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Preface

JATCO Technical Review 第5号に寄せて

Toward Publication of *JATCO Technical Review* No. 5

取締役 専務 押田眞克
Executive Vice President Masayoshi OSHIDA

現在、私共ジヤトコはCVTを経営戦略上最も重要な商品と位置付け、軽自動車用ベルトCVTから中型車用ベルトCVT、エクストロイドCVTまで、CVTのフルラインナップ体制を展開しております。このような中で2002年10月に、CVT戦略の第1歩としてSクラス車用ベルトCVT(150Nm)を日本市場に、Lクラス車用ベルトCVT(350Nm)を米国市場に投入し、商品力、品質等を市場に問うて来ました。1年余を経過した現在、JDPのIQSやAPEALの評価も良く、弊社のCVTを搭載した車輌の販売状況も好調であり、大変良い結果が得られております。

本号では、特集として「高トルク容量ベルトCVTを支える技術」が7件紹介されております。私自身もこのプロジェクトを生産技術担当として進めてまいりましたが、実現するにあたりこれまで経験したことも無いような数々の課題に直面しました。今回の論文のように、地道な研究によって解決が図られたものや、開発・生産・品質保証・サプライヤ果ては客先であるカーメーカーまで巻き込んで解決した課題等々、まさしくプロジェクトXで語られるような場面を数多く経験致しました。

このような活動を通して性能・品質面で及第点を取ったCVTも、コスト面ではまだまだ課題を残しております。CVT固有の部品のコスト高と、CVT化に伴う要求品質の厳しさが主な理由であると考えており、これらの課題をクリアし高い目標を達成するためには、製造技術、すなはち材料を含めた工法をブレークスルーしていくことが重要であると感じております。

At JATCO, we are working to provide a full product lineup of continuously variable transmissions (CVTs), which we have positioned as the most important products in our corporate business strategy. Our current lineup ranges from steel-belt CVTs for minicars to steel-belt CVTs for midsize cars and the Extroid CVT for large cars. As the first step in the deployment of this CVT strategy, we put a steel-belt CVT (150 N-m) for small cars on the Japanese market in October 2002 and a steel-belt CVT (350 N-m) for large cars on the U.S. market. We are letting the marketplace judge the attractiveness, quality and other attributes of our CVTs. In a little over a year since then, we have seen very satisfying results. Our products have received excellent evaluations in Initial Quality Studies (IQS) and Automotive Performance, Execution and Layout (APEAL) Studies conducted by J. D. Power and Associates, and cars fitted with our CVTs have been enjoying robust sales.

This issue of the *JATCO Technical Review* contains an seven-article special feature focusing on the "technologies supporting steel-belt CVTs with high torque capacity." I myself was involved in promoting this project as the person responsible for production engineering. In the course of developing these CVTs, we faced many challenging issues that we had never experienced previously. The articles in this special feature relate how these challenges were met through steady research efforts. They also describe how various issues were resolved through cooperative activities that involved not only our engineering, manufacturing and quality assurance departments, but also our suppliers and even the vehicle manufacturers that are our customers. We experienced any number of situations just like those talked about on the "Project X" television program in Japan.

While our CVTs have received high marks for their

一方、視野を外に向けてCVTを囲む環境を考えてみますと、市場の拡大、お客様の多様化、自動車技術の変革等、変化の著しい時代に突入しており、CVTそのものも、そのような変化に素早く対応せざるを得ないでしょう。

変化の激しい時代、先の読み難い時代であるからこそ、開発から立上げまでの期間短縮や、フレキシブルなモノ造りが必要とされてきており、CVTの造り方も、より一層の革新が必要であると感じております。

そのような革新の中にあっては技術者に負うところは多く、昨年合併を果たしたDMCも含めて、これまで培ってきた知識・知見をベースにさらなる飛躍をしなければなりません。その意味において、培ってきた知識・知見を形式化し、企業の財産として、技術の棚に残すことは重要なことであり、これまで全力で疾走してきたジヤトコの技術陣がこの時期にクリアした課題を論文として整理し、本号にまとめたことは大変意義深いことであると考えます。これを皆さんが熟読いただき、踏み台として更に大きな一歩を踏み出していただくことを期待します。

最後に、ジヤトコのCVT戦略が、我々ジヤトコ社員、サプライヤの方々、ユーザーであるメーカーの方々全員の努力と情熱で成功し、社会に貢献できることを切に願っております。

performance and quality as a result of those efforts, there are still many issues left to be resolved with regard to cost. The principal reasons for that include the high cost of the parts specific to CVTs and the rigorous quality requirements associated with the adoption of a CVT. In order to resolve these cost issues and attain our high goals, I feel it will be crucial to achieve breakthroughs in manufacturing technologies, that is, in engineering methods which also include materials.

If we shift our perspective to the external world and consider the environment surrounding CVTs, it is clear that we have plunged into a time of profound changes. For example, markets are expanding, customers' needs are diversifying, and dramatic innovations are occurring in automotive technologies. CVTs themselves must be capable of adapting quickly to these striking changes.

The very fact that this is a period of dramatic change and it is difficult to foresee the future makes it all the more important to shorten lead times from the start of development to the production launch and to have flexible design, engineering and manufacturing systems. I feel that increasing innovation is needed in the way CVTs are built.

Much of the work involved in achieving such innovations is shouldered by the engineers. We need to make further strides forward based on our accumulated knowledge and expertise, including that possessed by Diamondmatic Co., Ltd. (DMC), which merged with JATCO last year. In that sense, it is vital to give concrete form to our accumulated knowledge and expertise and to incorporate it in technologies that are passed on as our corporate assets. It is extremely significant, I think, that our engineering staff, who have been running at full speed all this time, have written articles for this issue, describing how various challenges were met in these changing times. I hope that everyone will read these articles thoroughly and that the information contained in them will serve as a springboard for taking further large steps forward in the future.

In closing, it is our fervent hope that we can successfully accomplish our CVT strategy, through the hard work and passion of everyone involved, including all of us at JATCO, our suppliers and the vehicle manufacturers that are our customers, and thereby contribute to the further benefit of society.

Historical Continuously Variable Transmission Developments: The past, the Present and the Future.

Professor Andrew A. Frank
The University of California-Davis

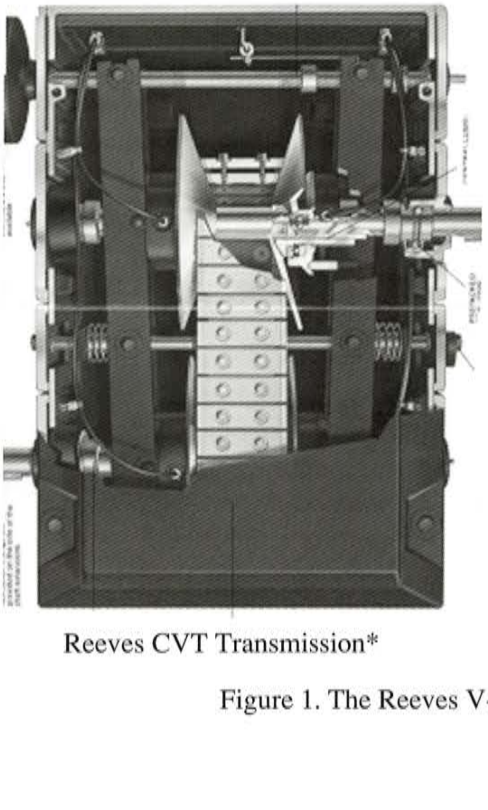
History and Background

The CVT began in the agricultural industry in the 1850's. These devices were generally very low power and relatively inefficient and required continuous maintenance, but they provided a useful function.

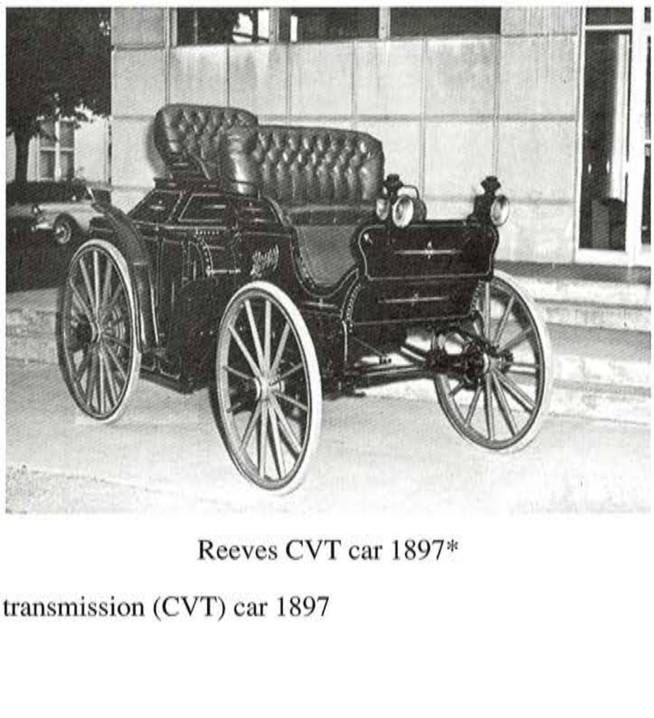
One of the more reliable CVT cars was the one designed by the Reeves Motor Company shown in Figure 1. Its' transmission was made with wooden blocks bolted onto a leather and later rubber composite belt.

As the automotive industry began to emerge at the turn of the last Century the early automobiles with low power (5 to 10 hp) used CVT's for their transmission to match the vehicle speed with the early gasoline engines which could only run in a very limited speed range. These transmissions worked but were not very durable due to the materials used for their construction. The materials were generally steel, wood, rubber and leather. They were unreliable and a source of low durability. As engines began to become better and more reliable these transmissions were no longer acceptable.

The invention and adoption of the multi-speed geared transmission and the refinement of the multi-cylinder gasoline engine reduced the need to have an infinite ratio control for drivability. In fact, transmissions have increased their numbers of gears and span of ratio over the years for both cars and trucks, and engines have increased the number of cylinders from one or two in the early days to as many as 16 by the 1930's. There is an inverse trade off between the number gears in the transmission and the number of cylinders in the engine for smooth driveability.



Reeves CVT Transmission*



Reeves CVT car 1897*

Figure 1. The Reeves V-belt-chain transmission (CVT) car 1897

In the search for an automatic transmission in the late 1920's and early 1930's, many automobile companies and inventors began to search the world for variable speed concepts that could be adapted to the automobiles and trucks of the day since the early CVT's were of such low rated power, torque and durability. The reason was motivated by the need for better fuel economy while providing an automatic drive system. This research in CVT's was slowed by the 2nd World War. At the end of World War II, there was an abundance of fuel in the USA so some of the reasons for an efficient transmission was no longer required. Thus the focus was on drivability and smoothness and not efficiency and emissions.

The torque converter was invented to make the discrete transmission appear as a continuously variable transmission system. The CVT's developed at the time were compared with the multi-speed gear box with torque converters and were found to be heavier, more complex to control, more expensive to manufacture, provide less isolation of the powertrain to the road, and required special fluids to operate but provided better fuel efficiency. The disadvantages made the CVT less competitive in the 1950's.

By the 1980's to the 1990's and now in 2004 the requirement for higher fuel economy and lower emissions and better performance has caused the transmission industry to increase the complexity of the multi-speed transmissions by adding more discrete ratios, increasing the ratio span and providing closer ratio spacing. These transmissions today have as many as 6 to 8 speeds with as many clutch packs and of course, more weight and control complexity. As a result, today the CVT can now be competitive to many Automatic Multi-speed transmissions. This is true in all the transmission performance parameters except torque in most cases in the automotive industry. There are new concepts that are addressing the torque and power issues.

Some of the reasons the CVT was not competitive in the past with Multi-speed transmissions were, complexity of control, weight, size, durability and cost. Today, many of these reasons no longer hold. Especially with the advent of modern electronics, control theory and new mechanical concepts, the CVT is now competitive and as volume builds will provide lower manufactured cost, lighter weight and higher performance than equivalent geared transmissions. Many CVT vehicles have shown fuel economies better than equivalent automatic transmissions vehicles but less than manual transmissions. The theory, however, says that the fuel economy should be better than both conventional manual and automatic transmissions if the control system and power transmission efficiency were equal and the vehicle drivability can be as good or better.

It is up to researchers and development engineers to create the new technology to make the CVT systems competitive and better than the contemporary multi-speed transmissions both manual and automatic. The target areas for improvement are:

1. Weight.
2. Cost.
3. Driveability as good or better than conventional transmissions.
4. Volume or size, similar or smaller.
5. Configuration similar or identical to existing transmissions for simple vehicle integration.
6. Lower complexity i.e. fewer parts.
7. Better durability than the automatic transmission.
8. More flexible control system.
9. No cooling requirements due to much higher efficiency
10. Much quieter and smoother in operation.

All these areas must be equal or better than conventional transmissions or why should any manufacturer consider a change in their manufacturing infrastructure?

This paper will explore some concepts that can make current CVT's compete favorably in these areas.

Control of CVT and IVT Concepts and Driveability

There are two important levels of control for the CVT. The first level is the clamping pressure and CVT system mechanics to control the traction system. The second is the energy management required for control and tuning of the CVT for driveability and low fuel consumption and emissions.

These two levels of control must interact with each other but can be considered separately if the systems are properly designed. The reason for separation is to simplify analysis and design and make conceptualization easier. The reason for the complexity is that the CVT is unlike the conventional Multispeed transmission in behavior. In addition the CVT is under continuous power transmission as shifts occur, where as the conventional transmission generally interrupts power in some way.

Control of traction elements in CVT's

Traction in CVT's is necessary to provide drive torque. The Coefficient of traction of rolling elements is on the order of 0.05 to 0.1 with steel on steel and a little higher for rubber on steel. This means that it takes 10 to 20 times more clamping force than force transmitted. To accomplish this, many current designs use loading mechanisms that are mechanical ramps or screws or hydraulic hydrostatic pressure.

These control mechanisms can consume enormous amounts of energy if not carefully designed. The advent of modern electronics, simple electric motors and low cost hydraulic pumps allow the design of a low energy controller. The following figure is a system that shows energy consumption less than 10% of conventional hydraulic pump-valve constant pressure controllers and eliminates all close tolerance components. The concept is known as the "Servo motor hydraulic pressure control system". In addition, since it is electronically controlled it can easily be combined with a higher level engine or powertrain controller. This is a great advantage in high level control. The advantage is even more emphasized with the parallel hybrid electric powertrain system because the electric traction motor can be used to aid in driveability.

The advantage of this system is the low power required to maintain high clamping pressure, and the fidelity of control. This is due to high power-high frequency pulse width modulation electronics. This kind of system reduces the need for calibration and development since there is no valve body or close tolerance moving parts except the pumps. But these gear pumps are extremely simple and the pressure transducer provides the proper information for control under all conditions. This means that a properly designed system will require no calibration except the pressure transducers which can be calibrated in the laboratory. The computer behind the servo amplifiers makes the system very robust.

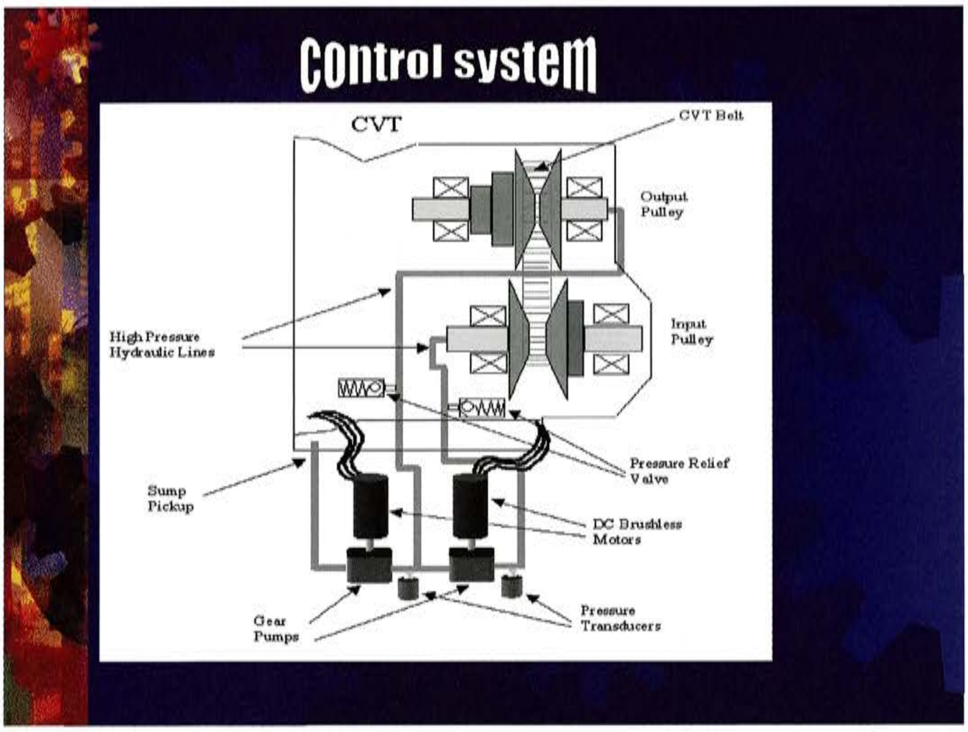


Figure 2. Servo hydraulic control system

Control of fuel efficiency and driveability

A second level of control is the integration of the CVT with the engine or power source. To understand this second level more clearly the dynamic equations governing the action of the CVT is discussed below.

The CVT and IVT dynamic control for a vehicle is governed by the following model and equation of motion.

The consequence of equations 1 and 2 and the model is that the acceleration of the output shaft or the car is dependent on two independent system inputs. These inputs are:

1. The torque from a power source. Which may come from the control of the I/C engine throttle or an electric motor or both.
2. The rate of change of ratio.

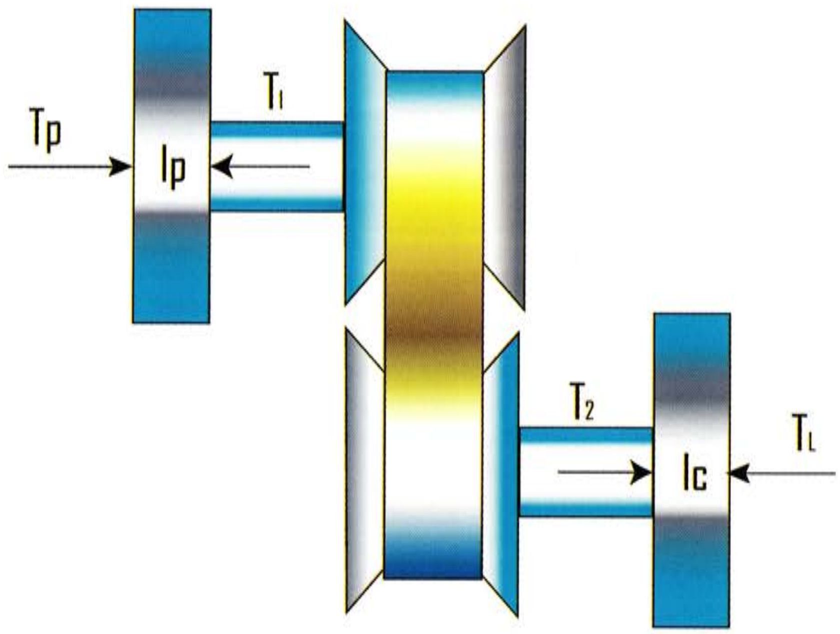


Figure 3. CVT dynamic model

$$\dot{w}_C = \frac{r \cdot T_P - \dot{r} \cdot w_P \cdot I_P - T_L}{I_C + I_P \cdot r^2} \qquad \text{Equation 1}$$

Where $r = \frac{\omega_{input}}{\omega_{output}}$ = ratio of the CVT Equation 2

R_{dot}=rate of change of ratio
T_P = power source torque input
T_L = load and loss torques
W_c=output speed
W_P=input speed
I_c = output inertia
I_p = input inertia

These two inputs can control the output torque equally at every instant in time. In fact the rate of change of ratio can provide a torque magnitude much bigger than the power source since it's origin is from the inertias of the load and the input momentum at a specific instant. This term can easily break the shafts or destroy the CVT.

The existence of the r_{dot} term can be viewed as a "friend" or a "foe". If it is chosen as a foe then it must be counteracted by an additional torque since it is generally in opposition to a desired positive acceleration of the output shaft or drive shaft. If it is treated as a friend then it can be used to aid in the control of driveshaft torque or vehicle acceleration. Thus it could and should be controlled explicitly and independently.

In either case the ratio rate term must be accounted for in order to control vehicle acceleration or deceleration characteristics as we are currently used to in conventional transmission vehicles.

Controlling ratio rate is dependent on the geometry of the CVT or IVT i.e. the ratio and the inertias of the engine and the vehicle. Thus, torque from the power source must be coordinated with the rate of change of ratio. The best way to handle this is to have two independent sources of torque. This could happen if the system is a parallel hybrid with both an internal combustion engine and an electric motor at the transmission input shaft. Then if there is enough torque available from the electric motor to compensate for the negative effect of the desired ratio rate as the transmission is shifted then the vehicle can accelerate smoothly with good acceleration response.

A parallel hybrid electric powertrain is ideal for this purpose. This kind of powertrain shown in Figure 4 below which provides the ideal engine-CVT control for both fuel efficiency and driveability.

The electric motor, of course, must be supplied energy to provide the shift characteristics. This energy must come from a separate source such as a battery. The energy from the battery must be replenished at some time in the future. These hybrids can return the energy used to compensate for CVT shifting when the ICE is asked to operate at very low torques and consequently low efficiencies by using the electric motor as a generator while the vehicle is in light load or deceleration or braking. By declutching the gasoline or Diesel engine, the system becomes a much more efficient electric motor powertrain.

The overall average efficiency of the CVT hybrid can have a very high average efficiency compared to the conventional ICE discrete transmission drive train providing less than 1/2 the driving cycle fuel consumption or more than double the fuel economy.

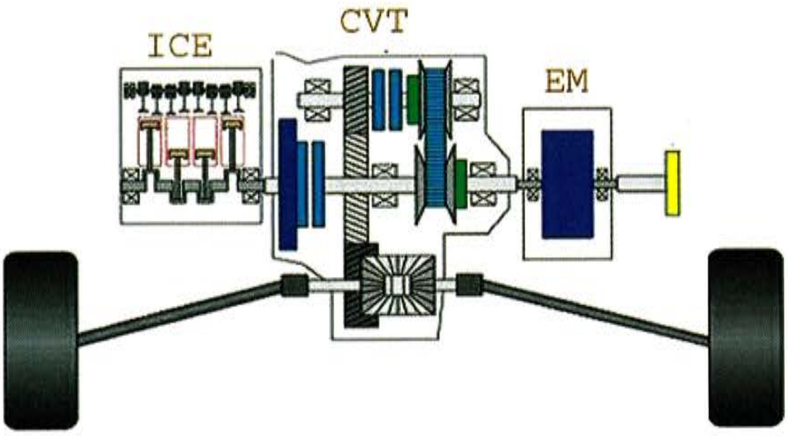


Figure 4. A parallel CVT hybrid electric motor powertrain with ideal shifting characteristics

CVT concepts and currently manufactured CVT's for automobiles

Some currently manufactured cars with CVT's are shown in the following figure.
These cars are shown thanks to Bosch-VDT. The other notable manufacture of CVT cars is Audi with a V-Chain CVT by LUK. There are a number of vehicle CVT's currently on the market and many new CVT concepts are in development.

The notable CVT's currently in high volume production use the Bosch-Van Doorne V-belt concept, The Luk V-Chain concept, and the Toyota Electric power-split CVT as in the Prius and the new Ford Escape Hybrid.

Some of these cars are shown in the Bosch VDT brochure below.

In the industrial market it is the hydrostatic CVT systems, and other low power concepts such as the Reeves variator* and a host of other types of variators such as ball disc systems the Bier drive system etc. None of these concepts are suitable for power transmissions in automobiles or trucks of the modern day.

A CVT not often considered is the electric power split system of the Toyota Prius. This electric generator and motor combination with a differential gear set is every way a CVT as any system and therefore must obey the dynamic equation above. The reason for the powersplit system is to increase the efficiency of the transmission from a pure electric generator-electric motor CVT as in Diesel locomotives. The other reason is that an electric motor vehicle launch system eliminates the need for a torque converter and provides low speed pure electric vehicle operation

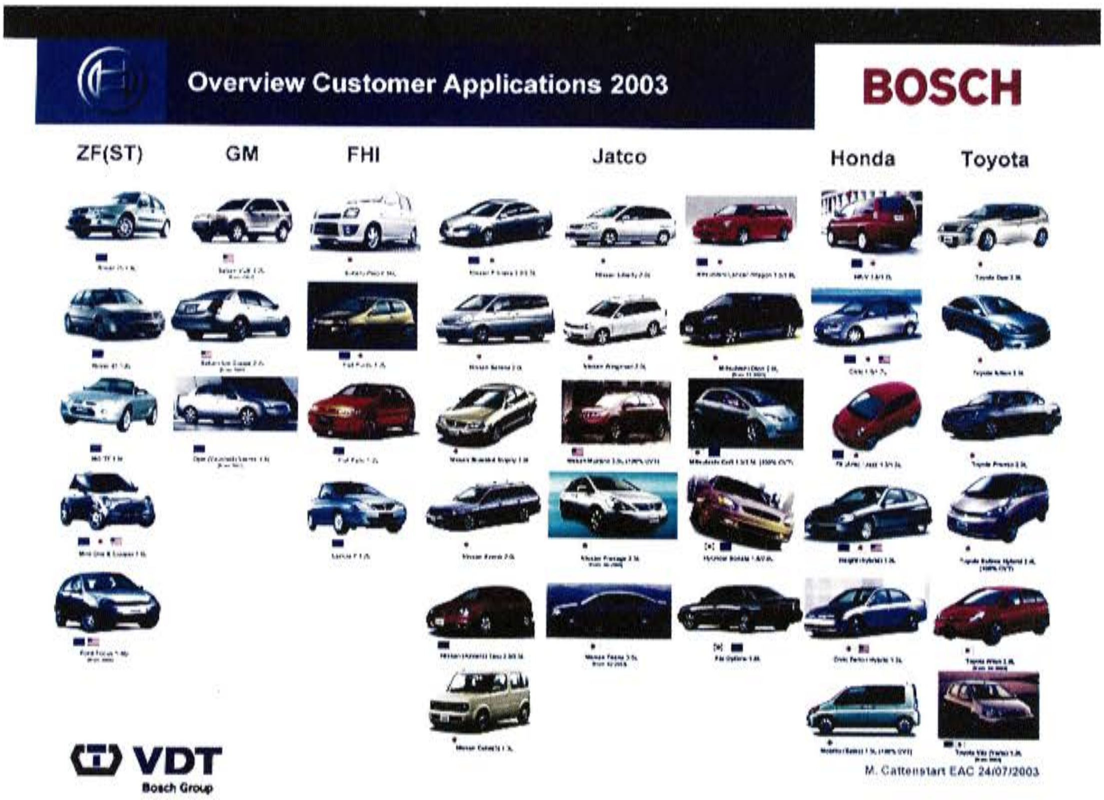
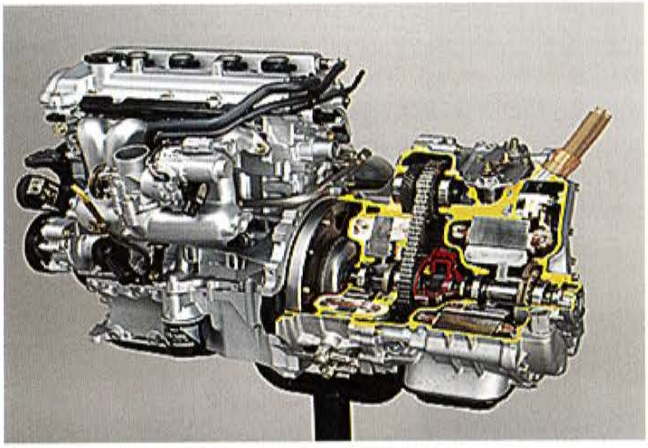
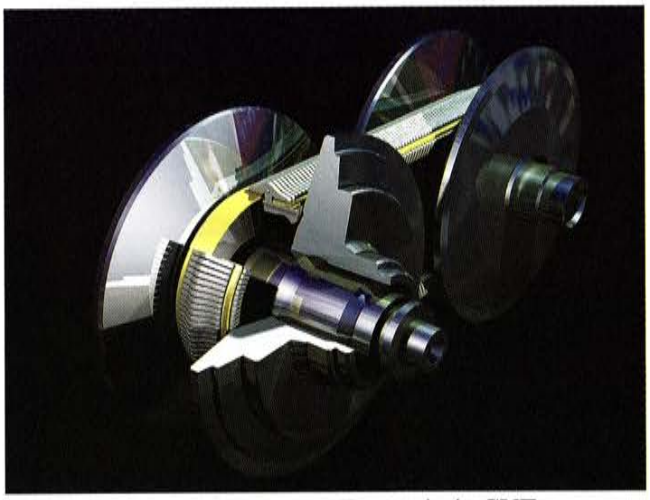


Figure 5. Cars and light duty vehicles with the Bosch-VDT CVT

This concept will appear soon in the Ford Escape, the Toyota Highlander and many other vehicles. There are other much simpler concepts that can accomplish the same function as achieved by this powertrain today such as the configuration shown above in Figure 4 above. At the time Prius concept was in development in 1980, there was no alternative mechanical solutions as there are today. The alternative available now is the high torque CVT's using chain technology. The Prius system and the Bosch- VDT CVT are shown in the following figure.



Toyota Electric CVT for the Prius



Bosch-Van Doorne Transmissie CVT

Figure 6. Toyota Prius Electric CVT ** and the Bosch-VDT CVT

Pure hydrostatic CVT's have been in production for many years but have never made it into the automotive market because of high noise and low efficiency. These have been and are currently used for industrial and military machines and in applications where noise and efficiency are not as important as in the current automotive market. These systems have proven to be very durable and reliable but produce very high noise level and are low in efficiency

The theory of the Van Doorne V-belt, the Luk V-chain are quite different and the efficiency, noise characteristics and control of these transmission systems differ. They offer the best efficiency, lowest noise and most durable of all CVT's to date.

A concept, that has been under research, for many years is the concept of the Toroidal CVT. The concept has two variations, a full Toroid and the 1/2 Toroid. The following figure shows the 1/2 Toroid system.

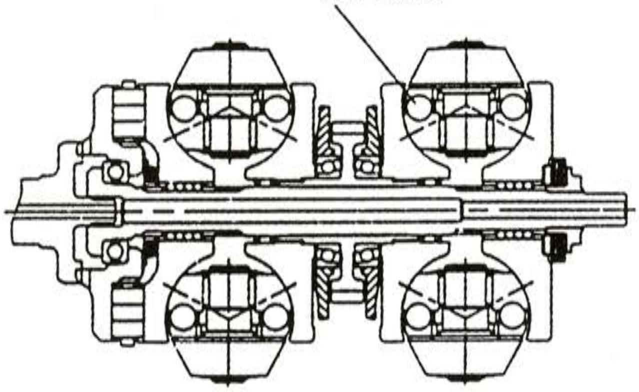


Figure 7. 1/2 Toroid CVT (Kraus)

These concepts have been researched for many decades but they suffered from durability, control, weight, complexity, efficiency and size issues. They represent some of the oldest researched concepts in the automotive market. These concepts are being improved and may eventually solve these deficits to be competitive with the V-belt or V-chain systems.

There are other concepts that are even more efficient and can provide yet simpler to manufacture, higher power, durability, reliability, and better performance at a much lower cost. One of these is the V chain concept of Gear Chain Industries, GCI***, of The Netherlands. GCI is currently researching concepts that can be implemented to over 1000 kw. This concept is showing that the efficiency can be higher, the noise lower, the durability higher, the control system simpler, and be designed with far fewer parts than any other transmission concept. This promising concept may be able to replace other concepts because it operates on a different principal. The principal is similar to the concept of gears but it is a CVT. The technology will be licensed to major manufacturers in the near future because it shows great promise to replace other current CVT concepts. The following figures compare the GCI chain and the Bosch-VDT belt for the same transmission.

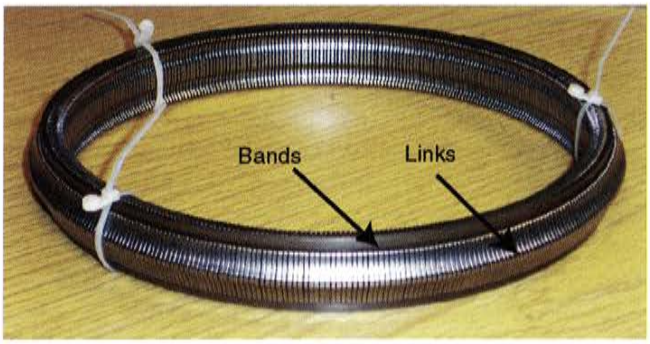


Figure 8. Bosch-Van Doorne CVT belt

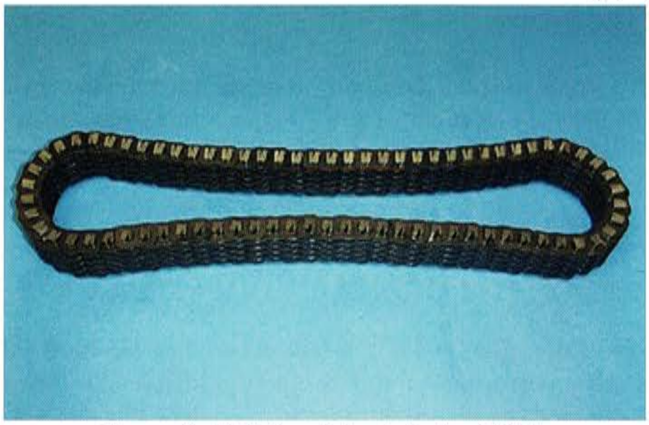


Figure 9. GCI Involute chain for CVT's

The Attractiveness of CVT's

The advantage of the CVT is potentially a very low mechanical part count and a much simpler control system. However, practical control is more complex because it requires coordination of the transmission with the engine since CVT efficiency and durability are dependent on the torque and speed being transmitted which determines the pressure between the Variable elements. In addition, the CVT must be protected against the sudden inertial shocks that occur as the vehicle is operated on real roadways and highways by the general public. Some manufacturers include torque limiting clutches to address these sudden large high frequency torques.

One of the disadvantages of the V belt or chain CVT in the past is that the input and output shafts are often not collinear. There are a number of ways to solve this problem that involve gear sets and additional fixed ratio chains etc. A simple In-Line CVT concept is required to make the CVT adapt to conventional drive vehicles. The requirement would be to design a CVT system to be a direct (bolt to bolt) replacement for an equivalent manual transmission and clutch system with about the same overall size constraint, at of course at a lower cost and lighter weight. This concept is being researched by the University of California Davis. A preliminary sketch of the transmission is shown in the following figure.

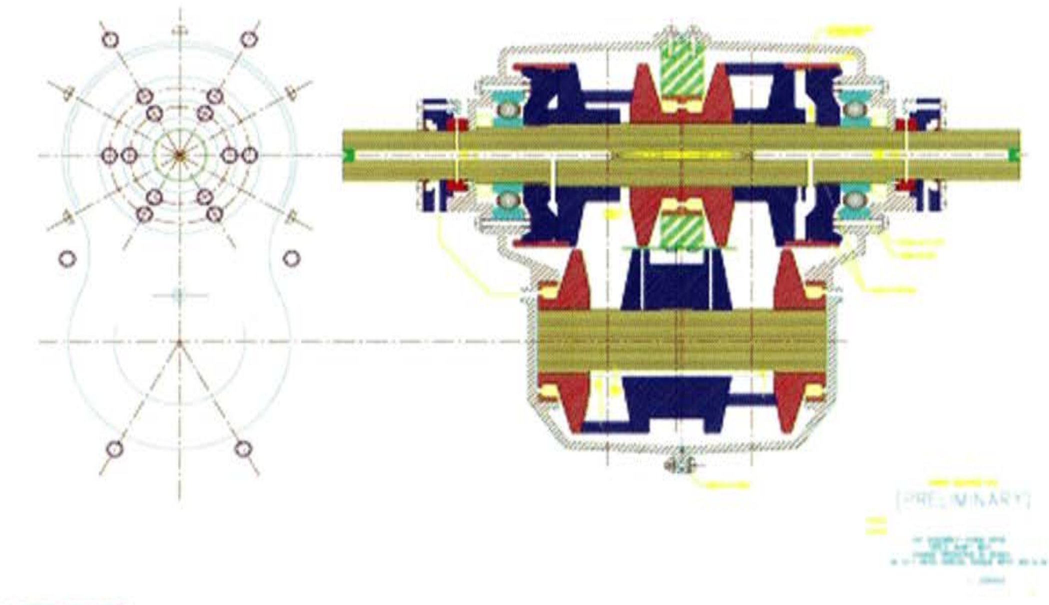


Figure 10. A new inline CVT designed by UCDavis for 500 NM and 200 kw

The Future

To summarize the development in the last 100 plus years in the development of CVT's, there have been many inventions and developments that have made it now possible for successful CVT systems, including the understanding of the dynamics of CVT-engine control systems as well as many new materials and intelligent controls.

The future for CVT's is bright because of the need for higher performance and better fuel efficiency, lower costs and broader applications. It is clear that the new concepts of the Hybrid Electric Vehicle, Hybrid Flywheel Vehicles, and Hybrid Hydraulic Vehicles all can use a CVT of some sort. The auto industry continues to increase the number of gears of the discrete transmission in the search for lower fuel consumption and smoother driveability. The CVT's of the past had been more expensive and less durable than discrete gear transmissions, but new technology now exists for CVT's with much higher efficiency and much lower costs and far fewer parts. These CVT concepts are beginning to compete with conventional multi-speed transmission concepts.

The concepts that will win the race will be the concepts that are the most flexible and efficient that can be adopted effectively to all powertrain configurations in vehicles with conventional transmissions and the evolving hybrid vehicle concepts. For example, exchanging a conventional manual or automatic transmission in a Sports Utility Vehicle directly with no mechanical mounting modification to a vehicle with a CVT or a hybrid with an electric motor/generator and CVT. This kind of flexibility will make the CVT of the future a compelling choice and the customer will have full flexibility of choice with no change in the OEM assembly plant. The transmission concept that will allow this flexibility will begin to capture market share of transmissions if the cost is equal or lower. This now appears to be possible with many new concepts as discussed.

Some notable systems in production today are the Honda Hybrid CVT's and the Toyota Prius Hybrid Electric CVT. Hybrid systems work much better with a CVT than with discrete ratio transmission systems because the engine and electric motor/generator and battery system can be much better optimized for efficiency and emissions when the ratios are infinitely variable.

Drivability issues are also much easier to solve and implement. Finally the CVT has the ability to reduce mechanical parts count leading to much longer life, higher efficiency, and lower costs. We at University of California-Davis have constructed hybrid electric- CVT vehicles with total powertrain mechanical parts count that is less than 20% of the conventional car with no sacrifice in performance. Of course these hybrids with the CVT have more electronic controls than the conventional car. Thus, we traded off mechanical complexity in conventional drivetrains for electronic complexity in Hybrid Electric CVT Drivetrains. This trade-off is now possible because it results in lower costs and higher reliability.

Some of the advantages that can be gained by the hybrid electric CVT are:

1. No reverse gear system is required.
2. No vehicle starting device such as a torque converter or clutch is required.
3. No need for hydraulic transmission system calibration since it can be accomplished electronically with sensors to provide information on temperature, age of the oil, and other parameters requiring calibration. .
4. The fewest number of mechanical parts in the powertrain and transmission system.
5. All accessory loads on the engine are removed to increase the net thermal efficiency of the engine. This can be done, since there is a large amount of battery energy.

Examples of this concept are shown in the vehicles constructed below. These vehicles all have fuel economy that is at least double the conventional vehicle but they do not weigh much more than the conventional car and they can go 100 km all electrically before the engine needs to come "on" to maintain the batteries at about 20% State of Charge.

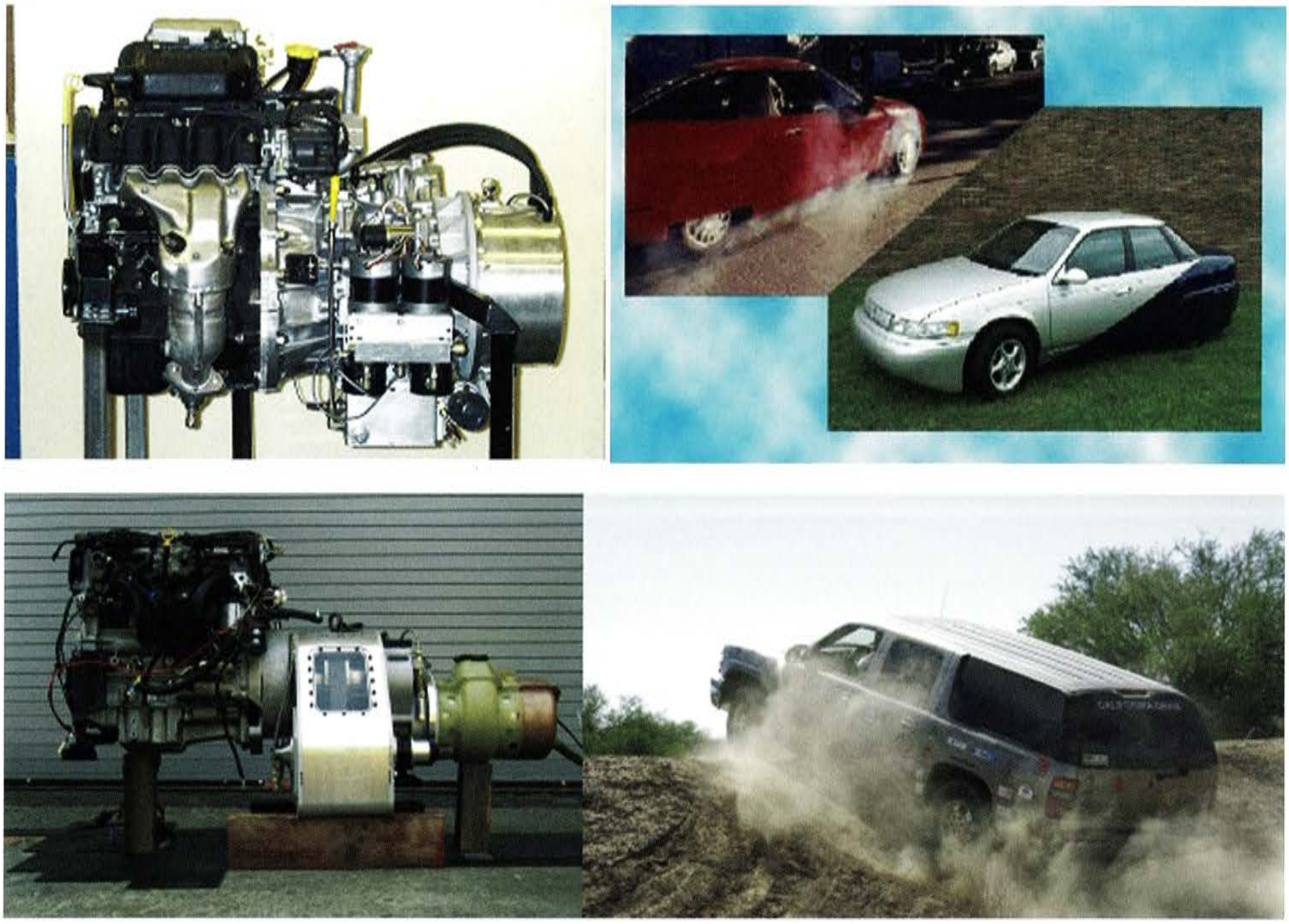


Figure 11. CVT hybrid electric powertrains for 100 kw 250 NM 3 liter vehicles and a 250 kw 6 liter 700 NM vehicle with double the fuel efficiency of the conventional vehicles constructed at the University of California at Davis Hybrid Electric Vehicle Center.

Conclusions

The CVT concepts for the future needs of the automobile and truck market needs to be designed in such a way that it is adaptable to all manner of powertrain concepts, from manual transmission vehicles, conventional torque converter automatic transmissions to hybrid electric powertrains for both front engine, front wheel drive vehicles to front engine rear wheel drive and all wheel drive vehicles . In addition, the CVT concept should be adaptable to other vehicular functions such as 4 wheel drive transfer boxes and differentials.

There are many more applications for CVT's in vehicles that have not been discussed. Some of these ideas are for example, constant speed drives for air conditioning and accessories, CVT's for differential drives for four wheel drive systems, etc.

Perhaps the biggest challenge will be large trucks and bus main transmissions where power and torque are high with steady high loads and durability is a stringent requirement. The challenges for this industry segment is far more difficult than the light duty vehicles in passenger automobiles. Some of the requirements include a different high level control system. For example, the engine is often used for braking down hill and the transmission must be able to transmit torque with absolute certainty. Failure under these conditions is not acceptable. Thus the control systems and the transmission elements must be very robust under all operating conditions and fail safe systems under all possible component failures must be designed.

Finally, the CVT can be the lowest cost, most efficient, the smoothest, quietest, and the most durable transmissions of the near future. Every vehicle, including electric vehicles can benefit from the CVT. Thus, future advance powertrains such as electric drive system with power supplied by hybrids and fuel cells will be made more efficient and more controllable by the CVT.

The final question will be, what form will the future take? It likely that many CVT and IVT forms will occupy the CVT space.

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** Takehisa Yaegashi Toyota Motor Co. SAE International Congress. Toyota presentation March 4, 2003

*** Jacques Van Rooij, Gear Chain Industry, the Netherlands.

特　集

高トルク容量ベルトＣＶＴの開発

Development of a New Belt CVT with High Torque Capacity

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抄　録　近年，地球環境の改善要求が増大し，車の燃費向上が全世界的な課題となっている．一方，自動変速機の需要も世界的に高まってきており，その中で燃費向上を達成するために普及してきたのがCVTである．従来，CVT車の排気量は，1.0 Lから2.5Lクラスが主であったが，ジヤトコが世界に先駆けて3.5LクラスのCVT(以下CVT3，Fig. 1)の開発に成功した．本稿では，CVT3の主要仕様，およびその制御，性能について紹介する．

Summary　Improving the fuel economy of vehicles has become an issue of worldwide concern in recent years in connection with the growing awareness of global environmental issues. On the other hand, global demand for automatic transmissions has also been increasing. Against this backdrop, the application of continuously variable transmissions (CVTs) has been spreading at a rapid pace in recent years as a means of attaining improved fuel economy. Until recently, CVTs have mainly been used on 1.0-2.5-liter class vehicles. At JATCO, we have succeeded in developing the world's first CVT (CVT3) capable of being used on 3.5-liter class vehicles (Figure 1). This paper describes the main specifications, construction, hydraulic circuit and control system of this new-generation CVT.

1. はじめに

このCVT3は，2002年11月より北米で発売開始した日産ムラーノ(Fig. 2)に搭載され，国内でもティアナ・プレサージュに搭載し，好評を得ている．

その開発の狙いは，以下の4項目である．

(1) コンパクト化
(2) 軽量化
(3) ワイドレシオ化や油圧制御などによる燃費向上
(4) 加速性能の向上

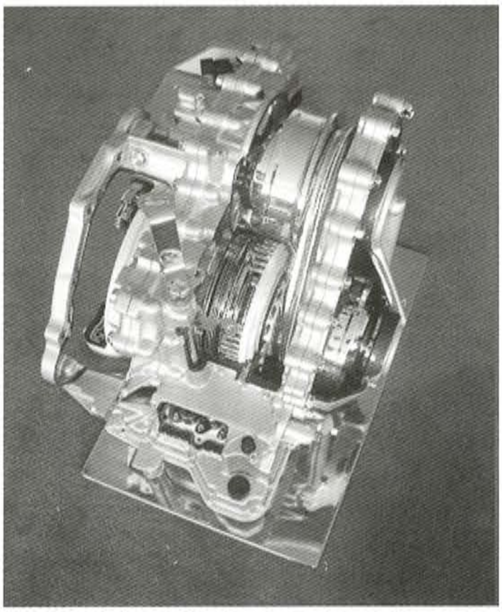

Fig. 1 CVT3

1. Introduction

The CVT3 is used on the Nissan Murano (Fig. 2) that was rolled out in the North American market in November 2002 as well as on the Nissan Teana and the Presage in the Japanese market. It has been popularly received by customers for its excellent performance. Four objectives were set for the development of this CVT:

(1) More compact size
(2) Lighter weight
(3) Enhanced fuel economy due to a wider ratio range, improved pressure control and other improvements
(4) Improved acceleration performance

Fig. 2 Nissan　Murano

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2. 主要諸元

CVT3の主要仕様について，当社製Hyper CVTと比較してTable 1に，またCVT3の主断面をFig. 3に示す．プーリ軸間の拡大とプーリ押力の増加により，CVT3のトルク容量を，Hyper CVTの250Nm(2.5L)に対して350Nm(3.5L)に増大した．さらに，燃費を向上するため，プーリ変速比幅を拡大するとともに，プーリ各部品の強度，ベルト単体強度などを増大した．

Hyper CVTの開発経験に基づいて，プーリ変速比幅の拡大，ユニットのコンパクト化，高効率化に注目し，バランスよく最適化した．

Table 1 Major specifications of CVT3 and Hyper CVT

Belt CVT		CVT3	Hyper CVT
Engine	Type	3.5L V6	2.5L L4
	Max torque (Nm/rpm)	350/4400	250/4400
	Position of starter motor	CVT side Engine side	Engine side
CVT Ratio	Ratio range	5.401	5.359
	Low～OD	2.371～0.439	2.326～0.434
	Final gear ratio	4.878～6.327	5.473, 5.743
	Manual-shift mode	6-speed	6-speed
CVT Dimensions	Overall length (mm)	386.9	396.0
	Distance between pulley shafts (mm)	178	168
Maximum pulley pressure (MPa)		5.7	4.6
Oil pump discharge capacity (cm^3/rev)		22.4	19.6

トルク容量が増大しているにも拘わらず，CVTの変速比幅を，Hyper CVTの5.359から5.401に増大した．また，各部品のコンパクト化やレイアウトの改善により，Hyper CVTに対して全長を約9.1mm短縮した．

3.駆動フィーリングと燃費との両立

4ATに対して駆動フィーリングと燃費とを同時に改善するため，ユニットのハードとしては，簡素化，軽量化，変速比幅の拡大の3つを実現するとともに，制御としては，変速スケジュールの最適化はもちろんのこと，運転条件に合わせて変速スケジュールや変速モードを自動選定するとともに，登降坂制御(自動エンジンブレーキを含む)やマニュアルシフトの採用など数々の新機軸を採用した．

(1)変速スケジュールの最適化

CVT3は，変速比幅が4ATに比べて大きいことから，高車速側でもエンジン回転を低く保ちながら走行でき，燃費が向上する．参考に変速線をFig. 4に示す．回転が低いと，アクセルペダルを踏込んだ瞬間の初期加速レスポンスが悪化しがちとなる

2. Main Specifications

The main specifications of the belt CVT3 with high torque capacity are compared with those of the Hyper CVT in Table 1. A main cross-sectional view of the CVT3 is shown in Figure 2. The torque capacity of the CVT3 has been increased to 350 Nm, compared with 250 Nm for the Hyper CVT, primarily by increasing the distance between the pulley shafts and the pulley pushing force. The ratio range of the CVT3 has also been expanded to reduce fuel consumption further. The strength of the pulley components and that of the belt have been increased.

In addition, the CVT3 design strikes a good balance based on previous experience acquired in developing the Hyper CVT among the measures for expanding the ratio range, its compact construction and higher efficiency.

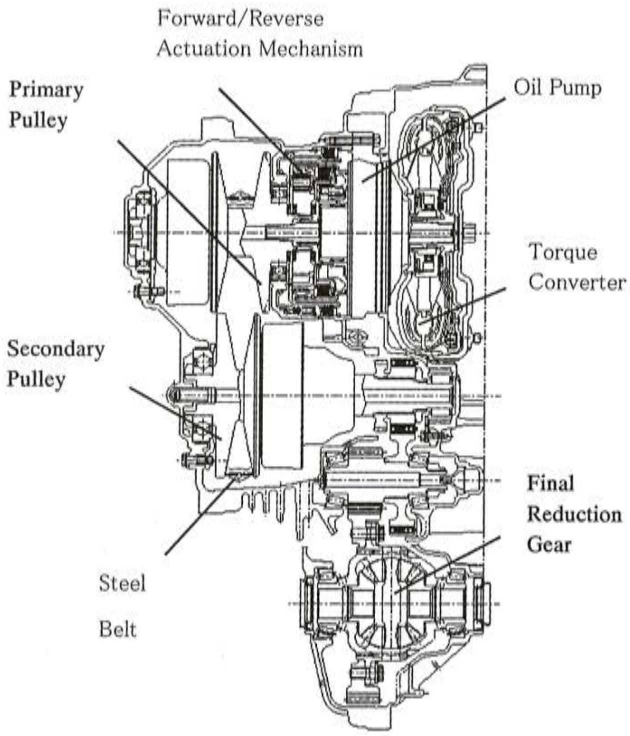

Fig. 3 Main cross-sectional view

The ratio range achieved by the belt and pulleys of the CVT3 has been expanded to 5.401 compared with 5.359 for the Hyper CVT and it has a wider ratio range as a CVT for 3.5-liter engines than CVTs for 2.5-liter engines. The CVT3 is 9.1 mm shorter in overall length than the Hyper CVT as a result of adopting more compact components and improving their layout.

が，CVT3の場合は，350Nmというエンジントルクに助けられて，平地では動力性能の低下を気にすることなくエンジンを低回転に維持することができた．

これらを含め，北米Combineモードにおける燃費向上効果は，4AT対比で約16%，5AT対比で約8%である．(Fig. 5)

(2)運転条件に自動適合する変速スケジュール

上記で設定した最適変速スケジュールに加えて，運転条件(ドライバーの意図)や走行条件(道路勾配など)に応じて変速スケジュールを自動選択することにより，あらゆる条件で動力性能を満足しつつ燃費を改善させることができた．

特に，登坂時には大きなパワー余裕を必要とするので，アクセルペダルから足を離してもエンジンを高回転に保つよう，変速比をLOWギア比側に維持するとともに，加速の際にも，同様に平坦路よりLOWギア比側にシフトさせ，より強力な加速が得られるようにした．これによって，登坂時の駆動フィーリングを格段に向上させることができた．

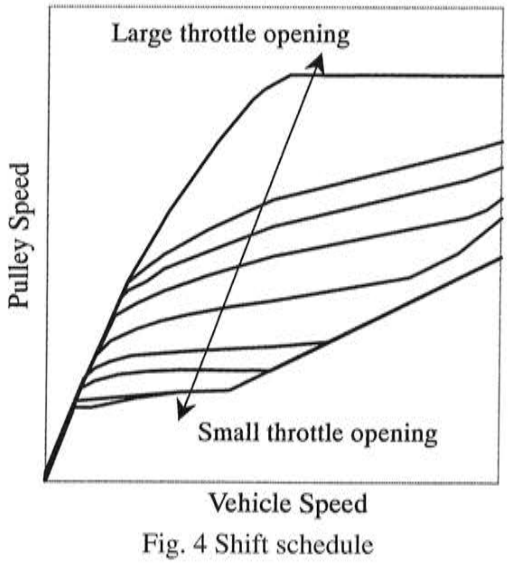

Fig. 4 Shift schedule

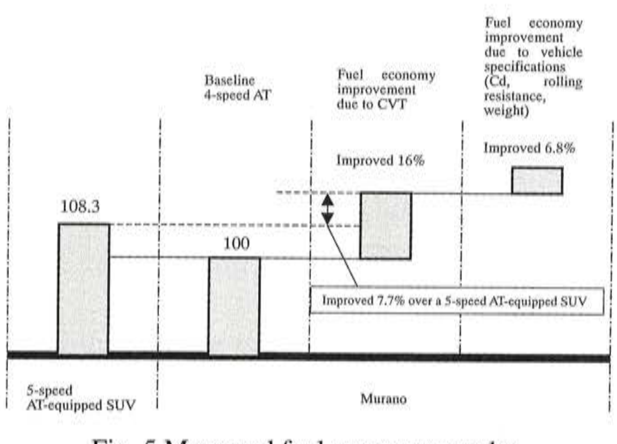

Fig. 5 Measured fuel economy results

3. Improvement of Both Driveability Feeling and Fuel Economy

Simultaneous improvement of the driveability feeling and fuel economy was addressed both in terms of the hardware and software. With regard to the hardware, steps were taken to simplify the construction, lighten the weight and widen the gear ratio range so as to achieve the three major development aims. As for the software, many new controls were adopted in addition to optimizing the shift schedule. A control was added for automatically selecting the shift schedule and shift mode to match the driving conditions. A hill-climbing control, including automatic engine braking, was also adopted along with a manual shift mode.

(1) Optimization of shift schedule

Because the CVT3 was designed from the outset with a wider gear ratio range than that of conventional 4-speed ATs, it allows the engine speed to be kept low while the vehicle is cruising, which works to improve fuel economy. The shift lines are shown in Fig. 4 for reference. With conventional CVTs, the initial acceleration response at the moment the driver depresses the accelerator pedal tends to deteriorate at low engine speeds. In contrast, the CVT3 allows the engine speed to be kept low when cruising on level terrain without any concern about a decline in power performance, thanks to its capacity to handle engine torque of 350 N-m.

As a result of the foregoing improvements, the CVT3 has the effect of improving fuel economy under the U.S. combined city/highway driving cycle by approximately 16% over a 4-speed AT and by approximately 8% over a 5-speed AT (Fig. 5).

(2) Shift schedule that adapts automatically to various conditions

In addition to the adoption of an optimized shift schedule, the CVT3 also automatically selects the right shift schedule matching the operating conditions (the driver's intentions) and the driving conditions (road grade, etc.). This adaptability improves fuel economy while still providing the power performance required under all sorts of conditions.

The feeling of driveability has been markedly improved especially when climbing hills. This is accomplished, for example, by maintaining a low transmission ratio during hill-climbing, when a large margin of reserve power is needed, so as to keep the engine speed high even if the driver releases the accelerator pedal. In addition, when the vehicle is accelerated, the CVT3 also shifts further to the low ratio side than on a level road so as to deliver more powerful acceleration.

4. 大排気量車の駆動フィーリングの向上

CVTは，軽自動車の非力なエンジン用のユニットとしてスタートしたことから，エンジン回転は高めにチューニングされてきた．

エンジン回転を高めに設定すると，以下の課題が顕在化する．

(1)エンジン音が大きく，騒々しい

(2)加速時のリズム感が乏しい

(3)燃費が悪化する

これらの課題に対して，大排気量エンジンに相応しい変速制御に改善することにより，CVT搭載車のこれまでのイメージを払拭することができた．すなわち，静粛性を確保するためにエンジン回転を低目に維持しても，駆動力余裕が十分にあるので，いざとなれば力強い走りが得られる．また，高速でアクセルペダル開度をほぼ一定に維持して走行している場合には，変速を感じさせないような穏やかな変速が可能となる．

また，従来の低排気量車では，アクセルペダル踏込み直後は専らエンジン回転の上昇にトルクを消費するため車両は殆ど加速せず，回転上昇後に加速するという加速遅れを感じる制御であったが，大排気量車では低速でもエンジントルクが大きいので，エンジン回転を急いで上昇させる必要が無く，アクセルペダル踏み込みの初期から充分な加速度が発生する．そのため，エンジン騒音が減少し高級感を維持できるとともに，車速上昇とエンジン回転上昇とをリニアに連動させることができる．これにより，加速時に変速比を大きく変えて駆動力を発生させようとする変速依存型から，変速比を小幅に抑制する仕様に改良できた．(Fig. 6)

4. Improved Driveability Feeling in Large-Engine Vehicles

Continuously variable transmissions began as units for use on minicars, though their torque capacity has steadily been improved in the intervening years. They were originally teamed with low-power engines, with the engine speed tuned on the high side. Setting a somewhat high engine speed, however, gives rise to the following issues:

(1) Large engine noise

(2) Lack of a feeling of rhythm when accelerating

(3) Deterioration of fuel economy

These issues have been resolved by improving the shift control of CVTs to make them suitable for use on vehicles with large-displacement engines, which has completely changed the previous image of CVTs. In short, because there is ample power in reserve even if the engine speed is kept low during cruising, quiet operation is achieved along with powerful driving performance whenever it is needed. In addition, when cruising at high speed while keeping the position of the accelerator pedal nearly constant, the CVT3 can execute subtle ratio changes imperceptible to the driver.

Previously, shift control in CVT applications on small-engine vehicles tended to produce a feeling of an acceleration lag because torque was mainly consumed for raising the engine speed right after the driver depressed the accelerator pedal. Consequently, there was very little vehicle acceleration, and the vehicle did not accelerate until after the engine speed had risen. In contrast, vehicles fitted with a large-displacement engine have high engine torque available even at low speeds, so sufficient acceleration is generated from the moment the driver depresses the accelerator pedal without any need to increase the engine speed hurriedly. That makes it possible to reduce engine noise and maintain a feeling of high vehicle quality. It also results in a feeling of synchronized linearity between the increase in vehicle speed and the rise in engine speed. As a result, the shift pattern specification has been improved from one that was dependent on the transmission ratio, whereby the ratio was substantially changed during acceleration so as to generate the demanded driving force, to one that allows the ratio to be changed within a narrow range (Fig. 6).

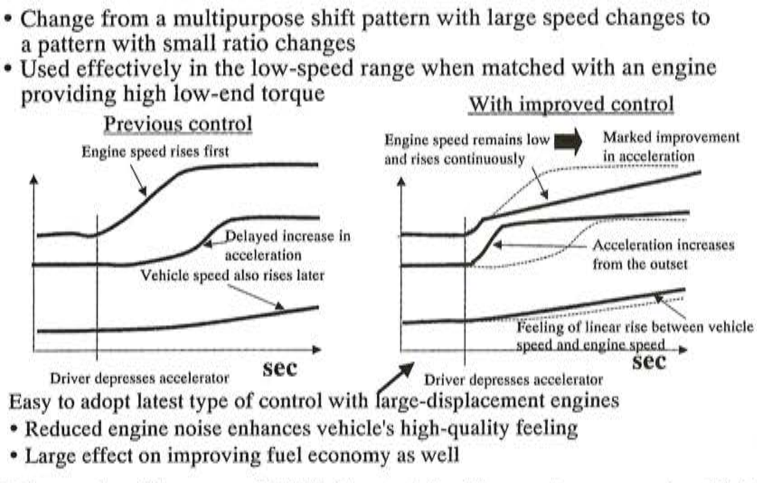

Fig. 6 Example of Improved Shift Control for Use on Large-engine Vehicles

すなわち，大排気量・大トルクエンジンになるほど変速比の変化を小さく設定でき，エンジン回転速度が低めに抑えられるため，以下のメリットがある．
(1)エンジン音が減少し，車両の高級感が増大
(2)燃費向上効果が大

In other words, with larger levels of engine displacement and torque, smaller ratio changes are possible, allowing the engine speed to be kept low and resulting in the following benefits.
(1) Engine noise is reduced to enhance the high-quality feeling of the vehicle.
(2) There is a large effect on improving fuel economy.

