

Development of a New Manual Transmission for a 3-Liter Class Vehicle

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ABSTRACT

A new 6-speed manual transmission for 3 L class engines has been developed as Honda's top-of-the-line manual transmission. To be mounted on different models, this transmission was given the capability to employ a wide range of gear ratios and to handle high torque while keeping its total length equal to transmissions for 2 L class. Computer aided engineering (CAE) analysis was used to enhance the gear stiffness in order to cope with higher power outputs. The inside surface of the case was made smooth and a force feed lubrication system driven by an oil pump has been used to reduce oil agitation loss in order to curb a rise in oil temperature at high travel speeds. Much emphasis was put on reducing the weight of each component, which resulted in a 3 L class transmission with the weight equal to 2 L class. In order to realize the level of quality appropriate for our top-of-the-line vehicles, the transmission has been equipped with a constant-mesh reverse gear and a dual mass flywheel.

1. Development Concept

In North America, in recent years, there has been an increasing demand for manual transmission vehicles with 3 L displacement engines. Honda R&D has, in this project, developed a new 6-speed manual transmission (referred to as 6MT hereafter), to meet the following targets.

- (1) Applicability to multiple models
- (2) Have measures to handle high output and high torque
- (3) While being in the 3 L class, actualize weight that is equivalent to that of the 2 L class
- (4) Achieve high quality appropriate to luxury vehicles

This report introduces the major technologies used to achieve these targets.

2. Basic Structure

The gear train is a parallel dual shaft type with constant

mesh type synchro-mesh structure from 1st to 6th and reverse gears. In addition, the case is a 2-piece structure – clutch case and transmission case. The major specifications of the 6MT developed in this project are given in Table 1, and Fig. 1 shows a vertical cross section diagram.

3. Applicability to Multiple Models

3.1. Change Control System

The change control system uses a cable control format. Its location is above the transmission giving more freedom to the layout of surrounding parts, such as the starter and wiring, and to fitting to the engine and vehicle body. In addition, in order to make wire layout easier, the structure is such that shift and selector cables are almost parallel making retrieval possible. Figure 2 shows the layout of the change control system.

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Table 1 Specification of transmission

Transmission type		Manual 6-speed
Maximum speed (rpm)		7 000
Transmissible torque (Nm)		317 over
Gear ratio	1st	3.933
	2nd	2.478
	3rd	1.700
	4th	1.250
	5th	0.975
	6th	0.770
	Reverse	4.008
	Final	3.285
Weight; Wet (kg)		67

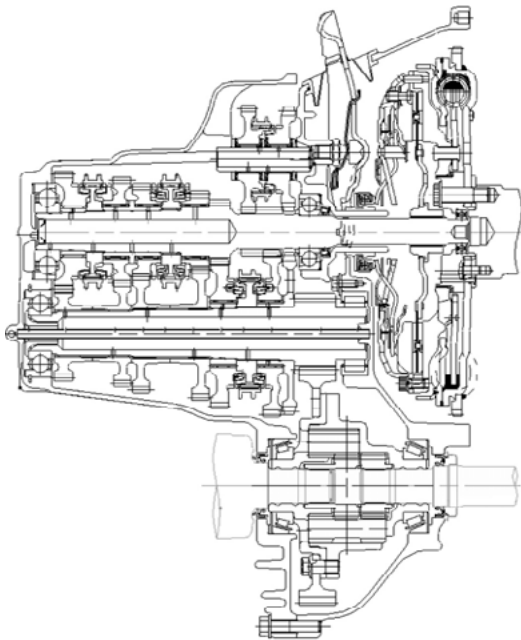


Fig. 1 Main section of transmission

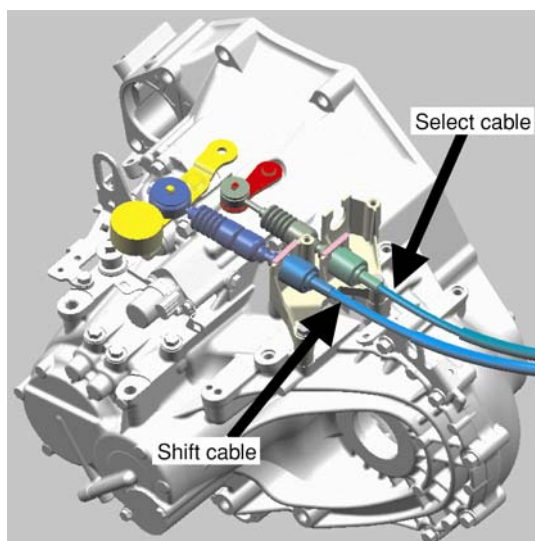


Fig. 2 Change cable layout

3.2. Gear Train

Widening the distance between the main shaft and the countershaft, the setting degrees of freedom are expanded from the sporty series' cross ratio to the sedan series' open ratio. It is possible to set final gear ratio across a wide range of gear ratio settings, from sedan series to SUVs. However, with conventional technology there has been the issue of a large increase in weight when taking measures for this. Therefore, measures have been taken utilizing the technologies described in sections 3.3, 4, and 5, to achieve the targets.

