Development of Lightweight and Compact Differential for Formula One Car

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ABSTRACT

The differentials used in Formula One vehicles comprise a differential mechanism and a differential restriction device (LSD), which together form a bias-adjusting mechanism, and a final reduction gear set. Because the differential is positioned close to the rear end of the vehicle, it contributes to dynamic performance. Reducing the weight of the differential helps to centralize the mass of the vehicle, and increasing its compactness boosts aerodynamic performance. Development efforts were concentrated on the achievement of a lightweight and compact differential, with the focus in the initial stage on the centralization of mass. Previously mounted on the final driven shaft, the bias-adjusting mechanism was positioned on the final drive shaft, helping to reduce the center of gravity and the yaw moment of inertia. A full pinion engagement planetary gear was employed as the differential gear. A study of the merits and drawbacks of the mechanism resulted in the development of an ultra-short differential (USD) employing a full pinion engagement double pinion planetary gear positioned on the final driven shaft as the differential gear. It was predicted that this would increase the compactness of the unit and reduce its weight by 1.2 kg.

1. Introduction

Normally, the bias-adjusting mechanisms of Formula One differentials are positioned on the final driven shaft, and transmit the drive force that has been reduced by the transmission to the left and right drive wheels. This necessitates a large number of transmission parts and a high level of torque transmission, and the differential therefore represents a significant percentage of the weight of the drivetrain. In addition, due to restrictions on the operating angle of the drive shaft, the final driven shaft is positioned close to the rear end of the vehicle, at approximately the same height as the centers of the wheels. This means that a heavy component is located in a high position, far from the vehicle's center of gravity. Reducing the weight and size of the differential would contribute to lowering the vehicle's center of gravity and reducing yaw moment of inertia; reductions in these parameters help to boost cornering performance. In addition, the bias-adjusting mechanism also incorporates a limited-slip differential (LSD) that controls the right-left differential restriction torque, and the performance of this unit is also an important factor.

This paper will discuss the development of a lightweight and low center of gravity differential reconciling the satisfaction of these performance demands with durability.

2. Development Aims

The following aim was established in the initial development project in order to realize a low center of gravity and a low yaw moment of inertia.

(1) Positioning weight as low and as close to the center of gravity as possible

The development project aimed to achieve a low center of gravity and a low yaw moment by shifting the bias-adjusting mechanism from the final driven shaft to the final drive shaft, closer to the vehicle's center of gravity, and to a lower position.

Furthermore, the following aim was established for the development of the ultra-short differential, based on knowledge gained from the initial development and consideration of advantages and disadvantages:

(2) Reducing the weight of parts located far from the center of gravity and in a high position

This development aimed to contribute to the realization of a low center of gravity and a low yaw moment by reducing the size of the differential in order to reduce its weight.

Regulations stipulate the total weight of Formula One vehicles, meaning that a weight reduction would be compensated for by using ballasts. By this means, it would be possible to realize an even lower center of gravity and yaw moment. Figure 1 shows an image of the aims of the developments discussed here.

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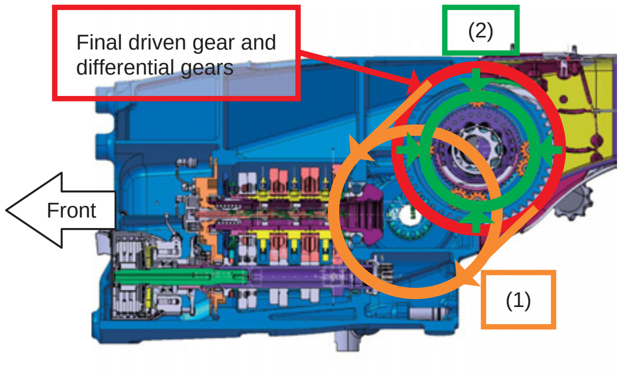

Fig. 1 Method of achievement of low center gravity and low yaw moment

3. Methods of Achievement of Development Aims

3.1. Efforts to Centralize Mass

The first stage of the development project focused on centralizing mass that would be effective in lowering the center of gravity and reducing yaw moment of the vehicle. A mechanism was developed that shifted the differential, the heavy component of the gearbox, from the final driven shaft to the final drive shaft, closer to the vehicle's center of gravity.

