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ABSTRACT

Toyota has developed the world's first 8-speed automatic transmission (AA80E) for RWD automobiles. The transmission will first be used in the all-new Lexus LS460. In addition, a novel control system has been developed to maximize the predictability, response, efficiency, and initial quality of the powertrain while utilizing the high number of gear steps.

INTRODUCTION

Mounting concern over high carbon dioxide emissions and the growing costs of crude oil continue to drive Toyota Motor Corporation to develop increasingly fuel efficient automobiles. On the other hand, in the luxury vehicle market, high dynamic performance is still one of the most significant factors of attraction. In the development of the LS460, the goal was to not only improve fuel economy and overall refinement, but also to realize a major improvement in drivability and dynamic performance. The present paper will provide an overview of the function and integration of the primary components that make up the AA80E, as well as the new powertrain control system behind it, which have contributed to the improvement in the above stated areas.

KEY DEVELOPMENTS

Several new and improved control elements applied to the AA80E distinguish it from previous systems:

1. A quantitative index of acceleration and deceleration, used to provide objective targets for smooth and stable acceleration with respect to accelerator operation.
2. A driving response and acceleration management system that determines a drive power target based on accelerator input. To meet the drive power target, the most suitable gear stage and engine torque level is calculated, while also taking into consideration drive power requirements from other vehicle systems.
3. A dual clutch-to-clutch shift control, in which two friction elements engage and two friction elements disengage simultaneously to change gear stages with a quick and shock-free response.
4. Improvement of fuel economy through expansion of the fuel cut area, adoption of neutral control, and expansion of the T/C lock-up area.
5. Improvement of the initial quality of gear shifts through increasing correction value accuracy and precision in transmission components during the manufacturing process.

Table 1: Technical Data of the AA80E

Torque Converter	φ272mm with Lock-up Clutch	
Gear Ratio	1st	4.596
	2nd	2.724
	3rd	1.863
	4th	1.464
	5th	1.231
	6th	1.000
	7th	0.824
	8th	0.685
	Rev	2.176
	(Final)	(2.937)
Shift Elements	4 Disc Clutches	
	2 Disc Brakes	
	1 O.W.C (for 1st to 2nd)	
Total Length	677mm	
Mass (Including ATF)	95Kg	
Max Torque Capacity	550Nm	

Table 2: Operation Chart of Friction Elements

Gear	Clutch				Brake		O.W.C.
	C1	C2	C3	C4	B1	B2	F1
1st	●					▲	●
2nd	●				●		
3rd	●		●				
4th	●			●			
5th	●	●					
6th		●		●			
7th		●	●				
8th		●			●		
Rev				●		●	

● :Engage ▲ :Engage For L range

Table 3: Solenoid Application Elements

SHIFT	Solenoid								
	ON/OFF		Linear						
	SR	SL	SL1	SL2	SL3	SL4	SL5	SLU	SLT
1st	●		●						●
(L range)			●					●	●
2nd	●		●				●		●
3rd	●		●		●				●
4th	●		●			●			●
5th	●		●	●					●
6th	●			●		●			●
7th	●			●	●				●
8th	●			●			●		●
Rev	●	●				●			●
Lock Up	●	●						●	●

