Jatco

The mission is passion.

JATCO TECHNICAL REVIEW

NO.20

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From DX to J.A.R.V.I.S.

Tomoyoshi SATO
Executive Vice President

My first encounter with digital technology goes back to my student days when I was using the boundary element method to analyze the fracture mechanism of steel bars. At that time computers ran FORTRAN programs and data were read into the machine using punch cards with perforated holes representing the data. Entering data was an arduous task because it required an enormous number of cards. It was really a nightmare if I happened to drop the cards while carrying them because it was hard to rearrange them back into the correct order.

Nonetheless, one thing that I felt in those days was that computers could do anything and that nothing was impossible if data were converted into information. Today even mobile phones have vastly more computing power than computers did at that time. I feel that an era when anything is possible is approaching.

Nowadays when I envision the future and hear the expression digital transformation (DX), I imagine the Just A Rather Very Intelligent System (J.A.R.V.I.S.) from Iron Man from the well-known Avengers movies. In response to requests from the protagonist, Tony Stark, this software program promptly calls up knowledge from around the world, analyzes it immediately and presents it as his own proposal. It sometimes makes Tony a cup of coffee to get in the mood and is scolded by him on the other hand if it tastes bad. All of our actions are governed by information. Computers are continually evolving with nothing being impossible for them in the world of information. I feel that the J.A.R.V.I.S. world is not so far off. Just as an aside, in the end though J.A.R.V.I.S. will take on a human-like form.

There are three major activities that I want to undertake now in aiming for that type of DX form.

The first one is to link digitization, which is changing the present, to digitalization that will create the future. If the relationships and continuity of data can be seen, it will lead to signs and predictions and the discovery of new wisdom. If digital twins can be achieved through the correlation of 3D data with physical realities, it may enable the discovery of new value. It is vital to continually review conventional wisdom and routine operations, enhance efficiency dramatically and also find new value and new wisdom.

The second one is to create our own technologies. Our very strengths at JATCO lie in the concept of three actualities, i.e., the actual place, the actual object and the actual condition. We are the only ones who can create new wisdom and value based on these three actualities. I believe that is precisely the source of our competitiveness and it leads to value that is also valid outside the company. For example, in the case of robotic process automation (RPA), we already have over 100 robots at work, all of which have been manufactured in-house. All of them are continuing to produce new wisdom and value. I think they are capable of creating even higher value for the very reason that we made them ourselves focusing on our own realities.

The third one is to develop the people and systems for supporting the first two activities. This fiscal year we organized a Promotion Digital Innovation Department. However, its members serve as assistants behind the scenes. The true principal actors are the divisions and departments responsible for the actual workplaces, actual objects and actual conditions. The members of this new department have large roles to play in supporting the DX of every division and department, in making clear the tools to be unified internally, in assisting the implementation of new digital technologies and in planning measures for improving the DX levels of the employees. Although few in number, this team is responsible for leading the entire company as they outline our DX form for the future.

Under the leadership of the Promotion Digital Innovation Department, I want to promote digitalization on our own as much as possible through these three activities and discover new wisdom and value befitting JATCO.

Everyone is well aware that the automotive industry is currently undergoing a profound transformation that occurs once in 100 years. At JATCO, we see this transformation as an excellent opportunity to provide new wisdom and value to our customers. This will involve the challenge of making ourselves more thoroughly familiar not only with mechatronics but also with electronics and pursuing digitalization of our products and services. As I mentioned earlier, by linking reality to information as is done by J.A.R.V.I.S., we will continue to provide our customers with JATCO's newly created value. This is what I want for our products that we put on the market. In the process of envisioning and dreaming of the kind of future described here, I would like to direct JATCO's digital transformation as we move forward.

Development of a new 9-speed automatic transmission for rear-wheel-drive vehicles

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Summary

JATCO launched production of a new 9-speed automatic transmission for rear-wheel-drive vehicles in September 2019. Since 2008, JATCO had been supplying a 7-speed automatic transmission for use on rear-wheel-drive vehicles, but higher environmental performance has been required of conventional transmissions in recent years. The new 9-speed automatic transmission was developed with the aim of achieving ultimate efficiency and ultimate response. In addition to environmental performance, driveability and vehicle mountability were also substantially improved.

1. Introduction

The new 9-speed automatic transmission (9AT) for rear-wheel-drive vehicles was developed around the themes of "ultimate efficiency and ultimate response". As vehicle manufacturers everywhere shift to transmissions with more speeds in pursuit of higher fuel economy, JATCO ultimately selected a 9-speed unit.

Adding more speeds enables wider ratio coverage for improved launch acceleration performance and a lower engine speed during high-speed cruising for improved fuel economy. On the negative side, the number of planetary gears, multi-plate clutches and other component parts increases, making the structure more complex and larger. Naturally, increasing the number of elements involved in transmitting torque also tends to worsen power transmission efficiency if nothing is done about it. In addition, shift patterns markedly increase, which can lead to more shift busyness and deterioration of response. In short, although there are advantages for launch acceleration and high-speed-cruising fuel economy, adding more speeds is meaningless unless efficiency and response are improved.

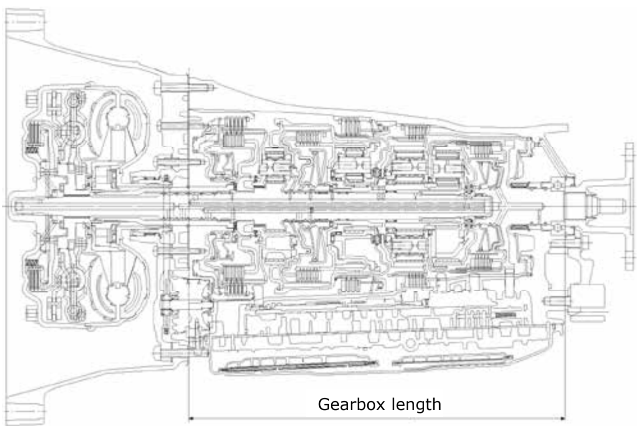

Fig. 1 Main cross-sectional view

This article describes the technologies adopted to achieve ultimate efficiency and ultimate response.

2. Overview of new 9AT

2.1 Development concept

The development concept consisted of the following three points, centered on the key themes of high efficiency and high response as mentioned above.

(1) To improve powertrain efficiency over that of the existing 7-speed automatic transmission (7AT) for rear-wheel-drive vehicles in order to ensure fuel economy competitiveness.

(2) To improve response and shift performance over that of the existing 7AT in order to provide immediate response to customers' driving operations and a smooth, delightful shift feeling.

(3) To suppress increases in size and mass due to the adoption of nine speeds.

2.2 Main cross-section and specifications

Figure 1 is a main cross-sectional view of the newly developed 9AT, and Table 1 lists the basic specifications of the unit in comparison with those of the existing 7AT. One major feature of the new 9AT is that it achieves class-leading ratio coverage of 9.1. In addition, despite having nine speeds, it has four planetary gear sets, the same number as the 7AT, and six shift elements, fewer than the

* Project Promotion Department

Table 1 Comparison of specifications

		7AT	New 9AT
Torque capacity		560 Nm	700 Nm
Torque converter size		UUF 260 mm dia.	UF 260 mm dia.
Lock-up		Torsional damper	Torsional + Pendulum
Gear ratios	Ratio coverage	6.3	9.1
	1st	4.887	5.425
	2nd	3.170	3.263
	3rd	2.027	2.250
	4th	1.412	1.649
	5th	1.000	1.221
	6th	0.864	1.000
	7th	0.775	0.862
	8th	-	0.713
	9th	-	0.597
	Rev	4.041	4.799
Shift control		Mechanical	Park/Shift by wire
Oil pump		Mechanical	Mechanical + Electric
Transmission case		Aluminum	Magnesium
Oil pan		Steel	Plastic
Shift elements		7 Clutches/Brakes	6 Clutches/Brakes
Planetary gear sets		4	4
Weight (wet)		109 kg	99.5 kg
Gearbox length		501.5 mm	439.5 mm

7AT. The gearbox is also shorter in overall length than the 7AT. Moreover, the torque converter is fitted with a pendulum damper and the new unit also weighs less than the 7AT even though the electric oil pump (including the inverter) and the parking brake actuator are built in.

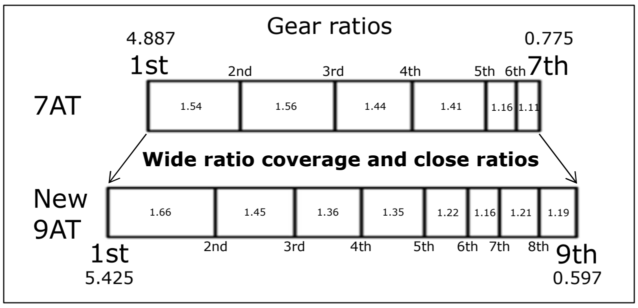

Fig. 2 Gear ratio comparison

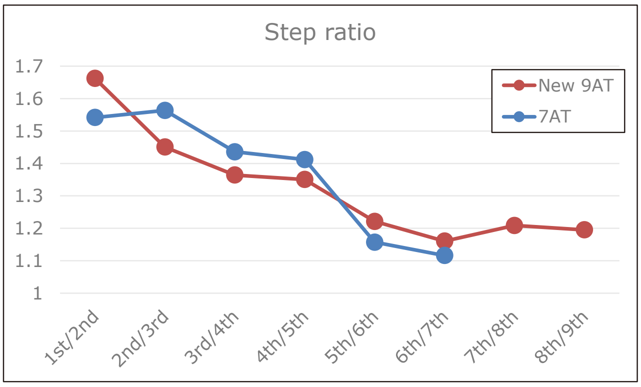

Fig. 3 Step ratio comparison

Figure 2 compares the ratio coverage of the 7AT and the new 9AT, and Fig. 3 compares their step ratios. The ratio coverage of the new 9AT has been expanded on both the Low and High gear ratio sides. The step ratio has been set with smaller differences between the preceding and following speeds than that of the 7AT and with close gear ratios on the frequently-used Low gear ratio side. This achieves delightfully rhythmical shifting during launch acceleration.

3. Technologies adopted to achieve the development concept

3.1 Contributions to higher efficiency

• Full bearing support structure

All of the rotating elements are positioned on the long input shaft. The support structure has been changed from the previous bushings to a full bearing support structure for reducing friction (Fig. 4).

• Pendulum damper

A pendulum damper has been adopted inside the torque converter in addition to the existing torsional damper.

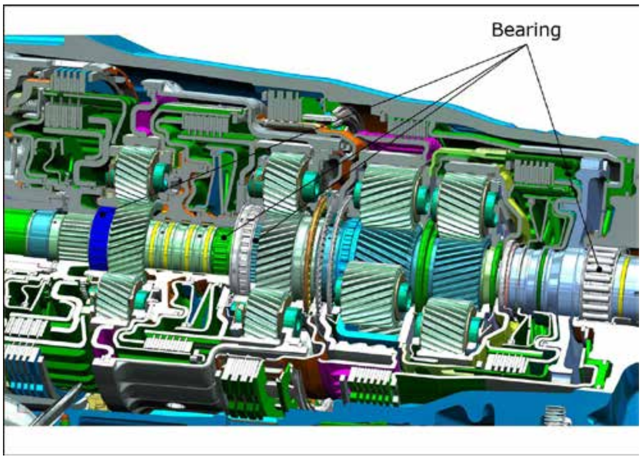

Fig. 4 Full bearing support structure

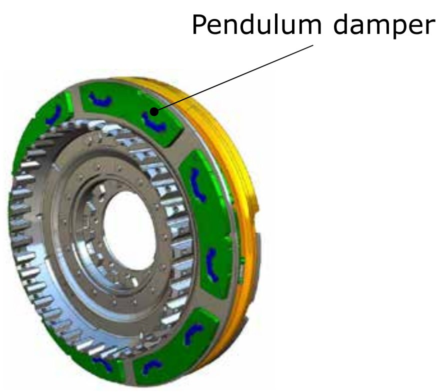

Fig. 5 Pendulum damper

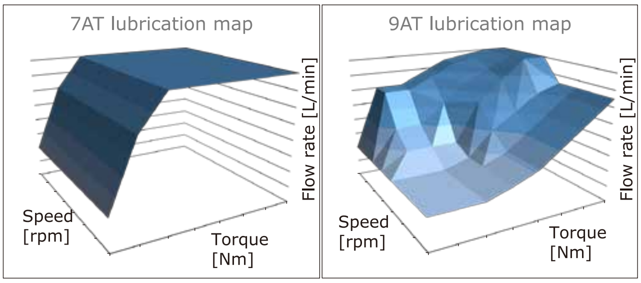

Fig. 6 Lubrication map comparison

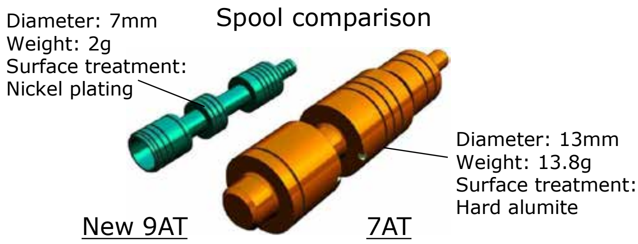

Fig. 8 Spool comparison

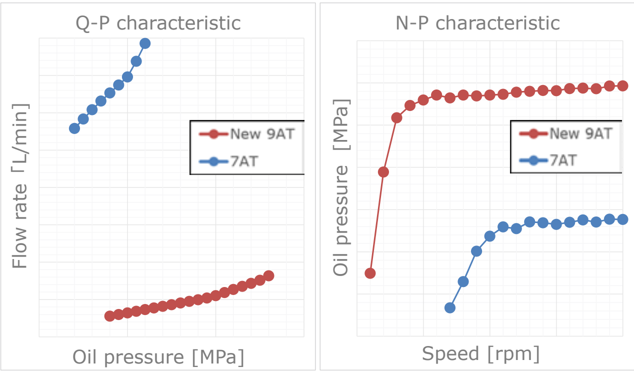

Fig. 7 Comparison of hydraulic characteristics

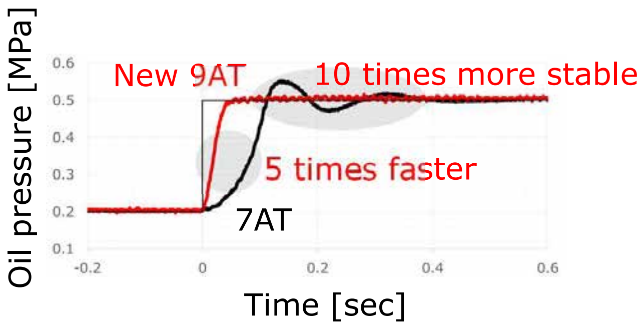

Fig. 9 Hydraulic response comparison

This has improved noise, vibration and harshness (NVH) performance and enabled expansion of the lockup region (Fig. 5).

• Waved drive plate

All shift elements adopt a waved drive plate that reduces drag friction when pulled away from the driven plate during disengagement. It also ensures durability in the high speed range owing to the optimum lubricant flow rate setting described below.

• Optimum lubricant flow rate setting

A lubricant pressure solenoid and a pressure regulator were adopted so that the lubricant flow rate can be controlled unrelated to the line pressure. This makes it possible to regulate the necessary lubricant flow rate to match the driving situation and achieves both a friction reduction and durability (Fig. 6).

3.2 Contributions to improving response

• Reduction of hydraulic circuit leakage

The clearance between the control valve bore and spools was made smaller for the purpose of reducing the amount of fluid flow leakage. The necessary level of hydraulic pressure is obtained at a low flow rate in combination with a small vane oil pump driven by a chain from the input shaft. This enables efficient hydraulic pressure control from the low speed range, thereby improving response (Fig. 7).

• Improvement of clutch control pressure response and stability

A lightweight, small-diameter nickel-plated spool was adopted for each clutch valve to enable variable dither control in all pressure ranges. It also dramatically improves response and stability (Figs. 8 & 9).

3.3 Contributions to size and weight reductions

The technical measures noted below were among those adopted to reduce the size and weight of the new 9AT, thereby suppressing increases in size and weight relative to the existing 7AT.

• Weight reduction through material substitution

  • Magnesium alloy case
  • Plastic oil pan
  • Aluminum bolts

• Use of high-density layout for shortening overall length (Fig. 10)

  • Thin-walled, press-formed parts made of high tensile strength steel sheet
  • Rotation detection by magnetic encoder

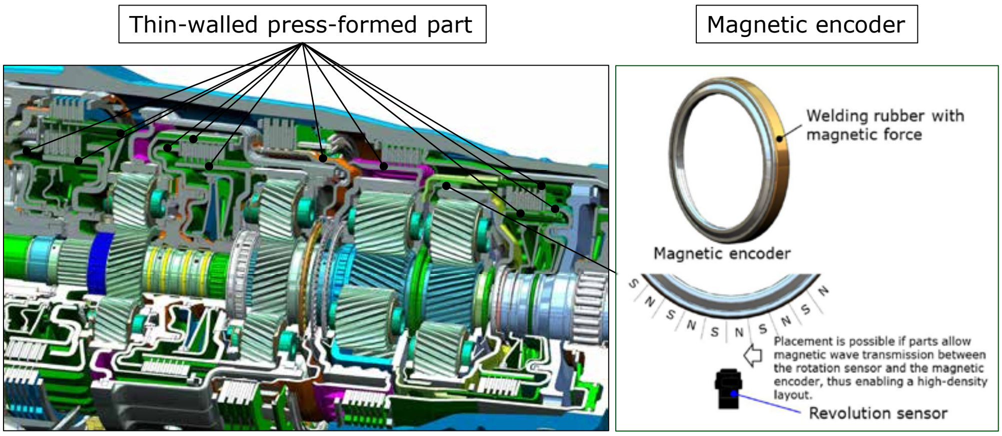

Fig. 10 Techniques for shortening axial length

4. Conclusion

Automotive transmissions today must be developed to provide higher power transmission efficiency. That was a prerequisite condition for the addition of more speeds to create the new 9AT described here.

Moreover, it was also essential to improve response in order to take full advantage of higher efficiency. The ultimate efficiency and ultimate response targets set for the newly developed 9AT were attained through the adoption of the technical measures described in this article.

There were many technical measures and engineering methods that JATCO adopted for the first time, thus providing a stockpile of technologies that can be fed back for developing and manufacturing new products in the future.

In future work, efforts will be made to further improve the competitiveness of conventional engine-AT combinations and also to prepare for a shift to electrification.

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Development of a torque converter for a new 9-speed automatic transmission for rear-wheel-drive vehicles

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Summary

A torque converter was newly developed for a new 9-speed automatic transmission for use on rear-wheel-drive vehicles for the purpose of improving the efficiency of the transmission. Various improvements were made compared with the torque converter of an existing 7-speed automatic transmission used on the same vehicle type. A pendulum dynamic vibration absorber was adopted as a damping mechanism in addition to the torsional damper. A hydraulic circuit structure for effectively cooling the lockup clutch friction material was adopted along with a structure for reducing the surface pressure of the friction material. As a result, quietness and anti-shudder durability were improved while expanding the region of lockup operation. This article describes the details of the newly developed torque converter.

1. Introduction

Various technical measures were taken to improve the vehicle fuel economy obtained with a new 9-speed automatic transmission (9AT) for rear-wheel-drive vehicles. More speeds were added, the vehicle speed for initiating torque converter (TC) lockup (LU) was lowered, and a low-viscosity automatic transmission fluid (ATF) was adopted for reducing friction.

The adoption of these technical measures made it necessary to solve two principal TC technical issues. One was to improve quietness in the speed range with large engine torque fluctuation owing to the lower vehicle speed for LU operation. The other was to improve the anti-shudder durability of the LU clutch friction material in connection with the use of a low-viscosity ATF different from existing transmission fluids.

This article describes the efforts undertaken to address these issues.

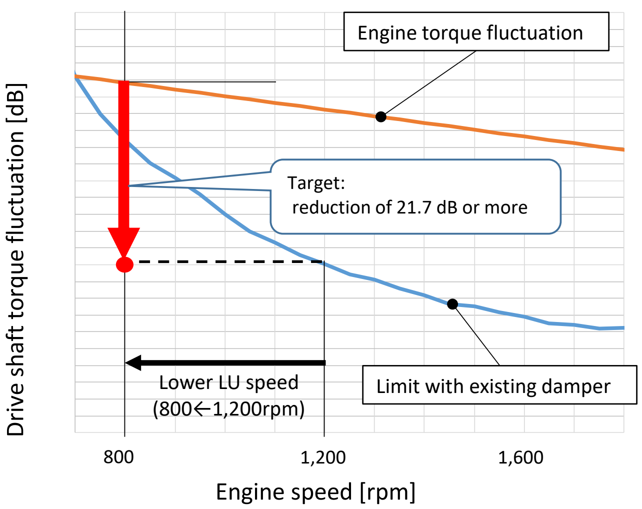


Fig. 1 Quietness target

2. Performance issues and solutions

2.1 Quietness

The reduction of the vehicle speed for LU operation involves using a low engine speed range with large torque fluctuation. This worsens booming noise, characterized as a vibration phenomenon producing a relatively low frequency noise with an oppressive feeling among various interior noises.

The engine speed for initiating LU operation of the new 9AT was changed to 800 rpm from 1,200 rpm of the existing 7-speed automatic transmission (7AT) for use on rear-wheel-drive vehicles. The lower speed increases the torque fluctuation level in the drive shaft, representing a typical characteristic of booming noise. In order to achieve the quietness target, it was necessary to obtain the same level of drive shaft torque fluctuation as that of the existing unit. To accomplish that, the target set for drive shaft torque fluctuation at 800 rpm was to reduce it by 21.7 dB or more compared with the level of engine torque fluctuation (Fig. 1).

The measures taken previously for the existing TC included increasing the amount of inertia, lowering the

* Hardware System Development Department

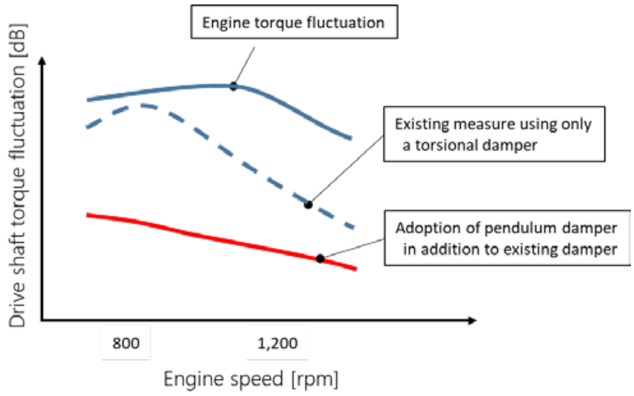


Fig. 2 Effect of adding a pendulum damper

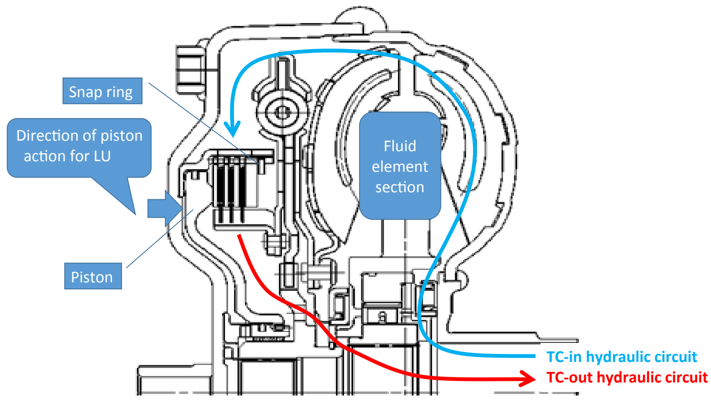
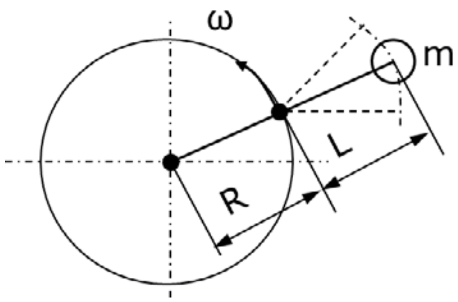


Fig. 4 Structure of hydraulic circuits and LU clutch of existing 7AT



ω: angular velocity
m: mass
R: distance from spindle rotation center to pendulum damper rotation center
L: distance from pendulum damper rotation center to mass center of gravity

Fig. 3 Principal pendulum damper parameters

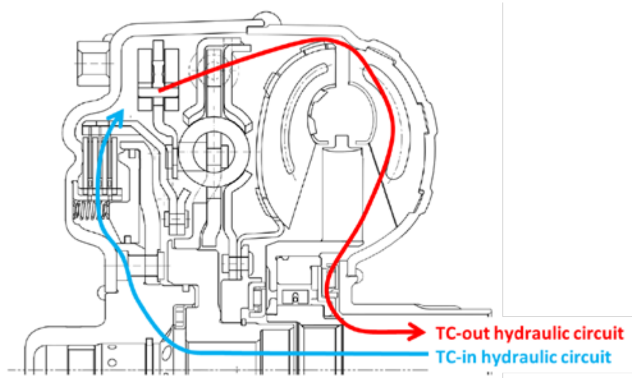


Fig. 5 Structure of hydraulic circuits and LU clutch of new 9AT

stiffness of the torsional damper and reducing hysteresis torque. However, because the quietness target could not be achieved with these measures, it was decided to newly adopt a pendulum damper as a damping mechanism for the new 9AT.

A pendulum damper generates damping force by activating the pendulum in the opposite phase according to the input vibration frequency. Reducing the damper stiffness and other previous measures only served to shift the resonance point, so a pendulum damper was added in addition to the existing torsional damper with the aim of reducing the level of drive shaft torque fluctuation itself (Fig. 2).

Figure 3 shows the principal parameters needed for designing the pendulum damper. In order to apply these principal parameters to the design of the pendulum damper, it was necessary to determine the natural frequency in Eq. (1) and the damping force T in Eq. (2).

$$f = \frac{\omega}{2\pi}\sqrt{\frac{R}{L}} \tag{1}$$

$$T = m(R+L)\omega^2 R\theta \tag{2}$$

The natural frequency is defined to match the number of engine cylinders using the ratio of R to L, where R is the distance from the pendulum center to the center of rotation and L is the distance from the mass to the pendulum center. In order to obtain large damping force, it is necessary to set the mass M and the distances R and L as large as possible.

Given the layout restrictions in this project, an effort was made to achieve the target set for drive shaft torque fluctuation by suitably allocating the damping force and order factors so as to obtain the maximum damping effect.

2.2 Anti-shudder durability

The most important factor for improving the anti-shudder durability of the LU clutch friction material is to reduce the sliding surface temperature when the friction material slides on the mating clutch plate. To reduce the sliding surface temperature of the friction material, the hydraulic circuit structure and the LU clutch structure were improved.

The LU clutch structure was improved so as to enhance the cooling performance of the LU clutch friction material. The existing hydraulic circuit is structured such that ATF heated in the fluid element section is supplied to the LU clutch section. In contrast, the improved hydraulic circuit supplies ATF directly to the LU clutch friction material from the cooled TC-in hydraulic circuit for supplying ATF to the TC.