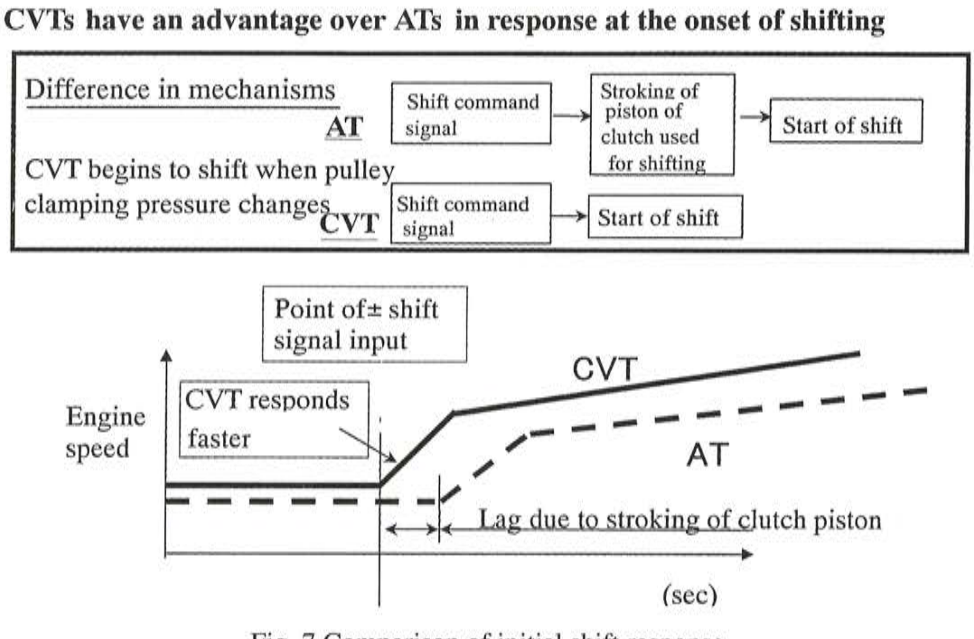

Fig. 7 Comparison of initial shift response

5. マニュアルシフトモードの大排気量車への適用

マニュアルシフトモードでは，変速のレスポンスが非常に重要な性能となる．CVTはATと比較して変速の初期レスポンスの点では，当然有利である．Fig. 7に示すようにATではシフト指令後，変速用のクラッチピストンに油圧が作動し，ピストンがストロークして初めて変速がスタートするのに対して，CVTではプーリのクランプ力を常時フィードバック制御していて，次の変速に備えていることから，シフト指令後直ちに変速をスタートすることができる．このようにクラッチピストンのストロークに要するラグがない分，変速の初動は早い．

HyperCVTでもマニュアルシフトモードを採用したが，今回ムラーノの'04モデルで設定したマニュアルシフトモードについては，大排気量エンジン用として，さらにその制御を改良した．

これまでのCVTのマニュアルシフトモードでは，Fig. 8のように，1速の高回転側でベルトが温度上昇し，熱的強度限界を超えることのないよう，高回転側ではHI側にシフトするように設定していた．ベルトの発熱はトルクが大きくなる程増大するため，大排気量エンジンほどその発熱量は大きくなり，より低い回転からHI側にシフトさせる必要があった．

5. Improvement of Manual Shift Control for Application to Large-engine Cars

Shift response is a critical performance parameter in a manual shift mode. Owing to their inherent mechanism, CVTs have an advantage over ATs with respect to response at the onset of shifting. An AT does not actually begin to shift until the piston of the clutch used in executing the shift strokes under the application of pressure after a shift command has been issued, as indicated in Figure 7. In contrast, the clamping pressure of the pulleys in a CVT is under constant feedback control in preparation for the next shift, enabling the unit to begin shifting immediately after a shift command is issued. Because there is no time lag needed for the stroking of the clutch piston, the shift response of a CVT is that much faster.

While the Hyper CVT also incorporated a manual shift mode, the shift control of the manual shift mode designed for the 2004 model year Murano was further improved for use with a large-displacement engine.

The manual shift mode of previous CVTs was designed to shift to the high ratio side under high engine speeds, as shown in Fig. 8. That was done to prevent the belt from exceeding its thermal strength limit due to the rise in the belt temperature during operation in the first speed range under high engine speeds. Because heat generation by the belt increases in proportion to the torque level, the quantity of heat produced increases with larger engine displacement, making it necessary to shift to the high ratio side at a lower engine speed.

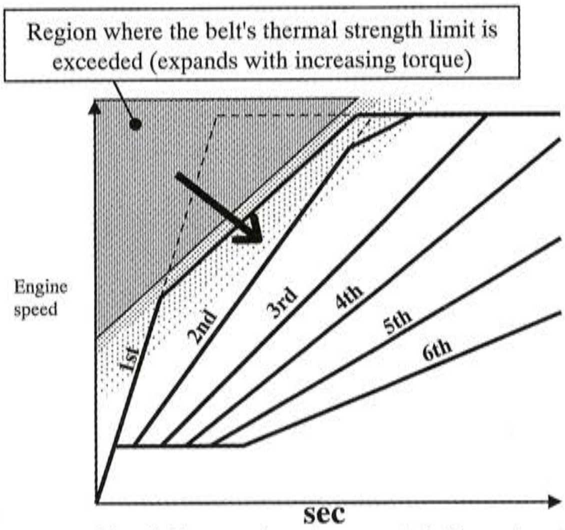

Fig. 8 Conventional manual shift mode

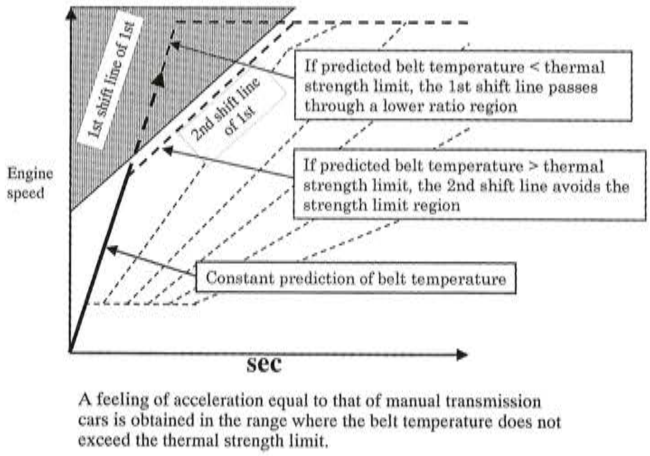

Fig. 9 Improved manual shift mode control

しかしこの様な変速線を設定すると加速時には本来のマニュアトランスミッション車の様な加速フィーリングが得られなくなる．

そこで今回大排気量車でもマニュアルトランスミッション車と同じフィーリングを得られるよう，ベルト温度を油温，プーリ回転数，入力トルクより常時予測してベルト温度が熱的強度限界以下の場合には，従来よりもLOW側に維持するような制御を追加した．(Fig. 9)

ベルトの温度を精度良く予測するために，油温センサの出力電圧を油温変換する際の遅れ回路（フィルター）の迅速化，またベルト冷却用オイルの油温を推定するためにエンジンの水温センサ値を代用し，新たなセンサを追加することなく制御することとした．

6. おわりに

CVTは，軽自動車などの小排気量車に搭載されてスタートしたため，小排気量車にしか適用できないと思われていたイメージが払拭され，中～大排気量車への採用が可能となってきた．

また北米で発売した日産ムラーノはJDパワーの評価でもクラス最高の評価(Fig. 10)を頂き，北米でも受け入れられることが証明できた．

However, with these conventional shift lines, a true feeling of acceleration like that of manual transmission cars cannot be obtained.

To resolve that issue, a shift control was newly added to the CVT3 to keep the transmission ratio more on the low side than in the case of previous CVTs, when the belt temperature is below the thermal strength limit. This control constantly predicts the belt temperature based on the fluid temperature, pulley speed and input torque level, and it enables the CVT3 to deliver the same feeling of acceleration as a manual transmission car even when applied to a car model fitted with a large-displacement engine (Fig. 9).

In order to predict the belt temperature accurately, the operating speed of the delay circuit (filter) has been increased at the time the output voltage of the fluid temperature sensor is converted to a fluid temperature. In addition, the output of the engine coolant sensor is used as a substitute value for predicting the temperature of belt cooling oil. These measures facilitate the above-mentioned shift control without the addition of any new sensors.

6. Conclusion

Because CVTs were initially applied to minicars and other vehicles fitted with small-displacement engines, it was thought that they could only be used on such vehicles. That image has now been dispelled by the development of CVTs suitable for use on vehicles powered by medium to large-displacement engines.

The Nissan Murano marketed in North America received the highest ratings in its class in a survey conducted by J. D. Power and Associates (Fig . 10), which testifies to the popular reception of the CVT3 among North American customers.

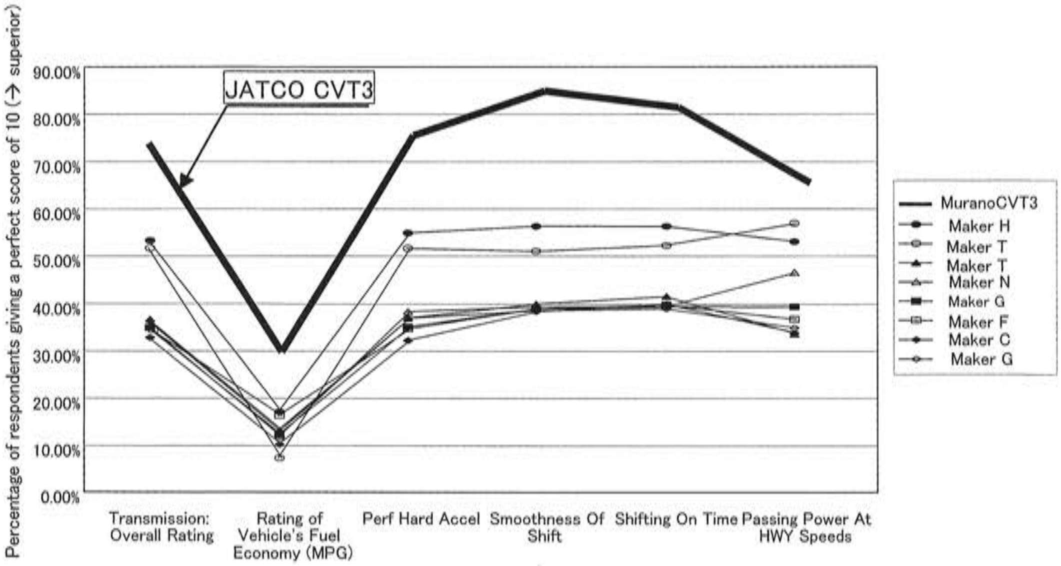

Fig. 10 Comparison of J.D. Power's APPEAL Scores for Transmission Performance

最後に本CVTを開発するにあたり，日産自動車(株)の方々をはじめ，多大なご協力頂いた関係者の方々に深く感謝申し上げる．

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特　集

3.5Lクラス金属ベルトCVTのトルク容量解析

A Study on the Torque Capacity of Belt CVTs for 2.0-liter and 3.5-liter Front Drive Cars

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抄　録　実機サイズのベルトBOX試験機を用いて，プーリ比が最LOW状態における2Lクラスと，3.5LクラスCVTのスリップ特性を測定し，この測定結果に基づいて両CVTのエレメント・プーリ間摩擦特性や，入力プーリ内でエレメント間圧縮力が作用する角度(アクティブアーク)を計算し比較検討した．また，ベルト／プーリの3次元動的挙動や，応力を解析する3次元FEM解析により，両CVTのバンド応力，エレメントPV値について比較した．その結果，3.5LクラスCVTは，2LクラスCVTとほぼ同等のトルク容量余裕率と強度限界があることがわかった．

Summary　An actual-size belt box tester was used to measure the slip characteristics of 2.0-liter and 3.5-liter class CVTs under a condition of their lowest pulley ratio. The measured results were then used to calculate and compare the element-pulley friction characteristics of both CVTs and the active arc of the element compressive force generated on their primary pulley. A 3-D finite element model, capable of analyzing the 3-D dynamic behavior of the belt and pulleys and stress, was applied to compare the band tensile force and element PV values of the two CVTs. As a result, it was found that the 3.5-liter class CVT has approximately the same percentage of torque capacity allowance and strength limits as the 2.0-liter class unit.

1. はじめに

近年，自動車の燃費と走行性能の両方を高いレベルで改善する技術として，CVTに対する期待が大きくなっている．

本報告では，実機サイズのベルトBOX試験機を用いて，トルク容量が限界となる最LOW状態(減速比最大状態)における2Lクラスと3.5LクラスのCVTのベルトスリップ特性を測定し，このスリップ特性の実験結果に基づいて，両CVTにおけるスリップ速度とエレメント/プーリ間の摩擦係数との関係（以後μ-V特性と称す)や，トルク容量余裕代について解析した．また最大トルク伝達状態におけるCVTベルトの3次元動的FEM解析により，エレメントのPV値や，バンド張力とエレメント圧縮力の分布，入力プーリ内のエレメント挙動などを明らかにした．

2. トルク伝達メカニズム

2.1. 金属ベルトの構造

Fig. 1にベルトCVTの構成，Fig. 2に金属ベルトの構造を示す．金属ベルトは2組のスチール製積層バンドに約400枚のスチール製エレメントを連続的に組み付けて構成される．また，各積層バンドは厚さ約0.2mmの無端バンドを9～12枚重ねて構成される．この金属ベルトは入出力一対の金属製のV溝プーリ間に巻きかけられ，エレメントの側面とプーリの円錐面との間に作用する摩擦力で，エレメント間圧縮力と，積層バンド張力の両方を発生させ，それらによりトルクを伝達する．

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1. Introduction

There have been growing expectations of continuously variable transmissions (CVTs) in recent years as a technology capable of significantly improving both the fuel economy and driving performance of vehicles. In this work, an actual-size belt box tester was used to measure the belt slip characteristics of a 2.0-liter and a 3.5-liter class CVT at the lowest pulley ratio (a condition of the maximum reduction ratio) where a transmission reaches its torque capacity limit. Based on the measured slip characteristics, an analysis was made of both CVTs with respect to the relationship between the sliding velocity and friction coefficient between the elements and pulleys (i.e., μ-V characteristic) and their torque capacity allowance. In addition, a dynamic 3-D finite element analysis was made of the CVT belt under a state of maximum torque transmission. The results made clear the PV values of the elements, the distributions of the band tensile force and element compressive force, and element behavior on the primary pulley, among other characteristics.

2. Torque Transmission

2.1. Metal Belt Structure

A cross-sectional view of a belt CVT is shown in Fig. 1 and the structure of the metal belt is shown in Fig.2. The metal belt consists of two laminated steel bands to which approximately 400 steel elements are consecutively attached. Each laminated band is composed of nine or twelve endless layers approximately 0.2 mm in thickness. The metal belt is wound around a pair of metal, V-shaped input and output pulleys. Torque is transmitted both by the compressive force between the elements, produced by the friction force acting between the sides of the elements and the cone-shaped pulley faces, and by the tensile force of the laminated bands.

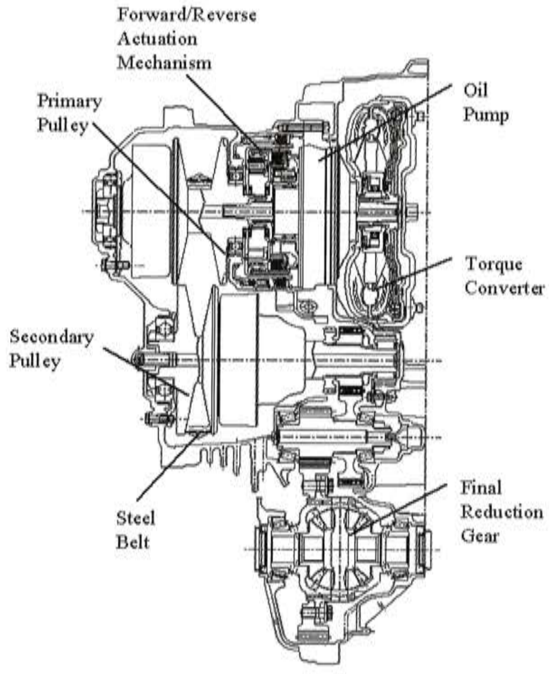

Fig. 1 CVT cross-sectional view

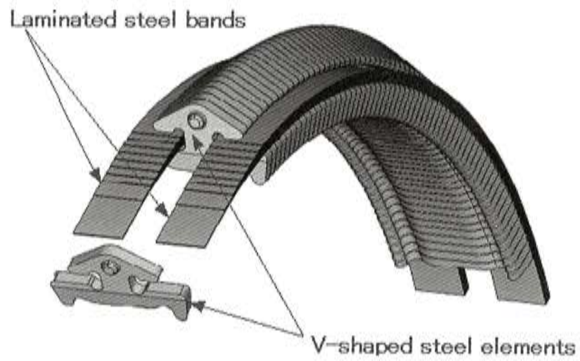

Fig. 2 Structure of a metal CVT belt

2.2. LOW状態でのエレメント圧縮力分布

最LOW状態では，駆動側のベルト巻き付き半径が小さいので，駆動側の積層バンド張力は出口部より入り口部で大きくなる．従って，このバンド張力差は駆動トルクを伝達する方向に作用し，そのトルクは主としてベルトクランプ力で決まる．

そこで，入力トルクがバンドの張力差により伝達するトルクより小さい場合は，エレメント/プーリ間摩擦力によって発生するエレメント間圧縮力Q_bはトルク伝達を邪魔する方向に機能する．(Fig. 3)一方入力トルクが増加して，積層バンド張力によって伝達するトルクT_{band}〔$T_{band}=(T_b-T_a)\times R_1$〕以上になると，エレメント間圧縮力$Q_a$はトルク伝達に寄与するように機能する．(Fig. 4)

2.2. Distribution of Element Compressive Force Under a Low Ratio

Under a low ratio condition, the radius of belt contact on the drive pulley is small, so the band tensile force at the entry side on the drive pulley is larger than that at the entry side on the driven pulley. Accordingly, the band tensile force acts in the direction to transmit torque. When the input torque is small, the element compressive force due to the friction force between the elements and pulleys is distributed on the side acting against the transmission of torque (Fig. 3). On the other hand, as the input torque increases and exceeds the level that can be transmitted by the tensile force of the laminated bands, T_{band} [$T_{band} = (T_b - T_a)\times R_1$], the element compressive force due to the friction force between the elements and pulleys is distributed on the side that assists the transmission of torque (Fig. 4).

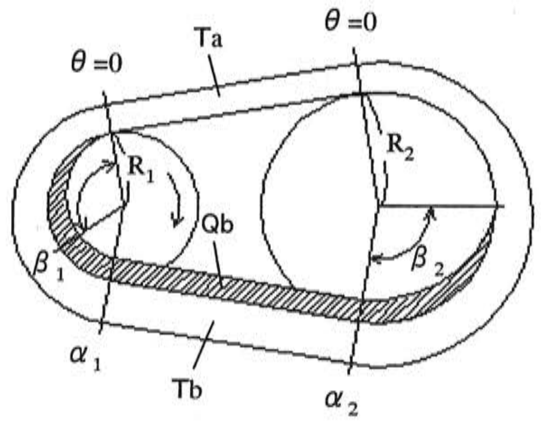

Fig. 3 Distribution of element compressive force and band tensile force ($T_{band}\geqq T_{in}$)

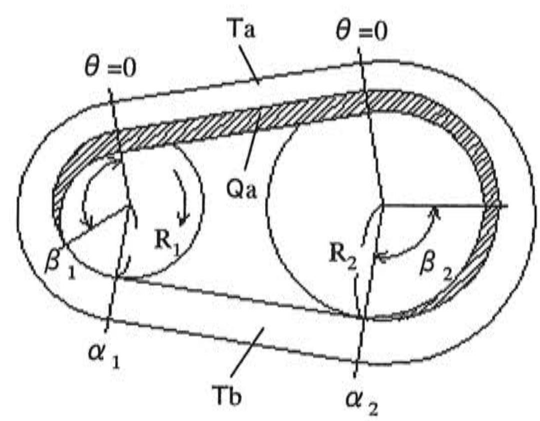

Fig. 4 Distribution of element compressive force and band tensile force ($T_{band}\leqq T_{in}$)

3. ベルトCVTのスリップ特性

3.1. ベルトBOX試験装置と実験方法

Fig. 5にベルトBOX試験機の概要を示す．実機のベルト/プーリを用いて，所定のプーリ比に設定する．このとき，セカンダリーピストン室へ供給する油圧でベルトクランプ力を調整する．駆動側モータは，トルクメータを介してプライマリーシャフトに結合され，セカンダリーシャフトはトルクメータを介して吸収ダイナモに接続されている．ベルト/プーリ部の潤滑用パイプは，実機と同様の位置に配置されている．実験では，プーリ比を最LOW状態，入力回転数を4000rpm一定にセットし，入力トルクを変化させ，そのときの出力軸のトルクと回転数(N_{out})を計測する．各トルク条件におけるスリップ率S(%)は下式により求める．

$$S(\%) = (1 - \frac{N_{out}}{N_o}) \times 100 \qquad (1)$$

ここで，N_oは無負荷時出力回転数(クランプ力一定)である．Table 1に，供試品である2Lクラスと3.5LクラスのベルトCVTの諸元と実験条件の一覧を示す．

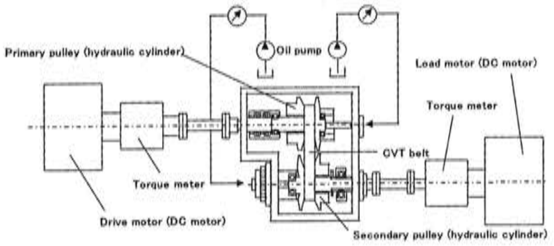

Fig. 5 Schematic of the test rig

3.2. 実験結果

Fig. 6は，2Lクラスと3.5LクラスのベルトCVTのスリップ特性を比較したものである．比較しやすくするため，横軸はトルクレシオ(入力トルク/公称最大トルク)とし，無次元化している.Fig. 6からわかるようにトルクレシオは，2LクラスCVTでは0.38，3.5LクラスCVTでは0.46程度のところで，スリップ率が不連続に変化しており，このトルクレシオの点で，エレメントの圧縮力分布がFig. 3からFig. 4の状態に入れ替わったと推定できる．このことから3.5LクラスベルトCVTの方が，2LクラスベルトCVTより，積層バンドの張力でトルクを伝達する割合が大きいと言える．

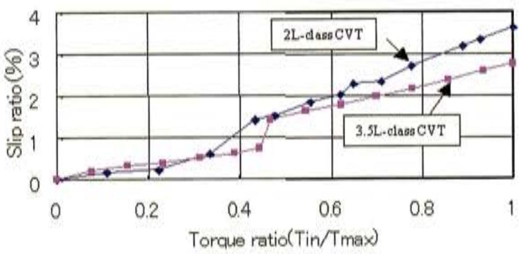

Fig. 6 Slip characteristics of 2L-class and 3.5L-class CVTs

3. Slip Characteristics of Belt CVTs

3.1. Belt Box Tester and Test Method

A schematic diagram of the belt box tester is shown in Fig. 5. The pulley ratio is set to a specified level using the belt and pulleys of the CVT to be tested. The belt clamping force is adjusted by the pressure supplied to the piston chamber of the secondary pulley. The drive motor is connected to the primary pulley shaft via a torque meter, and the secondary pulley shaft is connected to a load dynamometer via a torque meter. The lubrication pipe for the belt and pulleys is positioned in the same place as that of an actual CVT. During a test, the pulley ratio is set to its lowest value, and the input speed is kept constant at 4,000 rpm. Under these conditions, the load torque is varied, and the output speed (N_{out}) is measured for each torque condition. The slip ratio S (%) under each torque condition is found with the following equation.

$$S(\%) = (1 - \frac{N_{out}}{N_o}) \times 100 \qquad (1)$$

where N_o is the output speed under a no-load condition.

Table 1 shows the specifications of the 2.0-liter and 3.5-liter class CVTs used in the tests along with the test conditions.

Table 1 Major specifications and experimental conditions

			2L-class belt CVT	3.5L-class belt CVT
Pulley ratio		Low	2.326	2.371
		OD	0.434	0.439
Belt specs	Center distance of variator (mm)		168	178
	Element width (mm)		30	30
	Element thickness (mm)		1.8	1.8
	Nnmber of band layers		9	12
	Length of CVT belt (mm)		703	747.7
Experimental conditions	Input speed		4000 rpm	
	Oil temperature		110℃	
	Pulley ratio		Lowest	
	Input torque		No load～Max. torque	

3.2. Test Results

Figure 6 compares the slip characteristics found for the 2.0-liter and 3.5-liter class CVTs. To facilitate an easy comparison, the torque ratio along the horizontal axis is expressed as a non-dimensional value in terms of the input torque/nominal maximum torque. As is clear from Fig. 6, the slip ratio of the 2.0-liter and 3.5-liter class CVTs changed discontinuously at a torque ratio of approximately 0.38 and 0.46, respectively. It is estimated that the distribution of the element compressive force at these torque ratios switched from the state in Fig. 3 to that in Fig. 4. This would seem to indicate that the 3.5-liter class belt CVT transmitted a larger proportion of torque by the tensile force of the laminated bands than in the case of the 2.0-liter class belt CVT.

4. トルク伝達容量と摩擦特性に関する理論解析

ベルトCVTのトルク容量は，エレメント/プーリ間のμ-V特性に大きく依存し，摩擦係数μ_bが大きいほどトルク伝達容量は大きくなる．また限界トルク容量の大小は，小径側プーリのベルト巻き付き角(α_1)に対するアクティブアーク(β_1)の比率(β_1/α_1)とトルクレシオとの関係で評価できる．ここでアクティブアークとは，小径側巻き付き角の中で，エレメント間圧縮力に実質的に寄与する角度である．以下に数値解析により，2Lクラスと3.5LクラスのベルトCVTの各々について，μ-V特性とアクティブアーク比率β_1/α_1を求め比較検討した．

4.1. 基礎式

Fig. 7は，ベルトCVTにとってトルク容量的にもっとも厳しい最LOW状態で，入力トルクが大きい状態におけるエレメント圧縮力分布をモデル化したものである．

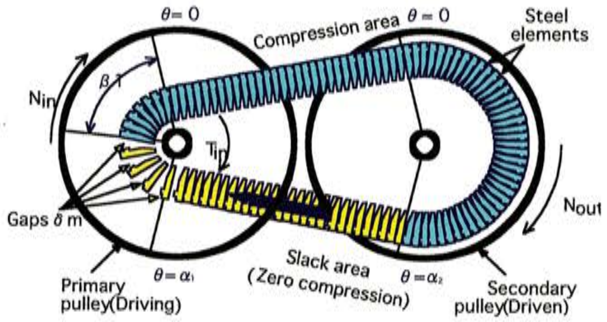

Fig. 7 Slip mechanism of a metal CVT belt

エレメント間には組み立て時から微小な隙間が存在し，この初期隙間の全体量(δ_0)をエンドプレーと呼んでいる．そして，入力プーリの巻き付き開始部ではエレメント間圧縮力が発生せず，このエンドプレーδ_0に基づいて発生するエレメント間隙間(δ_m)により，ベルトスリップが発生する[1]．従って，スリップ率S(%)は下式で表せる．

$$S(\%) = \left\{ \delta_m \Big/ (L_e + \delta_m) \right\} \times 100 \quad \text{―――――} \quad (2)$$

ここで*Le*はエレメント1枚あたりの板厚である．

つぎにベルトCVTのトルク伝達は，積層ベルト張力$T(\theta)$と，エレメント間圧縮力$Q(\theta)$により決定される．Fig.8に示すようなエレメントとバンドの微小巻きかけ部分dθについての力の釣り合いから以下の式が導かれる[2]．

$$T(\theta) = T_1 e^{\mu_a \theta} + Cr \quad (T \geq 0) \quad \text{―――――} \quad (3)$$

$$Q(\theta) = Q_1 e^{\mu\theta} - T_1 e^{\mu_a \theta} + Ce \quad (Q \leq 0) \quad \text{―――――} \quad (4)$$

$$\mu = \mu_b \cos\zeta \Big/ (\sin\varphi + \mu_b \sin\varsigma \times \cos\varphi) \quad \text{―――――} \quad (5)$$

4. Theoretical Analysis of Torque Capacity and Friction Characteristic

The torque capacity of a belt CVT depends greatly on the μ-V characteristic between the elements and pulleys and increases with a larger friction coefficient μ_b. The magnitude of the limit torque capacity can be evaluated on the basis of the relationship between the ratio of the active arc (β_1) to the belt angle of contact (α_1), (β_1/α_1), on the small-diameter pulley and the torque ratio. The active arc refers to the angle that contributes substantially to the element compressive force within the angle of belt contact on the small-diameter pulley. In the following numerical analysis, the μ-V characteristic and the active arc ratio b1/a1 of the 2.0-liter and 3.5-liter class CVTs will be found and compared.

4.1. Basic Equations

Figure 7 shows a model of the distribution of element compressive force under a condition of large input torque at the lowest pulley ratio, which is the severest state for a belt CVT in terms of torque capacity.

Micro gaps are present between the elements from the time the belt is assembled. The total amount of the initial gaps (δ_0) is called end play. Compressive force between the elements is not generated at the onset of belt engagement with the primary pulley. The element gaps (δ_m) resulting from this end play δ_0 give rise to belt slip.[1] Accordingly, the slip ratio S (%) can be expressed as

$$S(\%) = \left\{ \delta_m \Big/ (L_e + \delta_m) \right\} \times 100 \quad \text{―――――} \quad (2)$$

where Le is the plate thickness of one element.

The amount of torque transmitted by a belt CVT is determined by the tensile force of the laminated bands T(θ) and element compressive force Q(θ). The following equations are derived from the balance of forces acting on the elements and bands in a micro region of belt contact dθ, as shown in Fig. 8.[2]

$$T(\theta) = T_1 e^{\mu_a \theta} + Cr \quad (T \geq 0) \quad \text{―――――} \quad (3)$$

$$Q(\theta) = Q_1 e^{\mu\theta} - T_1 e^{\mu_a \theta} + Ce \quad (Q \leq 0) \quad \text{―――――} \quad (4)$$

$$\mu = \mu_b \cos\zeta \Big/ (\sin\varphi + \mu_b \sin\varsigma \times \cos\varphi) \quad \text{―――――} \quad (5)$$

where

C_r : centrifugal force per unit length of the laminated bands
C_e : centrifugal force per unit length of the elements
μ_a: friction coefficient between the bands and elements
μ_b: friction coefficient between the elements and pulleys
ϕ : pulley half vertical angle
ζ : deflection angle from the tangential direction of element sliding

ここで

C_r：積層バンド単位長さあたりの遠心力

C_e：エレメント単位長さあたりの遠心力

μ_a：バンド/エレメント間摩擦係数

μ_b：エレメント/プーリ間摩擦係数

ϕ：プーリ半頂角

ζ：エレメント摺動接線方向からの偏角

入力プーリ巻き付き角度内のエレメント間隙間δ_mとアクティブアークβ_1に関しては以下の関係式が成立する．

$$\beta_1 = \alpha_1 - \frac{1}{R_1}\left\{\left(\frac{L_e}{\delta_m}+1\right)\delta_t - L_e\right\} \quad (6)$$

ここで，δ_tはエンドプレーδ_0にベルトに力が作用することで発生する付加的な隙間量を加算したものである．

エレメント/プーリ摺動速度は以下の式で表せる．

$$V(m/\mathrm{sec}) = \frac{L}{60}\frac{N_{in}}{I_p}\frac{S}{100000} \quad (7)$$

ここでN_{in}:入力回転数(rpm)，I_p:プーリ比，L:ベルト全長(mm)

Fig. 6のスリップ特性実験結果を用いて，2Lクラスと3.5LクラスのCVTのエレメント/プーリ間μ-V特性ならびにアクティブアーク比率β_1/α_1を以下の手順で求める．

エンドプレーδ_0はエレメント圧縮分布が入れ替わる入力トルクT_{ic}でのスリップ率S_0を(2)，(6)式に代入し，(6)式におけるδ_tはδ_0と等しいとして求める．また，エレメント圧縮分布が入れ替わる時点におけるトルクは，バンド張力のみでトルクが伝達されるのでエレメント圧縮力はゼロと考え，(3)，(4)式を使ってバンド-エレメント間摩擦係数μaを求める．次にエレメント間圧縮力がトルク伝達に寄与するトルクレシオ($T_{in}/T_{max} \geqq T_{ic}/T_{max}$)におけるスリップ率の実験結果を逐次(2)～(7)に代入してエレメント/プーリ間摩擦係数μ_b，アクティブアークβ_1を求める．なお，トルクレシオがT_{ic}/T_{max}以上の入力状態におけるバンド張力によるトルク伝達分担は一定(=T_{ic})と考える．

4.2. エレメント/プーリ間のμ-V特性

Fig.9は，2Lクラスと3.5LクラスにおけるCVTのエレメント/プーリ間のμ-V特性について計算し比較したものである．2LクラスCVT用の作動油は添加剤としてZnDTPが含まれており，この添加剤により高いμを確保している[3]．今回3.5LクラスCVTではZnDTPを含まない新油を採用したが，ほぼ同等またはそれ以上に高いμを確保していることが推定できる．また，トルクレシオ1における摺動速度V_1は，2Lクラスの場合0.71m/s，3.5Lクラスの場合0.64m/sである．

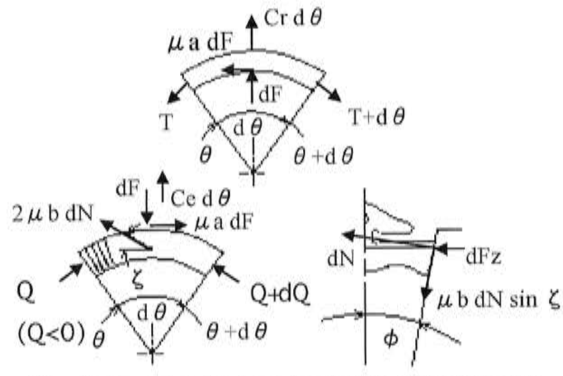

Fig. 8 Forces acting on the bands and elements

The following relational equation holds true between the band gaps δ_m and the active arc β_1 within the angle of belt contact on the primary pulley.

$$\beta_1 = \alpha_1 - \frac{1}{R_1}\left\{\left(\frac{L_e}{\delta_m}+1\right)\delta_t - L_e\right\} \quad (6)$$

where δ_t is the result of adding the amount of additional gaps produced by the force acting on the belt to the end play δ_0.

The sliding velocity between the elements and pulleys is given by

$$V(m/\mathrm{sec}) = \frac{L}{60}\frac{N_{in}}{I_p}\frac{S}{100000} \quad (7)$$

where N_{in} is the input speed (rpm), I_p the pulley ratio and L the total belt length (mm).

Using the results of the slip characteristics test in Fig. 6, the μ-V characteristic between the elements and pulleys and the active arc ratio β_1/α_1 of the 2.0-liter and 3.5-liter class CVTs were found according to the procedure explained below. The end play δ_0 is found by substituting into Eqs. (2) and (6) the slip ratio S_0 at the input torque T_{ic} where the element compressive force distribution changes, under the assumption that dt in Eq. (6) is equal to δ_0. The torque at the time the element compressive force distribution changes is transmitted only by the band tensile force, so the element compressive force is regarded as being zero. The friction coefficient μ_a between the bands and elements is found using Eqs. (3) and (4). The test results for the slip ratio at the torque ratio ($T_{in}/T_{max} \geqq T_{ic}/T_{max}$) where the element compressive force contributes to torque transmission are then substituted successively into Eqs. (2)-(7) to find the friction coefficient μ_b between the elements and pulleys and the active arc β_1. It is assumed that the share of the torque transmitted by the band tensile force under an input state where the torque ratio is T_{ic}/T_{max} or greater is constant (= T_{ic}).

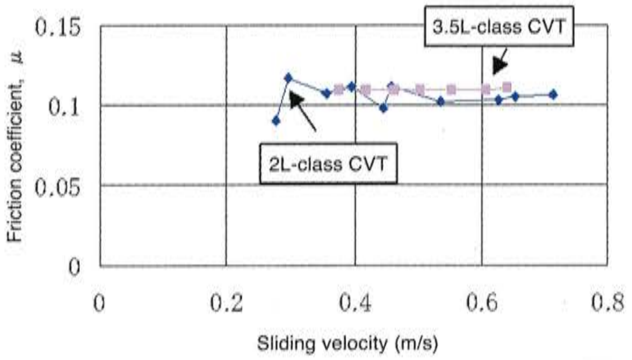

Fig. 9 μ -V characteristics of 2L-class and 3.5L-class CVTs

4.3. 限界伝達トルク容量に対する余裕度

ベルトCVTでは，アクティブアーク比（β_1/α_1）が1となる入力トルク以上のトルクを伝達することは理論的にできない．従って，アクティブアーク比とトルクレシオとの関係を調べることにより，ベルトCVTの限界伝達トルクに対する余裕度が推定可能となる．Fig. 10は，アクティブアーク比（β_1/α_1）とトルクレシオとの関係を，2Lクラスと3.5LクラスのCVT同士で比較したものである．アクティブアーク比が小さいほど，トルク容量の余裕度は高いと言える．Fig. 10から明らかなように，3.5LクラスCVTは2LクラスCVTと同等以上の余裕度を持っていることがわかる．

4.2. Element-Pulley μ -V Characteristic

Figure 9 compares the element-pulley μ -V characteristics calculated for the 2.0-liter and 3.5-liter class CVTs. The fluid of the 2.0-liter class CVT contains the Zn-DTP additive. It has been reported that this additive maintains a high μ [3]. A new fluid without ZnDTP has been adopted for the 3.5-liter class CVT recently. It can be inferred from the results in the figure that this fluid provides an equally high or even higher μ . The slip velocity V_1 at a torque ratio of 1 was found to be 0.71 m/sec for the 2.0-liter class CVT and 0.64 m/sec for the 3.5-liter class CVT.

4.3. Limit Torque Capacity Allowance

Theoretically, a belt CVT cannot transmit torque once the input torque reaches a level where the active arc ratio (β_1/α_1) is 1. Accordingly, the allowance with respect to the torque transmission limit of a belt CVT can be estimated by investigating the relationship between the active arc ratio (β_1/α_1) and torque ratio. Figure 10 compares this relationship for the 2.0-liter and 3.5-liter class CVTs. It is seen from the figure that the smaller the active arc, the higher the torque capacity allowance becomes. As is clear from Fig. 10, the allowance of the 3.5-liter class CVT is as large as or even larger than that of the 2.0-liter class CVT.

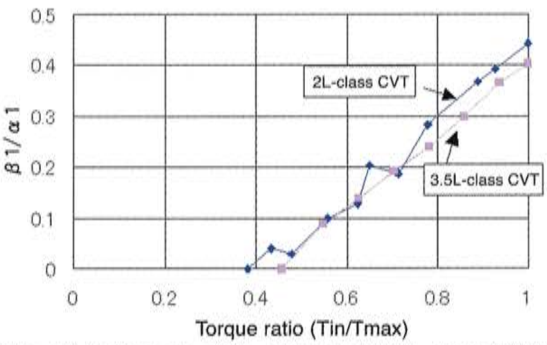

Fig. 10 Active arc of 2L-class and 3.5L-class CVTs

5. 動的挙動，応力比較

トルクレシオ1の状態における，3.5Lクラスと2LクラスのCVTそれぞれの動的挙動，応力を3次元FEMにより解析し，比較検討した．

5.1. モデルの構成 [4]

ベルトCVTを構成するエレメント，バンド，プーリの各部品1つずつの3次元有限要素データを連成させ，エレメント同士，エレメントとバンド，エレメントとプーリ間すべてに接触定義を設定している．積層バンドについては，計算精度と計算時間との兼ね合いから引張剛性が等価な単層モデルで近似している．またプーリと，エレメントは剛体として扱っている．Fig. 11にモデルの概要を示す．

5. Comparison of Dynamic Behavior and Stress

Three-dimensional finite element analyses were conducted to analyze and compare the dynamic behavior and stress of the 2.0-liter and 3.5-liter class CVTs under a condition of a torque ratio of 1.

5.1. Model Configuration[4]

Three-dimensional FEA data for each of the components making up a belt CVT, namely the elements, bands and pulleys, were coupled and the contact between the elements, that between the elements and bands and that between the elements and pulleys were all defined. The laminated bands were approximated with a single-layer model having equivalent tensile rigidity in order to strike a good balance between accuracy and CPU time. The pulleys and elements were treated as solids. The FEA model of the belt-pulley assembly is shown in Fig. 11.

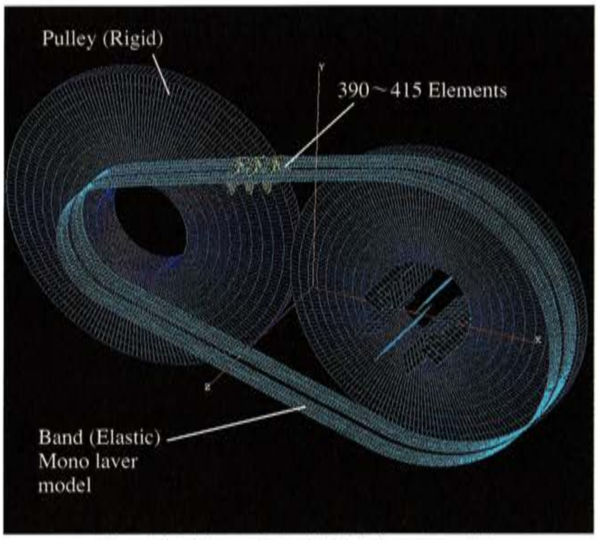

Fig. 11 FE model of V-belt assembly

5.2. 計算手順

ベルトCVTを実機と同様の拘束条件で多接触モデルとして計算するため，計算ツールには有限要素モデルPAM-SHOCKを用いた．所定のプーリ比におけるベルト巻きかけ状態で，入力トルク0，入力回転数0の状態から入力回転数を目標の回転数まで徐々に上昇させ，その後，入力トルクを所定値まで上昇させ，各部の挙動が定常状態に収束するまで計算を実行する．

5.3. 計算結果

入力回転数4000rpm，プーリ比最LOW，トルクレシオ1の条件における両CVTのエレメント/プーリ間，エレメント/バンド間，エレメント/エレメント間の相互作用力や，面圧，速度などについて計算した．Fig. 12は，定常状態のある時刻における各エレメント重心の速度分布と，エレメント重心に対応する回転半径でのプーリ速度，ならびにエレメント間押力分布とバンド張力分布を示したものである．上段が2Lクラス CVT，下段が3.5LクラスCVTである．両CVTとも入力プーリに巻き付き開始した直後のエレメントは，急速にプーリ速度まで加速される．その後エレメント間圧縮力が発生するアクティブアーク状態になるが，この領域にあるエレメントの速度は，プーリ速度より遅くなりスリップ状態になっていることがわかる．また，2LクラスCVTの場合，バンド張力で伝達するトルクは，入力トルクの38%である．一方3.5LクラスCVTの場合，バンド張力のトルク分担率は44%になっている．これは，3.2節の実験結果とよく一致している．Fig. 13は1個のエレメントに着目し，入力プーリに巻き付いてから入力プーリを出るまでの間のエレメント側面部荷重と，プーリに対するスリップ速度の推移を計算したものであり，上段が2LクラスCVT，下段が3.5LクラスCVTである．

エレメント/プーリ間の摺動速度は，2LクラスCVTの場合0.5m/sから0.8m/s程度であるのに対して，3.5LクラスCVTの場合0.5m/sから0.6m/s程度である．この結果は，4.2節で計算したトルクレシオ1における摺動速度とほぼ一致している．

5.2. Calculation Procedure

The PAM-SHOCK finite element program was used as the computational code in order to calculate the multi-contact model under the same restraints as an actual belt CVT. Under a state of belt contact at the specified pulley ratio, the input speed was gradually increased to the prescribed speed from a condition of zero input torque and zero input speed. The input torque was then increased to the specified level, and the calculation was run until the behavior of each component converged to a steady state.

5.3. Calculation Results

The interactive forces, contact pressures, velocities and other characteristics between the elements and pulleys, between the elements and bands and between the elements themselves were calculated for both CVTs under conditions of an input speed of 4,000 rpm, the lowest pulley ratio and a torque ratio of 1. Figure 12 shows the calculated velocity distribution at the center of gravity of each element at a certain time under a steady-state condition, the pulley velocity at the radius of revolution corresponding to the element center of gravity, the distribution of compressive force between the elements and the band tensile force distribution. The upper graph is for the 2.0-liter class CVT and the lower one is for the 3.5-liter class CVT. The results indicate that the elements of both CVTs were rapidly accelerated to the pulley velocity right after the belt began to wrap around the primary pulley. Subsequently, the active arc state developed under which compressive force was generated between the elements. The velocity of some elements in this region was slower than the pulley velocity, giving rise to a state of belt slippage. In the case of the 2.0-liter class CVT, band tensile force transmitted 38% of the input torque. In contrast, the band tensile force of the 3.5-liter CVT transmitted 44% of the input torque. The calculated results show good agreement with the experimental data mentioned earlier.

The results in Fig. 13 were calculated for one element from the time it began to wrap around the primary pulley until it exited the pulley. The figure shows the thrust force on the sides of the element and the change in its sliding velocity relative to the pulley. The upper graph is for the 2.0-liter class CVT and the lower one is for the 3.5-liter class CVT. For the 2.0-liter class CVT, the sliding velocity between the element and the pulley ranged between 0.5 to 0.8 m/s, whereas that of the 3.5-liter class CVT was around 0.5 to 0.6 m/s. These results nearly coincide with the sliding velocities mentioned earlier that were calculated under a torque ratio of 1. For both CVTs, the thrust force acting on the sides of the element reached its maximum value right after the onset of the active arc and was 940N-m for the 2.0-liter class CVT and 1,404 N-m for the 3.5-liter class CVT.

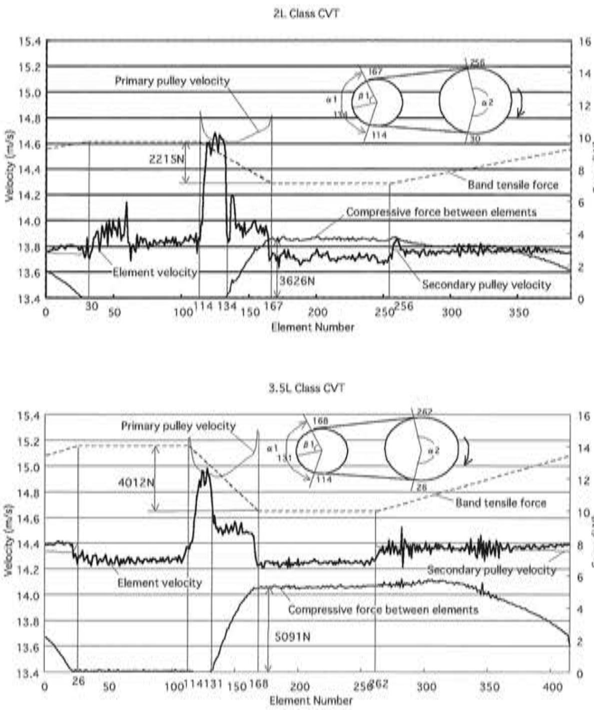

Fig. 12 Comparison of dynamic behavior and stress of metal CVT belts

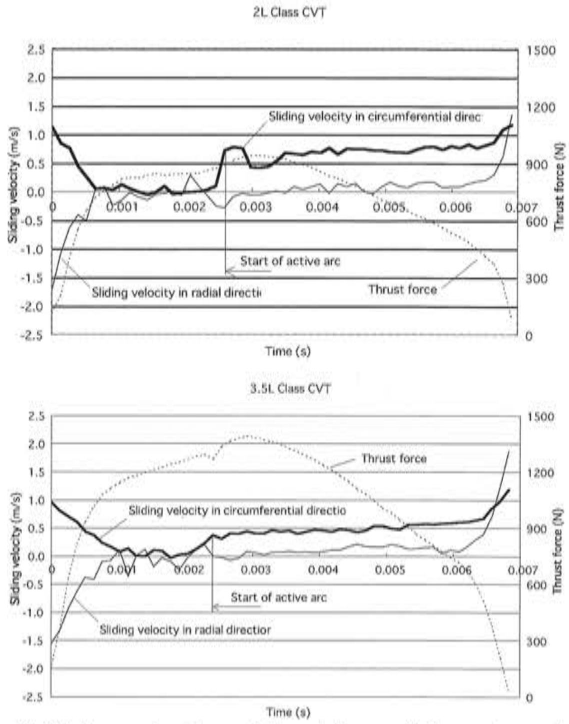

Fig. 13 Sliding velocity and thrust force of elements on the primary pulley

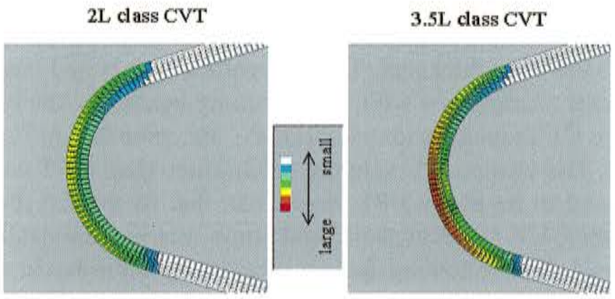

Fig. 14 Comparison of element contact pressure on the primary pulley

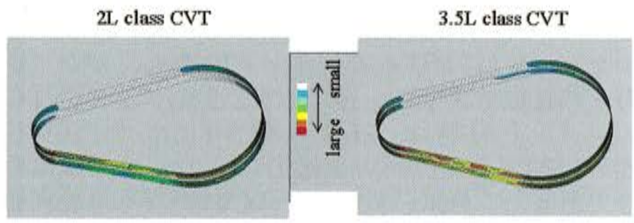

Fig. 15 Comparison of band tensile force

エレメント側面部に作用する荷重は，両CVTともアクティブアーク開始直後に最大となり，2LクラスCVTでは940N，3.5LクラスCVTでは1404Nである．

Fig. 14は入力プーリに接触しているエレメント側面部の面圧分布の比較結果である．2LクラスCVT，3.5LクラスCVTともエレメント側面部上側で高い面圧になっていることがわかる．

Fig. 15はバンド張力分布の比較結果である．両CVTとも入力プーリ巻き付き開始部でバンド引張り応力は最大となり，2LクラスCVTの場合，最大値は37MPa．3.5Lクラスの場合は39MPaである．

Figure 14 compares the distribution of contact pressure on the side faces of the elements in contact with the primary pulley for both CVTs. The results indicate that the contact pressure at the top of the element side faces was high for both the 2.0-liter and 3.5-liter class CVTs.

Figure 15 compares the band tensile force distribution of both CVTs. Band tensile stress reached its maximum level for both CVTs at the point where the belt began to wrap around the primary pulley. The maximum stress value seen for the 2.0-liter class CVT was 37 MPa and that for the 3.5-liter class CVT was 39 MPa.

5.4. エレメントPV値，バンド応力の比較

5.3.節の計算結果に基づき，トルクレシオ1の状態における，3.5Lクラスと2LクラスのCVTそれぞれのエレメントPV値とバンド応力を求める．エレメントPV値のPは，Fig.13で，アクティブアーク内にあるエレメント側面部荷重の平均値を用いてヘルツ面圧P_Hを下式で求めた．Vについてはトルクレシオ1におけるエレメント／プーリ間摺動速度V_lを用いた．

$$P_H=\sqrt{\frac{N\times E_e\times\sin(\varphi)}{2\pi\times L_s\times R_1\times(1-\upsilon^2)\times\varepsilon}} \qquad (8)$$

ここで，Nはエレメント側面垂直抗力，E_eはエレメント弾性係数，L_sはエレメント側面長さ，νはポアソン比，εはエレメント側面部凸凹のプラトー率である．

バンド応力τは，引張り応力σ_Tと曲げ応力σ_Mの和として下式で求める．

$$\tau=T(\alpha_1)/(2\times A_r)+E_B t/(2\times R_1) \qquad (9)$$

ここで，$T(\alpha_1)$は，Fig. 12に示す入力プーリに巻き付き開始する位置における引張り荷重である．A_rは積層バンド断面積であり，E_Bはバンド弾性係数，tはバンド板厚である．以上の計算により求めたトルクレシオ1におけるエレメントPV値と，バンド応力τの比較結果をFig. 16に示す．3.5LクラスCVTのエレメントPV値は，2LクラスCVTに比べて10％程度の増加であり，バンド応力については，ほぼ同等である．これは，3.5LクラスCVTのプーリ軸間距離が2LクラスCVTに比べ増加したこと，摺動速度V_lが小さいこと，および積層バンド数を9から12層に増やしたことが貢献したと思われる．

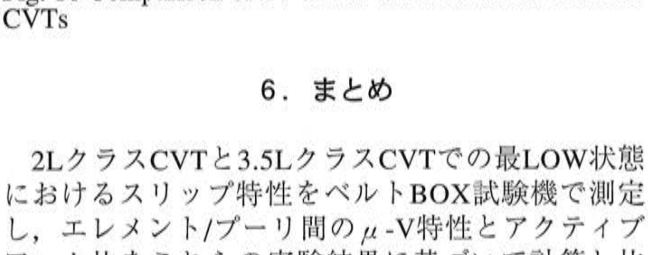

Fig. 16 Comparison of PV factor and band stress between CVTs

6．まとめ

2LクラスCVTと3.5LクラスCVTでの最LOW状態におけるスリップ特性をベルトBOX試験機で測定し，エレメント/プーリ間のμ-V特性とアクティブアーク比をこれらの実験結果に基づいて計算し比較した．また，トルクレシオ1における両CVTの動的挙動，応力について3次元FEMで解析し，以下のことが判明した．

5.4. Comparison of element PV values and band stress

Based on the foregoing calculated results, the element PV values and band stress were found for both CVTs under a condition of a torque ratio of 1. The value of P here was found with the following equation as the Hertzian contact pressure P_H, using the mean value of the thrust force acting on the sides of the elements in the active arc in Fig. 13. The element-pulley sliding velocity V_l under a torque ratio of 1 was used as the value of V.

$$P_H=\sqrt{\frac{N\times E_e\times\sin(\varphi)}{2\pi\times L_s\times R_1\times(1-\upsilon^2)\times\varepsilon}} \qquad (8)$$

where N is the vertical drag on the element sides, Ee is the element modulus of elasticity, Ls is length of the element sides, ν is Poisson's ratio and ε is the plateau ratio of the surface profile of the element sides.

Band stress τ was found with the following equation as the sum of tensile stress σ_T and bending stress σ_M.

$$\tau=T(\alpha_1)/(2\times A_r)+E_B t/(2\times R_1) \qquad (9)$$

where $T(\alpha_1)$ is the tensile force at the point where the belt begins to wrap around the primary pulley as shown in Fig. 12, Ar is the cross-sectional area of the laminated bands, E_B is the band modulus of elasticity and t is the band thickness. The element PV values and band stress calculated with the foregoing equations for the two CVTs under a torque ratio of 1 are compared in Fig. 14. The element PV value of the 3.5-liter class CVT was found to be about 10% larger than that of the 2.5-liter class CVT, whereas their band stress was approximately equal. The following factors presumably contributed to these results: the distance between the pulley shafts was longer for the 3.5-liter class CVT than for the 2.5-liter class CVT, the sliding velocity V_l was low, and the number of layers of the laminated bands was increased from nine to twelve for the 3.5-liter class CVT.

6. Conclusions

The slip characteristics of 2.0-liter and 3.5-liter class CVTs were measured under a condition of their lowest pulley ratio using an actual-size belt box tester. Based on the measured results, the μ-V characteristic between the elements and pulleys and the active arc were calculated and compared for the two CVTs. A 3-D FEA model was used to analyze the dynamic behavior and stress of both CVTs. The results obtained made clear the following points.

(1) The increased number of layers of the laminated bands of the 3.5-liter class CVT enabled the band tensile force to transmit torque up to a torque ratio of 0.45, which contributes to the greater torque capacity of this unit.

1. 3.5LクラスCVTの場合，バンド積層数を9から12層にしたことにより，バンド張力でトルク伝達が可能なトルクレシオは0.45まで増加したことがトルク容量増加に寄与している.
2. 3.5LクラスCVTのエレメント/プーリ間のμは，相対速度Vに関わらず2LクラスCVTとほぼ同等以上であることがトルク容量増加に寄与している.
3. 3.5LクラスCVTの限界伝達トルクに対する余裕度は2LクラスCVTの余裕度と同等以上であることが，アクティブアーク比率から推定できる.
4. 最LOW状態と，トルクレシオ1の状態とにおけるCVTベルトの動的挙動とバンド応力は，2LクラスCVTと3.5LクラスCVTとでほぼ同等であるが，エレメントPV値で比較すると，3.5LクラスCVTでは2LクラスCVTに比べて10%程度増加することが推定できた.

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(2) The element-pulley μ of the 3.5-liter class CVT was equal to or greater than that of the 2.5-liter class CVT regardless of the relative velocity V. This also contributes to the former unit's increased torque capacity.
(3) It was estimated from the active arc ratio that the torque capacity allowance of the 3.5-liter class CVT was equal to or greater than that of the 2.0-liter class CVT.
(4) Under the conditions of the lowest pulley ratio and a torque ratio of 1, the dynamic behavior of the CVT belt was approximately the same for both the 2.0-liter and 3.5-liter class CVTs. A comparison of their element PV values showed that the value of 3.5-liter class CVT was around 10% greater than that of the 2.0-liter class CVT.

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特　集

薄型トルクコンバータの開発

Development of the Super Ultraflat Torque Converter

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抄　録　トルクコンバータの内部流れをコンピュータ解析することにより，各ブレード形状を最適化しながら軸方向寸度を極力短縮した．本稿では，新開発JF010E型無段自動変速機用トルクコンバータへの適用例について解説する．

Summary　This paper describes the new super ultraflat torque converter adopted for our new JF010E automatic transmission. The internal flow characteristics of the new torque converter were invested by Computational Fluid Dynamics (CFD).

1. はじめに

トルクコンバータ(以下TCとする)は，入出力軸の回転速度差に応じて自動的に伝達トルクが決まる回転速度差感応型の流体継手である．

TCは，エンジンの爆発などによるトルク変動を吸収するとともに，出力軸が停止しているときには入力トルクを二倍程度に増幅する機能を有するほか，エンジントルクの急変に対して滑らかなトルク伝達特性を有すること，乾式・湿式・電磁紛クラッチ等他の発進要素に対して熱容量が大きく信頼性も高いことなど，優れた発進用継手として，無段変速機を含む各種の自動変速機に用いられている．

しかし，燃費や重量の改善要求から小型化やロックアップの高機能化のニーズは高く，小型化を目的としたTCの開発[1]が報告されている．さらに，より大型多気筒エンジンを前輪駆動方式に採用したり，変速機の多段化や無段化と高機能化が進むにつれて，TCに与えられるスペースはますます少なくなっており，TCの軸方向寸法を短縮する扁平化の要求も高い．

また近年，流体解析がハード・ソフトともに目覚しい発展を遂げているが，いまだTC内の複雑な三次元流れを正確に定量解析できているわけではない．しかし，現在の解析技術を用いて，現状から各パラメータを変更したときの変化分を定量解析することはでき，高性能なTCの最適設計が可能なはずである．

本稿では，上記の要求を背景に開発されたJF010E型無段自動変速機用TCについて紹介する．

1. Introduction

A torque converter is a type of hydrodynamic coupling that automatically varies torque in response to the rotational speed difference between the input/output shafts, which determines the level of torque that is transmitted. Along with absorbing torque fluctuations caused by engine explosions and other factors, a torque converter functions to multiply input torque by approximately twofold when the output shaft is stopped. Additionally, it is also characterized as having smooth torque transmission characteristics relative to sudden changes in engine torque, high reliability with a large heat capacity in relation to other start-off elements such as dry, wet and electromagnetic powder clutches, and other excellent features. A torque converter is now used with various types of automatic transmissions, including continuously variable transmissions (CVTs), as a coupling that provides outstanding start-off capability.

However, there are strong needs for smaller torque converters and enhancement of lock-up functionality in order to meet demands for fuel economy improvements and vehicle weight reductions. Efforts to develop downsized torque converters have been reported in the literature.[1] Moreover, the space allowed for mounting a torque converter has been increasingly reduced owing to the adoption of larger engines with more cylinders in front-wheel-drive systems and the higher functionality achieved for automatic transmissions, including the addition of more speed ranges and greater use of CVTs. Accordingly, a flatter torque converter design with a shorter axial length is strongly required.

Despite the remarkable advances seen in recent years in both the hardware and software aspects of fluid analysis, the complicated 3-D flow through the torque converter has still not been accurately analyzed quantitatively. However, it should be possible to obtain the optimum design of a high-performance torque converter by applying current analytical techniques to analyze quantitatively the amount of change that occurs as a result of varying each parameter from the present design value.

This paper describes the torque converter developed for the JF010E AT against the backdrop of the above-mentioned needs.

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2．流体要素の開発

2.1. 扁平化のトレンド

1980年代より，中小排気量車に前輪駆動方式を採用するものが多くなり始め，現在ではこちらの方が主流となっている．前輪駆動方式に搭載するため，軸方向寸度を短縮した扁平型TCが開発されている．車両への搭載性を考えると，Fig. 1に示すTCの薄型率(W/D)がポイントとなるが，流体性能の観点からはTCの扁平率(W/H)が焦点になる．ここでは，TCの扁平率と生産年度の関係(Fig. 2)から，扁平化のトレンドを見てみる．ここ数年で，扁平化が急速に進んでいることが分かる．

2. Development of Hydrodynamic Elements

2.1. Trend toward a smaller aspect ratio

Front-wheel-drive systems began to be adopted on many cars powered by small and medium displacement engines from the early 1980s and are now the mainstream drive configuration of such vehicles. Flatter torque converters with a shorter axial length have thus been developed for use with front-wheel-drive systems. In considering ease of mounting on a vehicle, the main point is the flatness ratio (W/D) of a torque converter, but from the standpoint of hydrodynamic performance, the key point is the unit's aspect ratio (W/H), as illustrated in Fig. 1. The trend toward torque converters with a smaller aspect ratio is shown here in terms of the relationship between the aspect ratio and year of production (Fig. 2). It is seen that rapid strides toward a smaller aspect ratio have been made these last few years.

