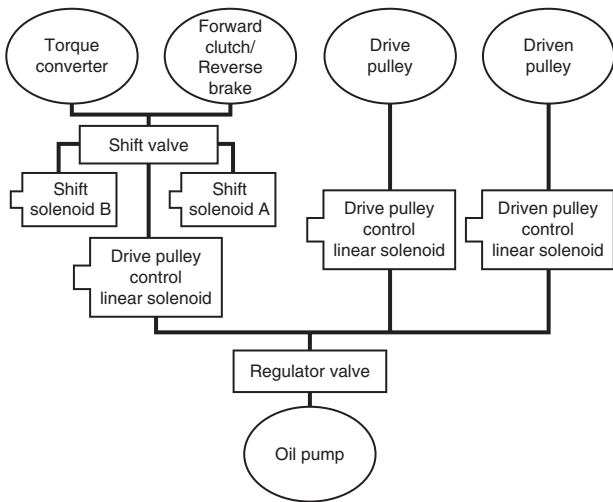


Fig. 13 Hydraulic circuit of previous CVT

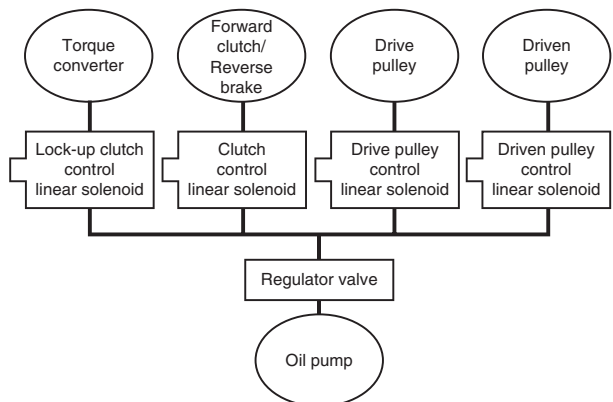


Fig. 14 Hydraulic circuit of developed CVT

4.4.2. Control for 4WD support

The adoption of constant clutch capacity control eliminated the difference between 2WD and 4WD belt input torque and allowed the use of common component parts. A new torque-on-demand control at launch was also adopted to comply with the rear differential input torque characteristics, achieving smooth launch characteristics.

4.4.3. Control for idle stop support

With the adoption of an electric oil pump, the engagement of the forward clutch is not completely prevented even during idle stop. This achieved a highly responsive and smooth launch when restarting. Figure 15 shows launch responsiveness from idle stop.

4.5. Increasing Efficiency

In addition to optimizing the primary reduction ratio, each part of the CVT was reexamined with a view to further enhancement of efficiency.

4.5.1. Reduction of oil agitation resistance

The adoption of the primary reduction mechanism resulted in the drive pulley being located above the oil level, and reduced the agitation resistance from rotation of the pulley. This was combined with adoption of a baffle plate to reduce the agitation resistance of the large-diameter final driven gear.

4.5.2. Oil pressure sensor

An oil pressure sensor was placed in the oil pressure control system to constantly sense the CVT working oil pressure, and learning control that provides feedback was adopted. This enabled reduction of the pressure margin and also lowered the oil pump drive torque and belt friction.

4.5.3. Cruising efficiency results

Figure 16 shows a comparison of the cruising efficiency of the present CVT and the CVT of other vehicles. An advantage of about 2% was obtained at vehicle speeds from 40 km/h to 100 km/h.

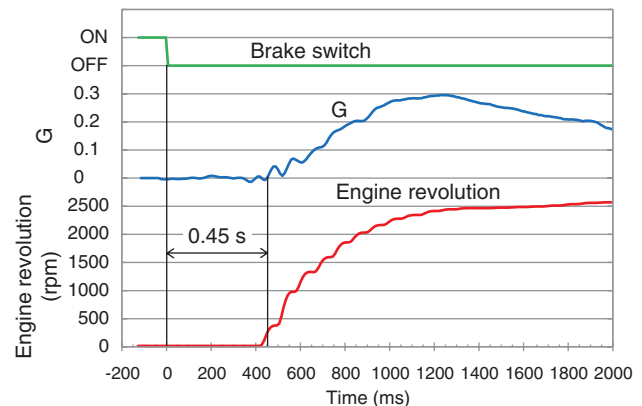


Fig. 15 Response time from idle stop

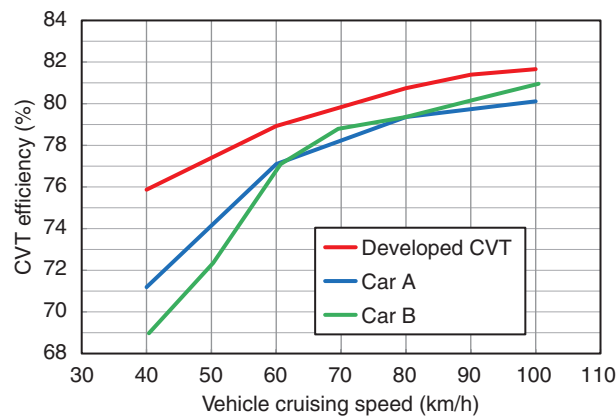


Fig. 16 CVT efficiency at cruise

5. Conclusion

A new model CVT was developed for K-car use and yielded the following results:

- (1) The adoption of a parallel-shaft, helical gear primary reduction mechanism achieved a CVT weight in the top class for CVTs in the 110-Nm allowable input torque class.
- (2) Optimal primary reduction ratio settings achieved cruising transmission efficiency in the top class for a K-car CVT.
- (3) Compared to the previous CVT, the gear-meshing frequency was lowered by 1000 Hz or more when cruising at 60 km/h, enhancing the quietness of the cabin.
- (4) The specifications for the torque converter and forward gear set were differentiated to support NA and TC engine versions. Support for 4WD was provided with the T/C case, the differential case for 4WD use, and a special transfer. Idle stop was supported with just the addition of an electric oil pump.

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