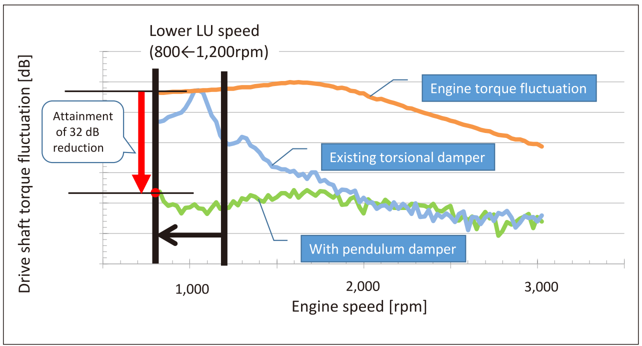


Fig. 6 Effect of quietness improvement

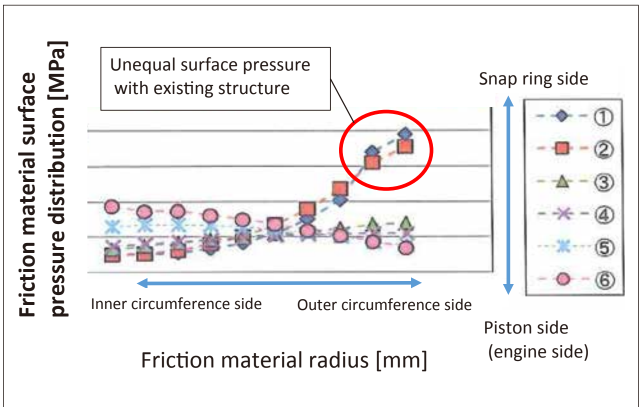


Fig. 7-1 Surface pressure distribution of friction material with existing structure

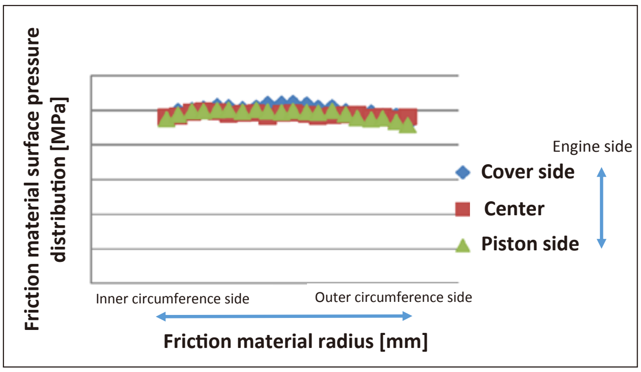


Fig. 7-2 Surface pressure distribution of friction material with new 9AT

In the existing 7AT, the LU clutch is structured such that a snap ring receives the load applied by the LU piston. In the new 9AT, the piston and the TC cover both receive the load at the center of the inner and outer diameter of the LU clutch friction material. The LU clutch structure was improved so that it equalizes the surface pressure on the sliding surface of the LU clutch friction material (Figs. 4 and 5).

3. Effects of improvements

3.1 Quietness

As a result of adding a pendulum damper in addition to the existing torsional damper, torque fluctuation was effectively reduced by 32 dB, exceeding the reduction target of 21.7 dB. This enabled the vehicle speed for LU operation to be reduced, thus contributing to improving vehicle fuel economy (Fig. 6).

3.2 Anti-shudder durability

As a result of making structural changes, the surface pressure distribution on the friction material was improved from that shown in Fig. 7-1 to the distribution in Fig. 7-2. In addition, together with the effect of the improved cooling

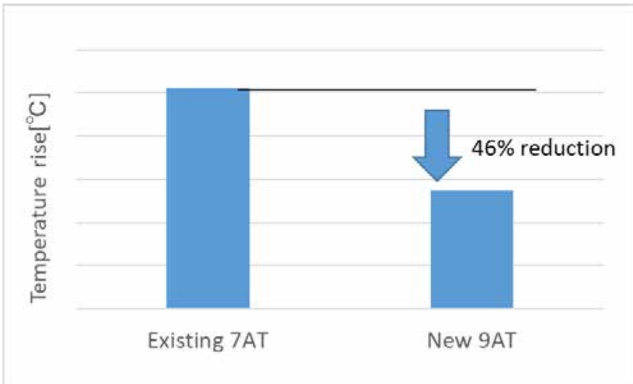


Fig. 8 Effect of reducing friction material temperature

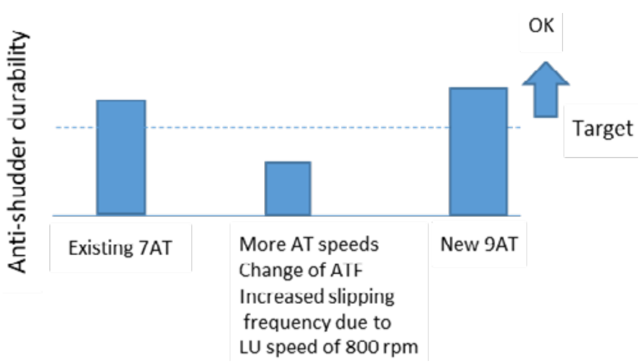


Fig. 9 Effect of improving anti-shudder life

performance, the rise in friction material temperature at the same level of heat generation was reduced by 46% with the improved structure compared with that of the existing AT (Fig. 8).

The reduction of the temperature rise resulted in a difference in ATF performance. As a result, anti-shudder durability was improved over that of the existing 7AT even under conditions where slipping occurs more frequently (Fig. 9).

3.3 Satisfying both performance and layout requirements

Simply combining the pendulum damper for improving quietness and the structural changes to the hydraulic circuit and the LU clutch for improving anti-shudder durability would increase the axial length of the new unit. Accordingly, the pendulum damper was positioned on the outer circumference side for the purpose of securing the desired damping force, while the position of the LU clutch was kept as it was on the inner circumference side. This had the effect of eliminating dead space inside the TC. In addition, with regard to the torsional damper, the parent spring that always receives torsional torque was positioned on the inner circumference, and the secondary spring that is only used against excessive inputs was positioned on the outer circumference. These specifications made it possible to satisfy the required performance without extending the axial length (Fig. 10).

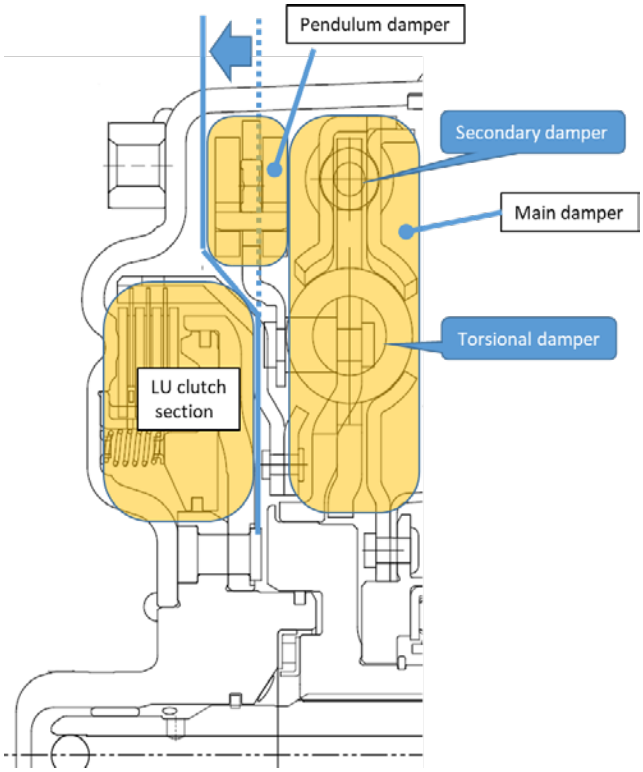


Fig. 10 Layout of new 9AT

4. Conclusion

The TC developed for the new 9AT adopts a pendulum damper to enhance quietness as well as improved structures for the hydraulic circuit and the LU clutch to improve anti-shudder durability.

The pendulum damper, hydraulic circuit and LU clutch structure designed for this TC represent basic JATCO technologies, and the application of the same structures to other new units is now being considered.

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Control techniques supporting fuel economy and driveability of a new 9-speed automatic transmission for rear-wheel-drive vehicles

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Summary

The new 9-speed automatic transmission for rear-wheel-drive vehicles that went into production in September 2019 incorporates numerous control techniques for eliciting the full benefits of additional speeds. Among them, this article describes control techniques adopted for the first time that contribute to improving fuel economy and driveability. Two driving situations are presented as examples to illustrate the shift control details and the issues that have been resolved.

1. Introduction

More speeds have been added to stepped automatic transmissions in recent years from the perspective of satisfying concerns about the environment and fuel economy while simultaneously providing the desired power performance. However, although adding more speeds can be expected to improve environmental friendliness, fuel economy and power performance, there is also concern that increasing the number of shifts will worsen shift busyness and response, among other adverse effects.

The new 9-speed automatic transmission (9AT) for rear-wheel-drive vehicles has been developed to resolve these issues, and it also incorporates numerous control techniques for eliciting the full benefits of additional speeds.

This article describes two driving situations as typical examples where the shift control techniques incorporated in the new 9AT for the first time contribute to improving fuel economy and driveability.

2. Configuration of new 9AT gear train

This section explains the configuration of the new 9AT gear train. Figure 1 is a skeleton of the new 9AT, and the chart in Fig. 2 shows the operation of the shift elements. The shift patterns of each gear are configured such that not only are one-gear upshifts and downshifts from the current gear possible, but also skip-shifting such as 1st gear⇔3rd gear, 3rd gear⇔5th gear, 5th gear⇔7th gear, 7th gear⇔9th gear can be executed. Shifts are possible by just releasing one element and engaging one element (referred to here as single transition shifts).

The arrows in Fig. 3 connect the shift patterns that can be executed by single transition shifts; the different colored lines indicate gears that are skipped. In addition to shifting between adjacent gears, it is seen that 11 shift patterns are possible by single transition shifts. The control techniques developed to solve the issues mentioned earlier by using these characteristics are explained in section 3.

3. Control techniques for obtaining both fuel economy and driveability

Specific examples are presented to show how the shift control techniques contribute to improving fuel economy and driveability in the following two driving situations. The issues and control measures for addressing them are explained.

1. Downshift when decelerating during fuel cut-off
2. Power-on downshift by depressing the accelerator pedal

3.1 Downshift when decelerating during fuel cut-off

3.1.1 Issue of lengthening fuel cut-off duration when decelerating

For the purpose of improving fuel economy, a longer

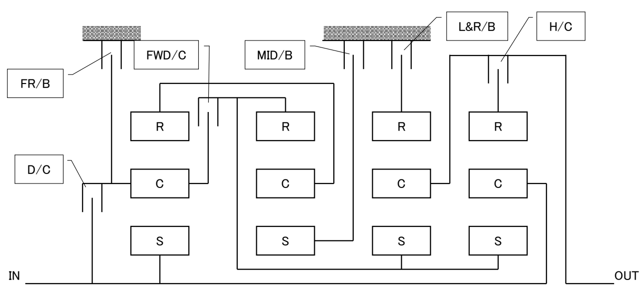

Fig. 1 Skeleton diagram of the new 9AT

* Control System Development Department

Gear	MID/B	L&R/B	FR/B	D/C	H/C	FWD/C
1	●	●				●
2		●		●		●
3	●	●		●		
4	●	●			●	
5	●			●	●	
6				●	●	●
7	●				●	●
8			●		●	●
9	●		●		●	

Fig. 2 Shift elements of the new 9AT

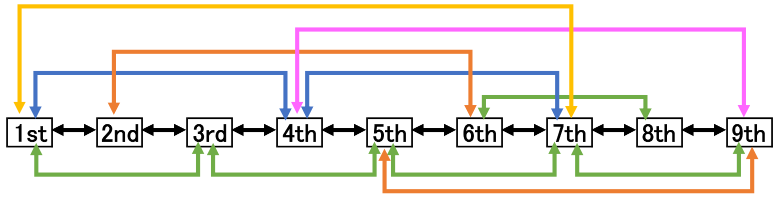

Fig. 3 Single transition shift patterns

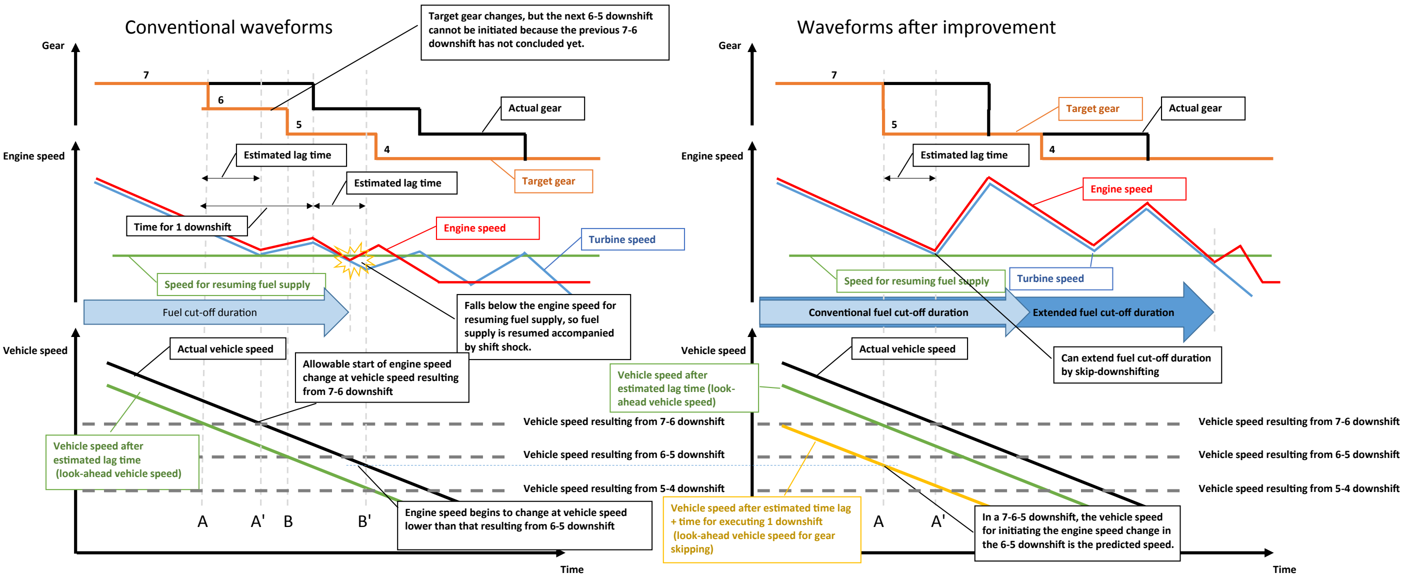

Fig. 4 Downshift during vehicle deceleration

fuel cut-off duration than before is required today during power-off deceleration. In order to continue fuel cut-off to the low gear range, it is necessary to downshift before the engine speed reaches the level for resuming fuel supply.

Adding more speeds, however, increases the number of shifts executed during deceleration. Consequently, in the conventional waveforms shown in Fig. 4, shifting cannot be initiated at the desired timing B depending on the vehicle deceleration rate and shift time. There was concern that fuel supply would be resumed, thus precluding any fuel economy improvement. There was also concern that shift shock might be worsened by an abrupt change in engine torque due to unintentional resumption of fuel supply during shifting.

Therefore, a new shift control technique was adopted to obtain both continuation of fuel cut-off and resumption of fuel supply at the desired timing, as explained below.

3.1.2 Attainment of longer fuel cut-off duration during deceleration

The downshift timing during deceleration has previously been commanded taking into account the lag time between the issuance of the shift command and the onset of the change in engine speed induced by downshifting. The control adopted this time for the 9AT also estimates whether the change in engine speed due to the next downshift can be achieved at the desired timing. If it is judged that it cannot be achieved at the desired timing, the transmission is commanded to skip-downshift aggressively such as by executing 9-7 and 7-5 shifts.

Skip-downshifting enables fuel cut-off to be continued without the engine speed unintentionally dropping to the level for resuming fuel supply. It also prevents worsening of shift shock caused by interference between shifting and fuel supply resumption.

The specific control method is explained here. As indicated by the conventional waveforms in Fig. 4, the conventional control technique calculates the estimated lag time from a 7-6 shift command to the onset of the change in engine speed and also the vehicle speed (referred to here as the look-ahead vehicle speed) after the estimated lag time based on the current rate of vehicle deceleration. The start of the 7-6 downshift is commanded at timing A where the look-ahead vehicle speed drops below the speed that results from the 7-6 downshift. If the 7-6 downshift can be initiated at timing A as commanded, the engine speed

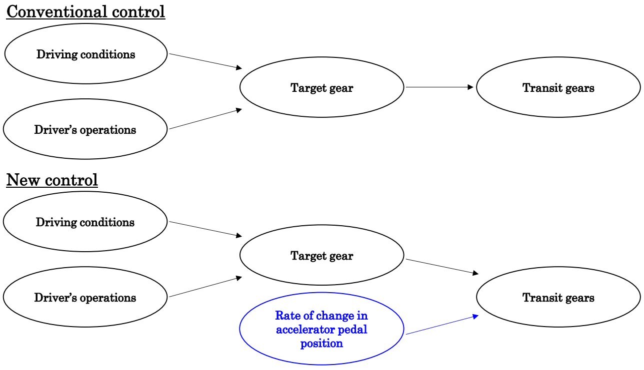

Fig. 5 Determination of transit gears

begins to change at timing A' where the actual vehicle speed is the speed produced by the 7-6 downshift. However, in the case of sudden deceleration or for some other reason, the subsequent 6-5 downshift cannot be initiated at the targeted timing B because the previous 7-6 downshift has not concluded yet. Accordingly, the vehicle speed when the engine speed begins to change is lower than the speed resulting from the 6-5 downshift.

As indicated by the improved waveforms in Fig. 4, in addition to calculating the look-ahead vehicle speed following the estimated lag time, the new control technique also estimates the vehicle speed (referred to here as the look-ahead vehicle speed for gear skipping) after the estimated lag time plus the time for executing one downshift, as noted in the conventional waveform in the figure. At timing A commanded by the conventional control technique for initiating the 7-6 downshift, if the look-ahead vehicle speed for gear skipping is below the speed resulting from the 6-5 downshift, in the event 7-6 and 6-5 downshifts are executed, it means that at the time of the 6-5 downshift the change in engine speed begins when the vehicle speed is below that resulting from the 6-5 downshift. Therefore, in this case the 7-5 downshift command makes it possible to decelerate as far as 5th gear without any resumption of fuel supply. In addition, because a 5-4 downshift can also be executed at the desired timing, fuel cut-off can be continued even during the 5-4 downshift.

As a result, assuming that fuel supply is resumed in 4th gear, the fuel cut-off duration can be continued and extended as indicated by the improved waveforms in Fig. 4.

3.2 Power-on downshift by depressing the accelerator pedal

3.2.1 Effect of adding more speeds on improving driveability and issues involved

Adding more speeds has the effect of making it possible to provide the fine-tuned driving force demanded by drivers. However, the transmission must pass through more gears than before to reach the driving force that drivers want. This poses the issue that a lag may occur before the driving force demanded by drivers is reached. The shift control of the 9AT was improved to provide the fine-tuned driving force that drivers demand and without any lag either.

3.2.2 Determination of transit gears

The 9AT incorporates a control technique that optimizes the number of transit gears in order to transition to the desired fine-tuned driving force without any lag. As shown in Fig. 5, the conventional control technique determines the optimum target gear based on the driving conditions and the driver's operations; it selects the minimum number of transit gears to be followed for reaching the target gear.

The new control technique refers to the rate of change in the accelerator pedal position when selecting the number of transit gears. Accordingly, when there is a large rate of change in the accelerator pedal position, the number of transit gears to reach the target gear is minimized, thereby enabling a quick transition to the driving force demanded by the driver. When there is a small rate of change in the accelerator pedal position, the number of transit gears to reach the target gear is increased, thus delivering fine-tuned driving force that traces the driving force desired by the driver.

Moreover, this control technique is capable of real-time judgments so as to provide driving force that follows the driver's various accelerator pedal inputs. For example, in shifting from 9th gear to 4th gear, the single transition shift patterns in Fig. 3 show that the following transit gears can be selected: 9-4, 9-5-4, 9-7-4, 9-7-5-4, 9-7-6-5-4, 9-8-7-4, 9-8-6-5-4, 9-8-7-5-4, and 9-8-7-6-5-4. The following are examples related to the accelerator pedal position. As shown in Fig. 6, a 9-4 shift is selected when the driver depresses the accelerator pedal fast and a 9-8-7-6-5-4 shift is selected when the accelerator pedal is depressed gently. When the depression of the accelerator pedal produces a curved pattern, the number of transit gears is reduced in the first half and increased in the second half for a 9-7-5-4 shift sequence. When it produces an arch-shaped pattern, the number of transit gears is increased in the first half and reduced in the second half for a 9-8-7-4 shift sequence.

Because the transit gears are selected in real time in this way to match the accelerator pedal position, the 9AT can provide the fine-tuned driving force demanded by the driver and without any lag.

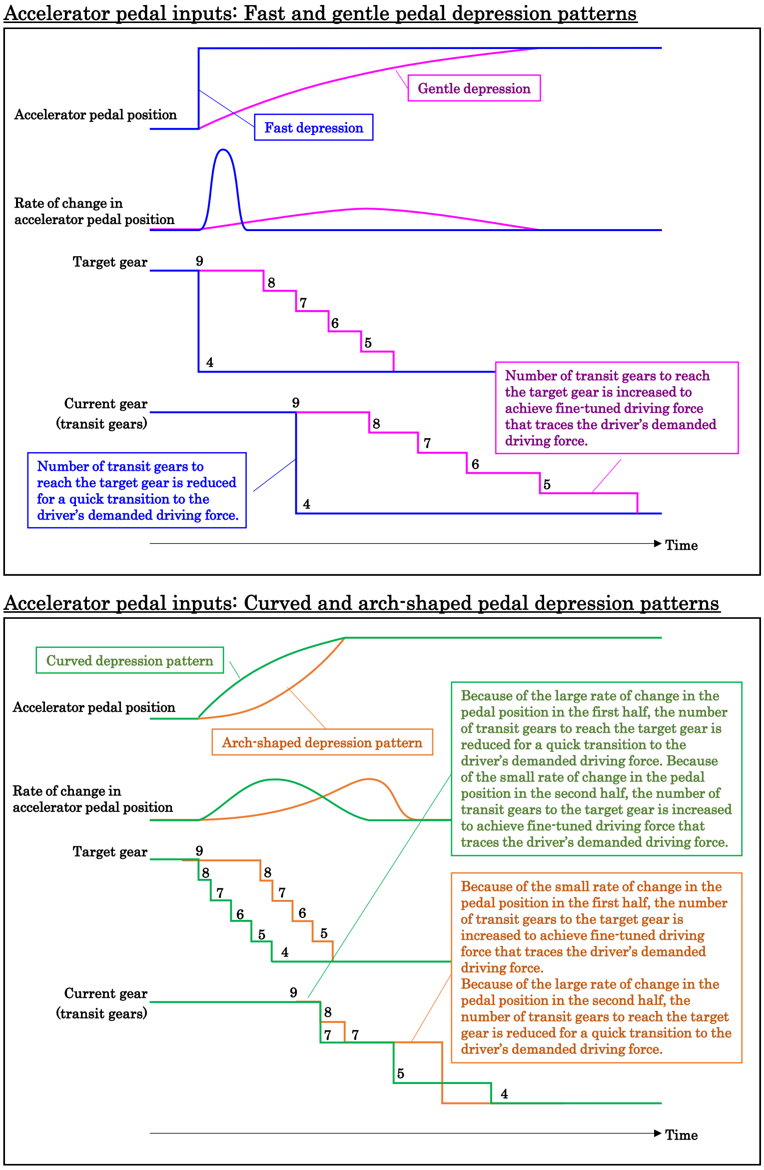

Fig. 6 Accelerator pedal inputs and transit gears

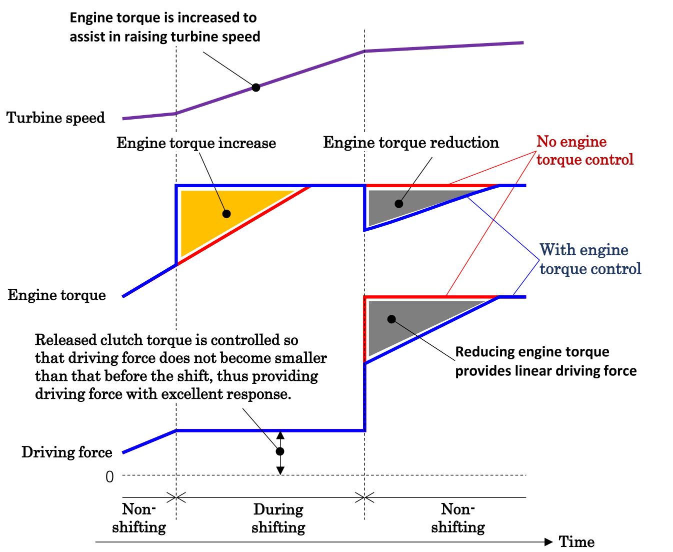

Fig. 7 Provision of smooth driving force with excellent response

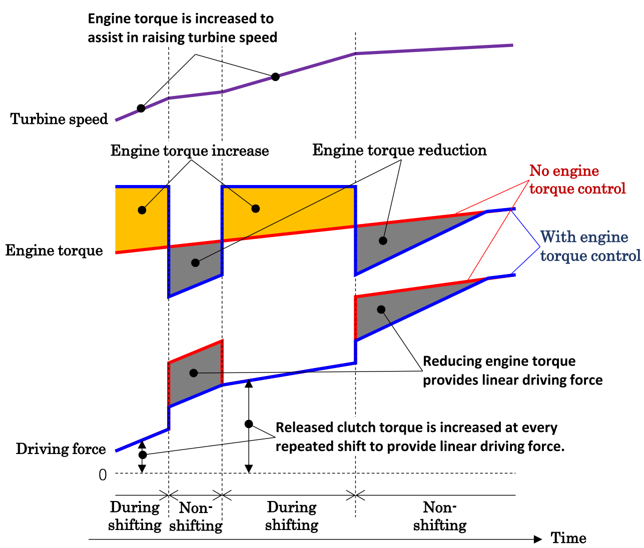

Fig. 8 Provision of smooth, fine-tuned driving force

3.2.3 Determination of clutches and engine torque

As explained in the preceding subsection, there are situations where priority is put on reaching the target driving force and situations where preference is given to providing fine-tuned driving force. The allocation of clutches and engine torque in each case must be optimized in order to obtain driving force matching the objective of the situation.

As shown in Fig. 7, in situations where priority is put on reaching the target driving force, engine torque is increased during shifting to assist in raising the turbine speed. The released clutch torque is also reduced so as not to obstruct the increase in turbine speed, which shortens the shift time. However, if the released clutch torque is reduced too much, driving force will decrease excessively during shifting, giving the driver a feeling of insufficient acceleration. To avoid that, the released clutch torque is controlled so that driving force does not become smaller than it was before the shift. Moreover, to ensure a smooth driving force transition after the completion of the shift, engine torque is reduced for resuming driving force smoothly. This provides smooth driving force with excellent response.