3.3. Shortening of Total Length

Utilizing the following technologies, the total length of the transmission has been shortened equivalent to that used in a 2 L class vehicle, while it is a 3 L class. Figure 3 shows total length relating to engine torque.

3.3.1. Clutch gear embedding structure

As shown in Fig. 4, concerning the counter 1st gear, while maintaining optimum synchronize capacity, the clutch gear which was formed on the conventional gear edge surface is embedded inside of the gear, and so the gear width is reduced. As a result, total length is reduced by 7 mm.

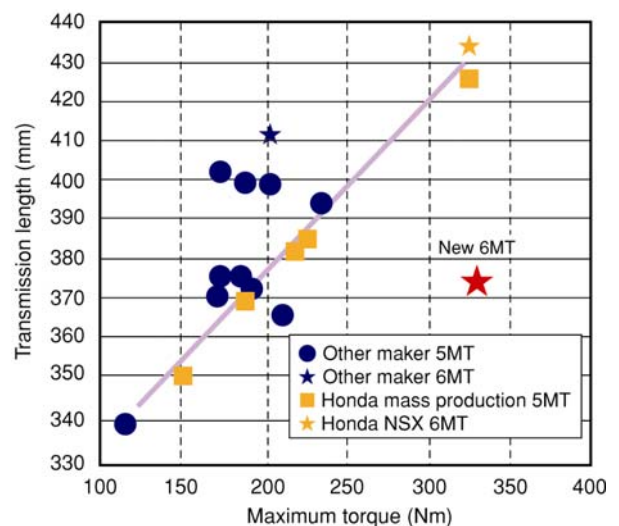


Fig. 3 Comparison of transmission length

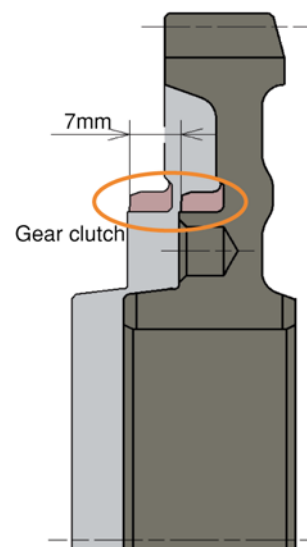


Fig. 4 Counter 1st gear

3.3.2. Self adjusting clutch system

The clutch uses an automatic wear adjustment system (self-adjusting clutch – referred to as SAC hereafter), and the system is designed such that the release bearing does not move back when it becomes worn. As a result, total length is shortened by 9 mm (Fig. 5).

3.3.3. Gear width shortening

Considering the 1st and 2nd gears, it is necessary to balance the issues of reducing tooth stress from impact load input while maintaining gear contact ratio, which greatly contributes to gear noise.

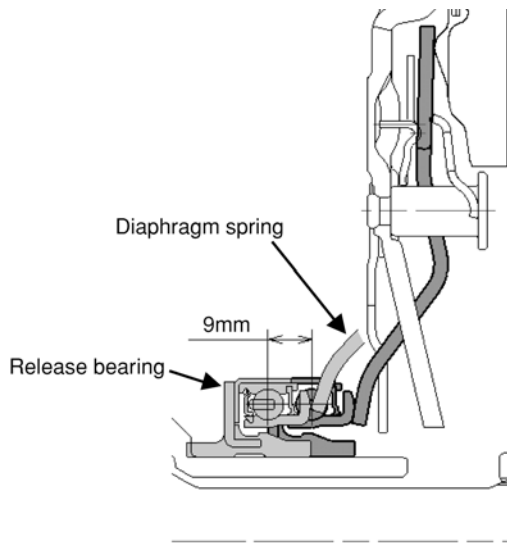


Fig. 5 Clutch

With this 6MT, at the same time as making it more compact, fully considering cost and weight, etc., and maintaining contact ratio with a small module, gear width is shortened by taking measures against high stress due to impact loads, including use of high strength gear material and shot peening treatment.