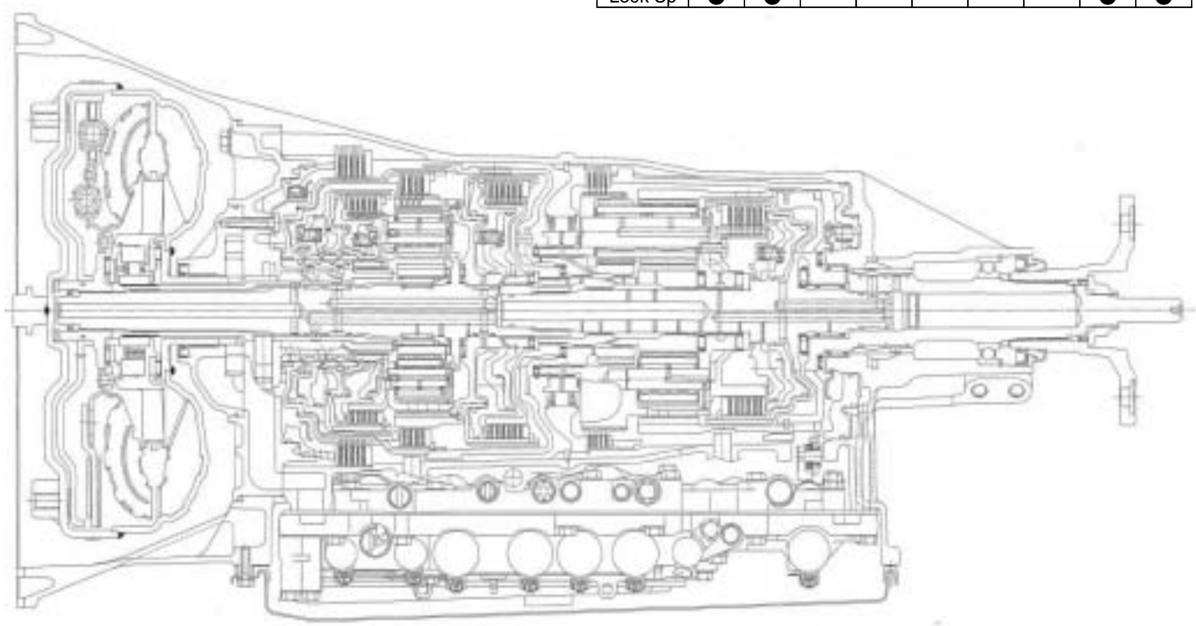

Figure 1: Cross Section of AA80E

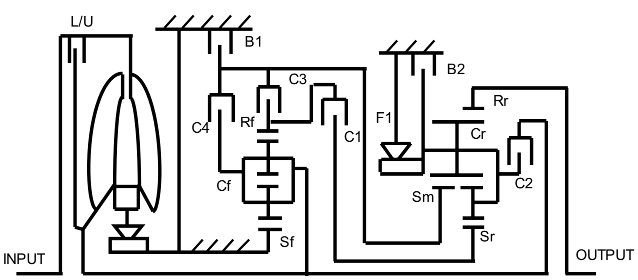

Figure 2: Gear Train Schematic

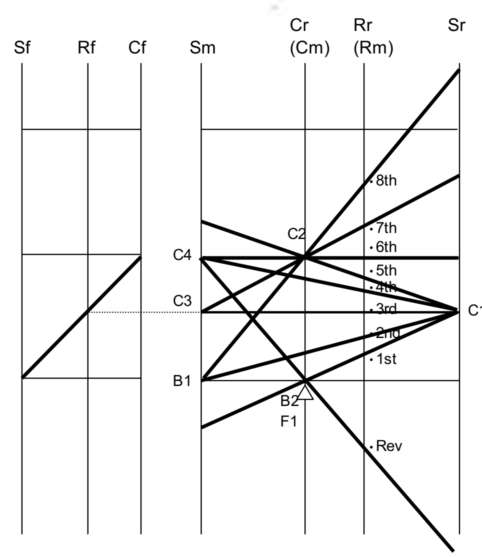

Figure 3: Nomographic Chart

BASIC TRANSMISSION STRUCTURE

The AA80E is composed of a high capacity torque converter, four clutch elements, two brake elements, one O.W.C., one under-drive planetary gear set, and one Ravigneaux planetary gear set. Table 1 lists further hardware specifications. A cross section of the AA80E is shown in Figure 1, and the basic structure of the gear sets and friction elements is revealed in Figure 2. Rotation sensors are installed in three components: the input shaft, the output shaft, and a sun gear (part Sm in Figure 2). The relationship between gear sets and friction elements for each gear stage is illustrated in Figure 3. Tables 2 and 3 show the operation of friction elements and solenoids, respectively, for each gear stage.