Figure 8 shows an example where preference is given to providing fine-tuned driving force. In this case, the released clutch torque in each shift is controlled so that driving force is increased in a step-like manner every time a shift is repeated. In addition, engine torque is increased during shifting to assist in raising the turbine speed and thereby shorten the shift time. Following the completion of the shift, engine torque is reduced and the driving force between shifts is reduced, thereby decreasing the difference in driving force between the intervals of shifting and non-shifting. This results in smooth, fine-tuned driving force.

3.2.4 Provision of driver's demanded driving force

Figure 9 presents the data measured for a vehicle incorporating the control techniques explained in subsections 3.2.2 and 3.2.3. When the accelerator pedal was depressed rapidly, smooth driving force was obtained with excellent response; when the accelerator pedal was depressed gently, smooth, fine-tuned driving force was obtained. The data confirm that the improved shift control techniques provided the smooth, fine-tuned driving force demanded by the driver and without any lag.

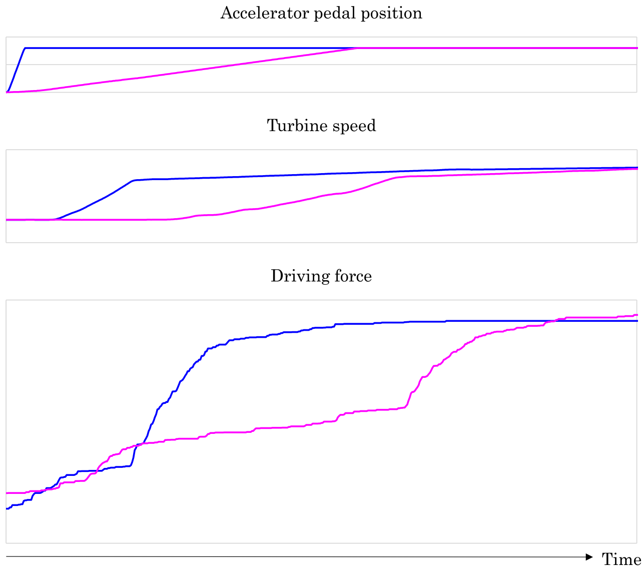

Fig. 9 Provision of driver's demanded driving force

4. Conclusion

The 9AT incorporates many control techniques that resolve the issues of increased shift busyness, worsened response and other undesirable aspects resulting from the addition of more speeds and also more fully elicits the benefits of additional speeds.

Among these techniques, this article focused on downshift control and showed how the adopted methods contribute to improving fuel economy and driveability.

In future work, we intend to develop control techniques that contribute to improving fuel economy and driveability further while also taking into account adaptation to the shift toward electrification.

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Memo

Application of external air temperature estimation to a variable lubrication control system for a new 9-speed automatic transmission for rear-wheel-drive vehicles

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Summary

The new 9-speed automatic transmission for use on rear-wheel-drive vehicles is fitted with a system that enables independent control of the lubricant flow rate. Ascertaining the temperature of the oil flowing in the lubrication circuit is essential in order to optimize the lubricant flow rate to match the driving situation. That requires an estimation of the oil temperature drop in the air-cooled oil cooler. To do that, an engine intake air temperature sensor is used to estimate the external air temperature for determining the oil temperature drop in the oil cooler, which enables variable lubrication control.

1. Introduction

Automatic transmission fluid (ATF) serves various purposes and one of its key functions is lubrication. The existing 7-speed unit is not able to regulate the lubricant flow rate independently, which could become excessive depending on the driving situation, resulting in increased friction.

The new 9-speed automatic transmission (9AT) for rear-wheel-drive vehicles can control the lubricant flow rate independently to reduce friction. The lubricant flow rate is greatly affected by the pressure loss due to the oil temperature drop that occurs in the air-cooled oil cooler. That makes it necessary to control the flow rate by taking into account the magnitude of the oil temperature drop. For the new 9AT, a method of controlling the lubricant flow rate was developed that monitors the magnitude of the oil temperature drop in the air-cooled oil cooler based on an estimation of the external air temperature.

2. Targeted system

The configuration of the lubrication system circuit that was the focus of this study is explained first. As shown in Fig. 1, after the ATF pressure is regulated by the control valve, the oil flows to the torque converter and then to the built-in oil cooler (BIOC). In the BIOC, heat exchange takes place between the ATF and the engine coolant. The BIOC

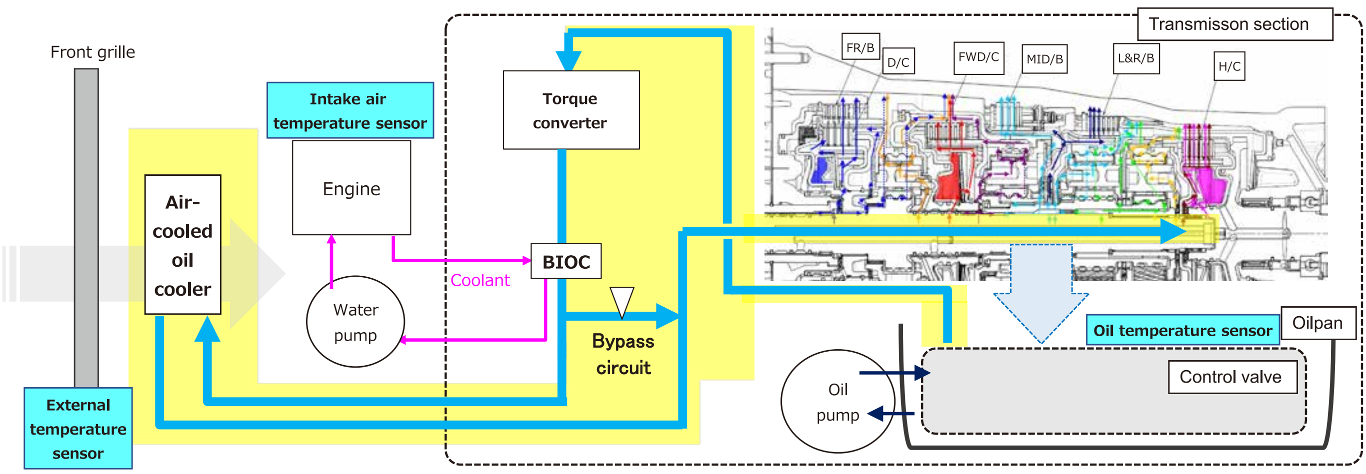

Fig. 1 Cooling and lubrication circuit

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functions as a warmer to warm the ATF when the inlet oil temperature is low and as a cooler to cool the ATF when inlet oil temperature becomes high.

A bypass circuit is provided right after the BIOC. A wax serves to open and close the bypass circuit by using the characteristic that its volume changes according to the temperature. The bypass circuit is open when the temperature of the oil flowing through it is low, and the ATF follows a return circuit back to the transmission without passing through the air-cooled oil cooler. The bypass circuit is closed when the oil temperature is high, so the ATF flows into the air-cooled oil cooler. The ATF is cooled there by the heat exchange that occurs with the external air flowing through the air-cooled oil cooler. The new 9AT is expected to be used on large vehicles such as pickup trucks for the U.S. market, so an air-cooled oil cooler with high cooling performance is necessary.

As shown in Fig. 2, the source pressure for supplying lubricant (i.e., lubricant pressure) is created by a lubricant linear solenoid valve, which is activated by a control signal from the controller, and by a lubricant regulator activated by the output pressure of the solenoid valve. A larger lubricant flow rate is needed to protect and cool the transmission

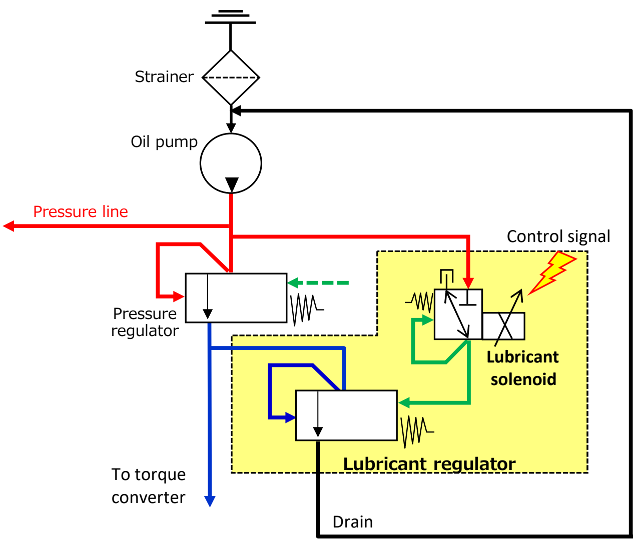

Fig. 2 Lubrication system in control valve

parts in high load driving situations where the transmitted torque and rotational speed are high, but a smaller flow rate is sufficient under low loads. Accordingly, a variable lubrication control system was built for the new 9AT that optimizes the lubricant flow rate by regulating the lubricant pressure according to the driving load.

3. Issues in targeted system

As explained in the preceding section, when the temperature of the ATF flowing through the bypass circuit is low, the bypass circuit is open so the oil does not pass through the air-cooled oil cooler; when the ATF temperature rises, the bypass circuit is closed so the oil begins to flow to the air-cooled oil cooler.

Figure 3 shows the lubrication circuit and the change in the oil temperature after the ATF becomes hot and the bypass circuit is closed. The red line shows the condition where the BIOC functions as a cooler under a high external air temperature. The blue line shows the condition where the BIOC functions as a warmer under a low external air temperature. Because the ATF is heated in the BIOC under this condition, the bypass circuit is closed and the ATF flows to the air-cooled oil cooler.

The magnitude of the oil temperature drop in the air-cooled oil cooler is greatly influenced by the temperature

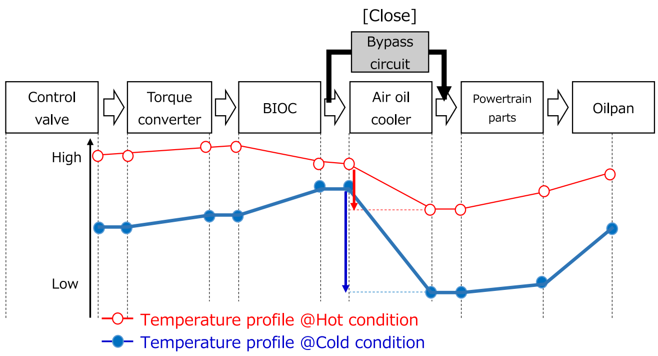

Fig. 3 Oil temperature fluctuation in lubrication circuit

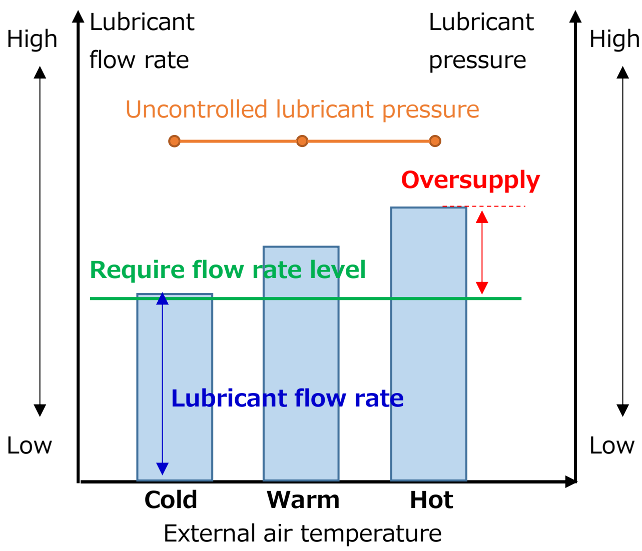

Fig. 4 Uncontrolled lubricant flow rate

Table 1 Decision analysis results

	Installation of new sensor	Diversion of existing sensor to estimate oil temp. at air-cooled oil cooler outlet	
	Proposal (1)	Proposal (2)	Proposal (3)
	Air-cooled oil cooler outlet oil temp. sensor	External air temp. sensor	Engine intake air temp. sensor
Accuracy	○	○	△
Difficulty of control	△	×	△
Sensor installation rate	×	△	○
Judgment	No GO	No GO	**GO**

of the external air flowing through the cooler. The oil temperature drop is larger with a low external air temperature than with a high external air temperature. That tendency is especially pronounced with a large-capacity air-cooled oil cooler.

In short, it was initially thought that only the driving load needed to be considered in the variable lubrication control system. However, it was found that the effects of the magnitude of the oil temperature drop in the air-cooled oil cooler and the resultant pressure loss also have to be considered. If the lubricant pressure is aligned with a low external air temperature condition where a large pressure loss occurs, the lubricant flow rate will become excessive under a high external temperature condition. Accordingly, this kind of situation must also be eliminated (Fig. 4).

4. Development of a solution

Using an estimated external air temperature value would make it possible to ascertain the magnitude of the oil temperature drop in the air-cooled oil cooler and also the resultant pressure loss. Reflecting the results in variable lubricant flow rate control would provide a countermeasure for the issues mentioned above.

4.1 Estimation of external air temperature using engine intake air temperature

The magnitude of the oil temperature drop in the air-cooled oil cooler must be reflected in the variable lubrication control procedure in order to optimize the lubricant flow rate. The oil temperature at the inlet of the air-cooled oil cooler is estimated from the operating temperature of the bypass circuit. Several proposed methods were investigated for estimating the oil temperature at the outlet of the air-cooled oil cooler.

Proposal (1) involved newly installing a temperature sensor to measure the outlet oil temperature directly. Proposals (2) and (3) involved estimating the oil temperature at the air-cooled oil cooler outlet indirectly based on the temperature of the external air passing through the cooler. Table 1 presents the results of a decision analysis (DA) that was conducted from the following perspectives: "accuracy" of calculating the outlet oil temperature, "relative difficulty of constructing a control system" using a sensor, and the "sensor installation rate" on vehicles for applying the system to all relevant vehicles.

The results revealed it would be difficult to adopt proposal (1) from the standpoint of the "sensor installation rate" and proposal (2) from the standpoint of the "relative difficulty of constructing the control system". It was decided to proceed with proposal (3) involving the use of the engine intake air temperature sensor to estimate the external air temperature and then using the result to calculate the oil temperature at the outlet of the air-cooled oil cooler. With respect to accuracy, it will be noted that there are driving situations where the engine intake air temperature and the external air temperature diverge. That divergence has been minimized by adding a temperature correction procedure to the control system.

4.2 Relationship between oil temperature drop in the air-cooled oil cooler and pressure loss

The method of estimating the external air temperature was selected. However, in order to estimate the oil temperature at the outlet of the air-cooled oil cooler, it was necessary to estimate the effects of the external air temperature and oil flow rate in the cooler on the magnitude of the oil temperature drop and the resultant pressure loss. It will be noted that with the bypass circuit closed, the oil flow rate in the oil cooler and the lubricant flow rate are equal. The selected method involves two steps: (1) to estimate the oil temperature drop in the air-cooled oil cooler due to the external air temperature; (2) to estimate the values of the pressure loss in the air-cooled oil cooler and in the following downstream section relative to the oil temperature drop in the oil cooler. They are explained in detail below.

First, the heat exchange characteristic in the air-cooled oil cooler is used to explain the relationship between the oil flow rate in the oil cooler and the oil temperature drop. The heat exchange characteristic E [kW] is defined by the following parameters.

- Oil temperature at air-cooled oil cooler inlet T_{IN} [°C]
- Oil flow rate in air-cooled oil cooler Q_{OIL} [L/min]
- Temperature of air flowing through air-cooled oil cooler = external air temperature T_{AIR} [°C]
- Air velocity in air-cooled oil cooler V_{AIR} [m/s]

The oil temperature at the air-cooled oil cooler outlet T_{OUT} [°C] has the following relationship with the oil temperature drop ΔT_{OIL} [°C] in the oil cooler.

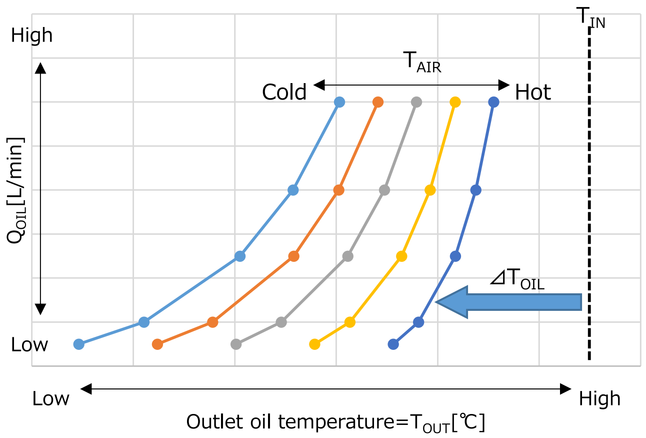

Fig. 5 Estimated oil temperature drop

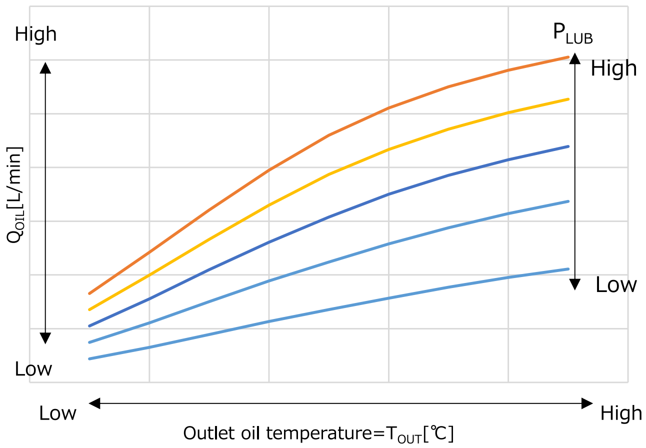

Fig. 6 Estimated oil flow resistance

$$T_{OUT} = T_{IN} - \Delta T_{OIL} \tag{1}$$

Letting η [kJ/kg K] represent the specific heat of the oil, ΔT_{OIL} can be expressed with the following equation.

$$\Delta T_{OIL} = \frac{E(T_{IN},\ Q_{OIL},\ T_{AIR},\ V_{AIR})}{\eta \times Q_{OIL}} \tag{2}$$

From Eqs. (1) and (2), the oil temperature at the air-cooled oil cooler outlet T_{OUT} can be expressed as shown below.

$$T_{OUT} = T_{IN} - \frac{E(T_{IN},\ Q_{OIL},\ T_{AIR},\ V_{AIR})}{\eta \times Q_{OIL}} \tag{3}$$

Figure 5 shows the oil temperatures that were calculated at the outlet of the air-cooled oil cooler based on Eq. (3). The horizontal axis is the oil temperature at the oil cooler outlet T_{OUT} and the vertical axis is the oil flow rate in the oil cooler Q_{OIL}. These temperatures were calculated under a condition of the lowest oil temperature at the inlet of the air-cooled oil cooler T_{IN} and with the bypass circuit closed. The air velocity in the air-cooled oil cooler V_{AIR} was calculated assuming a high driving speed. The blue arrow in the figure indicates the oil temperature drop ΔT_{OIL} in the air-cooled oil cooler. It is seen that the oil temperature at the oil cooler outlet T_{OUT} decreases with a smaller oil flow rate in the oil cooler Q_{OIL} and when the external air temperature is low.

Next, the relationship between the lubricant pressure and the lubricant flow rate is explained based on the pressure loss characteristic in the air-cooled oil cooler. The lubricant flow rate Q_{OIL} can be expressed in relation to the lubricant pressure P_{LUB} as shown in the equation below.

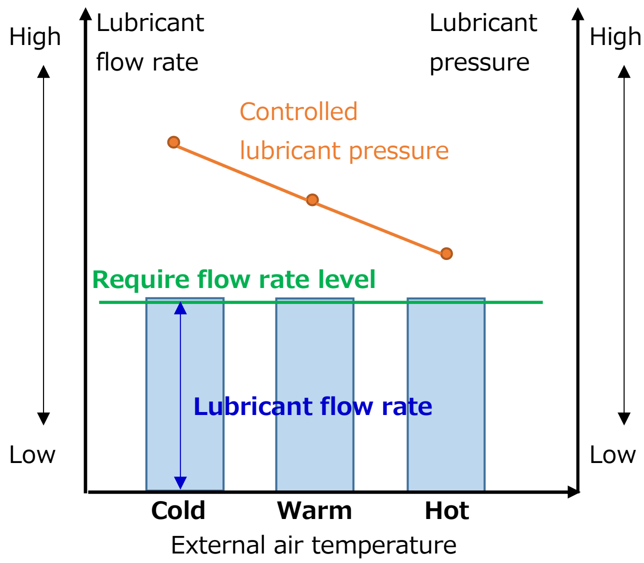

Fig. 7 Controlled lubricant flow rate

$$Q_{OIL} = R_1(T_{IN}, T_{OUT})P_{LUB} + R_2(T_{IN}, T_{OUT})\sqrt{P_{LUB}} \quad (4)$$

where R_1 and R_2 are coefficients of oil flow resistance in the lubrication circuit and are sensitive to T_{IN} and T_{OUT}.

As was done in the case of Fig. 5, by setting T_{IN} constantly at the oil temperature for closing the bypass circuit, Q_{OIL} becomes a function of T_{OUT} and P_{LUB}. The calculated results for Q_{OIL} are shown in Fig. 6 on the same axis as in Fig. 5. It is seen that the pressure loss in the lubrication circuit increases when T_{OUT} is low, so the lubricant flow rate decreases at the same lubricant pressure.

As is clear by looking at Eqs. (3) and (4), Q_{OIL} and T_{OUT} calculated with these equations, respectively, are parameters of the other equations, which means that each one cannot be calculated independently. It is necessary to find a solution as a set of Q_{OIL} and T_{OUT} that satisfies both equations. In this project, a solution under each set of conditions was found by numerical calculation and mapped. The necessary lubricant pressure was then derived by calculating back from the desired lubricant flow rate.

4.3 Attainment of variable lubrication control by estimating the external air temperature

As a result of the development work described above, a variable lubrication control system was built based on the estimated value of the external air temperature. This system provides variable control of the lubricant flow rate that takes into account the oil temperature drop in the air-cooled oil cooler and the resultant increased pressure loss.

The effects of the system are shown in Fig. 7. The lubricant pressure is regulated according to the estimated value of the external air temperature. This enables the necessary lubricant flow rate to be optimally supplied regardless of the external air temperature. Consequently, it is now possible to prevent an excessive lubricant flow rate even in environments with a high external air temperature.

5. Conclusion

- A variable lubrication control system was built based on the estimated value of the external air temperature. The system optimally controls the lubricant flow rate by taking into account the oil temperature drop that occurs in the air-cooled oil cooler depending on differences in the external air temperature. It thus achieves both durability and lower friction.
- An engine intake air temperature sensor is used in estimating the external air temperature. The use of this highly versatile, existing device will enable the new 9AT transmission to be applied to a wide range of vehicle models in the future.

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Memo

Development of a CVT without an electric oil pump to achive a start-stop system achived before the vehicles stops

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Masumi FUJIKAWA*** Jonghwan LEE **** Tatsuo SATO*****

Summary

For a mini-vehicle CVT without an auxiliary transmission, JATCO has developed a start-stop system that does not have an electric oil pump and can be activated before the vehicle stops. The new start-stop system was achieved by discontinuing the electric oil pump and improving CVT shift control in order to satisfy the requirements for vehicle fuel economy and re-acceleration performance. This article describes the CVT technologies that contribute to ensuring re-acceleration performance.

1. Introduction

Directly engaging the torque converter lock-up (LU) clutch while a vehicle is decelerating and cutting off fuel injection to the engine are techniques adopted for continuously variable transmissions (CVTs) to improve fuel economy for reducing carbon dioxide (CO_2) emissions. After the vehicle stops, the fuel supply is suspended by the start-stop system while the vehicle is stationary.

However, the idling of the engine consumes fuel as the vehicle decelerates in the interval from lock-up clutch release to the activation of the start-stop system when the vehicle becomes stationary. In order to reduce such fuel consumption by engine idling, JATCO initiated development of a start-stop system that is activated before the vehicle stops (Fig. 1).[(1)]

The new start-stop system has been applied to the Jatco CVT-S (CVT-S) for use on mini-vehicles to improve vehicle fuel economy and to ensure the desired re-acceleration performance after the engine is restarted. This article describes the development details.

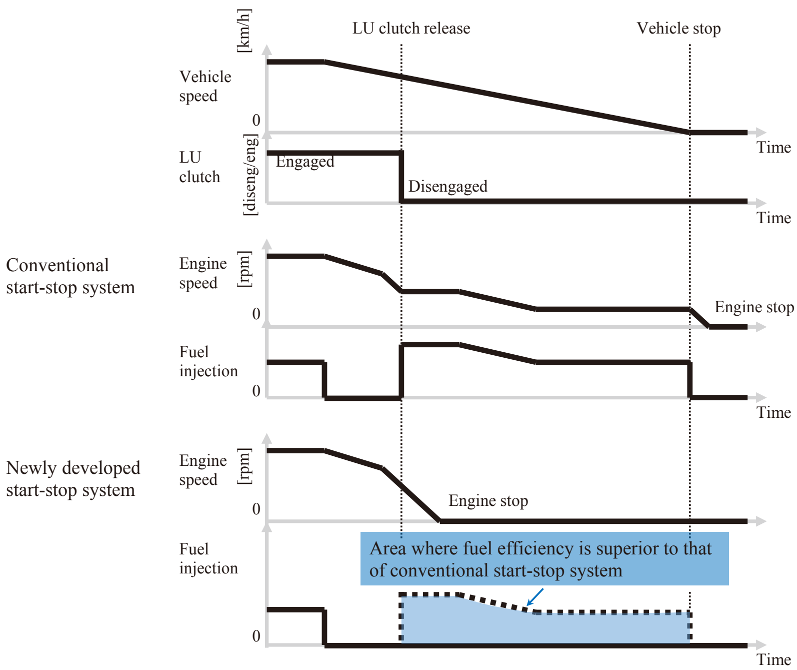

Fig. 1 Start-stop system timing chart

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** Unit System Development Department
*** Innovative Technology Development Department
**** Control System Development Office, JATCO Korea Engineering Corporation
***** Vehicle application Development Department, JATCO Engineering Ltd

Table 1 Specifications

Item			Jatco CVT7 (Current small CVT)	Jatco CVT-S (New CVT)
Torque capacity (Nm)			98	100
Control system			Electronic	←
Torque converter size (mm dia.)			185	←
Gear ratios	Ratio coverage		7.3	6.0
	Pulley ratio		2.200 - 0.550	2.411 - 0.404
	Final gear ratio		4.575	6.54 - 7.756
	Planetary gear ratios	1st	1.821	1.000
		2nd	1.000	
		Rev.	1.714	0.952
Total low ratio			18.33	15.77
Weight (wet) (kg)			65.0	60.8
Overall length (mm)			346	356
Distance between pulley shafts (mm)			147	←

Table 2 Comparison of conditions for activating start-stop system

	Jatco CVT7 (Current small CVT)	Jatco CVT-S (New CVT)
Vehicle speed [km/h]	13 or less	13 or less
Vehicle deceleration [-]	0.3G or less	0.3G or less
Pulley ratio [-]	Over 2.1 @High gear of auxiliary transmission	Over 1.5

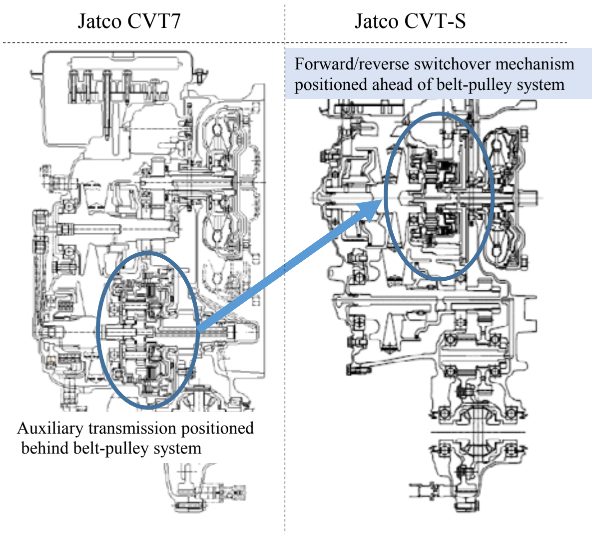

Fig. 2 Cross-sectional view of CVT7 and CVT-S

2. Development concept and structure

The CVT-S was developed to fit the engine compartment layout of mini-vehicles as a smaller, lighter unit than the Jatco CVT7 (CVT7) and thereby improve interior roominess, comfort and safety.