4. Technology for High Output and High Torque

4.1. Appropriate Case Stiffness

Development of the case was based on the concept of maintaining optimum stiffness and reducing oil agitation loss.

Conventionally, around the differential where high stress is generated, ribs have been created on the inside and outside of the case, thereby maintaining stiffness. In addition, because the differential is positioned below the transmission half in the oil, between the differential part ribs on the inside of the case and the final driven gear and differential, there is loss due to agitation, and in the high-speed range is a major factor in oil temperature rise. Focusing on this, as shown in Fig. 6, by smoothing the shape of the differential part ribs on the inside of the case and adding ribs outside the case, the stiffness required of the case was maintained. In addition, an oil chamber is formed by the ribs on the inside of the case, which also act as reinforcement, reducing oil agitation loss due to the gears.

The case shape was designed through CAE analysis, which allowed smoothing the shape of a differential part rib inside the case. When making the analysis, a clutch case and transmission case 3D model was connected by a simple model bolt, CAE analysis was made of the case as a whole, and the optimum shape designed for stress and stiffness. As a result, while it is a 3 L class transmission, the weight

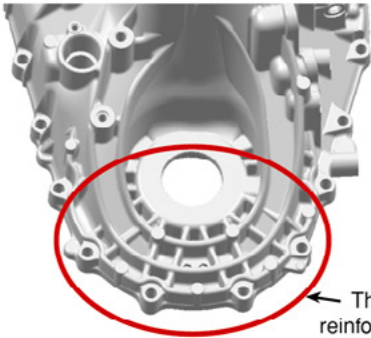
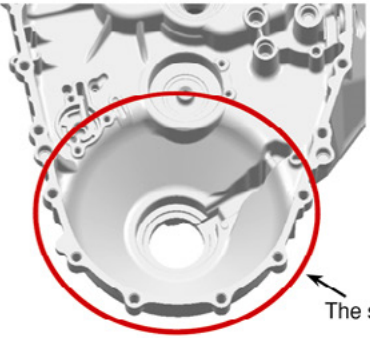
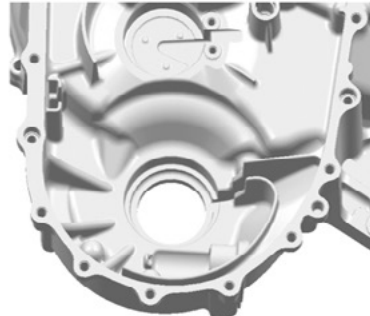
Newly developed model		
Current mass production model		
	Outside view of transmission case	Inside view of clutch case

Fig. 6 Comparison roughness of differential

equivalent to that for a 2 L class vehicle was achieved. Figure 7 gives the analysis results, and Fig. 8 shows the relationship between case weight and engine torque.

4.2. Case Mating Surface Pressure

On the surface in which the clutch and transmission case join together, wear occurs due to the friction accompanying large load input. As a countermeasure to this, using CAE analysis, the surface pressure distribution is made uniform across the case mating surface improving durability. As a result, it is possible to reduce the sealed surface width of the clutch and transmission case making it possible to reduce the case weight together with the benefits described in section 4.1.

4.3. Optimization of Gear Stiffness

In the gears, by extending the distance between gear shafts and increasing input torque from the engine, large moment loads are generated. Therefore, gear part stiffness greatly affects tooth meshing errors and this is a major cause

of tooth strength deterioration. It is possible to optimize gear part shape, fully considering gear meshing error and weight, by using CAE analyses. Figure 9 shows the results of tooth flank surface pressure analysis and the gear shape before and after measures were taken concerning counter 2nd gear. With the gear shape change, there is a reduction in tooth flank part meshing error and tooth flank surface pressure is reduced by 6.6%, thereby improving durability.

4.4. Lubrication

In order to have the appropriate oil distribution and effective oil supply for lubrication, a semi-dry sump structure (using both oil bath lubrication from gear agitation and force feed lubrication from an oil pump) was adopted. Figure 10 shows this lubrication structure.

As explained in section 4.1 on optimization of case stiffness, the oil chamber is created by the ribs of the transmission case. By locating the oil pump suction in this area, heat due to gear agitation and aeration in the high-speed range are prevented.

