Development of Gearbox Control during Honda Formula One Third Era

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ABSTRACT

One main development topic for the Honda Formula One third era gearbox was the seamless gearbox (Quick Shift), which reduced the interruption of acceleration to zero.

Gearshift control provided a way to control dog damage and excessive torque, which had been issues affecting Quick Shift feasibility, and this enabled the team to realize first deployment of the Quick Shift in racing, ahead of the other teams. The search for even smoother gear shifting continued after that, and led to the establishment of a mechanism that would reduce to a minimum the torque fluctuation resulting from correlation of the torsional vibration of the driveshaft and the inertial torque generated during gear shifts.

1. Introduction

The structure of the gearbox in the Honda Formula One third era was the seven-gear sequential gearshift of the constant mesh type widely used on motorcycles. This was electronically controlled using a hydraulic pressure actuator, and gearshift operations were carried out using a paddle attached to the steering wheel. Given this structure, shifting gears requires the engine torque to be first reduced to nearly zero. Reducing this gearshift time to the ultimate extent would contribute to increased racing competitiveness, and development of such control was therefore begun. The development effort sought, on the upshift side, reduction of acceleration loss, stabilization of wheel torque, and limitation of torque oscillation when shifting gears. On the downshift side, it sought reduction of minus torque when shifting gears, and reduction of idle running time. In this way, measures were taken to achieve a balance of hardware evolution combined with performance enhancement and establishment of reliability.

The Quick Shift mechanism, development of which started in 2004, was an innovative mechanism that enabled upshifting while still maintaining engine torque at the maximum level. It brought about a major evolution in performance when shifting gears. However, it was necessary to develop new controls to take advantage of this mechanism. Even before the introduction of the Quick Shift, the demand for better gearshift performance had been addressed by enhancing control equipment and changing the controlled subjects. With the seamless gearshift, however, there was a likelihood that significant damage will be made to the dog or the gears depending on the gearshift timing. Control that would eliminate this likelihood and assure a prescribed level of reliability was an absolute condition for the feasibility of this new mechanism. This paper will introduce the main controls that established the reliability of the Quick Shift and the control methods that realized smooth upshifting and downshifting.

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2. Development Goals

2.1. Establishment of Quick Shift Reliability

The Quick Shift is a mechanism that enables upshifting without lowering the engine torque. However, when one dog collided with another edge to edge, the shock was significant, and sometimes the dog edge would wear so that it could no longer provide driving power. If the next gear provided driving power when there was dog delta speed, an impact torque was also generated that resulted in the gear exceeding its torque limit so that the gear was damaged. The following two points were therefore defined as control objectives:

(1) Zero collision of one dog with another edge to edge
(2) Zero occurrence of gear over-torque

2.2. Realization of Smooth Upshifting and Downshifting

The inertial torque that occurred in gear shifting when shifting up or down could result in excessive wheel slip, rear tire lock-up, deterioration in drivability due to torque oscillation after gearshifts, and other such issues. The following objectives were defined to address these issues:

(1) Upshifting
- Maintain wheel driving torque
- Limit oscillations in engine speed and in torque after gearshifts

(2) Downshifting
- Limit minus torque after gearshifts
- Limit oscillations in engine speed and in torque after gearshifts

3. System Overview

Figure 1 shows the gearbox system configuration. The gear position is determined by the angle of the gearshift barrel, and the gear shifting is executed by input on the gearshift switch attached to the steering wheel. While shifting gears, the system controls the gear position and simultaneously requests the clutch transfer torque, engine speed, and engine torque.

The clutch operates according to analog input from the clutch paddle attached to the steering wheel. In addition to engaging and disengaging, the amount of clutch disengagement is also controlled according to gearshift and anti-stall requests.

The gearshift and clutch actuators are driven by 200-bar hydraulic pressure controlled by a Moog valve. (The control cycle is 1 ms.)

4. Establishment of Quick Shift Reliability

4.1. Avoiding Edge to Edge Collision of One Dog with Another

An overview of the control should include the development of Trigger Shift control. The Trigger Shift control uses rotation sensors for the drive shaft (layshaft), which is connected directly to the engine, and for the driven shaft (main shaft) to constantly calculate the angle of the gear and of the dog ring, memorize the phase difference between the dog ring and the gear when the gear is engaged and driven, and calculate the timing for driving the gearshift barrel so that it can occur in the interval between dogs at the next gearshift. Figure 2 shows a conceptual image of the target position.