2.1 Structure and specifications of CVT-S for mini-vehicles

The CVT-S specifications in Table 1 were determined on the basis of the development concept.[(2)] After analyzing the constituent elements of a CVT, a structure was selected that positions the forward/reverse switchover mechanism ahead of the belt and pulley system as traditionally done in JATCO's midsize CVTs. In contrast, the CVT7 has the auxiliary transmission located behind the belt and pulley system (Fig. 2). Moreover, the CVT-S was developed without an electric oil pump in order to reduce the cost.

2.2 Application of start-stop system activated before the vehicle stops

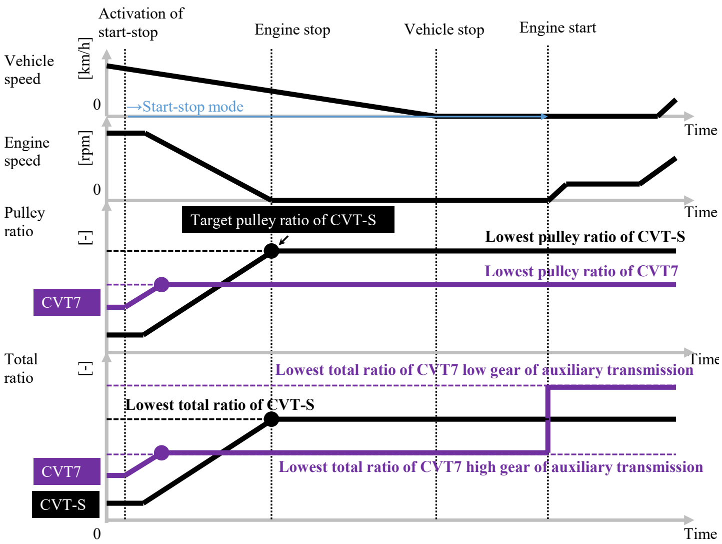

Fig. 3 Timing chart of the pulley ratio and through ratio of CVT7 and CVT-S

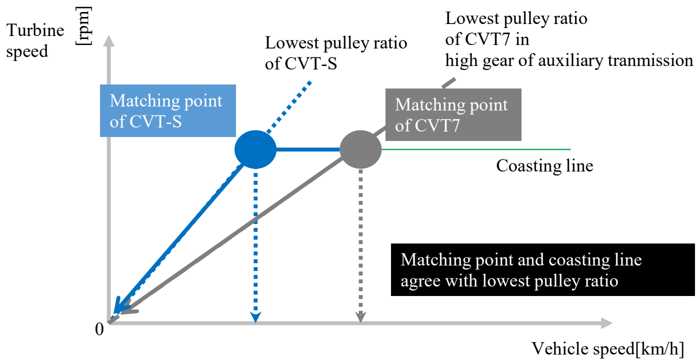

Fig. 4 Shift schedule of CVT-S and CVT7

The aim of the new start-stop system is to improve fuel economy by stopping the engine simultaneously with the release of the LU clutch. The conditions for activating the system are shown in Table 2. In order to ensure re-acceleration performance, the pulley ratio must return to a low gear ratio before the vehicle comes to a stop. However, because the CVT-S does not have an auxiliary transmission, it is necessary to continue shifting by means of the belt and pulley system until a lower vehicle speed is reached than with the CVT7 (Figs. 3 and 4).

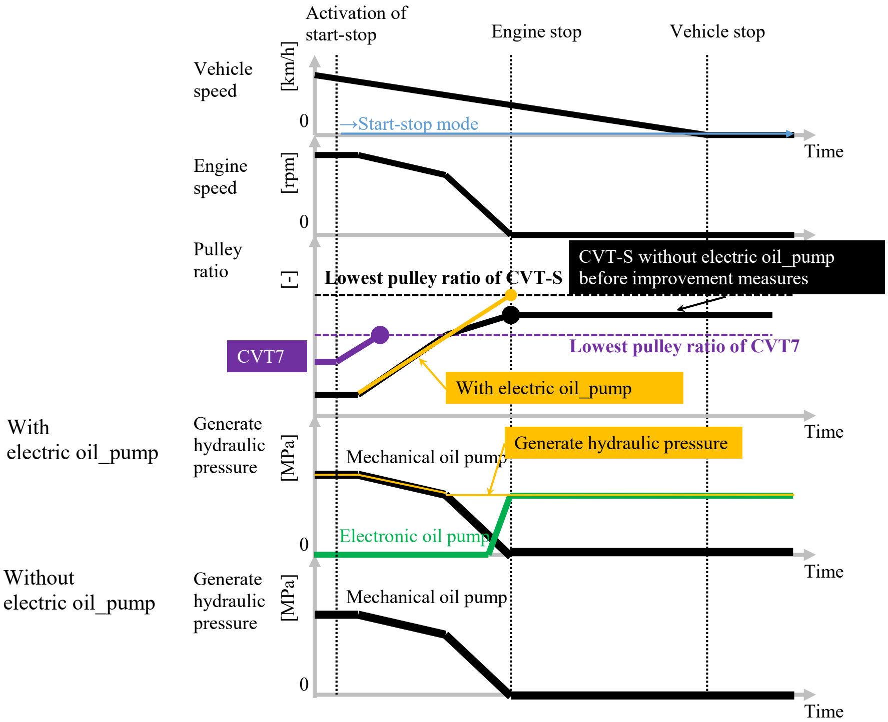

Fig. 5 Shift schedule timing chart of the belt-pulley system during vehicle deceleration

Table 3 Specifications of forward clutch

	Jatco CVT7 (Current small CVT)	Jatco CVT-S (New CVT)
Volume of hydraulic pressure chamber of forward clutch [cc]	16.5	21.4

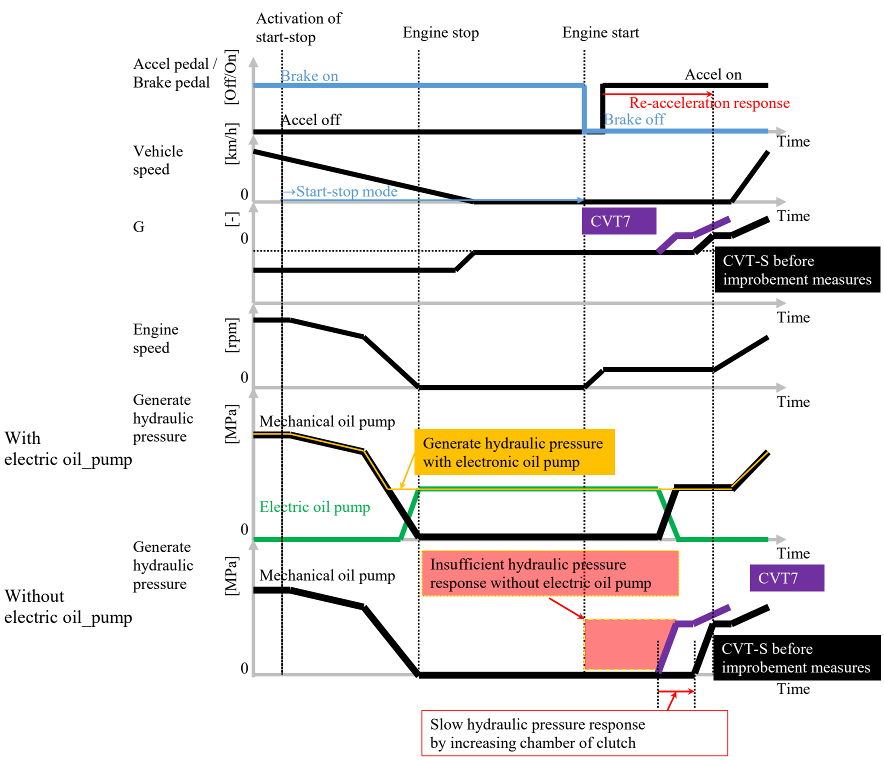

Fig. 6 Timing chart of hydraulic pressure response

3. Concerns about re-acceleration performance

One concern about re-acceleration performance was that the pulley ratio would not return to a low gear ratio before the vehicle came to a stop. Another concern was a possible response lag in generating hydraulic pressure after engine restart.

3.1 Concern about shift controllability of belt-pulley system during deceleration

The pulley ratio must return to a low gear ratio before the vehicle stops moving. However, because the CVT-S does not have an electric oil pump, if the engine was stopped while the vehicle was still in motion, the mechanical oil pump could not discharge any pressure. This would mean that no pulley pressure could be generated, making it difficult to return the pulley ratio to a low gear ratio (Fig. 5).

3.2 Hydraulic pressure response after engine restart

Because the CVT-S does not have an electric oil pump, CVT fluid flows out of the pulley pressure chamber while the engine is stopped and it leaks internally from the hydraulic pressure circuit. In addition, the forward clutch pressure chamber of the CVT-S has a larger volume than that of the CVT7 (Table 3). Consequently, after engine restart, it takes longer for the mechanical oil pump to supply a sufficient quantity of fluid to fill the pulley pressure chamber (Fig. 6).

4. CVT technologies contributing to re-acceleration performance

The adoption of the technologies described below has contributed to ensuring the desired re-acceleration performance.

4.1 Shifting to a low gear ratio before the engine stops

In order to ensure the desired re-acceleration performance, the pulley ratio must shift to a low gear ratio before the vehicle comes to a stop. In general, the ways of shifting to a low gear ratio include raising the hydraulic pressure of the secondary pulley or lowering the hydraulic pressure of the primary pulley. The hydraulic pressure of the primary pulley could not be reduced below the limit

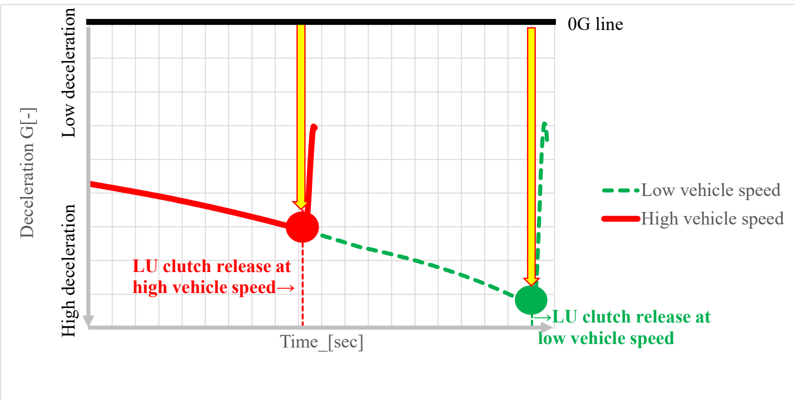

Fig. 7 Simulation results for deceleration feeling by changing vehicle speed for LU clutch release

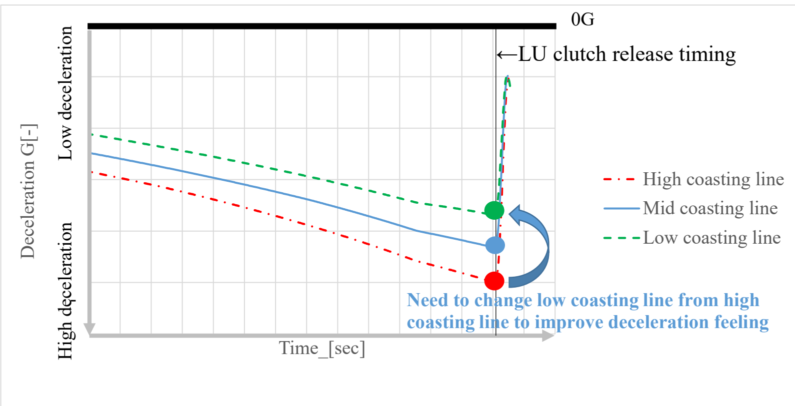

Fig. 8 Simulation results for deceleration feeling by changing coasting line

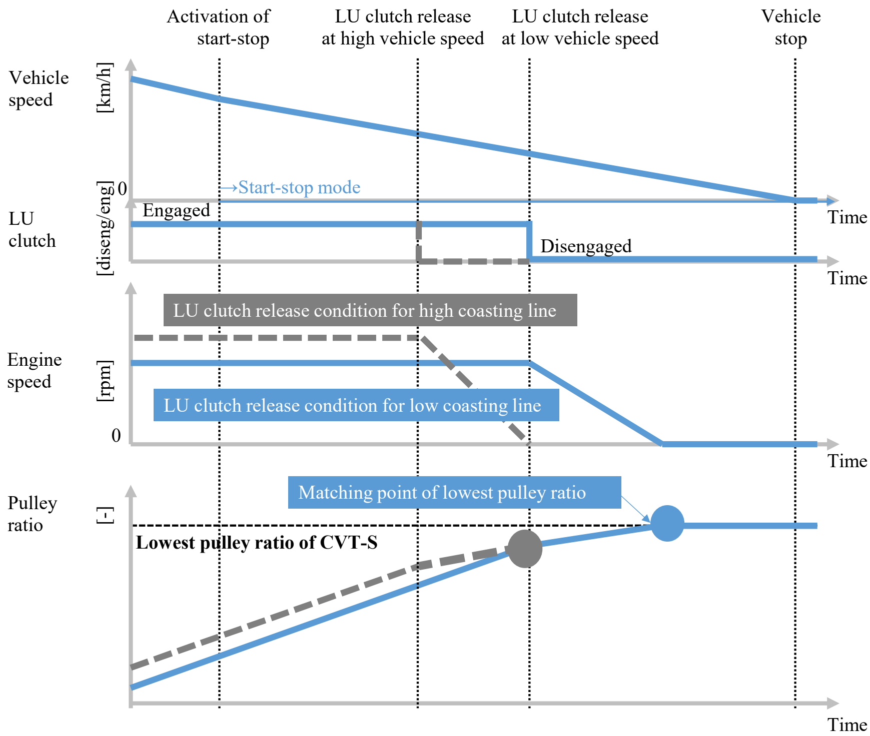

Fig. 9 Timing chart for changing LU clutch release condition to satisfy both deceleration feeling and shift control

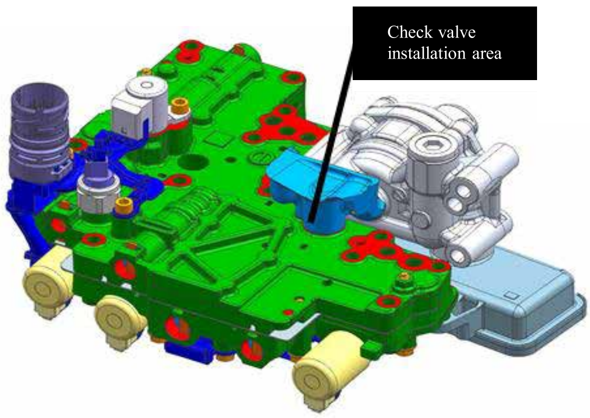

Fig. 10 Hydraulic system layout

needed to secure the torque transmission capacity desired for the belt. Consequently, in order to ensure the hydraulic pressure of the secondary pulley, the vehicle speed for releasing the LU clutch was reduced to a lower level and the timing for stopping the engine was delayed. However, lowering the vehicle speed for releasing the LU clutch would produce a stronger feeling of deceleration that might give the driver a sense that the vehicle was slowing down unintentionally (Fig. 7). Accordingly, the coasting line was lowered to improve the feeling of deceleration (Fig. 8). As the result, the vehicle speed condition for LU clutch release and the coasting line were both set so as to achieve an acceptable deceleration rate combined with shift control (Fig. 9).

4.2 Improvement of hydraulic pressure response by reducing internal fluid leakage

In order to ensure the desired vehicle acceleration performance following engine restart, the hydraulic pressure of the pulleys and clutches must be raised immediately so that driving force from the engine can be transmitted. However, the mechanical oil pump stops operating simultaneously with the stopping of the engine. Consequently, the CVT fluid filled in the hydraulic pressure circuit would flow backward and leak internally from the circuit, causing a response lag in the generation of hydraulic pressure. To avoid that condition, a check valve has been provided between the mechanical oil pump and the control valve to prevent backward flow (Fig. 10).[3] In addition, a low-leakage type of sealing ring has been adopted for the pulleys.

As a result, hydraulic pressure response has been ensured for the CVT-S that is equal to or better than that of the CVT7 even though the former unit has a larger clutch hydraulic pressure chamber than the latter unit.

5. Conclusion

A start-stop system that is activated before the vehicle stops has been developed without an electric oil pump and applied to a mini-vehicle CVT that does not have an auxiliary transmission. This was accomplished by improving the CVT shift control and hydraulic pressure response so as to satisfy customer demands for vehicle fuel economy and re-acceleration performance.

6. References

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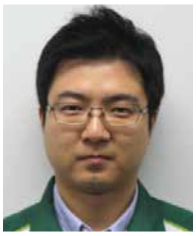
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From macro to micro thermal performance design —Estimation of lubricant temperature inside a CVT—

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Summary

In the process of lubricating internal transmission parts, the lubricant is heated and then partially cooled by the oil cooler before being returned and recirculated through the transmission. The lubricant temperature has traditionally been estimated at the oil cooler exit, but heat is transferred to the lubricant inside the transmission, presumably raising its temperature from the cooler exit before it is supplied to the parts. This article describes a method of modeling heat transfer and estimating the lubricant temperature in the passage from the oil cooler exit to the lubricated parts. Experimental results validating this method are also presented.

1. Introduction

Transmissions have traditionally been designed for macro thermal performance in order to ensure durable quality in real-world driving. The macro design includes the temperature in the oil pan, at the oil cooler exit and other parts based on the total amount of heat generated and radiated by the transmission. However, despite managing the oil temperature based on prior experience, it was found that there were driving situations where part temperatures exceeded the envisioned levels.

In order to improve the durable quality of all the parts inside the transmission, it is necessary to estimate part temperatures from a micro perspective and not just from a macro viewpoint. As one activity for estimating part temperatures, the lubricant temperature supplied to transmission parts was estimated and the estimation was validated in this study. This article describes the validation of the method used.

2. Transmission of interest and scope of lubricant temperature estimation

The transmission selected for investigation in this study was the Jatco CVT8 (CVT8), one of JATCO's representative products. Forced lubrication to the variator was selected for investigation. Because the variator is the most heat of all the CVT parts, managing its temperature is critical. Lubrication to the variator was treated as the main lubricant flow in the lubricant circuit of interest and lubrication to each part as branch lubricant flow. The dashed lines in Fig. 1 show the flow of the lubricant as seen from the side of the transmission. Figure 2 shows the lubricant passage extracted from Fig. 1. The actual path of lubricant flow from the oil pan to the parts to be lubricated is shown in Fig. 3. The transfer of lubricant from one part to another is divided into individual sections. Section number (i) indicates the traditional scope of temperature estimation and section numbers (ii) to (v) represent the scope of interest in this study.

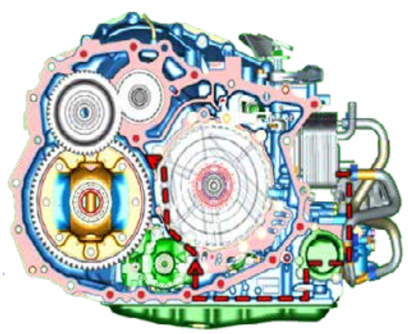

Fig. 1 Flow of lubricant in a side view of the transmission

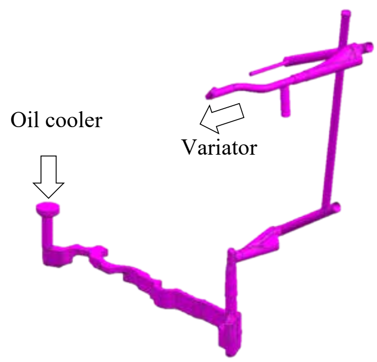

Fig. 2 Lubricant passage of CVT8

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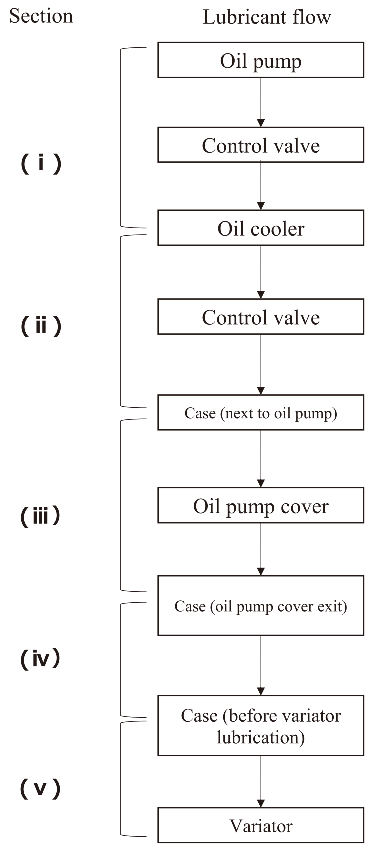

Fig. 3 Lubricant flow path and sectional divisions

3. Method of estimating lubricant temperature

In order to estimate the lubricant temperature, the change in the temperature in each section, dT, is found with the following equation.

$$\mathbf{dT} = \frac{Q}{G \times \gamma \times C} \quad (1)$$

dT: change in lubricant temperature
G: lubricant flow rate Q: amount of heat transferred
C: specific heat of lubricant γ: lubricant density

The amount of heat transferred Q to the lubricant received in each section must also be estimated in order to estimate the lubricant temperature. To do that, the lubricant passage shown in Figs. 1 and 2 was represented in a simple configuration, and the heat transfer paths to the lubricant were modeled (Fig. 4). It was considered that heat was transferred to the lubricant in the following ways:
I. heat transfer from heat-generating parts and
II. heat transfer due to the temperature difference between the passage inner wall and lubricant.

Ways I and II were defined as follows.
I. Heat transfer from heat-generating parts

This was defined as the transfer of energy loss incurred by heat-generating parts.

$$\mathbf{Q = EnergyLoss} \quad (2)$$

The oil pan is the heat-generating part that affects the lubricant passage in this transmission. In addition, since the lubricant passage goes through the transmission interior, it has no sections with a lower temperature than the lubricant temperature. Accordingly, it was assumed there was no heat radiation from the lubricant passage.

II. Heat transfer due to temperature difference between passage inner wall and lubricant

Assuming the flow in the lubricant passage was turbulent, heat transfer was defined using the Dittus-Boelter equation (Eq. (5)) for heat transfer of turbulent pipe flow.

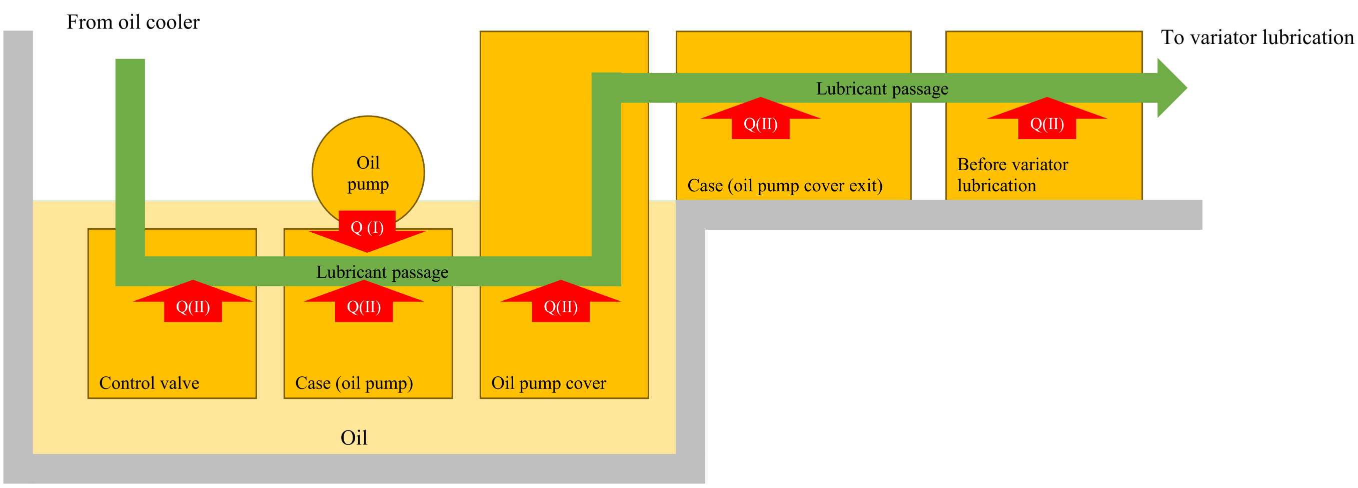

Fig. 4 Model of heat transfer to lubricant

$$\mathbf{Q} = \mathbf{A} \times \mathbf{h} \times (T_s - T_o) \quad (3)$$

$$\mathbf{h} = \frac{Nu \times \lambda}{d} \quad (4)$$

$$\mathbf{Nu} = 0.023 \times Re^{0.8} \times Pr^{0.4} \quad (5)$$

A: surface area of lubricant passage inner wall
Ts: temperature of lubricant passage inner wall
Nu: Nusselt number
d: pipe diameter
Pr: Prandtl number
h: coefficient of heat transfer
To: lubricant temperature
λ: thermal conductivity of lubricant
Re: Reynolds number

Oil remaining in the oil pan is considerably scattered by churning inside the transmission. For that reason, the temperature of the oil passage inner wall was assumed to be the same as that of the lubricant in the oil pan.

The equations defined above were applied to each section of the lubricant passage to formulate an equation for estimating the change in the lubricant temperature from the oil cooler exit until it was supplied to the variator.