HYDRAULIC CONTROL DEVICE (VALVE BODY)

The valve body features large-flow, compact linear solenoids which allow direct control of hydraulic pressure, thus eliminating separate pressure control valves for each engagement element. This system greatly reduces the hardware requirements in the valve body; the total number of valves has been reduced by 20% compared to conventional transmissions. Figure 4 shows the streamlined structure of the AA80E valve body versus conventional hydraulic control devices. In addition, precise control of oil pressure was achieved by improving response in the linear solenoids; this allowed further refinement of overall shift characteristics.

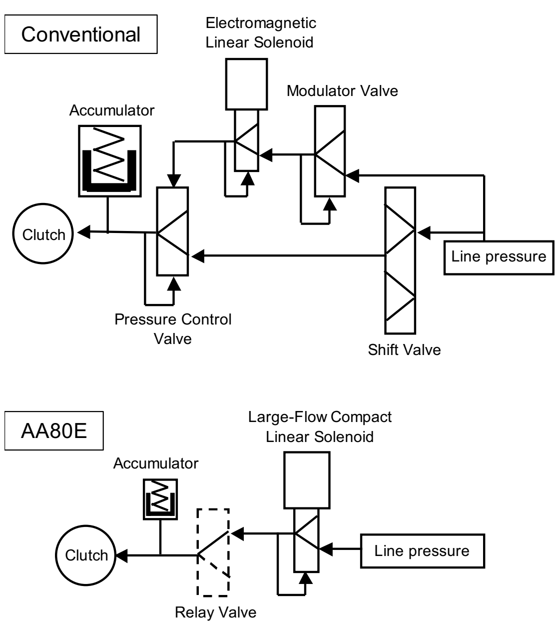

Figure 4: Hydraulic Control Structure Chart

CONTROL SYSTEM ARCHITECTURE

The powertrain control system features a newly developed driving response and acceleration management system. In this system, a Model Driver decides drive power targets based upon the accelerator operation, and a Powertrain Manager (PTM) sorts them into an engine torque target and gear stage target. As a result, both engine and transmission play an actuator-like role, dutifully realizing each target. The PTM also considers requests for drive power from other systems such as Vehicle Dynamics Integrated Management (VDIM), which contribute to improvements in vehicle safety as well as performance. Figure 5 shows the main differences between this control system and conventional control systems.

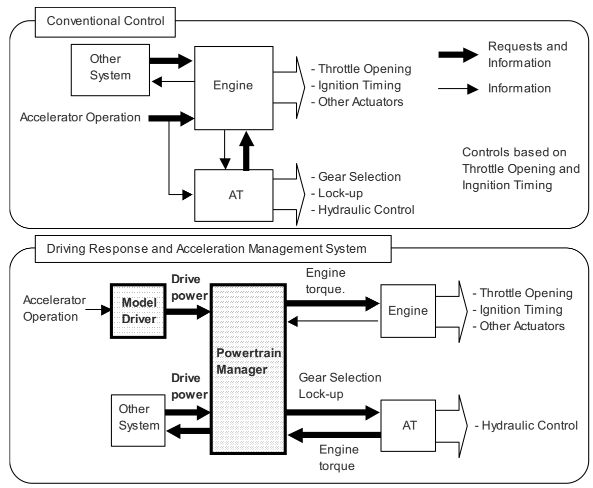

Figure 5: Conventional versus Driving Response and Acceleration Management System

ESTABLISHING OBJECTIVE TARGETS FOR DETERMINING SUITABLE DRIVE POWER

For the new control system to determine the most suitable drive power for a given situation, dynamic performance in the areas of acceleration and deceleration are quantified and broken down into unique scenarios. Specifically, drivers' operations are classified into five situations: standing-start, acceleration, deceleration, reacceleration, and manual shift operation. Each of these situations is broken down into further categories. For example, acceleration is separated into peak acceleration, smoothness, linearity, kick down response, and so on, in order to establish quantitative targets. For deceleration, the target values of engine brake force and response are determined for situations such as accelerator off and switching to manual mode operation.

For a given performance scenario, the Model Driver uses vehicle specifications, engine torque characteristics, T/C characteristics, gear ratios, etc., to determine the drive power necessary to achieve the performance target values. The required drive power is then used in an algorithm to decide engine torque and gear stage in the PTM. In Figure 6, an example of target parameters in the quantitative index during acceleration and deceleration is shown, and an example of a target value with respect to other models is shown in Figure 7. A graph of drive power versus accelerator pedal angle for both a conventional and new control system is shown in Figure 8.