The Trigger Shift learns the phase differences in the below order from (1) to (4), and drives the shift barrel accordingly.

(1) The rotation angle of the dog ring is calculated from the main shaft rotation.
(2) The rotation angles of gears 1 to 7 are calculated from the layshaft rotation.
(3) The angle difference between (1) and (2) (phase difference (1)-(2)) for each gear when engaged with the dog ring is calculated and memorized.
(4) The angle up to the point where the target phase difference is reached is predicted from the barrel movement time and the dog delta speed, and the barrel is driven accordingly to shift gears.

The above describes the basic logic involved.

Figure 3 shows a block diagram of the Trigger Shift control.

The following two data settings are necessary to implement this control:

(1) Response time until the barrel reaches the in-gear position
(2) Target position between dogs (upshift and downshift)

These values are determined by the structure, and they do not require fine tuning while the car is actually in operation. The control is efficient also in helping to ensure reliability by reducing the likelihood of operational errors during racing, and the like.

Figure 4 shows example Trigger Shift race results in graph form from the ninth competition in 2006, the Canada Grand Prix. The graph summarizes the in-gear positions between dogs when shifting up from 3rd to 4th gear. The triangle symbols (▲) in the figure represent the positions targeted at by the Trigger Shift system. The open circle symbols (◯) in the figure represent the actual in-gear positions. The figure indicates that the system generally functioned with in-gear positions as targeted

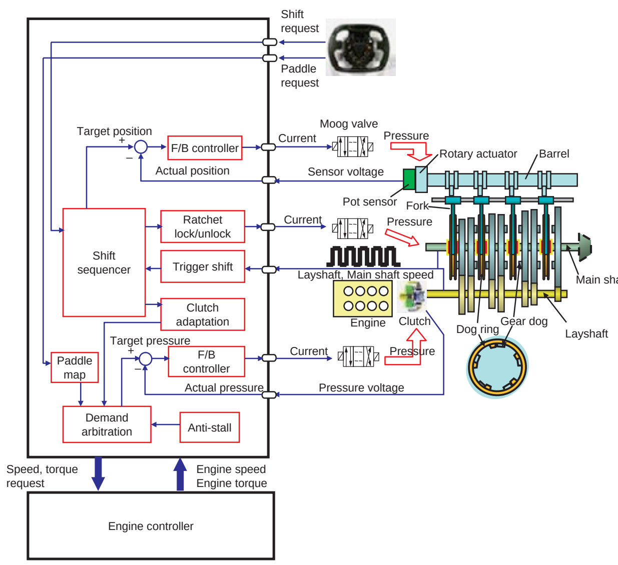

Fig. 1 System configuration

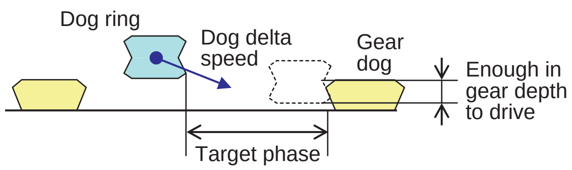

Fig. 2 Image of trigger shift

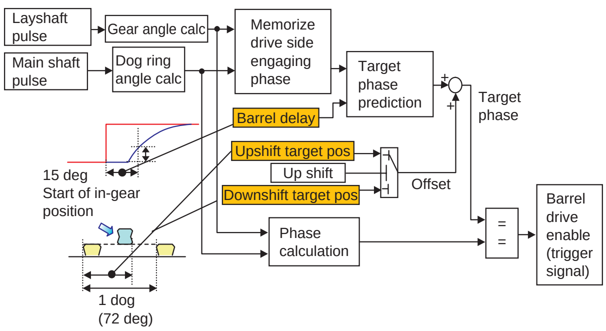

Fig. 3 Trigger shift control system

for, and did so repeatedly with good reproducibility.