4. Experimental validation

The lubricant flow rate and the temperature of each part were measured using an actual CVT, and the results were compared with the estimated change in the lubricant temperature. The factors affecting changes in the lubricant temperature are the amount of heat generated by the

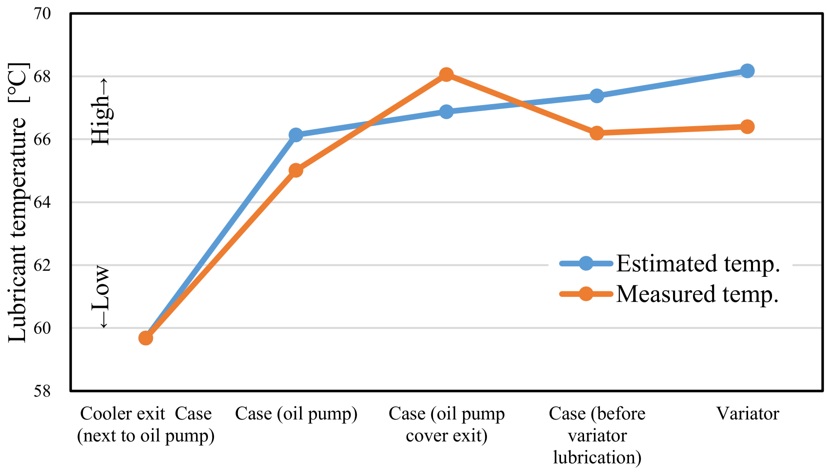

Fig. 5 Validation results

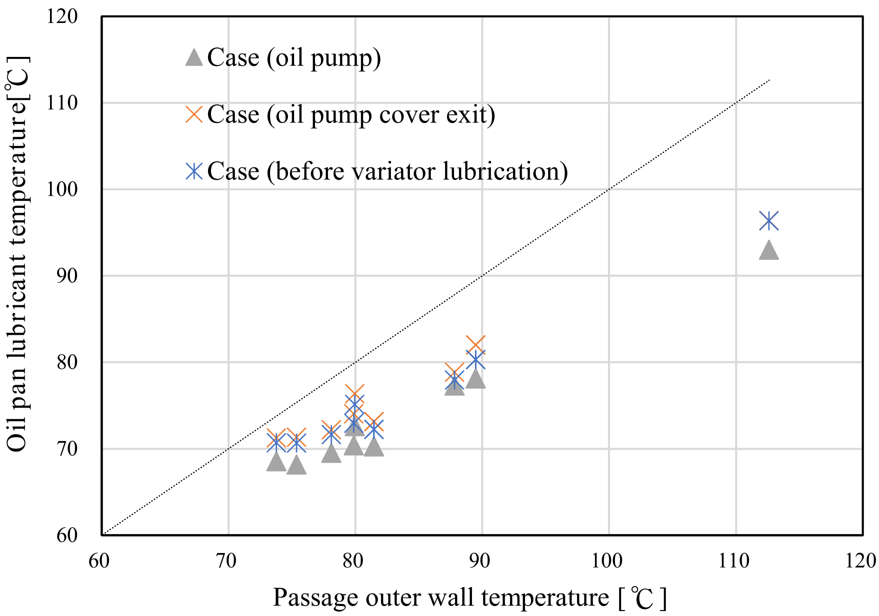

Fig. 6 Comparison of passage outer wall temperature and oil pan lubricant temperature

oil pump, the lubricant temperature in the oil pan and the lubricant flow rate. These factors were varied as the experimental conditions in the validation exercise in order to make a comparison with the estimated temperatures. The amount of heat generated by the oil pump is influenced by its rotational speed and discharge pressure.

5. Validation results

The difference between the estimated and measured temperatures for each part was a maximum of 2.4°C and a mean of ±1.0°C under each section. The oil pump displayed large heat generation in the validation exercise and the oil temperature in the oil pan was high. Figure 5 presents the experimental and estimated results for the highest rise in the lubricant temperature from the oil cooler exit to the variator under such conditions.

6. Discussion

A comparison of the measured and estimated results revealed that the lubricant temperature was often estimated higher than the actual measured temperature. As mentioned earlier, there are many places in the lubricant passage where heat is transferred to the lubricant from the passage inner wall. Among the variables in Eqs. (3) to (5) defined for heat transfer, the inner wall temperature (Ts) was assumed to be the same as the oil temperature in the oil pan. Therefore, the relationship between the passage inner wall temperature and the oil temperature in the oil pan was investigated. Because it was difficult to measure the inner wall temperature without blocking the lubricant flow, the temperature of the passage outer wall was measured. The results showed that the outer wall temperature tended to be lower than the oil pan lubricant temperature (Fig. 6).

It was thought that even places on the inner wall that were not submerged in the lubricant because of churning would have a temperature close to the oil pan lubricant temperature. However, it was found that the wall temperature was actually lower than the oil pan lubricant temperature. That difference was applied to Eqs. (1) to (5) and converted to the change in the lubricant temperature. The effect was found to be approximately 1°C. Accordingly, if the inner wall temperature was separately defined, it could further improve estimation accuracy. However, considering the resultant effect on durable quality, it was decided that the current level of accuracy within ±1.0°C was sufficient. Moreover, increasing the complexity of the estimation method would not be a good idea for its future use, so it was concluded that the present definition of the inner wall temperature should be left as it is.

7. Conclusion

Previously, macro temperature estimations were made using the overall heat generated and radiated by an entire CVT. In this study, a micro approach was taken to estimate the temperature of the lubricant supplied and the validity of this method was verified. This was undertaken as one activity for estimating the lubricant temperature at each part for the purpose of improving durable quality.

Modeling heat transfer to the lubricant made it possible to estimate the lubricant temperature to within a mean accuracy of ±1.0°C of the measured values, taking into account the passage inner wall temperature and heat-generating parts. Simplifying the equations and concept suppressed the additional volume to be studied, resulting in a method fully capable of practical application.

Since the estimation concept does not pertain uniquely to the CVT8, it can also be applied to estimate the lubricant temperature in other transmissions.

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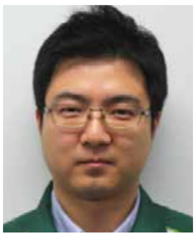
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Forming technology for integrated forging of parking gear with bottomless teeth and fixed pulley half

Ryosuke ONO* Shunsuke OHSHIMA* Yasuhiko YOSHIMIZU* Knwon YOUNGJO**

Summary

As a weight reduction measure for continuously variable transmissions, integrated forging of the parking gear and the fixed pulley half was previously adopted. Work was undertaken in this project to form a parking gear with bottomless teeth as a further weight reduction measure. This article describes the technical details of the practical application of a forming technology for integrated forging of the parking gear with bottomless teeth and the fixed pulley half. This was accomplished by using forging simulations to control workpiece deformation resistance.

1. Introduction

It has been necessary to further reduce the weight of continuously variable transmission (CVT) parts in recent years in order to improve fuel economy with respect to environmental performance. Because pulleys in particular are heavy CVT parts, reducing their weight is a key issue that should be resolved.

The parking gear has been integrated with the fixed pulley half as a means of reducing the weight of existing products. This paper describes the details of the practical application of a forming technology for integrated forging of the parking gear with bottomless teeth and the fixed pulley half for the purpose of achieving further weight reductions.

2. Parking gear performance requirements and forging method

2.1 Parking gear functionality and structure

Figure 1 shows the configuration of the integrated parking gear and fixed pulley half. When a vehicle is parked and the shift lever is moved to the P position, the parking gear functions to lock the transmission by engaging the tip of the parking pawl with the parking gear (Fig. 2).

Figure 3 illustrates the engagement of the parking gear and the pawl. In this locked position, the bottom of the parking gear teeth and the parking pawl are not in contact, so eliminating the tooth bottom can contribute to reducing the weight.

2.2 Mechanism of closed-die forging

Forging with burrs and closed-die forging are two types of forging methods (Fig. 4). Forging with burrs is a process in which excess material is pushed out to become burrs as forming of the part proceeds. Because this process provides excellent formability, underfill is not likely to occur and

Fig. 1 Integrated parking gear and fixed pulley half

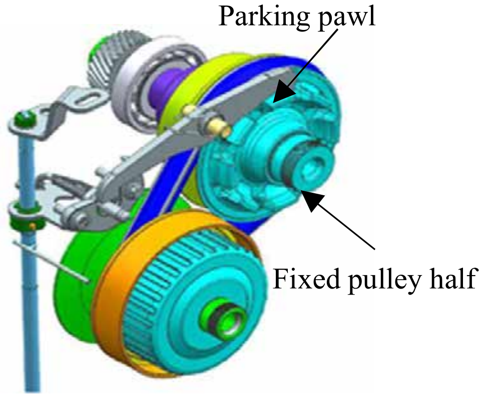

Fig. 2 Pulley assembly

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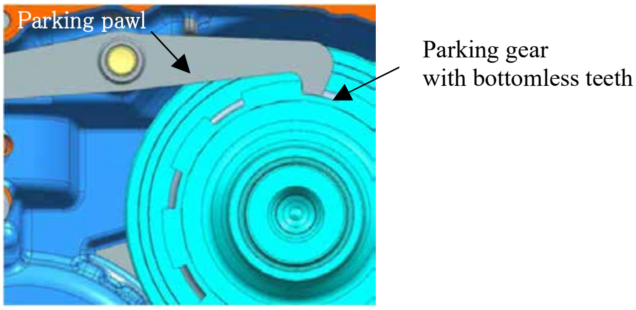

Fig. 3 Parking gear and pawl engagement

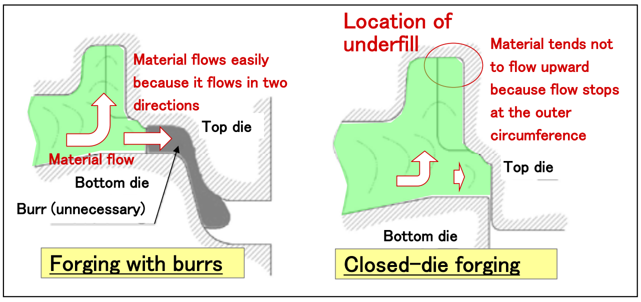

Fig. 4 Forging with burrs and closed-die forging

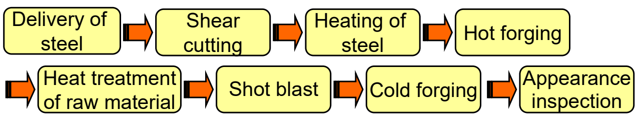

Fig. 5 Pulley production processes

low loading contributes to long die life. On the negative side, the input material weight increases because yield worsens to the extent that burrs occur.

In contrast, in closed-die forging the part is formed with the top and bottom dies engaged. This process has the advantage that yield improves because burrs do not occur. On the other hand, because forming is done in a closed-die state, it has the disadvantages that the load tends to increase, heightening the risk that the dies may crack or wear. In addition, because productivity declines, underfill is apt to occur, resulting in quality defects. Generally, the closed-die forging method entails a greater degree of difficulty.

2.3 Method of producing integrated parking gear and fixed pulley half

The production processes of the integrated parking gear and fixed pulley half are shown in Fig. 5. As mentioned in the preceding subsection, closed-die forging has been adopted for this product. It is formed through a combination of hot forging and cold forging.

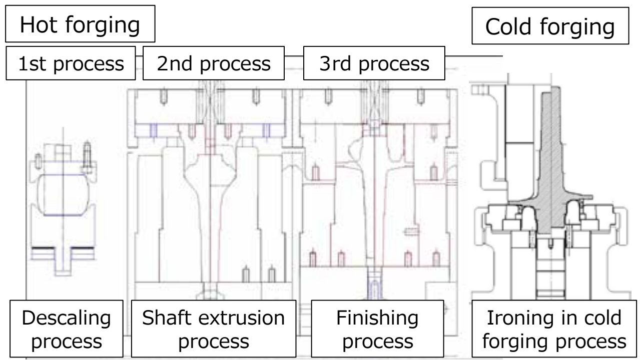

Fig. 6 Role and geometry of each process in hot and cold forging

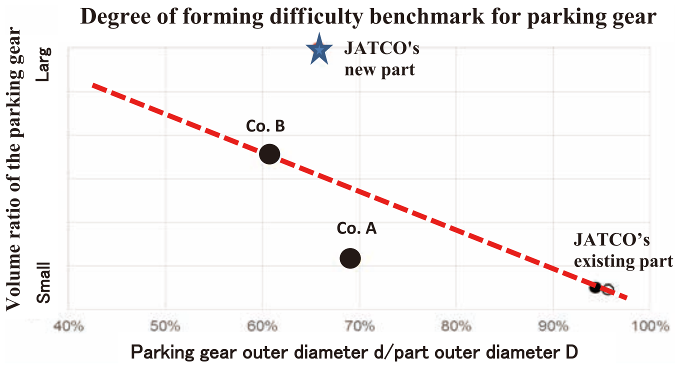

Fig. 7 Degree of forming difficulty of integrated parking gear and fixed pulley half

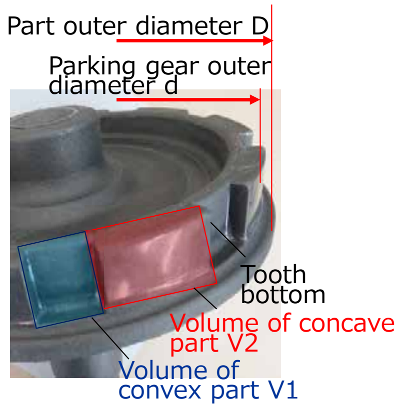

Fig. 8 Parking gear geometry

The product is formed in a total of four processes, three hot forging processes and one cold forging process (Fig. 6). The hot forging processes form the shafts of the fixed pulley half, which have a high rate of cross-sectional area reduction, and the parking gear with its difficult-to-form shape. Ironing of the parking gear is done in the cold forging process to form the tooth profile with high accuracy.

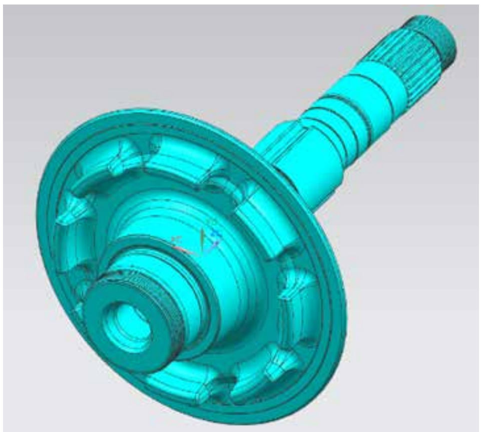

Fig. 9 Integrated parking gear with bottomless teeth and fixed pulley half

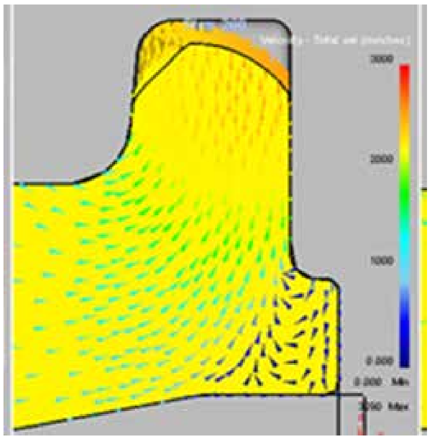

Fig. 10 Results of defect analysis by simulation

2.4 Degree of difficulty in forming integrated parking gear and fixed pulley half

Figure 7 shows the degree of difficulty involved in forming the integrated parking gear and fixed pulley half. The horizontal axis shows the ratio of the parking gear outer diameter d to the part outer diameter D (d/D). The vertical axis shows the volume ratio of the parking gear (V1/(V1+V2)) as indicated in Fig. 8. It is seen that the parking gear is positioned more to the outside as the diameter ratio d/D becomes larger. That makes it necessary to apportion more material volume to the outer side, which complicates the forming of the parking gear. In addition, as V1 becomes larger, it signifies a bigger difference in the convex and concave portions of the parking gear, making the gear more difficult to form.

The integrated parking gear with bottomless teeth and the fixed pulley half adopted in this project is shown in Fig. 9. Because this part shape is a top-level benchmark in the automotive industry, it was expected to be highly difficult to form and that problems like underfill, seizure and shortening of die life, among other things, might occur.

The following sections describe the closed-die forging methodology proposed for the difficult-to-form shape of this integrated parking gear with bottomless teeth and fixed pulley half.

3. Forging process design

3.1 Forging simulation

The finite element method was used to conduct a forming simulation in order to predict the material flow and forming timing in the forming process for this forged part. In the simulation, the dies were treated as rigid bodies and the material as an elastic body. The coefficient of friction, material temperature, die temperature and the force and speed applied by the press in the compression direction were defined as the parameters for conducting the forging simulation.

The forming load, forming flow and die stress are among the principal judgment criteria for determining the die geometry in each process for a forged part. A more excellent part shape can be expected as the forming load in the simulation decreases.

An ideal material flow is to fill the long and short shafts of the fixed pulley half, the parking gear and the outer diameter of the part in that order. A simulation is therefore conducted to determine the optimal die geometry. If the die geometry is unsuitable, the forming timing can differ, resulting in cases where the parking gear is underfilled or reverse material flow occurs. Reverse flow can cause marks or depressions in the product that become quality defects. This means that the die geometry is an extremely critical factor in the forging process. It is also necessary to determine the die geometry so that stress is below the threshold level because high stress can damage the die.

3.2 Issues in developing a parking gear with bottomless teeth

The first step toward developing a parking gear with bottomless teeth was to conduct a simulation of part forming in the forging process. Eliminating the tooth bottom would reduce the volume of the parking gear, thus quickening the filling timing of the outer diameter. Consequently, using the previous die geometry could worsen the underfill defect of the parking gear mentioned above or cause depressions in the back side of the sheave. These potential problems were revealed in a prior simulation.

The results of a forming simulation of the parking gear portion in the third forging process are shown in Fig. 10. It was predicted that underfill would likely occur because the upper part of the die was not fully filled.

The arrows show the direction in which forming proceeds. Generally, forming in the forging process ordinarily proceeds from the inside toward the outside. However, the results in Fig. 10 indicate that some of the arrows on the back side of the parking gear were in the opposite direction. This indicates reverse flow of material during forming. Because it is known that reverse material flow can cause marks or depressions, some measure had to be taken to prevent it.

As a corrective measure against underfill defects and depressions on the sheave back side, the die geometry in the second forging process was changed and a forming simulation was performed again. As a result, it was confirmed that underfill and depressions on the sheave back side did not occur.

Figure 11 compares the simulation results before and after the corrective measure was taken. A time history of the material flow begins from the left side. The yellow color indicates places that were not fully filled. Figure 11(a) shows that the corners of the parking gear portion were not fully filled before the die geometry was changed, and Fig. 11(b) shows that even the corners were filled after the die geometry was changed.

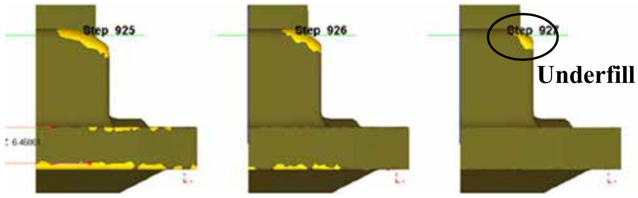

Fig. 11(a) Simulation results before corrective measure

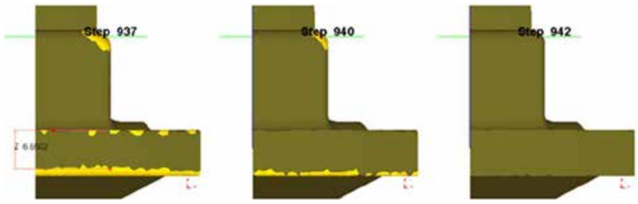

Fig. 11(b) Simulation results after corrective measure

Fig. 12 Parking gear underfill defect

4.1 Experimental results and forging simulation improvement

A forging trial was performed using the die modified on the basis of the simulation results in subsection 3.2. It was found that an underfill defect occurred in the parking gear portion of the die (Fig. 12), indicating that there was a significant difference in formability between the actual die and the simulation.

The following reason is presumed to account for the difference in the results between the simulation and the actual die. The adoption of bottomless gear teeth thinned the geometry of the parking gear portion, causing a decline in thermal energy. In addition, because the surface area of the parking gear was increased, it tended to cool more easily so the material temperature dropped. The decrease in the material temperature increased deformation resistance of the workpiece, which presumably caused more material flow defects than was predicted by the simulation.

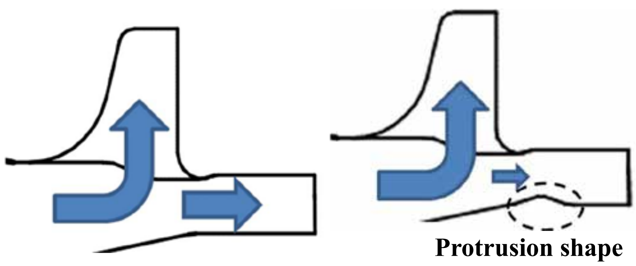

Fig. 13 Material flow with addition of small protrusion to modify finishing die geometry

Fig. 14 Simulation results following addition of protrusion to finishing die

Accordingly, the friction coefficient used in the simulation between the die and the workpiece was revised and a forging simulation was performed again. As a

Fig. 15 Parking gear after addition of small protrusion

result, the actual forging and the simulation showed good agreement. This confirmed that forging defects could be predicted by simulation even for this integrated parking gear with bottomless teeth and fixed pulley half.

4.2 Improvement of material flow by adding a protrusion in the finishing die

As mentioned in subsection 3.1, the ideal material flow is in the order of the short and long shafts of the fixed pulley half, parking gear and the part outer diameter. On the other hand, in the prototype forging trial explained in subsection 4.1, it was found that the parking gear portion tended to cool more easily, causing an increase in the deformation resistance of the workpiece. That gave rise to underfill defects as a result of the part outer diameter filling with material before the parking gear portion.

Therefore, a study was undertaken of a proposed measure for improving material flowability to the parking gear portion by increasing deformation resistance in the direction of the part outer diameter so as to delay the timing for filling with material.

A die geometry was devised with a small protrusion provided as shown in Fig. 13 for the purpose of delaying the forming timing of the part outer diameter. A simulation was conducted again using a die with this protrusion and the results are shown in Fig. 14. Good results were obtained, and no forging flow defect was observed that would lead to an underfill defect in the parking gear.

4.3 Validation results for improved forging simulation

Underfill occurred in the first prototype trial, but in the second prototype trial an underfill occurrence rate of 0% was obtained for the same forming load as in the first trial. No other quality problems were observed, including other underfill locations and marks (Fig. 15).

The defect rate following the production launch has continued to remain at a low level of no more than 0.2%, thereby confirming that good results were obtained with the improved forging simulation.

5. Conclusion

A forming technology was developed for integrated forging of the parking gear with bottomless teeth and the fixed pulley half in a closed-die forging process. This part has one of the most difficult shapes to form in the automotive industry. The following knowledge was gained in this project.

(1) A forming simulation was conducted for integrated forging of the parking gear portion with bottomless teeth and the fixed pulley half. It was found that the friction coefficient between the die and the material had to be increased because the parking gear portion tended to cool easily.

(2) Material flow to the parking gear portion was improved by providing a small protrusion in the die to increase workpiece deformation resistance in the direction toward the outer circumference.

6. References

Shunsuke OHSHIMA and Knwon YOUNGJO, Forging Dies, Japanese Unexamined Patent Application Publication No. 2019-76943 (filing date: October 26, 2017)

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Memo

Technological development for improving the cylindricity of the reaming process for CVT control valve spool holes

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Summary

It is necessary to reduce the amount of internal leakage in the control valve of continuously variable transmissions in order to improve vehicle fuel economy. That requires the use of high-accuracy machining. This article describes technical measures undertaken to improve the cylindricity of the reaming process for spool holes that require especially high accuracy.

1. Introduction

There are demands to improve the fuel economy obtained with continuously variable transmissions (CVTs). One effective measure for doing that is to reduce the amount of internal leakage in the control valve (Fig. 1). The control valve uses the hydraulic pressure generated by the oil pump to control the pressure level needed for shifting. The spool holes that were the focus of this work function as the fitting holes of the spool valves that regulate the hydraulic pressure. To reduce the amount of leakage, the clearance between the spool valves and spool holes is minimized and also high-strength material is applied to improve their durability. However, this measure worsens the reaming accuracy of the spool holes, making it especially difficult to ensure their cylindricity on the order of several μm. This article describes concrete technical measure adopted to improve cylindricity.

2. Spool hole machining method and issues involved

Spool holes are machined on a vertical machining center in two processes. In the rough casting in Fig. 2(a), pilot holes are first drilled in (b) and then reamed in (c). A single-blade reamer is used to suppress polygonal shape error of reamed holes[1] the accuracy of which is problematic. A hydraulic chuck holder with high rigidity, excellent runout accuracy and outstanding ease of attaching/releasing is used for chucking the tool. There are three main issues in spool hole reaming: cylindricity, finish surface roughness and tool life. Among them, improving cylindricity is the most difficult issue.

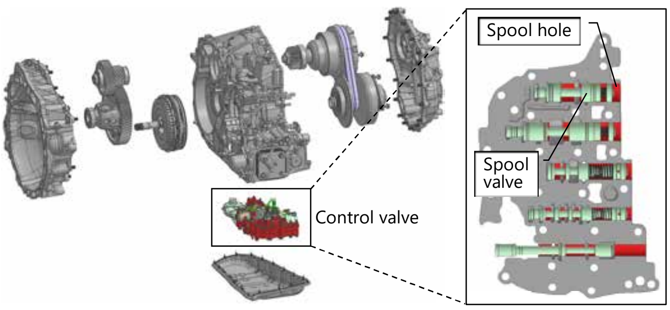

Fig. 1 Exploded view of a CVT

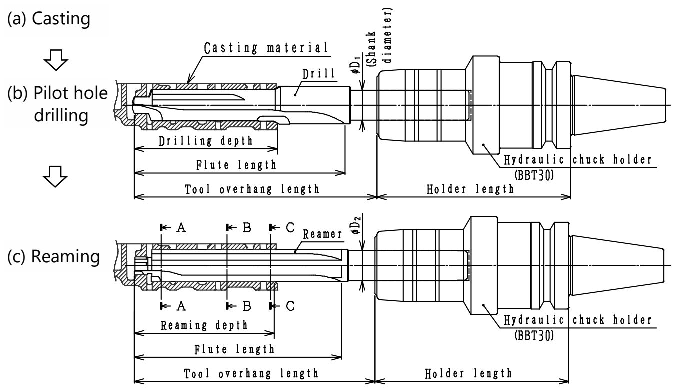

Fig. 2 Conventional tooling layout

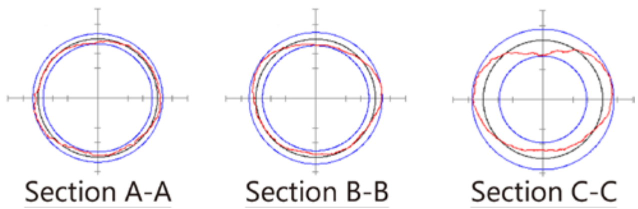

Fig. 3 Roundness profile of reamed spool hole

3. Factors influencing improvement of cylindricity

Cylindricity is defined as a combination of three

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elements: circumferential runout (hole diameter dimension), roundness and coaxiality (position deviation), which are measured with a dedicated measuring machine. As shown in Fig. 3, individual measurements are made of section C-C (mouth of hole), section B-B (middle of hole) and section A-A (bottom of hole). A comparison of the roundness of each section reveals that section C-C (mouth) has the worst roundness. Presumably, deterioration of mouth roundness is the principal cause of the deterioration of cylindricity. Therefore, based on the results of previous studies,[(1)-(5)] the factors influencing improvement of cylindricity were organized in a systematic cause and effect diagram as shown in Fig. 4. The factors influencing cylindricity include the tool/holder specifications, cutting conditions, workpiece material and the machining equipment used. This study focused on the cutting conditions (feed rate of pilot hole drilling) and tool/holder specifications (tool bending stiffness and rotational balance), and cutting tests were conducted to investigate their respective influence.