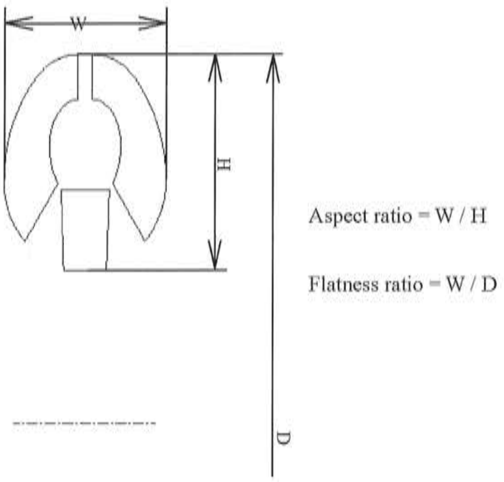

Fig. 1 Definitions of Flatness Ratio and Aspect Ratio

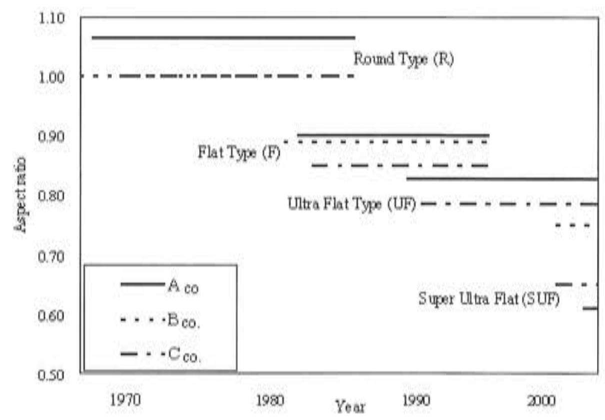

Fig. 2 Trend of Aspect Ratio

2.2. 扁平化へのトライアル

流路外径と各要素出入口角度を一定にして扁平化すると，Fig. 3に示す解析例のように，トルク比とトルク容量係数がともに低下する傾向となる．流体解析には市販コードであるSTAR-CDを用いた．乱流モデルは標準k-εモデル，スキームは一次風上差分である．各要素1ピッチずつであり，メッシュはヘキサで個数は約8万である．この解析手法は，他の文献(2)(3)と照らし合わせてみても，遜色なく，適切であると判断する．実際にTCの扁平率が流体性能にどのような影響を及ぼすかが報告(2)され，同様の傾向となっている．

2.2. Efforts to achieve a smaller aspect ratio

Figure 3 shows an example of a hydrodynamic analysis in which the aspect ratio was reduced while keeping the outer diameter of the flow passage and the inlet/outlet angles of each element constant. It is seen that both the torque ratio and the torque capacity coefficient tend to decline with a smaller aspect ratio. The calculations were performed with the commercially available STAR-CD code. The standard kappa-epsilon model was used as the turbulence model, and a first-order upwind differencing scheme was used to discretize the values. Each element was modeled with a mesh pitch of 1 mm, using a hexa mesh and approximately 80,000 grid cells in total. This analysis method is believed to be suitable and compares favorably with other methods described in the literature.[2-3] The actual influence of the aspect ratio on torque converter performance has been reported[2] and the same tendencies were seen in the present analysis.

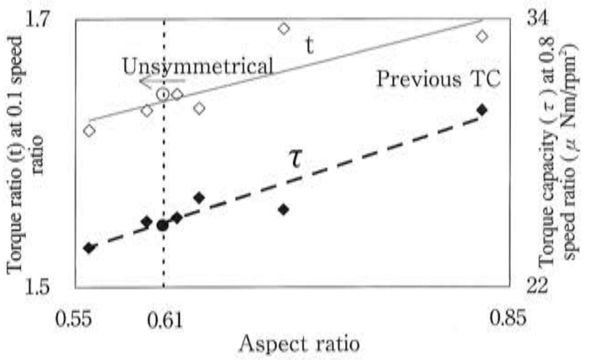

Fig. 3 TC Performance versus Aspect Ratio

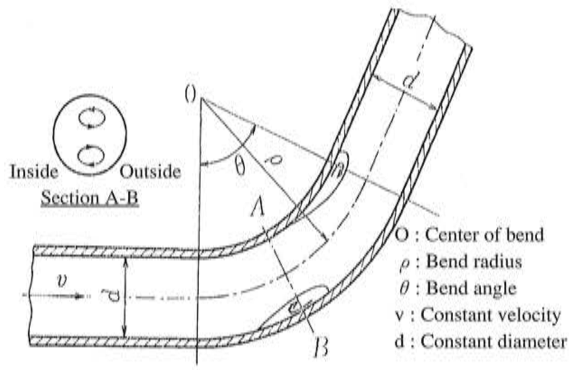

Fig. 4 Loss at Bend

性能低下率を抑えながら極力扁平化させるため，性能低下の限界から扁平率の限界を求めた結果，ポンプ，タービンを対称的に扁平化した場合の扁平率を0.62と決定した．さらに非対称に扁平化してみると，タービンランナのみ扁平率を0.60相当に扁平化しても，性能の低下が殆どないことが解った．以上より，全体としては扁平率0.61として開発を進めることにした．

2.3. 扁平トルクコンバータの性能向上

扁平化により低下した流体性能を改善するために，性能低下の原因を考察した．流路の曲率が大きくなることにより，曲りによる損失すなわち二次流れによる剥離が増大することが最大の原因であると推定し(Fig. 4)，この対策に主眼を置いた．三次元粘性流れモデルで検証し，各設計パラメータを適正化し，最終的に実験で確認した．流路面積，流路曲がり，羽根角度の分布などを単独に変化させ，その変化分をもとにして，所望の扁平率を持つTCの流路面積，羽根角度分布等を求めた．

2.4. ポンプインペラ出口面積比

曲がりによる損失を考慮し，ポンプインペラ出口流路総面積を変化させたモデルを作成し，計算した．流路外径Dのなす面積($D^2/4\pi$)に対するポンプインペラ出口流路面積の比をポンプインペラ出口面積比(以下PRとする)とする．ある扁平率以上ではPRが小さいとトルク容量係数は小さくなるが，扁平率がある程度以下の場合，トルク容量係数はPRが小さいときにピークを有することが解った．これは，PRを大きくしても剥離で循環流量が増えず，逆にPRを小さくすれば流路平均径が大きくなり，コア部の曲がりが小さくなって剥離しにくくなるためである．

Fig. 5の解析結果に示すように，PRを小さくすると剥離が少なくなることが解る．その他の性能もPRによって性能が変化する(Fig. 6)ため，PRはトータルバランスで決めた．

The limit of the aspect ratio was found from the acceptable limit of the performance reduction, with the aim of achieving the smallest possible aspect ratio while suppressing the degree of decline in torque converter performance. As a result, an aspect ratio of 0.62 was selected when the aspect ratios of the pump and turbine were reduced symmetrically. Additionally, in a study of an asymmetrical reduction of the aspect ratios, it was confirmed that there was virtually no decline in performance when the turbine runners alone were flattened to correspond to an aspect ratio of 0.60. Based on these results, it was decided to develop the new torque converter with an overall aspect ratio of 0.61.

2.3. Improvement of performance of a flatter torque converter

The factors causing a decline in performance due to a smaller aspect ratio were investigated in order to improve the hydrodynamic performance of a flatter torque converter. It was hypothesized that the biggest factor was an increase in losses at bends, i.e., separation caused by 2-D flow, due to the larger curvature of the flow passage (Fig. 4). An effort was made to optimize each design parameter, focusing on measures for addressing that factor. The parameters were first examined using a 3-D viscous flow analysis model and the optimized parameters were then verified experimentally. The flow passage area, bend curvature, blade angle distribution and other parameters were varied individually. Then, based on the amount of change observed, the flow passage area, blade angle distribution and other design parameters were determined for a torque converter having the desired aspect ratio.

2.4. Area ratio at pump impeller outlet

A model was created and calculated in which the total flow passage area at the pump impeller outlet was changed, taking into account losses caused by bends. The area ratio at the pump impeller outlet (PR) was defined as the ratio of the flow passage area at the pump impeller outlet to the area formed by the outside diameter (D) of the flow passage ($D^2/4\pi$). It was found that above a certain aspect ratio, the torque capacity coefficient decreases when PR is small;

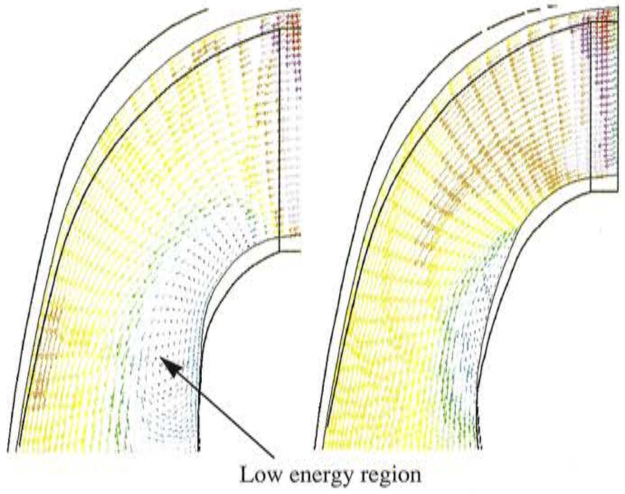

Fig. 5 CFD Results

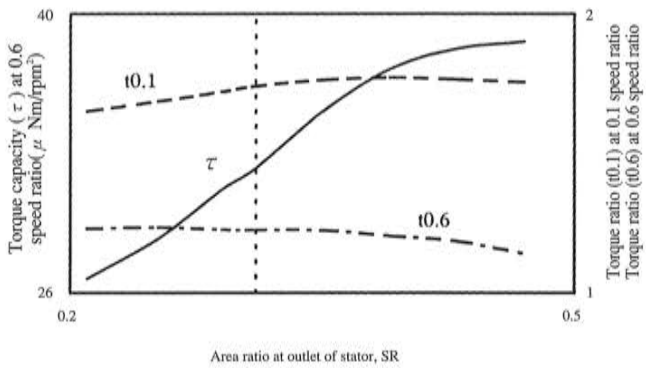

Fig. 7 TC Performance versus SR

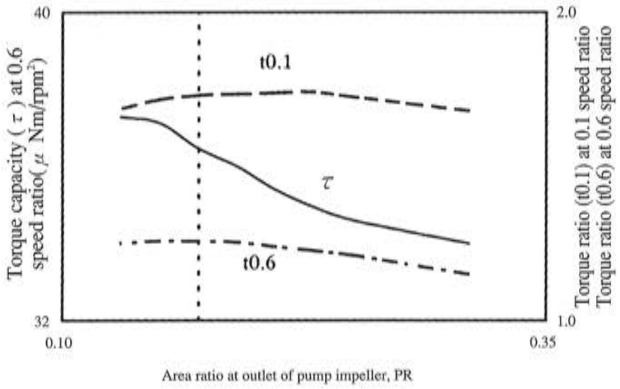

Fig. 6 TC Performance versus PR

2.5. ステータ面積比

ステータについてもPRと同様に流路面積を変化させたモデルを作成し計算した．流路外径面積に対するステータ出口流路面積の比をステータ出口面積比(以下SRとする)とする．トルク容量係数やストールトルク比は，SRを大きくすると増加を続ける(Fig. 7)．一方，効率は低下するため，SRを適当な値に設定する必要がある．これらの変化は，循環流量と半径比の影響によるものである．また，ポンプインペラ出口に比べステータは翼間が狭く，SRを大きくしても曲りによる二次流れ損失は大きくならないと考えられる．

however, provided that the aspect ratio is somewhat below that level, the torque capacity coefficient shows a peak value when PR is small. The reason for that can be understood as follows. Circulatory flow does not increase due to separation even if PR is increased; reducing PR, on the other hand, increases the mean diameter of the flow passage and the bend in the core section becomes smaller, making it less likely that separation will occur.

The CFD results presented in Fig. 5 indicate that separation is reduced when PR is made smaller. Since other performance parameters also change depending on PR (Fig. 6), the value of PR was determined by considering the overall performance balance.

2.5. Area ratio at stator outlet

A model of the stator with a changed flow passage area was created and calculated in the same way as for PR. The area ratio at the stator outlet (SR) was defined as the ratio of the flow passage area at the stator outlet to the area formed by the outside diameter of the flow passage. It was found that the torque capacity coefficient and stall torque ratio continue to increase with a larger SR (Fig. 7). On the other hand, because efficiency declines, SR must be set at a suitable value. These changes are influenced by circulatory flow and the radius ratio. Additionally, as the stator blade spacing is narrower compared with that at the pump impeller outlet, it is thought that secondary flow loss due to a bend will not increase even if SR is made larger.

2.6. 翼倒れ

扁平化して流路の曲率が大きくなると，二次流れが増大すると考えられるので，二次流れ抑制のため翼倒れを変化させたモデルを作成し，計算した．ここでは，ポンプインペラ翼において，シェル側とコア側の翼の中央部同士を結ぶ直線とTC中心軸とのなす角を倒れ角とする．翼倒れの変化による解析結果をFig. 8に示す．翼を倒すことにより二次流れが抑制されていることが解る．しかし，倒し過ぎると翼表面積の増大による摩擦損失の増大により性能が低下するため，適正値を求めた．代表特性として速度比0.6におけるトルク比の変化をFig. 9に示す．

2.6. Blade inclination

It was estimated that reducing the aspect ratio and increasing the curvature of the flow passage might result in greater secondary flow. Accordingly, a model with a changed blade bias angle was created and calculated in an effort to suppress secondary flow. The bias angle was defined here as the angle formed by a straight line connecting the centers of the pump impeller blades on the shell and core sides and the center axis of the torque converter. The CFD results obtained for a change in the blade bias angle are shown in Fig. 8. The results indicate that inclining the blade suppresses secondary flow. However, excessive blade inclination increases friction losses due to the larger blade surface area, causing performance to decline. Consequently, a suitable bias angle was determined. As a representative performance characteristic, the change in the torque ratio at a speed ratio of 0.6 is shown in Fig. 9.

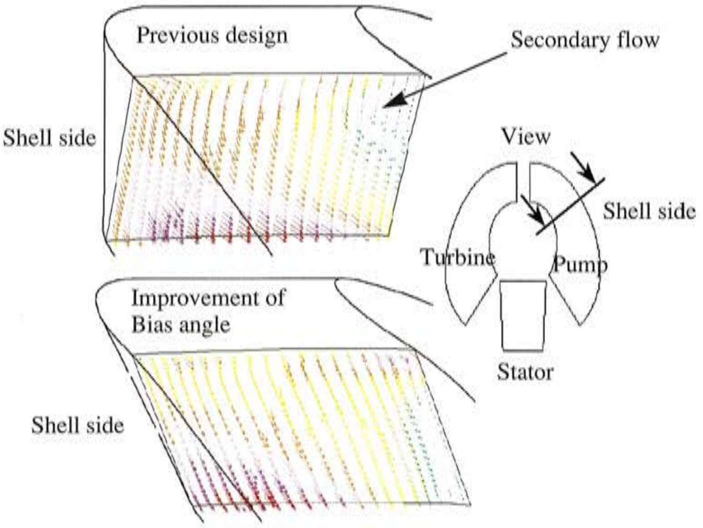

Fig. 8 CFD Results

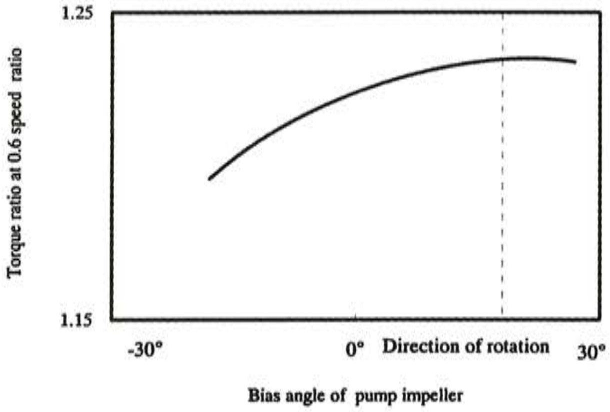

Fig. 9 TC Performance versus Bias Angle

2.7. ステータ軸長

扁平化に伴い，ステータの軸長を短くする必要がでてくる．軸長を短くすると、ステータ翼表面積が減少し、摩擦損失の低減により循環流量が増大して性能が向上するが，短過ぎるとスリップ角が増大して，回転方向の流速変換量が減少し，性能が悪化する．そこで、軸長の適正値を求めた(Fig. 10)．

2.7. Stator axial length

With a smaller aspect ratio, it becomes necessary to shorten the stator's axial length. Shortening the axial length reduces the stator blade surface area, thereby decreasing friction losses, which increases circulatory flow to improve torque converter performance. However, an excessively short axial length increases the slip angle, causing the velocity conversion factor in the direction of rotation to decrease, which degrades performance. Accordingly, an optimum axial length was determined (Fig. 10).

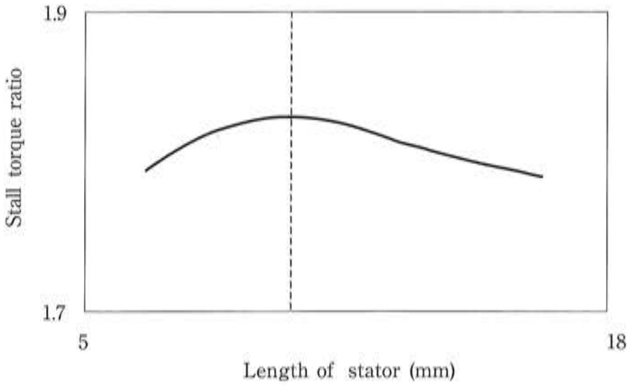

Fig. 10 TC Performance versus Length of Stator

2.8. その他の要因

その他の設計パラメータとして以下を検討した．

・各要素の入口・出口角
・翼負荷分布
・シェルーコアの翼角分布

流体解析により，これらの設計パラメータを適正化した．要因毎に性能変化ポテンシャルを比較し，Fig. 11に示す．ここでは，ポンプインペラの出口角の変更で得られたTC性能を等ポテンシャルとする．解析で得られた最適諸元のTCを実機確認した結果，現行品を26％扁平化した超扁平品（扁平率0.83→0.61）でも，従来TC並みの性能ポテンシャルが得られた．最適諸元のTCの実測性能を従来品と比較し，Fig. 12に示す．

今回のTCは，無段変速機に適用しているため，発進性能を重視して、低速度比域におけるトルク比を高くし、トルク容量係数を低くしている．今回のものは、性能ポテンシャルで比較しても，同等以上となっていることが解る（Fig. 13）．

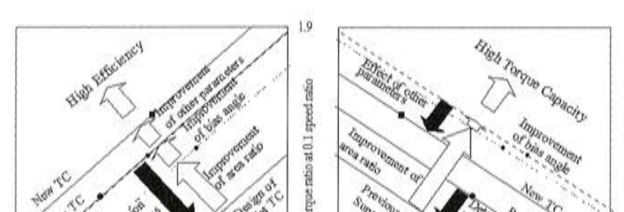

Fig. 11 Improvement of TC Performance

2.8. Other factors

Other design factors that were examined included:

- inlet/outlet angles of each element,
- blade load distribution, and
- blade angle distribution from the shell to the core.

These design parameters were optimized on the basis of the CFD results. Figure 11 shows a comparison of each factor's potential for changing torque converter performance. The performance obtained by changing the outlet angle of the pump impeller is regarded here as the equivalent potential. The optimum specifications obtained in the CFD analysis were used to build a super ultraflat torque converter, having an aspect ratio of 0.61, or 26% smaller than the previous aspect ratio of 0.83. Test results confirmed that this new super ultraflat unit has a performance potential equal to that of the previous torque converter. The measured performance of the new unit and previous torque converter are compared in Fig. 12.

Because the new torque converter is intended for use with a CVT, emphasis was placed on start-off acceleration performance. In this connection, its torque ratio in the low speed ratio range was increased and its torque capacity coefficient was lowered. A comparison with the previous torque converter shows that the performance potential of the super ultraflat unit is equal to or even better than that of the former unit (Fig. 13).

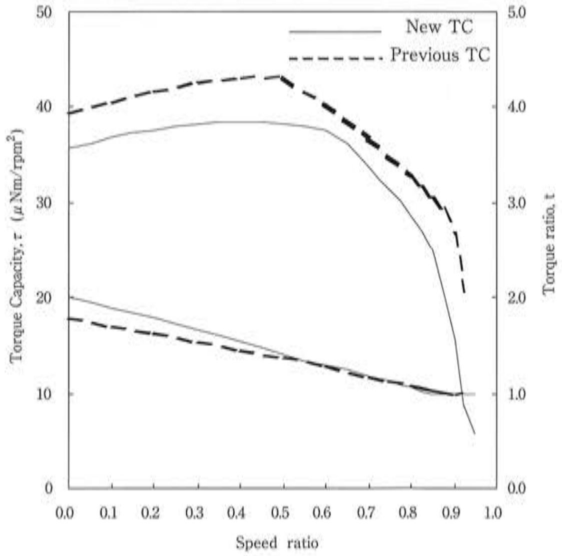

Fig. 12 Performance of Previous and New TCs

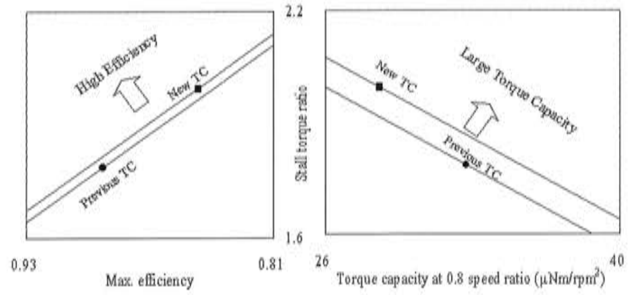

Fig. 13 Performance of Previous and New TCs

3. トルクコンバータの開発

従来のダンパは，スプリングが内周にあるタイプであったが，タービンランナの最も膨らんでいる位置とスプリングの位置が近いために，軸方向寸度面で不利となっていた.

今回は，スプリングを極力外周に配置することにより，タービンランナの膨らみとスプリングが干渉しないようにして，軸方向寸度を短縮した.

以上より，新流体要素とダンパスプリング外周配置LUを組み合わせた薄型TCを開発した．従来のF04B用TCに対し，TC全長を約21％低減することができた(Fig. 14).

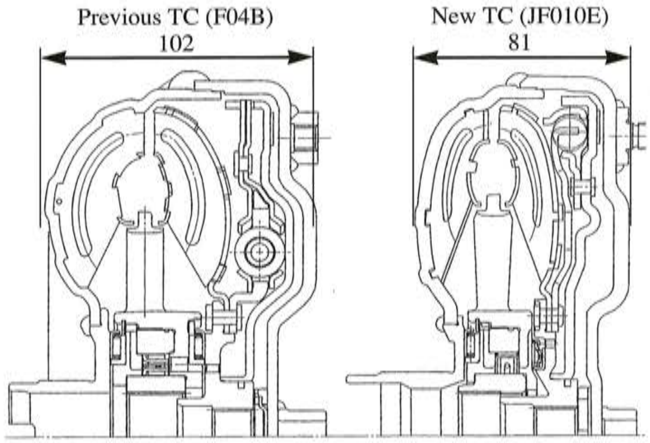

Fig. 14 Cross Sections of Previous and New TCs

4. おわりに

TCは，車両の動力性能や燃費性能に直接寄与する機能部品であるとともに，スペースや重量の占める割合も大きいので，TC性能と搭載性のトレードオフが重要な課題である.

3. Torque Converter Development

The previous torque converter was of the type with the damper spring of the lockup clutch on the inner perimeter. Because the position of the spring was near the largest bulge of the turbine runners, the previous type was at a disadvantage with regard to its axial length.

Every effort was made to position the spring on the outer perimeter of the new torque converter so as to avoid interference with the bulge of the turbine runners and allow the axial length to be shortened.

The new super ultraflat torque converter was thus developed by combining the new hydrodynamic elements with a lockup clutch having the damper spring located on the outer perimeter. Compared with the previous unit for the F04B automatic transmission, the overall length of the new torque converter has been reduced by approximately 21% (Fig. 14).

4. Conclusion

A torque converter is a functional component that contributes directly to the power performance and fuel economy of a vehicle. Moreover, because it accounts for a large proportion of the space and weight allocated for the drivetrain system, there is a critical trade-off between its hydrodynamic performance and vehicle mountability.

In developing a new torque converter that has a different geometry from previous units, it is not practical to build and test prototype units repeatedly because of the enormous cost and time involved. The use of numerical analysis, as was done in the present case, makes possible the quick development of challenging products that resolve the above-mentioned trade-off, although there may be some question concerning accuracy.

In the present development project, emphasis was put on shortening the axial length while still maintaining the desired hydrodynamic performance. In future work, this type of CFD approach will be applied vigorously to develop even more compact torque converters without being limited by previous performance considerations.

従来品に対して形状の異なるTCを開発する場合，試作・実験の繰り返しでは費用・期間ともに莫大であり，現実的ではない．今回のように、数値解析であれば，精度面での課題はあるものの，上記のトレードオフの課題を解決した挑戦的な製品を短期に開発することができる．

今回の開発では，性能を維持しつつ軸方向の短縮に重きをおいたが，今後は従来性能にとらわれない更なる小型化の開発にも，このような手法を駆使して積極的に取り組みたい．

最後に，本TC開発にあたりご協力いただいた日産自動車(株)パワートレイン開発本部ドライブトレイン開発部の皆様をはじめ、関係者の方々に深く感謝の意を表する．

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Finally, the authors would like to thank everyone involved for their cooperation in the development of this new torque converter, especially the people in the Drivetrain Engineering Department of the Powertrain Engineering Division at Nissan Motor Co., Ltd.

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特　集

大容量CVT油圧＆電子制御の紹介

Hydraulic System, Shift and Lockup Clutch Controls Developed for the Large Torque Capacity CVT3

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抄　録　3.5リッターエンジンへの適用を可能にした新世代ベルト式CVT（以下，CVT3と呼称）は，2002年11月，日産自動車（株）様の『Murano』に搭載され，その燃費・運転性の飛躍的な向上に貢献した．

本稿では，このCVT3の実現に向けて開発してきた油圧，変速，ロックアップに関する各種新制御，およびその性能ついて紹介する．

Summary　A new-generation steel-belt CVT (CVT3) for use with engines up to the 3.5-liter class was developed and fitted on the Nissan Murano in November 2002. The CVT contributes to improvements in both fuel economy and driving performance. This article describes the new hydraulic pressure, pulley ratio (shift) and lockup clutch controls developed for the CVT3 and the performance levels obtained.

1. はじめに

近年，燃費向上に対する要求は益々増大してきており，自動車メーカ各社のCVTへの期待は極めて大きくなっている．

ジヤトコは，長年のCVT開発や量産で培った技術ノウハウを基盤に，燃費と運転性能の両方を高いレベルで改善し，かつ大排気量エンジンへの適用が可能なCVT3を新開発した．

本稿では，CVT3の制御技術のうち，基本となる以下の三つの制御内容について，現行のCVT（以下，CK2と呼称）と対比しながらその概要を説明する．

(1) 油圧制御（油圧回路構成）
(2) プーリ比（変速）制御
(3) ロックアップ制御

2. 油圧制御（油圧回路構成）

油圧制御回路の全体構成としては，CK2と同様，使用油圧の高い方から，プーリ圧系，クラッチ圧系，トルクコンバータ圧系の順に階層的に配置した（Fig. 1）．このように配置すれば，各部の油圧リーク量を極力低減することができる．

この油圧制御回路の構成を基本とし，更に応答性，精度，信頼性などの性能向上技術を織り込んだ．以下にその性能向上技術の具体例を解説する．

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Functional Component Development Department No.1
** 商品企画統括部
Product Planning Administration Department

1. Introduction

Demands for fuel economy improvements have been continually increasing in recent years, and many automakers have strong expectations of continuously variable transmissions (CVTs) in this regard. At JATCO, we have newly developed the CVT3 on the basis of the technologies and expertise accumulated through many years of experience in developing and mass-producing CVTs. Besides improving both fuel economy and driveability, this new-generation steel-belt CVT is also applicable to large-displacement engines.

This paper describes the hydraulic pressure control (hydraulic circuit configuration), pulley ratio (shift) control and lockup clutch control developed for the CVT3. These three controls that constitute the CVT3's fundamental control technology are explained in comparison with those of the previous CK2 CVT.

2. Hydraulic Pressure Control (Hydraulic Circuit Configuration)

A schematic diagram of the overall hydraulic pressure control circuit is shown in Fig. 1. Like the CK2 unit, the pulley pressure system, clutch pressure system and torque converter pressure system are arranged in hierarchical order from the highest to the lowest pressure used. This arrangement minimizes the amount of pressure leakage in each system.

With this structure as the foundation of the hydraulic pressure control circuit, various technologies were incorporated to improve response, accuracy, reliability and other performance parameters further. The following sections present specific examples of the technologies adopted to improve the performance of the CVT3.

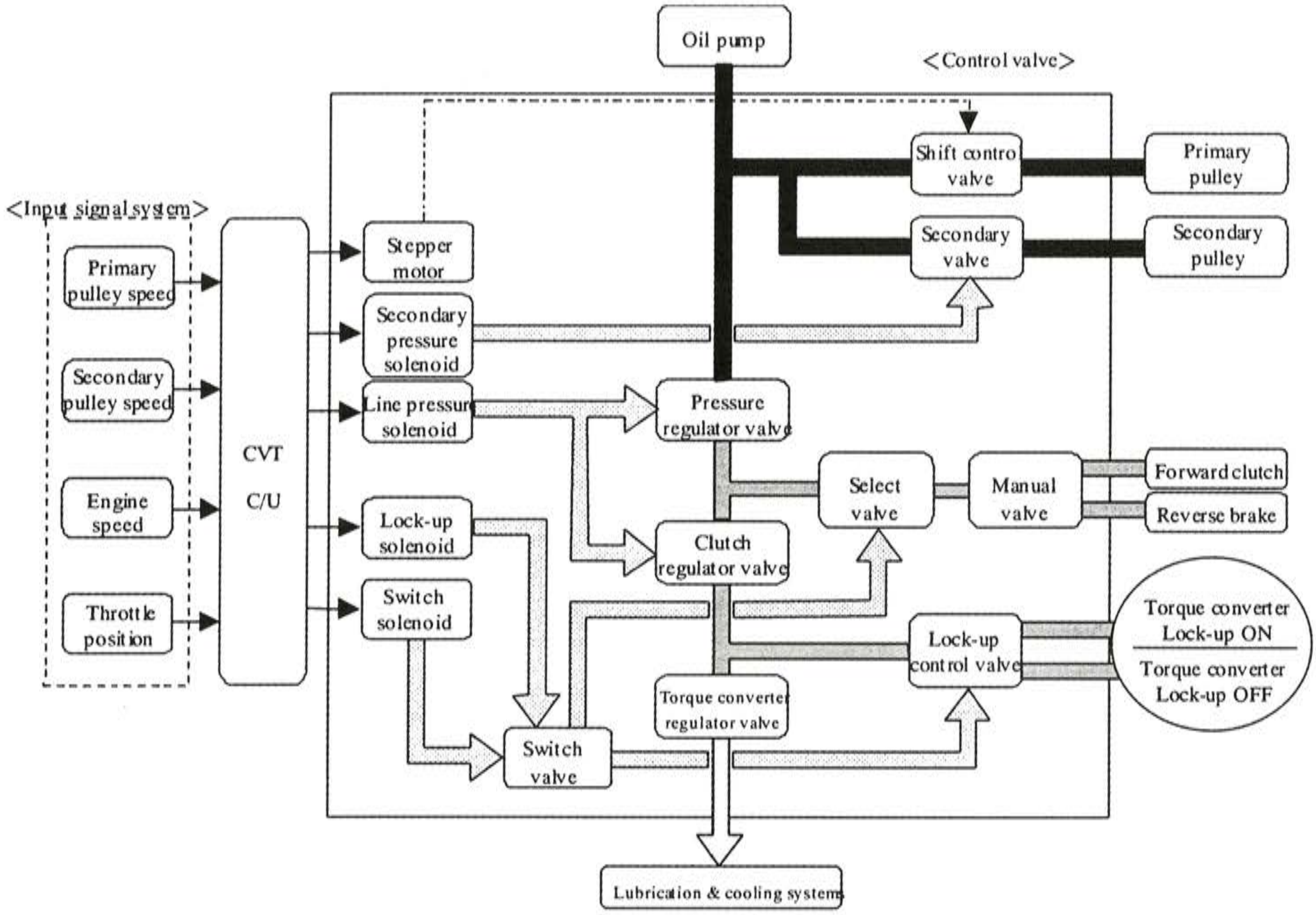


Fig. 1 Schematic diagram of the hydraulic circuit

2.1. セカンダリプーリ圧減圧機構

一般に，プーリ比(変速比)は，セカンダリプーリに対するプライマリプーリのクランプ力比によって制御する．具体的には，変速比が『LOW』の状態は，クランプ力比を小さく，すなわち，プライマリプーリへの供給油圧をセカンダリプーリへの供給油圧より低くすることで実現する．一方，変速比が『HIGH』の状態は，クランプ力比を大きくする必要があるため，プライマリプーリ油圧を高くする．この際，セカンダリプーリへの供給油圧を低くできれば，それとバランスするプライマリ油圧，および，全ての油圧の元圧であるライン圧をも低くすることができ，効率向上の観点からは望ましいことになる．

これに対し，CK2では，変速比の全域で，セカンダリプーリ供給油圧がプライマリプーリ供給油圧より高い設定にしかできない油圧回路構成となっているため，全体の油圧が必要以上に高くなる条件が存在していた．

CVT3では，Fig. 2に示すセカンダリ減圧機構を導入することにより，この課題を解決し，伝達効率を大幅に向上させた．

2.1. Secondary pulley pressure reducing mechanism

The pulley (shift) ratio is generally controlled by managing the ratio of the primary pulley clamping force to that of the secondary pulley. Specifically, the Low shift ratio state is achieved by reducing the clamping force ratio so that the hydraulic pressure supplied to the primary pulley is lower than that supplied to the secondary pulley. On the other hand, the clamping force ratio must be increased to achieve the High shift ratio state, which means raising the hydraulic pressure of the primary pulley. From the standpoint of improving efficiency, it is desirable to reduce the hydraulic pressure supplied to the secondary pulley at that time, because that allows a lower primary pulley hydraulic pressure, which is balanced with that of the secondary pulley. It also allows a lower line pressure, which is the source of all the pressure levels in the transmission.

The hydraulic circuit configuration of the CK2 CVT only allowed the hydraulic pressure of the secondary pulley to be set higher than that of the primary pulley in all shift ratio ranges. That condition meant that the overall hydraulic pressure was set at a higher level than was necessary.

As shown in Fig. 2, a secondary pulley pressure reducing mechanism has been incorporated in the hydraulic circuit of the CVT3 to resolve this issue, and transmission efficiency has been significantly improved as a result.

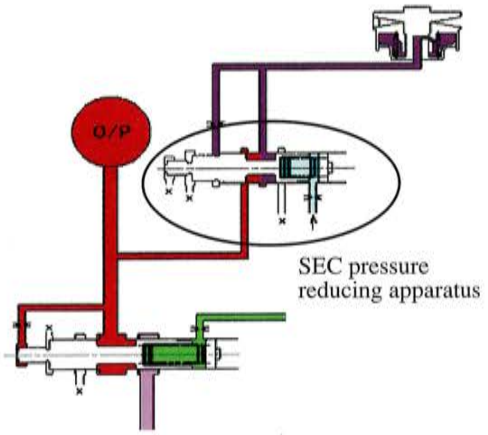


Fig. 2 Secondary Pressure Reducing Circuit :
○ is attached to the addition part newly

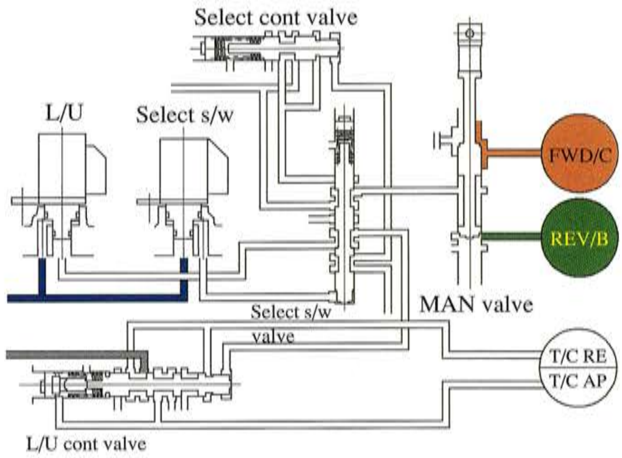


Fig. 3 Engagement Circuit

2.2. クラッチ圧の制御自由度の増大

CK2では，前・後進をセレクトする際のクラッチ供給油圧を制御するソレノイドは，セカンダリプーリ圧を制御するソレノイドと兼用しており，セレクト時の油圧設定の自由度が制約されていた．CVT3では，この制約を回避するため，ロックアップ用ソレノイドと兼用することとし(Fig. 3)，クラッチ圧の設定自由度を確保した．これにより，セレクト用アキュムレータを廃止することができ，本体構造の小型化にも寄与した．

2.3. 流量制御弁付大容量対応オイルポンプ

オイルポンプとしては，高油圧用のクレセントレス内接ギヤ式ポンプを，軽量化を目的としたアルミ材にて新開発した．吐出回路には流量制御弁を設け，高回転になってオイル吐出量が一定値以上になるとオイルをポンプ吸入側に戻すことにした．これにより，プレッシャレギュレータバルブへの供給オイル量が規制されて，油圧制御の精度が向上するとともに，メインストレーナの吸入流量が制限され，ポンプキャビテーションの発生を防止することができた．同時にポンプ効率の向上も実現した．

2.4. 2方リニアソレノイド

2.4.1. 油圧キャリブレーションシステム

油圧制御用アクチュエータとしては，従来用いてきたPWM式3方向ソレノイドに代えて，当社製5速AT(RXO)で実績のある2方向リニアソレノイドを採用した．また，このATと同様に，各油圧系統の油圧特性データをトランスミッションに内蔵したメモリに記憶し制御に利用するシステム(油圧キャリブレーションシステム)を採用した．これにより，ソレノイド単体としての精度に依存せずに，システム全体として高い油圧精度を確保した．

2.2. Increased degree of freedom for clutch pressure control

In the CK2, the solenoid for controlling the clutch pressure when selecting forward/reverse also doubled as the solenoid for controlling the secondary pulley pressure. That arrangement restricted the degree of freedom for setting the hydraulic pressure when making a forward/reverse selection. With the CVT3, the solenoid for controlling the lockup clutch pressure serves this double function (Fig. 3), thereby avoiding this restriction and securing an increased degree of freedom for setting the clutch pressure. As a result, that made it possible to eliminate the accumulator for forward/reverse selection, which helped to reduce the size of the CVT3.

2.3. Large-capacity oil pump with flow rate control valve

A high-pressure crescent-less internal gear pump was newly developed as the oil pump. The pump is made of aluminum to achieve a lighter weight. A flow rate control valve has been incorporated in the discharge circuit. When the oil flow rate reaches a certain level at high operating speeds, the valve functions to return oil to the suction side of the pump. This regulates the oil flow rate supplied to pressure regulator valve, which improves hydraulic pressure control accuracy. It also prevents the occurrence of pump cavitation by limiting the intake flow rate at the main strainer. Pump efficiency is also improved at the same time.

2.4. Two-directional linear solenoid

2.4.1. Pressure calibration system

Two-directional linear solenoids with a proven record of performance in our 5-speed RXO AT were adopted as the pressure control actuators. These solenoids replace the three-directional pulse-width modulation (PWM) type that was used previously. Like the ROX AT, the CVT3 adopts a pressure calibration system that stores the pressure characteristic data of each pressure system in a memory incorporated in the transmission for use in

2.4.2. リニアソレノイドを含めた油圧系の開発

一般に，リニアソレノイドの特性は，その負荷に相当する，スプールバルブや油圧流路回路抵抗，流路体積などから非常に大きな影響を受けることが知られている．このため，本開発にあたっては，全てのスプールバルブを含めた油圧回路構成を組込んだシミュレーションを構築，活用することで，開発効率の向上に役立てた．シミュレーション活用の例として，踏み込みダウン変速時の油圧応答に関し，実験値と比較した結果を示す(Fig. 4)．

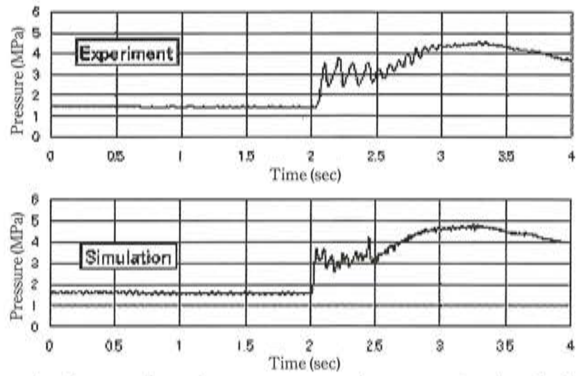


Fig. 4 Comparison between experiment and simulation

3. プーリ比(変速)制御部

プーリ比(変速)制御のハードウェアとしては，ステッピングモータ&油圧サーボリンクから構成されている．この構成を使えば，油圧変動やトルク変動といった外乱に強く，プーリ比を目標値どおりに維持・制御できるメリットがある．また，ソフトウェアとしては，内部モデルでの応答と実際の応答との差分に基づいてフィードバック補正するロバストマッチング制御を採用している(Fig. 5)．CVT3では，CK2で採用してきた上記基本構成を，ハード・ソフトの両面から，大幅に発展させた．

具体的には，プーリピストン面積比の変更に併せ，変速制御弁のテーパ角とオーバーラップ量(流量ゲイン)を再検討し，ハードウェアの応答特性を大幅に改善した(Fig. 6)．さらに，CPUの能力向上，制御ゲインの適正化といったソフト面での開発アイテムを織り込み，制御全体として変速の応答性と安定性の両方を向上させることができた．マニュアルアップシフトのプーリ比制御性能を比較すると，変速後のオーバーシュート量が改善されていることが分かる(Fig. 7)．

control operations. This system achieves high pressure accuracy for the entire unit, instead of relying on the accuracy of individual solenoids.

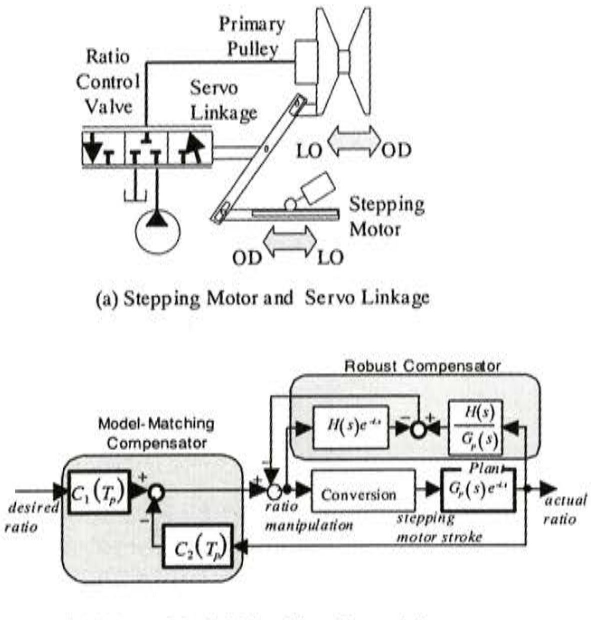
2.4.2. Development of the hydraulic system, including linear solenoids

It is well known that the characteristics of linear solenoids are greatly influenced by the spool valves, resistance in the hydraulic fluid passages, fluid passage volume and other properties, corresponding to the applied load. In this development project, a simulation model of the hydraulic system was constructed that included all the spool valves. The use of this model was helpful in improving development work efficiency. As one example of the application of the model, Fig. 4 compares the experimental and simulation results for the pressure response in a downshift accompanying the driver's depression of the accelerator pedal.

3. Pulley Ratio (Shift) Control

The hardware for pulley ratio (shift) control consists of a stepping motor and a hydraulic servo linkage. This configuration has the advantage of allowing the pulley ratio to be maintained and controlled according to the target value because of its robustness against disturbances such as pressure and torque fluctuations. As the software, a robust model-matching control method was adopted that provides feedback compensation based on the difference between the actual response and the response of an internal model (Fig. 5). This basic configuration of the shift control system is the same as that of the CK2 unit, but substantial improvements were made to both the hardware and software for the CVT3.

Specifically, the response characteristics of the hardware were significantly improved by revising the taper angle and amount of overlap (flow rate gain) of the shift control valve together with changing the area ratio of the pulley pistons (Fig. 6). The development objectives for the software included improvement of the capacity of the CPU and control gain optimization. These improvements made to the overall control system have enhanced both shift response and stability. A comparison of the CVT3 and CK2 in terms of their pulley ratio control performance in a manual 2-3 upshift shows that the new unit greatly reduces the amount of overshoot at the completion of the shift (Fig. 7).



(a) Stepping Motor and Servo Linkage

(b) Robust Model-Matching Control System

Fig. 5 Configuration of Shift Control System

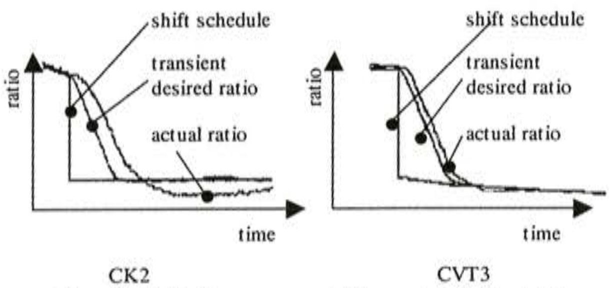


Fig. 7 Shift Response at Manual 2-3 Upshift

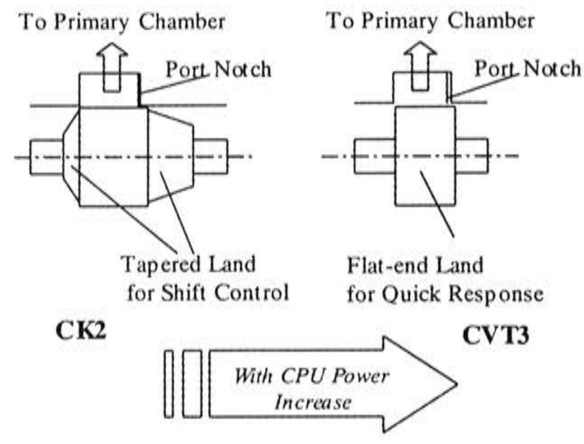


Fig. 6 Shift Control Valve

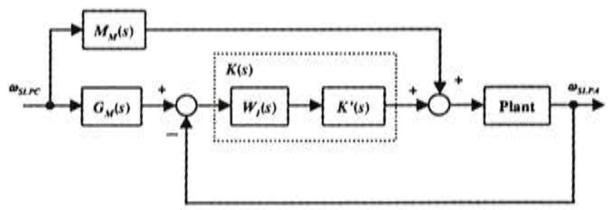


Fig. 8 Two-Degree-of-Freedom Control System

4. ロックアップ制御

ロックアップ領域を低速まで延長すると，燃費改善効果は大きくなるが，こもり音に加え，エンジンの回転数の急変に起因する不快感が副作用として問題視されることが多い．今回は，極低車速（概略20km/h以下）でロックアップさせて燃費を改善するとともに，エンジン回転の急変による不快感を克服するためロックアップ制御ロジックを大幅に改良した．発進時に滑らかで，かつエンジン回転数の変化をドライバの感性に合ったものとするため，ロックアップの開始指令車速を極微低速に設定するとともに，発進時専用のスリップロックアップ制御を採用した．

具体的には，応答性を設定する前置補償器と，規範モデルからの偏差値を修正するフィードバック補償器とを備えた2自由度制御系（Fig. 8）とし，制御対象であるロックアップ機構をLPV（Linear Parameter Varying）システムとして扱うことで，低次元の数学モデルで処理することができた．

4. Lock-up Clutch Control

Extending lockup operation to a lower speed range has a large effect on improving fuel economy, but it also entails certain side effects that are usually regarded as undesirable such as booming noise and an unpleasant sensation due to a sudden decrease in engine speed. For the CVT3, lockup operation is activated at an ultra-low vehicle speed of approximately 20 km/h to improve fuel economy. In addition, the unpleasant sensation due to the abrupt decline in engine speed has been overcome by substantially improving the lockup control logic. Start-off smoothness has been achieved along with a change in engine speed that matches the driver's sensibilities. That was accomplished by setting the vehicle speed for the onset of lockup operation at a very low level and by adopting a slip lockup control designed specifically for vehicle launch.

In more concrete terms, a two-degree-of-freedom control system (Fig. 8) was adopted that incorporates two compensators. One is a forward-positioned compensator for setting the response and the other is a feedback compensator for correcting the deviation from the reference model. The plant, i.e., the lockup clutch, is treated as a linear parameter varying (LPV) system, allowing it to be processed as a low-order mathematical model.

Fig. 9に，本制御の効果として，スロットル開度1/32における発進時の実験結果を示す．この結果からも，所定のタービン回転になるまでスリップ回転を継続制御することで，エンジン回転の大きな変化を伴うことなく，車速20km/h以下で完全ロックアップを達成できていることが分かる．

To illustrate the effects of this control system, Fig. 9 shows the experimental results obtained for vehicle start-off at a 1/32 throttle valve opening. The results indicate that the slip speed was continuously controlled until a specified turbine speed was reached. As a result, full lockup operation was achieved at a vehicle speed of less than 20 km/h without any large decline in engine speed.

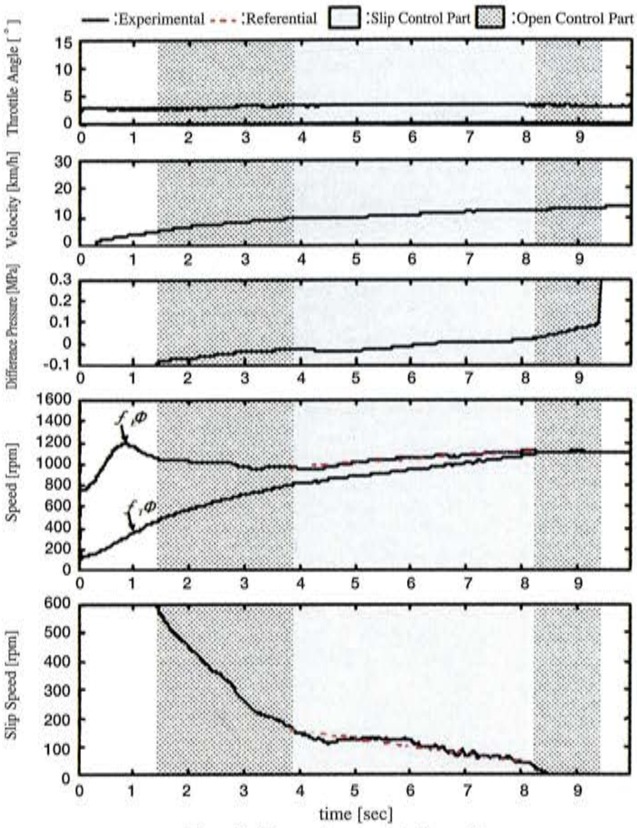


Fig. 9 Experimental Result

5. おわりに

CVT3で新規に開発，採用した油圧&電子制御技術について紹介した．これらの技術は，CVT3の燃費・運転性といった商品力の向上に大きく貢献したものと考える．なお，本制御の開発に当たり，多大なるご協力を頂いた開発メンバーをはじめ，社内外の方々に厚く感謝申し上げる．特に，今回採用した新ロックアップ制御の開発に際しては，日産自動車(株)電子電装システム開発部の安達主担殿，瀬川殿には多大なるご協力を頂いた．心より感謝申し上げる．

5. Conclusion

This article has described the hydraulic and electronic control technologies that were newly developed and adopted for the large torque capacity CVT3. These technologies contribute to substantial improvements in the attractiveness of the CVT3 in terms of fuel economy and driveability.

The authors would like to thank the development team members as well as various people inside and outside the company for their tremendous cooperation in connection with the development of these control technologies. Special thanks are due Messrs. K. Adachi and S. Segawa of the Electronics System Engineering Department at Nissan Motor Co., Ltd. for their invaluable cooperation concerning the development of the new lockup clutch control that was adopted for the CVT3.

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特 集

新ベルトCVTフルードの開発

Development of a New Steel-belt CVT Fluid

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荒川 慶江* Yoshie ARAKAWA

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抄 録 燃費や変速性能の向上といった市場の要求に対応するため，プッシュベルト式無段変速機（以下ベルトCVT）を採用する車種が増加している．これに伴い，ベルトCVTにはトルク容量の増大，使用温度範囲の拡大など多岐にわたる性能の向上が要求されてきており，CVTフルードにもこれに対応した性能向上が必要となった．本稿では，新たに開発したCVTフルードの性能について報告する．

Summary Push-belt continuously variable transmissions (CVTs) have increasingly been adopted on vehicles in order to meet market demands for improved fuel economy and shift performance. Accordingly, these CVTs are required to accommodate higher torque levels and to improve driving performance under a wider range of operating temperatures. A high-performance CVT fluid is needed to enable CVTs to deliver their full performance capabilities. This article describes the characteristics of a newly developed high-performance fluid for steel-belt CVT use.

1. はじめに

ベルトCVTが1987年に実用化されて以来，年々高まる燃費改善や運転性能向上などの市場要求に応えてベルトCVTを採用する車種が増加している[(1)]．また，適用エンジンの排気量も大きくなる傾向にあり，3.5Lエンジンに適用可能なベルトCVTも実用化された．このような背景のもとに，ベルトCVTの性能を最大限に発揮させるには高性能なベルトCVTフルードが必要となった．本稿では新たに開発したベルトCVTフルードの性能について報告する．

2. 開発の狙い

CVTを採用する車種の増加に伴い高トルクエンジンを搭載した車にもベルトCVTが採用されるとともに，日本や欧州に加えアラスカ，カナダなどの寒冷地を含む北米市場の車にもベルトCVTが搭載されてきている．このような使用環境の拡大に伴い，ベルトCVTには極寒地での燃費向上や運転性の確保に対応できる性能が必要となる．また，サービス性向上や地球環境保護の観点からFill for life化や高トルクエンジンへの対応も要求されている．

Table 1にベルトCVTフルードに要求される性能項目を示す．上記の要求を考慮し，開発油は特に以下の性能向上を目指した．

1. Introduction

A belt-drive CVT was first used on a production vehicle in 1987. Since then, an increasing number of car models have been fitted with steel-belt CVTs to meet market demands for improved fuel economy and driving performance, which continue to become more rigorous every year.[(1)] In addition, the displacement of the engines mated to CVTs has also tended to increase. A steel-belt CVT applicable to 3.5-liter engines has been implemented on production models at present. Against this backdrop, it has become necessary to have high-performance fluids for use in steel-belt CVTs in order to elicit their maximum performance. This article describes the performance of a newly developed steel-belt CVT fluid.

2. Development Aims

Along with the use of CVTs on a wider variety of car models, steel-belt CVTs are also being adopted on vehicles fitted with high-torque engines. Moreover, in addition to their use on vehicles in Japan and Europe, steel-belt CVTs are also being fitted on vehicles in the North American market, including cold weather regions like Alaska and Canada. As a result of this expansion of their usage environment, steel-belt CVTs must be capable of improving fuel economy and assuring acceptable driveability in extremely cold weather regions. They are also required to provide fill-for-life capability for the sake of improving serviceability and protecting the global environment and to be capable of accommodating high-torque engines.

Table 1 shows the various performance attributes required of a steel-belt CVT fluid. In view of the requirements mentioned above, the aims set for the development of the new fluid were to improve the following areas of performance in particular.

* 第一機能部品開発部
Functional Component Development Department No. 1

自動車技術会 2003年春季大会前刷集 No.59-03に掲載

(1) 燃費，運転性の向上
- 低温流動性

(2) 高トルクエンジンへの対応
- 高金属間摩擦係数

(3) Fill for life化への対応
- 剪断安定性
- 金属間摩擦係数の持続性
- シャダー防止性
- 摩耗防止性
- 焼付き防止性

Table 1 CVT fluid requirements

Fluids		Requirements		Characteristics
CVTF		Transmission of torque		High M/M* friction coefficient
	ATF	Improved fuel economy	Friction reduction	Lower viscosity
			Weight reduction	High friction coefficient
		Fill for life		Oxidation stability
				Thermal stability
				Shear stability
				M/M* μ stability
				Anti-shudder
			Component durability	Anti-wear property
				Anti-seizure property
				Rolling fatigue protection
				Anti-corrosion, Anti-rust
				Materials compatibility
		Better driveability		Low-temperature fluidity
			Noise and vibration suppression	Friction characteristics

*M/M: Metal-to-metal

3. 性能

3.1. 一般性状

開発油および性能比較に用いた市販油A，B，C，Dの一般性状をTable 2に示す．ここでA，BはCVT専用油，C，DはAT・ベルトCVT兼用油である．

3.2. 低温流動性，剪断安定性

低温時の燃費向上やエンジン始動性等の運転性向上のため，フルードには良好な低温流動性が求められる．また，CVTフルードは油圧の伝達という作動油としての役割を担っており，油圧を維持するため粘度を保持する必要がある．ベルトCVTはベルト-プーリ間でフルードを激しく剪断することから，ベルトCVTフルードとしてはATF以上の剪断安定性が求められる．

Fig. 1に各油の低温流動性と剪断安定性の関係を示す．低温流動性の評価として-40℃でのBrookfield粘度を測定し，剪断安定性の評価にはJASO M347-95[2]で定めている超音波剪断試験を実施した．開発油は良好な低温流動性と剪断安定性を有している．

(1) Improvement of fuel economy and driving performance
- Low-temperature fluidity

(2) Application to high-torque engines
- High metal-to-metal (M/M) friction coefficient

(3) Fill-for-life capability
- Shear stability
- Retention of M/M friction coefficient
- Anti-shudder property
- Anti-wear property
- Anti-seizure property

Table 2 Test fluids

Fluids		Developed CVTF	A	B	C	D
Kinematic viscosity (mm²/s)	40℃	33.8	30.3	33.7	36.6	29.4
	100℃	7.16	6.94	7.6	7.41	7.08
Viscosity index		183	202	205	175	218
Neutralization number (mgKOH/g)	Acid number	1.26	0.53	0.75	1.16	0.89
	Base number	1.13	1.07	2.13	1.76	2.42

3. Performance

3.1. General properties

The general properties of the developed CVT fluid and those of four commercial fluids (A, B, C and D) used in a performance comparison are shown in Table 2. Fluids A and B were dedicated CVT fluids, and fluids C and D were for use in both automatic transmissions (ATs) and steel-belt CVTs.

3.2. Low-temperature fluidity and shear stability

A CVT fluid is required to have good low-temperature fluidity in order to improve vehicle fuel economy and driveability, such as engine startability, at low ambient temperatures. Additionally, because a CVT fluid functions as a working fluid to transmit hydraulic pressure, it must retain its viscosity in order to maintain the desired pressure level. A steel-belt CVT fluid must have higher shear stability than AT fluids because it undergoes severe shearing between the belt and pulleys in this type of transmission.

Figure 1 shows the relationship between the low-temperature fluidity and shear stability of the five fluids examined in this study. Brookfield viscosity at -40℃ was measured to evaluate low-temperature fluidity, and shear stability was evaluated in sonic shear stability tests conducted according to the procedure specified in JASO (Japan Automobile Technologists Standard Organization) M347-95.[2] The results show that the developed CVT fluid possesses good low-temperature fluidity and shear stability.

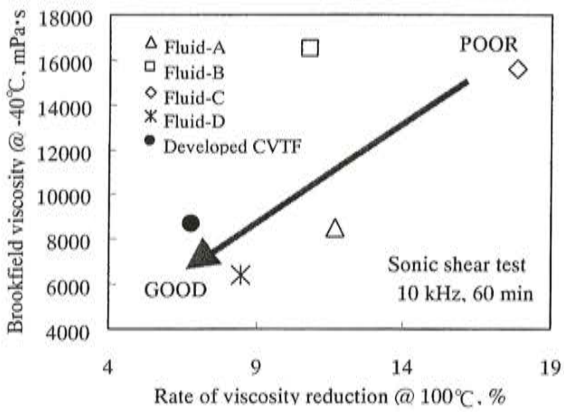

Fig. 1 Viscosity characteristics

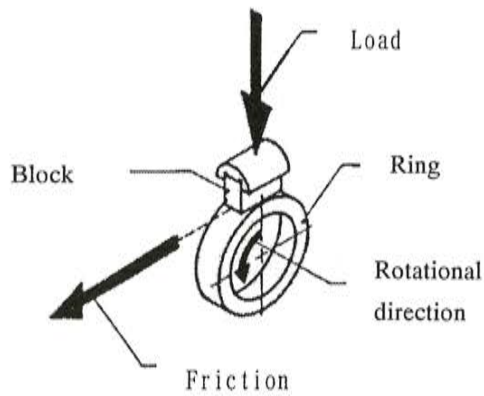

Fig. 2 LFW-1 block and ring

3.3. 金属間摩擦係数の持続性

ベルトCVTフルードのFill for life化には，経時劣化後においても新油時の金属間摩擦係数を維持していることが要求される．

経時劣化後の金属間摩擦係数の変化を評価するため，酸化劣化試験後の金属間摩擦係数を測定した．酸化劣化方法としてはJIS K 2514に定めているISOT試験を油温150℃で実施し，金属間摩擦係数の測定にはASTM D2714-94[3]に記載しているFalex Model 1摩擦摩耗試験機(以下LFW-1)を用いた．試験片の摺動状態をFig. 2に，今回実施した試験条件をTable 3に示す．なお，金属間摩擦係数の代表値として滑り速度0.25m/sでの値を用いた．結果をFig.3に示す．開発油の金属間摩擦係数は酸化劣化試験後でも初期の値を維持している．

Table 3 LFW-1 test conditions

Fluid temperature, ℃	110
Sliding speed, m/s	0.025, 0.075, 0.125, 0.25, 0.5, 1.0
Load, N	1112

3.4. シャダー防止性

ベルトCVT用トルクコンバータでは燃費向上のためロックアップ(以下L/U)クラッチの締結車速が低速化する傾向にあり，クラッチの締結頻度の増加に伴い摺動距離が長くなる．このため，ベルトCVTフルードにはATFと同様にシャダー防止性が求められる．

3.3. Retention of M/M friction coefficient

To provide fill-for-life performance, a steel-belt CVT fluid must maintain the same level of M/M friction coefficient as a fresh fluid even after aging with use. The change in the M/M friction coefficient following aging was evaluated by measuring this characteristic after conducting an oxidation stability test. Oxidation stability was examined by conducting an Indiana Stirring Oxidation Test (ISOT) at a fluid temperature of 150℃ according to the procedure prescribed in JIS K2514. A Falex Model 1 friction and wear testing machine (LFW-1), as described in ASTM D2714-94,[3] was used to measure the M/M friction coefficient. The sliding condition of the test piece is illustrated in Fig. 3, and the LFW-1 test conditions are given in Table 3. The value measured at a sliding speed of 0.25 m/s was used as a representative value of the M/M fiction coefficient. The results measured at this speed are shown in Fig. 3. It is seen that the developed CVT fluid maintained its initial M/M friction coefficient even after the oxidation stability test.