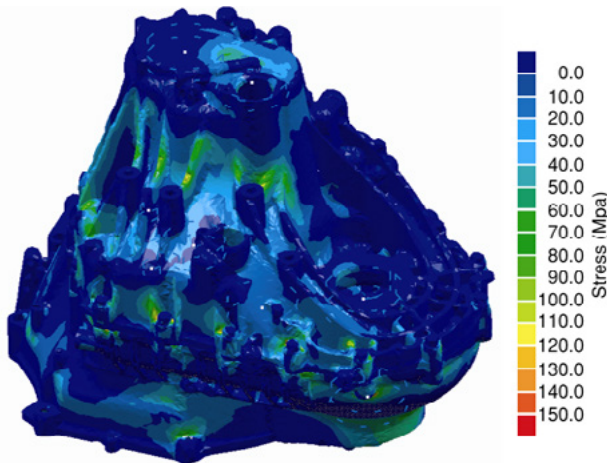


Fig. 7 Transmission case load analysis

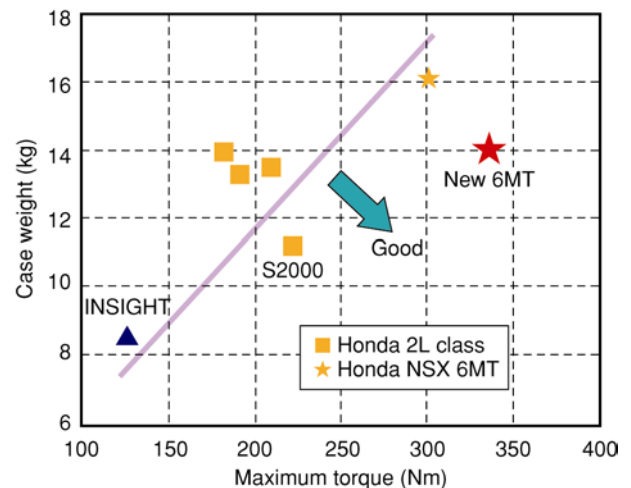


Fig. 8 Comparison of transmission case weight

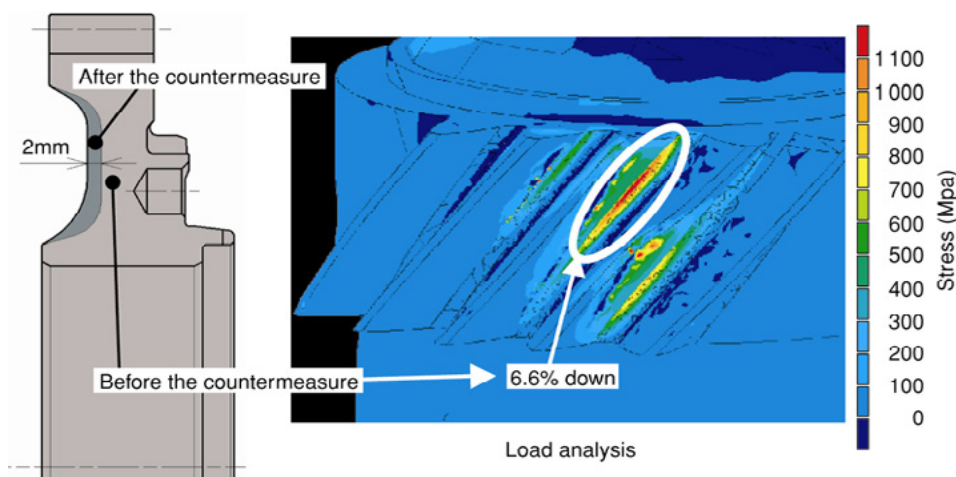


Fig. 9 Counter 2nd gear

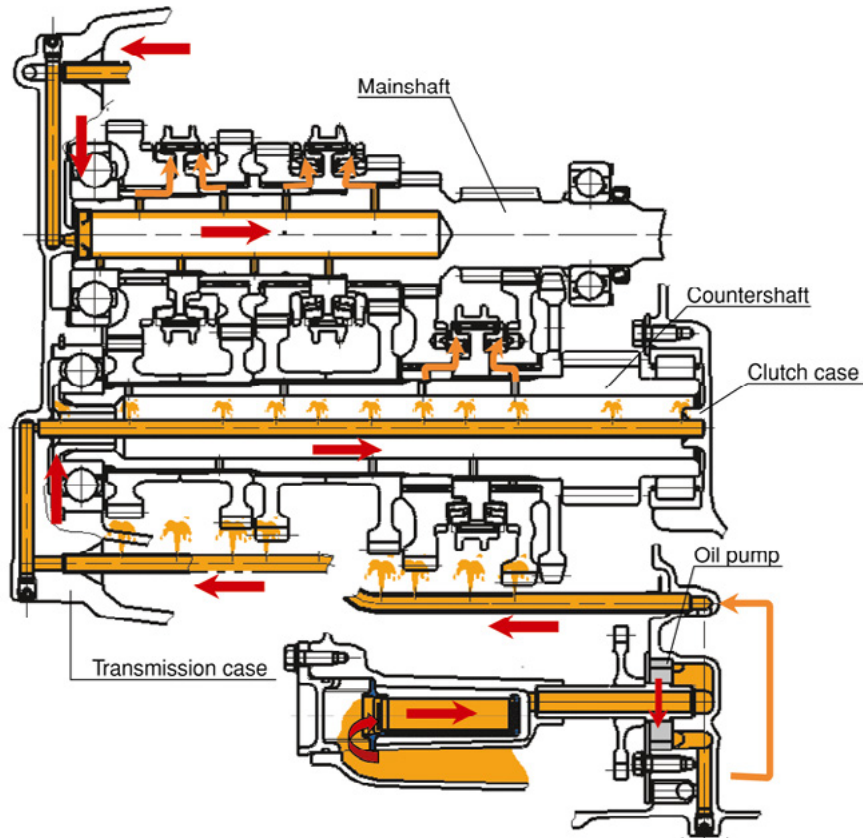


Fig. 10 Oil lubrication structure

In addition, the conventional countershaft lubrication structure is such that because oil is supplied from the edge of the transmission case, due to the effect of centrifugal force at high speeds, there is a large amount of lubricant on the transmission case side, while at the front edge of the clutch case it is difficult to maintain a sufficient amount of lubricant. As a countermeasure, piping is established within the countershaft to supply oil, the appropriate amount of lubricating oil required by the system, from holes in the piping.

Figure 11 shows a comparison of oil temperatures in the high-speed range. With the application of the technologies outlined above, there is a 15 °C drop in temperature, in comparison with conventional designs.

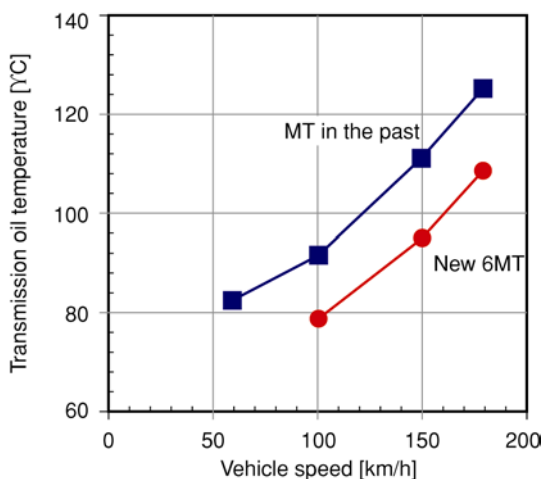


Fig. 11 Comparison of transmission oil temperature

5. Weight Reduction Technology

Together with the weight reduction of the frame part through total length reduction outlined in section 3.3, etc., weight reduction measures were taken for each part. The major technologies are introduced below.

5.1. Counter Reverse Gear Weight Reduction

The radial thickness of the gear flange part is determined by forging production limit dimensions. Furthermore, with hot forging, holes are made in the flange, and it is possible to reduce weight while maintaining the necessary rigidity (Fig. 12).



Fig. 12 Counter reverse gear

5.2. Hollow Countershaft

By making the external diameter of the countershaft large, the shaft's bending stiffness is maintained, and in order to reduce weight, the shaft's center hole diameter, which also acts as a lubricating oil supply path, is also made larger, thereby reducing weight.

5.3. Hollow Shaft and Pressed Shift Fork

A pressed fork was adopted for the shift fork and through CAE analysis the generated stress is equalized creating a lightweight, compact finished product that achieves stiffness balance of a shift fork, which is optimized to be almost symmetric. In addition, for a hollow shaft the welded shaft is used giving weight reduction.

Combining these technologies with frame weight reduction, a weight of a 3 L transmission was kept equivalent to that of a 2 L model. Figure 13 shows transmission weight against engine torque.