The transfer of the differential from the final driven shaft to the final drive shaft necessitated the use of two pairs of final gears. With respect to the width of the gears, because it was sufficient that one of the gears was able to transmit torque to the left or right wheel, the tooth width was reduced by approximately half against a conventional configuration.

The transfer of the bias-adjusting mechanism from the driven shaft side to the drive shaft side, which has a lower reduction ratio, helped to enable torque capacity to be reduced and the unit to consequently be reduced in size. Because the differential gears would be incorporated in the final drive bevel gears, a compact configuration was essential. A full pinion engagement planetary gear configuration that reduced the shared load through meshing of all contiguous pinion gears was adopted.

The cases of Formula One gearboxes must be made narrow, to help prevent interference with the flow of air to the rear of the vehicle. To help enable this, a configuration was adopted in which the LSD pistons were positioned opposite the clutch, and rods were introduced to connect them.

Figure 2 shows this bias-adjusting mechanism.

This configuration ultimately transferred approximately 2.5 kg of mass to the final drive shaft, reducing the yaw moment of inertia by 0.19% and lowering the center of gravity by 0.4 mm.

Figure 3 shows an image of this transfer of mass.

Rig tests showed that this mechanism achieved the necessary level of durability. However, the configuration was not employed in a vehicle, because the transfer of the differential to the final drive shaft would necessitate significant structural changes to the gearbox case, and the consequent increase in weight would nullify the merit of lowering the center of gravity.

This development project demonstrated that the bias-adjusting mechanism could be reduced in size through the use of a full pinion engagement planetary gear mechanism.

3.2. Efforts to Reduce Weight (USD)

It was proposed that the bias-adjusting mechanism should be positioned in the interior of the final driven shaft between the right and left drive shaft joints in order to achieve weight savings by reducing the size of the final gears and the differential.

Figure 4 shows the layout of a standard planetary bias-adjusting mechanism. This is a single-pinion planetary configuration that is composed of four pairs of mutually meshing single pinion gears, as differential gears, and left and right drive shaft drive gears positioned in parallel that mesh with the single pinion gears. The configuration is positioned on the final shaft outer circumference side and the inner side of the final