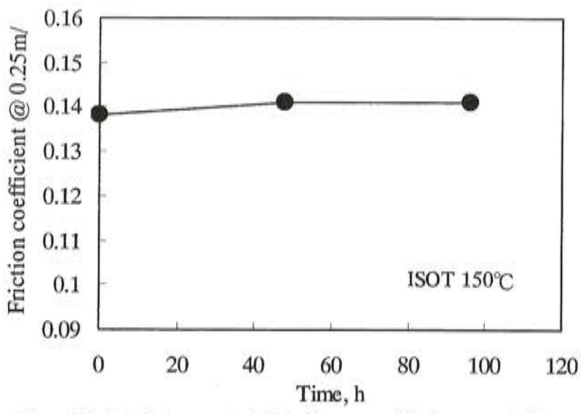

Fig. 3 Metal-to-metal friction coefficient stability

3.4. Anti-shudder property

There is a tendency to engage the lock-up clutch of the torque converter of a steel-belt CVT at lower vehicle speeds these days for the purpose of improving fuel economy. The increased frequency of clutch engagement results in a longer sliding distance. For that reason, a steel-belt CVT fluid must provide anti-shudder performance just like AT fluids.

Fig.4に開発油およびJASO標準油の低速滑り試験(LVFA)でのシャダー防止性評価結果を示す．試験方法としてはJASO M349-98[4]を参照し，開発油の評価には摩擦材として当社CVTのL/Uクラッチ使用材を用いた．また，シャダー寿命の評価にはμ-V特性が負勾配($\mu 1/\mu 50>1$)となる時間を用いた．開発油のシャダー寿命はJASO標準油以上であり，良好な摩擦特性が長期にわたって維持されている．

The anti-shudder properties of the developed CVT fluid and JASO standard fluid were evaluated using a low-velocity friction apparatus (LVFA) and the results are shown in Fig. 4. The test procedure was designed in reference to JASO M349-98[4] and the lock-up clutch friction material of JATCO CVTs was used as the friction material for evaluating the developed CVT fluid. Shudder life was evaluated on the basis of the time until the μ-V characteristic showed a negative slope (μ 1/μ 50 > 1). The developed CVT fluid displayed a longer shudder life than the JASO standard fluid, indicating that it is capable of maintaining an excellent fiction characteristic over a long period of time.

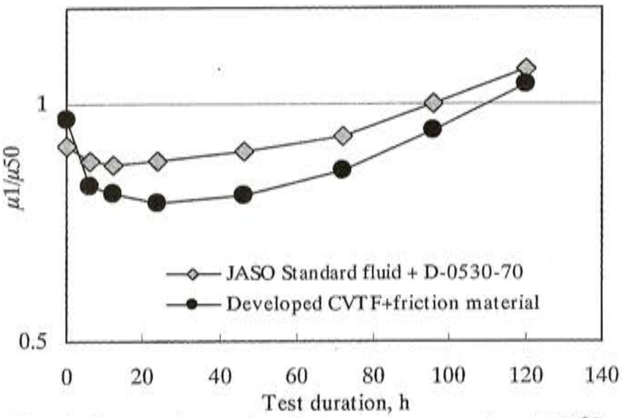

Fig. 4 Comparison of anti-shudder properties @40℃

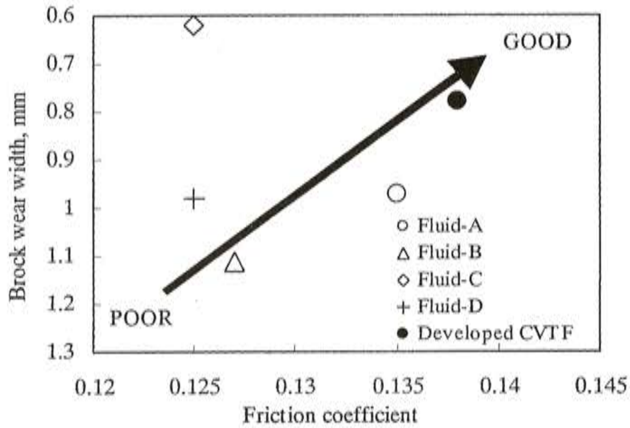

Fig. 5 Friction coefficients and wear widths

3.5. 金属間摩耗防止性

高トルクエンジンへのベルトCVTの適用に伴い，ベルト-プーリ間は高面圧で摺動することになる．このため，ベルトCVTフルードには高い金属間摩擦係数を長期間維持すると同時に，良好な摩耗防止性能が求められる．

Fig. 5に開発油および市販油の金属間摩擦係数と摩耗防止性能の関係を示す．摩耗防止性能の評価にはLFW-1試験後のブロック試験片の摩耗痕幅を用いた．開発油は高い金属間摩擦係数と金属摩耗防止性とを併せ持っており，良好な性能を有している．

3.6. 金属焼付き防止性

先に述べたベルト-プーリ間の高面圧化に対応するため，ベルトCVTフルードには摩耗防止性と併せて優れた焼付き防止性を長期間持続することが必要である．

各油の金属焼付き防止性を評価するため，酸化劣化試験後に3.3.と同じ条件でLFW-1試験を実施し，ブロック-リング間で焼付きが発生する酸化劣化試験時間を比較した．酸化劣化方法としては3.3.と同様に150℃でのISOT試験を採用した．

Fig. 6に開発油および市販油の酸化劣化時間とLFW-1試験における焼付き発生状況を示す．市販油ではISOT時間48hおよび96hの酸化劣化後に焼付きが見られたが，開発油ではISOT試験186hの酸化劣化後でも焼き付きが発生せず，焼付き防止性を持続している．

3.5. M/M anti-wear property

When a steel-belt CVT is mated to a high-torque engine, sliding between the belt and pulleys takes place under higher contact pressure. Consequently, the CVT fluid must maintain a high M/M friction coefficient over a long period of use and also provide excellent anti-wear properties.

Figure 5 shows the relationship between the M/M friction coefficient and anti-wear properties of the developed CVT fluid and the four commercial fluids. The wear width on the block test piece after the LFW-1 test was used in evaluating their anti-wear properties. The results indicate that the developed CVT fluid has both a high M/M friction coefficient and good anti-wear properties, enabling it to provide excellent performance.

3.6. Anti-seizure property

In order to handle the higher contact pressure between the belt and pulleys mentioned above, a steel-belt CVT fluid must be able to maintain an excellent anti-seizure property over a long period of time, in addition to providing good anti-wear performance. To evaluate the anti-seizure property of each fluid, LFW-1 tests were conducted under the same conditions as in section 3.3 following the oxidation stability test, and a comparison was made of the oxidation stability test time until seizure occurred between the block and ring. Oxidation stability was evaluated in ISOT tests conducted at a fluid temperature of 150℃ and under the same conditions as in section 3.3.

Figure 6 compares the seizure time of the developed CVT fluid and the commercial fluids following the oxidation stability test using the LFW-1 test procedure. The results for the commercial fluids indicate that seizure occurred after 48 hours and 96 hours of the ISOT oxidation stability test. In contrast, seizure did not occur with the developed CVT fluid even after 186 hours of the ISOT oxidation stability test, indicating that it maintained good anti-seizure properties.

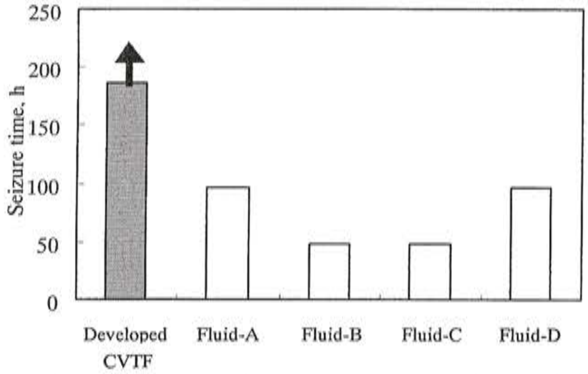

Fig. 6 Comparison of anti-seizure properties after oxidation

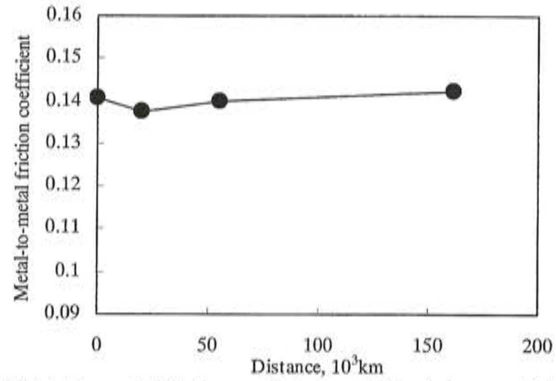

Fig. 7 Metal-to-metal friction coefficients stability during a vehicle test

3.7. 実車試験

開発油の金属間摩擦係数の持続性について，ベルトCVT搭載車両の走行試験で評価した．

Fig. 7に開発油の金属間摩擦係数の経時変化を示す．3.3.でのラボ試験の結果と同様，実車使用時の開発油は走行距離の増加に対しても新油時と同等の金属間摩擦係数を維持しており，市場におけるFill for life化の要望に対応できる性能となっている．

4. まとめ

高トルクエンジンへの対応，極寒地への適用，Fill for life化を目的として，高性能ベルトCVTフルードを開発した．

謝辞

新CVTフルードの開発にあたって多大なご協力を頂いた関係会社各位に，深く感謝の意を表する．

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(1) Y. Mogi, et al., Required Characteristics of fluids for future belt CVTs, Proceedings of TVT, pp. 10-15, (2001).
(2) JASO M347-95: Test Method for Shear Stability of Automatic Transmission Fluids.
(3) ASTM D2714-94: Testing Methods for Calibration and Operation of the Falex Block-on-Ring Friction and Wear Testing Machine.
(4) JASO M349-98: Test Method for Anti-shudder Performance of Automatic Transmission Fluids.

3.7. In-vehicle testing

A driving test was conducted with a steel-belt CVT-equipped test vehicle to evaluate the ability of the developed CVT fluid to retain its M/M friction coefficient. A time history of the M/M friction coefficient of the developed CVT fluid is shown in Fig. 7. Similar to the results seen in the laboratory tests described in section 3.3, the developed CVT fluid maintained the same level of M/M friction coefficient in the driving test as when it was fresh, despite the continued increase in the distance driven. This result indicates that the new CVT fluid has sufficient performance to meet the demand for fill-for-life capability in real-world driving.

4. Conclusion

A high-performance steel-belt CVT fluid has been developed with the aim of achieving performance characteristics allowing application to high-torque engines, use in cold weather regions and fill-for-life capability.

Acknowledgements

The authors would like to thank various individuals at the companies involved in the development of this new steel-belt CVT fluid for their invaluable cooperation.

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(2) JASO M347-95: Test Method for Shear Stability of Automatic Transmission Fluids.
(3) ASTM D2714-94: Testing Methods for Calibration and Operation of the Falex Block-on-Ring Friction and Wear Testing Machine.
(4) JASO M349-98: Test Method for Anti-shudder Performance of Automatic Transmission Fluids.

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微粒子ピーニングによるCVTプーリの耐摩耗性向上

Improvement of CVT Pulley Wear Resistance by Micro-shot Peening

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抄 録　近年，乗用車の燃費改善，運転性能向上に対する要求から，金属製プッシュ式ベルトを用いた無段変速機(ベルトCVT)が普及しつつある．ベルトCVTでは，ベルトとプーリ間の摩擦力で動力を伝達することから，高トルクエンジンや大重量車に適用するには，プーリの耐摩耗性が重要な技術課題となる．本報では，プーリの耐摩耗性向上技術として，微粒子ピーニング処理に着目し，田口メソッドによってピーニング条件を最適化するとともに，その効果を明確にし，微粒子ピーニングがこの技術課題の解決に有効な技術であることを見出したので報告する．

Summary　Continuously variable transmissions (CVTs) with a metal pushing V-belt have been penetrating the market in recent years in response to demands for improvement of vehicle fuel economy and driveability. Because metal-belt CVTs transmit torque by means of friction force between the belt and pulleys, the wear resistance of the pulleys is a key technical issue that must be addressed in order to apply these transmissions to high-torque engines and heavy vehicles. This article describes the application of micro-shot peening as a technique for improving pulley wear resistance. The peening conditions were optimized by using the Taguchi Method, and test results confirmed the effectiveness of this approach. It was found that micro-shot peening is a promising technology for resolving the issue of pulley wear resistance.

1. まえがき

近年，乗用車の燃費改善や運転性能向上の要求から，VDT社(Van Doorne's Transmissie b.v.)の金属製プッシュ式ベルトを用いた無段変速機(以下ベルトCVTと呼ぶ．CVT, Continuously Variable Transmissions)が普及しつつある[(1)]．ベルトCVTはベルトとプーリ間の摩擦力により動力を伝達するので，特に高トルクエンジンや大車重の車両に適用するには，プーリのベルトを挟み込む力(以下クランプ力，Clamping forceと呼ぶ)を増加し，両者の大きな滑り(巨視的滑りと呼ぶ)を防止し，耐久性を確保する必要がある．

しかし，ベルトとプーリ間には，元々正常な状態でも微小な滑りは存在しており[(2)]，微小滑り状態でのクランプ力の増大に伴うプーリの摩耗防止が，大容量化に際しての重要な技術課題となっている．本稿では以上のような負荷を受けるプーリの耐摩耗性向上技術として，微粒子ピーニング処理の適用について解説する．

1. Introduction

In recent years, demands for improvement of vehicle fuel economy and driveability have led to expanded use of continuously variable transmissions with a metal pushing V-belt made by Van Doorne's Transmissie b.v. (VDT).[(1)] Metal-belt CVTs transmit torque by means of friction force between the belt and pulleys. Accordingly, in applying these units to high-torque engines or heavy vehicles, the clamping force that squeezes the belt between the pulley faces must be increased. It is necessary to prevent large micro slip from occurring between the belt and pulleys and to assure their durability.

However, tiny micro slip typically occurs between the belt and pulleys even under normal operating conditions.[(2)] Prevention of pulley wear under a condition of greater clamping force in the presence of micro slip is a critical technical issue with respect to increasing the torque capacity of CVTs. This article describes the application of micro-shot peening as a technique for improving the wear resistance of CVT pulleys that have to withstand higher clamping forces.

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2. 大容量ベルトCVT用プーリの開発

2.1. ベルトCVTの変速機構

ベルトCVTでは，Fig. 1に示すように，ベルトの構成要素であるエレメント(Element)とプーリ間の摩擦力でトルクを伝達する[3]．変速させるには，可動プーリ(Slide pulley)を油圧で軸方向に移動させると，固定プーリ(Fixed pulley)との間の隙間が変化し，エレメントがプーリ半径方向(Fig. 1の上下方向) に移動することにより，ベルト巻き付き半径が変化する．Fig. 2に示すように，エレメント側の摩擦接触面をフランク面(Flank surface)，プーリ側接触面をシーブ面(Sheave surface)と呼び，これらの接触面は，トルク伝達容量に大きな影響を与える．フランク面には摩擦係数を高めるため，板厚方向(巻き付き部の接線方向)には，微細溝を設けている．

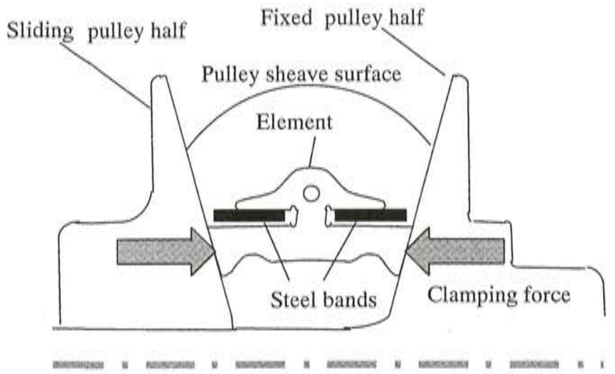


Fig. 1 Schematic diagram of metal-belt CVT

2.2. シーブ面の剥離摩耗について

ベルトCVTのトルク容量を増大するには，エンジンまたはタイヤから入ってくるトルクの増大に対して，クランプ力を増大することによって，エレメントとプーリ間に巨視的な滑りが発生しないようにしておく必要がある．クランプ力が増大すると，フランク面やシーブ面には高面圧が作用する．さらに，このベルトCVTでは，個々のエレメントとプーリ間に，正常状態でも微小な滑りが存在しているため，過酷な運転条件下では，シーブ面に深さ10μm程度の亀裂を伴う疲労摩耗(以下，剥離摩耗，peeling wearと呼ぶ)が発生する．

Fig. 3に実機の過酷耐久試験後におけるシーブ面の摩耗状況を示す．本摩耗が進行するとシーブ面の半径方向に段差が発生し，ベルトの半径方向の移動を妨げようとするため，最終的には変速が困難となる．また，プーリの軸心に対するベルトの直角度は，ベルトが正常に張られているときに，厳密に調整されているので，段差ができるとその直角度に狂いが生じ，ベルトの耐久性に悪影響をおよぼす．

2. Development of Pulleys for Large-capacity Metal-belt CVTs

2.1. Ratio change mechanism of metal-belt CVTs

As illustrated in Fig. 1, a metal-belt CVT transmits torque by means of friction force between the pulleys and the elements that are major components of the belt.[3] A ratio change is executed by moving the sliding pulley half axially under hydraulic pressure to change the distance between it and the fixed pulley half; the movement of the elements in the pulley's radial direction (downward direction in Fig. 1) changes the radius of belt contact on the pulley. The contact friction surface of the elements is referred to as the flank surface (Fig. 2) and that of the pulleys as the sheave surface. These contact surfaces have a large effect on the torque capacity of a metal-belt CVT. Tiny grooves tangential to the area of pulley contact are provided in the flank surface for the purpose of increasing its friction coefficient.

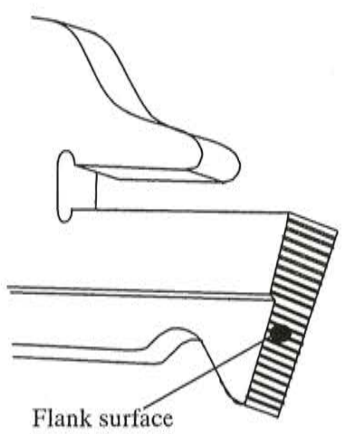


Fig. 2 Schematic diagram of element flank surface

2.2. Peeling wear of sheave surface

In order to increase the torque capacity of a metal-belt CVT, micro slip between the elements and pulleys must be prevented by increasing the clamping force in relation to the higher torque entering the transmission from the engine or the tires. Increasing the clamping force causes high contact pressure to act on the flank and sheave surfaces. In addition, micro slip is typically present between the individual elements and the pulleys of a metal-belt CVT even under normal conditions. Consequently, fatigue wear (referred to here as peeling wear) stemming from a crack about 10 μm in depth can develop in the sheave surface under harsh driving conditions.

Figure 3 shows the peeling wear condition of the pulley sheave surface of an actual CVT following a severe durability test. Further progression of this peeling wear produces a level difference in the radial direction of the sheave surface, which can hinder the radial movement of the belt and ultimately make it difficult to execute ratio changes. Additionally, the orthogonality of the belt relative to the center axis of the pulley is rigorously adjusted when the belt runs around the pulleys under a normal condition. The development of a level difference can cause the orthogonality to deviate and thereby adversely affect the durability of the belt.

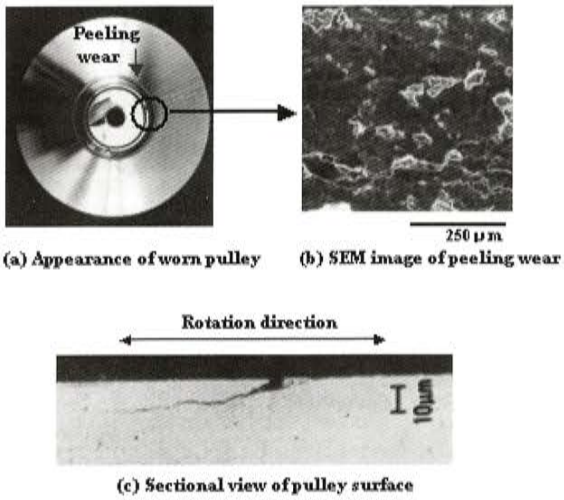


Fig. 3 Peeling wear of pulley sheave

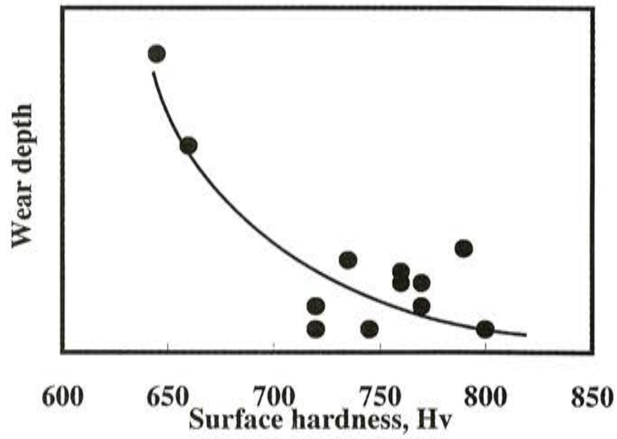


Fig. 4 Relations between surface hardness and wear depth

2.3. シーブ面硬さと剥離摩耗の関係

Fig. 4に，実機耐久試験におけるシーブ面硬さと剥離摩耗量との関係を示す．剥離摩耗深さはシーブ面硬さの増加に伴って減少している．

通常，プーリはFig. 5に示す工程で製造される．すなわち，肌焼鋼の表面を浸炭焼入れ焼戻しで硬化させた後，シーブ面を研削し，規定の精度と面粗度を得る．

耐摩耗性を向上させるため，硬度を従来品より飛躍的に増大させるため，この工程に加えて，シーブ面に微粒子ピーニング処理を施した．この処理により，研削後のシーブ面粗度を大きく悪化させることなく，表面硬さの増大が可能となる．以下に，著者らが開発した微粒子ピーニング条件の最適化について紹介する．

2.3. Relationship between sheave surface hardness and peeling wear

Figure 4 shows the relationship between the sheave surface hardness and amount of peeling wear that occurred in a CVT durability test. It is seen that the peeling wear depth decreases with increasing hardness of the sheave surface.

The general manufacturing process of the primary pulley is outlined in Fig. 5. After the surface of case-hardened steel is hardened by carburizing, quenching and tempering, the pulley sheave surfaces are ground to the specified accuracy and surface roughness.

With the aim of improving pulley wear resistance, we added a micro-shot peening treatment to this process in order to increase sheave surface hardness dramatically compared with previous pulleys. This treatment makes it possible to increase the surface hardness without substantially affecting the surface roughness of the ground sheave surfaces. The following section explains how we optimized the conditions of this micro-shot peening method that we developed.

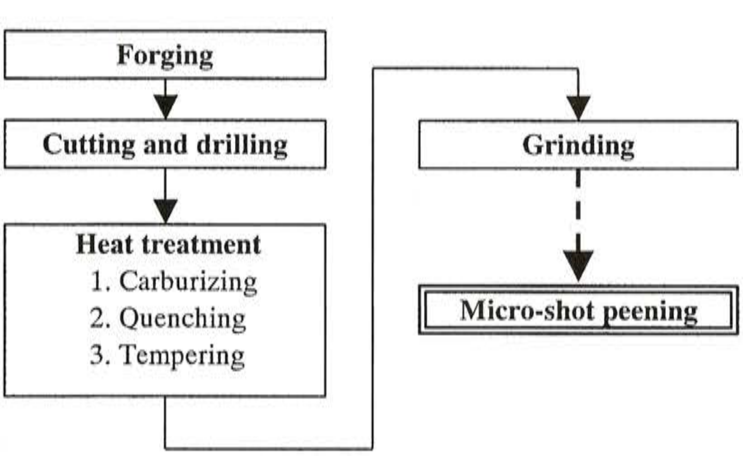


Fig. 5 Manufacturing process of primary pulley

3. 微粒子ピーニング条件の最適化

3.1. ピーニング処理条件選定の考え方

シーブ面に微粒子ピーニング処理を施す際には，面粗度の悪化による相手材への攻撃性や，μ －V特性の変化，およびシーブ面内での硬さ不均一による摩耗のばらつきなどに注目する必要がある．そのため，田口メソッド(L18)を適用し，これらの副作用を確認しながら，最適なピーニング条件を模索することとした．プーリを想定した円盤試験片に，微粒子ピーニング処理を施し，この試験片とエレメントとを摺動させ，処理条件が各種特性にどのように影響するかを調査した．

3.2. 田口メソッドによる実験評価

3.2.1. パラメータの選定

本評価では，微粒子ピーニング処理の際に耐摩耗性や面粗度に影響を及ぼすと推定される因子としてTable 1の7項目を選択し，L18直交表に割り付けた．影響因子には，以下のように処理条件因子とその他の因子とがある．

・処理条件因子：粒子硬さ，エア圧，粒径，投射時のノズル角度，試験片回転速度

・その他の因子：試験片の材質，潤滑油種

調査項目として，①円盤試験片の表面硬さ，②面粗度(Ra)，③μ －V特性，④フランク面の最大粗さ(Ry)，を設定し，特に摺動試験前後の変化に注目した．ここで，各特性について，望ましい特性別に分類すると，以下の3種類がある．

・大である程良好(望大特性)：表面硬さ，フランク面摩耗量(最大粗さで代用)

・小である程良好(望小特性)：円盤試験片の面粗度(Ra)，

・目標値に近い程良好(望目特性)：μ －V特性

3. Optimization of Micro-shot Peening Conditions

3.1. Concept for selection of peening conditions

In applying micro-shot peening to pulley sheave surfaces, care must be taken not to increase aggressivity toward the mating material by degrading surface roughness, not to change the μ-V characteristic and not to produce uneven sheave surface hardness that would cause wear variation, among other concerns. For that reason, we decided to use the Taguchi Method (L18) to confirm these secondary effects in the course of seeking the optimum peening conditions. Micro-shot peening was applied to disk-shaped test pieces representing CVT pulleys, and the pieces were then subjected to sliding contact with belt elements to investigate the effect of the peening conditions on various properties.

3.2. Experimental evaluation using the Taguchi Method

3.2.1. Selection of parameters

The seven parameters shown in Table 1 were selected as factors that were presumed to affect wear resistance and surface roughness in the micro-shot peening process. These factors were arranged in an L18 orthogonal array. As mentioned below, the selected factors include ones related to the peening conditions as well as other factors.

- Peening condition factors: particle hardness, air pressure, particle diameter, nozzle projection angle and test piece rotational speed
- Other factors: test piece material and type of lubrication oil

The items selected for investigation were the (1) surface hardness, (2) surface roughness (Ra) and (3) μ-V characteristic of the disk-shaped test piece and (4) maximum roughness (Ry) of the flank surface. Special attention was paid to changes in the characteristics before and after the sliding test. The following three types of characteristics are desirable here.

- Larger-is-better characteristic: surface hardness and flank surface wear (represented by maximum roughness)
- Smaller-is-better-characteristic: surface roughness (Ra) of disk-shaped test piece
- Nominal-is-best characteristic: μ-V characteristic

Table 1 Parameters of L18 experimental design

	Parameter	Unit	1st. level	2nd. level	3rd. level
A	Particle hardness	Hv	≧ 750	< 750	-
B	Air pressure	-	Low	Medium	High
C	Particle diameter	-	Small	Medium	Large
D	TP material	-	Steel A	Steel B	Steel C
E		-	-	-	-
F	Nozzle angle	Deg	45	60	90
G	Fluid	-	Oil A	Oil B	Oil C
H	Rotational speed	-	Low	Medium	High

3.2.2. 微粒子ピーニング方法

プーリ試験片には,同様の熱処理工程で作製した直径50mm，厚さ7mmの円盤試験片を使用し，Table 1のL18直交表に従って微粒子ピーニング処理を施した．ここで，投射材は鉄系粒子とし，0.2mm未満の直径の範囲で，粒度分布や硬さを変更しながらピーニング修理した．この際，直圧式のピーニング機を使用し，Fig. 6に模式的に示すように，ディスク面に対する投射角度を3種類(45°, 60°, 90°)に変更して，所定のエア圧で投射した．

また，円盤試験片の材料には，クロム肌焼鋼2種(クロムの含有量が異なる)，クロムモリブデン肌焼鋼1種の計3鋼種を使用し，潤滑油には，添加剤成分の異なる3種類の自動変速機用潤滑油を使用した．

3.2.2. Micro-shot peening method

Disk-shaped test pieces, measuring 50 mm in diameter and 7 mm in thickness, were produced in the same heat treatment processes as actual CVT pulleys. Micro-shot peening was performed according to the L18 orthogonal array given in Table 1. The micro-shot projectiles used in the peening treatment were steel particles; the particle size distribution and hardness were varied in a diameter range of less than 0.2 mm. A peening machine of the direct-pressure type was used. Shot particles were projected under the specified air pressure at the surface of a disk-shaped test piece at three different angles (45°, 60° and 90°), as shown schematically in Fig. 6.

The disk-shaped test pieces were made of three types of steel. Two types were case-hardened chromium steel containing different amounts of chromium, and the third type was a case-hardened chromium-molybdenum steel. Three types of automatic transmission fluid containing different additive packages were used as the lubrication oil.

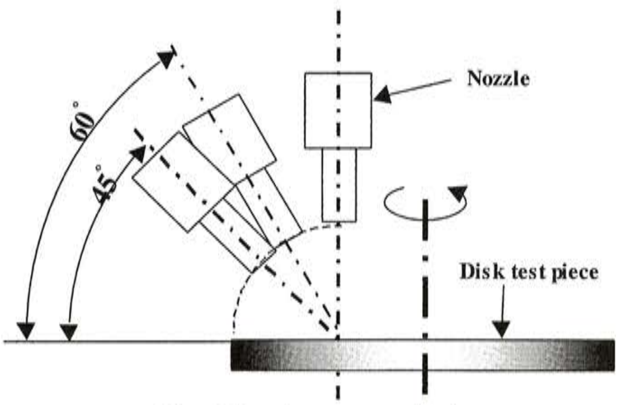


Fig. 6 Peening test method

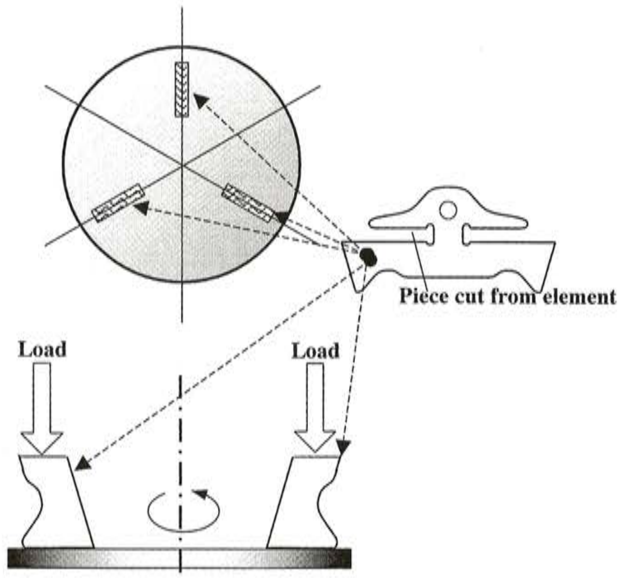


Fig. 7 Experimental setup of wear test

3.2.3. 摺動試験方法

Fig. 7に示すような試験方法で摺動させ，トルク容量の変化に注目して，相手材のフランク面の摩耗量，μ −V特性を計測した．フランク面には微細な溝があるため，フランク面摩耗量については，全ての試験片において試験後にも溝が残存していることを確認した後，その最大粗さ(Ry)を測定し，測定値が大である程，溝の摩耗量が少ないことを意味することから，摩耗量の代用値とした．また，μ −V特性の変化を調べるため，摺動試験の前後で，6箇所の滑り速度における摩擦係数を計測した．Fig. 9に測定例を示す．

実車の耐久走行においても，μ −V特性が途中で変化しないことが望ましいので，摺動試験前後で摩擦係数に変化がないことを目標とした．

3.2.3. Sliding test method

Sliding tests were conducted using the experimental setup shown in Fig. 7. The amount of wear on the mating flank surface and the μ -V characteristic were measured while paying attention to the change in torque capacity. A flow chart of the wear test is given in Fig. 8. Since tiny grooves are provided in the flank surface, we first confirmed that all the test pieces retained their grooves following the sliding test and then measured the maximum roughness (Ry). That value was used as a substitute for the wear amount, because a larger measured value means a smaller amount of groove wear. To investigate the change in the μ -V characteristic, the friction coefficient was measured at the sliding velocity of six locations before and after the test. An example of the measured results is shown in Fig. 9.

The target set for the μ -V characteristic was that it should not show any change before and after the sliding test because it is desirable that this characteristic should not display any change midway through a durability test of an actual vehicle.

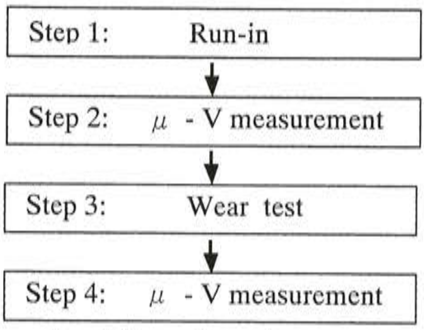


Fig. 8 Flow chart of wear test

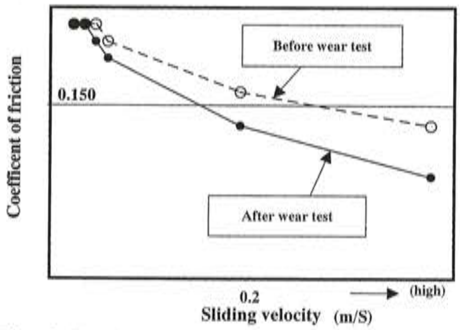


Fig. 9 Typical results of μ -V measurement

3.3. 評価結果

田口メソッドによる各特性の解析結果を，Table 2に示す．なお，表中の感度(Intensity)とSN比(SN ratio)は，各々が，個々の特性値の大小，およびそのばらつきに対する各水準の影響度合いを示しており，Table 2中に◎または○で示した水準を選択した場合に，特性値への感度が大であること，或いはばらつき低減に効果的であることを示している．結果の整理にあたっては田口メソッドの解析手法に従い，望小特性である円盤試験片の面粗度に関しては感度とSN比の両者を分離して評価する必要性がないことから，SN比のみを用いている．また，エレメント摩耗量については，試験条件を摩耗に対して最も厳しいLOW変速比相当の1条件としたため，試験条件の違いによる誤差の評価は行わず，感度のみの整理としている．解析結果をまとめると以下のようになる．

まず，シーブ面の耐摩耗性向上に有利な組み合わせとしては，高い表面硬度がばらつきなしで得られるTable 2中の○となる水準の組合せである，エア圧：大，粒子硬さ：大，投射方向：90°が望ましい．

また，処理条件因子の中で，投射粒径を小さくすることがエレメントの摩耗の低減に有効であることが分かった．試験片の面粗度(Ra)の影響も同様であることから，投射粒径が小さいと面粗度が悪化せず，エレメントが摩耗しにくくなると言える．

μ －V特性の変化に対しては，潤滑油種が支配的であり，ピーニング条件の影響は僅かである．また，シーブ面摩耗，フランク面摩耗に関する全ての特性に対して優れる試験片材質は認められない．

以上より，エレメントの摩耗を抑制し，シーブの耐摩耗性を向上させるための最適ピーニング条件は，以下のようになる．

エア圧：中間，平均粒径：最小，粒子硬さ：大，投射角度90°，試験片回転速度：中間

3.3. Evaluation results

The analysis results obtained with the Taguchi Method for each property are shown in Table 2. The (intensity) (sensitivity) and SN ratio given in the table indicate the degree of influence of each level relative to the magnitude and dispersion of the individual property values. If the levels indicated by the double (◎) or single open circles (O) are selected, it indicates either that their intensity toward the property values is large or that they are effective in reducing dispersion. In line with the analysis procedure of the Taguchi Method, the intensity and SN ratio did not have to be separated in making an evaluation of the surface roughness of the disk-shaped test pieces, which was a smaller-is-better characteristic. Accordingly, the SN ratio alone was used in organizing the evaluation results. The results for the element wear amount were organized on the basis of the intensity alone. Error attributable to differences in the test conditions was not evaluated because only one test condition was selected, i.e., a condition corresponding to the Low ratio state, which is the severest condition with respect to wear. The analysis results are briefly summarized below.

A combination of high air pressure, high particle hardness and a 90° projection angle is advantageous for improving the wear resistance of pulley sheave surfaces. This desirable combination, indicated by the open circles (O) in Table 2, provides high surface hardness without any dispersion.

Among the peening condition factors, it was found that reducing the shot particle size is effective in reducing element wear. The same effect was also observed for the surface roughness (Ra) of the test pieces. This suggests that using small shot particles reduces element wear without causing surface roughness to deteriorate.

The type of lubrication oil had a governing effect on changes in the μ -V characteristic, while the peening conditions had only a slight effect. None of the test piece materials examined was found to be superior with respect to all the properties affecting sheave and flank surface wear.

Based on the analysis results, the following conditions were found to be optimum for suppressing element wear and improving the wear resistance of pulley sheave surfaces.

Air pressure: medium level; mean particle diameter: smallest; particle hardness: high; projection angle: 90°; and test piece rotational speed: medium.

Table 2 Results of experimental design (L18) analysis

Parameters		Surface hardness Larger-is-better characteristic		Roughnes Ra Smaller-is-better characteristic		μ-V characteristic Nominal-is-best characteristic		Elements wear (Ry) Larger-is-better characteristic	
		Intensity	SN ratio	Intensity	SN ratio	Intensity	SN ratio	Intensity	SN ratio
Particle hardness (Hv)	≧ 750	-	○		○	-	-	-	
	< 750		↑		↑				
Air pressure	Low				○			△	
	Medium	↓	-		↑	-	-	○	
	High	○						△	
Particle diameter	Small				◎			◎	
	Medium	-	-		↑	-	-	↑	
	Large								
TP material	Steel A				△			△	
	Steel B	-	-		△	-	-	○	
	Steel C				○			△	
Nozzle angle (deg)	45							△	
	60	-	↓		↓	-	-	△	
	90		○		◎			○	
Rotational speed (rpm)	Low				△		△	△	
	Medium	-	-		○	-	○	○	
	High				△		△	△	
Fluid	Oil A					△		◎	
	Oil B					◎	-	△	
	Oil C					△		◎	

(◎: strongly effective, ○: effective, △: somewhat effective, -: ineffective)

4. 剥離摩耗の低減効果

4.1. 4円筒試験

微粒子ピーニング条件の変更による剥離摩耗の低減効果は，Fig. 10に示す4円筒試験で確認した．ここで，ピーニング処理条件としては，前述の最適条件をベースとしたが，試験片の面粗度は剥離摩耗への影響が大きいと推定されるので，粒径を変更させる条件も加えた．

なお，試験条件は面圧：2.59GPa，相対すべり速度：1.26m/secとし，一定時間後の剥離摩耗量で比較した．

4. Effect on Reducing Peeling Wear

4.1. Four-roller test

The effect of changing the micro-shot peening conditions on reducing peeling wear was confirmed in tests conducted with a four-roller tester (Fig. 10). The peening conditions used here were based on the above-mentioned optimum conditions, but the shot particle size was also varied. That was done because it was estimated that the surface roughness of the test pieces would have a large effect on peeling wear.

The test conditions were a contact pressure of 2.59 GPa and a relative sliding velocity of 1.26 m/sec. The amount of peeling wear that developed was compared after a given test time.

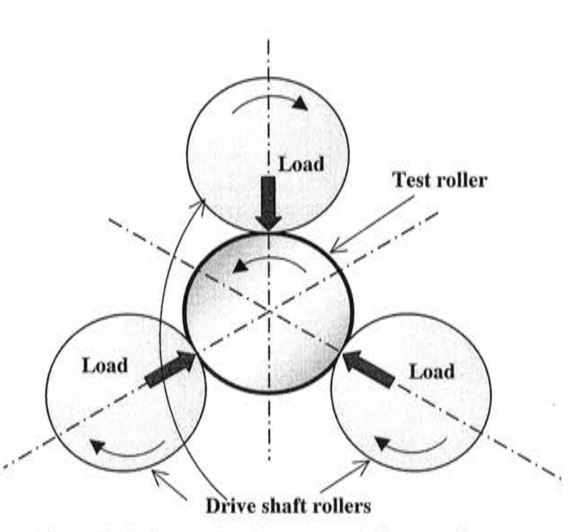


Fig. 10 Schematic diagram of four-roller test

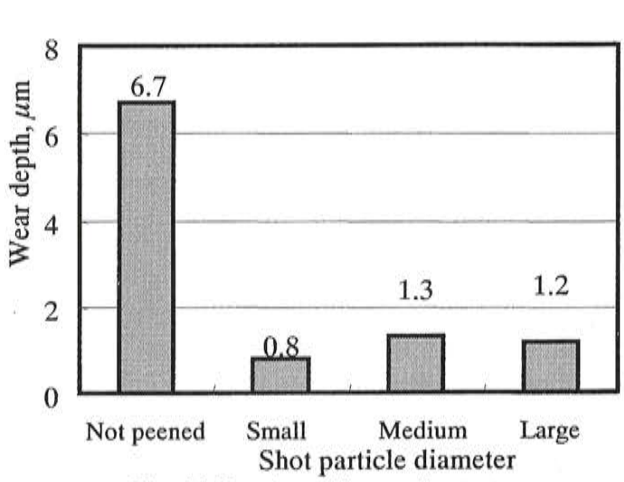


Fig. 11 Results of four-roller test

4.2. 試験結果

Fig. 11に試験後の摩耗深さを，Fig. 12(a), (b)に試験片表面のき裂発生状況を，(c)に剥離摩耗部断面のき裂進展状況を示す．

Fig. 11では，ピーニング処理をすればいずれの条件でも，処理をしない場合に比べ，剥離摩耗深さが大幅に減少し，剥離摩耗が飛躍的に減少したことが認められる．特に，小粒径で処理すれば，試験片の面粗度の悪化を防止できるので，剥離摩耗は最小となる．また，本試験では，Fig. 12(c)に示すように，約10μmの深さの微小なき裂から剥離摩耗が進行している点で，実機での剥離摩耗と同一の現象が再現していると考える．

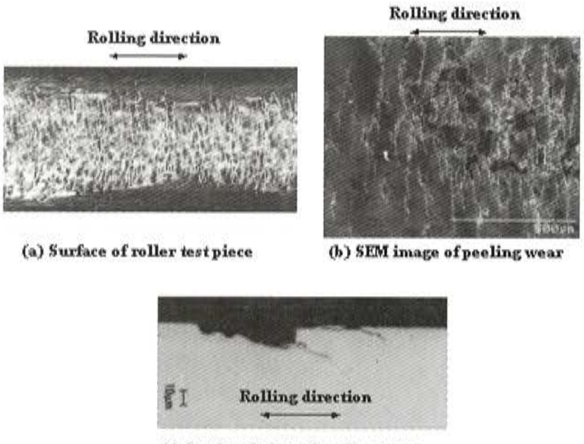
以上の結果から剥離摩耗の低減のためには，面粗度を小さくすることが有効であり，前章で示した最適条件は剥離摩耗の低減に対しても有利な条件となっていることがわかる．

4.2. Test results

Figure 11 shows the wear depth following the tests. The crack formation condition in the test piece surface is shown in Fig. 12 (a) and (b), and the condition of crack growth is shown in a cross-sectional view of the peeling wear area in Fig. 12 (c).

It is seen in Fig. 11 that all the peened test pieces, regardless of the shot particle diameter, showed a large reduction in peeling wear depth, indicating that peeling wear was dramatically reduced, compared with the unpeened test piece. The use of small shot particles in particular avoided degradation of the test piece surface roughness and resulted in the smallest amount of peeling wear. As seen in Fig. 12 (c), peeling wear developed from a micro crack of about 10 μm in depth. This four-roller test thus reproduced the same peeling wear phenomenon that was observed for the sheave surface of an actual CVT pulley.

The roller test results indicated that reducing the surface roughness is effective in reducing peeling wear. It was also confirmed that the optimum peening conditions mentioned earlier are advantageous for reducing peeling wear.



(a) Surface of roller test piece　(b) SEM image of peeling wear

(c) Sectional view of peeling wear

Fig. 12 Peeling wear of roller test piece

5. 実機での効果

5.1. 実機試験結果

微粒子ピーニング処理した場合の剥離摩耗に対する低減効果は，実機耐久試験でも確認した．剥離摩耗が発生しやすい入力側プーリ対(Primary pulley halves)のシーブ面に微粒子ピーニング処理を施し，変速比一定の条件で試験し，剥離摩耗深さを未処理品と比較した．

Fig. 13に示すように，実機においても微粒子ピーニング処理することにより，剥離摩耗深さが大幅に減少している．なお，エレメントのフランク面の摩耗量についても問題のないレベルであることを確認している．

5. Wear Reduction Effect with Actual Pulleys

5.1. Wear test results

The effect of micro-shot peening on reducing peeling wear was confirmed in a durability test of an actual CVT. The sheave faces of the primary pulley halves, which are susceptible to peeling wear, were micro-shot peened and tested under a condition of a fixed CVT ratio. The peeling wear depth was then compared with that of unpeened sheave faces.

As shown in Fig. 13, the wear depth of the actual pulley sheave faces was markedly reduced by micro-shot peening. It was also confirmed that the amount of wear on the element flank surfaces was at an acceptable level.

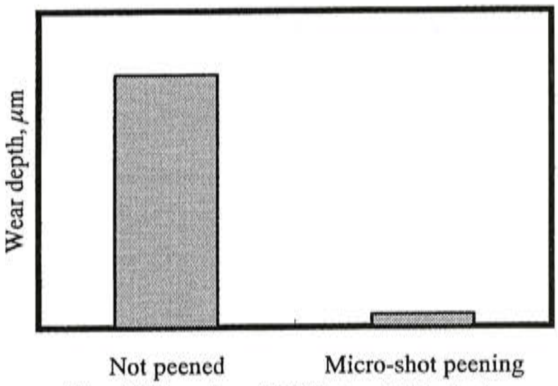


Fig. 13 Results of CVT durability test

5.2. 実機への適用状況

Fig. 14に，入力側プーリ対に微粒子ピーニング処理を施した排気量2.5 L（エンジントルク250Nm）クラスのベルトCVTの断面図を示す．微粒子ピーニング処理によりプーリの剥離摩耗が大幅に減少し，金属ベルトの高強度化と相まって，従来，排気量2.0L（同190Nm）クラスまでであったベルトCVTの大容量化を実現している．

6. おわりに

以上，ベルトCVTプーリの剥離摩耗の減少を目的とした微粒子ピーニング処理について解説した．

今後は，微粒子ピーニング処理によるフリクション低減や高強度化の可能性についての研究開発が進展し，ベルトCVTの特長である燃費効率のさらなる向上や，軽量・コンパクト・低コスト化を実現するための重要な技術の一つとして発展していくものと期待する．

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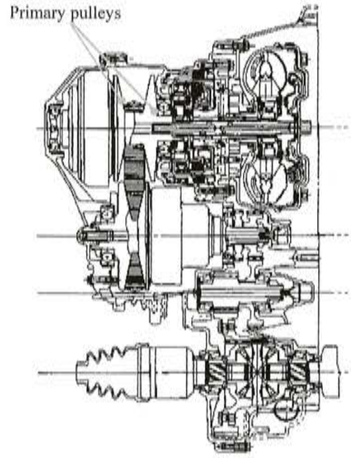

Fig. 14 Main cross-sectional view of belt-CVT

5.2. Application to actual CVT pulleys

Figure 14 is a cross-sectional view of a metal-belt CVT for use with 2.5-liter engines (engine torque of 250 N-m). Micro-shot peening is applied to the sheave faces of the primary pulley halves to reduce peeling wear substantially. That improvement, along with the adoption of a stronger metal belt, has made it possible to increase the torque capacity of metal-belt CVTs, which previously were used only with engines up to 2.0 liters in displacement (engine torque of 190 N-m).

6. Conclusion

This article has described a micro-shot peening process that is designed to reduce the peeling wear of metal-belt CVT pulleys. In future R&D work, we will examine the possibility of using this micro-shot peening process to reduce friction and increase strength. It is expected that this process will become a key technology for further improving the excellent fuel economy that is characteristically obtained with metal-belt CVTs and also for achieving further reductions in weight, size and cost.

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特　集

CVT1&CVT3混流組立ラインの紹介

Overview of the CVT1 and CVT3 Mixed Assembly Line

土屋　伸介* Shinsuke TSUCHIYA　永澤　泰介* Taisuke NAGASAWA　池田　淳一* Junichi IKEDA　斉藤　寿* Hisashi SAITO　高橋　潤* Jun TAKAHASHI

抄　録　当社は，'02年7月にCVT1とCVT3というトルク容量も部品構成も異なるCVTを混流して生産できる組立ラインを新設した.本ラインでは，キューブ，ムラーノ，ティアナ，プレサージュなどの日産自動車(株)殿向け車両に搭載する上記CVTを生産しており，生産台数の変動にも柔軟に対応するとともに高品質を確保している.

本稿では，混流組立ラインの概要と工程設計の考え方を紹介する.

Summary　The CVT1 and CVT3 assembly line can produce CVTs having different component parts, torque capacities and target vehicle applications. This mixed assembly line assures high product quality while allowing flexible adjustment of production volumes. It produces the CVTs used on the Nissan Cube, Murano, Teana and Presage models. This article presents an overview of the CVT1 and CVT3 mixed assembly line and process design approach.

1. はじめに

当社初のFF車用ベルト式CVTであるCK2(ADO型)をベースとして，3.5Lエンジンにも対応できるCVT3と2.0Lエンジンまで対応するCVT1とを同時開発し，短期間で立上げて，'02/7に生産を開始することとなった.

これに対応するため，同一のラインで両ユニットを効率的に生産可能とする2機種・混流組立ラインを計画した.

CVTユニットでは，ステップATより高い清浄度が要求されたことから，夾雑物の発生や混入の防止には，それまで以上に取り組んだ．また，2機種のCVTを混流生産するための生産設備の設計にあたり，設備や治具を極力共用し，投資や所要面積を削減するよう努めた.

2. ライン概要

2.1. 概要

所在地　：CVT事業所　第一工場
生産開始：2002年7月
設備台数：101台
自動化率：34 %
生産能力：12,000台／月　'03/7時点(増強可能)
生産機種：CVT1　1機種
　　　　：CVT3　5機種

1. Introduction

The CVT3 and the CVT1 were developed simultaneously in a short period of time and put into production in July 2002. The former CVT can accommodate even 3.5-liter engines, while the latter CVT can be mated to engines displacing up to 2.0 liters. Both models are based on the CK2 (ADO model) CVT, our first steel-belt CVT designed for use on front-wheel-drive cars.

To handle the production of these two CVTs, we designed a two-model mixed assembly line capable of producing both units efficiently on the same line. Because CVTs require a higher level of cleanness than stepped ATs, greater efforts than ever before were devoted to preventing the occurrence and incursion of foreign matter. In addition, in designing the production facilities for mixed production of these two CVT models, every effort was made to use the equipment and fixtures in common so as to reduce the capital investment and required factory floor space.

2. Assembly Line Overview

2.1. Outline

Location: Plant No. 1 at CVT Manufacturing Department
Production launch: July 2002
No. of machines: 101
Rate of automation: 34%
Production capacity: 12,000 units/month as of July 2003 (can be increased)
Models produced: CVT1: 1 type
CVT3: 5 types

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2.2. 組立工程の流れ

Fig. 1に，組立ラインの工程の流れを示す．各サブラインで組み立てた構成部品をメインラインへ移し，順次CVTユニットに組み立てる．

完成したユニットを，ファイナル組立ラインで性能試験し，出荷準備工程を経て出荷する．

2.2. Flow of assembly processes

The flow of the assembly line processes is outlined in Fig. 1. The constituent parts assembled in the sub-assembly processes are transferred to the main line and assembled into the CVTs in turn. Assembled CVTs undergo performance tests in the final test stage of the assembly line and are then prepared for shipment.

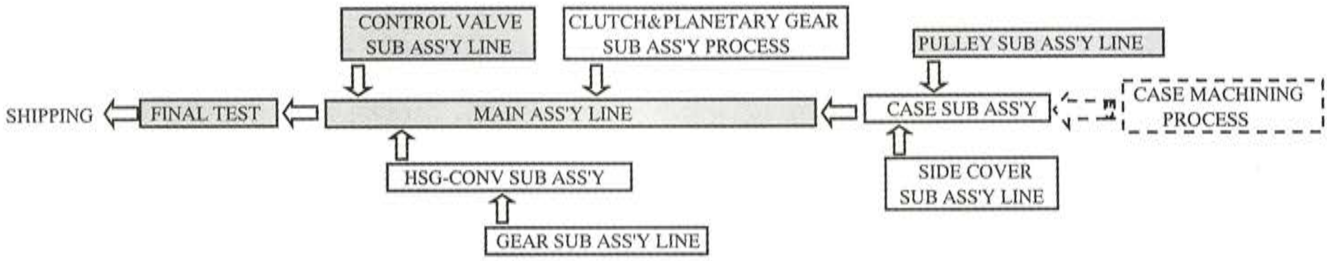


Fig. 1 Flow of CVT assembly processes

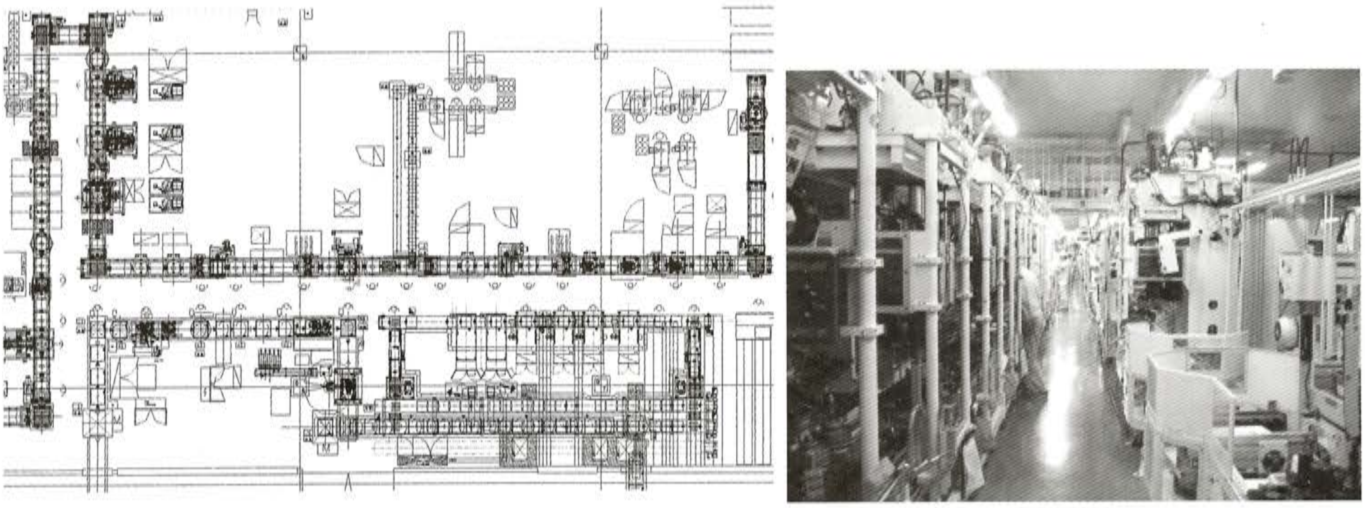


Fig. 2 Part of the assembly line

2.3. レイアウト

組立ラインのレイアウトの一部をFig. 2に示す．主組立エリアを防塵室内に設け，組立部品への粉塵の進入や夾雑物の混入を防止するとともに，作業環境の改善に努めた．

自動化ゾーンをコンベアを迂回させてライン外に出し，メインラインと主要サブラインの手作業ゾーンとを隣接させた二の字構成とし，作業編成効率を向上させている．

2.3. Line layout

A portion of the assembly line layout is shown in Fig. 2. The main assembly area is enclosed in a dustproof room to prevent dust and foreign matter from getting into assembled parts. This also helps to improve the working environment.

The automated assembly zone is located outside the line, bypassing the conveyor. The manual assembly zones of the main line and the principal sub-assembly lines are located next to each other in a parallel arrangement for improved work organization efficiency.

3. ラインの特徴

以下に，主要ラインの特徴点を紹介する．

3.1. プーリ組立サブライン

PRI側，SECD側の両軸に対して，BRGボールの選択，油圧室の組立，ギヤの圧入，アライメント調整用寸法の測定，ベルトの組付の各工程が，順次並んでいる．

Fig. 3に示すように，組立基準となる搬送治具(以下ワークベース)に各種の可変機能を持たせることにより，軸間や全長の異なる2機種の混流生産を，最小限の設備投資で可能とした．

3.2. コントロールバルブ組立サブライン

コントロールバルブ組立サブラインは，コントロールバルブの組立室と，油圧キャリブレーションや機能評価をするテストラインとで構成している．

夾雑物の進入をより一層防止するため，バルブ組立室を，他の組立エリアから独立させた. (Fig. 4)

CVT1とCVT3の制御システムや油圧ポート位置の統合，数多くの部品の共用などにより，設備投資，所要面積とも大きく削減した．バルブテスタについては，無段取り，全自動化により，共用化を実現した．

3. Features of the Assembly Line

The distinctive features of the major assembly lines are outlined below.

3.1. Pulley sub-assembly line

The processes for selecting bearing balls, assembling the pressure chambers, press-fitting the gears, measuring dimensions for making alignment adjustments and assembling belts are arranged in order on both the primary and secondary shaft sides. As illustrated in Fig. 3, the transfer jig (called the work base) that serves as the assembly reference standard incorporates various adjustable functions, enabling mixed production of the two CVTs, having different overall lengths and distances between pulley shafts, on one line with a minimum investment in facilities.

3.2. Control valve sub-assembly line

The control valve sub-assembly line consists of a control valve assembly room and a test line where pressure calibration and functional evaluations are done. The control valve assembly room is separated from the other assembly areas for more rigorous prevention of foreign matter incursion (Fig. 4).

Both the investment in facilities and the required floor space were greatly reduced by unifying the control systems and pressure port positions of the CVT1 and CVT3 and by sharing many parts in common between the two models. Valve testers have been fully automated and require no setup work, thus facilitating common use for both CVT models.

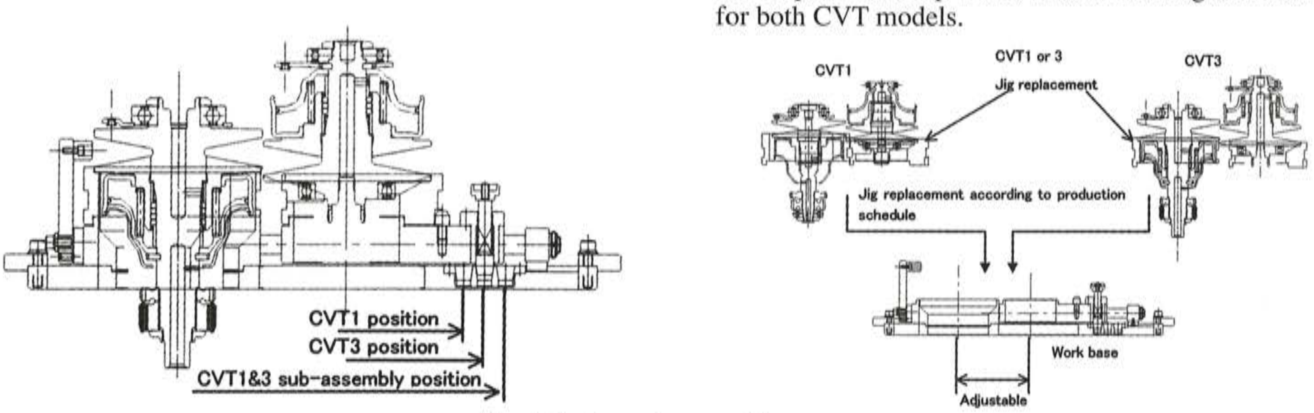


Fig. 3 Pulley sub-assembly process

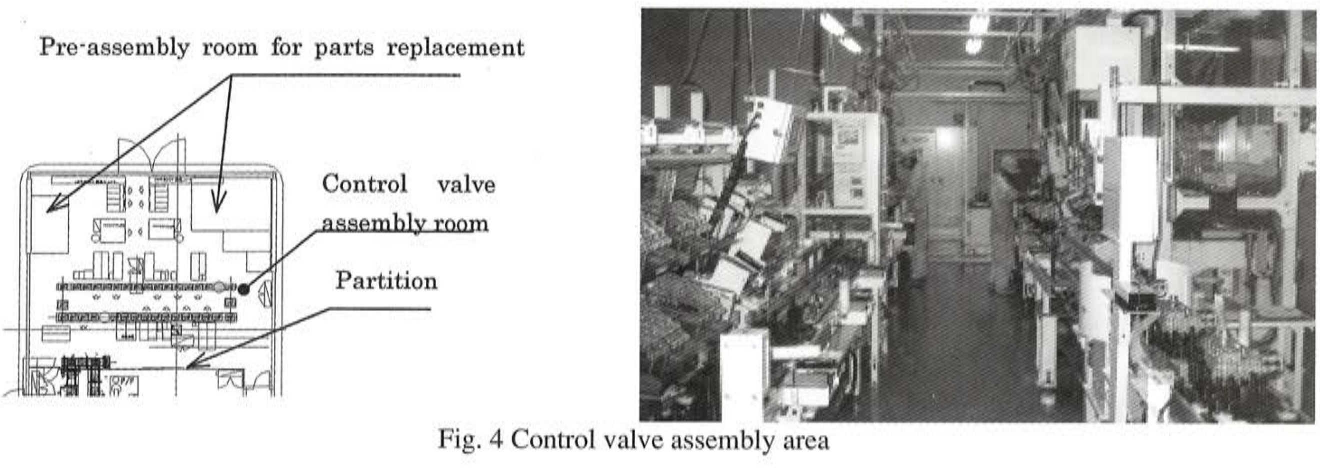


Fig. 4 Control valve assembly area

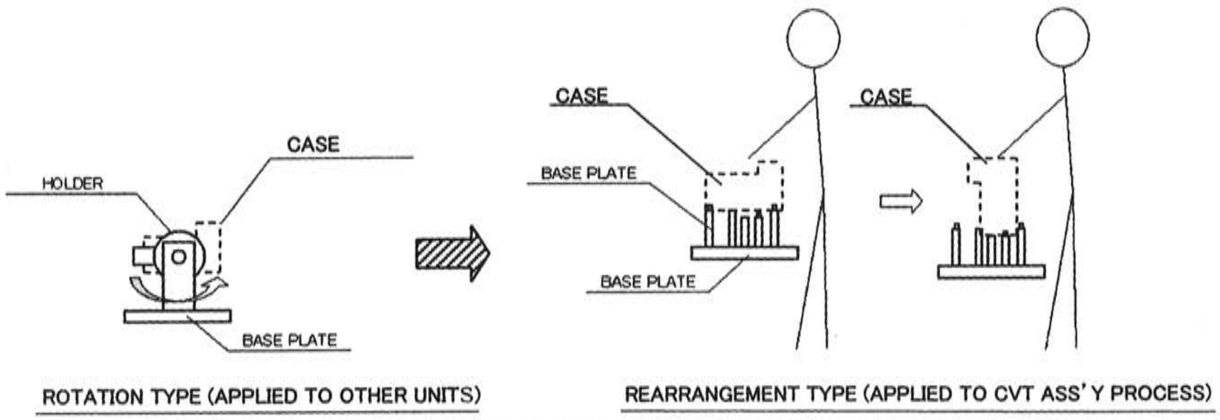


Fig. 5 CVT assembly line fixtures

3.3. メイン組立ライン

母部品となるケースに，各部品を順次組付けていくが，ケースへの組付方向が複数存在するので，ライン上でCVTユニットの姿勢転換が必要となる．このため，ケースに基準座を設け，簡易な載せ換え式のワークベースを用いて，姿勢転換を容易にした．(Fig. 5)

このワークベースは，複雑な機構を持つ回転式のワークベースに比べて，投資削減と各設備の位置決め精度向上に寄与している．

また，組付部品指示システムを採用し，部品の仕様違いなどの不具合を防止した．

3.4. ファイナルテストライン

CVT1とCVT3とでは，入出力軸の軸間距離，油圧検出口や電装コネクタの配置など，外部接続部の構造が大きく異なる．従って，ユニットとの接続を効率化するため，ユニット構造の異なる部位の接続は可変とし，Fig. 6に示すように，CVT1とCVT3を共用でテストできるファイナルテスタ構造とした．

更に，コンバータハウジングの専用ロケート穴を全機種共通ピッチとし，レンジセレクトレバー操作部への設備駆動穴の設置により，無段取り，全自動化を実現している．

これにより，小ロットでの混流生産が実現できた．なお，ワークベースも共用化している．

3.3. Main assembly line

Various parts are attached in order from different directions to the case that serves as the mother component, making it necessary to change the attitude of the CVTs on the line. To accomplish that, the case is provided with a reference base, and a work base that allows simple rearrangement of the case is used to facilitate easy attitude changes (Fig. 5).