The Trigger Shift performance limits will also be described here. If the barrel response is taken to be constant, then the dogs engage with a shallower intermeshing when the dog delta speed is greater, so that the dogs are more likely to collide edge to edge with each other. Furthermore, even when the intermeshing depth is the same, there will be greater damage to the dog edge when the dog delta speed is greater. Figure 5 shows the results of testing on wear limits by intermeshing gear depth and dog delta speed. The maximum value for dog delta speed between dogs is determined from the upshift speeds and the gear ratios in Formula One racing to be approximately 2000 rpm. The intermeshing gear depth necessary to reduce the occurrence of wear in this case, as shown in Fig. 5, is 1.5 mm. By compiling fork speeds in all the 2006 races, it was found that the four sigma point of the distribution was 1.54 mm/ms, and this was taken as the worst case. Given this, the in-gear position capable of avoiding dog-to-dog collision could be expressed in terms of time as 1.5/1.54 ≈ 1 ms, which is the necessary safety margin when the dog delta speed is 2000 rpm.

In actuality, as shown in Fig. 6, it is necessary to take into account variations in the target position due to barrel response times and dog delta speed changes. Taking these factors into consideration, together with the fact that the dogs are placed every 72 degrees, the limit dog delta speed at which the Trigger Shift can assure performance is expressed by formula (1):

(Dog pitch) - (Dog delta speed variation) / (Control cycle error) + (Barrel speed error) + (Safety margin) (1)

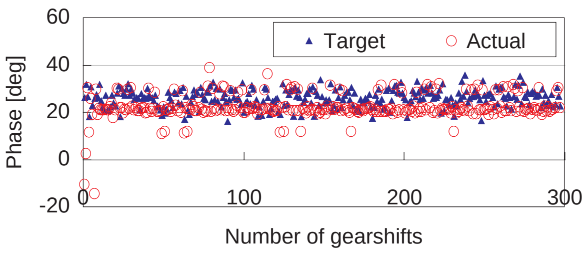

Fig. 4 Trigger shift result (2006 Canada GP)

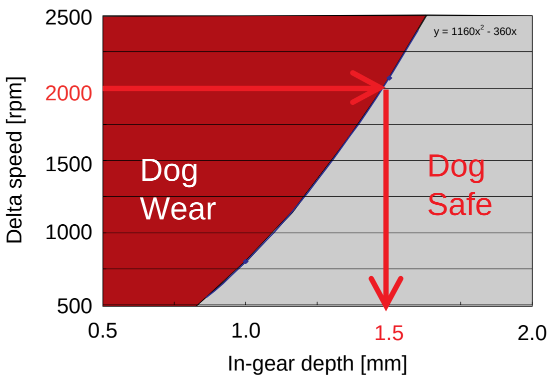

Fig. 5 Relation of dog delta speed and in-gear depth

In this formula, the value will be low for 2000 rpm. The recommended target position will be 32 degrees at the middle of the variation.

4.2. Avoiding Gear Over-Torque

Adjustment of the amount of clutch disengagement when shifting up so that the clutch transfer torque will be between the engine torque and the gear limit torque allows the impact torque to be absorbed by the clutch. However, the torque characteristics of the clutch lacked stability while the car was in operation due to changes in the coefficient of friction μ and the clamp load (the load pushing against the disk) resulting from disk wear. Therefore, stable absorption of the impact torque could not be realized with a fixed amount of clutch disengagement, so an auxiliary control of the clutch disengagement amount called clutch adaptation was applied. This control is focused on the amount of clutch slippage while shifting up, and it applies cumulative adjustments to the amount of clutch disengagement so that it will match the target slip amount. The clutch slippage amount while shifting up is calculated, and if it has not slipped more than the target amount, then the clutch will be adjusted in the direction of disengagement for the next shift up. If it has slipped more than the target amount, then it will be adjusted in the direction of tightening the clutch. Figure 7 shows a block diagram of the clutch adaptation control.