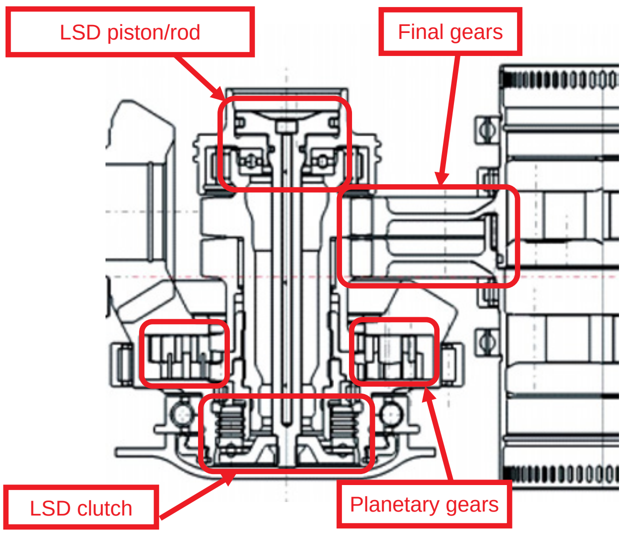

Fig. 2 Bias-adjusting mechanism on final drive shaft

Fig. 3 Image of movement of weight

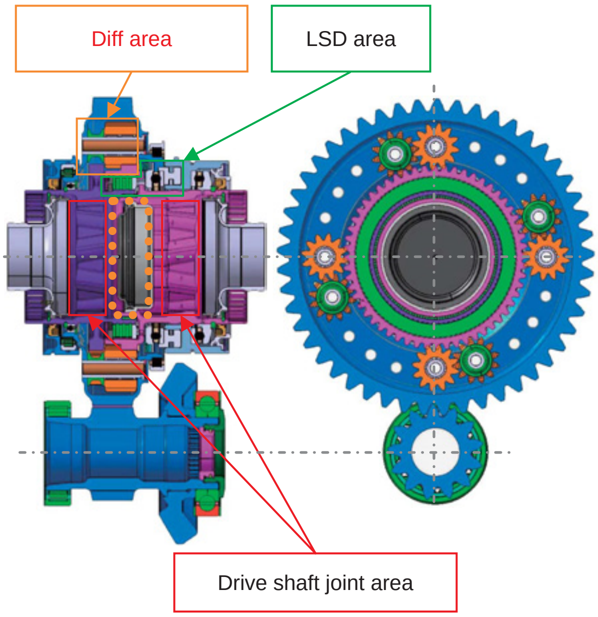

Fig. 4 Standard differential

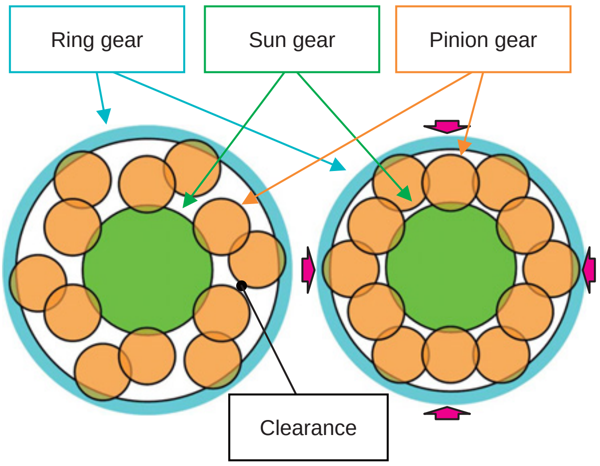

Fig. 5 Standard double pinion planetary gear

Fig. 6 Full pinion engagement double pinion planetary gear

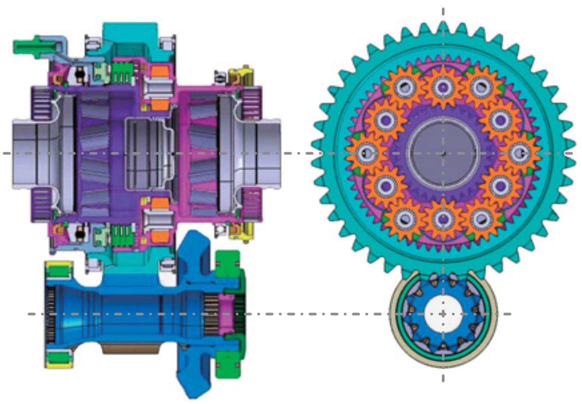

Fig. 7 USD

driven gears (the differential area) to drive the pinion gears. The unit is also provided with an LSD that produces left-right differential restriction torque using hydraulic control (LSD area).

From considerations of strength, the existing external diameters of the drive shaft joints could not be reduced. In addition, an increase in the total width of the differential was unavoidable if the existing mechanism was to be made more compact (the diameter of the final gears reduced), as it is fitted inside the final driven gear. The configuration of the LSD had also become complex, and no weight saving benefit could be obtained.

In order to resolve these issues, a double pinion planetary bias-adjusting mechanism was adopted to help enable the mechanism to be positioned between the drive shaft joints (in the drive shaft joint area) and reduced in size.

In the conventional single pinion planetary configuration, three gear meshing positions – left sun gear-pinion gear, pinion gear-pinion gear, and pinion gear-right sun gear, were arrayed in parallel. By contrast, the gear meshing positions in the double pinion planetary gear – ring gear-pinion gear, pinion gear-pinion gear, and pinion gear-sun gear - were arrayed vertically, and in-line engagement was obtained in parallel. This helped to enable the width of the bias-adjusting mechanism to be reduced. In addition, in order to further reduce the width and diameter of the double pinion planetary gears, the number of pinion gears was increased to six pairs from the former four pairs, reducing the shared load. However, increasing the number of pinion gears made it necessary to respond to interference between contiguous gears. If the diameters of the pinion gears were reduced in order to help prevent interference, it would be necessary to increase the width of the teeth to maintain strength. Again, if the angles of the interior and exterior pinion gears were increased to help prevent interference, the diameter of the ring gears would increase (Fig. 5). A full pinion engagement pinion gear configuration was therefore adopted to help enable the size of the ring gears and the final gears positioned outside the ring gears to be reduced. Figure 6 shows this configuration.