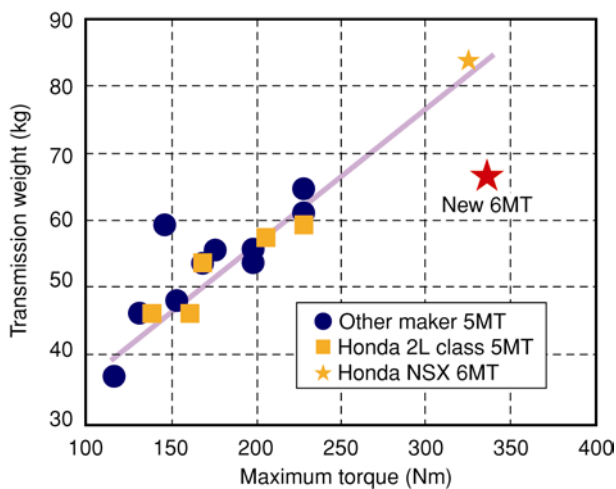


Fig. 13 Comparison of transmission weight

6. Technologies for Luxury

6.1. Shift Feeling

Shift feeling is an important element of a luxury vehicle, and development was carried out with the target being a nimble and smooth shift feeling as a high quality product.

In order to actualize a light shift feeling measures were taken to both shorten shift stroke and to reduce shift load. From 1st through 4th, a multi-cone synchronizer is used; and for 5th and 6th a high friction coefficient synchronizer is used. In addition, in 2nd, the triple-cone synchronizer, the largest developed in our company, is used to give it high capacity. A dual mass flywheel (referred to as DMFW hereafter) has been adopted leading to the elimination of a torsion spring mechanism on the friction disk making it possible to reduce the inertia moment on the synchronization side by 20%. However, increasing the synchronizer capacity (synchronization torque) increases the torsion generated during synchronization on the side being synchronized, and after synchronization is complete, that torsion is released generating a rotational difference. In this state, if the synchronizer sleeve advances when there is engagement with the clutch gear, the synchronizer sleeve bumps (as double bump) due to the rotational difference and the vibration generated is transmitted to the shift knob deteriorating the smooth shift feeling⁽¹⁾.

In order to solve this issue, the small diameter shape of the synchronizer sleeve that contacts the synchronizer spring was modified so that even after synchronization there is synchronization torque being generated by the synchronization spring, though it is very small. By reducing the rotational difference, shift knob vibration is reduced and shift feeling improved. Figure 14 shows a comparison of shift knob load and Fig. 15 shows synchronization operation state.