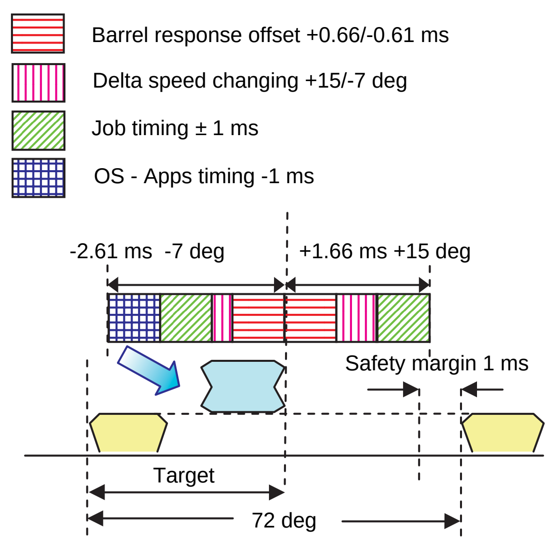

Fig. 6 Control tolerance of trigger shift

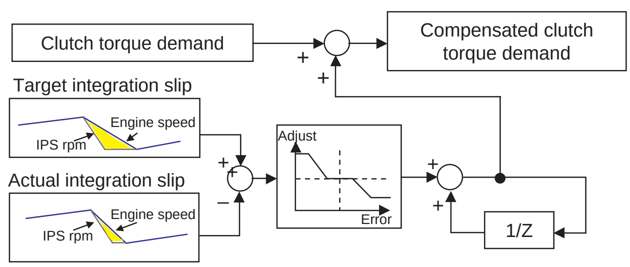

Fig. 7 Configuration of clutch adaptation control

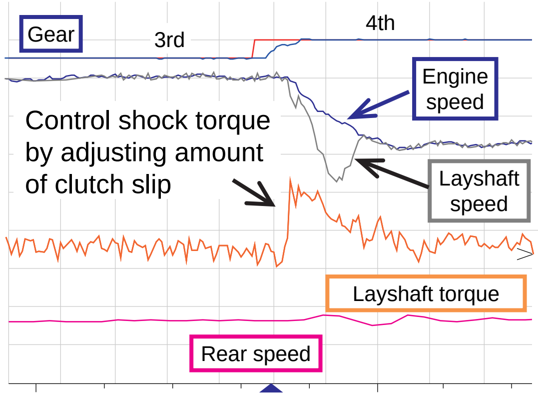

Fig. 8 Example of upshift behavior

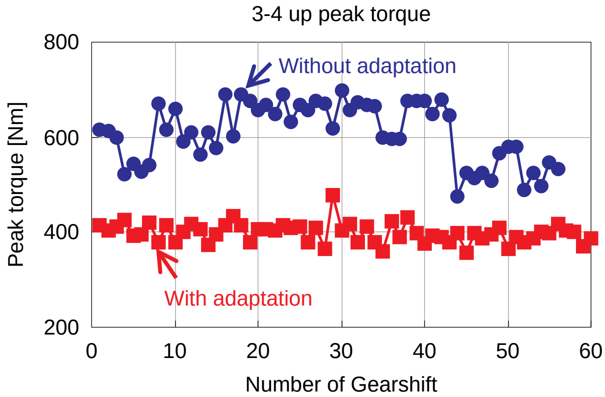

Fig. 9 Clutch adaptation result

Figure 8 shows the waveforms for actual shifts up from 3rd to 4th gear that absorb impact torque by making the clutch slip. Figure 9 shows a comparison of impact torque in the course of driving approximately 10 laps. It is apparent from this figure that when clutch adaptation is applied, the impact torque is held in the vicinity of 400 Nm with stability as intended.

5. Realizing Smooth Up and Down Shifting

5.1. Upshift

When shifting up with Quick Shift, as shown in Fig. 10, the inertial torque when shifting gears results in the wheel driving torque that exceeds the tire limits and induces excess wheel slipping as well as oscillation in the engine speed. The torque oscillation following a gear shift can lead to impaired drivability. Therefore, the following two points take on importance for upshift control:

(1) Limiting engine speed oscillation

(2) Maintaining optimal wheel driving torque

In order to quantitatively evaluate the engine speed oscillation, therefore, the oscillation level was represented by the deviation from a logical value for engine speed in the next gear integrated and depicted as an area, as shown in Fig. 11.

Figure 12 shows the summarized results of the correlation of this oscillation level with the slope when the engine speed is dropping. The horizontal axis of the graph shows the rate at which the engine speed is dropping, while the oscillation level is plotted on the vertical axis. It is apparent that the oscillation increases whether the slope of the engine speed is steep (part A in the figure) or too shallow (part B in the figure), and that it reaches its minimum value at -90 rpm/ms.