As a result, the distance between the final gear shafts was reduced from 125 mm to 100 mm, and reductions in size and weight were realized while the width of the total assembly remained unchanged (Fig. 7).

4. Verification of Performance and Durability

4.1. Differential Friction

The purpose of a hydraulically controlled differential mechanism is to restrict the differential motion between the left and right wheels in response to the operating status. If differential restriction torque exists during steady cornering with no drive force acting, then understeer is generated and this impedes drivability. Because friction exists in the differential limiting mechanism itself, even if oil pressure is entirely removed, differential restriction occurs corresponding to the amount of this friction. Given this, the development aimed to achieve the same differential friction as that of the proven 2008 specifications, and achieved almost identical performance.

4.2. LSD Control Performance

The differential restriction torque demanded from the differential during a race changes moment by moment in response to the operating status of the vehicle. A sufficiently short response time from the torque command to the generation of differential restriction torque is essential to controlling vehicle behavior as desired.

Figure 8 compares the level of displacement of the clutch assembly and torque response time for two differential specifications with different levels of clutch assembly stiffness. When engaged, the time from commencement of hydraulic pressure rise to 80% of maximum demand torque was evaluated, and when disengaged, the time from the commencement of hydraulic pressure drop to 5% of the same torque figure was evaluated. The vertical axis of the graph shows the level of axial displacement of the clutch assembly under a specific clamp load. The higher this figure, the lower the stiffness of the clutch assembly. Response time when the clutch was engaged satisfied the target, bettering the response time of the 2008 specifications. Neither specification satisfied targets when the clutch was disengaged, but a correlation with stiffness was verified. The clarification of stiffness targets guided development.

4.3. Durability

Verification of gearbox durability was essential, given the regulation stipulates that gearboxes must be used for four consecutive races. Rig tests were therefore conducted using loads corresponding to four races on the Monaco circuit, on which the frequency of use of the differential is highest. In the initial durability test, fatigue breakage occurred in the ring gears at approximately 60% of the target load. The following measures were effected before the next durability test:
(1) The tooth root shape was modified.
(2) Shot peening treatment was applied.

The modification of the shape of the tooth roots resulted in gear teeth being connected with a single radius at the bottom. An image is shown in Fig. 9.

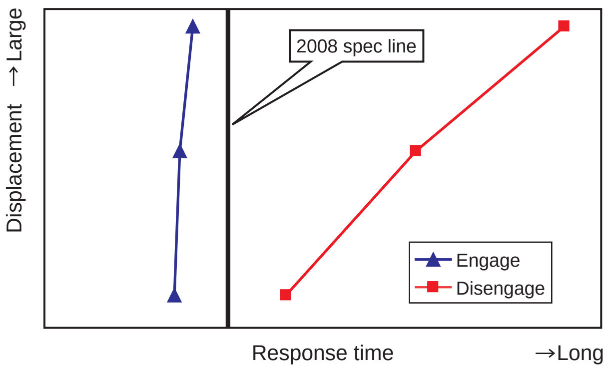

Fig. 8 Torque response time

The destruction rate of damaged final gears is 100%. Figure 10 shows an S-N curve for the ring gears in this development estimated based on this fact from the rig test results for the final gears up to this point.

Structurally speaking, minimal benefit would be obtained from reducing the width of the teeth of the final gears in the USD against the previous final gears, and some margin was thought to exist in terms of tooth root bending stress. Easy-to-manufacture shapes were therefore selected for the tooth roots, but target durability could not be achieved due to a decline in strength.

The effect of the single radius produced by the modification of the tooth root shape was verified in the same rig tests that had been conducted previously on the final gears. As in the case of the single radius, the research experience confirmed that the application of shot peening treatment to the tooth root increased the life of the gears by approximately 15%. Applying these results to the USD, the destruction rate corresponding to four races, the period for which regulations stipulate that the gears must be used, was 2.5%. The use of a single radius for the tooth roots and the shot peening treatment applied to the tooth roots would therefore increase durability. These results are shown in Fig. 11.

The effectiveness of these measures was, however, not verified due to the termination of the development program.