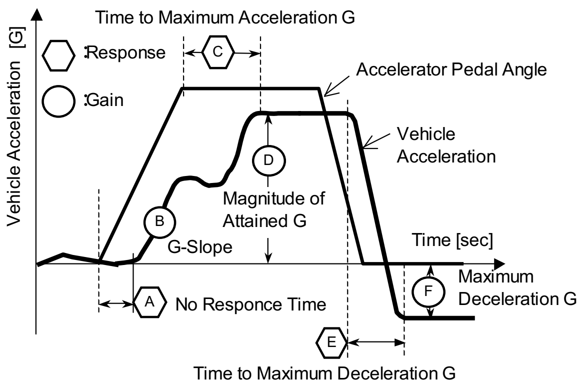

Figure 6: Example of quantitative index for acceleration and deceleration

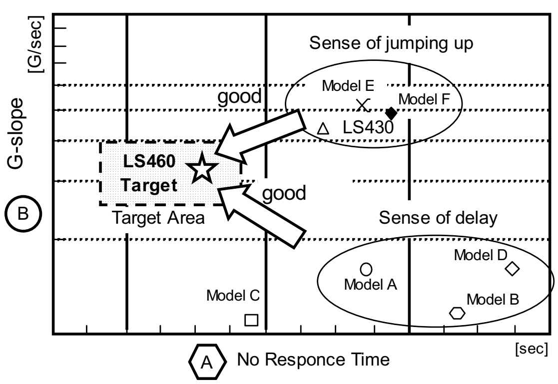

Figure 7: Example of calculated target value for acceleration with respect to other models

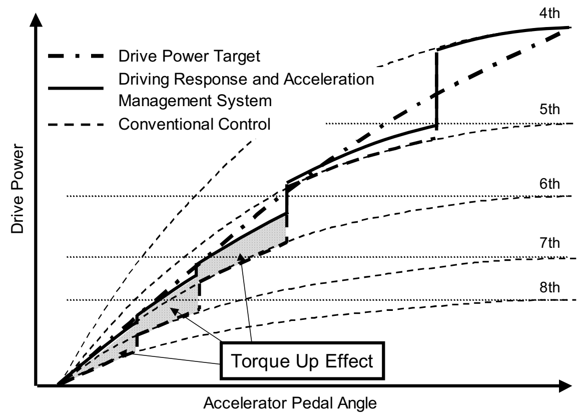

Figure 8: Comparison of conventional control to new control system with respect to the drive power target

SELECTION OF OPTIMAL GEAR STAGE FOR REQUIRED DRIVE POWER

Although there is more than one solution of gear stage and engine torque for realizing a drive power target, the PTM will select the gear stage that not only realizes the drive power target, but also prevents a sense of "busy shifting". For example, when slowly pressing on the accelerator pedal while cruising in eighth gear, the PTM will downshift one stage at a time, 8-7-6, to smoothly connect the drive power with engine rotation speed. However, if instead the accelerator were quickly applied for some small angle, the PTM will predict that the drive power suitable for sixth gear is required, and will command an 8-6 direct shift, skipping the seventh gear entirely. Even if the accelerator is pressed further and the drive power suitable to fifth gear is required, as long as the time elapsed from the initial 8-6 shift is small, the downshift to fifth is not performed and the drive power is increased by engine torque alone. By delaying the downshift to fifth for an appropriate time, a sense of "busy shifting" is thus prevented. The differences between these two strategies are illustrated in Figure 9. In the same scenario, if the accelerator pedal were quickly pressed all the way down, the control system would recognize the immediate demand for maximum drive power and command a direct, 8-3 kick down shift.

Regarding the gear stage selection process during deceleration, the new powertrain control system achieves reduced operation frequency of the brake pedal by the driver and enhancement of reacceleration performance. This has been made possible by expanding both the Grade Logic and the Navigation-based Artificial Intelligence (Navi-AI) shift control, which allow fine-tuned engine brake control. In other scenarios, when the driver exhibits sportier driving, acceleration and deceleration is enhanced by delaying upshifts to maintain higher engine speeds. Further, during the manual shift mode (range-hold), the gear stage is selected according to the speed of the vehicle and the previous gear stage to enhance the capability to decelerate as expected by the driver.