Figure 13 shows the summarized results of the correlation of the oscillation level and the delta engine speed with the next gear when the clutch is engaged. The graph has the difference between the engine speed when the clutch slippage ends and the engine speed at the next gear plotted on the horizontal axis, and the oscillation level plotted on the vertical axis. It is apparent from these results that engine speed oscillation when shifting up depends on both the engine speed when the clutch is engaged and the engine speed at the next gear. It is also apparent that reducing the differences between these speeds can limit oscillation, as well.

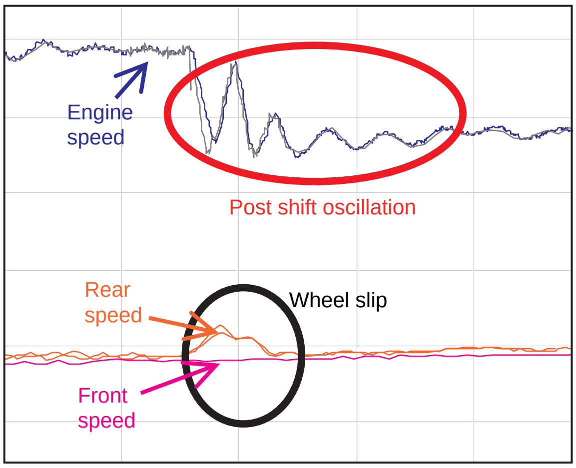

Fig. 10 Upshift oscillation issue

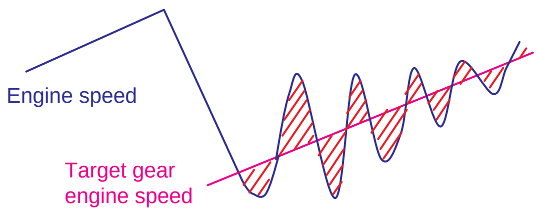

Fig. 11 Definition of oscillation quantity

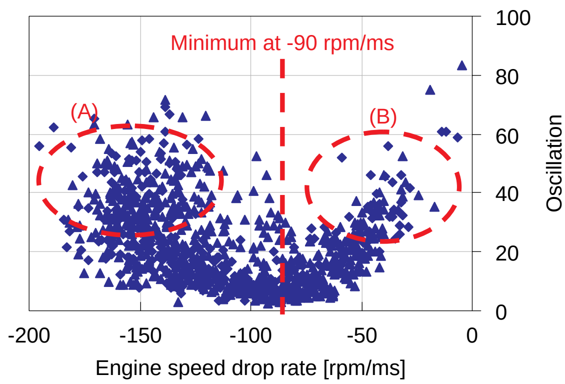

Fig. 12 Oscillation and engine speed drop rate

Analysis based on the above results showed that the oscillation level was smaller in the vicinity of the drop in engine speed at the rate of around -90 rpm/ms, which largely coincided with the torsional oscillation cycle in the drivetrain approximately 20 ms after the engine speed began to drop. Therefore, as shown in Fig. 14, it was found that oscillation could be limited by synchronizing it with the torsional oscillation of the drivetrain so that the clutch slippage is stopped at the point when the speed of the layshaft (which is on the other side of the clutch from the engine) is the same as the speed of the engine at the next gear, thus canceling out the torsional oscillation. It is necessary, therefore, to have the period of time until the engine speed reaches the next gear speed match with the torsional oscillation cycle, and this necessitates appropriate control of the engine torque reduction and the clutch slippage amount.

Control of wheel driving torque will next be described. The torque when shifting gears is subject to the influence of the inertial torque generated when the engine speed is reduced to the speed at the next gear. It is expressed by the following formula.

$$T_{Lay} = T_{ENG} - I_{ENG}\, d\omega / dt \qquad (2)$$

(T_{Lay}: Layshaft torque. T_{ENG}: Engine torque. I_{ENG}: Engine inertia. ω: Engine angular velocity)