Compared with a rotating work base with complex functions, this simple work base is less expensive and helps to improve the positioning accuracy of various pieces of production equipment.

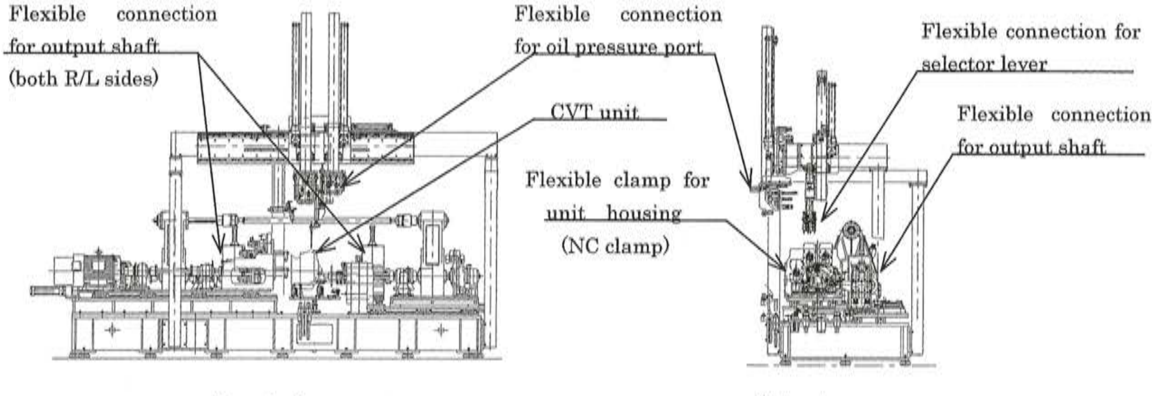
A system of indicating which parts should be assembled was also adopted to prevent incorrect selection of part specifications, among other problems.

3.4. Final test line

The CVT1 and CVT3 differ greatly in terms of the structures of their external connections, including the distance between their input and output shafts, fluid pressure detection port and layout of wiring harness connectors. Therefore, flexible connections were adopted at places where the structures of the two CVTs differ in order to facilitate efficient connections to each CVT model. As shown in Fig. 6, a flexible final tester was constructed so that it could be used for both the CVT1 and CVT3.

In addition, the special locating holes of the torque converter housing share a common pitch on all models, and a flexible connection is provided for preparing the shift slots for the selector lever. These measures achieve full automation without any setup work.

As a result of the foregoing measures, mixed production of both CVTs in small lots became possible. The work base is also shared in common for both models.



Front view Side view

Fig. 6 Flexible final tester

3.5. 品質保証

工程設計の段階でFMEAを行い，過去の不具合や推定不具合を全て洗い出し，設備対策，ポカヨケ，標準作業に織り込んだ．

ライン立上げ前に，各不良モードに対する工程確認会を開催し，計画段階で織り込んだ仕様で不良の出ないライン，または不良が出たとしても検出可能なラインであることを検証した．

なお，主要部品については，購入部品を含め，ロット毎に履歴を管理し，トレーサビリティを向上した．

4. おわりに

本稿ではCVTユニットの組立ラインについて，ライン概要と工程設計について紹介した．しかしながら，生産活動に“人”の要素は不可欠である．“2機種の新規ユニットを同一の生産ラインで同時に立ち上げる”といったこれまでにない組立ラインを立ち上げることができたのは，直接生産に携わる方々の倍する努力があったからに他ならない．この場を借りて関係者に感謝の意を表したい．

立ち上り後2周年を迎えようとしている今，搭載車両の拡大，ユニット性能の向上，原価・品質向上活動などの活動を継続している．今後，生産ラインとしての更なる発展を期待する．

3.5. Quality assurance

Failure mode and effect analysis (FMEA) was used at the process design stage to identify all previous and projected problems so that corrective measures could be fully incorporated in the facilities, in foolproof measures and in standard work procedures.

Before the production launch, process confirmation meetings were held concerning various failure modes. It was confirmed that the specifications adopted at the design stage would prevent defects from occurring on the assembly line and that any defects which might occur could be detected on the line.

For all major parts, including purchased parts, thorough records are kept for each production lot to improve the traceability of problems.

4. Conclusion

This article has presented an overview of the mixed production line for two CVT models and the process design. The human element, however, is still indispensable to production activities. Two new CVT models were launched simultaneously on the same production line, something that had never been done before. The successful launch of the assembly line was due to the sole fact that everyone directly involved in the production of these CVTs redoubled their efforts to accomplish it. The authors would like to take this opportunity to thank everyone involved.

As the second anniversary of the production launch approaches, various activities are still under way to expand the model applications of these CVTs, to improve their performance further and to reduce costs and enhance quality further, among other goals. It is expected that this assembly line will undergo further development in the coming years.

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論文・解説

歯車噛合テスタを用いた噛合非整数次音の計測システム

New Measuring System of Gear Noise at Non-integer Components of Meshing Frequency using the Gear Checker

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Yasuhiro UENISHI　　Masaru ARIYASU

抄　録　本論は，歯車の噛合時に発生する非整数次音の評価方法に関する．筆者らは，先ず歯面の最終仕上げ工程であるシェービング加工によって発生する噛合非整数次音のメカニズムを明確にした．次に，2歯面噛合テスタにFFT(高速フーリエ変換)機能を付加した評価システムを使えば，非整数次音の原因となる歯面の周期的なうねり成分が抽出可能であることを明らかにした．さらに，この評価システムは，噛合伝達誤差測定機より安価で実用性が高く，非整数次音を精度よく評価できるシステムであることを示した．

Summary　This paper dealt with an inspection method of gear noise at non-integer components of meshing frequency. First, it was confirmed that this noise was generated by tooth surface error formed by the vibration of shaving machine tools in shaving process. Second, we proposed the introduction of FFT processor in a gear checker system in order to measure the periodic component of tooth surface error. The gear with this tooth surface error was estimated by this system. On the other hand, the meshing transmission error (MTE) and the vibration on the gear housing were measured. From comparing these results, it is shown that the proposed measuring system is effective to estimate the gear noise at non-integer components of meshing frequency.

1. はじめに

最近，自動車用の歯車製造ラインでは，多種多量に対応した生産技術力が求められており，特にコスト低減のため，加工能率の向上や工具寿命の延長の取り組みがなされている．

特に熱処理前の歯面仕上げ工程であるプランジ方式シェービング(以降シェービングと呼ぶ)工程においては，安価で低騒音のギヤを製造するため，シェービングカッターのリグラインドや歯形管理の方法などに，各社が様々なノウハウを有している[1-3]．

しかしながら，このシェービング工程で，加工能率の向上のためにカッター回転数を高速化するなど加工条件を厳しくさせると，この工程で使用するシェービング盤の治具や芯押台の固有振動の影響が歯面に転写され，トランスミッションに組込んだ状態で非整数次音が発生することがある[4]．そして，一般的な歯車製造ライン(Fig. 1)の最終工程に設置されている通常の噛合テスタや歯車測定機では，この非整数次音を評価することは困難[5,6]であり，未だ当該発生周波数に関して種々評価した例がない．

そこで筆者らは，隣接している歯面に渡って周期的に転写されている歯面誤差を抽出するため，従来の2歯面噛合テスタにFFT機能を追加し，周波数軸上の振幅成分の加算平均を用いる手法を導入した．その結果，非整数次音の原因となるうねりを抽出することに成功した．さらに，本方式で抽出された値とロータリーエンコーダ方式の噛合伝達誤差検出装置から得られる値とを比較検討し，本方式を使えば十分な精度で非整数次音が評価できることを確認した．

1. Introduction

The gear productions for cars require the high-technique of production engineering. Therefore, it is important to improve processing efficiency for cost reduction at each company's. Recently, at the plunge shaving process, which is finish process of tooth surface before the heat treatment, various know-how's are needed for low cost[1,2] and low noise gear productions, which make frequency peak, such as the "meshing frequency" generated gear meshing smaller.

However, in this process, tooth surface may be effected by a periodic motion by vibration in a natural mode of the machine (shaving machine) when improving the cutting conditions for the improvement of processing efficiency. Consequently, it seems that the non-integer sound of meshing frequency is generated when meshing the gears[3,4]. By the way, it is considered that is difficult to evaluation at the gear checker in the last process of the gear production line (Fig. 1) or the tooth surface measuring system.

Therefore Car Company's engineers are troubled by this noise. And there is no report evaluated about the non-integer components of meshing frequency noise.

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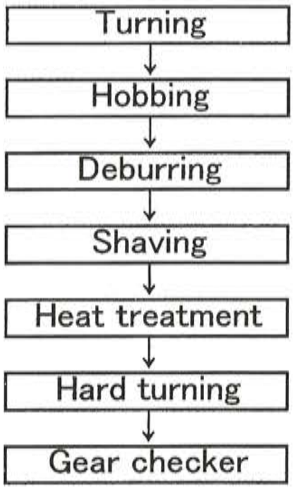


Fig. 1 Processes of gear production

2. 非整数次音の発生メカニズム

歯車噛合時の振動発生メカニズムをFig. 2に示す．強制外力として，以下の2項がある[7-9]．
① ばね剛性の変化（$-K(t)\triangle$）
② 歯面誤差による強制変位（$F(t)$）

一般的には，噛合率を増大させるなどにより，ばね剛性の周期的な変化を低減するなどして，噛合整数次の振動を低減している．

一方，歯面加工時に歯面と工具との相対位置が正常な位置から変化すると，歯面にはうねり成分が転写される．しかし，その変化がランダムであれば，歯面の転写成分もランダムとなり，歯面誤差に起因する振動はホワイトノイズ的になり，全体の振動レベルは大きくなったとしても，特定の周波数の振動が目立つということはない．ところが，歯面と工具との相対位置がある特定の周波数で振動すると，周期的なうねりが歯面に転写されることになり，非整数次音が発生する．

シェービング工程における歯車の回転数をn(rpm)，歯車の基礎円直径をd_g(mm)，シェービングカッターの振動数をf(Hz)とすると，歯車の歯面上に転写される振動成分の1周期の法線方向長さl(mm)は，以下のようになる．

$$l=(2\pi n/60)\cdot(d_g/2)/f \qquad (1)$$

また，歯車の軸直角モジュールをm(mm)，軸直角圧力角をα，正面噛合率をeとすると，一歯の法線方向の噛合長さLは以下のようになる．

$$L=e\cdot m\cdot \cos(\alpha) \qquad (2)$$

法線ピッチt_eは以下の式で表わされる．

$$t_e=m\cdot \cos(\alpha) \qquad (3)$$

The author's clarified the follows. First, it was confirmed that this noise was generated by tooth surface error formed by the vibration of shaving machine tools in shaving process. Second, we proposed the introduction of FFT processor in a gear checker system in order to measure the periodic component of tooth surface error. The gear with this tooth surface error was estimated by this system. On the other hand, the meshing transmission error (MTE) and the vibration on the gear housing were measured. From comparing these results, it is shown that the proposed measuring system is effective to estimate the gear noise at non-integer components of meshing frequency.

2.Generating Mechanism of Non-integer Components of Meshing frequency

A model of the vibration at the gear meshing is shown in Fig. 2. There are two items as meshing force, change of tooth spring rigidity (-K (t)D) and the tooth error (F (t))[5,6]. The meshing frequency of gear noise (vibration at the gear meshing: this frequency = (number of gear teeth) × (gear rotation speed)) has been decreased, because of smaller change of tooth spring rigidity as increasing of gear meshing ratio.

On the other hand, the tooth surface error is generated the motion (vibration component) at the change of relative motion between gear and shaving tool at gear cutting. However, in the case of random vibration, the tooth surface error is generated by the random motion, so that the noise due to tooth error component at gear meshing is white noise. Although, its over all level becomes large, the noise is not generated on a certain specific frequency.

However, if the relative vibration between tooth surface and a shaving tool occurs on specific frequency, it will be considered that a periodic wave is generated on tooth surfaces.

If the cutting rotation of gear is n (rpm), base circle diameter of gear is d_g (mm) and vibration of tool is f (Hz) at the shaving process, the length l (mm) of vibration component indicated on tooth surface at one cycle will be as follows.

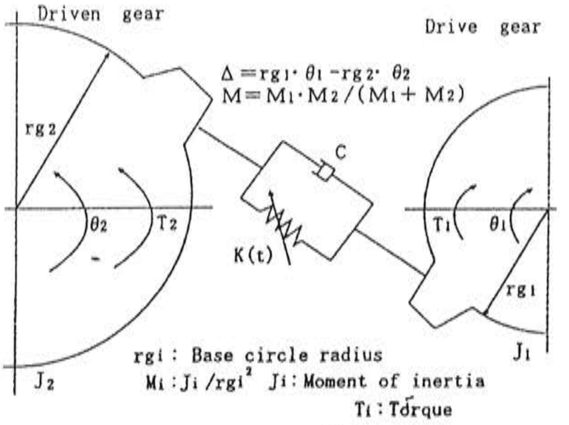
$$l=(2\pi n/60)\cdot(d_g/2)/f \qquad (1)$$

Here, module is m, pressure angle is α and meshing ratio is e at gears pair. The meshing length L of one meshing is as follows.

$$L=e\cdot m\cdot \cos(\alpha) \qquad (2)$$

The normal base pitch t_e (mm) is as follows.

$$t_e=m\cdot \cos(\alpha) \qquad (3)$$



$$M\ddot{x} + C\dot{x} + \bar{k}x = -\tilde{k}(t)\bar{\Delta} + \tilde{F}(t)$$

M : The equivalent mass
C : The attenuation coefficient
K : The mesh stiffness
Δ : The relative displacement
— : Average
～ : Variation

Fig. 2 The analytical

従って，この歯車の噛合音の次数Hは以下となる．

$$H = t_e/1 \quad (4)$$

従って，Hが整数となるチャンスは殆ど無く，シェービング盤の振動に起因する噛合音は，殆どが非整数次の振動となる．

ここで，L<1の場合には，一歯面に転写されるうねり成分は，振動の1周期に満たないため，一歯面上に凸が残る歯面と残らない歯面とが存在することになる．また，凸の位置も歯先から歯元まで，どの位置にでも発生する可能性があり，各歯面形状を個々に調べても，非整数次の振動が発生するかどうかを判別することは難しい．

また，L>1の場合には，各歯面に何らかの凸が観察されるが，歯先から歯元のどの位置に発生するのかがその歯によってばらつくので，その凸部が非整数次の周期的振動の原因になるかどうかを簡単には判定できない．

隣接歯面間の周期的なうねりを測定する手法として，噛合テスタを用いる手法が有効であると考えられる．噛合テスタとは，マスターギヤと呼ぶ高精度歯車と，評価歯車とをノーバックラッシュで噛合わせ，回転させたときに発生する軸間距離の変化を計測する試験機で，歯面に垂直な法線方向のうねり振幅をAとすれば，軸間距離の変化として下記式のEの量が計測できる．

$$E = A \cdot \sin(\alpha) \quad (5)$$

The meshing vibration component at gear meshing is H.

$$H = t_e / 1 \quad (4)$$

Therefore, H becomes non-integer components of meshing frequency mostly. Here, because the motion generated on the tooth surface is not finished within a cycle of vibration in the case of L<1, the convex or concave of error wave on the teeth does not remain. And, the position of a convex will be predicted (this position is decided by the phase of the motion). Even if evaluating the profile of one tooth, the occurrence of non-integer components cannot distinguish. In case of L>1, although the convex and concave of non-integer components on the teeth are observed, the phase of error wave will change at each tooth.

The technique using gear checker considered as the technique of measuring the existence of the periodic surface error. Gear checker is an evaluating machine, which measures change of the center distance generated due to meshing with a master gear with no backlash. Here, Eq. (5) shows the relation between surface error amplitude A and measuring value E.

$$E = A \cdot \sin(\alpha) \quad (5)$$

Moreover, FFT processing of the signal is used as the technique of detecting the periodicity of the E, with performing an addition average on a frequency do main. That is, the motion without periodicity become $1/SQRT(N_1)$, and only the periodicity motion can be extracted by performing the addition average of N_1 time. Here, if the gear's rotation in gear checker is N_1, evaluating frequency is as follows.

$$f_1 = f \cdot n_1 / n \quad (6)$$

3. Measurement Principle and Method

In general, at the last process of gear manufacturing line, there is gear checker in order to measures run-out, nick and over ball diameter (tooth size). That is the change value of center distance between the test gear and master gear (whose accuracy is 4th class (DIN) in our company) is measured with no backlash meshing operations.

3.1. New Evaluation System Including FFT

The new evaluation system with FFT make it feasible to measure not only the run-out, the nick, and over ball diameter of gear, but also the periodic surface error of teeth. The outline figure is shown in Fig. 3.

そして，Eの信号をFFT処理し，連続的な測定データを周波数軸上で加算平均すれば，その信号周期を検出することができる．すなわち，N_1回の加算平均値のうち，周期性のない成分は$1/N_1^{1/2}$となって消えていくので，周期性のある成分のみを抽出することができる．

ここで，噛合テスタにおける歯車の回転数をn_1とすれば，周波数軸上に発生する周波数f_1(Hz)は以下の式で計算できる．

$$f_1=f(n_1/n) \tag{6}$$

3. 測定の原理と方法

現在，一般的に用いられている噛合テスタでは，製造された歯車とJIS0級(JIS B 1702)程度の高精度なマスターギヤとをノーバックラッシュ状態で噛合わせ，中心間距離の変動値から以下を定量的に検出している．

① 歯溝の振れ
② 打痕の有無
③ オーバーボール径

3.1. 新評価方式

非整数次振動を評価するため上記の定量検出値に加えて，

④ 非整数次周波数の振動

も評価できる仕様とした．その模式図をFig.3に示す．

なお，今回使用したベース機は，大阪精密製噛合テスタであり，測定条件は以下のとおりである．

測定歯車の回転数：12rpm
負荷トルク　　　：2.6N･m

FFT分析機には小野測器製CF-360を用い，窓関数はフラットトップを選択した。そしてランダムに計測した16回の加算平均値のデータを評価値とした．このデータは中心間距離の変動値を示しているので，実際にはこのデータを噛合線方向成分に換算し，新方式の評価値として用いている．

3.2.MTE方式

歯車の噛合伝達誤差をMTE(Meshing Transmission Error)と呼び，入力側の歯車の回転速度を一定としたときの出力側歯車における回転変動量を表す．

噛合伝達誤差は，歯の曲げ剛性変化に起因する変位と歯面誤差による変位との合成値であり，実機運転での噛合起振力の原因となる(10-14)ので，MTE方式は歯車噛合音の評価指標として広く使用されている(15-17)．

今回，新方式の上記噛合テスタ方式に加えて，ロータリーエンコーダを使用したMTE測定機でも比較評価した．MTE方式の模式図をFig.4に示す．

なお，MTE測定機には，ロータリーエンコーダ分解能が4,500P/Rの国際計測機製を使用し，以下の測定条件で計測した．

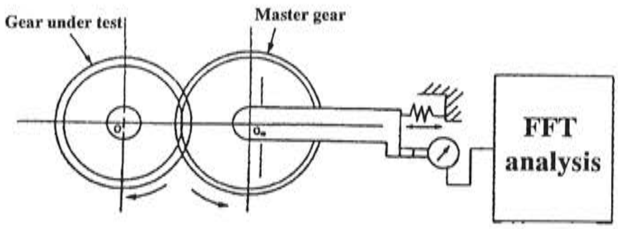


Fig. 3 Gear checker (The new type)

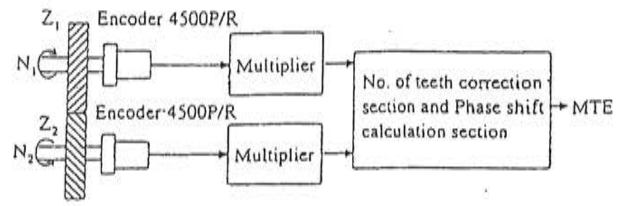


Fig. 4 Gear checker (MTE type)

The base gear checker is made from Osaka-Seimitsu-Kikai. So, the measurement conditions are as follows.
Measurement gear rotation: 12 rpm
Load torque: 2.6 N･m

FFT analysis machine (CF-360) is made from Ono-Sokki. The measuring condition is the addition at average random 16 times with a flat top window. The evaluation value A is the change of center distance (A=E/sin α) in meshing line direction.

3.2. MTE (Meshing Transmission Error)

Recently, MTE measuring is often used as evaluation of meshing conditions of gears(7-11), because Meshing Transmission Error in rotation transfer is caused meshing vibration forces due to spring variation of tooth and tooth error. The outline figure is shown in Fig. 4. Therefore, we also use the MTE system made from Kokusai-Keisokuki, including encoder specification 4500 P/R. The measurement conditions are as follows.
Measurement gear rotation: 15 rpm
Load torque: 1 N･m

測定歯車の回転数：15rpm
負荷トルク　　　：1N·m

3.3.台上試験での振動測定

上記二種類の測定方法における評価値と，実機のトランスミッションケース振動値との相関を明確にするため，同一の歯車を使用して比較試験した．

ケース振動値として，車内音との相関が高いシフトケーブルブラケットのケーブル方向加速度を測定した．測定条件としては，高速道路を100km/hで走行している運転を想定し，以下とした．
入力回転数：3,000rpm
負荷トルク：29.4N·m

3.4.計測歯車の諸元

供試したはすば歯車は，当社FF用5速ATの1次減速噛合部の被動側歯車であり，そのATの縦断面をFig.5に,歯車諸元をTable 1に示す．なお，歯車の材質はSCr420で，浸炭ソルト焼入れが施されている．

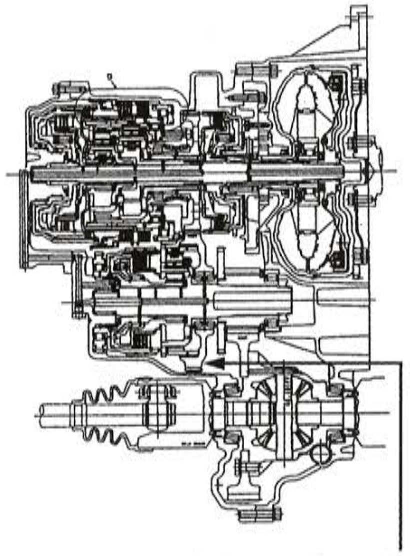


Fig. 5 Cross section of automatic transmission

使用した歯車4ケのうち，2ケはリジッドな治具で加工した歯車，他の2ケは剛性小の治具で加工した歯車とした．リジッドな治具とは、Fig.6-a,bに示すように，シェービング盤の固有振動が歯車に転写しにくくなるように，治具自体の質量(102kg)を約20%軽量化するとともに、芯押台を点支持から面支持に変更し，治具の取り付け剛性を約2.3倍に増大している．また加工条件は以下のとおりである．
加工回転数：230rpm
送り：0.6mm/min

なお，以降，固有振動が転写されている歯車を「GOOD歯車」、転写されている歯車を「NG歯車」と呼ぶ．

3.3. Vibration Measuring Method of Transmission Assembly

It is very important to grasp correlation between value measured by gear checker and transmission assembly vibration value. So, vibration of housing of transmission assembly included the measuring gears is measured at meshing frequency and non-integer components of meshing frequency by the acceleration pickup, which is attached by the shift cable bracket on a transmission housing. There is high correlation between vibration on shift cable direction and car interior noise. Here, the measurement conditions are input rotation: 3000 rpm and load torque: 29.4 N-m, that is operation on a highway (about car speed 100 km/h).

3.4. Dimensions of Gears

The Helical gears of this research are driven gears of 1st meshing in automatic transmission, whose cross sectional view is shown in Fig. 5. The dimension of gear is shown in Table 1. The material of gear is JIS SCr420, so it is hardened by carbonized with salt quenching. The four kinds of gears used in this research. Among those, two pieces; peculiar vibration of shaving machine may not be transcribed on the tooth surface of gear, which are held by the rigidly jig (mass of jig: 102 N, shown in Fig. 6-a). The other two pieces; peculiar vibration of shaving machine was transcribed on the tooth surface of gear, which are held by point support of simple jig on center (mass of jig: 121 N, shown in Fig. 6-b). The cutting conditions of shaving process are as follows.
Cutting rotation: 230 rpm
Cutting feed: 0.6 mm/min

Comparing with simple jig, the rigidly jig attains about 20% lightweight and 2.3 times stiffness (in gear radius direction).

Here, the gears on which peculiar vibration is not generated are called "GOOD gear", and the generated others are called "NG gear."

Table 1 Dimension of gear

Module		1.75
Pressure angle	(deg.)	17.5°
Number of teeth		89
Helix angle	(deg.)	31.5°(LH)
Pitch circle dia.	(mm)	182.668
Reference circle dia.	(mm)	171.329
Whole depth	(mm)	5.5
Outer dia.	(mm)	186.1

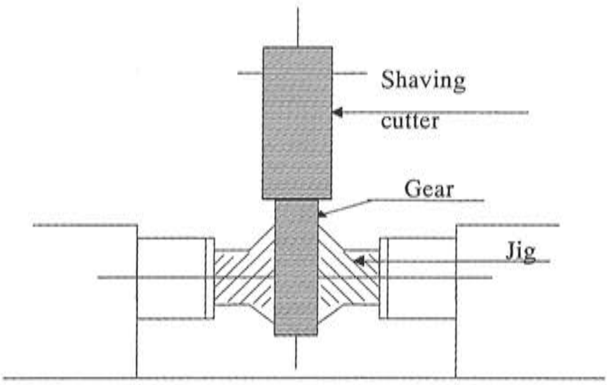


Fig. 6-a The rigid clamp processing

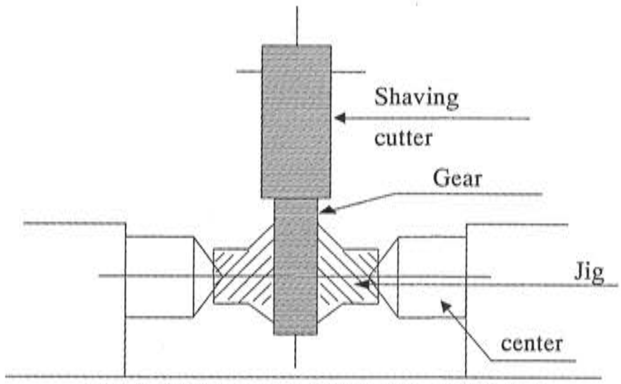


Fig. 6-b The center addition construction

4. シェービング盤の振動特性

シェービング加工時の一般的な加工軌跡とシェービング盤の振動レベルをFig.7に示す．振動レベルを測定するため，Fig.8に示すように，シェービング盤の芯押台部に加速度ピックアップを設け，鉛直方向の振動を計測している．振動レベルはシェービング加工時における時間あたりの切削体積に影響される傾向にあり，切削体積が増加する加工完了位置直前(Fig.7 finish時)に，振動が極めて大きくなる．

シェービング加工の完了直前での振動スペクトラム波形をFig.9に，被加工歯車の歯面法線方向の加速度に対する芯押台支持部の鉛直方向加速度の伝達関数をFig.10に示す．

治具の軽量化や取り付け剛性の増大により，シェービング盤の振動，および伝達関数のオーバーオール値の両者とも，GOOD歯車に対してNG歯車の方が高くなっていることが判る．また，NG歯車ではシェービング盤の共振周波数付近に伝達関数のピーク周波数があり，非整数次噛合音を悪化させていることも判る．

従って、シェービング加工治具は，軽量化と高剛性化に注力して設計されるべきであると考える．

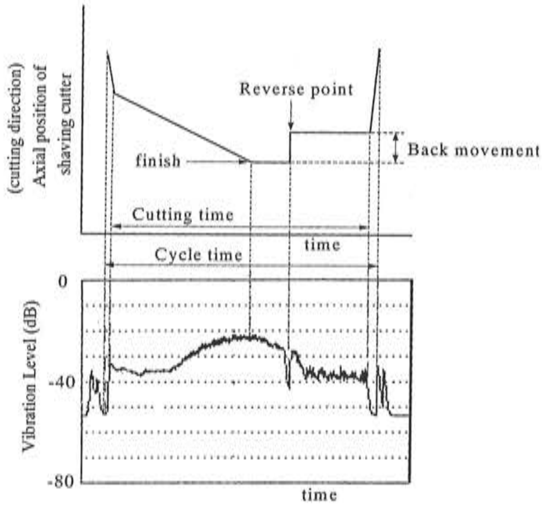


Fig. 7 Vibration mechanism of shaving

4. Vibration Characteristics of Shaving Machine

In this report, there are two kinds of gears ("GOOD gear" and "NG gear"). The former has no tooth surface error due to shaving process using the rigidly jig. The latter has tooth surface error due to the simple jig.

Here, the relation between the cutting process and the vibration at shaving is shown in Fig. 7. More, vibration of shaving machine is measured in direction of perpendicular by the acceleration pickup attached on head stocks on the bed of shaving machine, as shown in Fig. 8. Although, the vibration level of shaving depends on the position of shaving cutter, the vibration indicates high level near the finishing position of processing where the amount of cutting volume per time become maximum. Therefore, the spectrum of vibration of shaving at finishing position of processing is shown in Fig. 9. The transfer function on shaving machine between the jig and the gear are shown in Fig. 10. From these figures, it is clear that vibration level of "NG gear" is higher than that of "GOOD gear" on both vibrations in shaving process and overall of transfer function. These differences between rigidly jig and simple jig is due to lightweight ratio about 20% and rigidly ratio about 230%.

Furthermore, it is also clear from Fig. 10 that there are similar peak tendency between vibration in shaving and transfer function of shaving machine at "NG gear". This peak indicates the vibration of coupled modes occurred between shaving cutter and gear in shaving, because the peak frequency is similar to peculiar vibration of shaving machine, jig and gear. Therefore, the design of the jig on the shaving machine can be judged from the ratio between lightweight and support rigidity.

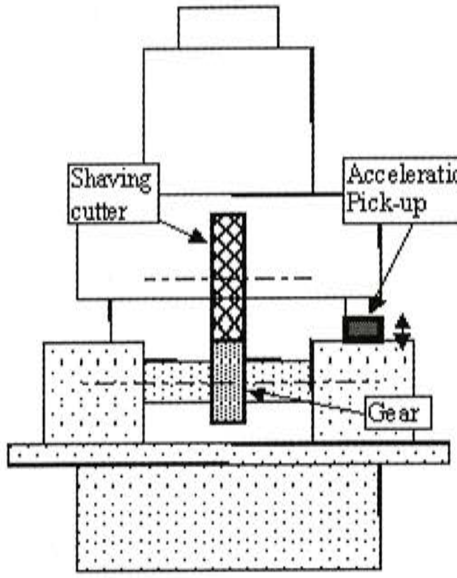


Fig. 8 Measuring position of acceleration pick-up

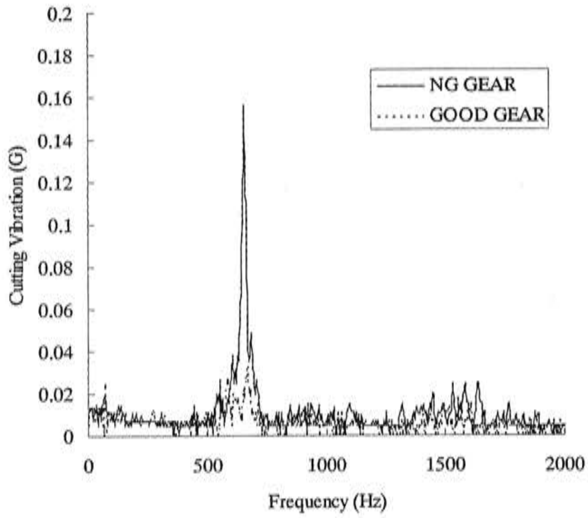


Fig. 9 Spectrum of cutting vibration in shaving

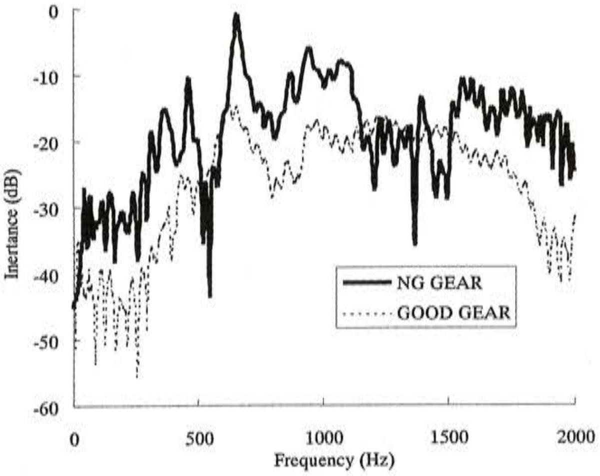


Fig.10 Transfer function on the shaving machine

5. 各評価方式の比較

5.1.新評価方式

5.1.1. シェービング盤の振動

GOOD歯車とNG歯車における回転方向の振動変位の周波数分析結果をFig.11に示す．

この図では，NG歯車にはシェービング盤の固有振動数に相当する33Hz付近にピークがあり，Fig.9の振動ピーク周波数650Hzに回転数比12/230（=測定機のマスターギヤ駆動回転数／シェービング盤のカッター回転数）を掛け合わせるとよく一致し，(6)式の関係が正しいことが判った．

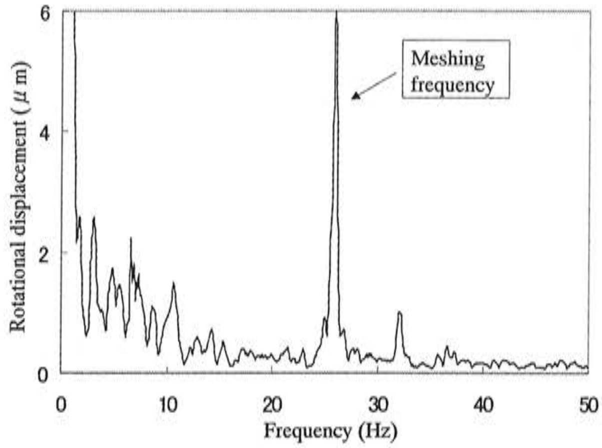
この方式を噛合テスタとして導入し，シェービング盤の振動レベルとケース振動との関係を品質管理値とするため，両者の振動変位間の相関を求めた．

5. Characteristics of Each Evaluation System

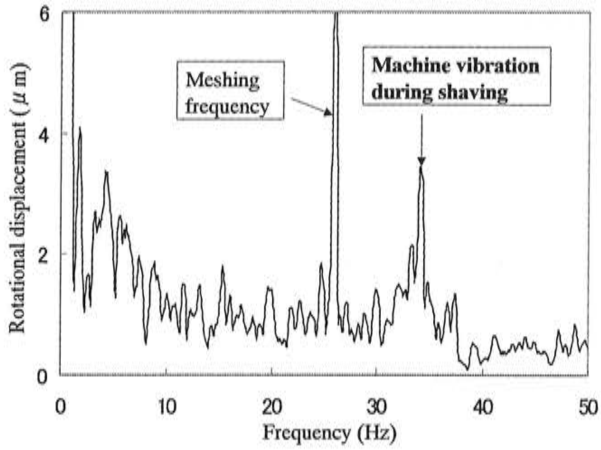
5.1. New Evaluation System using gear checker

5.1.1. Vibration Evaluation of Shaving Machine

First, we evaluate the generated tooth surface error caused by peculiar vibration of shaving machine by means of new evaluation system for "GOOD gear" and "NG gear". The result (frequency analysis result of the rotation direction A by Eq. (5)) is shown in Fig. 11. In the case of "NG gear", peculiar vibration generated by a shaving machine can be seen at (near 33Hz) and this frequency is equal to rotation ratio (approximately 12/230) of the peak frequency at 650Hz shown in Fig. 9. This ratio is equal to ratio of cutting rotation in shaving to measuring rotation. Therefore, it is clear that it is possible to evaluate tooth surface error generated the peculiar vibration of shaving machine in high accuracy. Therefore, the Eq. (6) is confirmed. Furthermore, it is necessary that correlation among the vibration of shaving machine, measured value by the new evaluation system and the vibration on housing (transmission assembly) is proved.



a) Good type gear



b) NG type gear

Fig.11 Relation between rotational displacement and frequency (In case of new measuring system)

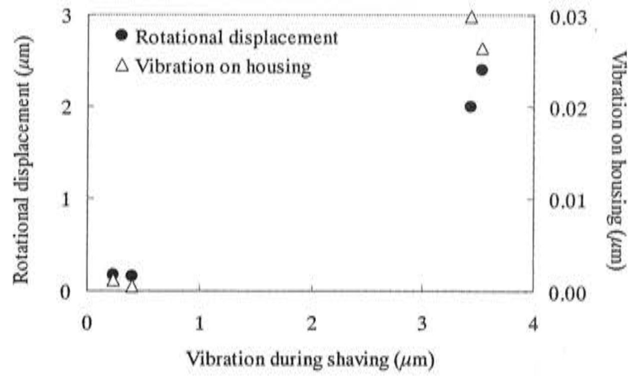


Fig. 12 Relation between vibration during shaving, rotational displacement of new measuring system and vibration on T/M housing

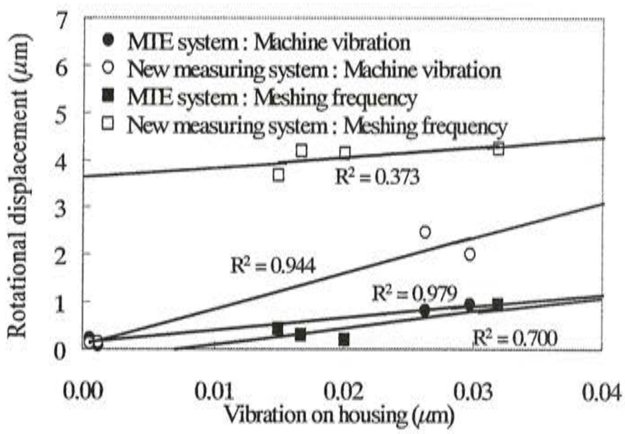


Fig. 13 Relation between vibration of T/M housing and rotational displacement

5.1.2. 新評価方式の正当性

Fig.12に，シェービング盤の振動に対する，新評価方式における歯面法線方向振動，および実機でのケース振動を示す．両者ともシェービング盤の振動とよく相関しており，最小自乗法による相関係数(R^2)は0.9以上である．

5.2.MTE方式

MTE方式での測定結果をFFT分析した結果からも，新評価方式と同様，シェービング盤の固有振動数と同じ振動が確認されるとともに，上術の相関係数が0.9以上あることが確認できた．従って，MTE方式でも非整数次の振振動評価が十分な精度で可能であることが判った．

5.1.2. Correlation between the shaving vibration, new evaluation vibration and vibration of transmission assembly

The correlation among the shaving vibration, vibration measured by new system and vibration of transmission assembly is shown in Fig. 12. In this figure values of vibration measured by new evaluation system and vibration of transmission are converted into equivalent values of the shaving machine vibration. As shown in this figure, it is clear that measured value of the new evaluation system and vibration on housing are also high value when shaving vibration indicated high values. The coefficient of correlation indicates more than 0.9(R^2).

5.2. Evaluation by MTE System

The frequency component, which is equivalent to peculiar vibration of shaving machine, can be also observed from measurement with MTE system. The coefficient of correlation also indicates more than 0.9. By MTE measuring system, it can also be checked that evaluation of the non-integer components of meshing frequency is possible enough.

6. 新評価方式の有効性

新評価方式の有効性を検証するために，ハウジング振動値に対するそれぞれの方式で得られた回転方向変動値の相関関係を，シェービング盤の固有振動に相当する周波数(かみあい非整数次周波数)および噛合１次周波数にて評価した図をFig. 13に示す．

これらより，シェービング盤の固有振動の評価は，両方式共，相関係数R^2＝0.9以上を満足しており，どちらでも評価可能であることがわかる．これは，駆動，従動両歯面噛合である新評価方式での測定時の噛合状態が，シェービング加工時と同様であるためであり，また，1歯面噛合であるMTE方式においては，トランスミッション駆動時と噛合状態が同様なためであると考えられる．

しかしながら，噛合１次周波数を評価する場合，特に新評価方式では，相関係数がR^2＝0.373と極めて低く，ハウジング振動値との相関性が低いことを示しており，本評価方式による評価は難しいと判断できる．これは噛合整数次の周波数に関しては，シェービング盤の振動の影響(歯のうねり)が小さく，歯車対の噛合動特性(歯のたわみ)の影響を受けるためであると考えられ，すなわち，駆動，従動両歯面噛合である新評価方式での計測時と，1歯噛合であるトランスミッション運転時の噛合状態が異なるためであると考えられる．

7. まとめ

シェービング工程で発生する振動に起因する噛合非整数次音の発生メカニズムを明確にし，噛合テスタを用いた当該音の新評価法を考案し，以下の結論を得た．

(1) 非整数次振動の発生を避けるには，シェービング加工治具を軽量・高剛性化し，加工時の共振を回避することが有効である．

(2) 非整数次振動の評価には，一方の歯車に高精度なマスターギヤを用いて噛合わせ，両歯車の中心間距離の変動を測定するとともに，FFT分析する新評価方式が高精度で有効な方式である．

(3) 非整数次音だけでなく，噛合周波数音までを評価するには，MTE方式を採用する必要がある．

最後に，本研究を進めるにあたり，多大なご指導，ご協力を頂いた同志社大学工学部 助教授 廣垣俊樹工学博士，同教授 青山栄一工学博士，および当社経営企画部 穴田能文氏をはじめ，関係部門の多く方々に深く感謝する．

6. Efficiency by using New Evaluation System

In order to investigate the efficiency by using a new evaluation system based on gear checker at the non-integer components of meshing frequency (peculiar vibration of shaving machine) and meshing frequency, the correlation among housing vibration and values on these system are shown in Fig. 13. These show that the values by each system have satisfied more than correlation coefficient R^2=0.9 at peculiar vibration of shaving machine.

This is the reason why meshing condition under running with no-backlash meshing is similar to both new evaluation system and cutting in shaving. Moreover, in the case of the MTE system, it is thought that meshing condition is also similar to running of transmission in motor car. However, when evaluating integer meshing frequency component, it is confirmed that evaluation value of the new system has very low correlation coefficient R^2=0.373 to housing vibration. According to this new evaluation system, it is difficult to evaluate the meshing frequency component. The integer components of meshing frequency cannot be easily influenced of the vibration of shaving machine, but can be influenced of meshing characteristics of gears. Therefore, it is thought that this is the reason why the meshing condition is different between new evaluation system with no-backlash meshing and transmission running with backlash meshing.

7. CONCLUSION

(1) The design of the cutter holding jig on the shaving machine can be judged from the ratio between lightweight and support rigidity for avoiding the non-integer components of meshing frequency occurred on transmission in running.

(2) The measuring results by new evaluation system are high accuracy on evaluating the non-integer components of meshing frequency caused by cutting vibration in shaving.

(3) If it is necessary to evaluate not only the non-integer components of meshing frequency but also the integer components of meshing frequency, it is necessary to adopt a MTE system.

Finally, the authors would like to thank Dr. T.Hirogaki and Dr. E.Aoyama at Doshisha University and Y.Anada of the Management planning department and numerous other individuals for their invaluable advice and copperation in connection with this research.

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論文・解説

AT制御ソフトウェアにおける品質開発の体系化と適用

Systematization and Application of the Quality Development Process for AT Control Software

佐藤 雅行*
Masayuki SATO

抄 録 近年，自動変速機には，小型軽量化と同時に高度な変速性能が求められており，制御システムの複雑さが増大している．制御システムはソフトウェアの形でマイクロコンピュータに組み込まれている．最近は，制御ソフトウェアの規模と複雑さが，急速に増大しているため，ソフトウェアの品質がきわめて重要な課題となっている．高品質なソフトウェアを確実に開発するためには，緻密に定義されたプロセスを厳密に実行管理することが不可欠である．本稿では，制御ソフトウェアに関する品質開発プロセスの体系化の考え方と具体的活動の概要，効果などについて説明する．

Summary The need for more compact and higher performance automatic transmissions has resulted in more sophisticated electronic control systems. The software installed in a microcontroller chip constitutes the major part of an AT control system. Rapid increases in the scale and complexity of control software have made software quality a crucial issue. The development of high-quality software requires rigorous execution and management of an elaborately defined development process. This article describes the concept and specific activities undertaken to systematize the quality development process for AT control software. Some activity examples and their benefits are also presented.

1. はじめに

近年の自動車においては，排気，燃費，快適性といった諸性能を高次元にバランスさせることが必要となっている．これに伴い，自動変速機(以下ATと呼ぶ)に対しては，小型軽量化やマニュアルシフト制御のような付加機能を実現しながら，レスポンスの向上や変速フィーリングの改善といった性能向上も実現するなど，相反する要求への対応の必要性が増大してきている．要求される機能・性能を達成するための手段のひとつとして，ATの電子制御化が推進され，最近ではワンウェイクラッチレス化，クラッチ油圧の直動化など，より高水準の機能・性能を達成している[1-2]．ATの電子制御システムにおける制御ロジックは，AT電子制御ユニット(以下ATCUと呼ぶ)にソフトウェアとして実装される．制御内容の急速な高度化に伴い，このソフトウェアの大容量化，複雑化が加速度的に進行しており(Fig. 1)，この傾向は今後も継続するものと予測される．[3]

また一方で，こういった製品をより早い時期に市場に投入し，社会の要請に応える必要性も非常に高く，種々の開発ツールを利用することによる開発効率向上へのアプローチも実践されている．[4-6]

1. Introduction

In recent years, vehicles have had to provide the highest possible balance of performance in terms of exhaust emissions, fuel economy and comfort. This has increased the necessity for automatic transmissions (ATs) to satisfy various conflicting performance requirements. These include, for example, improved response and shift feel while achieving a smaller, lighter package and providing additional functionality such as manual shift control.

The application of electronic control to ATs has been promoted as one approach to achieving these contradictory functions and performance capabilities. Recently, higher levels of functionality and performance have been attained through the elimination of one-way clutches, adoption of direct control of clutch pressures and other technologies.[1-2] The control logic of an electronic control system for ATs is implemented as software in the AT electronic control unit (ATCU). The rapid enhancement of control features has dramatically accelerated the increase in software volume and complexity (Fig. 1). It is expected that this trend will continue in the future.[3]

Meanwhile, it is imperative to respond to societal demands by putting improved products on the market as quickly as possible. Various types of development tools are being used in an effort to boost the efficiency of development work.[4-6]

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一般に，極度に複雑化した制御用ソフトウェアには，設計意図どおりに動作しないなど，種々の欠陥が潜在する可能性がある．このような欠陥は，ソフトウェアが人間の思考や意思をプログラムという形にコード化したものであり，その設計・製作過程がきわめて複雑かつ繊細であることに起因するものである．しがたって，無欠陥の制御用ソフトウェアを高い生産性を維持しつつ開発するためには，技術者同士の緊密な連携に加えて，緻密に定義されたソフトウェアの開発プロセスを厳格に管理することが必須となる．

当社ATCU開発部門ではこのような視点から，制御用ソフトウェアの品質開発プロセスに着目し，その改善と体系化を基軸として，ソフトウェア品質の向上を目的とした活動に取り組んできた．本報告では，その具体的方法と効果について論述する．

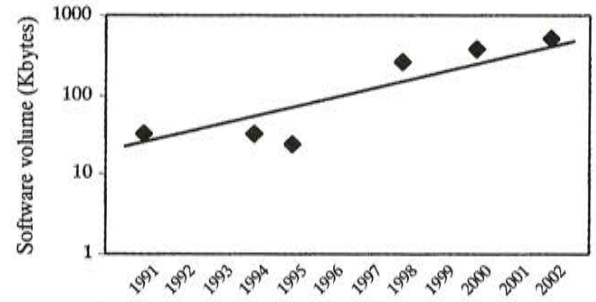


Fig. 1 Trend in ATCU software volume

2. ソフトウェア開発プロセスと課題認識

ATCU開発部門における制御用ソフトウェアの開発プロセスの概要をFig. 2に示す．

制御用ソフトウェアは，制御システムの設計，制御ロジック・ソフトウェアの設計，制御用プログラムの作成とデバッグ，制御ロジック検証と実車試験によるキャリブレーションという一連の工程を経て開発される．制御用プログラムにおける従来の主要な課題は以下のとおりである．

(1) 予定したスケジュールに対し，社内評価用の制御用プログラムの供給遅れが発生する．

(2) 評価用に供給した制御用プログラムで，動作不良が発生する．

これらの現象の発生要因を分析した結果，以下のような場合があることがわかった．

(1) 制御用プログラムの開発中に，要求仕様が頻繁に追加・修正される．

(2) 制御用プログラム開発の進捗状況を正確に追跡できていない．

In general, extremely complex control software may contain various latent defects that prevent the control system from operating as it was designed. Control software represents the codification of human ideas and wishes in the form of a program, and defects originate in the extreme complexity and intricacy of software design and manufacturing processes. Consequently, developing defect-free control software while maintaining high productivity requires close teamwork among the engineers involved. In addition, it also requires rigorous management of the elaborately defined software development process.

In the ATCU development group at JATCO, we have been working to enhance the quality of control software by implementing measures focused on improving and systematizing the quality development process for AT control software from these perspectives. This article describes some of the specific methods adopted and the resulting benefits.

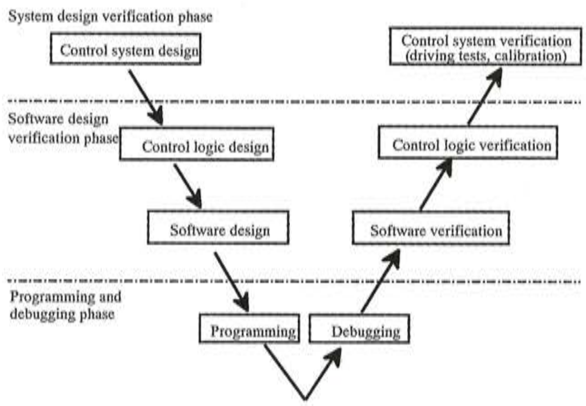


Fig. 2 Control system development process

2. Recognition of Issues in the Software Development Process

An outline of the control system development process in the ATCU development group is given in Fig. 2. An AT control system is developed through a series of steps that include control system design, control logic and software design, control program creation and debugging, control logic verification and control system verification and calibration on the basis of driving tests conducted with an actual vehicle. The main issues that typically occur in the development of a control program include the following:

(1) Provision of the control program for in-house evaluation is delayed beyond the originally planned schedule.

(2) The control program provided for evaluation does not operate properly.

An analysis of the factors giving rise to these issues revealed the presence of the following circumstances:

(1) Frequent revisions or additions are made to the required specifications during the development of the control program.

(2) The progress of control program development cannot be tracked accurately.

AT開発評価用プログラムの供給遅れや動作不良を防止するには，このような要因に確実に対処できる方策が必要とされる．個々の分析結果から方策を検討，決定し，実行していくことが通常の解決方法であるが，この方法では，未経験の現象や要因の場合，効果が期待できない．したがって，より確実な効果を狙い，ソフトウェア業界で実質的な標準となっている品質プロセスのフレームワークを利用，個別，独自の方策も織り込んだうえで，制御用ソフトウェアの品質開発プロセスを再体系化することとした．

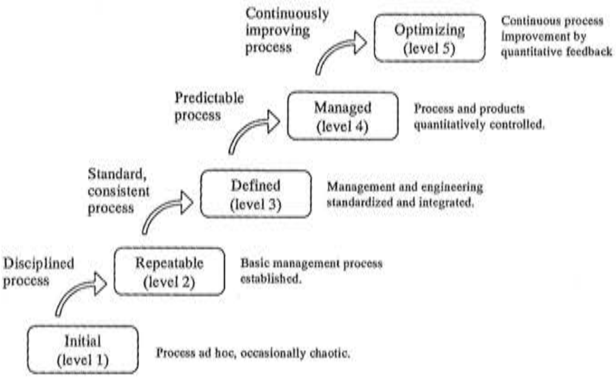
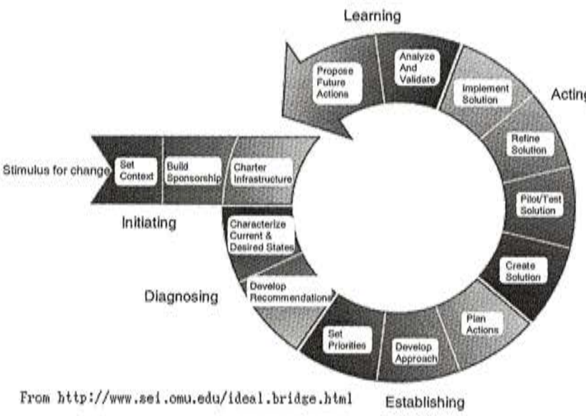


Fig. 3 Five levels of software process maturity

3. ソフトウェアプロセス改善モデル

品質プロセスのフレームワークとして，現在，世界的に広く普及しているCMM[7-10]を利用した．CMMはCapability Maturity Modelと呼ばれ，Fig. 3に示されるように，ソフトウェア開発組織のプロセス成熟度を1から5の5段階に分類，定義している[7]．適用される組織の能力程度に応じ，フレームワークを選択的に応用することで，段階的にレベルアップを図ることができる特徴をもつ．一般にプロセス成熟度が2以上であれば，その組織は安定した品質のソフトウェアを開発，供給する能力があるとされている．また，CMMには，IDEAL (Initiating, Diagnosing, Establishing, Acting, Learning) モデルが用意されており(Fig. 4)，活動の開始，診断，改善計画，改善実施，学習の各フェーズに沿った活動の推進が推奨されている．このうち，診断(Diagnosing)フェーズで，CMMフレームワークによるプロセス診断が実施される．これにより，品質プロセス上の弱みが認識され，改善の端緒が開かれることとなる．

Measures for dealing with such issues reliably are needed to prevent delays in the provision of a control program for use in evaluating an AT under development and the occurrence of operational defects. The ordinary approach to solving problems is to examine possible solutions based on the results of individual analyses and then decide and implement a suitable corrective measure. However, this method cannot be expected to be effective in the case of phenomena or factors that have never been experienced before. Therefore, it was decided to systematize once again the quality development process for AT control software with the aim of obtaining more definite benefits. Toward that end, we adopted the framework of the quality development process that has become the de facto standard in the software industry and also incorporated our own individual and original measures into it.



From http://www.sei.cmu.edu/ideal.bridge.html

Fig. 4 The IDEAL^{SM} model

3. Model for Software Process Improvement

The framework used for the quality development process was the Capability Maturity Model (CMM), which is currently widely employed throughout the world.[7-10] As illustrated in Fig. 3, the CMM classifies and defines the process maturity of a software development organization in terms of five levels from Level 1 to Level 5.[7] One of the model's features is that the framework can be selectively applied to an organization according to its level of ability and thereby promote progressive improvements to a higher level. In general, process maturity at Level 2 or higher indicates that an organization has sufficient capability to develop and supply software of stable quality. The CMM includes the IDEAL model, which stands for the phases of Initiating, Diagnosing, Establishing, Acting and Learning (Fig. 4). It is recommended that organizations promote activities in line with each of these phases. Among them, the CMM framework is applied to diagnose the software process in the Diagnosing phase. As a result, weaknesses in the quality development process can be identified, which is the starting point for making improvements.

4. 品質開発プロセスの再体系化

Fig. 2の既存の基本プロセスに対して，まず，このプロセスとソフトウェア開発業務の実態が，CMMフレームワークに適合しているかどうかを初期診断した．

4.1. 初期診断

初期診断の結果，以下が必要であることがわかった．

(1) 開発管理活動に対するレビューの強化．
(2) 作業の見積りとスケジューリング，およびリスク管理の強化．
(3) 習慣的業務運営方法の制度化．
(4) 実績データの収集，分析，活用の強化．
(5) 一部の担当者のスキルへの依存体質からの脱却．

なお，CMMフレームワークに基づくプロセス成熟度は，1.45であった．

4.2. 改善計画

抽出された課題に基づき，改善計画を立案した．改善計画の要点は以下のとおりである．

(1) 品質保証のための基盤整備

管理作業を安定化し，高いレベルの結果が反映されるように，ソフトウェアプロジェクト管理の手順を文書化し，体系化する．また，ソフトウェアプロジェクトの組織標準への遵守状況や欠陥の発生状況などを，定量的に管理する．

(2) 納期遵守のための基盤整備

作業の正確な見積りとスケジューリングのために，科学的な見積り手法を開発するとともに，スケジューリング，進捗管理ツールを導入する．また，不測の事態に対する対応力を強化するために，リスク管理手法を標準化し実践する．

(3) プロセス管理体制の整備

制御ソフトウェアに対する要求がある程度確定していることを前提に，要求変更が発生することも踏まえたプロセスの再構築を進める．また，SQA(Software Quality Assurance)チームを設置し，ソフトウェアプロジェクトとは独立した立場で，プロジェクトの管理活動および活動成果物に関して，客観的に判断することが可能な体制とする．また，SQAチームには，全体の計画・推進機能を集中してもたせ，一元的な改善活動を展開する．

(4) 人材教育の充実

制度化された標準類やツール類を，担当者ひとりひとりが十分に理解し，使いこなせるよう，研修プログラムの整備を進める．

4. Reorganization of Quality Development Process

An initial diagnosis was conducted to see if the existing development process in Fig. 2 and the actual practices of our software development activities complied with the CMM framework or not.

4.1. Initial diagnosis

The results of the initial diagnosis revealed that the following actions were needed.

(1) Reinforcement of the review of development management activities
(2) Reinforcement of work estimates and schedules and risk management
(3) Institutionalization of habitual methods of process management
(4) Reinforcement of collection, analysis and utilization of previous project data
(5) Extrication from the practice of depending on the skills of certain engineers

The CMM framework analysis indicated that the level of our software process maturity was 1.45.

4.2. Improvement plan

An improvement plan was drawn up based on the issues identified by the analysis. The major points of the plan are outlined below.

(1) Implementation of quality assurance foundations

The procedures for managing software development projects should be explained in a manual and systematized so as to stabilize management activities and enable high-level results to be reflected in other projects. In addition, the extent to which software project standards are faithfully observed, the amount of defects that occur and other aspects should be managed quantitatively.

(2) Implementation of foundations for meeting deadlines

Scientific methods should be developed for estimating and scheduling work accurately and tools should be implemented for managing work schedules and progress. Additionally, risk management methods should be standardized and followed so as to strengthen abilities for dealing with unforeseen situations.

(3) Implementation of a process management system

The development process should be reconstructed on the premise that the requirements for AT control software are definite to a certain extent and also by taking into account the fact that requirements change. In addition, a Software Quality Assurance (SQA) team should be formed to provide a mechanism for judging project management activities and activity results objectively from a position separate from the software development project. The overall planning and project promotion functions should be concentrated in the SQA team to facilitate the implementation of centralized improvement activities.

(4) Improvement of human resources development

Training programs should be created and implemented to enable all of the development engineers to fully understand and effectively use the institutionalized standards and tools.