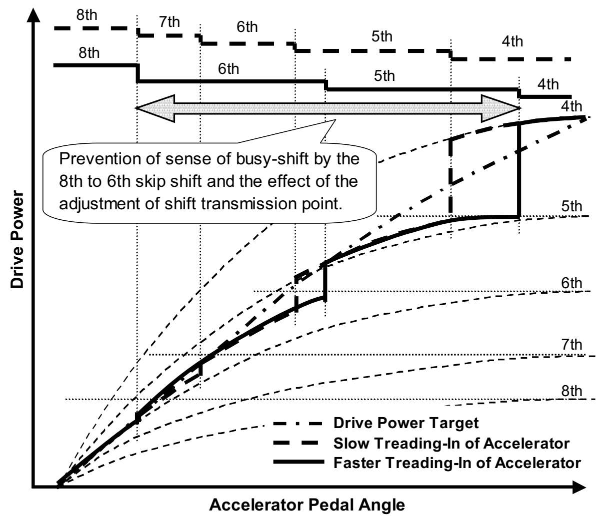

Figure 9: Shift transmission and drive force according to treading-in of accelerator.

ACHIEVEMENT OF QUICK RESPONSE AND HIGH QUALITY SHIFT CHARACTERISTICS

In order to quickly achieve a drive power target, a fast and smooth power-on downshift is necessary. The AA80E has a structure in which the input is switched from the C1 to C2 friction element at sixth gear. For certain downshifts (going from a gear stage of sixth or higher to one of fourth or lower) this structure requires two elements to disengage and two elements to engage simultaneously. For example, during the 8-3 downshift, the input C2 and reactive element B1 disengage while the input C1 and reactive element C3 engage (see Table 2). On the other hand, for a transmission capable of only single clutch-to-clutch shifts, a similar downshift from eighth to third gear would require two gear shifts, such as 8-5-3 or 8-7-3 shift events. Such a process delays the response time and fails to meet the target value (item C in Figure 6). To achieve the equivalent response of a single clutch-to-clutch shift event for all downshifts, the new powertrain control system manages both the timing and shock control for disengaging two elements and engaging two elements.

For a dual clutch-to-clutch shift, the upper limit of the AT output torque change is decided by the torque capacity of C2. This relationship is used to determine the engagement force of a reactive element while C2 is being disengaged so as to minimize shift shock. The relationship between the torque capacity of each engagement element and the output torque is shown by Formula (1)-(15). To achieve a quick response from dual clutch-to-clutch shifts, it is necessary to change reactive force elements faster than changing the input from C2 to C1, so the clutch-to-clutch timing of the reactive elements must be quicker than the normal timing for a single clutch-to-clutch shift event. In order to control the slippage of C2 and reactive elements during engagement transitions, a revolution sensor mounted within the gear train is used along with revolution sensors at the input and output shaft. Output signals from these three sensors enable precise control of the engagement timing for the involved friction elements to achieve the required response characteristics. An example showing characteristics of a 6-3 direct shift is shown in Figure 10, and an example of an 8-3 direct shift is shown in Figure 11. In comparison to two, consecutive shift events, a substantial reduction in gear stage transition time can be realized using the above stated method, while still maintaining the high quality shift characteristics of a single clutch-to-clutch shift.

Notes:

$d\omega_{in}/dt$: Angular acceleration of each member.

T_*: Clutch torque effect on each member.

Value ρ_*: The number of each planetary gear ratio.

Value I_*: Inertia moments of each member.