In the above formula (2), the term $I_{ENG}d\omega/dt$ normally had a negative value when shifting up, so that it would be added on top of T_{ENG}. This increased the wheel driving torque, which would induce excessive wheel slipping. It was necessary, therefore, to control T_{Lay} so as to keep the wheel torque constant, and the target T_{Target} would then be expressed by formula (3):

$$T_{Target} = T_{ENG} \times \text{Previous Ratio / Next Ratio} \qquad (3)$$

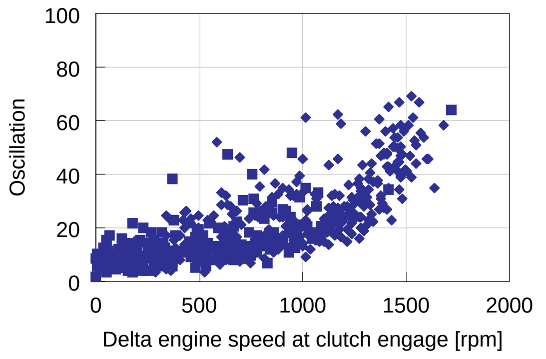

Fig. 13 Oscillation and speed error at clutch engage

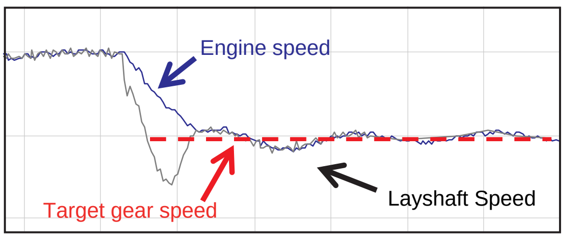

Fig. 14 Upshift example

The logic that realizes these aims can be described as follows. The target time for the engine speed to drop to the next gear speed is replaced by the downward slope of the target engine speed. The inertial torque $I_{ENG}d\omega/dt$ is defined by this slope of the target engine speed. In order to realize T_{Target}, therefore, T_{Lay} + $I_{ENG}d\omega/dt$ is supplied as the engine torque reduction demand during the gear shift.

T_{Target} is the clutch transfer torque T_{Clutch}, and where clutch adaptation had conventionally modified the clutch slippage amount, here instead it modified the slope of the engine speed by applying a correction quantity defined as a coefficient equivalent to the clutch coefficient of friction μ, thereby enhancing the accuracy of T_{Clutch}. Figure 15 shows a conceptual image of these elements of control.

Figure 16 shows the superposed actual upshift waveforms for the conventional upshift and the upshift with the new control applied. A smooth upshift was realized by the optimization of the slope of the engine speed and the engine torque reduction.

5.2. Downshift

Figure 17 shows the waveforms for a conventional downshift. The issues when downshifting are to limit the locking of the rear tires and to stabilize vehicle behavior when braking. The mechanism by which rear tire lock occurs during downshifting is described below.

(1) The dog delta speed immediately before being driven by the dogs is large.

(2) The dogs' engagement with each other leads to rapid convergence of the dog delta speed.