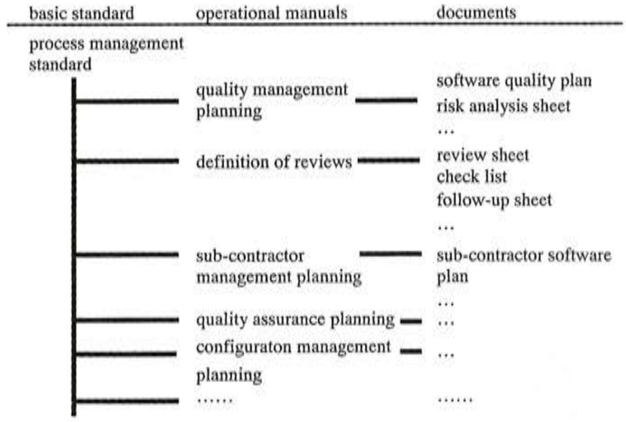


Fig. 5 Systematic scheme for standardizing documents

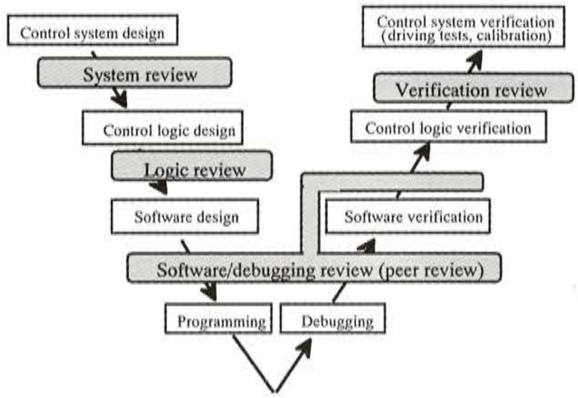


Fig. 6 Software reviews in the V-process

4.3. 体系化プロセス

Fig. 2の基本プロセスを軸として，前述のように改善したうえで，以下に述べる体系化プロセスを確立した．

(1) 標準文書体系

従来，必要の都度制定していた業務処理手順を記載した標準文書を，CMMフレームワークに沿った形に再体系化した． ATCU開発部門としての開発プロセス基本ルールを新たに文書化し，標準文書体系の根幹とするとともに，各工程のインプット，アウトプットとしての成果物，帳票を再定義した．(Fig. 5)

(2) ソフトウェアレビュー体系

ソフトウェア関連の設計状況を審査する仕組みとしてのソフトウェアレビューを，Fig. 6に示すように再定義し，各レビューの目的，実施時期，レビューの視点，審査資料の要件等を明確化した．

(3) プロセス監査体系

プロジェクトの活動状況を把握するためには，体系化された標準・基準の遵守状況をチェックする体制が不可欠である．今回，ATCU開発部門内に，Fig. 7に示す監査体制を確立した．SQAチームおよびシニアマネジメント層による確実かつタイムリーなフィードバックが可能となった．

5. 効果

5.1. プロセス成熟度診断

改善活動の進捗状況については，随時，CMMフレームワークへの適合度の評価診断という形で確認した．Fig. 8はその変化の様子である．2003年3月には，ひとつの節目としてCMM公式アセスメントを実施し，結果，プロセス成熟度2を達成した．

4.3. Systematized process

The systematized process described here was established on the basis of the previous development process and by instituting the improvements outlined above.

(1) System of standard documents

The operational manuals that had been previously created whenever necessary were reorganized into standardized documents in a format consistent with the CMM framework. The basic rules of the software development process in the ATCU development group were newly spelled out in documents to provide the foundations for systematizing the standardized documents. In addition, the inputs and outputs of each process were redefined in terms of results and forms.

(2) Software review system

Software reviews were redefined as indicated in Fig. 6 to provide a system for examining the status of design activities related to control software development. The objectives, time of implementation, review perspectives, requirements for the materials to be examined and other details were defined for each review.

(3) Process audit system

In order to ascertain the status of project activities, it is necessary to have a system for checking how faithfully the instituted standards and norms are observed. The audit system newly established in the ATCU development group for that purpose is shown in Fig. 7. This system facilitates reliable and timely feedback by the SQA team and senior management.

5. Benefits

5.1. Diagnosis of process maturity level

The progress of the improvement activities was confirmed from time to time by diagnosing and evaluating their consistency with the CMM framework. Examples of changes tracked over a 16-month period are shown in Fig. 8. An official CMM assessment was made in March 2003, and the results showed that process maturity of Level 2 had been attained at that juncture.

5.2. 品質への直接的効果

ATCU制御ソフトウェア開発プロジェクトにおける，直接的品質指標の概要を説明する．品質指標のひとつとして，ソフトウェアバグ発生率の変化を，Fig. 8にプロセス成熟度とともに示す．これからわかるように，プロセス成熟度の向上に呼応して，品質指標が著しく向上している．

ATCU開発部門では，プロセスの全体の体系化を進めながら，ソフトウェアレビューで使用するチェックシートの改良，ソフトウェア検証装置の自動化など個別技術の改善活動も並行して推進している．個別技術のみが改良されても，これを活用するプロセスなくしては，個別技術は十分に機能することができない．逆に，技術的裏付けのない体系化プロセスは，何ら機能することはない．また，整備されたプロセスの中で，個別技術を使いこなしていく人材なしでは，プロセス，技術ともに，意味をなさない．今回，これらプロセス，技術，人材の各領域における改善活動の相乗効果により，顕著な成果が得られたと考える．

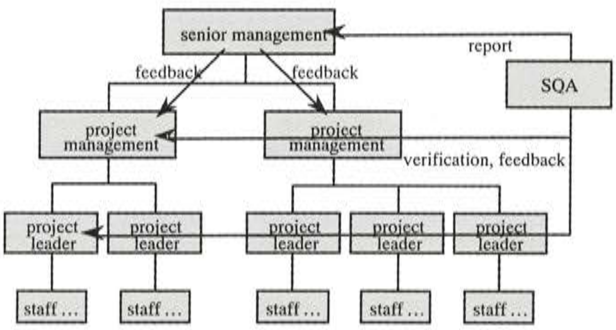


Fig. 7 Systematic scheme of management and verification

6. おわりに

ATCU開発部門における，制御用ソフトウェアの品質開発プロセスを体系化し，実効的な品質向上効果を得た．プロセスを体系化するにあたり，未だ顕在化していない課題にも対応できるよう，ソフトウェア業界での実質的世界標準であるCMMをベースとして，自身の直面している課題を解決するため，技術的方策を織り込み，品質プロセスとしての完全性を極力高めた．また，プロセスと技術を十分に活用できる人材の育成も同時に進めることにより，組織としての総合力を向上させた．

5.2. Direct effect on quality

A brief explanation is given here of the direct effect on certain quality indexes used in development projects for ATCU control software. As one index of quality, the change in the software defect rate is shown in Fig. 8 along with the process maturity level. As is clear from the figure, the quality index improved markedly in tandem with process maturity improvement.

Along with promoting systematization of the entire development process, the ATCU development group has also been proceeding with parallel activities to improve individual technologies. This includes improvement of the check lists used in the software reviews and automation of software verification devices, among other improvements. Simply enhancing individual technologies alone without improving the overall process in which they are used would not enable them to function fully. Conversely, merely systematizing the development process without providing the necessary technological support would not enable the system to function as it should. Moreover, unless there are people who can use the individual technologies effectively in the systematized development process, neither the process nor the individual technologies have any meaning. Under this improvement program, significant results were obtained through the multiplied effects of the improvement activities carried out in the three areas of the development process, individual technologies and human resources.

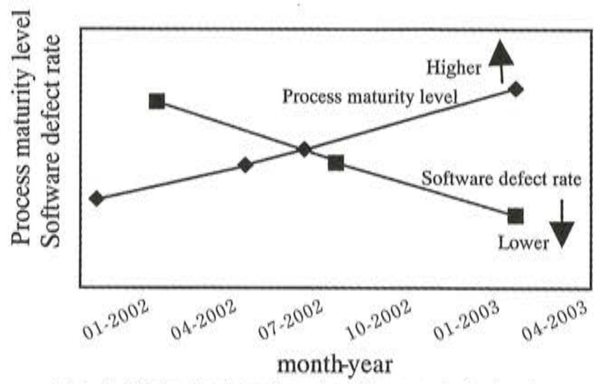


Fig. 8 Maturity level and software defect rate

6. Conclusion

The quality development process for AT control software was newly systematized in the ATCU development group and various benefits effective in improving quality were obtained. In order to allow flexibility for dealing with issues that have not surfaced yet, process systemization was based on the Capability Maturity Model (CMM), which is the global software industry's de facto standard. Technical measures for resolving specific issues facing the ATCU development group were incorporated in the CMM framework to maximize the level of perfection of the quality development process. In addition, simultaneous efforts were also made to develop human resources capable of fully using both the systematized process and individual technologies, thus enhancing the organization's overall capabilities.

ATの電子制御システムおよび制御用ソフトウェアの高度化が加速されるに従い，ソフトウェアのさらなる品質向上のためには，プロセス，技術，人材を一体システムとして捉え，改善していくことが重要である．(Fig. 9) 当社ATCU開発部門は，今後も継続的に，制御ソフトウェアの高度化を進めるとともに，プロセス体系のスパイラルアップを積極的に推進していく．

最後に，本報告に関する活動にあたり，多大なご協力をいただいた方々に感謝の意を表する．

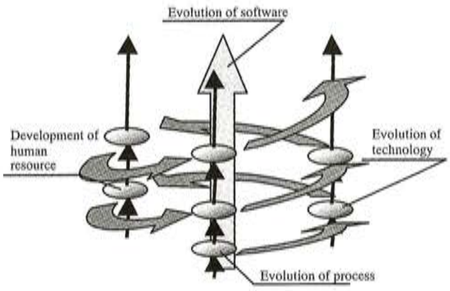


Fig. 9 Upward spiral of evolution

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As the sophistication of AT electronic control systems and control software continues to accelerate, it is essential to view the development process, individual technologies and human resources as an integrated system and improve all three elements together, in order to enhance the quality of control software further (Fig. 9). In JATCO's ATCU development group, we plan to enhance AT control software continuously and will also undertake vigorous efforts to promote an upward spiral of all elements of the process.

Finally, the author would like to express his appreciation to various individuals for their tremendous cooperation in connection with the activities described in this article.

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■ Author ■

Masayuki SATO

論文・解説

非線形有限要素法ソルバABAQUSによるトロイダルCVTの熱解析

A Heat Analysis of a Toroidal CVT using ABAQUS Nonlinear Finite Element Analysis Software

住　泰夫*
Yasuo SUMI

抄　録　トロイダルCVTのトラクション部の温度はトラクション性能に大きな影響を及ぼすことが知られている．トラクション部の温度はスピンによる発熱以外に他の発熱部からの熱伝導や冷却性能に大きく影響される．このため，汎用非線形有限要素法ソルバABAQUSを用いて，トロイダルCVTの熱伝導，熱伝達解析を行った．
本稿では，解析手法の解説と計算結果例，ならびに，得られた知見を記述する．

Summary　It is well known that the temperature of the traction unit in a toroidal CVT greatly affects traction performance. The traction unit temperature is substantially influenced by heat conduction from other heat sources and by the cooling performance, in addition to heat generated by spinning at the contact points. A general-purpose nonlinear finite element analysis (FEA) tool, ABAQUS, was used to analyze heat conduction and heat transfer in a toroidal CVT. This article explains the analysis method, examples of the simulation results and insights gained through this study.

1. はじめに

トロイダルCVTはその優れた動力性能と燃費性能によって，市場では高い評価が得られている[1]．しかし，更なる性能向上にはトラクション性能の向上が必要で，そのための手段の1つとして，トラクション係数の向上が考えられる．このため，高性能トラクションオイルの開発と共に，トラクション部の温度の低下が求められている．トラクション部の温度に及ぼすパラメータの影響を明確にするため，ABAQUSによりモデルを作成し，解析することを可能とした．

本手法ではトロイダルCVTの心臓部であるトラクション部をほぼ忠実にモデル化し，種々の要因の分析を可能とすると共に，実際の測温実験値とのコリレーションを可能とした．

2. 開発のねらい

今回解析したトロイダルCVTの主断面をFig. 1に示す．エンジンのトルクはトルクコンバータ，正逆転機構を経由して，バリエータ部に伝達される．バリエータ部は，1組の入出力ディスクに2個のパワーローラを有するキャビティ2組を出力ディスク部で背中合わせに配置している．すなわち，大容量化に対応するため，ダブルキャビティとしている．

1. Introduction

Toroidal CVTs have been by evaluated highly by the marketplace for their outstanding power performance and fuel economy benefits.[1] However, the traction performance, i.e., ability to transmit power, of toroidal CVTs must be enhanced to improve transmission performance further. One conceivable way of doing that is to improve the traction coefficient. Toward that end, it is necessary to develop a high-performance traction fluid and to reduce the temperature of the traction unit. ABAQUS software was used to create a simulation model for conducting a heat analysis in order to make clear the effects of various parameters on the traction unit temperature.

This software is capable of modeling the traction unit, representing the heart of a toroidal CVT, with high fidelity, making it possible to analyze many different factors and to correlate the results of the analysis with actual temperatures measured experimentally.

2. Development Objective

A main cross-sectional view of the toroidal CVT that was analyzed in this study is shown in Fig. 1. Engine torque is transferred from the torque converter to the forward/reverse actuation mechanism and then to the variator. The variator has two cavities positioned back-to-back in the output disk section, with each cavity consisting of a pair of input/output disks and two power rollers. In other words, it adopts a dual-cavity design to provide greater torque capacity.

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バリエータ部は，動力伝達原理としてトラクションドライブを用いている．トラクションドライブでの動力伝達面はトラクションオイルの剪断力で伝達しており，動力伝達面の法線力とトラクション係数が大きいほど大きな力を伝達できる．

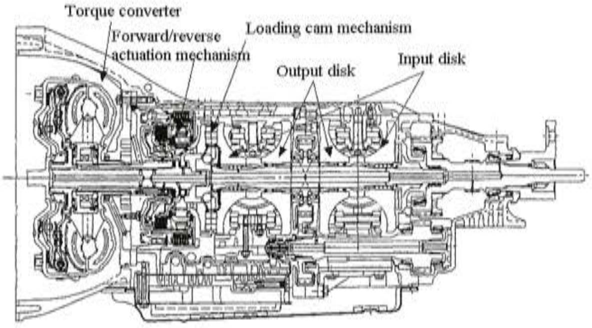

Fig. 1 Main cross-sectional view of toroidal CVT

法線力は，入力トルクにほぼ比例した押し付け力を発生するローディングカム機構で得ている．

変速は，パワーローラをトラニオン軸まわりに傾転させて，入力ディスク接触点半径と出力ディスク接触点半径を変化させることで行われる．

パワーローラと入出力ディスクの間の動力伝達面には，スピンが生じることがある．このスピンの量は傾転角すなわち変速比によって変化するが，このスピンにより生じた発熱がパワーローラと入出力ディスク間の動力伝達面の温度を上昇させる．

さらに，トラクションドライブ部の法線力により，パワーローラにはスラスト方向の力が発生し，その力を，パワーローラベアリングを介してトラニオンにより支持されている．このパワーローラベアリングはその構造上スピンを発生する．このスピンにより生じた発熱がパワーローラと入出力ディスク間の動力伝達面に伝達される．

また，トラクション性能に大きな影響を及ぼすトラクション係数は温度の上昇に伴い低下する傾向を持っている[2]．従って，良好なトラクション性能を維持するためには，動力伝達面の温度を低減することが重要となる．

トラクションドライブの動力伝達面やパワーローラベアリング部の発熱は弾性流体潤滑(EHL)理論によりかなり正確に求めることができるが，その発熱による熱がパワーローラや入出力ディスクなどにどのように伝達するかを定量的に解析することが重要な課題である．従って，開発の狙いとして，発熱部位からの熱がどのように伝導・伝達するかをシミュレーションにより解明することである．

The variator utilizes a traction drive as its power transmission principle. Power transmission in a traction drive is accomplished by the shear force of the traction fluid between the power-transmitting surfaces. The larger the normal force and traction coefficient of the power-transmitting surfaces, the more power the unit is capable of transmitting.

Normal force is obtained by a loading cam mechanism that generates loading force nearly proportional to the input torque. A ratio change is accomplished by tilting the power rollers around the trunnion axis to change the radii of contact on the input and output disks.

Spinning can occur at the power-transmitting surfaces between the power rollers and the input/output disks. While the amount of spinning varies depending on the tilt angle of the power rollers, i.e., the CVT ratio, the heat generated by spinning raises the temperature at the power-transmitting surfaces between the power rollers and the input/output disks.

Moreover, the normal force of the traction drive unit causes the power rollers to produce force in the thrust direction. That force is supported by the trunnions by means of the power roller bearings. Because of the nature of their construction, the power roller bearings also experience spinning that generates heat which is transferred to the power-transmitting surfaces between the power rollers and the input/output disks.

Additionally, another factor that greatly affects traction performance is that the traction coefficient tends to decrease with increasing temperature.[2] Accordingly, reducing the temperature at the power-transmitting surfaces is an important factor in maintaining good traction performance.

Heat generation at the power-transmitting surfaces of the traction drive and at the power roller bearings can be found with rather good accuracy on the basis of elastohydrodynamic lubrication (EHL) theory. However, a critical issue is to analyze quantitatively how the generated heat is transferred to the power rollers, input/output disks and other components. Therefore, the development objective here was to elucidate by computer simulation the ways in which such heat is conducted and transferred from each heat source.

3. Simulation Model

3.1. Selection of FEA tool

Heat conduction and heat transfer are phenomena with strong nonlinearity, and the problem to be analyzed involves contact between the power rollers and the input/output disks. For these reasons, the ABAQUS general-purpose FEA tool, which has a proven reputation in this field, was selected for use in the analysis.

3. 解析モデル

3.1. ソルバの選択

熱伝導，熱伝達は非線形性の強い現象であるのと，パワーローラと入出力ディスク間の接触問題であるので，この分野で定評のある汎用非線形有限要素法ソルバであるABAQUSを用いた．

ABAQUSには陽解法(explicit)であるABAQUS／Explicitと陰解法(implicit)であるABAQUS／Standardが用意されている．陽解法は時刻tあるいはそれ以前における既知の項を用いて解を求めるのに対し，陰解法では時刻$t+\Delta t$とそれ以前の項を用いて，時間増分毎に非線形代数方程式を解くことにより解を求める．この解析では，両者で実際に解析した結果，CPU時間の観点から，主として後者のABAQUS／Standardを用いた．

3.2. 解析モデル

トラクションドライブ部は主として，入出力ディスク，パワーローラ，パワーローラを傾転自在に支持するトラニオン，パワーローラとトラニオンの間に作用する押し力を受け持つボールとアウターレース，および，入出力ディスクをパワーローラに押し付けるローディングカム等で構成されているが，解析に用いたモデルは，モデルの簡単化のため，Fig. 2で示すように，入力ディスク，出力ディスク，2組のパワーローラ，ボール，アウターレースで構成した．また，全てのモデルは弾性体として取り扱っている．

また，パワーローラ，ボール，アウターレースは変速比の状態に応じて傾転角を任意に変えられように，傾転角をABAQUSの機能であるPython記述を用いたパラメータ手法を活用し，傾転状態のモデル化が容易にできるようになっている．

3.3. 解析のステップ

解析のプロセスは実際のトロイダルCVTが作動する時と同様に設定している．まず始めに，入出力ディスクのディスク間距離を合わせ，その後，アウターレース，ボール，パワーローラを所定の押し圧で押し付ける(Fig. 3参照)．その後，各部を所定の回転速度に達するまで回転させ，接触面間のすべりにより生じた熱が，各部に伝導，伝達し，各部の温度が定常状態に達したところで計算を終了する．

ABAQUSでは発熱による熱変形が接触状態の変化を生じさせ，その結果，発熱状態が変化する完全熱－応力連成解析が備わっているが，この解析の目的から非連成として解析した．すなわち，接触は発熱に影響を及ぼすが，逆に発熱は接触状態に影響を及ぼさないこととしている．その他，解析のリソース(CPU時間，メモリ，ハードディスク容量など)を低減するための工夫を幾つか施した．

ABAQUS includes ABAQUS/Explicit for obtaining explicit solutions and ABAQUS/Standard for obtaining implicit solutions. With the explicit method, a solution is found using time t or a term known in advance before t. In contrast, with the implicit method, a solution is found by solving a nonlinear algebraic expression for every increment of time using time $t+\Delta t$ and a term known prior to t. Based on the results of actual analyses performed with both methods, the latter ABAQUS/Standard method was chiefly used in this analysis from the standpoint of acceptable CPU time.

3.2. Simulation model

The traction drive unit mainly consists of the input/output disks, power rollers, trunnions that enable the power rollers to tilt freely, the balls and outer race responsible for the loading force that acts between the power rollers and trunnions, and the loading cam that produces the loading force which pushes the input/output disks against the power rollers. For the sake of simplification, the model used in the analysis consisted of an input disk, an output disk, two power rollers, balls and an outer race, as shown in Fig. 2. In addition, the entire model was treated as an elastic body.

The power rollers, balls and outer race were modeled such that their tilt angles could be varied according to the CVT ratio. That was done by using a parametric technique to simplify the modeling of the tilted condition. This technique made use of the Python programming language embedded in ABAQUS to describe the tilt angles.

3.3. Simulation steps

The simulation procedure was designed in the same way as the actual operation of a toroidal CVT. First, the distance between the input/output disks was adjusted, and the outer race, balls and power rollers were then loaded with the specified loading force. (The assembled model is shown in Fig. 3.) The various components were then rotated until they reached their specified operating speed. The heat produced by slipping at the contact surfaces was conducted and transferred to the different parts, and at the point when their temperature reached a steady state, the calculation was completed.

ABAQUS incorporates the capabilities for a fully coupled thermal-stress analysis, in which thermal deformation induced by heat generation changes the state of contact, with the result that the condition of heat generation changes. However, an uncoupled analysis was performed here because of the objective of this study. In other words, it was assumed that contact influenced heat generation, but that heat generation did not affect the state of contact. Several other resourceful measures were also taken to reduce the resources (CPU time, memory capacity, hard disk capacity, etc.) needed to conduct the analysis.

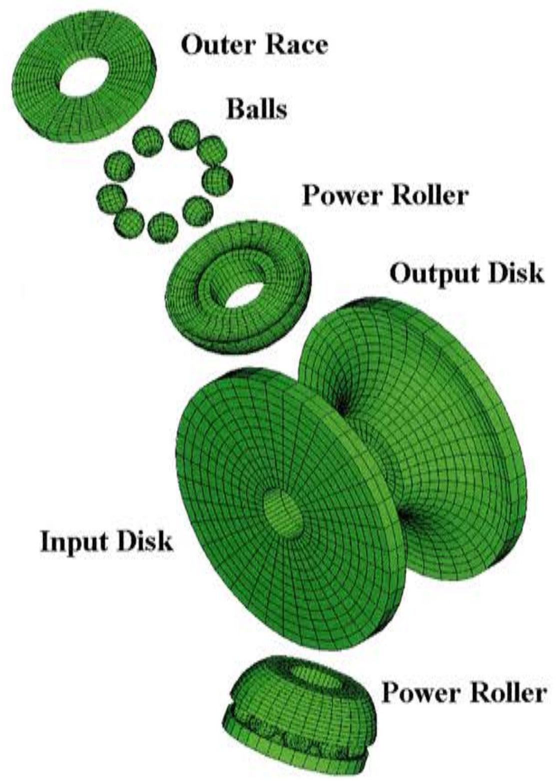

Fig. 2 Parts of simulation model

3.4. 発熱の取り扱い

入力ディスク～パワーローラ，出力ディスク～パワーローラ，パワーローラ～ボール，ボール～アウターレース各間の発熱は接触問題として，ヘルツの接触部の相対すべり速度，接触面圧を求め，これと摩擦係数とにより熱流束を求めることが可能であるが，要素分割を細分しなければならないことによるCPU時間の増大を避けることと，前述のEHL理論を用いた動力伝達面の損失がかなり正確に推定できることから，この損失から求めた熱流束を入力値としてモデル化した．

3.5. 実験値とのコリレーション

前述の理論的な手法で求めた各接触部の発熱による熱流束の計算結果はトラクション部のみで構成された実験装置の動力伝達効率の測定から検証することができる．また，モデルの熱伝導，比熱などの物理特性は材料特性から得ることができる．一方，各部の要素表面からの熱伝達は冷却放熱条件などにより左右される．これらの表面からの放熱を左右する熱伝達係数は金属～オイル間の環境条件によるが，解析には文献などに記載の値を参考に，若干補正して用いた．Fig. 4に示す測温点(図中赤点が測温点，数値が測温値)とのコリレーションを行った．

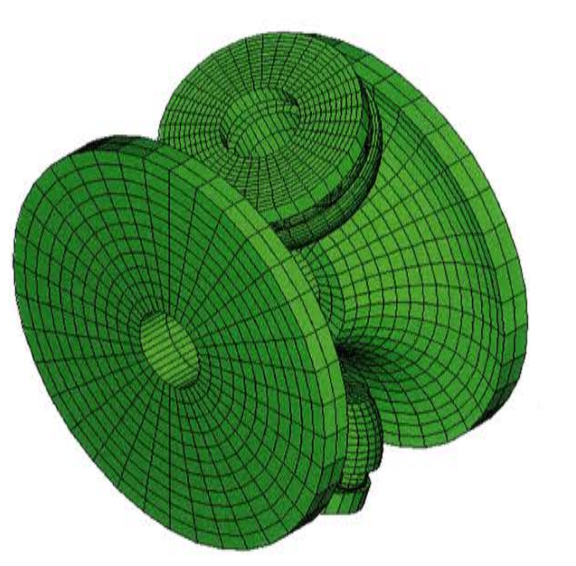

Fig. 3 Assembled simulation model

3.4. Treatment of heat generation

It is possible to determine the heat flux by using the friction coefficient and the relative sliding velocity and contact pressure found at the points of hertzian contact, by treating heat generation between the input disk and power roller, output disk and power roller, power roller and balls and balls and outer race as a contact problem. However, the heat flux value input into the simulation model was found in this study from the losses estimated at the power-transmitting surfaces using the EHL theory mentioned earlier. That was done to avoid increasing the CPU time on account of the need to divide the elements finely and also because such losses can be estimated rather accurately using the EHL theory.

3.5. Correlation with experimental data

The heat flux due to heat generation at each contact point was calculated using the theoretical method explained above. The calculated results can be validated by comparing them with the power transmission efficiency measured using an experimental rig consisting only of the traction unit. In addition, the heat conduction, specific heat and other physical properties of the model can be obtained from the material properties. On the other hand, heat transfer from the surface of the elements of each component is affected by cooling and heat radiation conditions, among other factors. The heat transfer coefficient that governs heat transfer from the component surfaces depends on the environmental conditions between the metal and the oil. The values given in the literature were taken as reference and slightly modified for use in the analysis. The correlation with the measurement points was then examined as shown in Fig. 4, where the red dots are the measurement points and the numbers are the measured temperatures.

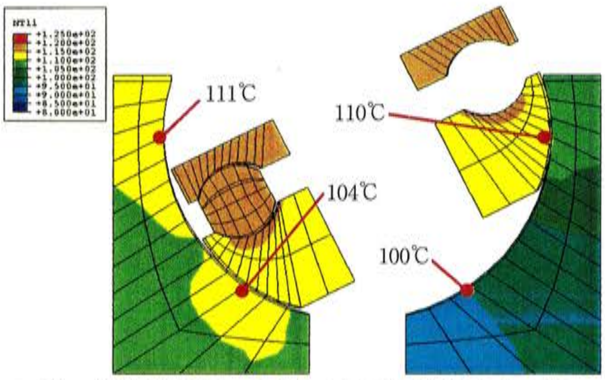

Fig. 4 Calibration points and simulation results

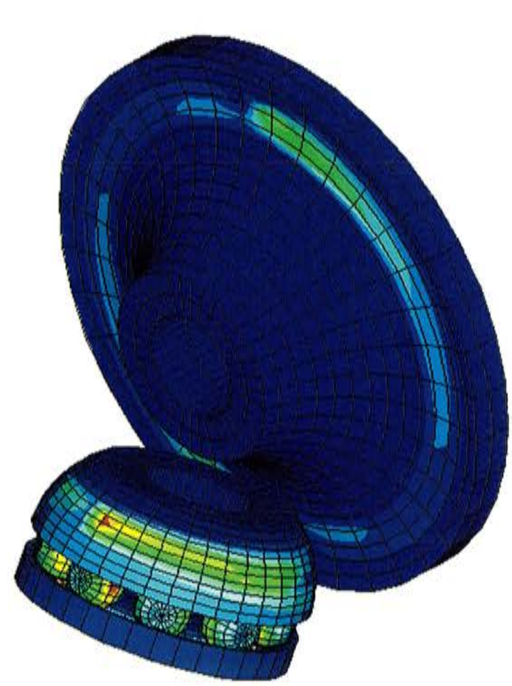

Fig. 5 Surface temperature after 1/4 rotation

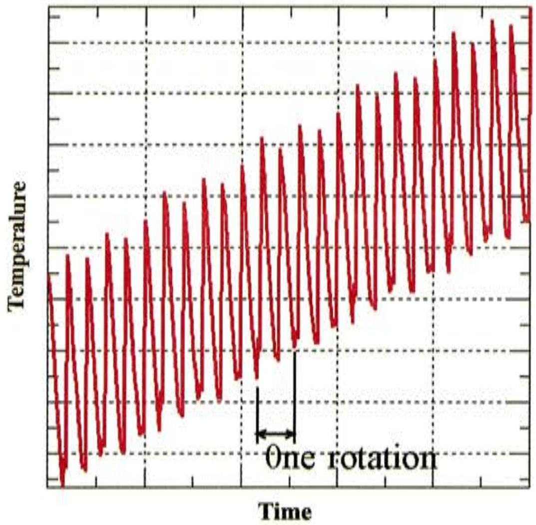

Fig. 6 Time history of contact point temperature

4.解析結果

Fig. 5は，Fig. 3で示したモデルの入力ディスクと上側に配置したパワーローラ，ボール，アウターレースを描かないことで接触部の温度を解りやすく表示した図であるが，変速比がLowの状態で，計算開始時から出力ディスクが90度回転した状態における出力ディスクおよび下側に配置したパワーローラ部の表面の温度分布の変化を示す．ディスクに着目した点の温度はFig. 6に示すように，1/2回転毎に上昇～下降を繰り返し，平均としては時間と共に上昇することとなる．

Fig. 7は，変速比Lowにおいて，キャビティ当たり62.5 k Wが入力された時の出力ディスク表面温度分布の計算結果を計算開始から20秒までを示すが，青色が低温部，赤色が高温部を表している．Fig. 8にほぼ定常状態に達した時刻における断面の温度分布を示す．最も温度が高い部位はボール～アウターレース，ボール～パワーローラの接触部位であり，この部分の損失により発熱した熱がパワーローラ～ディスクの転動面を経由してディスク側に伝わっていく様子がわかる．

4. Simulation Results

Figure 5 shows the change in the surface temperature distribution of the output disk and the power roller located below it. These results are for 90° (1/4) rotation of the output disk following the start of the calculation under a condition of a low CVT ratio. For easier understanding of the temperature at the contact point, the input disk and the power roller, balls and outer race located above it in the assembled model in Fig. 3 have not been depicted in Fig. 5. A time history of the temperature at the contact point on the disk is shown in Fig. 6. It is seen that the temperature rises and falls with each 1/2 rotation of the disk and that the average temperature increases with the elapsed time.

Figure 7 shows the calculated surface temperature distribution of the output disk at different elapsed times in the first 20 seconds following the start of the calculation. These results are for a low CVT ratio and an input force of 62.5 kW per cavity. The blue and red portions indicate low and high temperature regions, respectively. The temperature distribution in a cross section at the moment the CVT components reached nearly a steady-state condition is shown in Fig. 8. The places showing the highest temperatures are the contact points between the balls and outer race and between the balls and the power roller. It is observed that the heat generated by the slipping losses at these points is transferred to the disk via the tilted contact surface between the power roller and the disk.

一方，バリエータ部の冷却は，Fig. 9に示すように，
(1) パワーローラ軸芯を経由してキャビティ表面を冷却するフロー．このフローの一部は高温となるボール部を冷却したのち図示の如く，比較的高い温度となり，ディスク外側に流出する
(2) 両ディスクの軸を経由してディスクの両側面を冷却するフロー
(3) 上側のリンクポストに設けられたノズルより両ディスクのキャビティ面を冷却するフロー

に分けられるが，特に(1)のフローの内，ボールを冷却したフローは，入力ディスクの冷却を妨げる作用をすると考えられる．

The cooling oil flow paths of the variator are illustrated in Fig. 9. It is seen the oil flows are divided such that:
(1) An oil flow for cooling the cavity surface passes through the core of the power roller axis. As shown in the figure, part of this oil flow cools the high-temperature balls, thus giving it a relatively high temperature, and it then flows out to the disks.
(2) An oil flow for cooling the sides of the disks passes through the axis of each disk.
(3) An oil flow for cooling the cavity face of each disk comes from the nozzles provided at the link posts at the top.

The portion of oil flow (1) that cools the balls is especially thought to have the effect of hindering the cooling of the input disk.

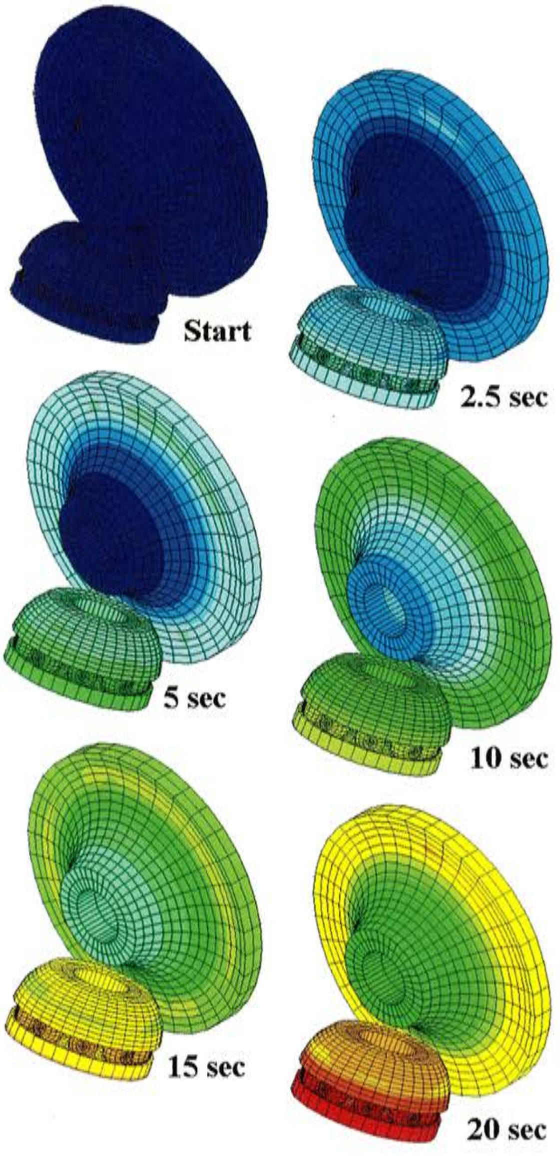

Fig. 7 Surface temperature

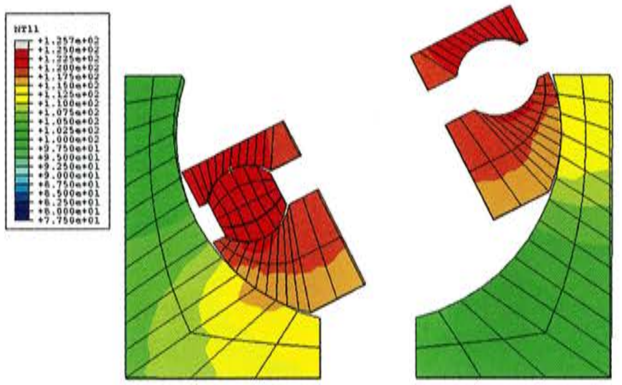

Fig. 8 Cross-section temperature

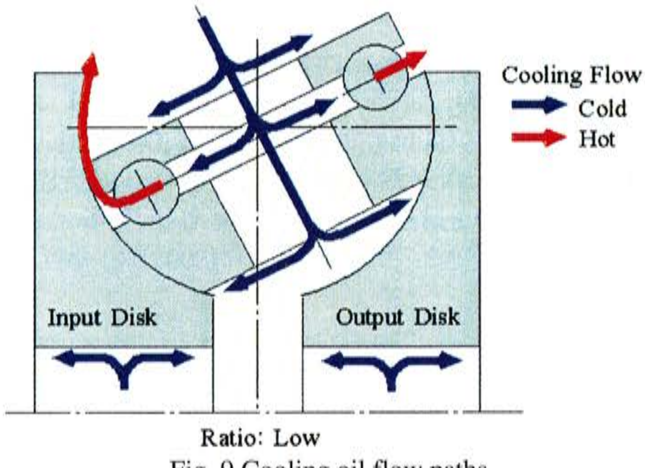

Fig. 9 Cooling oil flow paths

Fig. 10は各接触部の温度の変化履歴を示す．約40秒で平衡状態に達していることがわかる．

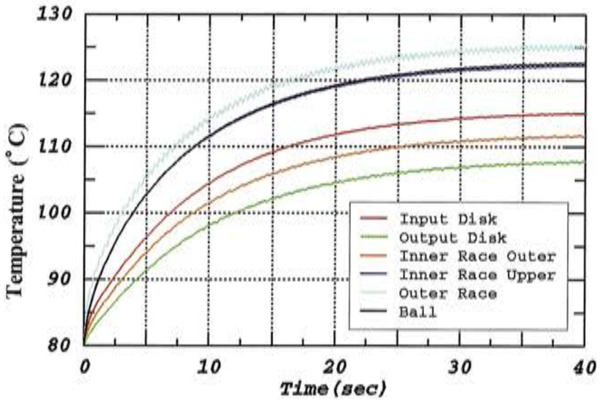

Fig. 10 Time histories of typical contact point temperatures

5. まとめ

トラクションドライブの性能を予測するために，接触部の発熱による各部の温度状態を求めるシミュレーション技術を開発した．

これにより，次の知見を得た．

(1) ABAQUSを用いて，実際の作動状態に近いモデルを作成することにより，トロイダルCVTの温度を予測することが可能となった．

(2) トロイダルCVTにおける最大の発熱部位はパワーローラのスラストを支持(ボールとアウターレース，ボールとパワーローラ)する軸受けの発熱であり，この箇所の発熱がパワーローラとディスクの接触部に伝導していることから，この発熱を低減することが重要である．

(3) 更に，パワーローラのスラスト軸受けを冷却したオイルがディスク表面に流出し，例えば，変速比Lowにおける入力ディスクのキャビティ外側が加熱され，その結果，接触部近傍の温度を上昇させることがわかった．

(4) 実時間20秒の計算に必要なCPU時間は約120時間で，実用の限界と言える．しかし，最新のCPUを用いれば1/2～1/3に短縮が可能である確認を得ており，将来は更に短縮できると考える．

6.あとがき

トロイダルCVTにおいても，トルク容量の拡大やコンパクト化の要請が高まる中で，トラクション性能の基本性能を決定づけるトラクション面の温度解析技術は重要な位置づけにある．今後も，冷却手法との関連を含めた精度向上と，設計ツールとしての実用的なシミュレーション技術の確立に努力いたしたい．

Figure 10 shows time histories of the temperatures at typical contact points. It is seen that the temperatures reach an equilibrium state in approximately 40 seconds.

5. Conclusion

A simulation tool has been developed for determining the temperature condition resulting from heat generation at various contact points in order to predict the performance of a traction drive system. The results of this study have made the following points clear.

(1) The temperature distribution of a toroidal CVT can be predicted by using ABAQUS to create a simulation model that resembles the unit's actual operational state.

(2) The places that generate the most heat in a toroidal CVT are the bearings (contact between the balls and outer race and between the balls and power rollers) that support the thrust of the power rollers. It is important to reduce the heat generated at these places because this heat is conducted to the contact points of the power rollers and input/output disks.

(3) Additionally, the oil that cools the thrust bearings of the power rollers flows out to the surface of the disks. For example, the results showed that under a low CVT ratio the outer surface of the cavity of the input disk was heated, which had the effect of raising the temperature in the vicinity of the contact point with the power roller.

(4) It took approximately 120 hours of CPU time to calculate the temperature distribution for 20 seconds of operation, which is about the limit for practical use. However, it was confirmed that the use of a recent CPU might be able to shorten that time by 1/2 to 2/3, so it is thought that the time can be further reduced in the future.

6. Closing Remarks

Against a backdrop of rising demands to increase the toque capacity and reduce the size of toroidal CVTs, the technology for analyzing the traction surface temperature, which basically determines the traction performance, occupies an important position. In future work, efforts will be made to improve the simulation accuracy, including the connection with the cooling method, and to establish a practical simulation technology as a design tool.

Finally, in connection with the development of this simulation technology, the author would like to thank various individuals in the Powertrain and Environment Research Laboratory, Nissan Research Center, Nissan Motor Co., Ltd. for their advice and cooperation in providing the measured temperature data and at ABAQUS Inc. for their cooperation with the development of the programming method.

最後に，本シミュレーションの開発にあたり，アドバイスと測温実験データの提供にご協力を頂いた日産自動車株式会社総合研究所動力環境研究所の関係各位，ならびに，プログラミング手法開発でご協力頂いたABAQUS Inc. の関係各位に深く感謝の意を表します．

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■ Author ■

Yasuo SUMI

論文・解説

アイスショットによるばり取り洗浄工法

Deburring by Ice Blasting

片岡　誠司*
Seiji KATAOKA

抄　録　近年，アルミ部品のばり取りには，様々な媒体(メディア)を投射する工法が試みられている．
本稿では，BHO，BKO(FF大型4AT,5AT)Case-Transmission #2ラインで初めて採用したアイスショットによるばり取り工法と，その工法導入時の不具合対策などを紹介する．

Summary　Various media and methods have been tried in recent years for deburring aluminum parts. This article describes the ice-blasting equipment used for deburring transmission cases on the No. 2 line of the BHO 4-speed AT and the BKO 5-speed AT for use on large, front-wheel-drive cars. The ice-blasting method and measures taken to deal with initial problems are discussed.

1. はじめに

夾雑物対策は，振動・騒音対策とともに，AT，CVTメーカにとっては最重要な課題である．
BHO，BKO Case-Transmissionには、変速制御用の油圧回路が組み込まれており，ばりが脱落してバルブに噛み込めば，変速異常という重要不具合となる．

2. ばり取り工法の推移

古くはブラシによるばり取りが主であったが，その後，高圧ウォータージェットによるばり取り工法が普及した．この工法では，高圧ポンプの故障やノズル詰まりが発生しやすく，頻繁なメンテナンスと高額の費用を要するという欠点を抱えている．
またカスケードまたはクルミショットというメディアを使った工法も採用されるようになってきたが，これらの工法は使用後のメディア残りが大きな課題となり，それを除去するための洗浄に神経を使わなければならない．
本稿で紹介するアイスショットは，氷をメディアとしているため，メディア残りが無いことが最大のメリットであり，今後注目される工法である．

1. Introduction

Along with measures against noise and vibration, the elimination of foreign matter from ATs and CVTs is one of biggest issues that a manufacturer of these units must address. The hydraulic circuit for shift control is incorporated in the transmission case of the BHO and BKO ATs. If burrs fall into the fluid and get caught in the valves, it could give rise to a serious problem of abnormal shift performance.

2. Tend in Deburring Methods

Formerly, burrs were mainly removed with a brush, and deburring by means of a high-pressure water jet later came into widespread use. This method has the disadvantages of being rather expensive and of requiring frequent maintenance because the high-pressure pump tends to break down and the nozzles are apt to plug up.
Deburring methods that make use of a cascade system or walnut shells as a deburring medium have also been adopted, but a major issue with these approaches is that the deburring medium can remain on workpieces after the operation. Meticulous cleaning is then required to remove any remaining deburring medium.
One of the biggest advantages of the ice-blasting method described here is that there is no residual deburring medium because ice is used. This method is expected to attract attention in the coming years.

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Unit Production Engineering Department

3. アイスショットの原理

3.1. 製氷の方法

氷というと，家庭の冷凍庫にあるブロック型のものを想像するかもしれないが，これをばり取り洗浄機に使用するには，氷の貯蔵庫，細かく砕くための強力なクラッシャが必要となり，大掛かりな設備となってしまう．

今回採用した製氷方法は，水中に半分浸っている冷えたドラムが1回転する間に，そのドラムの表面に付着した水が氷となり，スクレーパで掻き出されるというもので，氷は必要分だけ連続的に作られる(Fig. 1)．これを簡単なクラッシャで砕いて最終的な形状とする．貯蔵庫はなく，できた氷は数秒後にはワークに投射される．

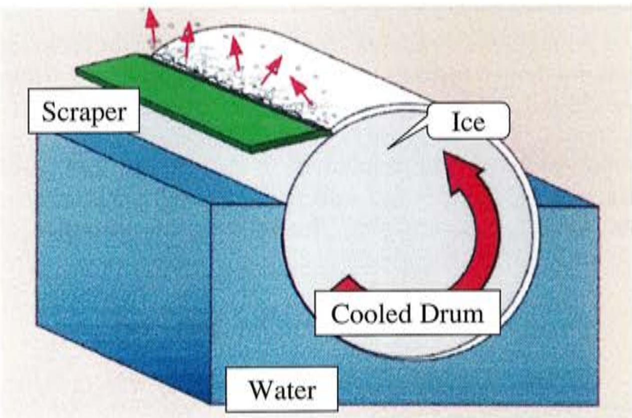

Fig. 1 Continuous Ice-making Process

3.2. アイスショットの動力源

氷をワークに投射させる方法としては，氷をエアーや高圧水に混ぜる方法が考えられる．

氷をなるべく固体の状態を維持したまま投射させるには，エアーに混ぜる方法が良いが，氷を高速に加速するには，多量のエアーが必要であり，大きな動力を消費する．

高圧水に混ぜて投射する方法を採れば，圧力の発生と制御が容易であり，これを選択した．

3.3. 氷の搬送方法

高圧水に氷を混ぜる方法としては，高圧水によって引き起こされる負圧を利用した．高圧水の配管に製氷機からの配管を合流させておき，高圧水をワークに向かって流すことで，氷の配管に負圧を発生させ，氷を吸い込む仕組みである(Fig. 2)．このとき，エアーも巻き込むので，エアーミキシング効果も期待できる．

3. Principle of Ice Blasting

3.1. Ice-making method

The word ice may suggest the ice cubes made in a household refrigerator. Attempting to use such ice in the deburring machine would require large-scale facilities, including an ice storage chamber and a powerful crusher for pulverizing blocks of ice into fine pieces.

The ice-making method we have adopted uses a cooled drum that is half-submerged in water. Water that sticks to the drum surface during one revolution becomes ice, which is then scraped off with a scraper. This continuous process can produce the exact quantity of ice required (Fig. 1). The ice is then pulverized by a simple crusher into its final form. Because the manufactured ice is projected against the workpiece a few seconds later, there is no need for a storage chamber.

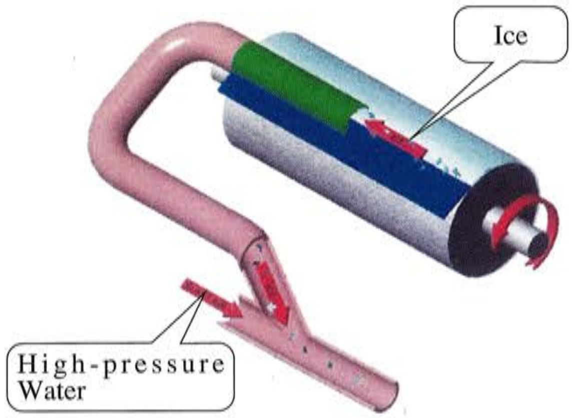

Fig. 2 Suction of Ice

3.2. Working principle of ice blasting

Two methods can be considered for projecting ice against the workpiece, either by mixing it in air or in pressurized water. Mixing ice in air is preferable in terms of maintaining it in a solid form as much as possible at the time of projection. However, this method consumes considerable power because it requires a large air supply to accelerate ice to high projection speeds.

We selected the method of projecting ice mixed in pressurized water because it is easy to generate and control the pressure.

3.3. Ice transport method

The vacuum generated by the pressurized water is utilized for mixing the ice in the water. The pipe from the ice-making machine is joined to the pressurized water pipe. Spraying the pressurized water toward the workpiece creates a vacuum in the ice pipe, which works to suck ice into the water (Fig. 2). Because air is simultaneously entrained, an air mixing effect can also be expected to occur.

4. アイスショット採用の経緯

4.1. BHO, BKO Case-Transmission ＃1ラインの課題

BHO，BKOは1994年から生産を開始したユニットで，そのCase-Transmission加工の＃1ラインは現在も稼動中であり，三菱自動車のFF乗用車に搭載している主力機種である．

ばり取り工法としては，高圧ウォータージェットを採用しているが，前述2項の通り，高圧ポンプの頻繁なメンテナンスに加え，品質面でも切粉残りという課題を抱えており，オペレータが全数チェックして取り除いているのが実状である．

切粉残りの原因としては，複雑なワーク形状が大きく関わっている．バルブボディの取付面には，バルブボディと同様の複雑な溝が掘られているため，切粉がこの溝に高圧ウォータージェットによって押し込まれて取れない状態になってしまうことがある(Fig. 3)．

洗浄機としては，この切粉残りの課題を解決することが最重要であった．

4. Background behind Adoption of Ice Blasting

4.1. Issues on the No. 1 line of BHO/BKO transmission cases

The BHO and BKO ATs went into production in 1994, and the No. 1 line for machining these transmission cases is still operating. These ATs are the mainstay transmissions used on the front-wheel-drive cars of Mitsubishi Motors Corp.

The deburring method makes use of a high-pressure water jet. As mentioned earlier in section 2, the high-pressure pump requires frequent maintenance. In addition, because of the quality problem related to residual chips, operators currently check every transmission case to make sure burrs have been removed.

The cause of residual chips is closely related to the intricate transmission case geometry. The surface where the valve body is attached is provided with the same intricate grooves as the valve body. There are times when chips are pushed into the grooves by the high-pressure water jet and cannot be removed later (Fig. 3). The biggest issue for the deburring washer was to be able to remove such chips.

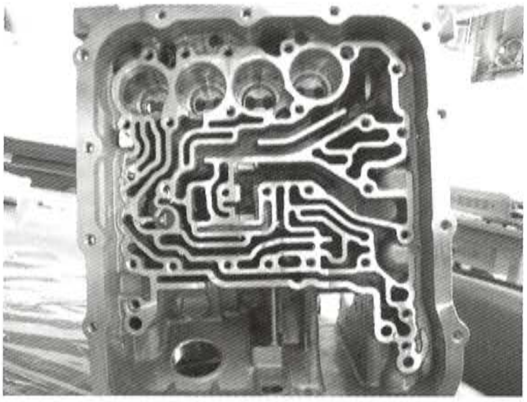

Fig. 3 Control Valve Face of Transmission Case

4.2. ＃2ライン洗浄機の仕様検討

増強ラインとして＃2ラインを新設するにあたり、転用改造可能なブラシ洗浄機と高圧洗浄機は保有していたが，4.1.項の課題を解決するため、新工法を取り入れることとした．

洗浄機の工法を決定するにあたり，カスケード及びクルミショットは，ワーク形状の複雑さからメディア残りの恐れがあるため除外し，高圧ウォータージェットとアイスショットとを比較評価した．

4.2. Study of No. 2 line deburring washer specifications

At the time the No. 2 line was built, we had brush washers and high-pressure washers that could be modified and adapted for use on this new additional line. However, we decided to adopt a new deburring method in order to resolve the problem mentioned in the preceding section.

In deciding what deburring method to adopt for the washer, we eliminated a cascade system and a walnut shell method from consideration because of the possibility that the deburring medium would remain in the intricate geometry of the transmission cases. Accordingly, a comparative evaluation was made of the high-pressure water jet and ice blasting.

4.3. ばり取り工法の比較試験

4.3.1. 比較試験条件

高圧ウォータージェットでは，＃1ラインでの実績を基に30MPa，ノズル径ϕ0.7mmを中心とし，またアイスショットでは，15MPaの水に60L/hの氷を混ぜ，ノズル径ϕ30で投射するという条件を中心として比較試験した．テストピースとしては，現物のCase-Transmissionを使用し，ばり除去性能，切粉除去性能を比較した．

4.3.2. 比較試験の結果(Table 1)

まず，ばり除去性能については，高圧ウォータージェット，アイスショットの両者とも，差異がなかった．

バルブボディの取付面の溝に切粉を無理やり挟ませたワークでの切粉除去性能試験においては，ウォータージェットでは圧力を上げても，＃1ラインと同様に切粉を押し込んでしまうだけの結果となったが，アイスショットでは，このような切粉でも除去することができた．固体である氷を当てることで，細い切粉なら分断し，また頑丈な切粉なら弾き飛ばすことになったのであろう．

この切粉除去性能が決め手となり，アイスショットを導入することとした．

4.3. Comparative tests of deburring methods

4.3.1. Test conditions

Based on the performance record of the high-pressure water jet on the No. 1 line, the pressure was set at 30 MPa and the nozzle diameter at 0.7 mm. For ice blasting, ice was mixed at a rate of 60 L/h in water under pressure of 15 MPa and projected through a 30-mm-diameter nozzle. These were the main conditions under which the comparative tests were carried out. Actual transmission cases were used as the test pieces, and a comparison was made of the deburring performance and chip removal performance.

4.3.2. Comparative test results

There was no difference in deburring performance between the high-pressure water jet and ice blasting (Table 1). Tests of chip removal performance were conducted on workpieces that had chips forcibly lodged in the grooves on the surface where the valve body is attached. The water jet only pushed the chips into the grooves, as it did on the No. 1 line, even though the pressure was raised. In contrast, ice blasting was able to remove such chips. It is presumed that fine chips were broken into pieces by the impact of the solid ice and that strong chips were blown away.

This difference in chip removal performance was the deciding factor in the decision to adopt ice blasting.

Table 1 Test Results for Deburring and Chip Removal

Method	Ice blasting	High-pressure water jet
Nozzle dia. (mm) and number	30 mm × 1	0.7 mm × 4
Pressure (MPa)	15 (pressure of water into which ice is mixed)	30
Deburring	○	○
Removal of lodged chips	○: Effective in breaking and blowing chips away	×: Pushes lodged chips further into grooves
Positive factors	○: Excellent for deburring and removal of residual chips	○: Excellent for deburring ○: Large flexibility for nozzle placement ○: Ample experience and proven record of use
Negative factors	×: Difficult to use multiple nozzles ×: Cannot angle nozzles ×: Difficult to detect hardness of ice ×: Requires control of water temperature and density to obtain hard ice ×: Little experience and record of use	×: Cannot remove chips lodged in grooves → Must be combined with another method ×: High-pressure pump is expensive to maintain

○: Good ×: Poor

5. 比較試験時の課題への対応

比較試験によって，アイスショットで，ばりや切粉が除去されることは判ったが，同時にTable 1のマイナス要因を克服することが必要となった．

5. Solutions to Problems Observed in Comparative Tests

The comparative tests showed that ice blasting was able to remove burrs and chips, but at the same time it was also necessary to overcome the negative factors listed in Table 1.

5.1. ばり発生箇所への確実な氷の投射

氷はワーク全面に投射される必要がある．このため，ノズルの移動とワークの回転は，マシニングセンタのように，NC制御とした．

5.2. 氷の検知方法

氷は水と混合して投射される．そのとき，ばり取りに有効な硬い氷になっていることを検知する必要がある．#1ラインのウォータージェットの管理に用いている荷重値管理では，水だけと氷入りとの差が無いことが判った．荷重の振動を計測したところ，水だけと氷を混入したときとでは数値に差があったことから，これを採用することにした．サイクル毎に設定しきい値をクリアしていれば，ワークに投射する仕組みとした(Fig. 3，Fig. 4)．

5.3. 水温管理

製氷機では前述のように連続的に氷が生成されるが，供給される水の温度が上昇すると，シャーベット状となり，ばり取り性能は損なわれてしまう．この対策として，水を冷却するためのチラ-装置を追加し，製氷機に設定温度以下の水を送ることとした(Fig. 5)．

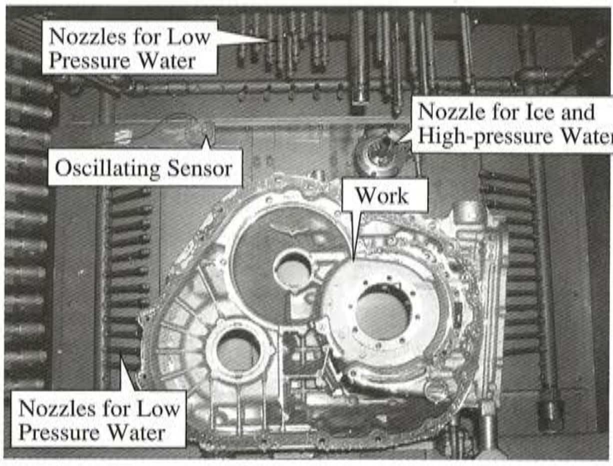

Fig. 4 Arrangement of Nozzles and Oscillating Sensor

5.4. クーラント液の影響の対策

通常，水は0℃で凝固し氷になるが，これに不純物が混入すると凝固点が下がり固まりにくくなる．アイスショット洗浄機の前工程にある加工設備からクーラント液が混入してしまうと，この現象で硬い氷ができなくなる．

当初，洗浄機から回収した水を製氷機に導く予定であったが，確実に硬い氷を得るため，チラー装置から製氷機に供給する水については，洗浄機への水とは別系統で供給することとした．

5.1. Reliable projection of ice toward burr locations

Ice must be projected against the entire surface of the workpiece. To accomplish that, the movement of the nozzle and the rotation of the workpiece are numerically controlled like the operation of a machining center.

5.2. Ice detection method

At the time the water-ice mixture is projected against the workpiece, it is necessary to detect whether the ice is frozen hard, as that state is effective for deburring. With the load value management method used for the water jet on the No. 1 line, it was found that there was no difference between water alone and the water-ice mixture. Measurement of the fluctuation of the load revealed that there was a numerical difference between water alone and the water-ice mixture, so it was decided to adopt that method. Provided that the threshold set for each cycle is attained, ice can be reliably projected against the workpiece (Figs. 3 and 4).

5.3. Water temperature control

As described earlier, ice is produced continuously by the ice-making machine, but if the water temperature rises, it results in sherbet-like ice that degrades the deburring performance. To prevent that, a chiller was added to chill the water below a certain given temperature before it is supplied to the ice-making machine. (Fig. 5).

Fig. 5 Control of Ice and Water

5.4. Measure against the influence of the coolant

Water ordinarily freezes and becomes ice at 0℃. The presence of impurities, however, can cause the freezing temperature to drop, making it more difficult for ice to harden. If coolant from the machine tools used in the process upstream of the ice-blasting washer is mixed in the water, it can prevent the formation of hard ice.

Initially, it was planned to recover water from the washer and feed it into the ice-making machine. However, to obtain hard ice reliably, it was decided to provide water from the chiller to the ice-making machine via a separate line from that for the water supplied to the washer.

5.5. 夾雑物スペックの確保

これらの諸策により，BHO,BKO Case-Transmission残留夾雑物重量スペックをクリアした．また，＃1ラインで課題となっていたバルブボディの取付面の切粉残りは格段に改善された．

6. 量産設備としての課題への対応

品質面では良好な結果となったが，量産ラインとしては超えるべきハードルがいくつもあった．

6.1. 製氷機の保証

製氷機は高額なため予備品は持たず，緊急時には設備メーカで所有している試験機の製氷機と交換することとした．

6.2. 結露対策

アイスショット洗浄機の配管類は5.3.項で説明したチラー装置を追加したことから全体的に低温になり，夏になると結露が激しくなった．この対策として，配管に断熱材を巻き，タンクや周辺機材にも断熱塗料を塗り，それでも結露の発生する部分ではオイルパンを追加した．

6.3. ランニングコストの低減

以前のブラシ洗浄機及び高圧洗浄機の電力量に比較し，今回導入したアイスショット洗浄機の電力量は2/3に低減し，また，洗剤を使わないのでフィルタ以外には消耗品が発生しないことから，ランニングコストが低減した．

7. アイスショットばり取り洗浄機の概要(Fig. 6,7)

(1)対象ワーク：BHO,BKO Case-Transmission 4機種
(2)サイクルタイム：2.0分/個
(3)要求スペック：清浄度，ばり高さ，残留物大きさ

5.5. Confirmation of foreign matter specification

The adoption of the above-mentioned measures made it possible for the BHO and BKO transmission cases to clear the weight specification for foreign matter. In addition, the issue seen on the No. 1 line concerning residual chips on the surface for attaching the valve body was markedly improved.

6. Resolution of Production Line Implementation Hurdles

Good results were obtained with respect to quality aspects, but several hurdles still had to be cleared in order to implement the ice-blasting system on our mass production lines.

6.1. Assured availability of ice-making machine

Because the ice-making machine is expensive, a replacement machine is not kept on hand. The manufacturer that installed the machine will replace it with a prototype machine in an emergency.

6.2. Prevention of condensation

The entire piping of the ice-blasting washer is at a low temperature because a chiller has been added, as explained in section 5.3. It was found that severe condensation occurred in the summer. To prevent that, the piping has been wrapped in insulation and insulating paint has also been applied to the water tank and peripheral equipment. An oil pan was added for locations where condensation still occurred in spite of those measures.

6.3. Reduction of operating costs

The newly introduced ice-blasting washer consumes one-third less electric power than the previous brush washers and high-pressure washers. In addition, it is also cheaper to operate because it does not require any detergent and has no consumables other than the filters.

7. Overview of Ice-blasting Deburring Washer

Views of the machine are shown in Figs. 6 and 7. The machine is used to clean four types of transmission cases for the BHO and BKO ATs. It has a cycle time of 2.0 min./case. It meets the specifications with regard to cleanliness, burr height and machining residue size.