[basic kinetic equations]

$$T_{in} = I_{input} * d\omega_{in}/dt + T_{Cf} + T_{C4} + T_{C2} \quad (1)$$

$$T_{Rf} = T_{Cf} / (1\text{-}\rho 1) \quad (2)$$

$$T_{Rf} = T_{C3} + T_{C1} \quad (3)$$

$$T_{out} = T_{Sm} + T_{Sr} + T_{Cr} \quad (4)$$

$$T_{Cr} = T_{C2} - T_{B2} - T_{F1} \quad (5)$$

$$T_{C3} + T_{C4} + T_{B1} = T_{Sm} \quad (6)$$

$$T_{C1} = T_{Sr} \quad (7)$$

$$T_{Rm} = -T_{Sm} / \rho 2 \quad (8)$$

$$T_{Rr} = T_{Sr} / \rho 3 \quad (9)$$

$$T_{out} = T_{Rm} + T_{Rr} \quad (10)$$

[e.x. condition of 8-3 downshift]

$$T_{C1} = T_{C4} = T_{B2} = T_{F1} = 0$$

from (1)-(10)

$$T_{in} = I_{input} * d\omega_{in}/dt + (1\text{-}\rho 1) * T_{C3} + T_{C2} \quad (11)$$

$$T_{out} = T_{C3} + T_{B1} + T_{C2} \quad (12)$$

$$T_{out} = -(T_{C3} + T_{B1}) / \rho 2 \quad (13)$$

Condition of shift progression

$$d\omega_{in}/dt > 0$$

from (11)

$$T_{C3} < 1 / (1\text{-}\rho 1) (T_{in} - T_{C2}) \quad (14)$$

(The relation of each torque capacity.)

from (12)(13)

$T_{out} = 1 / (1+ \rho\ 2) * T_{C2}$ (15)

(AT output torque is relative to C2 torque capacity.)

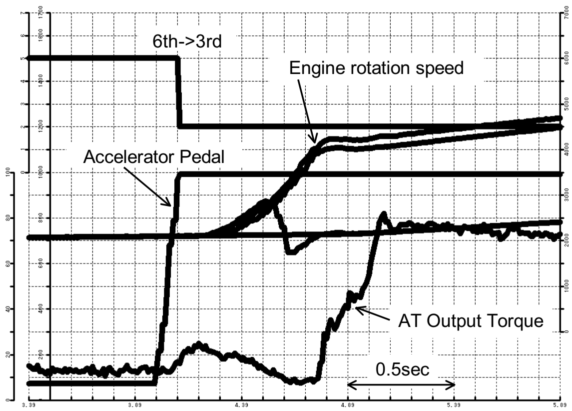

Figure 10: 6th->3rd Down shift.

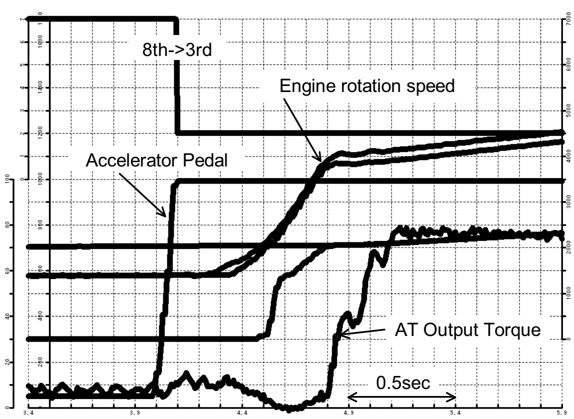

Figure 11: 8th->3rd Down shift.

FUEL ECONOMY IMPROVEMENT

The high number of gear steps and high capacity torque converter allow the AA80E to use an rpm region of the engine with better combustion efficiency at the time of acceleration. Combined with improvements in the powertrain control system, this has enabled the LS460 to realize further gains in fuel economy.

EXPANSION OF FUEL CUT RANGE

To perform a fuel cut during deceleration is effective for improving fuel economy. During deceleration, the powertrain control system can command a fuel cut from fifth through eighth gear, expanding the vehicle speed range where fuel cut is enabled. The AA80E maintains this fuel cut even during a coast down shift event. Moreover, by improving the response of the lock-up control when the driver releases the accelerator, the time from accelerator off to the start of the fuel cut is reduced by 1 second or more, and the vehicle speed when fuel cut is canceled is lowered seven km/h or more than the previous model. Combined, these changes have achieved expansion of the fuel cut activation range and improvement in deceleration response. Figure 12 illustrates these improvements.