(3) Inertial torque is generated, this applies torsion to the

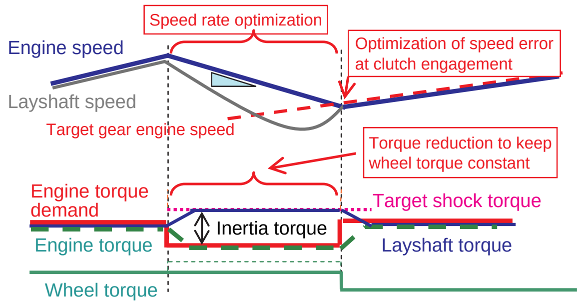

Fig. 15 Image of smooth upshift control

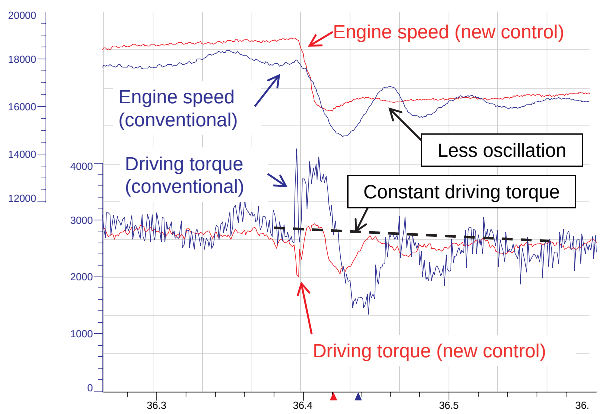

Fig. 16 Upshift comparison

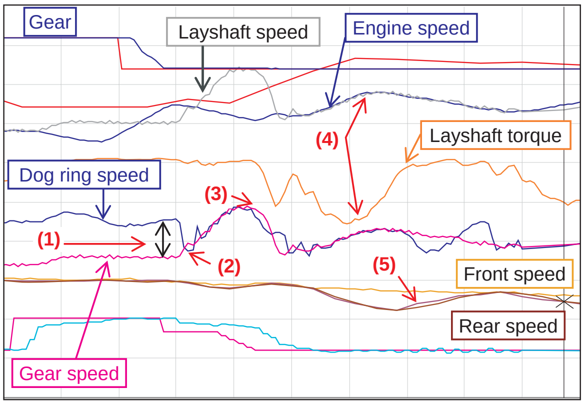

Fig. 17　Example of bad downshift

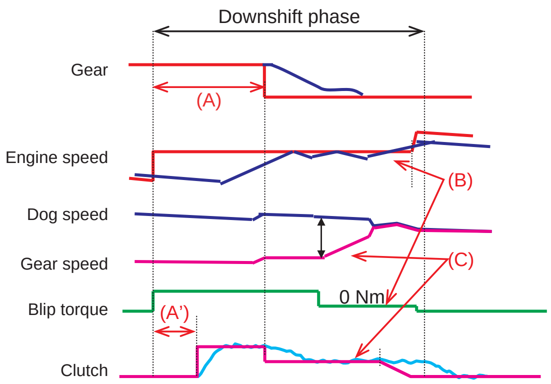

Fig. 18　Downshift control concept

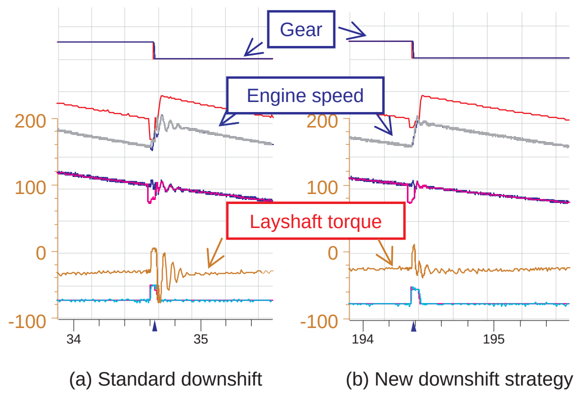

Fig. 19　Downshift comparison

driveshaft in the direction of deceleration, and oscillation occurs.

(4) Engine speed is increased and the minus torque exceeds the rear tire drive torque limit.

(5) The rear tires lock.

The above item numbers correspond to the numbering in Fig. 17.

The major reason for deterioration of downshifting is the dog delta speed when being driven by the dogs while in gear. A smoother downshift was realized by the following kind of control of this dog delta speed.

(1) Make the dogs drive while the gear speed is increasing. (Cancel out the inertial torque while being driven by the dogs.)

(2) While being driven by the dogs, make the gear speed lower than the dog ring speed.

In (1), the clutch is disengaged and the engine is blipped to raise its speed. Then, the clutch is engaged, with timing matched to when dog driving occurs. The clutch torque raises the gear speed, and the dog delta speeds converge. In (2), the engine speed demand is set lower than the next gear speed to control the gear speed so that it will be lower than the dog ring speed.

Figure 18 shows a conceptual diagram of this control.

In the diagram, the barrel-driven timing (A) and the clutch disengagement initiation timing (A') are determined by the engine blip response. As noted above, it is important that convergence of the dog delta speed between dogs should occur gently, which is why the engine torque and speed instruction values are defined so as not to decrease the gear speed in (B). Also, as shown in (C), the clutch transfer torque is controlled appropriately so that the gear speed increases gradually, thus relaxing the impact torque when in gear.

Figure 19 shows the results of comparison between these controls and conventional control. Figure 19 presents the waveforms for the conventional downshift (a) and the downshift (b) with the control described above. Comparison of the torque behavior and the oscillation in the engine speed after downshifting confirmed that these are reduced when the new control is applied.

6. Conclusion

The Quick Shift that was applied in racing from 2005 had enhancements made to the Trigger Shift in 2006, and it achieved a record of zero retirement for gearbox reasons in races during the 2007 and 2008 seasons.

The new control was applied from 2007, and the enhancement of gearshifting yielded a zero level of related driver complaints.

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