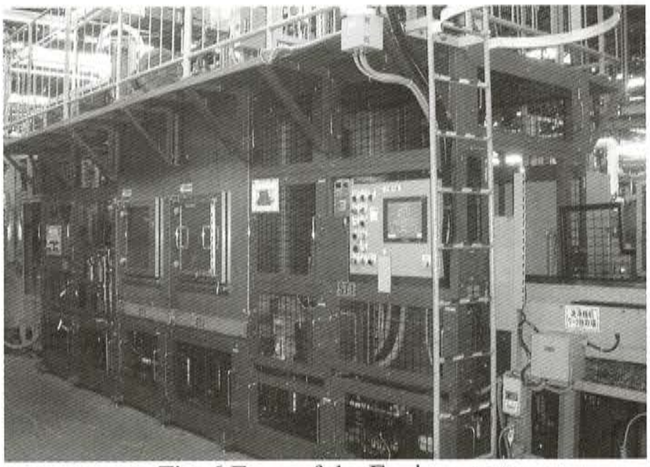

Fig. 6 Front of the Equipment

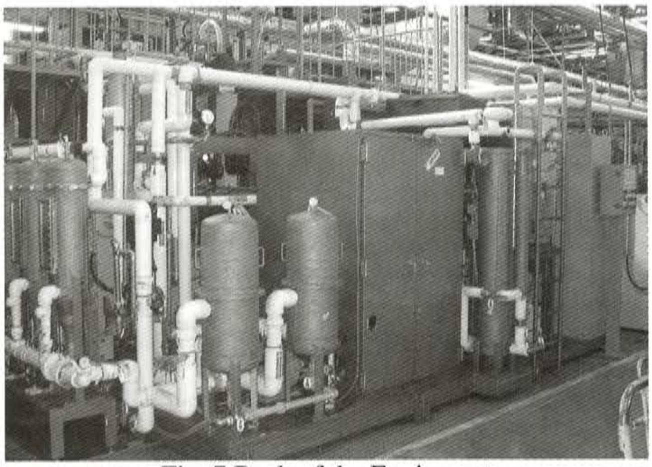

Fig. 7 Back of the Equipment

8. おわりに

アイスショットという工法を業界に先駆けて量産ラインに採用し，挟まった切粉まで除去できたこと，メディア残りが生じず，後処理が簡単であること，洗浄液やメディアなどの消耗品が少ないことなど，成果を挙げることができた．しかし製氷機に送る水の管理が当初の予想以上にデリケートであり，管理項目が多くなったことが課題として残った．

今後，同タイプのばり取り洗浄機を導入する際には，この課題を最初から配慮した仕様とすれば，より良い工法，設備となることを確信する．

当工法の実用化に際し，試験段階から設置後の改善活動にまでご協力いただいたBHO,BKO Case - Transmission #2ラインの関係者各位に厚く御礼申し上げる．

8. Conclusion

JATCO has been the first in the industry to implement ice-blasting equipment on mass production lines. This equipment offers the advantages of being able to remove chips lodged in grooves, of not leaving any deburring medium behind, of allowing a simple aftertreatment process and of using few consumables such as detergent and deburring medium. However, control of the water supplied to the ice-making machine has proved to be more delicate than initially imagined. There are still many such control issues to be resolved.

We believe that a better deburring method and equipment can be attained by factoring those issues into the specifications from the beginning when the same type of deburring washers are installed in the future.

The author would like to thank the members of the No. 2 production line of the BHO and BKO transmission cases for their invaluable cooperation from the testing stage to post-installation improvement activities in connection with the implementation of this deburring method.

■ Author ■

Seiji KATAOKA

論文・解説

統合新工場システムについて

Introducing the Jatco Enterprise Manufacturing System

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抄　録　新工場システム(JEMS: Jatco Enterprise Manufacturing System)は，業務統合プロジェクトの1つとして，現場主体となった自立運営ができる仕組みの確立を目指したシステムである．

本稿では，システムの概要及びシステムを運用する上で欠かすことのできないネットワークの再構築について紹介する．

Summary　The Jatco Enterprise Manufacturing System (JEMS) was developed in a project to integrate the company's operations. This new manufacturing system is designed to provide a framework that facilitates autonomous operation initiated by the plant floor.

This article gives an overview of JEMS and of the reconfigured network that is indispensable to the operation of this system.

1. システム構築の背景

旧トランステクノロジー社と旧ジャトコ社の合併にあたり，全社のシステム統合が必要だった．工場側のシステムでも，KONG(加工管理)・SKONG(粗材管理)・JMICS(実績収集)等の個々のシステムを統合し，組立ラインへの生産指示と直結したYOURS(旧トランステクノロジー　組立管理)・FCIM(旧ジャトコ　富士宮組立管理)等のインターフェースを有する新工場システムを構築することになった．

新システムの構築に際し，従来の見込み生産ではなく，生産指示から出荷までの生産リードタイムを短縮し，顧客ニーズへ迅速に対応できるよう，現場主体となった自律運営ができる仕組みを確立したいと考えた．

また，システム構成面でも，将来の事業拡大に対応できる拡張性を持たせた構成が必要だと考えた．

2. 機能概要

2.1. 新工場システム概要

生産管理システムをはじめ，新工場システムのDBサーバ，アプリケーションサーバ(以下APサーバとする)，監視サーバなどは全てセンタ側で設置・管理する．このセンタ側のシステムで決定された生産計画を，WAN(広域通信網)を経由して各工場に伝達し，現場での実績データを再びセンタ側へ返すという仕組みにした．

2.2. システム構成

新工場システムの機器構成をFig. 1に示す．

* 情報システム部
Management Information Systems Department

1. Motivation for Construction of JEMS

The merger of the former TransTechnology Ltd. and the former JATCO Corporation brought about a need to integrate our company-wide systems. The systems then in use at the plants included KONG (Kakou Online Network Growing system) for machining operations management, SKONG (Sozai-KONG) for raw materials management and JMICS (Jatco Manufacturing Information Control System) for collecting production results, among others. Those individual systems were integrated to construct the new JEMS manufacturing system that interfaces with YOURS (YOshiwara Unit Realtime management System) and FCIM (Fujinomiya Computer Integrated Manufacturing), which are directly linked to the production instructions given to the assembly lines. (YOURS is the assembly operations management system used previously by TransTechnology, and FCIM is the assembly operations management system used formerly by JATCO at the Fujinomiya Manufacturing Department.)

In constructing JEMS, we wanted to establish a system capable of being operated autonomously by the plant floor. Instead of the previous system of estimated production volumes, the new system would be able to respond quickly to customers' needs and shorten production lead times, from the issuing of production instructions to shipment of finished products.

Moreover, it was thought that the system should be configured with extensibility for coping with future expansion of the company's business.

2. Functional Requirements

2.1. Overview of JEMS

All elements of JEMS are installed and managed at a center, including the production management system, the database (DB) server, application (AP) server and supervisory server. The production plan determined by the system at the center is sent via a wide-area network (WAN) to each plant, and the actual production data at each plant are sent back to the center.

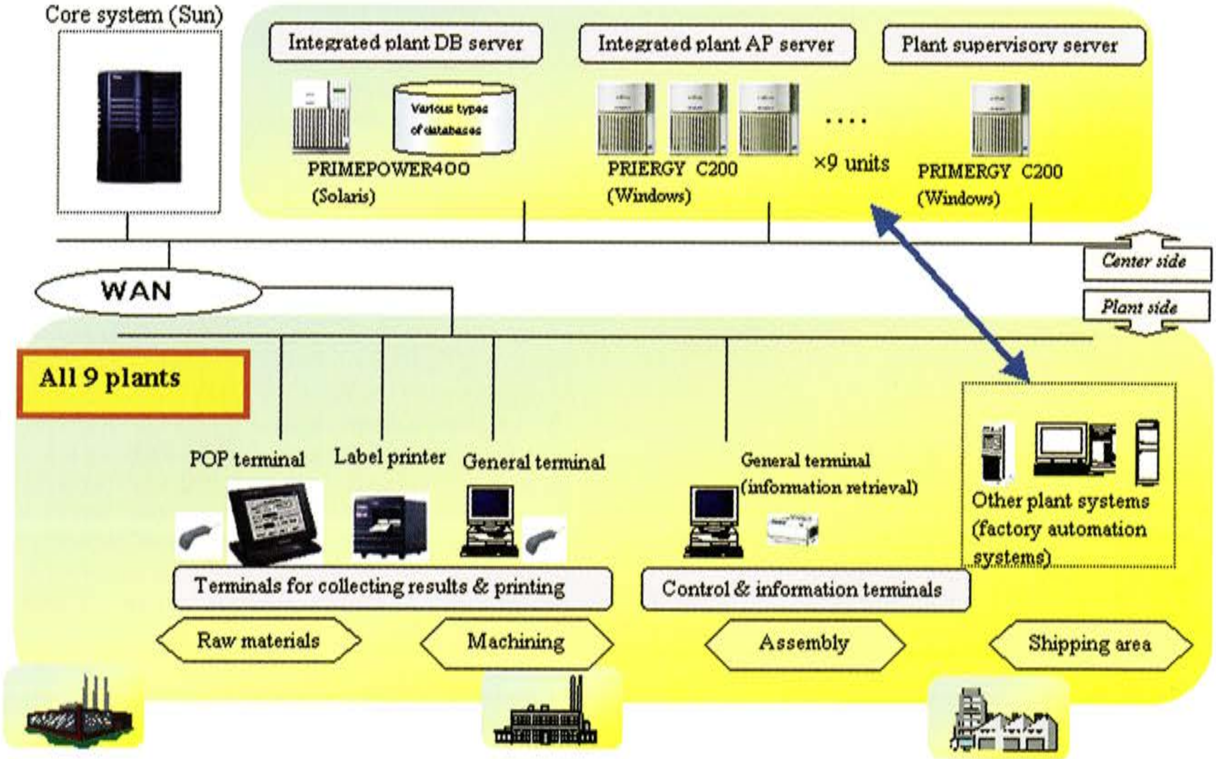


Fig. 1 System configuration

2.3. システムの特徴

(1)統合DBサーバ，APサーバによる集中制御

・9工場を集中管理

・3階層構成によるレスポンス確保(注1)

(2)監視サーバによる集中監視

・DBサーバ，APサーバの稼動状況を監視

*注1 3階層構成とは，P層(GUI制御(画面表示関係))，F層(業務ロジック)，D層(データ操作)の3層に明確に分けることで(1)ネットワークの高負荷を抑え，レスポンスの向上(2)メンテナンス性の向上(3)開発生産性の向上(4)プログラムの再利用性の向上が図れる．

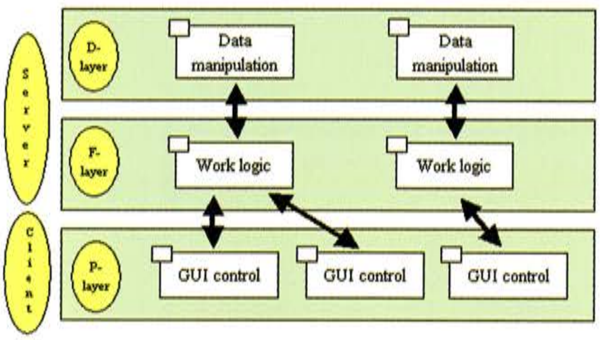


Fig. 2 Three-layer application configuration

2.2. System configuration

The configuration of the equipment making up JEMS is shown in Fig. 1.

2.3. Features of JEMS

(1) Centralized control by means of an integrated DB server and AP server
- Centralized control of nine plants
- Assured response based on a three-layer configuration*1

(2) Centralized supervision by means of a supervisory server
- Monitoring of the operating status of the DB server and AP server

*1The three-layer configuration refers to an explicit division into three levels⅄the P-layer (GUI control and associated screen displays), the F-layer (work logic) and the D-layer (data manipulation). This configuration is designed to (1) hold down high network traffic loads and enhance system response, (2) improve maintainability, (3) boost development work productivity and (4) increase the reusability of existing programs.

2.4. 新工場システムの機能

新工場システムの機能のアウトラインをFig. 3に示す．

2.4.1. 機能紹介

新工場システムのいくつかの機能について説明する．

(1)生産順序表

生産管理システム(JPICSII)にて立案された組立順序情報を受け，メイン組立ラインおよびそれに従属するサブ組立ライン，加工ラインに対して組立順序情報の提供および作業指示(組付指示，ピッキング/納入指示，生産指示)を行う．

(2)自律運営

生産管理システム(JPICSII)にて立案された所要量，独立需要，造溜計画の各情報を提示し，それらを基に製造現場が自工程の生産計画を自ら立案するとともに，それを実行するための材料の取り入れ(納入指示)も合わせて立案することを目的とする．基本的に加工工程で主に使用される機能．

(3)カンバン

製造現場にて現物を主体として生産および供給指示を行うための各種カンバンのコントロール(枚数管理，カンバン出力)を行うことを目的とする．

(4)ロット計画

自律運営で立案された納入指示や，生産管理システム(JPICSII)より受けた所要量，独立需要，造溜計画を所要情報として，在庫・供給の過未達を考慮しながら該当する工程のロット生産計画を作成する．

2.4. Functions of JEMS

The functions of JEMS are outlined in Fig. 3.

2.4.1. Functional overview

Several of the key functions of JEMS are explained here.

(1) Production scheduling chart

Based on the assembly scheduling information drawn up by the JPICS-II (Jatco Production Integrated Control System Version II) production management system, this function provides assembly sequence information to the main assembly line, the sub-assembly lines subordinate to it and the machining lines. It also issues work instructions (installation instructions, parts picking/delivery instructions and production instructions) to the lines.

(2) Autonomous operation

This function presents information on the necessary quantities, independent demand and production-for-storage plan. Based on that respective information, the plant floor draws up a production plan for its own process. A further objective of this function is also to enable each process to prepare a plan for securing the materials (delivery instructions) needed to execute its production plan. Fundamentally, this function is primarily used in the machining process.

(3) Kamban control

The purpose of this function is control the various types of kamban (signboards) used on the plant floor to give production and supply instructions concerning actual objects. It is used to manage the number of kamban and when they are issued.

(4) Lot plan

This function is used to create the lot production plan of the respective process, based on the delivery instructions generated by the autonomous operation function and the information on the necessary quantities, independent demand and production-for-storage plan received from the JPICS-II production management system. In doing so, it takes into account excess and insufficient inventory and supply volumes.

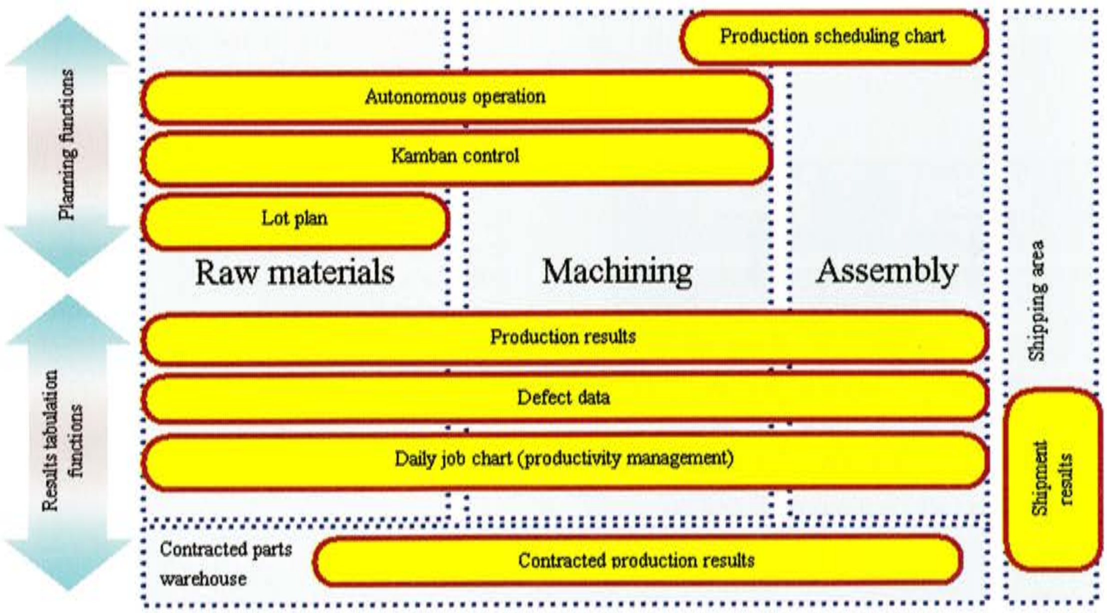


Fig. 3 Outline of plant system functions

(5)生産実績

組立・加工・粗材の各工程での生産実績を収集し，生産管理システム(JPICSII)に対して生産実績として計上する．また，組立工程においては組立バーコード，パレタイズリスト，製品ラベルの出力も行う．

(6)出荷実績

生産管理システム(JPICSII)より受けた出荷計画および自律運営・カンバンでの工場間をまたぐ納入指示を基に，出荷作業者への荷揃え指示を行うとともに，ユニット・パーツ・支給品・工場間移動の各出荷実績を収集し，生産管理システム(JPICSII)へ計上する．

(7)デイリーショップカルテ

日々の作業日報の入力を行い，得られた情報と，生産実績機能にて収集された生産実績，実活動計画値等の情報にて，日々の生産状況や月間での実活動率の計算等を行い提示する．

(5) Production data

This function collects the production results of each process—assembly, machining and raw materials—and uploads the production data to the JPICS-II production management system. It also issues assembly bar codes, palletizing lists and product labels to the assembly process.

(6) Shipment results

This function issues packing instructions to the shipper based on the shipping plan received from the JPICS-II production management system and the instructions generated by the autonomous operation and kamban control functions for deliveries between the plants. In addition, it also collects shipment results for transmissions, parts, supplied provisions and transfers between plants, and uploads the data to JPICS-II.

(7) Daily job chart

This function is used to input daily work information. Based on the input information and the production results collected by the production data function and the actual activity plan data, this function calculates and shows the daily production conditions and the actual monthly operating rates.

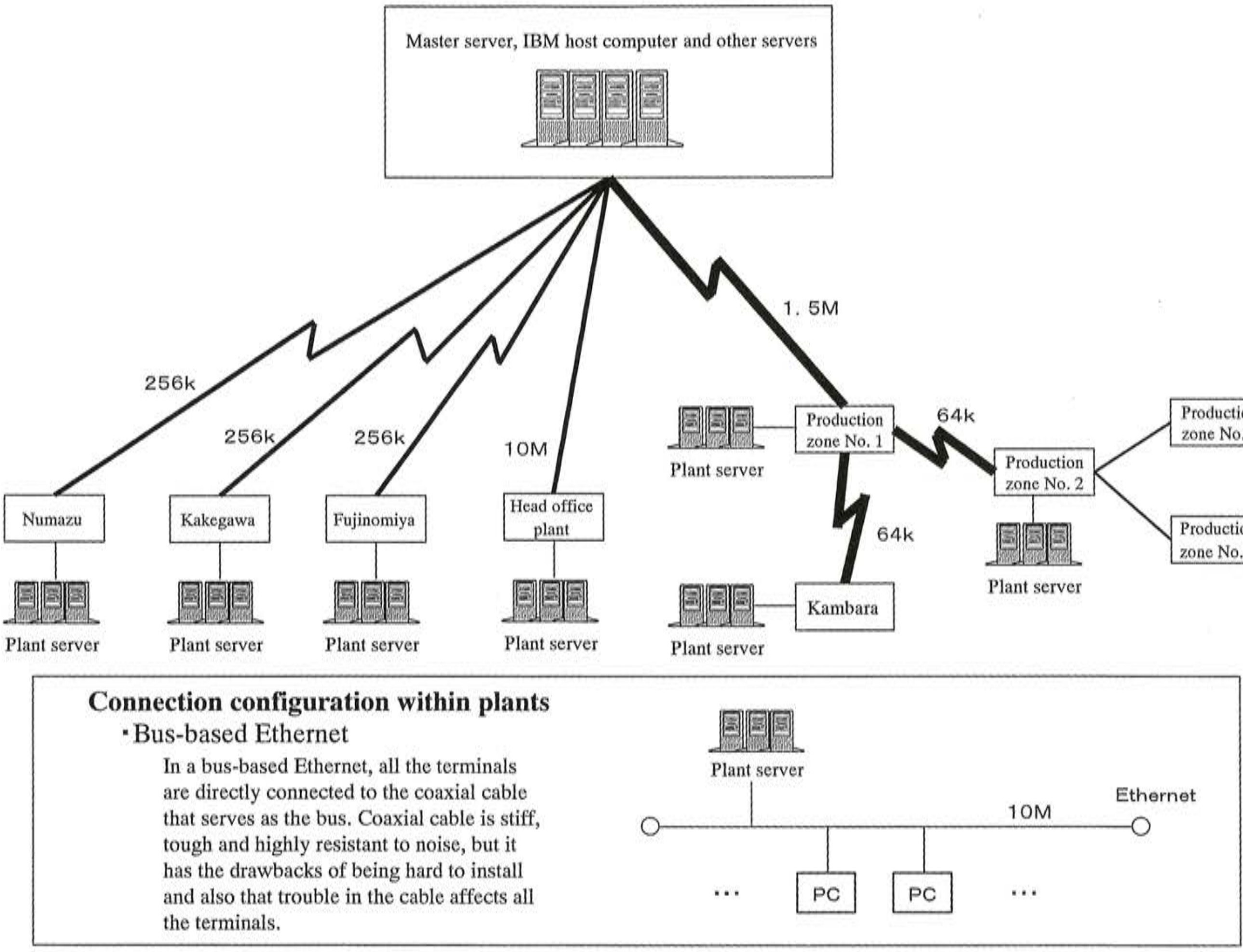


Fig. 4 Configuration of previous network

3. ネットワーク再構築について

旧システムでのネットワーク構成は各工場に工場サーバを設置し，本社にある基幹サーバやホストコンピュータとデータの授受を行っていた．(Fig. 4参照)

工場間の接続についてはNTTの専用回線を使用，工場内についてはバス型のイーサネットで接続，速度は10Mであり，約10年前に敷設した．そのため老朽化が激しく，トラブルが発生し工場全体に影響を与えることがあった．

3. Reconfiguration of the Network

The network of the former system was configured such that the master server and host computer at the head office exchanged data with a plant server installed at each plant (Fig. 4). NTT's leased lines were used to connect the plants, and a bus-based Ethernet was used for the connections within the plants.

That network had been built approximately ten years ago and provided a transmission speed of 10 MB. The network had become severely antiquated and the occurrence of troubles sometimes impacted all the plants.

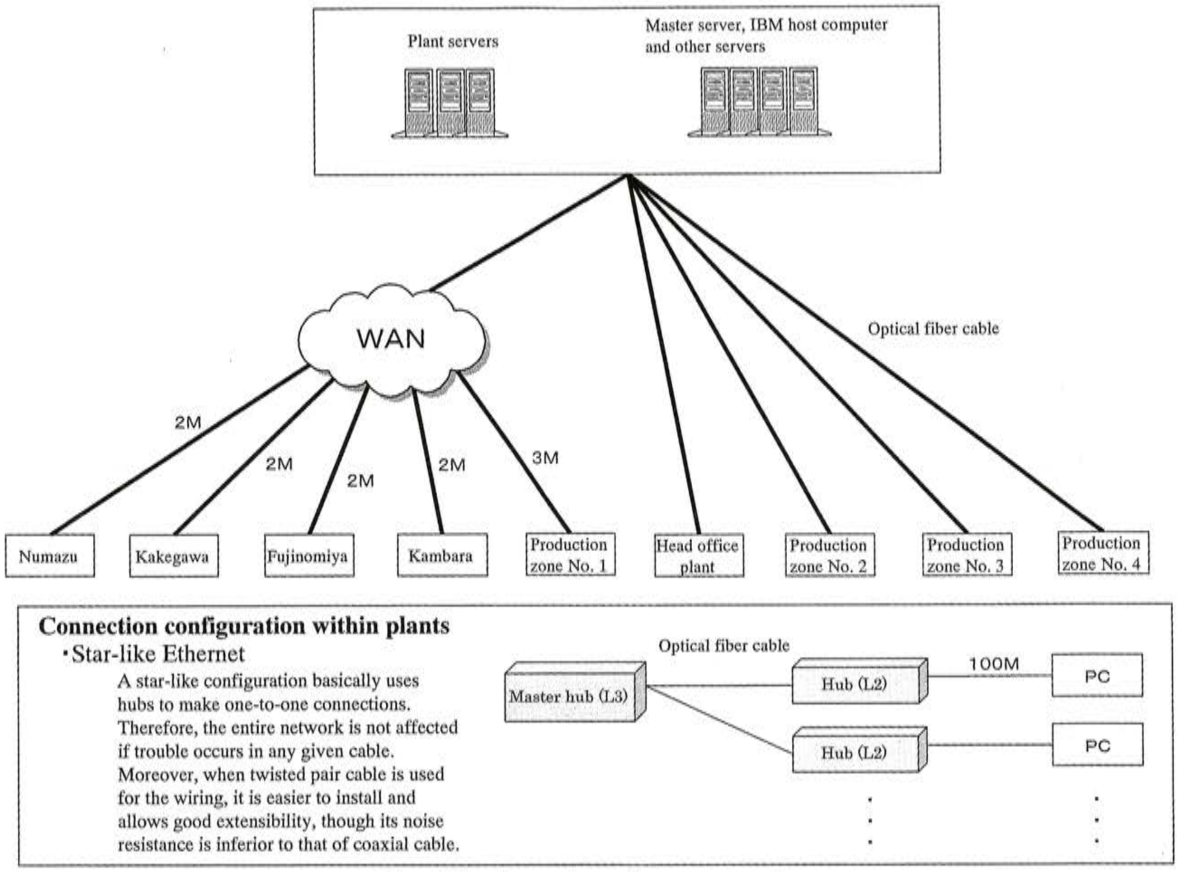


Fig. 5 Configuration of new network

新システムでのネットワーク構成は各工場に分散していた工場サーバを本社に一極集中させたため，工場間の回線速度を重視したネットワーク構成の構築を行った．(Fig. 5参照)

工場間接続については本社に隣接した2・4地区については直接光ケーブルで敷設，基幹部分の2重化を行い，信頼性を向上させた．
遠隔地については日本テレコムのIP-BPN(Solteria)で接続，回線速度を従来の256Kから2Mへ増速した．これにより，日本テレコムが監視でき，トラブル時の迅速な対応が可能となり信頼性が向上した．

For JEMS, the network has been configured with emphasis on the line speed of the connections to the plants. That was achieved by concentrating the plant servers solely at the head office, instead of distributing them among the plants as was done previously (Fig. 5).
For connections to the plants, optical fiber cable has been installed directly to production zones No. 2 and No. 4 located adjacent to the head office, and the core transmission lines have been made redundant for enhanced reliability.

More distant plants are connected via Japan Telecom's Solteria IP-VPN service, and the line speed has been increased from 256 KB to 2 MB. As a result, Japan Telecom can monitor the network and respond immediately whenever a problem occurs, which enhances reliability.

工場内についてもトラブル発生時の影響を極力少なくするよう接続形態を従来のバス型からスター型イーサネットへ変更した．基幹配線はギガビットイーサネット(1000M)を採用，支線配線は従来の10Mから100Mへ増速した．

また，機器が高度化したことにより，ネットワーク監視ができるようになり，トラブル発生時の迅速な対応が可能になった．

4.今後の課題

本システムは生産ラインと密接な関係にあり，システムが停止すると，組立ラインの停止につながり，ひいてはお客さまへも影響を及ぼしかねない．ネットワークのトラブルでも同様の現象が発生してしまう．

今後は，システム及びネットワークのトラブル発生時でも組立ラインを停止させないようシステムに改善していく．

5.終わりに

最後に，本システムを開発するにあたりご協力頂きました富士通(株)および，社内関係者の方々にお礼申し上げます．

The connection configuration within the plants was changed from the previous bus-based Ethernet to a star-like Ethernet so as to minimize as much as possible the impact of any network trouble. A gigabit (1,000 MB) Ethernet was adopted as the core wiring, and the capacity of the branch wiring was increased to 100 MB, up from 10 MB previously.

Thanks to the use of more sophisticated devices, the network can now supervise the equipment, making it possible to respond immediately whenever a problem occurs.

4. Future Issues

Because JEMS is integrally related to the production lines, a system shutdown can cause the assembly lines to stop, which, in turn, might also adversely affect our customers. Network troubles could also have the same impact.

In the future, we want to improve JEMS so that troubles in the system or network do not cause the assembly lines to shut down.

5. Concluding Remarks

The authors would like to thank various individuals at Fujitsu Ltd. and within the company for their cooperation in connection with the development of JEMS.

■ Authors ■

Yuji YONEKURA   Masakazu KANAMORI   Mitsuaki OMURA

Topics

第37回東京モーターショー2003への出展

Exhibitions at the 37th Tokyo Motor Show in 2003

小野田　司 *
Tsukasa ONODA

抄　録　'03年開催の東京モーターショーに出展し，『Intelligent Powertrain System Supplier』として,先進技術・新商品をPRした．展示会における駆動系技術の動向及び当社の展示内容について解説する．

Summary　We presented exhibitions at the 2003 Tokyo Motor Show to publicize our advanced technologies and new products as an intelligent powertrain system supplier. This article reviews the trends in powertrain technologies at the motor show and describes our exhibits.

1. はじめに

21世紀に入り，世界の自動車産業は，メーカ間のアライアンス・グループ化，地球環境問題への取り組み，クルマとITの融合，一層高度化が求められている安全技術，加えてユーザーニーズの多様化と顧客満足度の要求など様々な課題に直面している．本社会の要請に応えるべく，第37回東京モーターショー(注1)は『いま，挑む心．Challenge & Change(希望，そして確信へ)』をテーマに開催され，世界の乗用車・二輪車・部品メーカが次世代技術，新商品の発表・展示を競った．

(注1) 2003年10月22日から11月5日開催．
(プレスデー10月22・23日，特別招待日24日一般公開日25日より)出展社数約300社,入場者数142万人．

1. Introduction

The global automotive industry is currently facing a myriad of issues at the beginning of the 21st century. These include the formation of strategic alliances and groups among automakers, efforts to address global environmental concerns, the fusion of vehicles and information technology (IT), demands for increasingly more advanced safety technologies, and also the diversification of user needs and more rigorous requirements for customer satisfaction. The 37th Tokyo Motor Show*1 was organized around the theme of "The challenge: driving toward a better future." Passenger car and two-wheeled vehicle manufacturers and parts makers from around the world vied in announcing and presenting next-generation technologies and new products designed to meet society's needs.

*1 Held from October 22 to November 5, 2003. (October 22 and 23 were Press Days, October 24 was designated as a Special Guest Day, and general admission began on October 25.) Exhibiting companies numbered about 300, and show attendance totaled 1.42 million visitors (Fig. 1).

Fig. 1 Makuhari Messe

* 営業業務部
Sales Planning & Administration Department

2. トレンド

国内での車両販売の本格的な回復が遅れている背景もあり，カーユーザの購入を促進し，'03年国内車両販売目標の585万台を達成するため，未来のドリームカーの出展に加えて，'03年に販売を開始した新車，市場投入を予定しているプロトタイプ車の展示が増えている傾向にある．そのため，華やかな中にも，欧米でのモーターショーのように落ち着いたトレード・ショウの意味合いが各所に見られた．

今回はモーターショーに足を運んだことのない若年層，女性などの新規の来場者が多く，特に，若者に人気のカロッツェリアの展示，家族で楽しめるフェステバル・パークなどイベントの用意，小学生の無料化・高校生を大人の半額に料金設定するなどの工夫がなされていた．また，燃料電池・ハイブリッドカーなどへの試乗を体験できる来場者参加型の双方向コミュニケーションのイベントや将来の日本を担う子供にモノ作りに触れてもらう工夫がなされていた．

また，グローバルなカーメーカ間のアライアンス化も年々加速しており，日産・ルノー，三菱・ダイムラークライスラー，スズキ・富士重・GM，マツダ・フォード等が，各社の独自性を保ちながらも，各グループ単位で隣接したエリアに出展していた．

地球環境との共生への取り組みに関しては，様々な研究・開発が紹介されていた．

燃料電池車は1801年に英国で原理が発見され，将来のエネルギー源の本命として水素と酸素を化学反応させて，電気を発生させる機構であるが，日産『EFFIS』のように，燃料電池スタックと新開発のスーパーモータを組み合わせたシテイ・コミュータを始めとして，各社がコンセプト・カーを出展していた．

2. Trends

There tended to be more exhibits of new models rolled out in 2003 and of prototype models scheduled for near-future release, in addition to future dream cars. That was intended stimulate purchases by car users and thus attain the sales target of 5.85 million units set for the domestic market in 2003, partly against a backdrop of the delay in the full-fledged recovery of vehicle sales in Japan. As a result, a subdued trade show atmosphere like that of motor shows in Europe and North America was evident everywhere amid the spectacular displays.

The motor show this time drew many first-time visitors, including young people, women and others who had never gone to a motor show before. Various measures were taken to attract them, such as by displaying carrozzeria that are especially popular among young people, organizing special events like a festival park for families with children, charging no admission fee for elementary school pupils and reducing the ticket price for high school students to one-half the admission fee for adults. In addition, two-way communication events involving visitor participation were also organized such as test-drives of fuel cell vehicles and hybrid vehicles, and activities were staged for children to enable these future leaders of Japan to experience the world of manufacturing.

The formation of alliances between automakers has been gaining momentum every year. Nissan and Renault, Mitsubishi and DaimlerChrysler, Suzuki-Fuji Heavy Industries-GM, and Mazda and Ford, among others, presented exhibitions on a group basis in adjacent areas, while still retaining each company's unique identity.

The results of various research and development activities were introduced in connection with efforts to promote harmony with the global environment. Various companies exhibited eco-friendly concept cars, such as the Nissan EFFIS (Fig. 2) that combined a fuel cell stack with the newly developed Super Motor to create a city commuter. The working principle of a fuel cell vehicle was discovered in the United Kingdom in 1801. As a device that produces electricity through chemical

Fig. 2 Nissan EFFIS

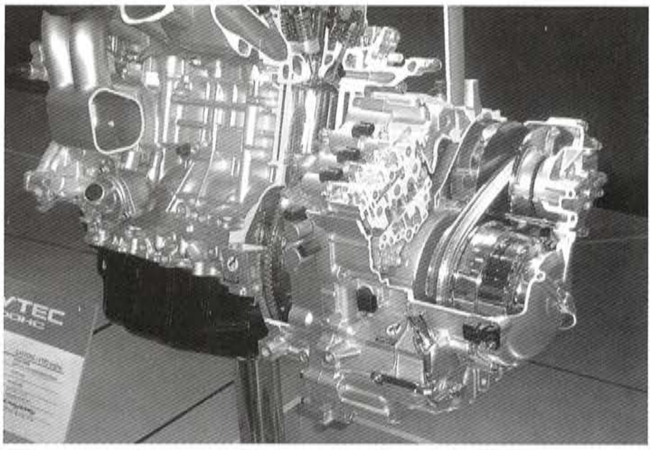


Fig. 3 Honda CVT

燃料電池車はクルマとしての走りの機能・性能は向上しつつあるが，コスト課題・水素供給のインフラ整備などの観点から，普及に時間が掛かると見られ，当初中継ぎ的な位置付けであったハイブリッド車も脚光を浴びていた．

トヨタ・ホンダ・日産が'97年頃からハイブリッド車を市場投入し，スズキが当社製4速ATを搭載した新システムを軽自動車に採用したことなどにより，益々，一般ユーザに身近な自動車となりつつある．欧州カーメーカが，デイーゼル・エンジンを当面の燃費改善の本命としている中で，日系メーカ及び，部品メーカ各社は，今回，パラレル方式，シリーズ・パラレル方式のハイブリッド車を紹介していた．

また，駆動系として，ベルト式CVTを各社が出展しており，ホンダは新規に市場投入した2.4Lエンジン用，日産は北米市場などで高い評価を受けた3.5L用エクストロニックCVT(当社製CVT)，三菱は優れた居住性と斬新なスタイルを実現したMRコンセプトモデル『i』へのCVT搭載を訴求していた．

駆動系部品メーカとして，LuKがAudi製のチェーン・ベルト式CVT，アイシンAWが小型車用ベルト式CVT，FR車用に日本精工はハーフ式，光洋精工はフル・トロイダルを出展し，実用化を目指した開発品を訴求していた．

日本市場は，'03年には約80万台のCVT車が販売され，今後，普及が急速に加速し，北米市場がこれに追随する見込みである．一方，欧州などの市場で普及しつつある多段AT，AMT(自動MT)と関連電子部品として，アイシンAWがFF，FR用6速AT，LuKブースではAMT，Siemensはベンツが発表した内製FR7速AT用起電一体型ATCUを展示していた．

3. 当社出展のコンセプト

3.1. テーマ

'03年4月1日のジヤトコ(株)とダイヤモンド・マチック(株)の事業統合により誕生した新生ジヤトコとして，初めてのモーターショーへの出展となった．世界最大規模のAT専門メーカとして，今後のグローバルな市場の拡大と多様化するニーズに的確に対応できる開発・生産技術力の高さと，魅力ある新商品の訴求をコンセプトに織り込んだ．

reactions between hydrogen and oxygen, the fuel cell is the odds-on favorite to become the energy source of the future.

Fuel cell vehicles are steadily improving in their operational functions and capabilities as automobiles, but their diffusion is projected to take considerable time owing to cost issues, construction of the supply infrastructure for hydrogen and other factors.

Also in the limelight were hybrid vehicles, which from the beginning have been positioned as playing an intermediary role. Toyota, Honda and Nissan began putting hybrid vehicles on the market around 1997, and Suzuki has adopted a new hybrid system in a minicar fitted with a JATCO 4-speed AT. These developments are making hybrid vehicles increasingly more familiar to ordinary users. European car makers view the diesel engine as the favorite technology for improving fuel economy in the foreseeable future. Japanese automakers and parts manufacturers, on the other hand, displayed hybrid vehicles with parallel systems or with series and parallel systems at the 2003 show.

Many companies emphasized steel-belt CVTs as their drivetrain systems. Honda showed a newly released CVT for use with a 2.4-liter engine (Fig. 3). Nissan exhibited the XTRONIC CVT (made by JATCO) that can accommodate a 3.5-liter engine and has been highly acclaimed in North America and other markets. Mitsubishi displayed its i concept model fitted with a CVT and built on the MR (midship/rear-wheel-drive) platform that provides outstanding interior comfort and fresh vehicle styling.

Drivetrain component manufacturers displayed the CVT products they have developed aimed at production vehicle implementation. LuK exhibited a chain-drive CVT with an Audi-made steel chain, Aishin AW displayed a steel-belt CVT designed for small-car application, and for use on rear-wheel-drive cars Nippon Seiko presented a half-toroidal CVT and Koyo Seiko showed a full-toroidal CVT.

Approximately 800,000 CVT-equipped vehicles were sold in the Japanese market in 2003. Their diffusion is expected to accelerate rapidly in the coming years, and the North American market is projected to follow this trend. On the other hand, there were also exhibits of electronic components for stepped ATs and automated manual transmissions (ATMs), which are steadily spreading in Europe and other markets. Aishin AW displayed such components for front- and rear-drive 6-speed ATs. The LuK booth exhibited ATM components, and Siemens presented an AT control unit (ATCU) integrated with an electric motor for application to a rear-drive 7-speed AT manufactured in-house by Mercedes-Benz.

当社は'97年に2Lクラス・'02年に3.5Lクラスエンジンに適応するベルト式CVT，並びに，'99年にはトロイダル式CVTの量産化に世界で初めて成功した実績を持っている．CVTのリーディング・カンパニーとして，排気ガスの低減・燃費向上などの環境への前向きな取り組みの一例として，軽自動車から3.5L用エンジンをカバーする新世代ベルト式CVTシリーズを中心に，トロイダルCVT，多段AT，HEVモデルを展示した．

また，一般の来場者には，ブラック・ボックスである自動変速機の作動と変速の原理を理解して頂くため，『Touch & Try』コーナを設け，遊星ギア，ベルト式CVT，トロイダルCVTの各原理モデルを来場者が自由に作動させることのできる体験型コーナを設けた．

3.2. ブース・デザイン

ダーク・グリーンの社名ロゴマークの視認性を高め，先進的な展示商品を効果的に訴求するため，ブース構造物は，他社に類のない純白を基調としたものとした．

メイン通路に面したコーナには，遊星歯車をイメージした企業理念の大型電照オブジェを設置し，その両側に次世代ベルト式CVT4台を配備した．

ブースは,カーメーカ等のお客様と落ち着いて懇談・プレゼンテーションができる会議室エリアと一般の来場者の方々が気軽に展示品を見学でき，技術に興味のあるエンジニア系学生の方々が気楽に懇談できるオープン・エリアの2つの機能に分離した．

オープンエリアには，各変速機グループ別の展示コーナを設け，それぞれの構造と特長を理解し易いものとした．特に，変速原理モデルの体験コーナを，通路側に配備することで，来場者が気楽に手で触れ，ブース内の混雑を回避することができる工夫をした．

Fig. 4 JATCO booth

3. Concept of JATCO's Exhibits

3.1. Theme

This was our first Tokyo Motor Show since the creation of the new JATCO through the integration of our transmission business with Diamondmatic Co., Ltd. on April 1, 2003. The concept embodied in JATCO's booth was to emphasize the high level of our development and production engineering capabilities and our attractive new products as one of the world's largest specialized AT manufacturers. Our aim was to show visitors that we have the technologies and products for accurately meeting users' diversifying needs and the requirements of the expanding global market in the coming years.

JATCO has a track record of being the world's first manufacturer to successfully mass produce a steel-belt CVT for 2.0-liter class engines in 1997, a steel-belt CVT applicable to 3.5-liter class engines in 2002 and a toroidal CVT in 1999. As a leading CVT manufacturer, we showcased our next-generation steel-belt CVT series, ranging in application from minicars to vehicles fitted with a 3.5-liter engine, along with displaying exhibits of toroidal CVTs, stepped ATs and transmissions for use on HEVs. These exhibits were examples of our proactive efforts to address environmental concerns by reducing exhaust emissions and improving fuel economy.

The booth also featured a "Touch & Try" area designed to give ordinary visitors a better understanding of the operation and shifting principle of automatic transmissions, which seem like a black box to many people. Working models of a planetary gearset, a steel-belt CVT and a toroidal CVT were displayed to enable visitors to gain hands-on experience by operating the models freely by themselves.

3.2. Booth design

The structural elements of the booth were designed with a pure white motif that was without parallel at other companies' booths. That was intended to enhance visitor recognition of the dark-green corporate logo expressing the company's name and effectively impress visitors with our advanced product exhibits (Fig. 4).

A large illuminated object modeled after a planetary gear and expressing our corporate philosophy was displayed in the exhibition space facing the main aisle. Flanking that object on both sides were four exhibits of next-generation steel-belt CVTs.

The booth was functionally divided into two areas. One was a conference room area where discussions and presentations could be conducted with automakers and other customers in a relaxed atmosphere. The other was an open exhibition area where ordinary visitors could readily view the exhibits, and engineering students interested in technology could converse casually.

The open exhibition area was divided into separate exhibition spaces for each group of transmissions so that their respective construction and features would be easier to understand. The working transmission models for hands-on experience in particular were located along the aisle so as to avoid the congestion inside the booth and readily enable visitors to touch these exhibits with their hands.

4. プレス・ブリーフィング

国内外数十社の報道機関の方々が当社ブースに来られた．内容は1)『グローバル・サプライヤとしての当社の事業状況』2)『新技術・新商品』に関するものに大別される．前者は世界トップレベルの技術力とAT生産台数を有する専門メーカの事業状況，特に,年間販売台数の約30%以上が，完成車両・ATユニットとして北米市場に輸出され，本市場でも今後CVT化が進むこと，当社が既にプレス発表したメキシコ新工場でのCVT生産の具現化に関してであり，一般紙・業界誌の期待と質問が寄せられた．

後者は軽自動車から大型車をカバーする次世代のベルト式CVTシリーズ，世界で初めて量産に成功したトロイダルCVTの取材は勿論のこと,日本市場を中心に普及しつつあるハイブリッド車への関心の高まりを反映して,従来のトランスミッションのスペースでハイブリッド化できるI-HATシステムに興味が集まった．

また，NHK特集として，各カーメーカの過去・現在・将来のクルマ作りと環境対応などをテーマとした『車を通して未来を見つめる東京モーターショー』のインタビュウを部品メーカとして唯一当社小島社長が受け，11月1日(2日・3日は再放送)に70分番組として放映された.(Fig. 5)

Fig. 5 NHK interview

5. 展示品

5.1. ベルト式ＣＶＴシリーズ

FF車用２Lクラスの CVTなど100万台以上の市場実績を基に，既に量産化，または開発中の次世代CVTをフルラインナップで出展した.従来のCVTに比べ，コンパクトで，しかも，超ワイドな変速比巾とすることで,発進加速性能と燃費の両方を大幅に向上した次世代商品.

4. Press Briefing

Journalists from several tens of domestic and overseas media companies came to our booth for a press briefing. They were generally interested in two broad subjects: (1) the current status of JATCO's business as a global supplier and (2) our new technologies and new products.

The first subject concerned the situation of JATCO's business as a specialized manufacturer possessing world-class technologies and with one of the largest AT production volumes worldwide. Representatives of general print media and automotive industry publications especially voiced questions and expectations about the future penetration of CVTs in the North American market. Currently, approximately 30% of the transmissions we sell annually are exported to that market either in finished vehicles or as AT units. They were also interested in the details surrounding the launch of CVT production at our new plant in Mexico, which we had announced earlier in 2003 that we were planning to establish.

The second subject naturally concerned our next-generation steel-belt CVT series, ranging in application from minicars to large cars, and toroidal CVTs that JATCO was the first in the world to mass-produce successfully. In addition, attention was also focused on the Integrated Hybrid Automatic Transmission (I-HAT) system that enables a hybrid powertrain to be mounted in the space of a conventional AT. That reflected the rising interest in hybrid vehicles, which are beginning to penetrate the market, especially in Japan.

NHK (Japan Broadcasting Corporation) created a TV special entitled, "The Tokyo Motor Show--Gazing at the Future through Vehicles." The program looked at the past, present and future of automobile manufacturing and environmental protection measures at vehicle manufacturers. JATCO President and CEO Hisayoshi Kojima was the only head of a parts manufacturer to be interviewed by NHK (Fig. 5). The 70-minute program aired on November 1 and was rebroadcast on November 2 and 3.

5. Exhibits

5.1. Steel-belt CVT series

The full lineup of our next-generation CVTs was displayed, including units already in mass production and others still under development. This series is based on our experience of having sold over one million CVTs already, including CVTs for 2.0-liter class front-wheel-drive cars. These next-generation products are more compact than previous CVTs and also have an ultra-wide ratio range, among other features that contribute to substantial improvements in both acceleration performance and fuel economy (Fig. 6).

(1)軽・サブコンパクト車用次世代ＣＶＴ
富士重工業(株)と当社との共同出資で『富士AT株式会社』を設立し，富士重工業と当社との共同開発により，世界最高レベルの軽自動車・サブコンパクト車用CVTの供給を予定.(外観モデルを展示)

(2)小型車用次世代CVT(JF009E)
'02年にフルモデル・チェンジした日産新型キューブに搭載された軽量・コンパクトで超ワイドな変速比幅を有するCVT(カットミッションを展示)

(3)中型車用次世代CVT
2.5Lクラスのエンジンに適合する新規開発中CVTであり，変速比幅の拡大などにより，クラストップレベルの高効率を実現(外観モデルを展示)

(1) Exterior model of a next-generation CVT for minicars and subcompacts
Fuji Heavy Industries Ltd. (FHI) and JATCO jointly financed the establishment of Fuji AT Ltd., which is scheduled to supply world-class CVTs for minicars and subcompacts. These CVTs will be jointly developed by FHI and JATCO.

(2) Cutaway model of the JF009E next-generation CVT for small cars
This CVT was adopted on the new Nissan Cube when it underwent a full model change in 2002. Lightweight and compact, it features an ultra-wide ratio range.

(3) Exterior model of a next-generation CVT for midsize cars
Now under development, this all-new CVT will be mated to 2.5-liter class engines. By expanding the ratio range and making other improvements, this CVT will achieve the highest levels of efficiency in its class.


Fig. 6 Next-generation CVTs

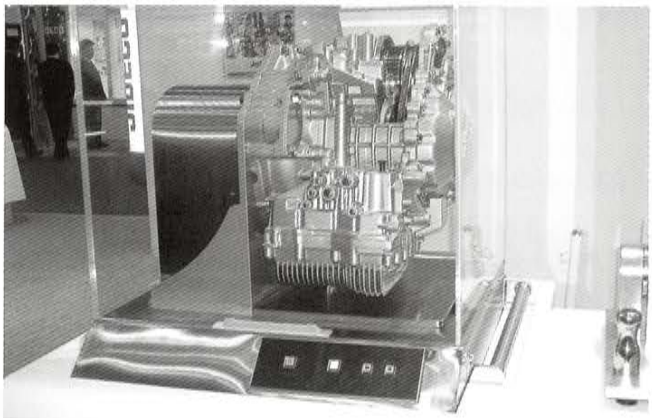


Fig. 7 High-torque capacity CVT

(4)大型車用次世代CVT(JF010E)
'02年に世界で初めて，FF3.5Ｌ用の量産に唯一成功し，日産 ムラーノ(北米SUV)，テイアナに搭載されているFF大型車用新世代CVT(JF010E)をベースに制作した作動モデル．プーリの溝の幅を変化させることで，プーリに掛かるスチール・ベルトの入力側と出力側各々の直径を滑らかに変化させ，無段階の変速の機構を視覚的に理解できるため，各来場者から，変速状況が判り易いとの評価を頂いた．CVTは小型車用の変速機との認識が覆り，このクラスまでCVT化が進んだのかとの驚嘆の声が多い．

(4) Working model of the JF010E next-generation CVT for large cars
This working model was created around the JF010E new-generation CVT for large front-wheel-drive cars that is used on the Nissan Murano SUV sold in North America and the Nissan Teana marketed in Japan. In 2002, the JF010E became the world's first and only successfully mass-produced CVT for use on 3.5-liter front-wheel-drive cars. The JF010E varies the width of the pulley grooves to smoothly change the diameter of the steel belt on the pulleys on both the input and output sides. Because the seamless ratio change mechanism could be understood visually, visitors rated the model highly for making it easy to understand how ratio changes are executed. Many visitors were astonished that CVTs have advanced as far as the 3.5-liter engine class. This model overturned the general perception that CVTs are transmissions for small cars.

5.2. トロイダルＣＶＴ(JR006E)

トルクコンバータ内に組み付けた 展示品用モータの駆動により,内部のパワーローラ4セットがロー側からハイ側に自動的に動き,滑らかな変速状態が視認できる. また,ボタン操作による8段変速のマニュアルシフト機構をつけ,スポーティ走行への対応技術をPRした.

5.2. JR006E toroidal CVT

A display-use motor was incorporated into the torque converter to drive the four sets of rollers in the CVT automatically from low to high transmission ratios, enabling visitors to see its smooth shifting action. An 8-speed manual shift mechanism operated by buttons was incorporated in the CVT to publicize its technological capability for providing sporty driving.

5.3. ステップ式AT

多段化に対応した5速・6速ATの中から，4台を出展した.

(1)FF車用5速AT

三菱エアートレック・ターボ車などに搭載されているエンジンの高性能化に対応する大トルク容量用ATであり，INVECS－Ⅲに対応した走行状況に応じたシフトコントロールを実現.(カットモデルを展示)

(2)大型FR車用5速AT(JR507E)

世界トップレベルの変速性能・高効率・軽量設計を実現し,日産シーマ/スカイライン搭載された5速AT(カットモデルを展示)

(3)FF車用6速AT

大トルクと中トルク容量エンジン用の2台(開発中)を展示. Lepelletier system採用によるコンパクトな形状により優れた搭載性を実現し,超ワイドの変速比を有する6速AT.(外観モデル)

5.3. Stepped ATs

Four models were chosen for display from among our 5-speed and 6-speed ATs featuring additional speed ranges.

(1) Cutaway model of a front-drive 5-speed AT

With its large torque capacity, this AT is designed to handle high-performance engines like those fitted on the Mitsubishi Airtrek Turbo and other models. It provides shift control that matches the operating conditions and is compatible with the INVECS-III shift schedule.

(2) Cutaway model of the JR507E 5-speed AT for large rear-wheel-drive cars

Featured on the Nissan Cima and Skyline, this 5-speed AT ranks among the world's best in terms of shifting performance, high efficiency and lightweight design.

(3) Exterior models of front-drive 6-speed ATs

Two models of 6-speed ATs now under development were displayed, one for large-torque engines and the other for medium-torque engines. These ATs adopt a Lepelletier gearset system to achieve a compact package for superior vehicle mountability and also have an ultra-wide ratio range.

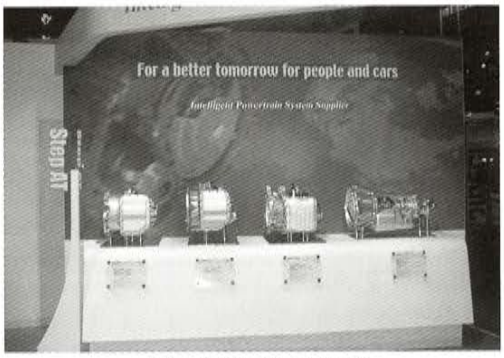


Fig. 8 Stepped ATs

5.4. ハイブリッド用トランスミッション(T/ M)

当社は燃費向上のため，上記のCVTや多段ATの開発，ミッション内のフリクション低減などの様々な対応，更に，ハイブリッド車に対応した開発も進めている．今回，当社独自に研究・開発のI-HATシステムとカーメーカが開発されたハイブリッド・システムに適合するT/Mの2例を出展した.

5.4. Transmissions for hybrid vehicles

JATCO is taking various steps to improve vehicle fuel economy, such as by developing the above-mentioned CVTs and stepped ATs and by reducing internal friction in transmissions, among other things. Moreover, we are also developing transmissions for hybrid vehicles. At this motor show, we exhibited the I-HAT system that we are researching and developing independently and a transmission compatible with the hybrid system developed by an automaker (Fig. 9).

(1) I-HATシステム(研究開発中)

従来の自動変速機とほぼ同じスペースでエネルギ回生機能を持つ当社独自のハイブリッド・システム.遊星歯車の採用により,スタートクラッチやトルクコンバータを廃止し,駆動・発電兼用の小型モータを採用.(カットモデルを出展)

特に,日本市場でのニーズの高まりにより,近い将来,駆動系のひとつの柱となるハイブリッド用T/Mに一般来場者の方々の関心が極めて高い.

(2)スズキ・ツイン用HEVシステム

スズキ(株)の開発・量産されたシステムに当社のFF4速AT(JF405)が採用され，エンジンの後部に回生モータなどを組み込んでもジャスト・フィットするコンパクトで高効率なATを実証.(エンジン外観モデルと，回生モータ部・AT部のカットモデルを展示)

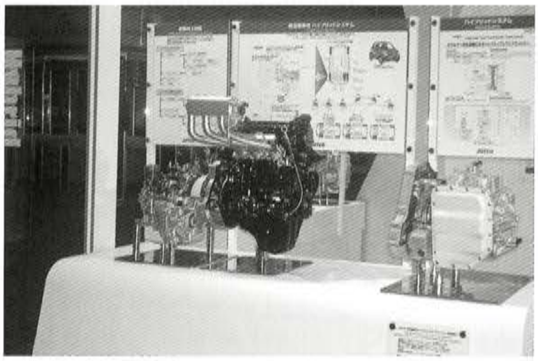


Fig. 9 HEV systems

5.5. 手回し動モデル

エンジン駆動の代わりに入場者自らがハンドルを回し，各変速機構部のローとハイ側切り替えレバーを動かす3種類の体験型モデルを設置し,変速原理の理解を深めた.

(1) 遊星歯車セット
(2) トロイダルCVTパワーローラ部
(3) CVT金属ベルト・プーリ部

6. まとめ

(1)モーターショー全体の基本コンセプトとして，お客様参加型の新しいスタイルを取り入れながら，各社が環境・IT・安全の3つのキーワードの技術革新の具現化を訴求した展示会となった．また，メーカ間のアライアンス・グループ化の傾向が見られた.

(1) Cutaway model of the I-HAT system under development

I-HAT is JATCO's unique hybrid system that includes an energy regeneration function, yet it installs in virtually the same space as a conventional AT. The use of a planetary gearset eliminates the need for a start-off clutch and a torque converter. A small motor has been adopted that doubles as a power source and a generator. Ordinary visitors showed an extremely strong interest in this transmission for use with hybrid systems, which are expected to become a core powertrain technology in the near future, owing to growing needs for such systems in the Japanese market in particular.

(2) HEV system for the Suzuki Twin

JATCO's JF405 front-drive 4-speed AT was adopted for the Twin hybrid car that Suzuki Motor Co., Ltd. has developed and is mass producing. The fact that this AT just fits in the space behind the engine, even when a motor for regenerating kinetic energy is installed, testifies to its compact design and high efficiency. An engine exterior model was displayed along with a cutaway model of the energy-regenerating motor and AT.

Fig. 10 Dynamic model in Touch & Try area

5.5. Manually operated models

Three types of transmission models were exhibited that visitors themselves could operate by turning a handle, instead of being driven by an engine. The hands-on experience of moving a lever to switch the ratio change mechanism between low and high ratios helped to deepen their understanding of the shifting principle.

(1) Planetary gearset
(2) Power rollers of a toroidal CVT
(3) Steel belt and pulley assembly of a CVT

6. Recap

(1) The basic concept running throughout the motor show was one that emphasized the exhibitors' technological innovations in the three key areas of the environment, IT and safety, while also incorporating a new style of visitor participation exhibits designed for hands-on experience. The recent trend toward the formation of alliances and groups among automakers was also evident.

(2) 地球環境との共生への取り組みの中で，高効率化・軽量化の要請などトランスミッションへの期待は年々高まっており，各社独自の新商品・コンセプトモデルの展示をしていた.

(3) '03年4月に発足した新生ジヤトコとして，初めてのモーターショーへの出展となった. Intelligent Powertrain System Supplierとして,先進的な商品である次世代ベルト式CVT, トロイダルCVT,5速・6速の多段AT, I-HATなどのハイブリッド・システムを展示し， カーメーカ,及び自動車産業の方々を始め，プレス関係者に技術力の高さと商品力を訴求することができた.

(4) 当社ブースは，市場のニーズの多様化と技術革新が進む中で，CVT，多段AT，ハイブリッドシステムの展示エリアを独立させ，トランスミッションの知識がない一般の来場者の方々も理解し易いものとした.前回に引き続き,欧米の博物館のコンセプトを取り入れ，各種トランスミッションの体験型作動モデルを配備し，来場者自らその機構・原理を理解できるもとした.

7. 最後に

今回，世界有数のカーメーカ,報道機関などのご来賓をはじめとして,多くの一般来場者の方々に当社ブースにお立ち寄り頂き,また,長期間の出展企画・展示品の諸準備に携わって頂いた関連会社,及び,商品企画統括部・先行技術開発部・商品開発各部・材料工法開発室・人事部・サービスサポート部・経営企画部・総務部・営業各部などの多くの社内関連部門，全体事務局　第二営業部企画グループ　若月・安田両氏のご協力・ご尽力に対して，感謝の意を表します.

(2) Amid efforts to promote harmony with the global environment, expectations of transmissions are rising every year, including demands for higher efficiency and lighter weight. Each manufacturer displayed its own new products and concept models.

(3) This was our first Tokyo Motor Show since the creation of the new JATCO in April 2003. As an intelligent powertrain system supplier, we displayed our advanced products represented by next-generation steel-belt CVTs, toroidal CVTs, 5- and 6-speed stepped ATs, and hybrid systems such as the I-HAT system. These exhibits emphasized our cutting-edge technological capabilities and attractive products, especially to the automakers and others in the automotive industry as well as to media representatives.

(4) Our booth was divided into separate exhibition areas for CVTs, stepped ATs and hybrid systems, against a backdrop of diversifying market needs and ongoing technological innovations. This layout made it easier even for ordinary visitors without any knowledge of transmissions to understand the exhibits. Continuing the approach used at the 2001 show, we adopted the concept of a European or American museum and displayed working models of various types of transmissions that enabled visitors to understand the structure and operating principles through personal hands-on experience.

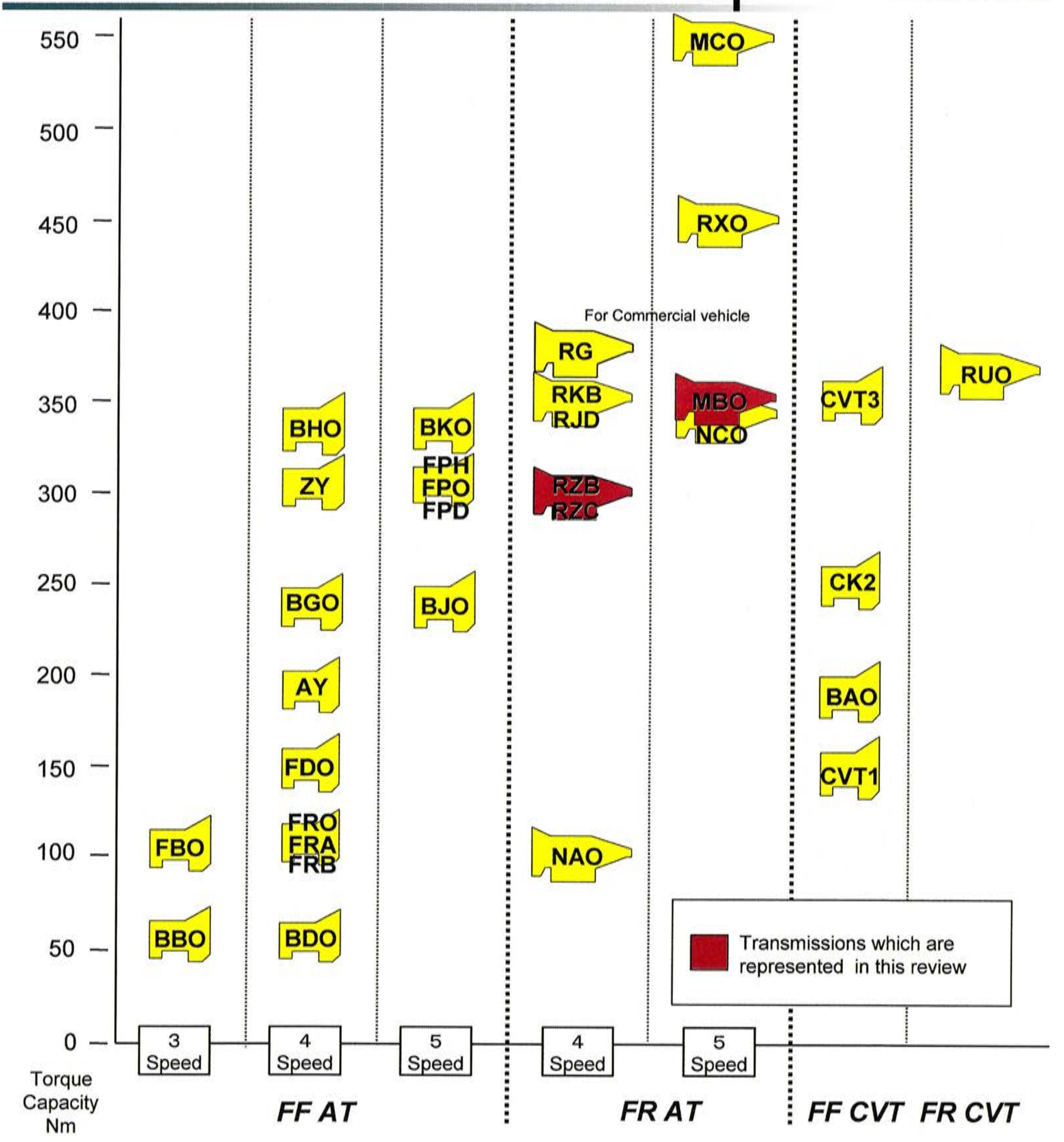
7. Concluding Remarks

Our booth was visited by representatives of the world's leading automakers, media organizations and other special visitors as well as by large numbers of the general public. The author would like to take this opportunity to thank various individuals at affiliated companies and many related departments within JATCO for their valuable cooperation and assistance with the planning and preparation of the exhibits over many months. Thanks are due the Product Planning Administration Department, Advance Technology Development Department, Product Development Groups, Materials & Process Development Department, Human Resources Development Department, Services Support Purchasing Department, Corporate Planning Department, General Administration Department, Sales & Marketing Departments, Motor Show Secretariat and to Messrs. Ms. T. Wakazuki and Mr. T. Yasuda of the Planning Group in Sales & Marketing Department No. 2.