In addition, with our company's conventional

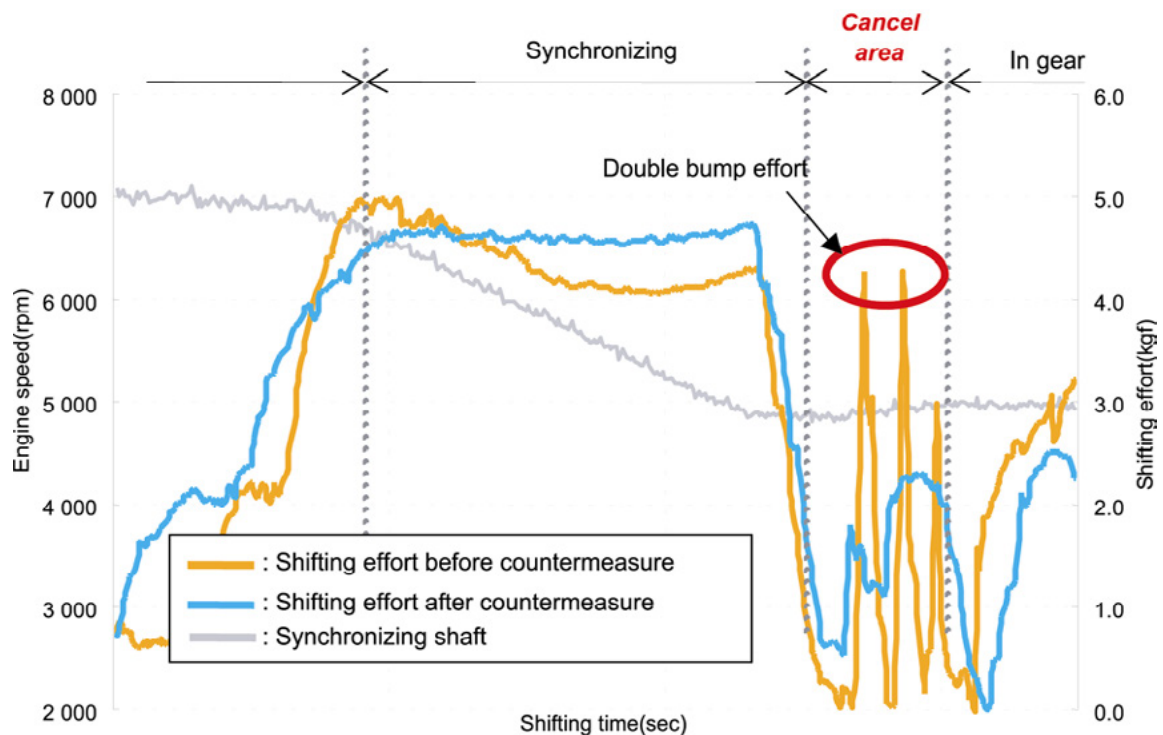


Fig. 14 Time chart of shifting

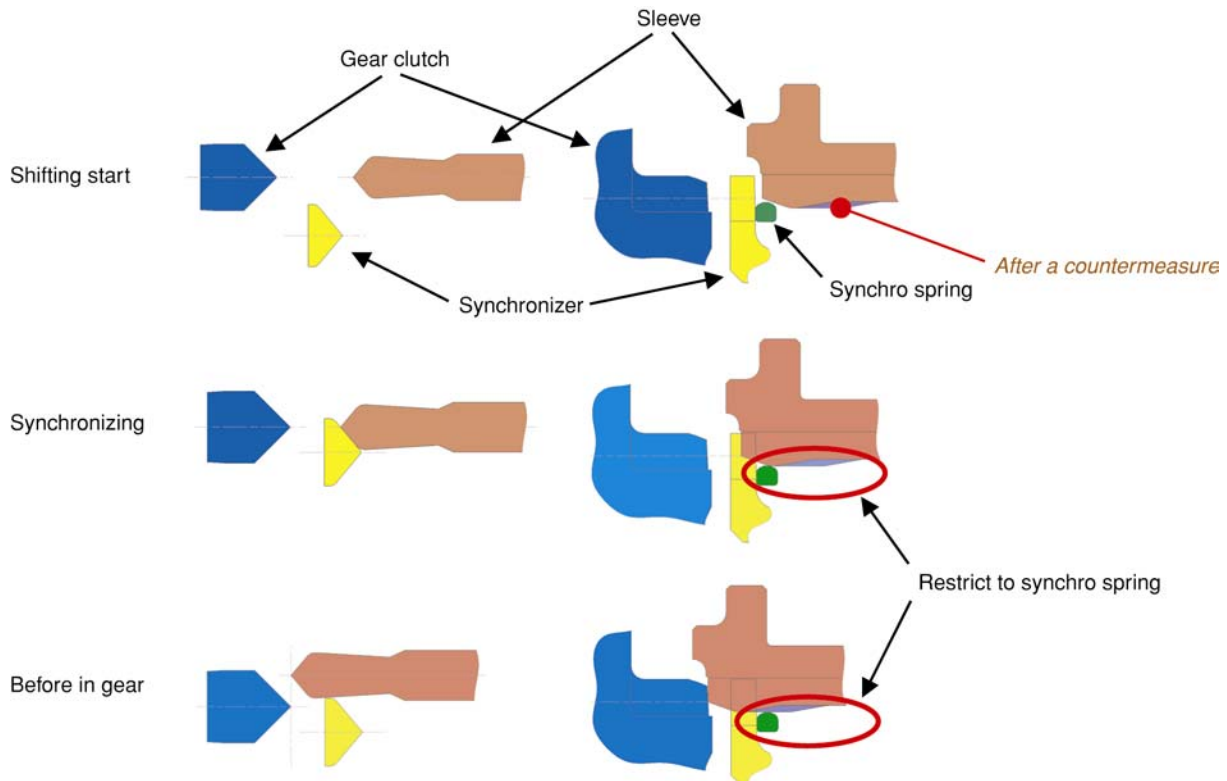


Fig. 15 Comparison of transmission length

transmission, to countermeasure rattling due to gear fluctuation of angular velocity when idling, a friction damper was used in counter low gear and counter 2nd gear.

In this new transmission, by adopting a DMFW, it is possible to establish wide-angle torsion properties, and by setting the spring constant low in the low torque range, fluctuation of angular velocity when idling is dramatically reduced. Through this, the friction dampers that were countermeasures for rattling were eliminated and the friction generated when the synchronization sleeve pushes its way through the clutch gear chamfer is reduced. Combined with the shift knob vibration reduction mentioned previously, shift feeling is improved and higher quality is achieved.