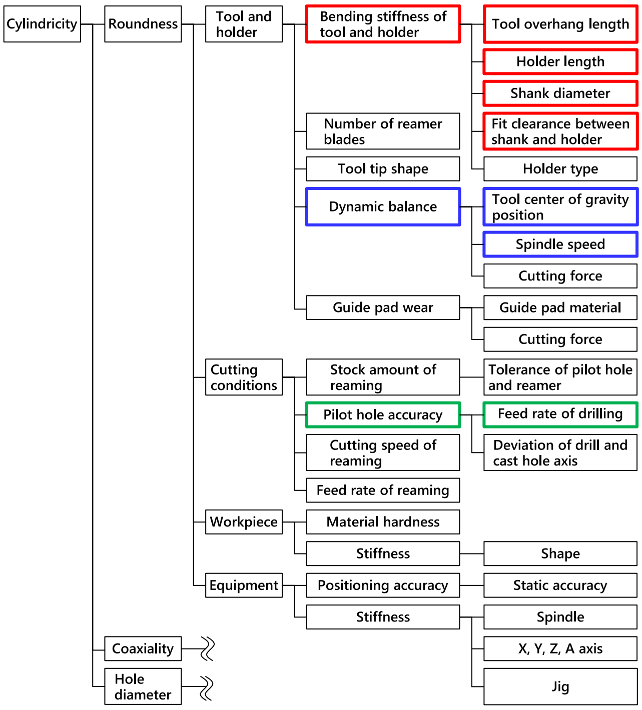


Fig. 4 Cause and effect diagram for improving cylindricity

4. Overview of cutting tests

4.1 Influence of feed rate of pilot hole drilling on cylindricity

Machining tests were conducted with the equipment shown in Fig. 5 to confirm the influence of the feed rate of

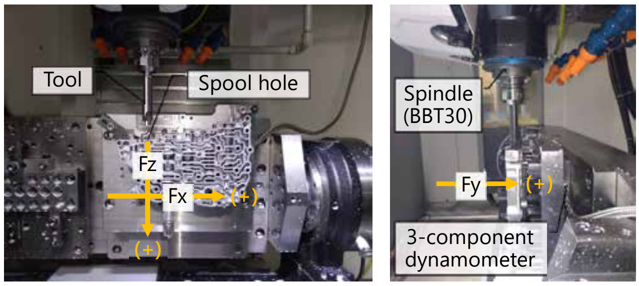


Fig. 5 Schematic diagram of cutting test

Table 1 Cutting conditions

Process	Cutting speed Vc m/min	Feed rate fz mm/tooth	Tooling
Drilling	311	0.025, 0.05, 0.1	Fig. 2(b)
Reaming	150	0.089	Fig. 2(c)

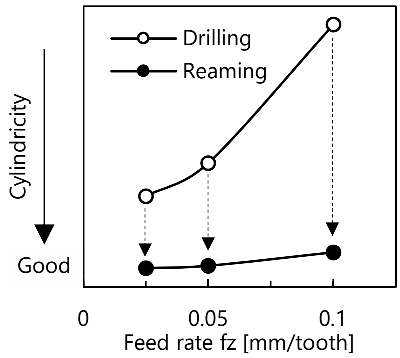
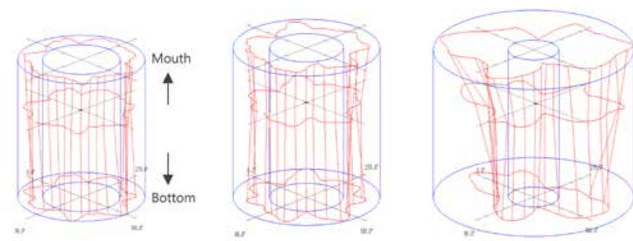


Fig. 6 Results of cutting test



(a) fz 0.025 mm/tooth (b) fz 0.05 mm/tooth (c) fz 0.1 mm/tooth

Fig. 7 Cylindricity profile of pilot hole

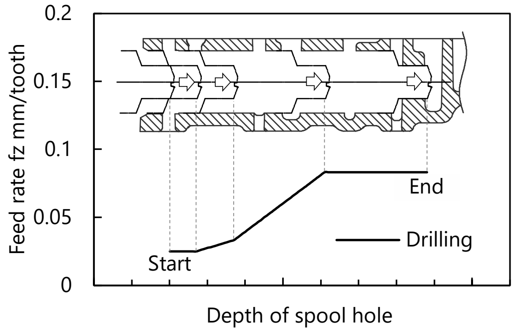


Fig. 8 Variable feed control of drilling

pilot hole drilling on the cylindricity of reamed holes. The cutting conditions are shown in Table 1. Pilot holes were drilled using three feed rate levels and the workpiece was then removed from the machine to measure cylindricity. The workpiece was then mounted on the machine again, and the holes were reamed using a single set of conditions. Since high-accuracy positioning was possible with this machine, it was assumed that position deviation between the pilot hole and the reamed hole due to removing/remounting the workpiece was negligible.

The measured cylindricity results are plotted in Fig. 6. The results confirmed that the cylindricity of the reamed holes was influenced by that of the pilot holes and also that the cylindricity of the pilot holes was influenced by the drilling feed rate.

Figure 7 shows profiles of the pilot hole cylindricity. As the feed rate was increased, the roundness of the mouth markedly deteriorated. According to a previous study,[4] that was presumably caused by unstable tool behavior at the time the tool cut into the workpiece. It was reported that using a highly rigid tool and cutting with tiny feeds in the initial phase of machining are effective in preventing such deterioration. Therefore, variable feed control was adopted in order to obtain both high accuracy and high machining efficiency. With this approach, a low feed rate is applied in the initial phase of pilot hole drilling and the feed rate is gradually raised as the drilling depth increases.

4.2 Improvement of tool bending stiffness

The bending stiffness of the tool and holder was measured with the method shown in Fig. 9 to confirm the amount of tool displacement[5] relative to the bending load of the tool when held in the holder. The measured results for the existing drill and holder specifications are shown in Fig. 10(a). Sharp displacement gradients are seen from the fulcrum where the shank is held by the holder to the tool tip, suggesting that high stiffness at the place where the shank is gripped is important. Therefore, as indicated in Table 2, an effort was made to reduce the amount of

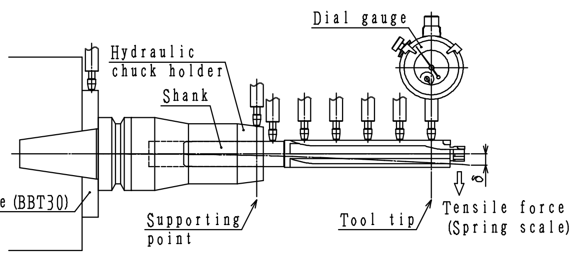


Fig. 9 Method of measuring bending stiffness of tool and holder

Table 2 Improved specifications

Contents	Drill	Reamer
Tool overhang length	↓ 30.2% shorter	↓ 35.9% shorter
Shank diameter	↑ 25.0% larger	↑ 12.5% larger
Holder length	↓ 28.6% shorter	
Fit tolerance of shank and holder	H4 (holder)/js4 (shank)	

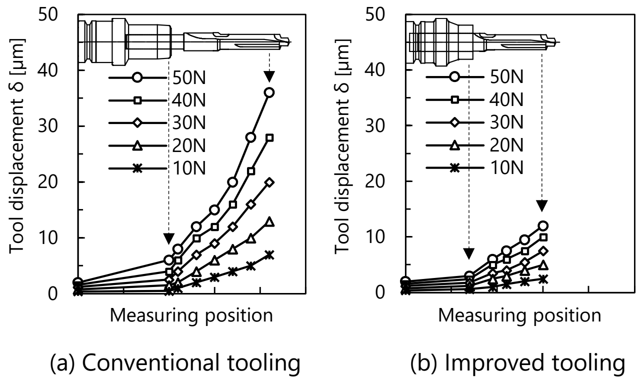


Fig. 10 Measured bending stiffness of drill and holder

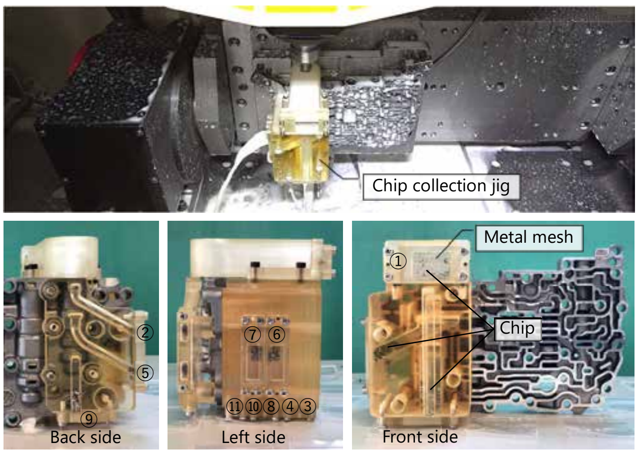


Fig. 11 Chip collection jig

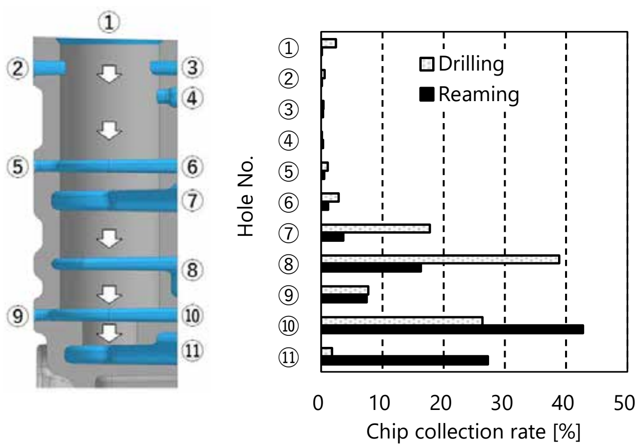


Fig. 12 Chip collection results

displacement through a combination of measures, including reducing the tool overhang length, increasing the shank diameter, shortening the holder length and minimizing the fit clearance between the shank and the holder. The results obtained are shown in Fig. 10(b).

Conceivable ways of reducing the tool overhang length include shortening the flute length of the drill and reamer as shown in Fig. 2(b) and (c). However, that could worsen the evacuation of chips toward the mouth of the spool hole, giving rise to concern that chip clogging might gouge the machined surface. To prevent that, a fluid simulation was used to optimize the positions and angles of the coolant discharge ports of the tool with the aim of evacuating chips from the spool hole mouth toward the bottom of the hole. Figure 11 shows a chip collecting jig made of resin by 3D printing, which was used to confirm the effects of this improvement. This jig fits tightly without any gaps to the spool hole mouth (1) and each of the side openings ((2)-(11)) shown in Fig. 12. A metal mesh incorporated at the exit ports makes it possible to separately collect only the chips that are evacuated together with the coolant. The results obtained with the jig are also shown in the figure. The results confirmed that chips were concentrated at side openings (7) to (11) toward the bottom of the hole as was intended.

4.3 Improvement of tool rotational balance

In addition to improving tool bending stiffness as explained in 4.2, an investigation was made of the influence of the rotational balance on tool tip runout. Although a single-blade reamer is advantageous for avoiding polygonally shaped reamed holes, there was concern that the tool rotational balance might worsen for the following reasons. (1) The center of gravity (G) on the guide pad side might deviate due to gouging of the cutting edge as shown in Fig. 13(a). (2) Reamer behavior at the onset of cutting might become unstable due to the application of cutting force (P) on the guide pad side which is in reverse phase of the cutting edge, thereby worsening the roundness of the hole mouth. To prevent that, the reamer outer circumference was designed with a shape so that the spindle rotation center (O) would be positioned at the center of gravity (G) as shown in Fig. 13(b).

Figure 14(a) shows the appearance of the tool measuring device based on an optical line sensor along with the method of measuring tool tip runout. The measured results are shown in Fig. 14(b). It is seen that the execution of a balanced design effectively reduced tool tip runout at high rotational speeds.

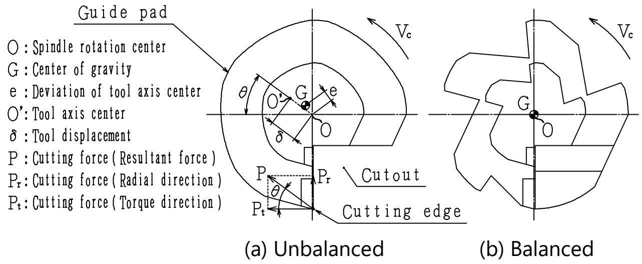


Fig. 13 Cross section of reamer

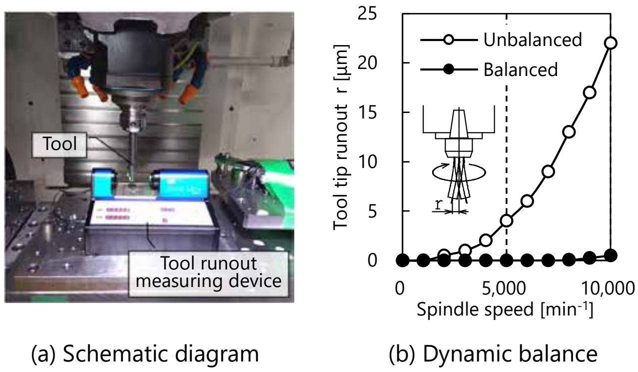


Fig. 14 Results of dynamic balance test

5. Optimized specifications and application results

Based on the results presented in section 4, the stiffness of the tool and holder was increased by the measures listed in Table 2 and combined with the balanced reamer rotational design shown in Fig. 13(b). Figure 15 presents the results of a cutting test conducted with the cutting conditions shown in Table 3. Compared with the existing

Table 3 Improved cutting conditions

Process	Cutting speed Vc m/min	Feed rate fz mm/tooth	Tooling
Drilling	311	0.025-0.083 (Variable feed control)	Table 2 and balanced
Reaming	225 (50% faster)	0.1 (12% faster)	

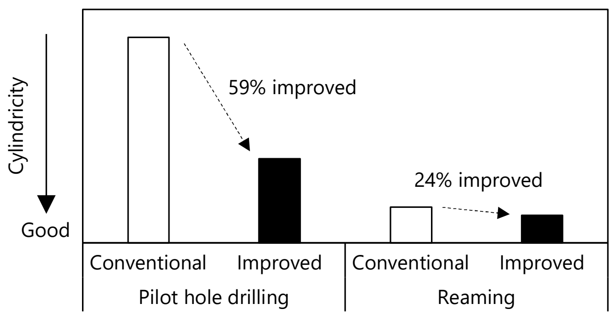


Fig. 15 Results of cutting test

cylindricity, it is seen that the cylindricity of both the pilot hole and the reamed hole was improved, thereby confirming the effectiveness of these specifications.

6. Conclusions

Optimizing the tool and holder specifications used in machining the control valve spool holes improved hole cylindricity by 24% over the existing dimension. The following insights were gained in this study.

(1) The cylindricity of reamed holes is greatly influenced by that of pilot holes. Reducing the feed rate for drilling pilot holes is effective in improving their cylindricity.

(2) Shortening the tool overhang length, increasing the shank diameter and minimizing the fit clearance between the shank and holder are effective measures for suppressing tool displacement.

(3) The tool flute length can be shortened by optimizing the positions and angles of the coolant discharge holes in the tool so as to promote chip evacuation from the spool hole mouth toward the bottom of the hole.

(4) Tool tip runout at high rotational speeds can be reduced by designing a single-blade reamer with high stiffness and excellent rotational balance.

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Practical application of an outdoor AGV system

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Summary

For the production of a new transmission, it was necessary to transport parts between separated buildings. In order to reduce logistics and transport costs, an outdoor Automated Guided Vehicle system was constructed based on a towing Automated Guided Vehicle and implemented to handle the transporting and loading/unloading of cargo fully automatically. This article describes the measures developed and adopted to deal with transport issues and safety issues so as to achieve fully automated parts supply with this new towing Automated Guided Vehicle system.

1. Introduction

Two types of Automated Guided Vehicles (AGV) are the towing type and the low-bed type. The former type transports things on a cart that the vehicle tows by means of a towing hook connection. The latter type transports things by sliding under the cart and is secured to it by a pin. The towing type is not much affected by floor distortions or undulations, but it has no record of use at JATCO for automated parts supply. In contrast, the low-bed type has an ample record of automated parts supply use, but floor distortions or undulations can make it difficult for the vehicle to travel stably.

A recent plan for a new transmission required AGV use for automated parts supply, in addition to involving travel outside the plant over a surface with many distortions and undulations. Moreover, measures had to be adopted for preventing any impact on product quality, including protection against wind and rain.

This article describes the technologies that were developed to achieve fully automated parts supply using an AGV system based on the towing type. It explains the measures implemented to deal with outdoor transport issues, to prevent parts supply delays and to ensure safe operation.

Fig. 1 Towing type AGV

2. Vehicle travel and parts supply issues

2.1 Measures supporting AGV travel on slopes

The towing type of AGV in Fig.1 is well-suited to traveling on distorted surfaces. However, there is a risk of deviating from the path when traveling on a downward slope because the vehicle is pushed by the weight of the load on the cart. It was necessary to implement measures to prevent such deviation because the outdoor path for the intended use contained a downward slope of 4° along the way.

According, a system was devised for automatically braking the cart. A mechanism was adopted for sliding the towing bar 60 mm longitudinally. An automatic braking system was built for moving the towing bar via a linkage so as to press rubber pads against the middle fixed tires of the cart. Figure 2(a) shows images of the system when traveling on a level surface. The towing bar is in the "pulled-out position" to retract the rubber pads. Figure 2(b) shows images of the system when traveling on a downward slope. The towing bar is moved to the "downward slope pushed-in position" so as to press the rubber pads against the tires via the linkage mechanism. This braking system works to brake the cart, enabling the towing AGV to travel stably on downward slopes.

2.2 Outdoor transport issues

With the previous method of transporting parts between buildings, a cover was placed over the cart to keep out wind

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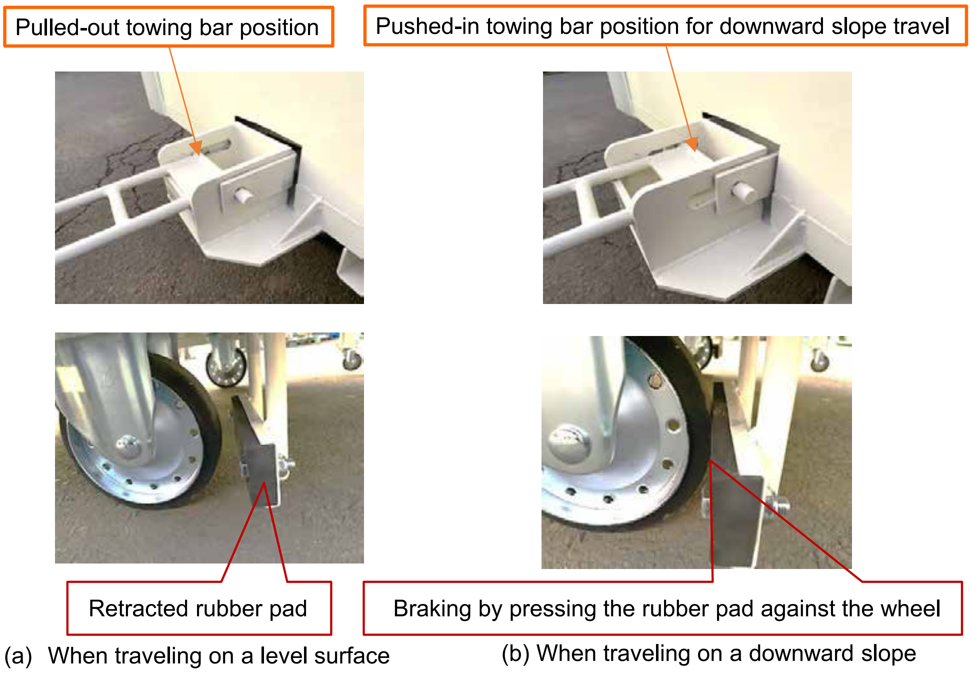

Fig. 2 Automatic braking system

and rain. The cover was opened and closed manually and a forklift was used to handle the parts. In order to achieve fully automated parts supply, it was necessary to have the same level of functionality for keeping out wind and rain as that provided by previous truck transport in an outdoor environment. In addition, the cart had to be fitted with a mechanism for automatically opening and closing the cargo door. To do that, a framework was constructed using aluminum frames. It was covered with a 3D molded tarp made of the same quality of material as that of the truck bed tarp and fastened. A gull-wing structure was adopted for automatically opening/closing the cargo door. The cart is equipped with two actuators to which air is supplied from the conveyor via an automatic coupling, making it possible to open/close the cargo door automatically (Fig. 3).

2.3 Fully automated parts supply

In order to achieve fully automated parts supply, it is necessary to prevent any problems of something getting caught when transferring from the conveyor to the cart. That could be caused by changes over time in the slope of the floor or by wear of the cart casters. Accordingly, it requires continuous monitoring and control of the heights of the conveyor and the cart so that their levels always coincide perfectly.

To ensure that the heights of the conveyor and cart always coincide, the AGV system adopts a towing bar that swings and floats vertically. This makes it possible to pull the entire cart onto hoist rollers for raising the cart. Figure 4(a) shows the cart before it is positioned on the hoist rollers. Because the casters are still in contact with the floor, the cart tilts along the slope of the floor. Figure 4(b) shows the cart positioned on the hoist rollers. Because the cart casters are off the floor, the cart is not affected even if the floor has a slope. Moreover, this system ensures stable automated parts supply even if the cart casters are worn.

Fig. 3 Gull-wing structure for automatically opening/closing the cargo door

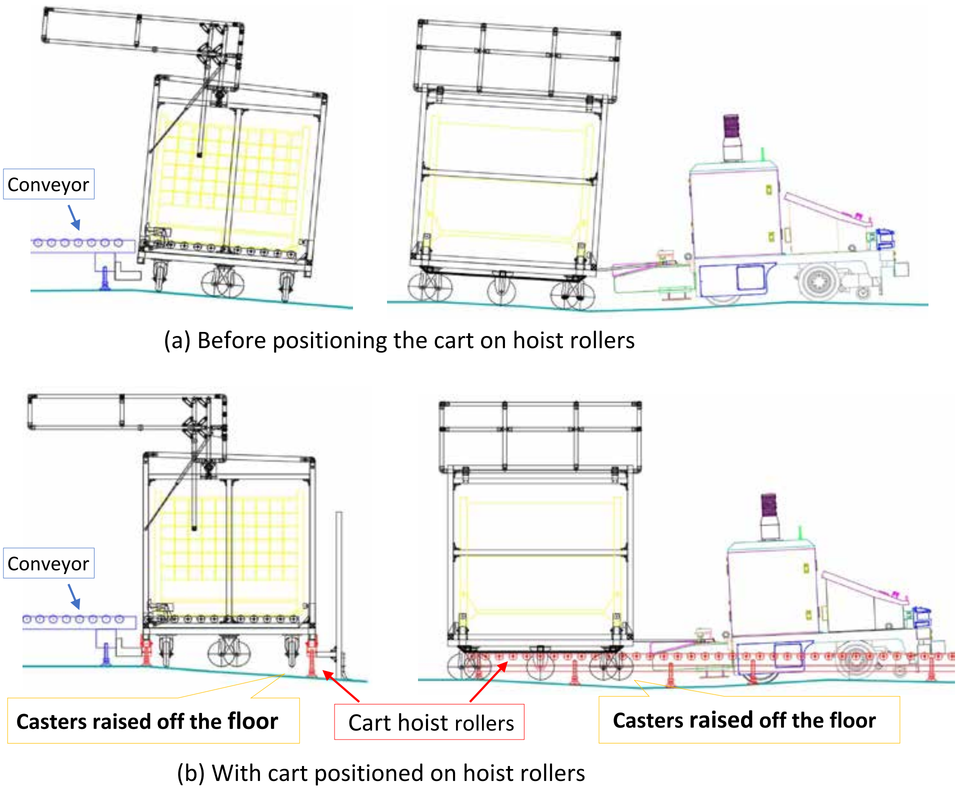

Fig. 4 System for raising the cart on a sloped floor

3. Outdoor AGV monitoring system

In this project, the AGV travels along an outdoor path out of sight of the employees and, in addition, there are few employees at the relay point in the path. There was concern that if some problem occurred in the AGV system, it might not be discovered immediately. In case a problem was not discovered right away, it might delay the transport of parts, thus causing a supply delay that could possibly stop production. To avoid that situation, it was considered necessary to continuously convey information on the condition of the outdoor AGV to the employees.

Systems for monitoring indoor AGVs have already been implemented at other JATCO production plants. However, one monitoring system is ancillary to an AGV acceleration control system and another system was adopted for the purpose of analyzing the uptime rate of the AGV. No system had been implemented previously for the purpose of notifying employees of an AGV problem like the function needed in the present project. Therefore, it was necessary to newly investigate an AGV monitoring system.

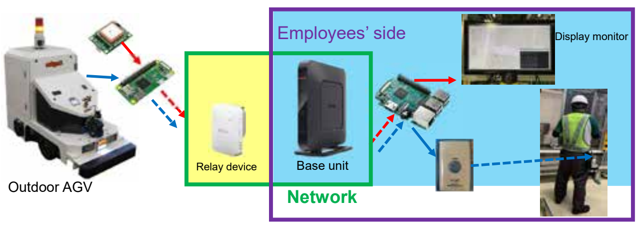

Fig. 5 Configuration of AGV monitoring system

The requests of the shop floor employees who would be using the system are summarized below.

(1) A function for notifying employees in real time of an AGV problem

(2) A function for showing the AGV's position

A study was then undertaken of a system for satisfying these requests.

The parts to be used in the system were then selected. Because the necessary input/output pins were on hand, it was decided to adopt a single board computer that allows easy control and is inexpensive. For the first function, a local area network was built between plant buildings so that a trouble signal issued by the AGV could be transferred close to the vicinity of the employees. This achieves real-time notification of a problem. For the second request, a small GPS module is used to enable position information to be obtained and displayed. The monitoring system for the outdoor AGV was constructed in this way (Fig. 5).

4. Safety measures

In order to operate the outdoor AGV safely, a traffic control system is needed for avoiding collisions with other ordinary vehicles traveling on the plant premises. At other JATCO plants, traffic signal facilities connected to a wireless network are used at intersections traveled by both outdoor AGVs and ordinary vehicles. In this project as well, traffic signal facilities connected to a wireless network, rotating red lights and other equipment were installed as safety measures.