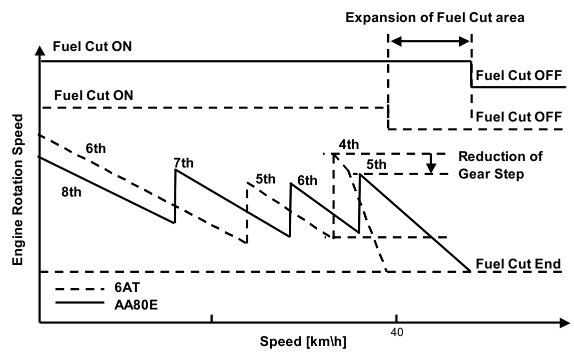

Figure 12: Coast Down Shift Control with Fuel Cut

NEUTRAL CONTROL

Neutral control is a control strategy to reduce fuel consumption during the idle condition, when energy is wasted in the form of heat in the T/C. In the LS460, this control is accomplished by weakening the engagement force of the C1 input clutch to bring the turbine speed of the T/C closer to that of the pump (engine) speed. This alone can improve the fuel economy by 10 % when the vehicle is at a standstill. A problem with neutral control in the past has been the acceleration response at restart; however, through precise control of the linear solenoids in the valve body and the engine torque level, restart characteristics without a loss in performance have been realized.

LOCK-UP AND FLEX LOCK-UP CONTROL SYSTEM

Toyota's Flex Lock-Up Control allows the clutch in the torque converter to maintain a partially-engaged position, enhancing fuel efficiency and increasing the lock-up clutch's operation range. The AA80E is capable of performing this control from the fourth through eighth gear stage. The range of full lock-up has been expanded as well, starting as low as fifth gear. To maintain high quality shift characteristics for an upshift event during full lock-up, the control system momentarily adopts flex lock-up control.

Together, these developments have improved the overall fuel economy by 6.5% over the previous six-speed automatic transmission. This is illustrated in Figure 14.

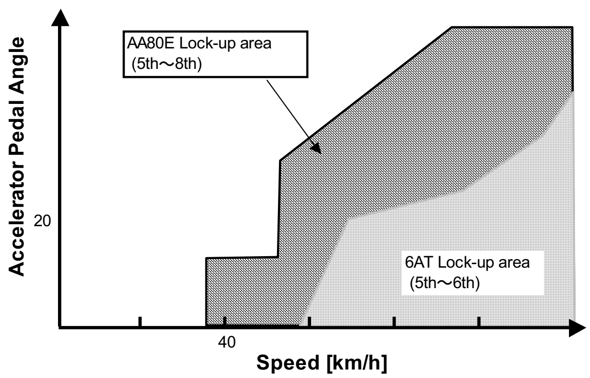

Figure 13: Lock-up Area

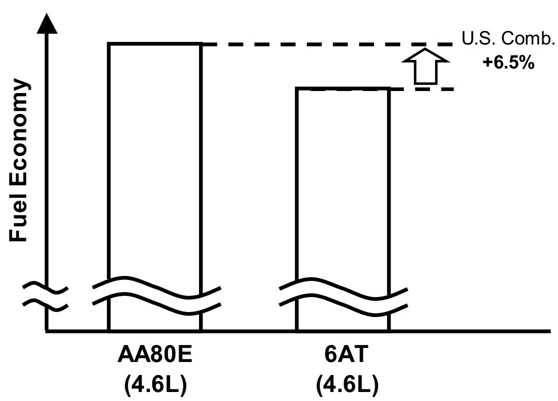

Figure 14: Comparison of Fuel Economy

IMPROVEMENT OF INITIAL QUALITY

In order to improve the initial quality of shift characteristics, variation must be minimized. The LS460 aims to amaze customers from the first drive by using manufacturing data to minimize variation in electrical hardware. Figure 15 shows the main contributors to variation (in terms of oil pressure) during an upshift event in gradual application of the accelerator. In the development process it was found that the top contributors included variances due to the i-P characteristics of both the linear solenoids and the ECU. Based on these results, a plan for reducing the effects of the variations was implemented as follows.