6.2. Reverse Structure

For the reverse structure, a constant mesh type synchro-mesh structure was adopted of the same type as for the drive gears. The purpose of a constant mesh type is to reduce gear noise when driving in reverse and the purpose of the synchro-mesh structure is to improve engagement during reverse shifting and to prevent abnormal noise from occurring.

Thanks to these measures engagement during reverse, which was 95% with the conventional sliding selection type, is dramatically improved.

6.3. Clutch Performance

Upon examination of clutch torsion properties, which greatly affect marketability of gear vibration and noise, it was found that by increasing idle gear count through the use of a constant mesh type for 6MT and reverse and by increasing idle gear rotation inertia accompanying an expansion of the distance between gear shafts, gear tooth contact energy is

increased. Fully considering this, another company has adopted a DMFW for this class of vehicle, too. A DMFW has two large masses – one on the engine side and one on the transmission side. Between them are spring and hysteresis elements so fluctuation of angular velocity input from the engine is attenuated restraining fluctuation of angular velocity of the gear and reducing rattling noise.

A SAC was also adopted for the clutch, and in addition to the benefits outlined in 3.3.2, there is a reduction in clutch pedal force variation.

Conventionally, because there are changes to diaphragm spring usage position (set position) due to clutch disk wear a large change in load during use is generated. In contrast, with the adoption of SAC, it is possible to have the diaphragm spring operate in a fixed set position following the clutch disk wear, thereby reducing clutch pedal force variation. Figure 16 gives the results of a comparison between the conventional structure and SAC release load elapsed change.

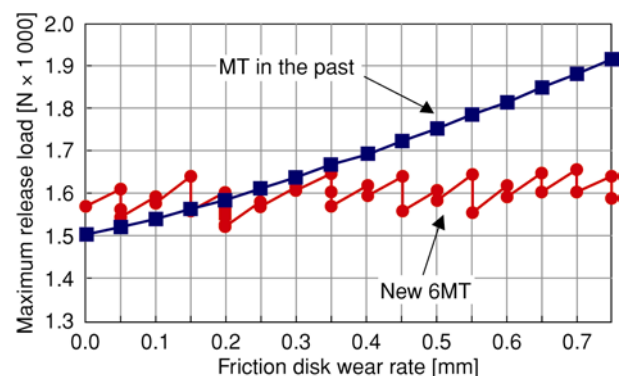


Fig. 16 Comparison of clutch release load

A new clutch disk facing material was adopted, which is environmentally friendlier.

7. Summary

Using the technologies outlined in this paper, the development targets for the transmission for Honda's luxury models were achieved through the following means.

- (1) Wider gear ratio settings were made possible by extending the distance between shafts. By adopting a clutch gear embedded structure and clutch automatic wear adjusting structure, a compact size was realized. As a result, while at the same time handling high torque, total length was equivalent to that of the 2 L class, and it is possible to fit the transmission to multiple models.
- (2) CAE analysis was carried out, enhancing gear stiffness and maintaining gear tooth flank durability. While maintaining case stiffness, case inner surface smoothness was implemented. In addition to the smooth case inner surface, by adopting a force feed lubrication structure, oil agitation loss was reduced and there was a 15 °C drop in oil temperature at high speeds. As a result, it is possible to deal with both high output and high torque.
- (3) Weight reduction measures were taken for each part, including weight reduction of the frame parts and weight reduction by making holes in the gear flange through hot forging and making the countershaft center hole diameter larger. As a result, weight reduction was achieved while maintaining high capacity.
- (4) Working towards shift feeling improvement, rattling noise reduction, and reduction in changes to clutch stepping force, a high quality product was achieved appropriate to a luxury sporty car.

8. Conclusion

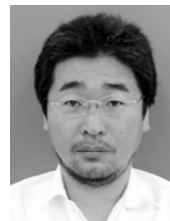
This 6MT is mounted to the ACURA 3.2CL TYPE S and has received favorable evaluations in the market. In addition, based on this 6MT and in response to the requirement for further weight reduction, a smaller size, or cost reduction, etc., this transmission can be fitted to multiple models.

Finally, the authors would like to take this opportunity to express their deep gratitude to the many people both in Honda and in other companies, who gave their generous cooperation in the development of this 6MT.

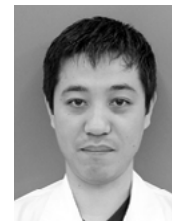
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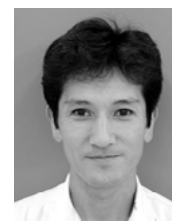
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