The obstacle sensors used heretofore for preventing collisions with outdoor AGVs cannot detect the wheelbase of the trailer bed. Consequently, the height of the trailer bed must also be detected. An obstacle sensor capable of detecting the trailer bed height was added, thus improving safety with respect to the trailer.

5. Conclusion

The outdoor AGV system described here transports parts between separated buildings and loads/unloads cargo fully automatically. Practical application of this system was achieved by implementing the following measures.

(1) Development of an automatic braking system to enable travel on downward slopes.

(2) Development of a cart equipped with an automatic opening/closing door and a function for preventing incursion of wind and rain so as to enable outdoor transport.

(3) Development of an automated parts supply system constructed with cart hoist rollers for raising the entire cart.

(4) Development of a function for notifying employees in real time of an AGV problem and a function for displaying the vehicle's position in order to enable outdoor transport out of sight of the employees.

As further automation is promoted in the future, it is expected that it will be necessary to transport products of various sizes and difficult-to-handle products as well as to transport them over longer distances. We intend to adapt the present AGV system so that it can meet such needs.

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Improvement of product quality by introducing the Quality Design Sheet

Kumiko ANDERSSON* Makoto UCHIDA** Masao ARIMATSU**

Summary

In order to deliver products of the highest quality to customers, JATCO has regularly improved its quality assurance tools and endeavored to apply the optimal processes at all times. These efforts have markedly improved both R&D and manufacturing quality. However, various problems have remained such as rework caused by poor communication between departments and not identifying all quality risks in the early stages of the development process.

This article describes the Quality Design Sheet that has been created for the purpose of promoting “active communication” and “early visualization of quality risks” in order to solve these issues. The introduction of the Quality Design Sheet is contributing to the formation of a powerful quality assurance system throughout the entire company.

1. Motivation for creating QDS

In order to put products of higher quality and performance on the market, it is essential that the intentions of product planners and development engineers are transmitted accurately to manufacturing engineers, including suppliers, and made into definite realities. One commonly used method to accomplish that is quality function deployment (QFD), and one of the tools used in this process is a quality assurance (QA) table.

Like other companies, JATCO has used QA tables to improve product quality. In handing over the responsibility for a product from the previous process to the next one, problems have sometimes occurred when intentions were not fully and accurately conveyed owing to poor communication on both sides, among other factors. In addition, communication from the R&D department to the manufacturing department has often been one-way. When manufacturing issues occurred, depending on the schedule, there was not enough time to feed them back to the R&D department. There were not a few instances when manufacturing engineers had to resign themselves to proceeding without getting issues resolved.

In order to solve such problems, it was necessary to develop some tool to facilitate sufficient communication on both sides and to enable all issues, including ones related to manufacturing, to be identified and resolved in the early stages of the development process. The Quality Design Sheet (QDS) described here was thus created as a tool that further expanded QA tables.

Figure 1 presents examples of issues that occurred due to poor communication. Gear damage is listed as an example of a potential quality risk that can occur due to exceptionally high surface pressure caused by a large

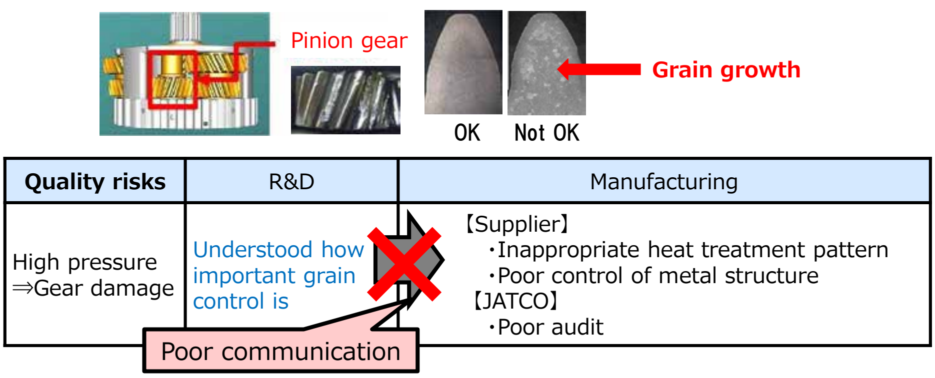


Fig. 1 Examples of issues due to poor communication

* New Business Develpoment Department ** Corporate Quality Assurance Department

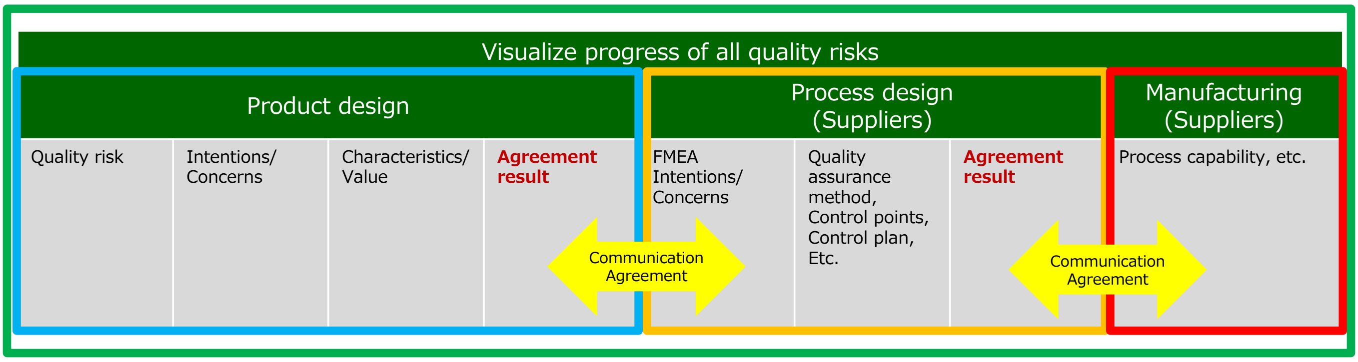


Fig. 2 Quality Design Sheet concept

torque input. R&D engineers understood the importance of crystal grain control as a crucial aspect of preventing such a risk. However, that information was not correctly conveyed to the manufacturing department so the metallographic structure was not properly controlled by applying an appropriate heat treatment pattern nor was sufficient care taken when conducting a process audit.

2. Aim of QDS and improvement

The QDS concept is outlined in Fig. 2. The QDS is divided into three stages: product design, process design and manufacturing. It is created by listing identified quality risks along the vertical axis and indicating along the horizontal axis the results of studies done by the responsible departments to solve the risks as the project progresses. The QDS has improved several processes compared with those of the traditional QA table for the purpose of solving issues originating in poor communication between departments. The following describes three major points in particular that have been improved.

1) Expanding the perspective for identifying quality risks

The QA characteristics that were identified previously were ones deemed important with regard to product functionality. They included characteristics that might possibly lead to serious risks in the field and ones concerning risks that previously occurred in the field for similar products. However, characteristics having a low possibility of becoming a risk in the field were not the target of identification, even ones that posed issues at the R&D stage and ones given close attention in design reviews of products have a high degree of novelty. Consequently, there was a problem that not all risks extending from R&D through manufacturing were included.

To solve this issue, the areas for identifying quality risks were expanded to include all aspects of concern regarding R&D and manufacturing. It was decided to call items that were newly identified as a result of this expansion of areas for identifying risks as quasi-QA characteristics.

2) System for promoting recognition of the importance of communication and agreement

In order to avoid quality risks, it is necessary for all processes to have the same awareness by accurately communicating to the production engineering department the quality risks identified by the R&D department and further to communicate them to the manufacturing department without fail. Previously, it was left to downstream processes to try to understand and avoid quality risks that occurred in upstream processes.

Signature columns (indicated as agreement result in red in Fig. 2) were provided on the QDS for indicating “who, when and what” was agreed at the time of the transition at each stage from R&D to production engineering and then to manufacturing. This was done to create a system so that a project would only proceed to downstream processes after obtaining their understanding and agreement concerning quality risks identified in upstream processes.

3) Visualizing the whole picture and progress of quality risks

Latent quality risks in a product are all listed on the QDS, making it possible to visualize the whole picture of risks and to understand at a glance how much progress is made at each stage in addressing them. This enables quality risks to be managed.

The QDS was initially applied as a tool at the stage prior to making a QA table. First, all the quality risks identified at the R&D stage were listed on the QDS and the related QA characteristics were investigated. Subsequently, they were transferred to a QA table at the time of the transition to the manufacturing stage. The QA table alone was then used in the processes from manufacturing onward.

This made it possible for all processes to share quality risks. However, mistakes occurred in transferring information to the QA table, and both the QDS and the QA table were used together at the transition stages from R&D to manufacturing, making it difficult to manage the latest information on quality risks. These and other new issues occurred because of double management of the two forms. Therefore, an improvement was made by combining the best points of the QDS and QA table into one integrated form (Fig. 3).

This created a comprehensive QDS that incorporated the function of the QA table in a new form. Internally, the name of this form is officially registered as "QA Table (Quality Design Sheet)", but QDS will be used as the unified designation from section 3 below.

3. Composition of QDS

Monozukuri mainly consists of the stages of R&D, production engineering and manufacturing. The departments responsible for each stage bear the responsibility for delivering products of excellent quality to customers. Here, we will explain the role that each department plays in this process.

3.1 Role and responsibility of R&D department (in blue framework on left side of Fig. 2)

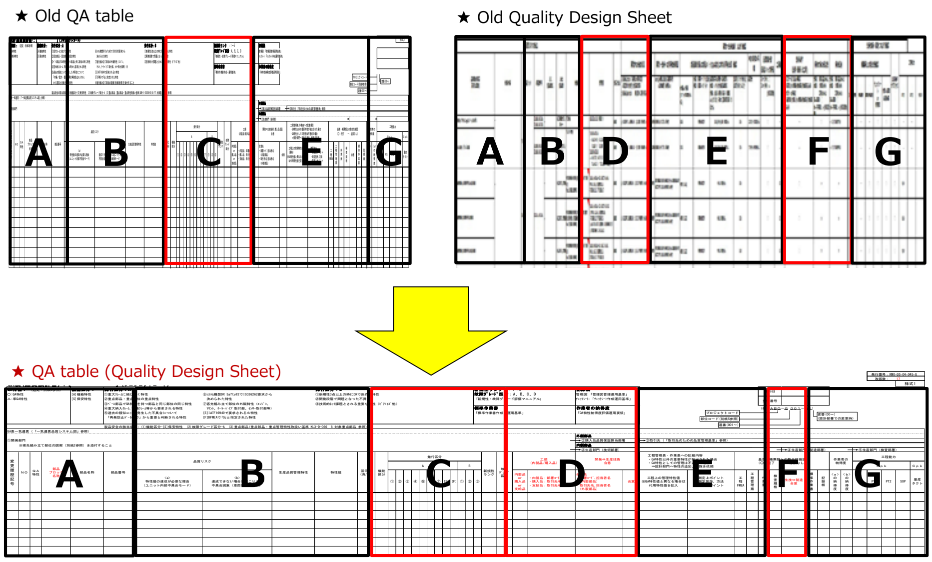
This department identifies the quality risks of all parts, assesses the degree of influence of the risks in terms of functional characteristics, and determines the categories of characteristics according to their level of importance. The categories of characteristics are classified into three levels: QA characteristics, quasi-QA characteristics and general characteristics.

How these characteristics are related to risks pertaining to the vehicle and the AT/CVT is communicated to the production engineering department along with the intentions and concerns of the R&D department.

Discussions are held until both the R&D department and the production engineering department are satisfied. At the stage where an agreement is reached, the production engineering department signs the agreement column of the QDS (Red letter in Fig. 2).

3.2 Role and responsibility of production engineering department (yellow center frame in Fig. 2)

After receiving and agreeing to the intentions



A: Risk
B: Characteristics/Value
C: Perspective for identifying characteristics
D: Agreement between R&D and production engineering
E: Document for manufacturing
F: Agreement between production engineering
G: Process capability

Fig. 3 Integration of QA table and Quality Design Sheet and concerns of the R&D department, the production engineering department studies process designs and management methods corresponding to the QA characteristics, quasi-QA characteristics and general characteristics. Management methods are specified according to the respective categories of characteristics. The production engineering department provides feedback to the R&D department about concerns raised by the manufacturing department in cases where the methods of measuring dimensions or strength are difficult to apply or the values set for characteristics are stricter than before.

In the course of repeating these studies, the items agreed with the R&D department are incorporated in a process failure mode and effects analysis (PFMEA) and reflected in process designs. In addition, process management aspects with regard to concrete methods of measuring and assuring each characteristic are determined, and the intentions and concerns of the production engineering department are communicated to the manufacturing department.

The manufacturing department and the production engineering department hold discussions until the former understands and accepts the latter's ideas. At the stage where an agreement is reached, the manufacturing department signs the agreement column of the QDS (Red letter in Fig. 2).

3.3 Role and responsibility of manufacturing department (right red frame in Fig. 2)

After receiving and agreeing to the intentions and concerns of the production engineering department, the manufacturing department aligns the forms of the former department, including the PFMEA results, with its own forms such as process management charts, work procedure charts and others. It also evaluates the level of acceptance among manufacturing workplace employees and describes the results on the QDS.

Moreover, after all the production preparations are completed and production trials begin at the plant, the manufacturing department inspects the values of each characteristic using the assurance methods agreed with the production engineering department. Manufacturing processes and methods are continuously improved until the process capability for each characteristic satisfies the specific standard. It is confirmed that the quality risks communicated by the R&D and production engineering departments do not occur.

Process capability values are also measured in the initial production control phase following the launch of mass production, the results are recorded on the QDS and the measurements are concluded (measurement of QA characteristics continues until the end of production).

3.4 Role of corporate quality assurance department (green peripheral frame in Fig. 2)

In order to proceed with monozukuri based on the QDS, it is necessary to monitor the overall progress and to promote the smooth transfer of information and agreements between departments.

That is the role of the corporate quality assurance department. When difficult-to-solve issues occur or agreements cannot be reached between departments, the corporate quality assurance department clarifies the factors involved and works together with the person in charge of each issue to propose methods of promoting solutions; it leads this activity until the issues are closed.

This department assures the quality of products by conducting confirmation activities at various places throughout the company after issues have been closed at the R&D, production engineering and manufacturing stages, respectively. That is done to confirm that quality is being assured as agreed by the departments involved and that all the concerns have been solved, among other things.

4. Effects of QDS

This section presents specific examples of the effects obtained at each stage from the quality improvements made by using the QDS.

1) Expansion of the perspective for identifying quality risks

As explained earlier, previously, the R&D department was mainly responsible for identifying quality risks. Consequently, quality risks in production engineering and in manufacturing were not completely identified, and risks appeared after production trials were initiated.

The introduction of the QDS provides opportunities for the production engineering and manufacturing departments to confirm early on the quality risks identified by the R&D department and for both departments to provide feedback to R&D. Accordingly, quality risks in production engineering and manufacturing can now be included from the early stage of R&D activities.

Moreover, feedback is also possible from the cost planning department, after-sales service department and others, enabling risks to be identified that take into account cost attainment, field serviceability and other aspects. The expanded perspective for identifying risks has therefore been greatly effective in improving quality.

No.	Model	Part name Number	Quality characteristic	Reason	Possible defects
1	CVT	FIX PRIMARY 3--- -----	Slot contact angle ○° +○°/-○°	Belt slip, roller wear	Vehicle does not move forward and shift failure
2	CVT	FIX PRIMARY 3--- -----	Diameter of roller slot Ø○mm +○/-○	Belt slip, roller wear	Vehicle does not move forward and shift failure
3	CVT	FIX PRIMARY 3--- -----	Sheave waveness ○%, ○μm ≥ ±○%	Belt wear, sheave face wear and belt slip	Vehicle dose not move
4	CVT	FIX PRIMARY 3--- -----	Sheave roughness ○○	Belt wear, sheave face wear and belt slip	Vehicle does not move
5	CVT	FIX PRIMARY 3--- -----	Sheave roughness ○ min	Belt wear, sheave face wear and belt slip	Vehicle dose not move
6	CVT	FIX SECD 3--- -----	Slot contact angle ○○° +○°/-○°	Belt slip, roller wear	Vehicle does not move forward and shift failure
7	CVT	FIX SECD 3--- -----	Diameter of roller slot Ø○mm +○/-○	Belt slip, roller wear	Vehicle does not move forward and shift failure
8	CVT	FIX SECD 3--- -----	Sheave waveness ○%, ○μm≥±○%	Belt wear, sheave face wear and belt slip	Vehicle dose not move
9	CVT	FIX SECD 3--- -----	Sheave roughness ○○	Belt wear, sheave face wear and belt slip	Vehicle dose not move
10	CVT	FIX SECD 3--- -----	Sheave roughness ○ min	Belt wear, sheave face wear and belt slip	Vehicle dose not move

Fig. 4 Quality risk card

2) Improvement of understanding and awareness in manufacturing workplaces through communication of information and agreements

Quality risks are now communicated from the R&D department to the production engineering and manufacturing departments in the early stages of R&D for the purpose of reaching agreements on the items, categories and values of characteristics. As a result, the production engineering and manufacturing departments can now provide feedback to the R&D department approximately one year earlier than before.

In addition, the R&D department now explains the meaning and importance of the values of the various characteristics directly to the production engineering and manufacturing departments. This eliminates any discrepancies between them and has improved understanding on the part of the production engineering and manufacturing departments and raised awareness of the importance of workplace management.

An activity undertaken to improve quality at JATCO MEXICO S.A.DE C.V. is explained here as one example.

After receiving a detailed explanation of quality risks from the R&D department, the production engineering and manufacturing departments created a quality risk card (Fig. 4) based on the QDS. This card has made it possible for employees to know at a glance what types of risks might occur in the event that the processes they are responsible for do not satisfy the required characteristics. Employees always carry this quality risk card around together with their employee ID card, enabling them to confirm risks at any time. This activity has heightened all the employees' awareness of quality and has served to improve quality significantly.

It has also enabled preceding and following processes to thoroughly discuss quality risks that might occur and to reach agreements on their respective roles and measures for solving them. That has worked to heighten awareness of responsibilities in each process. For example, responsibilities are handed off to the next process like a baton in a relay race, so the locus of responsibility is clearly defined. Not only the R&D, production engineering and manufacturing departments, but also other related departments are able to become involved in resolving issues while cooperating constructively.

It has also led to more active daily communication not only about items on the QDS, but also about the production schedule, costs and other work procedures. Thus, it also has a secondary effect of enabling smooth consultations.

3) Visualization of the whole picture and progress of quality risks

Information is entered on the QDS from left to right as a project proceeds, making it easy to see the progress being made in dealing with each issue. It is now possible to understand at a glance at which stage and in what way quality has been assured against the risks identified by the R&D department.

As a result, the whole picture of which specific parts are unfinished in which processes can now be seen, enabling action to be taken immediately. This includes dealing with tiny issues that tend to be obscured by much larger ones and avoiding putting off action on risks deemed to have a low rate of occurrence, as well as matters lacking an agreement such as the values of characteristics, methods of assuring quality and process capabilities not yet attained.

5. Future efforts

We want to undertake efforts to deal with the following two points as specific issues.

1) Incorporating this process in related departments and continuous improvement

The corporate quality assurance department regularly holds meetings to explain the QDS to related internal departments and suppliers for the purpose of promoting the penetration of the QDS. Questions and opinions expressed at such meetings are gathered for promoting further improvements.

A company-wide improvement activity was launched in fiscal 2019 in which efforts were undertaken to improve the R&D process and manufacturing process in cooperation with related departments. This activity revealed certain problems in the operation of the QDS as well as many points requiring improvement, which have been consolidated into the following six issues.

(1) Reduction of variation in the extraction level of quality risks

(2) Clearly defining the methods for validating characteristics and their values

(3) Method of agreeing on how to classify the categories of characteristics

(4) Clearly defining the rules for issuing standards

(5) Method of agreeing on process management characteristics and quality assurance procedures

(6) Penetration of rules for deploying the results of identifying process capabilities

Persons responsible for these issues were determined and working groups were formed to solve them. The results of studies undertaken by the working groups were incorporated in criteria for applying the QDS and in other related standards and are already being put to practical use. They will also be reflected in new projects in the future.

2) Company-wide penetration including overseas production plants and suppliers

Regular meetings and educational activities are continuously being undertaken so that overseas production plants and suppliers also understand the importance and practical application methods of the QDS. It is also planned to include explanations of the QDS in new employee training programs so as to improve quality awareness on a company-wide basis.

6. Conclusion

As explained here, the QDS is a tool for smoothly connecting the entirety of monozukuri from the most upstream process to the final process. It contributes to improving quality by strengthening the ties between the R&D, production engineering and manufacturing departments.

The products to which the QDS has been applied in R&D and manufacturing processes have attained even better quality, confirming that this tool has produced significant results. The corporate quality assurance department naturally takes measures against issues that occur in the process of creating the QDS. It is also actively involved in dealing with any issues that occur as the various departments proceed with their studies. Not only does it create standards for the QDS, it also works vigorously to help with the revision of standards issued by each department.

Going forward, we aim to further improve quality by penetrating the QDS thoroughly throughout the company.

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Accelerated product quality verification through real-world high-mileage driving in a short period of time

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Summary

The model cycle of JATCO's transmissions has been shortened in recent years in order to cope with rapid changes in customers' vehicle requirements and in the environment surrounding vehicles. Consequently, it has been necessary to shorten the period for verifying the real-world quality of our new products and ones that incorporate new technologies. We have so been investigating high-mileage transmissions recovered from customers. However, that method alone requires considerable time to verify the real-world quality of our products in the field. Consequently, it has become necessary to recover high-mileage products more quickly in order to cope with the need for a shorter model cycle.

To ensure that we supply products that satisfy customers, it is absolutely essential to eliminate any dissatisfaction with the transmission perceived by customers while driving. Toward that end, we initiated an activity to install our new products and ones featuring new technologies in vehicles that undergo high-mileage driving in a short period of time in real-world environments. This approach enables quicker verification of product quality in the field than the previous method. Data are constantly collected during driving in a wide range of environments and situations for subsequent analysis, enabling confirmation of the degree of customer satisfaction with the transmission and the occurrence of high input loads that might lead to transmission failure. This article describes the market fleet activity that enables high-mileage driving in the real world in a short period of time and presents typical results obtained with this method.

1. Introduction

JATCO has been manufacturing automatic transmissions for over 50 years. In order to continue to provide products satisfactory to customers, we have been recovering used transmissions that accumulated high mileage during customer use and investigating them for the purpose of improving the quality of our products in the field.

The model cycle of our transmissions has been shortened in recent years in order to respond to rapid changes in customers' vehicle requirements and in the environment surrounding vehicles. In this connection, we are continuing to verify the quality of our transmissions in the field in relation to product deterioration revealed by investigations of high-mileage transmissions recovered from customers as has been done to date. In addition, in

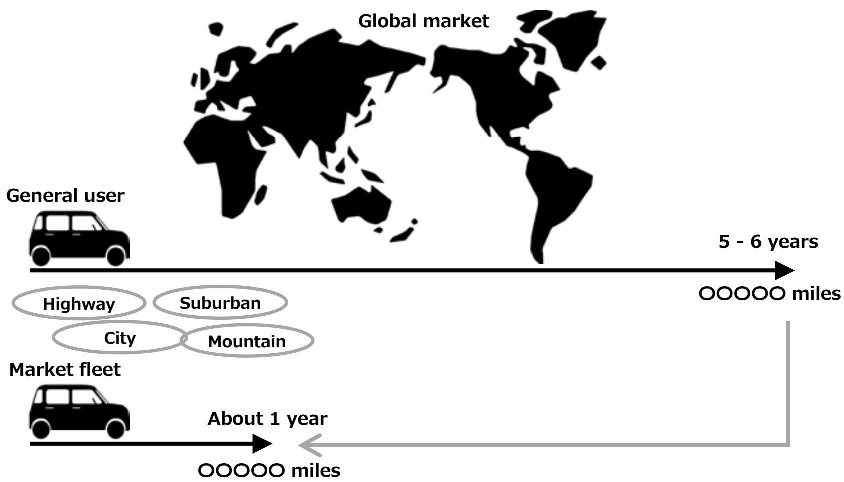


Fig. 1 Market fleet overview

* Corporate Quality Assurance Department

order to cope with the shorter model cycle, it has become necessary to recover high-mileage products in a shorter period of time and to provide immediate feedback.

Moreover, what drivers demand of transmissions while driving has also become more rigorous owing to diversification of driving environments and the ways vehicles are being driven nowadays. To solve these issues, we decided to initiate a market fleet activity in order to recover and investigate high-mileage transmissions quicker than with the previous method and thereby accelerate verification of product quality in the field.

2. Overview of market fleet activity and driving conditions

This section gives an overview of the market fleet activity and the driving conditions. Market fleet refers to a method of shortening the period for recovering high-mileage transmissions by having a staff of local professionals drive fleet cars approximately 500-1,000 km a day. In this way, the high mileage that ordinary users would accumulate in 5-6 years can be reached in approximately one year.

The routes driven include city roads, suburban roads, highways, mountain roads and others so as to reproduce the driving environments and ways of driving vehicles actually experienced by customers. This approach achieves highly accurate verification of product quality in the field (Fig. 1 and Table 1).

In addition, data on the vehicle and transmission are constantly collected during driving for subsequent analysis. This enables verification of the accuracy of the input loads that were assumed in the product development process.

The transmissions of market fleet vehicles that have reached the mileage target are then recovered from the field, disassembled and analyzed. If the results show that the component parts have deteriorated more than was expected or reveal deterioration suggestive of a potential cause of a failure, the input loads inducing such deterioration are analyzed based on the driving data recorded for the vehicle and transmission. This makes it possible to verify the progression of such deterioration and to make improvements so as to prevent potential problems.