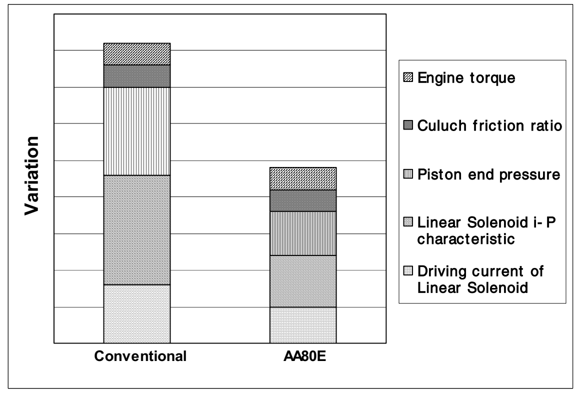

Figure 15: Affect of Variations

Accurately determining the ECU characteristic- During the manufacturing process of the ECU, the ECU is tested in both high and low temperature chambers. The amperage variations due to temperature are used to determine correction values which are then written to the ECU. Using this process, the effect of variations and temperature on the drive current has been reduced by up to 70%.

Reducing variations in effective pressure at the time of clutch engagement- In order to minimize variations in the effective pressure of a clutch which directly influence its torque capacity, the variation in i-P characteristics of the linear solenoids and piston end pressure, must be compensated for. At the transmission plant, the i-P characteristics are measured by a valve body tester, and piston end pressure is measured by a clutch pack tester for each engagement element. The correction values are calculated with the results from these tests,. A bar code reflecting the correction values is then added to the AT body. At the final assembly plant, the correction values in the bar code for each specific AT are written to the matching vehicle ECU. By doing this, minimal effect to the engagement pressure due to variation in the valve body is achieved. In addition, upon completion of each AT, an inspection pattern is performed while applying the calculated correction values, and the shipping judgment is based on a strict standard for the shift quality.

The differences in i-P characteristics of the ECU's map (ex. based on the nominal linear solenoid) and valve body lead to inaccurate valve body pressure for a pressure request of the ECU. In reference to Figure 16, assuming the i-P characteristic 1 of the ECU's map and the i-P characteristic 2 the valve body do not coincide, a drive current from the ECU that may seem accurate to the ECU will still result in inaccurate piston pressure. To resolve this, the two i-P characteristics are combined to calculate a final correction values.

Further, in order to improve the correction accuracy, the same calibration equipment is used for ammeters in the AT factory, linear solenoid factory, and ECU factory, and quality improvement activities are regularly performed with related suppliers.

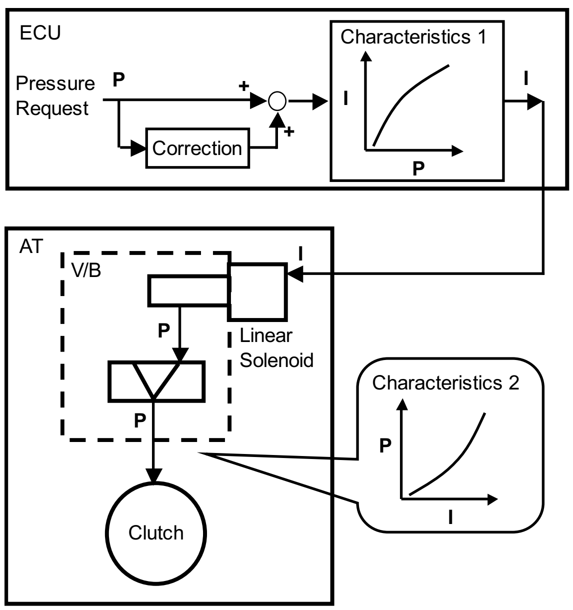

Figure 16: Differences in i-P characteristics

CONCLUSION

This paper has described the basic architecture of the AA80E, as well as the new powertrain control system behind it. The key developments in this powertrain have allowed significant improvements in the areas of dynamic performance, fuel economy, and initial quality. By employing these developments in other powertrains and vehicles, Toyota aims to continue offering driving excitement to its customers, but with minimal environmental impact.

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