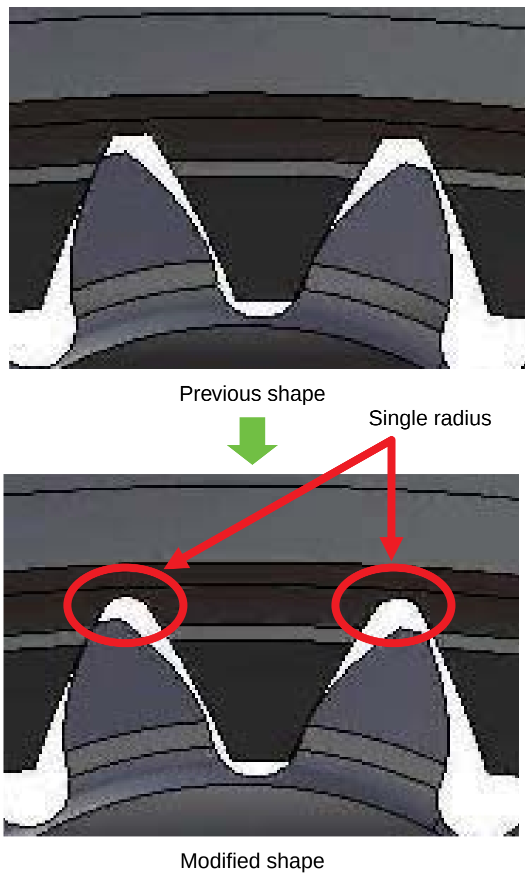

Fig. 9 Modification of root shape

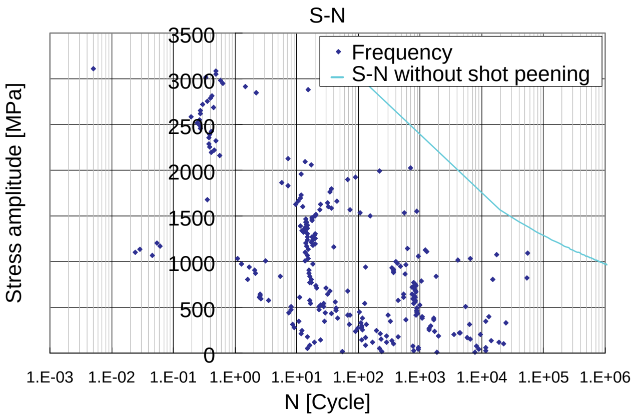

Fig. 10 Until fatigue destruction

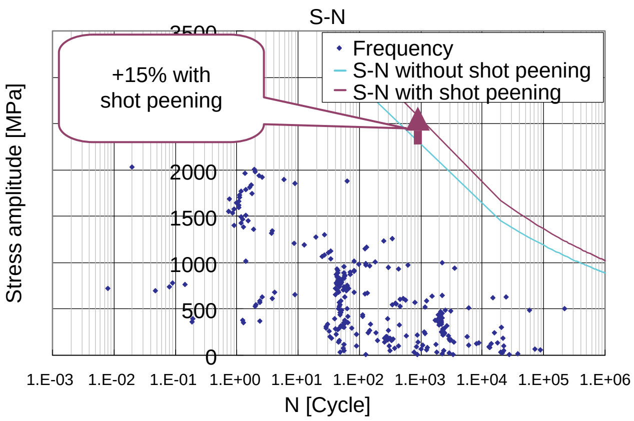

Fig. 11 4 races with large root R and shot peening

5. Conclusion

A full pinion engagement double planetary gear was employed in a differential mechanism in order to reduce the weight of the mechanism, lower the center of gravity, and reduce yaw moment of the vehicle. The following results were obtained:

(1) Compared to the 2008 specifications, the weight of the unit was reduced by 1.2 kg (-14% in terms of weight ratio), and the vehicle's inertial moment was reduced by 0.30% and its center of gravity lowered by 1.1 mm.
(2) The distance between the final drive shafts was reduced from 125 mm to 100 mm.
(3) Tests showed that the performance of the LSD could match the performance of the 2008 specifications.
(4) Tests showed that a sufficient level of reliability could be achieved to satisfy the regulation that gearboxes were not to be replaced for a four-race period.

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