Fleet vehicle drivers are also asked to comment on vehicle behavior from a customer's perspective. Vehicle operating conditions relevant to their comments are

Table 1 Driving conditions

Vehicle	Mini-vehicles ~ Ordinary passenger cars
Regions	US & JPN
Routes	City roads, Suburban roads, Highways, Mountain roads, etc.
Mileage / time	High mileage that a customer accumulates in 5 or 6 years is reached in about 1 year.
Procedure	Vehicles are driven by professional staff to focus on the ways of driving in the test area

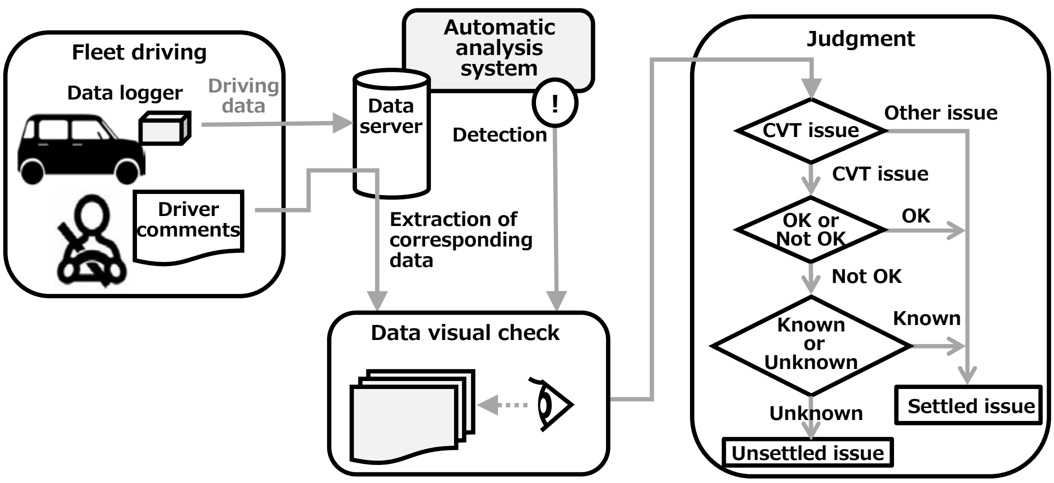


Fig. 2 Driving information analysis method

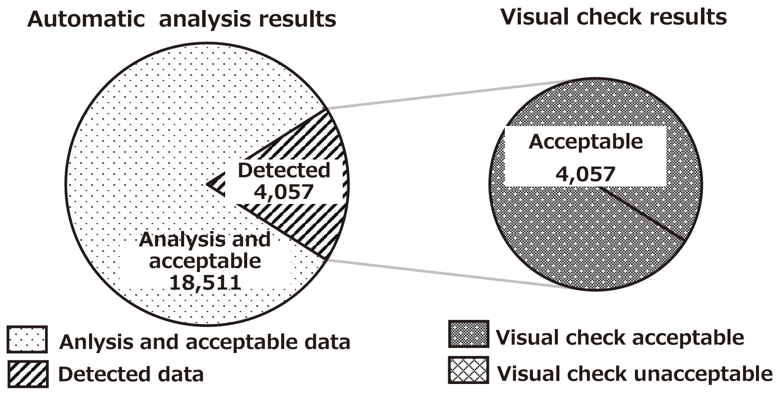


Fig. 3 Automatic analysis results (example for one vehicle)

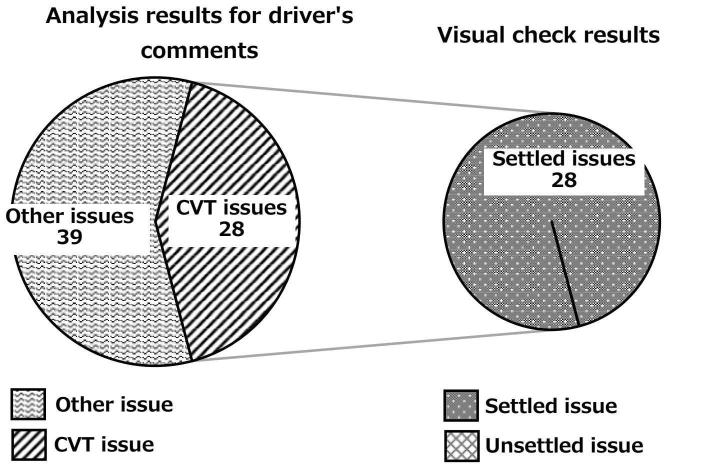


Fig. 4 Analysis results for drivers' comments (example for one vehicle)

analyzed based on vehicle and transmission data. This enables comments concerning the transmission to be separated from others pertaining to the vehicle itself. Comments can thus be divided into those at a level that customers might perceive as unsatisfactory vehicle behavior, already known phenomena that are not a problem, and unknown phenomena for which improvement is necessary, among others. This leads to the development and manufacturing of products satisfactory to customers (Fig. 2).

3. Results of market fleet implementation

3.1 Results of automatic analysis of driving data

Data on the vehicle and transmission collected during driving are analyzed by an automatic analysis system. This enhances detection sensitivity for extracting data without omitting or overlooking anything with respect to high input loads that might cause a transmission to fail during customer use. The staff then analyze the extracted data in detail in order to judge whether or not high input loads were actually applied.

All the data collected during driving pertaining to the vehicle, location and mileage target were automatically analyzed and also analyzed in detail. The results showed that there were no high input loads to the transmission as described in the following example. The automatic analysis system detected approximately 4,000 cases of data suspected of indicating high input loads that might lead to acceleration failure or inability to drive. However, the results of the detailed analysis revealed that there was no problem with regard to any of the data (Fig. 3).

3.2 Analysis results for drivers' comments

Concerns expressed by the fleet vehicle drivers about vehicle behavior from a customer's perspective were analyzed and the results revealed that there were no unknown phenomena requiring improvement. As one example, there were 28 cases of concern about the shifting of the transmission and other aspects. However, as a result of comparing the drivers' comments with the phenomena in question, it was found that all cases involved known phenomena that did not require any improvement (Fig. 4).

3.3 Results of disassembly and inspection

Transmissions were recovered from vehicles that reached the mileage target in each region and the degree of deterioration was confirmed quantitatively by disassembling the units for a visual inspection and precision measurements. The results of the visual inspection of the appearance of disassembled parts confirmed that the gears, clutches and other components were not broken or damaged. No deterioration was found that would lead to transmission failure (Fig. 5).

Precision measurements were made of wear and other conditions. The data were compared with the results of previous investigations of high-mileage transmissions recovered from the field. It was confirmed that there was no abnormal deterioration (Fig. 6).

As described here, for transmissions that completed the mileage target without any problem, the level of deterioration of the component parts was within the expected range. There were no signs of anything that might become the cause of transmission failure. The results thus verified that the investigated transmissions possessed sufficiently high quality for high-mileage driving in the real world.

4. Conclusion

The implementation of the market fleet activity has markedly shortened the time needed to investigate used transmissions compared with the previous method of recovering transmissions from customers. This has made it possible to verify the real-world quality of our products much quicker than before.

Data on the vehicles and transmissions recorded during driving were comprehensively analyzed automatically and drivers' comments about vehicle behavior were analyzed in detail. This method provides highly reliable and accurate verification of real-world product quality as well as enabling valuable feedback to the quality assurance standards used in product development and manufacturing.

Accordingly, the market fleet activity described here is an extremely effective method of assuring product quality in the real world.

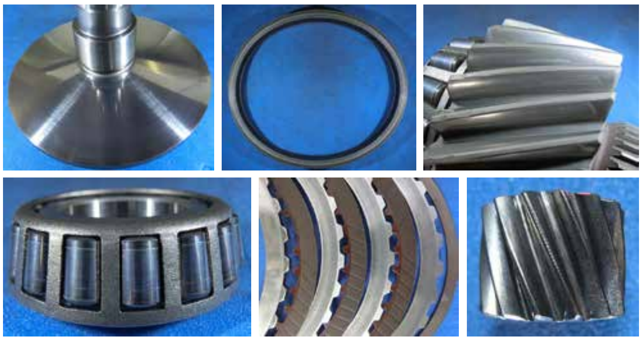
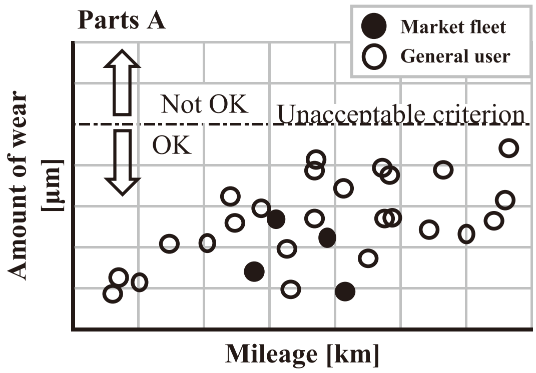


Fig. 5 Disassembly and visual confirmation



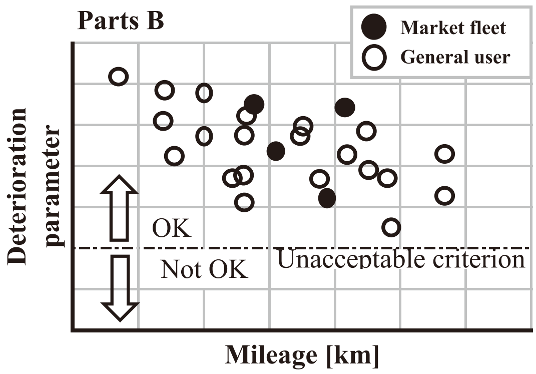


Fig. 6 Precision measurement result

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Introducing the Jatco CVT7 (JF015E) for the Renault Samsung XM3

The Jatco CVT7 (JF015E) is equipped on the new-generation XM3 that Renault Samsung Motors Co., Ltd. released in South Korea in March 2020. Mated to Renault Samsung's third-generation 1.6L 4-cylinder engine, the JF015E is distinguished by low friction and wide ratio coverage achieved with its auxiliary transmission. In addition, a thorough review of the control system has contributed to outstanding fuel economy and driveability. A notable feature of the XM3 is the application of a new electrical/electronic (E/E) architecture shared by Renault and Nissan. The JF015E has been optimally tuned to match the crossover vehicle concept that reflects the needs of the South Korean market. As a result, the XM3 delivers both the dynamic performance of an SUV and the interior quietness of a sedan, making the vehicle highly popular with customers.

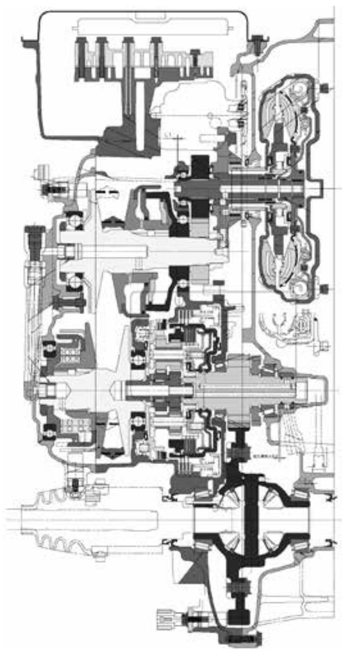


Fig. 1 Main cross-sectional view

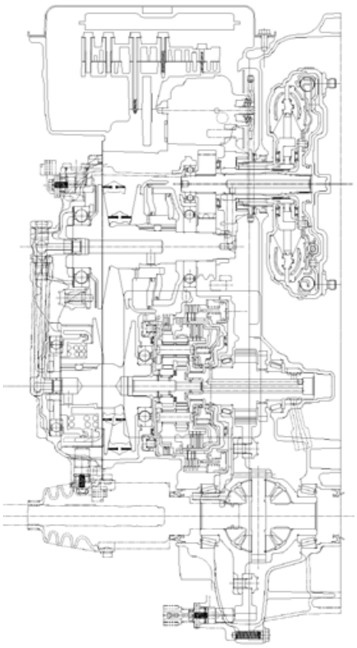
Table 1 Specifications of JF015E

Torque capacity	150 Nm
Torque converter size	205 mm
Pulley ratios	2.2 - 0.55
Auxiliary transmission gear ratios	1st 1.821 2nd 1.000 Rev. 1.714
Ratio coverage	7.3
Final gear ratio	3.882
Selector positions	P, R, N, D + Paddle shift mode
Overall length	334 mm
Weight (wet)	70.1 kg (2WD)

Renault Samsung XM3

Introducing the Jatco CVT7 (JF015E) for the Nissan Magnite

The Jatco CVT7 (JF015E) is equipped on the new-generation Nissan Magnite that Nissan Motor Co., Ltd. released in India in December 2020. The JF015E is distinguished by wide ratio coverage achieved with its auxiliary transmission and low friction. These features combine with a 1.0L 3-cylinder turbo engine to provide improved driveability, including launch and acceleration performance with excellent response, along with class-leading fuel economy and compliance with India's Bharat Stage VI-1 exhaust emissions standards.



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Ratio coverage		7.3
Final gear ratio		3.882
Number of selector positions		5(P-R-N-D-L)
Overall length		342.1 mm
Weight (wet)		67.3 kg

Nissan Magnite

Introducing the Jatco CVT8 (JF016E) for the Nissan Rogue

The Jatco CVT8 (JF016E) is equipped on the new-generation Rogue that Nissan Motor Co., Ltd. released in North America in October 2020 for the 2021 model year. The JF016E incorporates a newly designed control valve that dramatically improves shift response and also further reduces friction. These improvements help to elicit the full performance of the newly developed PR25 engine and contribute to improved power performance, an enhanced feeling of driveability and higher fuel economy. The adoption of shift-by-wire also contributes to improved interior roominess.

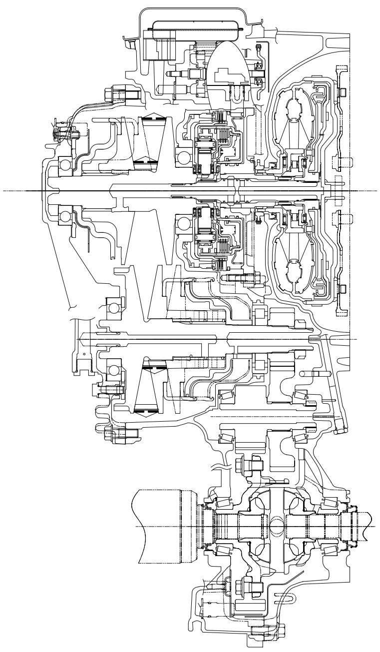


Fig. 1 Main cross-sectional view

Table 1 Specifications of JF016E

Torque capacity	250 Nm
Control system	Electric
Torque converter size	236 mm
Pulley ratios	2.631-0.378
Ratio coverage	7.0
Reverse gear ratio	0.745
Final gear ratio	5.694
Number of selector positions	5(R ← N ← H → N → D/M) + P Button
Overall length	362.3 mm
Weight (wet)	94.0 kg (2WD), 94.8 kg (4WD)

Nissan Rogue

Introducing the Jatco CVT7 (JF015E) for the Changan EADO PLUS

The Jatco CVT7 (JF015E) is installed on the Changan EADO PLUS that Changan Automobile Co., Ltd. released in China in September 2020. This marks the first application of a JATCO CVT on Changan vehicles. The combination of the JF015E, featuring wide ratio coverage thanks to its auxiliary transmission, and Changan's Bluecore 1.6L GDI 4-cylinder engine delivers high fuel economy. Moreover, the JF015E provides the smooth driveability that distinguishes a CVT as well as dynamic driving performance achieved with its 8-speed manual shift mode, qualities that are highly popular with customers.

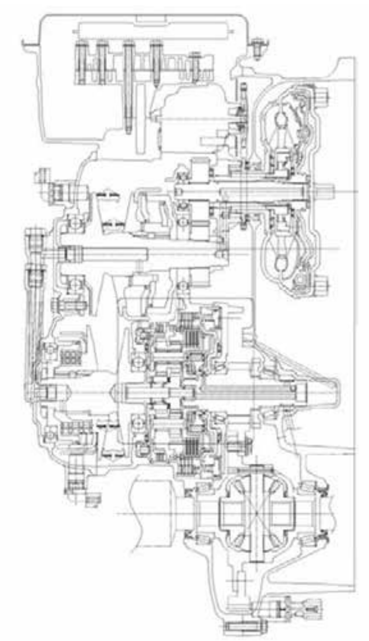


Fig. 1 Main cross-sectional view

Table 1 Specifications of JF015E

Torque capacity		156 Nm
Torque converter size		218 mm
Pulley ratios		2.2 - 0.55
Auxiliary transmission gear ratios	1st 2nd Rev.	1.821 1.000 1.714
Ratio coverage		7.3
Final gear ratio		3.882
Selector positions		P, R, N, D + Manual mode
Overall length		347.4 mm
Weight (wet)		70 kg

Changan EADO PLUS

Highlights of 2020

1. Participation in KSAE's annual conferences

Two technical presentations in total were given at the annual spring and autumn conferences sponsored by the Korean Society of Automotive Engineers (KSAE). Owing to the impact of the novel coronavirus pandemic, the annual spring conference was convened from July 1-4, approximately one month later than the original schedule. An employee of JATCO Korea Engineering Corporation presented a technical paper entitled, "Development of a Continuously Variable Transmission without electric oil pump for Start and stop system," in which he explained start-stop coasting. At the annual autumn conference that was held from November 18-21, a JATCO Korea Engineering Corporation employee presented a technical paper entitled, "A Study on Bolt Fastening Structure for Plastic Valve Body of CVT," in which he explained a new bolt fastening structure designed for plastic valve bodies.

2. JATCO's 50th Anniversary Ceremony

A ceremony to celebrate JATCO's 50th anniversary was held on January 28 at Fujisanmesse in the city of Fuji, where the company originated and the headquarters is still located today. Approximately 2,000 people, including present and former employees and other related persons, attended the ceremony to mark the half-century since Japan Automatic Transmission Co., Ltd. was founded in January 1970. Four representative company officers gave presentations about JATCO's history and future. Following an opening message by President Teruaki Nakatsuka, historical talks were given by successive presidents and the current chairman. In addition, Yoshikatsu Kawaguchi, a native of Fuji and the former goalkeeper of Japan's national soccer team, gave a message to JATCO about going forward toward the future based on his own personal experience. After the conclusion of a gigantic balloon relay in which everyone participated, JATCO's new employees made a powerful declaration for the future, and the ceremony closed with a commemorative photograph of all the attendees.

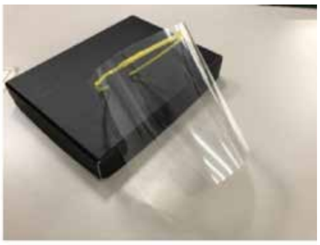
3. Production of medical gowns and face shields

As part of JATCO's support for medical workplaces that are dealing every day with the novel coronavirus pandemic, the company produced medical gowns from May through June and provided them to the Fuji Medical Association in the city of Fuji where the headquarters is located. The gowns were made from polyethylene sheets used as packaging material at plants and other places and were produced in the clean room of the CVT plant assembly line in the Fuji No. 1 Area.

Medical face shields were also produced from April to June and were provided through the Fuji Medical Association to medical workplaces of the Fujinomiya Medical Association, Fuji Dental Association and other organizations.

Medical gowns produced

Frames produced with a 3D printer



Face shield

4. Recipient of GM's 28th annual Supplier of the Year Award

JATCO received a 28th annual Supplier of the Year Award from General Motors (GM). This was the third time JATCO has received this award since fiscal 2016. JATCO was one of 116 suppliers from 15 countries, not including the United States, that were recognized for making continuous contributions exceeding GM's expectations, providing outstanding value and bringing about novel innovations. GM selects companies once a year for its Supplier of the Year Award based on the results of the Strategic Supplier Engagement (SSE) program under which GM regularly evaluates its suppliers, among other criteria. This difficult-to-win award is given only to approximately 100 companies, representing just 1% of the some 10,000 suppliers that are eligible for evaluation. A virtual awards ceremony was held on June 24.

5. Jatco CVT-S-equipped minivehicles named "2020-2021 K Car of the Year"

The Japan Car of the Year awards for 2020-2021 were announced on December 7. The Nissan Roox, the Mitsubishi eK X space and Mitsubishi eK space, all equipped with the Jatco CVT-S, won the K Car of the Year award. This marks the second consecutive year for winning this award following the Nissan Dayz last year. The Japan Car of the Year Executive Committee cited the following reasons for giving these vehicles the K Car of the Year award.

"The cars raised the standard for minivehicles with their highly stable, easy-to-control driving performance, while also providing the outstanding practicality of a super height wagon. The excellent quality of the interior trim and seat comfort were also highly evaluated. Moreover, another notable attractive feature is the adoption of the ProPILOT advanced safety/driver assistance system that delivers performance equal to that of larger registered vehicles."

Source: Japan Car of the Year website

6. JATCO Group companies certified as "Health and Productivity Management Organization 2020"

The Ministry of Economy, Trade and Industry (METI) announced on March 2 the enterprises certified as a "Health and Productivity Organization 2020". The Nippon Kenko Kaigi certified 1,481 organizations in the large enterprise category and 4,723 organizations in the small and medium-sized enterprise category. This time all JATCO Group companies were certified, including JATCO Ltd and JATCO Engineering Ltd in the large enterprise category and JATCO Plant Tec Ltd and JATCO Tool Ltd in the small and medium-sized enterprise category. It is an especially notable achievement that JATCO and JATCO Engineering were recognized as "White 500" organizations, representing the top 500 organizations in the large enterprise category.

7. QC Circle Congress of the Technology Promotion Association (Thailand-Japan)

The QC Circle Activities Congress for Fiscal 2020 Thailand Domestic Quality Awards was held in an event hall at the Bangkok International Trade & Exhibition Centre (BITEC) in Bangkok from September 1-4. The Gleung-Gat-Jor Team participated in the congress as the representative of JATCO (Thailand) Co., Ltd. The team presented its QC activity entitled, "Eliminate Dent inside Diameter Dai 45.920 at SLIDE-PRI Line" and received the Silver Prize.

8. Receipt of Shizuoka Prefecture Governor's commendation award for safety achievement in fiscal Reiwa 2

An employee at the AT Plant in the Kambara Area received a safety achievement commendation award given by the Governor of Shizuoka Prefecture. The award was presented at an awards ceremony sponsored by the Shizuoka Prefecture Association for Safety of Hazardous Materials on November 18. This award is given to individuals who have contributed to promoting improvement of safety standards through efforts over many years to prevent hazardous material accidents in workplace facilities and to foster voluntary safety activities.

Memo

Patents

1. OIL PUMP

(Fig. 1)

Application Number : 2015-63701
Application Date : 26.3,2015
Patent Number : 6381469
Registration Date : 10.8,2018
Title : OIL PUMP
Inventors : Atsuyuki Ide

【SUMMARY OF THE INVENTION】

An oil pump is provided with an inner rotor having external teeth, an outer rotor disposed in a loose-fit state within a pump chamber and having internal teeth meshed with the external teeth, a ring-shaped pressure chamber provided adjacent to the pump chamber in a direction of a rotation axis, a discharge opening for connecting the pump chamber and the pressure chamber, and a cylindrical discharge passage having one end connected to the pressure chamber and the other end serving as a connection opening. A portion of the discharge passage is located to open inside of the outer periphery of the pressure chamber when viewed in the rotation axis direction. The one end reaches to a middle of the pressure chamber when viewed in a radial direction of the rotation axis, and the discharge passage and the pressure chamber are in direct communication with each other.

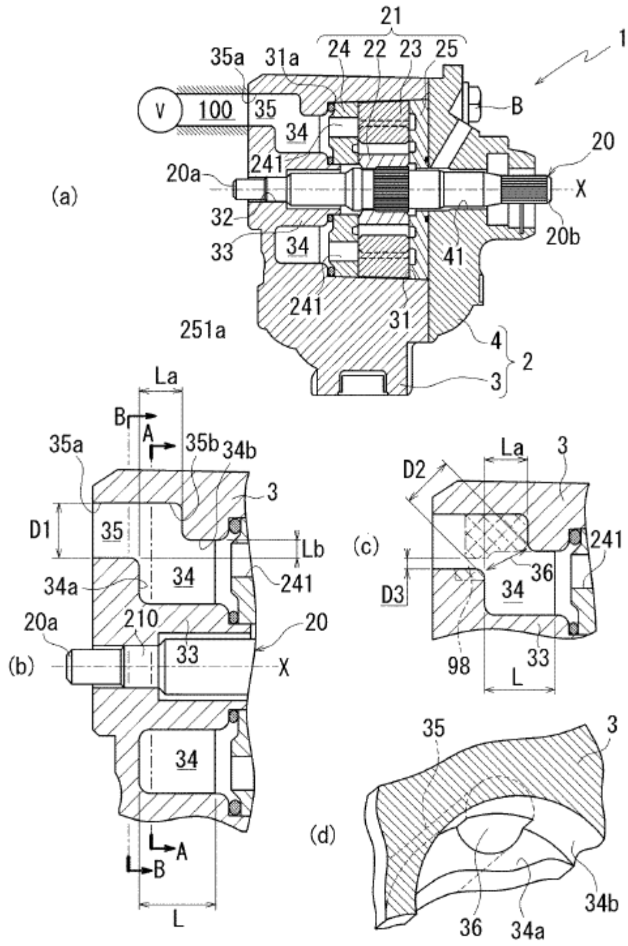


Fig. 1

2. Vacuum carburization method and vacuum carburization device

(Fig. 2)

Application Number : 2015-26472
Application Date : 13.2.2015
Patent Number : 6412436
Registration Date : 5.10.2018
Title : Vacuum carburization method and vacuum carburization device
Inventors : Kosuke Ida, Nobuhiko Inoue, Kojiro Suzuki, Shigeru Kano

【SUMMARY OF THE INVENTION】

The present invention provides a vacuum carburizing method capable of carburizing a workpiece with high quality while preventing the occurrence of spot-like excessive carburization.

A vacuum carburizing method for carburizing a workpiece disposed in a carburizing chamber by injecting a carburizing gas into a carburizing chamber in a decompressed atmosphere, wherein a gas injection amount of the carburizing gas injected into the carburizing chamber is: Calculated based on the volume of the workpiece in the carburized state in the carburizing chamber, the volume of the carburizing chamber, the total surface area of the workpiece, and a constant set based on the type of carburizing gas, A gas injection amount of carburizing gas is injected into the carburizing chamber.

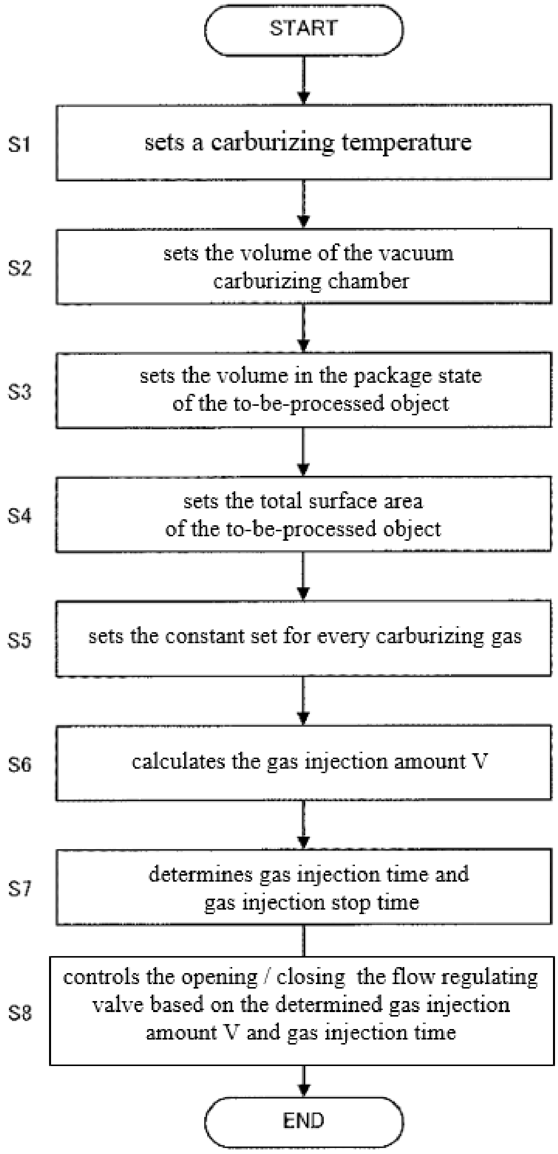


Fig. 2

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