■ Author ■

Tsukasa ONODA

Product Line-up



新商品紹介

スポーツカー適用の4速AT JR405E(RZC)の紹介

Introducing the JR405E (RZC) 4-speed AT for Sports Car Applications

斉藤　譲 *　　斉藤　祐司 **
Yuzuru SAITOU　　Yuuji SAITOU

抄　録　JR405E型FR用4速自動変速機(以下ATとする)は，従来のJR405E-M(RZO)型FR用4速ATをベースにマツダ様のスポーツカーRX-8に搭載するべく開発され，2003年4月9日に発表・発売された．
本報では，このAT開発の成果などを紹介する．

Summary　The rear-drive JR405E 4-speed automatic transmission (AT) was developed for use on the Mazda RX-8 sports car, based on the previous JR405E-M (RZO) 4-speed AT for application to rear-wheel-drive cars. This new AT was announced and released on April 9, 2003. This article describes the results achieved through the development of the JR405E AT.

1. はじめに

JR405E-M(RZO)型FR用4速ATは，1999年にマツダ様の車両(その後日産自動車様，三菱自動車様にもOEM供給)に搭載されたクラッチ油圧直接電子制御(Direct Electronic Shift Control : DESC)式ATであった．今回新開発のJR405E型ATは，これをベースに，マツダ様の本格的スポーツカーRX-8(Fig. 1)に搭載するべく開発された．

本ATは，7,500rpmという高回転に対応しながら，スポーツカーにふさわしくクィックさと滑らかさとが両立するシフトクォリティを実現した．

2. 適用車両及びATの主要仕様

適用した車両とATの仕様をTable 1に示す．

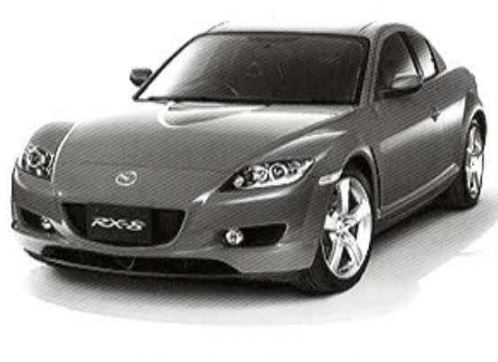


Fig. 1 MAZDA RX-8

1. Introduction

The rear-drive JR405E-M (RZO) 4-speed AT, incorporating the Direct Electronic Shift Control (DESC) system for direct electronic control of clutch pressures, was adopted on a Mazda model in 1999. It was also subsequently supplied to Nissan Motor Co., Ltd. and Mitsubishi Motors Corp. on an OEM basis. Based on that existing transmission, the JR405E AT was newly developed for use on Mazda's RX-8 authentic sports car (Fig. 1).

The JR405E AT achieves shift quality that provides both smoothness and quickness befitting a sports car, while also accommodating high operating speeds up to 7,500 rpm.

2. Major Specifications of Vehicle and AT

Table 1 gives the major specifications of the RX-8 model and the JR405E (RZC) AT.

Table 1 Technical specifications

Vehicle	Model		RX-8
	Drive type		Rear-wheel drive
	Weight		1,330 kg
	Engine type		Rotary engine (RE) 654 cc×2 13B-MSP
	Max. power (DIN)		154 kw (210 ps)/7,200 rpm
	Max. torque (DIN)		222 Nm/5,000 rpm
	Max. speed (rpm)		7,500
AT	Torque converter		250 mm dia.with lock-up
	Gear ratio	1st	2.785
		2nd	1.545
		3rd	1.000
		4th	0.694
		rev	2.272
	Dry weight (kg)		66.4
	Communication with vehicle		CAN
	Shift lever positions		P-R-N-D

* 第二商品開発本部
Product Development Center No.2
** 適用開発部
Application Development Department

3. 構造

本ATの縦断面図をFig. 2に示す.
パワープラントフレーム(PPF)*に対応するため,軽量化に配慮しつつケース類のリブ・R等を最適設計することで強度を確保した.
*PPFについて
急加速時にデフの上下揺動が,駆動力の伝達を遅らせる.このデフ揺動を規制するために,ATからデフまでをリジッドに結合したのがPPFである.それにより,アクセル操作に対するダイレクト感と優れた加減速レスポンスを実現した.

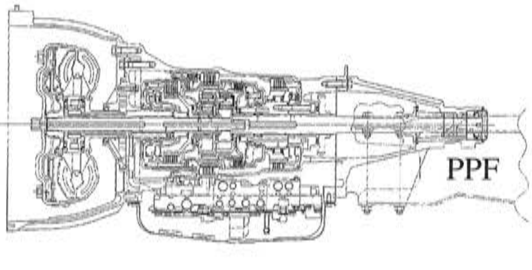


Fig. 2 Cross-sectional view of JR405E

4. 変速性能向上

4.1. 変速応答性の向上

マニュアルモードでのシフト操作や,アクセルペダル踏み込み時のダウンシフトでは,特に素速い応答性が望まれるので,以下を改良した.

(1)油圧応答性の改良
各クラッチへ供給する油圧を直接制御するソレノイドの回路部に設けた防振用アキュムレータの油圧特性をばね剛さの高い特性とした.

(2)マニュアルモード専用データの設定
マニュアルモードでのシフト操作の応答性を改良するため,クラッチピストンの無効ストローク時間とイナーシャフェーズ時間の短縮を狙って,Dレンジよりも高目の油圧を設定した.

(1),(2)の変更により,以下のように変速判断からイナーシャフェーズ開始までの時間をマニュアルモードで0.3sec以下(Fig. 3,4),踏込みダウンシフトで0.2sec以下(Fig. 5)とすることができた.

3. Construction

A cross-sectional view of the JR405E AT is shown in Fig. 2.

While giving consideration to weight reductions, the case was designed with optimal rib reinforcements and radii (R) to ensure sufficient strength for accommodating the powerplant frame (PPF).

The PPF serves to provide a rigid connection from the AT to the rear differential. This rigidity works to suppress the vertical vibration of the differential that occurs during hard acceleration and delays the transmission of driving force. As a result, the vehicle provides a feeling of direct response to the driver's accelerator inputs and achieves outstanding acceleration/deceleration responsiveness.

Comparison of G waveforms for manual downshifts
RX-8 (4→2 downshift) Model B (6→4 downshift)

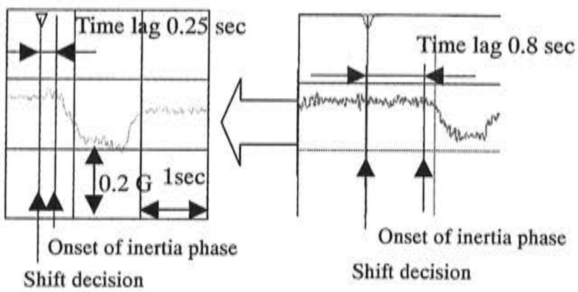


Fig. 3 Manual downshift waveforms

4. Improvement of Shift Performance

4.1. Improvement of ratio change response

Quick shift response is especially desired in manual shifting and when the transmission downshifts in response to the driver's depression of the accelerator pedal. The following improvements were made to provide that quickness.

(1) Improvement of oil pressure response
An accumulator for suppressing pressure oscillations is provided in the solenoid circuit that directly controls the pressure supplied to each clutch. The pressure characteristic of the accumulator is characterized by high spring rigidity.

(2) Dedicated settings for manual shift mode
To improve the responsiveness of manual mode shifting, the pressure is set slightly higher than the D range level. The purpose of this is to shorten the ineffective stroke time and inertia phase time of the clutch pistons.

The improvements in (1) and (2) above have made it possible to reduce the time from a shift judgment to the onset of the inertia phase to less than 0.3 second in the manual shift mode (Figs. 3 and 4) and in a power-on downshift to less than 0.2 second (Fig. 5).

4.2. 滑らかなアップシフト

オートモードのアップシフトでは，クイックさと滑らかさとが両立したシフトクォリティが得られるよう，以下のように対応した．

(1) ピストンストローク中の供給油圧の増大

ピストンストローク中の供給油圧を大幅に増大し，締結側クラッチのピストンストローク終了後，速やかにイナーシャフェーズに進行させることにより，トルクフェーズの時間を短縮した．

また，本変更により，イナーシャフェーズ開始時の締結側の容量が過多になることによる突き上げショックが発生しやすくなるが，学習制御の導入より，この問題を解決した．

(2) 棚圧（イナーシャフェーズ進行油圧）の適正化

油圧や摩擦材のμ，エンジントルクなどがばらついても，常時適切な変速時間が得られるように，変速の状況に応じてイナーシャフェーズの進行油圧を決定できる制御を織り込んだ．

また，イナーシャフェーズ終了時の出力軸トルクの段差を小さくするため，フィードバック制御を用いてイナーシャフェーズ終盤における変速の進行度合いを適切に制御するようにした．

(3) クラッチフェーシングのμ-v特性のフラット化

クラッチの滑り速度vの増加とともに摩擦係数μが減少する摩擦特性の場合，振動方程式においてネガティブダンピングを与えてシャダーの発生が懸念されるだけでなく，トルクフェーズの引き時間や変速終了時のトルク段差が増加してしまう．従って，これらを防止するため，μ-v特性のフラットな摩擦材を使用した．

4.2. Smooth upshifts

The following measures were taken to obtain upshift quality in the automatic mode that is characterized by both quickness and smoothness.

(1) Increase in hydraulic pressure supplied during piston stroke

The hydraulic pressure supplied while a clutch piston is stroking has been substantially increased to promote a quick progression to the inertia phase following the completion of the piston stroke of the engaging clutch. This works to shorten the duration of the torque phase.

This increase in hydraulic pressure could result in excessive capacity on the engaging side at the onset of the inertia phase, thereby inducing an end bump. However, that problem was resolved by adopting adaptive learning control.

(2) Optimization of hydraulic pressure during the inertia phase

A control procedure was adopted that enables the progression of the hydraulic pressure in the inertia phase to be determined according to the shift status. As a result, a suitable shift time can always be obtained even if the hydraulic pressure, μ of the friction materials, engine torque or other parameters vary.

In addition, feedback control is used to suitably control the degree of shift progress in the end stage of the inertia phase so as to minimize any discontinuities in the output shaft torque at the conclusion of the inertia phase.

(3) Use of clutch facings with a flat μ-V characteristic

In the case of a friction characteristic where the friction coefficient μ decreases with increasing sliding speed V of the clutches, there is concern that the application of negative damping in the vibration equation could give rise to shudder. In addition to that, the draw-down time of the torque phase and the torque discontinuity at the completion of a shift might increase. Accordingly, friction materials with a flat μ-V characteristic are used to prevent such undesirable effects.

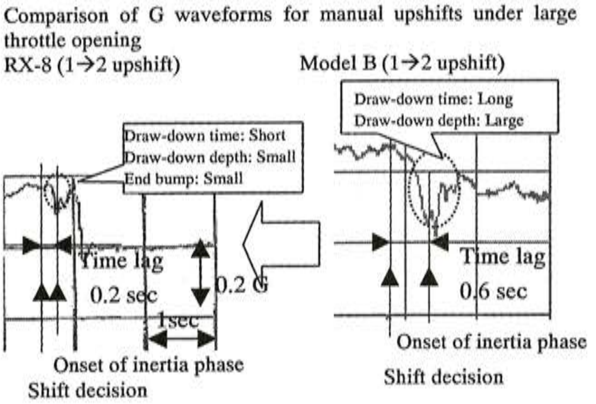


Fig. 4 Manual upshift waveforms

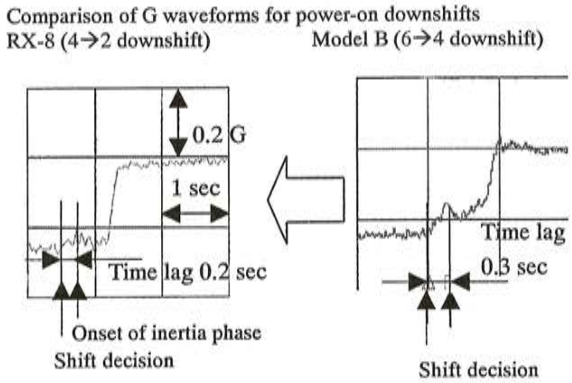


Fig. 5 D range downshift waveforms

これらの対応により，以下のように極めてクイックかつ滑らかなシフトクォリティが得られた．

- トルクフェーズが短く，引き感を殆ど感じない(Fig. 6中のA)
- イナーシャフェーズ開始時の突き上げショックがない(Fig. 6中のB)
- トルクフェーズ開始時とイナーシャフェーズ開始時の出力軸トルク段差がほとんどない(Fig. 6中のC)
- イナーシャフェーズ終了時と変速終了後の出力軸トルク段差が殆どない(Fig. 6中のD)

The adoption of the foregoing measures has resulted in exceptionally quick and smooth shift quality with the following characteristics.

- A short torque phase with virtually no feeling of draw-down (A in Fig. 6)
- No end bump at the onset of the torque phase (B in Fig. 6)
- Virtually no discontinuities in the output shaft torque at the onset of the torque phase and at the onset of the inertia phase (C in Fig. 6)
- Virtually no discontinuities in the output shaft torque at the completion of the inertia phase and at the completion of shifts (D in Fig. 6)

G waveform for automatic 1→2 upshift with large throttle opening

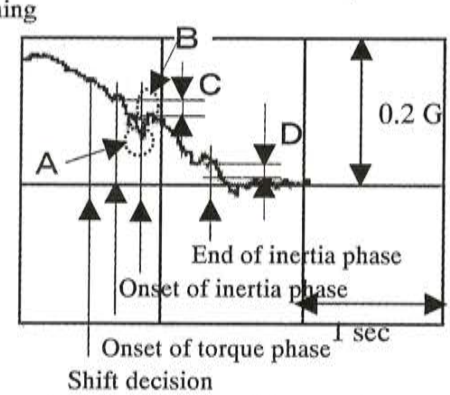


Fig. 6 D range upshift waveforms

4.3. アクティブシフトへの対応

アクティブシフトとは，スポーツ走行時などにより強力な駆動力が得られるように，低速ギヤを積極的に選択するシフトモードであり，従来のオートパワーモードに比べ，変速時のトルクや車速の領域が大幅に拡大している．従って，変速時のクラッチ供給油圧を幅広い領域できめ細かく設定した．これによって，あらゆる変速条件で良好なシフトクォリティを実現することができた．

4.3. Active shifting capability

Active shifting refers to a shift mode that actively selects a lower speed gear so that more powerful driving force can be obtained in sporty driving, for example. Compared with the previous Auto Power Mode, the torque level and vehicle speed range when shifting are substantially increased. Accordingly, the hydraulic pressure supplied to the clutches when shifting has been fine-tuned over a wide range. This makes it possible to achieve excellent shift quality under all sorts of shift conditions.

4.4. シフトクォリティ全般の改善

幅広い各種の変速条件にきめ細かく対応できる制御ロジックの開発とデータ構造の見直しを行い，全般にわたって良好なシフトクォリティを実現することができた．

4.4. Improvement of overall shift quality

Overall shift quality has been improved by developing control logic capable of providing fine-tuned control in response to the wide range of various types of shift conditions and by revising the control data structure.

5. 環境保護対応

排気ガス規制をクリアするため，以下の適合を実施した．

(1) ロックアップ車速の低速化

(2) コーストロックアップの採用

5. Environmental Protection Measures

The following measures were implemented to comply with exhaust gas regulations.

(1) Reduction of the vehicle speed for the onset of lock-up operation

(2) Adoption of coasting lock-up control

6. 診断システム

CAN(Controller Area Network)を通じてATの診断機能をもつ，WDS(World Wide Diagnostic System)テスタとの通信を可能とした.

7. あとがき

新しいコンセプトのスポーツカーにATを適用するため，マツダ様のマネージメントの方々にご協力いただきチーム活動で取り組むことで本ATは，さまざまな狙いを達成できた.

是非　試乗してスポーツカーにふさわしいクイックな変速レスポンスと滑らかなシフトクオリティの両立を実感していただきたい.

最後にマツダ様をはじめとする社内外関係者の方々には，紙面を借りて厚くお礼申し上げます.

6. Diagnostic System

Communication with a Worldwide Diagnostic System (WDS) tester, which incorporates a function for AT diagnosis, is now possible via a Controller Area Network (CAN).

7. Conclusion

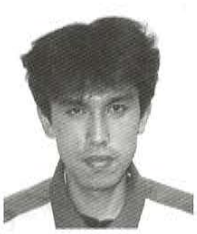
Various objectives set for the JR405E (RZC) AT were attained through the team activities that were undertaken, thanks to the cooperation of members of Mazda's management, to adapt this transmission to their new sports car concept.

The authors hope that readers will definitely test-drive the RX-8 and experience first-hand the quick shift response and smooth shift quality of this AT, characteristics befitting a sports car.

Finally, the authors would like to take this opportunity to thank various related persons at Mazda and in the departments concerned at JATCO for their cooperation in connection with the development of this automatic transmission.

■ Authors ■

Yuzuru SAITOU



Yuuji SAITOU

新商品紹介

AWD 5速オートマチックトランスミッション(MBO)の紹介

Introducing the AWD 5-speed AT (MBO)

渡辺　康敏 *
Yasutoshi WATANABE

抄　録　富士重工業(株)の新型レガシィ(Fig. 1)に搭載した新開発AWD(ALL WHEEL DRIVE)5速ATは，レガシィの開発テーマである『お客様に感動していただける性能』の達成の一翼を担って開発された．

この5速ATは，軽量，小型化により車載性に優れ，高効率化により燃費を向上させるとともに，ダイレクトな油圧制御方式により，「走り」の質を高めた．

本稿では新型 AWD 5速AT(以下MBOと称す)の開発の狙い，基本構成，特長について述べる．

Summary　The all-wheel-drive (AWD) 5-speed AT featured on Fuji Heavy Industries' new Legacy (Fig. 1) was developed to play a key role in accomplishing the car's development theme, which was "to provide performance that impresses customers." This 5-speed AT offers excellent vehicle mountability, thanks to its light weight and compact size, and delivers high efficiency for improved fuel economy. In addition, its direct pressure control system also enhances the quality of the car's driving performance. This article describes the development objective, basic construction and features of the new AWD 5-speed AT (MBO).

1.　はじめに

富士重工業(株)向けには，現在AWD 4速AT用部品のRAMがあるが，今回新型レガシィ用に新たなAWD 5速ATを開発することとした．

今回開発したMBOは，FR 5速AT(JR507E型)をベースとしてAWD対応した新型5ATであり，FRのベース部分をジヤトコが開発した．軽量，小型で高効率なJR507E型との部品共用率を高めることにより，高信頼性と低コストとを両立させた．

なお，富士重工業(株)には，トルクコンバータ，プラネタリギア，クラッチ，コントロールバルブ，ATコントロールユニットを提供している．

1. Introduction

JATCO has been supplying Fuji Heavy Industries, Ltd. with the RAM for the current AWD 4-speed AT. It was decided to develop a new AWD 5-speed AT for use on the latest generation of the Subaru Legacy.

The newly developed MBO is a 5-speed AT with AWD capability and is based on the JR507E 5-speed AT for application to rear-wheel-drive cars. JATCO was responsible for developing the basic parts for the rear-wheel-drive layout. Increasing the percentage of parts shared in common with the small, compact and high-efficiency JR507E AT made it possible to achieve high reliability combined with low cost.

The parts supplied to Fuji Heavy Industries include the torque converter, planetary gears, clutches, pressure control valve and the AT control unit.

Fig. 1 New Legacy

* 適用開発部
Application Development Department

2. 開発の狙い

JR507E型をベースに，AWD ATとして最適化し，『走り』と『燃費』の両立を高次元でバランスさせることを狙いとした．具体的には，以下の要素部品について，ベース5ATの特長を活かしながら，新たな改良を加えて性能，機能を向上させた．

① トルクコンバータ
② プラネタリギア・クラッチ
③ コントロールバルブ(油圧制御)
④ ATコントロールユニット(電子制御)

3. 構造と特徴

AWD 5ATの主断面図をFig. 2に，主要諸元をTable 1に示す．

以下に開発の狙いで紹介した項目について，「従来5ATの踏襲部」と「改良部」とに分けて、要素ごとに概説する．

2. Development Objective

The principal development objective set for the MBO was to achieve the highest possible balance of driving performance and fuel economy, using the JR507E as a base and creating an optimized AWD AT. Specifically, improvements were newly added to the following key parts to enhance performance and functionality while making the most of the distinctive features of the base 5-speed AT.

(1) Torque converter
(2) Planetary gears and clutches
(3) Pressure control valve
(4) AT electronic control unit

3. Construction and Features

A main cross-sectional view of the AWD 5-speed AT is shown in Fig. 2, and its major specifications are given in Table 1.

A general explanation is given here for each of the key elements noted in the preceding section, with the discussion divided between the "parts inherited from the base 5-speed AT" and "improvements."

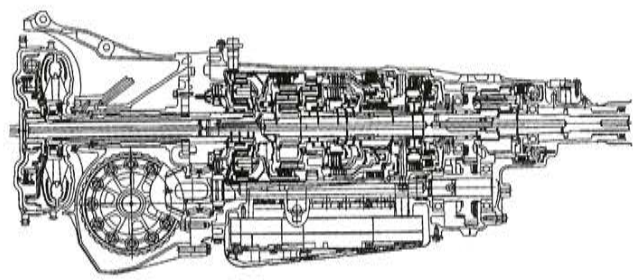


Fig. 2 Main cross-sectional view

Table 1 Specifications

<table>
<tr><td colspan="3">Drive system</td><td>AWD</td></tr>
<tr><td colspan="3">Applied engine</td><td>EJ20
(191 kW/6000,
350 Nm/2400)</td></tr>
<tr><td colspan="3">Transmission model</td><td>TG5C</td></tr>
<tr><td colspan="3">Torque converter</td><td>Torque converter with wet multi-plate lock-up clutch</td></tr>
<tr><td rowspan="7">Gear ratios</td><td rowspan="6">Main ratios</td><td>1st</td><td>3.540</td></tr>
<tr><td>2nd</td><td>2.264</td></tr>
<tr><td>3rd</td><td>1.471</td></tr>
<tr><td>4th</td><td>1.000</td></tr>
<tr><td>5th</td><td>0.834</td></tr>
<tr><td>Rev</td><td>2.370</td></tr>
<tr><td colspan="2">Final gear ratio</td><td>3.272 (GT),
3.583 (GT spec B)</td></tr>
<tr><td colspan="3">Transfer case</td><td>Compound planetary gearset with center differential (VTD)</td></tr>
<tr><td rowspan="2">Oil used</td><td colspan="2">Fr diff</td><td>Subaru Gear Oil
(FFX 75W-90)</td></tr>
<tr><td colspan="2">ATF</td><td>ATF RED-1</td></tr>
</table>

3.1. トルクコンバータ

【従来5ATの踏襲部】

ロックアップクラッチ部に多板式クラッチ，3way式油圧制御回路，独立制御可能油圧室などを踏襲採用し，スリップロックアップ領域を広範囲で可能とした．

【改良部】

前輪デフ内蔵のAWD用AT構造とするため，流体要素を超々扁平化するとともに，クラッチのダンパスプリングのレイアウトを最適化し，トルクコンバータと主変速機構との間にデフを配置した．(Fig. 3)

3.2. プラネタリギアレイアウト

【従来5ATの踏襲部】

基本となるギアレイアウトを踏襲することで，プラネタリギア，クラッチ類の各要素を最適配置でき，『小型軽量化』と『走りの良さ』を実現した．

【改良部】

小径化のために，バンドブレーキを多板ディスク化し，高回転対応のために，滑り軸受けを転がり軸受化した．また，軸受の潤滑性を向上するため潤滑構造を変更し，AWD対応のため潤滑回路などを変更した．(Fig. 4)

3.1 Torque converter

[Parts inherited from the base 5-speed AT]

The lock-up clutch assembly continues the multi-plate clutch, 3-way pressure control circuit and independently controllable pressure chambers, among other parts, making it possible to expand the region of slip lock-up operation.

[Improvements]

In order to construct the AWD AT with a built-in front differential, the hydrodynamic elements were made super ultraflat and the layout of the clutch damper spring was optimized. The differential was positioned between the torque converter and the principal shift mechanism (Fig. 3).

3.2 Planetary gear layout

[Parts inherited from the base 5-speed AT]

The continuation of the gear layout, which is the basis of the design, allowed the various elements such as the planetary gears and clutches to be optimally positioned, thereby achieving a compact, lightweight unit that provides excellent driving performance.

[Improvements]

Multi-plate discs were adopted as the band brakes to reduce the transmission's diameter, and rolling bearings were adopted in place of sliding bearings in order to cope with higher rotational speeds. The structure of the bearing lubrication system was also changed to improve lubrication performance, and the lubrication circuit and other parts were changed to make them compatible with AWD (Fig. 4).

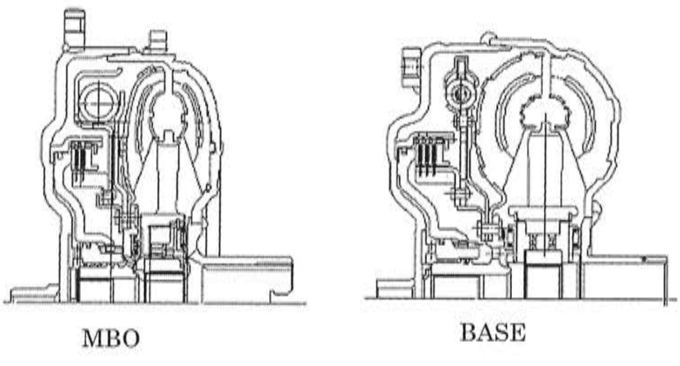


Fig. 3 Torque converter

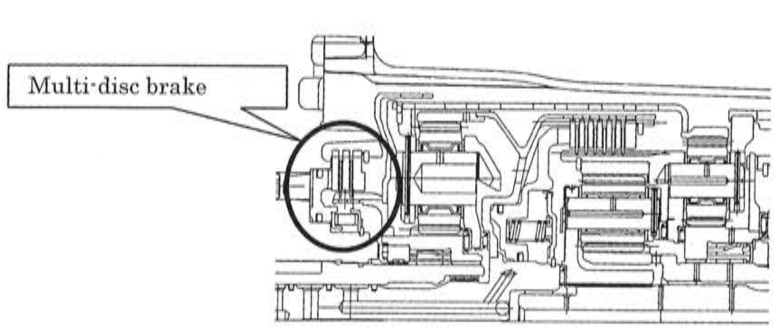


Fig. 4 Schematic of planetary gearset

3.3. コントロールバルブ構造

【従来5ATの踏襲部】

各クラッチ圧を直接クラッチ毎に制御可能なダイレクトクラッチ圧制御方式を踏襲し，入力トルクの変化にも瞬時かつ高精度で対応できる仕様とした．

また，低温性能に優れた2方向型リニアソレノイドを採用するとともに，ソレノイドの個体ばらつきを補正して安定した油圧を確保するため，ソレノイドの油圧特性をコントロールバルブ上面に配置したメモリーBOXに書き込み，車載時にATコントロールユニットにデータを転送するシステムを採用した．

【改良部】

従来4ATでは別配置としていたトランスファの制御バルブとソレノイドとを，コントロールバルブ内に配置し，車載性を改良した．

3.4. ATコントロールユニット

【従来5ATの踏襲部】

変速品質を確保するため，基本制御については踏襲した．

【改良部】

AWD制御部を合体させるとともに，性能向上のために，以下の新制御を追加した．

3.4.1. エンジンとの総合制御

変速時のエンジントルク制御に関しては，変速ごと，トルク別などの制御条件を細密にし，電制スロットルによる駆動トルク制御を駆使することで，変速前後の駆動力とイナーシャフェーズの駆動力との段差が少ないスムーズな変速を実現した．

3.4.2. 出荷時学習

AT組立完成後の性能検査(ファイナルテスト)時の検査データをもとに，本来，車両実走行で進行するであろう学習値を先行してコントロールバルブのメモリBOXに書き込む出荷時学習を採用した．

これにより，車載初期から安定した高い変速品質を実現し，実走行による学習を短時間で済ませるようにした．

3.4.3. 新アダプティブ制御

アダプティブ制御を，スポーティー走行への対応に特化した制御とするため，制御の作動領域はスポーツモードに限定した．新アダプティブ制御の内容を次に示す．

3.3 Control valve construction

[Parts inherited from the base 5-speed AT]

The direct clutch pressure control system was continued to allow each clutch pressure to be controlled directly. The control valve specifications allow instantaneous and highly accurate response to changes in input torque.

Two-way linear solenoids with excellent low-temperature characteristics were adopted along with a system for ensuring stable hydraulic pressure by compensating for individual solenoid variability. This is accomplished by inputting solenoid pressure characteristics into a memory device positioned on top of the control valve and sending the data to the AT control unit during vehicle operation.

[Improvements]

The control valve, solenoid and other parts of the transfer case were incorporated in the control valve to improve vehicle mountability, instead of positioning them separately as in the case of the previous AWD 4-speed AT.

3.4 AT control unit

[Parts inherited from the base 5-speed AT]

The basic control system was continued to ensure good shift quality.

[Improvements]

Along with incorporating the AWD control unit in the system, the following new controls were added to improve performance.

3.4.1 Integrated control with the engine

Detailed conditions for controlling engine torque during shifting were separately specified for every type of shift and torque level. Drive torque control by the electronically controlled throttle is effectively utilized to achieve smooth shifting without any discontinuities between the drive torque before and after a shift and the drive torque in the inertia phase.

3.4.2 Learning control at time of shipment

A learning control system was adopted that enters learned values in the memory device of the control valve at the time of shipment in anticipation of the values that will be acquired in the course of real-world driving. The values are based on the inspection data obtained in the final performance tests performed on each AT following assembly.

This system achieves stable, high-quality shifting from the initial period of vehicle use and enables the learning process based on actual driving to be completed in a short period of time.

①ブレーキングダウン制御：急なブレーキング時，ドライバの減速意図を察知して，自動的にダウンシフトさせエンジンブレーキを強化する．
②コーナーリング時ホールド制御：コーナーリング時に不用意なアップシフトやダウンシフトを避けるため，ギヤ段をホールドする．
③アクセルペダル急戻し時ホールド制御：急なアクセルペダル戻し時，ドライバの減速意図を察知して，シフトアップさせない制御．

また上記以外にも，ドライバの運転状況，道路状況(上り坂，峠など)を常にモニタし，必要に応じて変速線を自動的に切換える自動変速線切換え制御を，全ての自動変速領域において実施している．

3.4.4. 新開発ステアリングシフト対応制御

従来のマニュアルシフト制御に加えて，セレクトレバーがDレンジ位置のままでも，ステアリングに装備されているUP/DOWNスイッチの操作により，瞬時にマニュアルシフトモードへの切換えを可能とする新ステアリングシフト対応制御を採用した．

同制御では，マニュアルシフトモードへの切り替わり後，規定時間内に再操作がない場合や車両が加速状態になると，通常のDレンジ制御に自動復帰するもので，より手軽なマニュアルシフト操作を可能とした．(Fig. 5)

Fig. 5 Photo of steering wheel switches

4. まとめ

新型レガシィの商品力向上に寄与するべく，新型AWD 5ATを開発し，それを満足する仕様を完成させることができた．

5. 謝辞

本MBOは短期開発であったが，関係部署の方々のご協力により，無事開発できた．

多大なご協力を頂いた富士重工業(株)様，その他の社内外の関係部署各位に深く御礼申し上げる．

3.4.3 New adaptive control

The operating region of adaptive control was limited to the sports mode in order to make the control procedure specifically match sporty driving. The details of this new adaptive control are explained below.

(1) Braking downshift control: The transmission infers the driver's intention to decelerate from sudden braking action and automatically downshifts to provide stronger engine braking.

(2) Gear-holding control during cornering: The gear range is held during cornering to avoid unexpected upshifts or downshifts.

(3) Gear-holding control at sudden release of accelerator pedal: This control infers the driver's intention to decelerate from a sudden release of the accelerator pedal and prevents the transmission from shifting up.

In addition to the control features mentioned here, a control is also provided that constantly monitors the driver's driving actions and the road conditions (uphill road, mountain pass, etc.) and automatically changes the shift lines if necessary. This shift line control is provided in every automatic shifting region.

3.4.4 Newly developed steering wheel shift control

In addition to the conventional manual shift mode, a new steering wheel shift control was adopted for instantaneously switching to the manual shift mode even when the selector lever is in the Drive position. This is done by operating the Up/Down switches provided on the steering wheel.

Following a switch to the manual shift mode, the transmission automatically reverts to the ordinary Drive range control again if there is no further operation of the switches within a specified period of time or if the vehicle begins to accelerate. This control thus facilitates easier manual shifting (Fig. 5).

4. Conclusion

The new AWD 5-speed AT was developed with the aim of helping to enhance the product appeal of the new Legacy, and specifications for satisfying that objective were embodied in the unit.

5. Acknowledgments

The MBO was successfully developed in a short period of time, thanks to the fine cooperation of all the departments involved.

The author would like to thank various individuals at Fuji Heavy Industries, Ltd. and also in the departments concerned both inside and outside the company for their tremendous cooperation in connection with the development of the MBO.

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■ Author ■

Yasutoshi WATANABE

商品紹介

FF車用4速AT JF404E-W (FDO) の紹介

Introducing the JF404E-W (FDO) 4-speed AT for Front-drive Cars

JF404E-W (FDO) 型自動変速機は，従来２個使っていたシンプソンタイプのプラネタリギアをラビニョウタイプにすることで1個とし，ワンウェイクラッチも2個から1個にすることで，軽量，コンパクトを実現した，小型の前輪駆動乗用車のための4速自動変速機です．95年にVolkswagen様の Poloに初めて採用され，2000年12月にはSKODA 様のFABIAにも拡大採用されています．
更に2002年1月にはフルモデルチェンジしたVolkswagen様のPoloにも継続採用されました．

The JF404E0W (FDO) AT features a lightweight, compact design as a result of adopting one Rabinow planetary gearset instead of the two Simpson planetary gearsets used previously and also reducing the number of one-way clutches from two to one. This 4-speed AT is designed for use on front-drive compact cars. It was first adopted on the Volkswagen Polo in 1995 and its application was expanded to the Skoda Fabia in December 2000. Moreover, it has continued to be used on the new generation of the Volkswagen Polo that was released in January 2002.

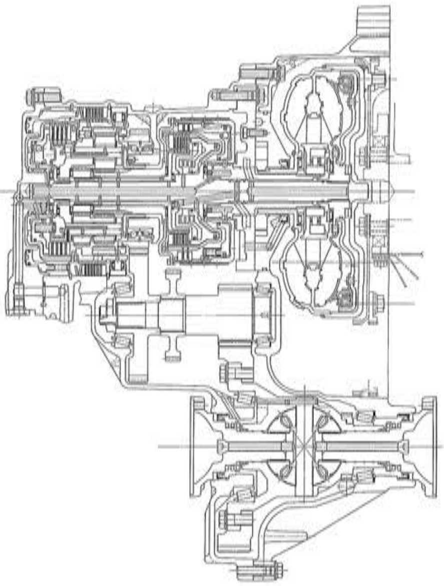

Fig. 1 Main cross-sectional view

Table 1 Specifications

Max. input torque		150 Nm
Max. input speed		7,000 rpm
Max. vehicle weight (GVW)		1,550 kg
Control system		Electronic
Torque converter		215 mm dia.
Gear ratios	1st	2.875
	2nd	1.512
	3rd	1.000
	4th	0.726
	Rev.	2.656
Final drive gear ratio		3.47～4.38
No. of selector positions		7(P, R, N, D, 3, 2, 1)
Overall length		360.8 mm
Center distance between engine and differential		174 mm
Dry weight		60 kg

■ Typical model fitted with the JF404E-W (FDO) AT ■

Polo

FF車用4速AT　JF405E-H(FRO)の紹介

Introducing the JF405E-H (FRO) 4-speed AT for Front-drive Cars

'98年10月にスズキ(株)様のワゴンR RRに初めて搭載されたJF405E-H(FRO)型自動変速機は，世界トップレベルの小型・軽量で，小型乗用車にジャストフィットしたFF自動変速機です．'03年10月からのワゴンRにも引き続き採用されています．

The front-drive JF405E-H (FRO) 4-speed AT was first adopted on the Suzuki Wagon R RR in October 1998. As one of the world's smallest and lightest units, this AT is just the right size for small cars. It has continued to be used on the Wagon R since October 2003.

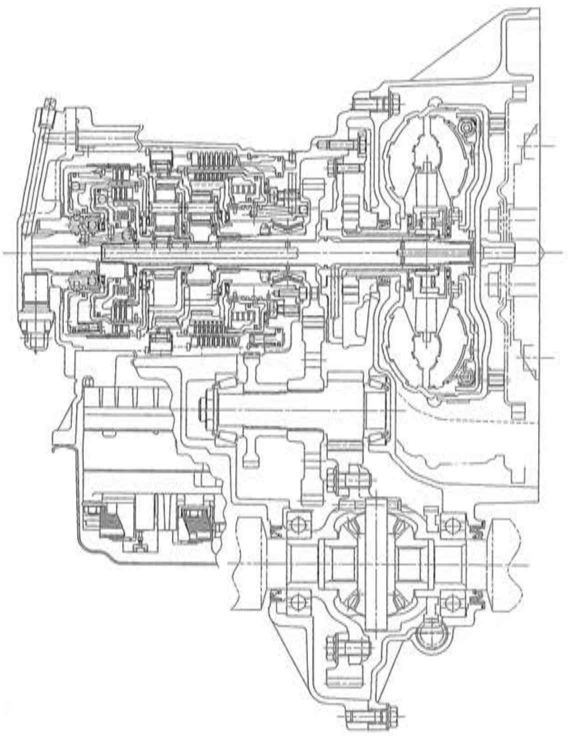

Fig. 1 Main cross-sectional view

Table 1 Specifications of JF405E-H

Max. input torque		110 Nm
Max. input speed		7,800 rpm
Max. vehicle weight (GVW)		1,235 kg
Control system		Electronic
Torque converter		186 mm dia.
Gear ratios	1st	2.914
	2nd	1.525
	3rd	1.000
	4th	0.725
	Rev.	2.642
Final drive gear ratio		4.017～5.804
No. of selector positions		6 (P,R,N,D,2,L)
Overall length		359.9 mm
Center distance between engine and differential		172 mm
Dry weight		45.7 kg

■ Typical model fitted with the JF405E-H (FRO) AT ■

WAGON R RR

FF車用4速AT　JF405E-G(FRB)の紹介

Introducing the JF405E-G (FRB) 4-speed AT for Front-drive Cars

'02年 7月に，韓国のGM大宇オート&テクノロジー社のMatizに搭載されたJF405E-G(FRB)型自動変速機は，JF405E-H型をベースに，解析技術、適用開発技術を活用し，ギアノイズの改善，変速性能を向上させた，FF車用4速ATです。

Based on the JF405E-H automatic, the front-drive JF405E-G (FRB) 4-speed AT was adopted on the Matiz, produced by South Korea's GM Daewoo & Technology Company, in July 2002. Analysis techniques and applied engineering technologies were utilized to reduce the gear noise and improve the shift performance of the JF405E-G AT.

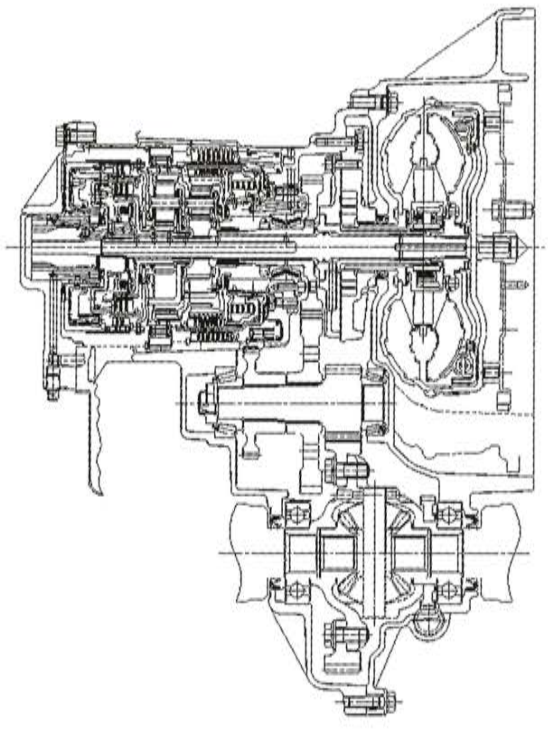

Fig. 1 Main cross-sectional view

Table 1 Specifications of JF405E-H

Max. input torque		71.5 Nm
Max. input speed		6,500 rpm
Max. vehicle weight (GVW)		1,217 kg
Control system		Electronic
Torque converter		186 mm dia.
Gear ratios	1st	2.914
	2nd	1.525
	3rd	1.000
	4th	0.725
	Rev.	2.642
Final drive gear ratio		4.709
No. of selector positions		6(P, R, N, D, 2, L)
Overall length		359.9 mm
Center distance between engine and differential		172 mm
Dry weight		46.0 kg

■ Typical models fitted with the JF405E-G (FRB) AT ■

Matiz

FF車用4速AT　JF405E-Q(FRD)の紹介

Introducing the JF405E-Q (FRD) 4-speed AT for Front-drive Cars

'04年1月に，韓国の起亜自動車様MORNING/PICANTOの1.0L&1.1L　NA車両に搭載されたJF405E-Q(FRD)型自動変速機は，韓国軽自動車初の4速自動変速機 FRAをベースに，電子制御を高度化し，シフトクオリティーを向上させたFF車用4速ATです．韓国市場だけでなく 欧州の各地域にも輸出されています．

The front-drive JF405E-Q (FRD) 4-speed AT was adopted on the Kia Morning/Picanto model, fitted with a 1.0- or 1.1-liter naturally aspirated engine, in January 2004. This transmission is based on the FRA 4-speed AT that was the first 4-speed automatic to be used on Korean minicars. The electronic control system and shift quality of the JF405E-Q have been further enhanced. In addition to the South Korean market, this AT is also exported to various markets in Europe.

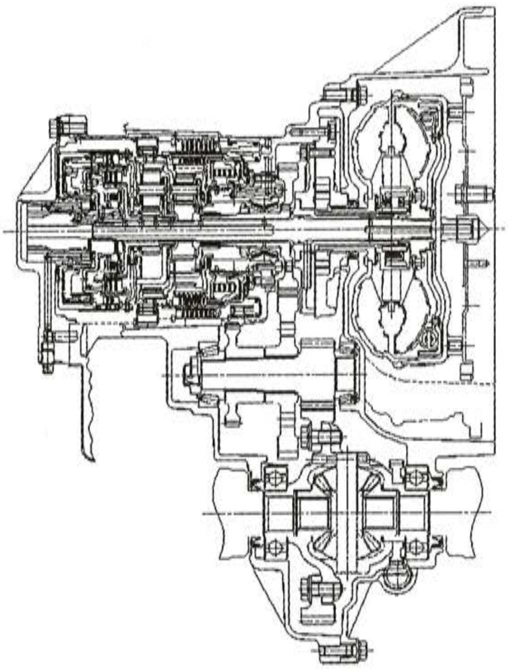

Fig. 1 Main cross-sectional view

Table 1 Specifications

Max. input torque		96 Nm
Max. input speed		6,000 rpm
Max. vehicle weight (GVW)		1,350 kg
Control system		Electronic
Torque converter		186 mm dia.
Gear ratios	1st	2.914
	2nd	1.525
	3rd	1.000
	4th	0.725
	Rev.	2.642
Final drive gear ratio		3.977
No. of selector positions		6 (P, R, N, D, 2, L)
Overall length		359.9 mm (Excluding mounting bracket)
Center distance between engine and differential		172 mm
Dry weight		49.5kg

Typical model fitted with the JF405E-Q (FRD) AT

FF車用5速AT JF506E-R (FPO)の紹介

Introducing the JF506E-R (FPO) 5-speed AT for Front-drive Cars

JF506E-R型自動変速機は，'99年7月にRover MG Group様のRover75とLand Rover Group様のFreelanderに初めて搭載されました．'01年7月にはRover MG Group様のRover75 Tourerに拡大採用されました．

The JF506E-R 5-speed AT was first adopted on Rover MG Group's Rover 75 in July 1999 and on Land Rover Group's Freelander in August 2000. Its application was further expanded to Rover MG Group's Rover 75 Tourer in July 2001.

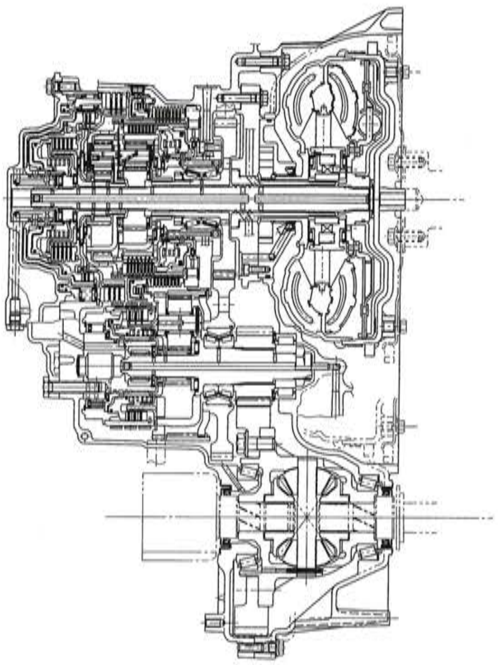

Fig.1 Main cross - sectional view

Table 1 Specifications

Max. input torque		310 Nm	
Max. input speed		7,000 rpm	
Max. vehicle weight (GVW)		1,900 kg	
Control system		Electronic	
Torque converter		250 mm dia.	
Gear ratios	1st	3.474	3.801
	2nd	1.948	2.131
	3rd	1.247	1.364
	4th	0.854	0.935
	5th	0.685	0.685
	Rev.	2.714	2.970
Final drive gear ratio		2.9～4.1	
No. of select positions		7 (P, R, N, D, 4, 3, 2)	
Dry weight		95 kg	

■ Typical models fitted with the JF506E-R (FPO) ■

Rover 75 Tourer

Freelander

商品紹介

FF車用5速AT　JF506E-L (FPD)の紹介

Introducing the JF506E-L (FPD) 5-speed AT for Front-drive Cars

JF506E-L型自動変速機は，'01年1月にJaguar様のX-Typeに搭載されました．X-TypeはJaguar様初のコンパクト・プレミアムカーで，Jaguar様の伝統的なドライビングを継承するためトラクション4と呼ばれる全輪駆動システムを備えています．JF506E型自動変速機として初のガソリン3.0L車への適用となり，高トルク，高回転での耐久性を確保するため，各部位の強化を実施し，ラグジュアリーカーにふさわしい変速性能を得るため各種の新制御を導入しています．さらに環境保護，運転性に対する客先ニーズに応える改良を行いました．

The JF506E-L 5-speed AT was adopted on the Jaguar X-Type in January 2001. Representing Jaguar's first compact premium car, the X-Type is fitted with the Traction 4 all-wheel-drive system to continue the Jaguar tradition of driving performance. This was the first application of the JF506E AT to a car powered by a 3.0L gasoline engine. Various parts of the transmission were reinforced to secure sufficient durability for handling high torque and rotational speed levels, and new controls were adopted to provide shift performance befitting a luxury car. Improvements were also made to meet the customer's needs with respect to environmental protection and driveability.

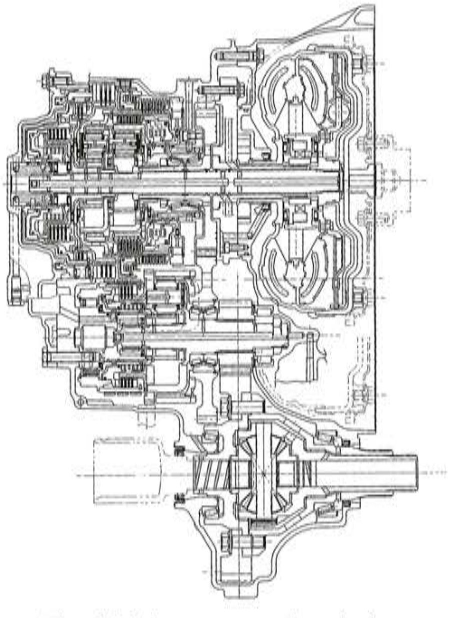

Fig. 1 Main cross-sectional view

■ Typical model fitted with the JF506E-L (FPD) AT ■

Jaguar X-Type

Table 1 Specifications

Vehicle	Model		X-type 2.1L	X-type 2.5L	X-type 3.0L
	Drive type		Front-wheel drive	All-wheel drive	All-wheel drive
	Weight (kg)		1,489	1,620	←
	Engine type		V6 DOHC	←	←
	Max. power (DIN)		117 kw (159 ps)/6,750 rpm	145 kw (198 ps)/6,800 rpm	172 kw (234 ps)/6,800 rpm
	Max. torque (DIN)		198 Nm/3,750 rpm	244 Nm/4,000 rpm	284 Nm/3,000 rpm
AT	Totque converter		236 mm dia.with lock-up	250 mm dia.with lock-up	←
	Gear ratios	1st	3.802	←	←
		2nd	2.132	←	←
		3rd	1.365	←	←
		4th	0.935	←	←
		5th	0.685	←	←
		Rev	2.97	←	←
	FDR		4.153	3.898	←
	Center distance(mm)		205.9	←	←
	Dry weight (kg)		95	←	←

商品紹介

FF車用5速AT　JF506E-V (FPH)の紹介

Introducing the JF506E-V (FPH) 5-speed AT for Front-drive Cars

JF506E-V型自動変速機は，'02年6月にFord モンデオのマイナーチェンジから搭載されました．V6 2.5Lガソリンと I4 2.0TDCiディーゼルの2機種への適応で，ディーゼル向けにはコーストでの応答性向上として，FF車用では当社初となるコーストスリップロックアップを採用しています．

The JF506E-V AT was adopted on the Ford Mondeo when the car underwent a minor model change in June 2002. The transmission is mated to a 2.5-liter gasoline V6 and two types of 2.0-liter inline 4-cylinder TDCi diesel engine. To improve response under a coasting condition on the front-wheel-drive models fitted with a diesel engine, the transmission adopts JATCO's first-ever coasting slip lock-up control.

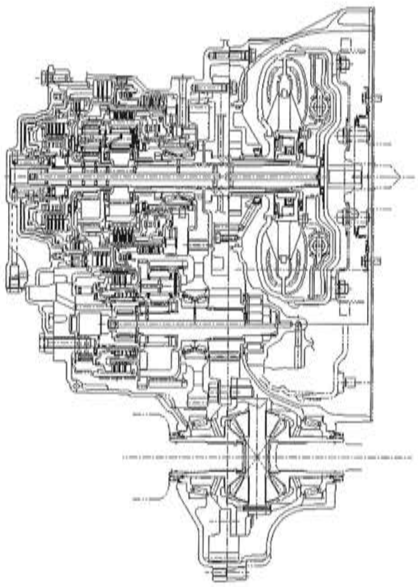

Fig. 1 Main cross-sectional view

■ Typical model fitted with the JF506E-V (FPH) AT ■

Ford Mondeo

Table 1 Specifications

	Model		Mondeo 2.5L V6	Mondeo 2.0L	Mondeo 2.0L
Vehicle	Drive system		2WD	←	←
	Weight (kg)		2,190	2,230	←
	Engine type		**Duratec 2.5L**	**Puma Diesel 2.0L 115 PS**	**Puma Diesel 2.0L 130 PS**
	Max. power (DIN)		170 PS / 6,100	**115 PS** / 4,000	**130 PS** / 3,800
	Max. torque (DIN)		222 Nm / 2,746	280 Nm / 1,900	310 Nm / 1,800 - 2,100
AT	Torque converter		EFJ	AAD	
	Gear ratios	1st	3.801	←	
		2nd	2.131	←	
		3rd	1.364	←	
		4th	0.935	←	
		5th	0.685	←	
		Rev	2.970	←	
	FDR		3.712	3.491	
	Center distance (mm)		200	←	
	Dry weight (kg)		96	95	

FR車用4速AT JR405E-K(RZB)の紹介

Introducing the JR405E-K (RZB) 4-speed AT for Rear-drive Cars

JR405E-K(RZB)型自動変速機が，いすゞ様のタイ向けピックアップトラックD-MAXに採用されましたので紹介します．D-MAXは日本国内では販売はされておりませんが，タイではシェアNo.1であり，量産開始以来順調に生産を重ねています．

Introduced here is the JR405E-K (RZB) 4-speed AT that has been adopted on the D-Max pickup truck marketed in Thailand by Isuzu Motors Ltd. Although the D-Max is not sold in Japan, it ranks No. 1 in market share in Thailand. The production volume has grown steadily ever since the model went into mass production.

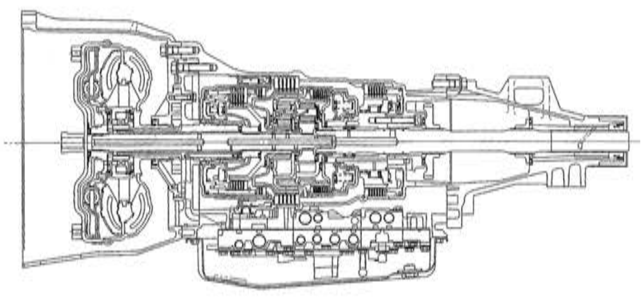

Fig. 1 Main cross-sectional view

Table 1 Specifications

Max. input torque		294 Nm
Max. input speed		7,000 rpm
Max. vehicle weight (GVW)		3,250 kg
Control system		Electronic
Torque converter		250 mm dia.
Gear ratios	1st	2.785
	2nd	1.545
	3rd	1.000
	4th	0.694
	Rev.	2.272
No. of selector positions		7 (P, R, N, D, 3, 2,1)
Dry weight		63 kg

■ Typical model fitted with the JR405E-K (RZB) ■

D-Max

FR車用5速AT JR509E(MCO)の紹介

Introducing the JR509E (MCO) 5-speed AT for Rear-drive Cars

JR509E型自動変速機は，北米フルサイズピックアップ、SUV用ATとして、JR507E型ATをベースに強化を行い、'03年10月に日産自動車(株)様のパスファインダー アルマーダに採用されました.

Developed around the JR507E AT, the JR509E 5-speed AT was further improved for use on full-size pickup truck and sport utility vehicle (SUV) models in the North American market. It was adopted on the Nissan Pathfinder Armada SUV in October 2003.

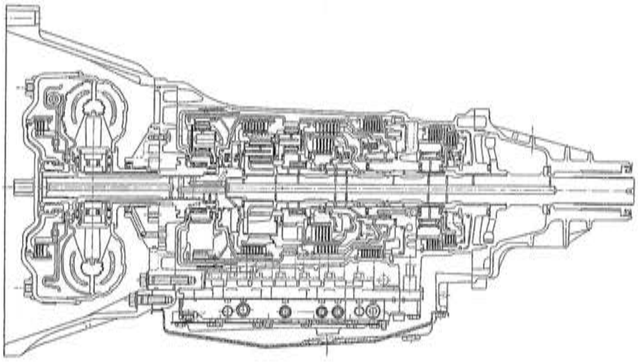

Fig. 1 Main cross-sectional view

Table 1 Specifications

Max. input torque		550 Nm
Max. input speed		6,000 rpm
Max. vehicle weight (GVW)		3,230 kg
Control system		Electronic
Torque converter		260 mm dia.
Gear ratios	1st	3.827
	2nd	2.368
	3rd	1.519
	4th	1.000
	5th	0.834
	Rev.	2.613
Final drive gear ratio		2.937
No. of selector positions		6 (P, R, N, D, 3, 2)
Dry weight		85 kg

■ Typical model fitted with the JR509E (MCO) ■

Pathfinder Armada

FF車用ベルトCVT F1C1(BAO)の紹介

Introducing the Steel-belt F1C1 CVT (BAO) for Front-drive Cars

F1C1型CVTは，'00年に三菱自動車(株)様初のCVTとしてランサーセディアに搭載され，その高いシフトクオリティと低燃費で好評を得ています．今回，各部の改良によりさらなる燃費の向上，軽量化を達成し，コルトにも拡大採用されました．

The F1C1 CVT was adopted on the Mitsubishi Lancer Cedia in 2000 as the first CVT ever to be used on a Mitsubishi car. It has been highly acclaimed for its outstanding shift quality and low fuel consumption. Further improvements have now been made to attain better fuel economy and a lighter weight, and application of this unit has been expanded to the Mitsubishi Colt as well.

Table 1 Specifications

Engine		For 1.3L Engine		For 1.5L Engine	
2WD/4WD		2WD	4WD	2WD	4WD
Max. input torque		125 Nm	←	150Nm	←
Max. input speed		6,000 rpm	←	←	←
Max. vehicle weight (GVW)		1,375 kg	←	←	←
Control system		Electronic	←	←	←
Torque converter		236 mm dia.	←	←	←
Ratios	FWD	2.319~0.445	←	←	←
	REV	2.588	←	←	←
Ratio coverage		5.211	←	←	←
Final gear ratio		5.219	5.686	5.219	5.686
No. of select position		6 (P,R,N,D,Ds,L)	←	←	←
Dry weight		76.8 kg	77.8 kg	77.8 kg	78.8 kg

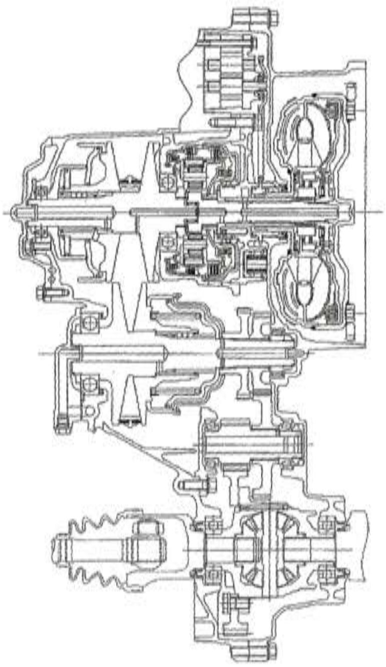

Fig. 1 Main cross-sectional view

■ Model fitted with the F1C1 CVT ■

COLT

事業所紹介

ジヤトコ岡崎開発事業所の紹介

Introducing JATCO's Okazaki Product Development Department

大西　豊二*
Toyoji OHNISHI

1. 概要

所在地：　愛知県 岡崎市 橋目町 字 中新切 1番地
従業員数：　147名(2003年11月1日現在)

2002年4月に三菱自動車株式会社(以下MMC)からトランスミッション部門を分離して設立したダイヤモンドマチック社は，2003年4月にジヤトコと合併した．新たにジヤトコに加わった京都，八木，水島，岡崎各事業所のうち，ここ岡崎開発事業所には，開発部門及び営業部門があり，設計(含む開発管理)67名，実験74名，営業6名の社員が，主にMMC向けユニットの開発，改良，折衝業務に取り組んでいる．(実験の一部は京都事業所内にも在席) (Fig. 1, 2)

設計(含む営業)はMMC岡崎の技術本館5F，実験はMMC岡崎の駆動系試験棟1～4F及びエンジン耐久試験棟の1Fで業務をしている．

MMCの開発本部内に居住していることから，MMCと密接に連携でき，MMCの要請にもタイムリーに対応できるため，効率良い開発作業が可能となり，MMC/ジヤトコ双方にとって大きなメリットとなっている．またテストコースに近いため，すぐに試験車両での走行確認ができ，MMC向けユニット開発としては最高のロケーションにある．

1. Overview

The Okazaki Product Development Department is located at 1, Nakashinkiri, Hashime-cho, Okazaki City, Aichi Prefecture. The workforce numbered 147 employees as of November 1, 2003.

Diamondmatic Co., Ltd., formed in April 2002 when Mitsubishi Motors Corporation (MMC) spun off its transmission division, merged with JATCO in April 2003. As a result, the Kyoto, Yagi, Mizushima and Okazaki manufacturing and product development departments newly joined JATCO's organization. The Okazaki Product Development Department has both product development and sales groups, with 67 employees engaged in design (including R&D administration), 74 employees engaged in testing and six employees engaged in sales activities. They are mainly responsible for developing, improving and conducting negotiations for transmissions intended for delivery to MMC. Some of the testing personnel are also located at the Kyoto Manufacturing Department.

Design personnel (including sales staff) work on the fifth floor of the MMC Car Research & Development Center (Fig. 1). Testing personnel work on the first through the fourth floors of MMC Okazaki Drivetrain Testing Laboratory and the first floor of the Engine Durability Testing Laboratory (Fig. 2).

Being located in the MMC Car Research & Development Center (MMC Okazaki) allows close teamwork with MMC and also timely response to MMC's requests, thereby facilitating efficient development activities, which has large benefits for both JATCO and MMC. Moreover, because of the close proximity to the proving ground, confirmation driving tests can soon be conducted in test vehicles. All of these factors make the location ideal for developing transmissions for MMC.

Fig. 1 Design office

Fig. 2 Test laboratory

* 岡崎開発事業所
Okazaki Product Development Department

2. 組織と業務内容

設計には，CVTを担当する第一商品開発本部，ATを担当する第二商品開発本部と開発管理部，テクニカル原価低減推進室の一部がある．さらに設計フロアの一画には，MMCを担当する第二営業部も席をおいており，MMCの要請に日々対応している．

実験については，設計同様CVTを担当する第一実験部，ATを担当する第二実験部，材料開発を担当する材料開発グループが在席している．

主な担当機種は，従来MMCが保有していた，FF車用AT，FR車用AT，FF車用CVT，軽自動車用FF/FR用ATの全てに渡り，これらに加え富士/厚木地区で開発したユニットのMMC車両適用業務も担当している．

テストコースに近いことから，MMC殿の秘匿試験車の走行確認も比較的容易で，特に実験業務の効率化に役立っている．

3. 環境

ジヤトコ岡崎開発事業所のあるMMC岡崎は，愛知県岡崎市の西端に位置し，豊田市，安城市にも近い半農/半工地域で，隣接地には果樹園などの田園風景が広がる比較的閑静な地域である．(Fig. 3)

2. Organization and Development Activities

The design division comprises Product Development Center No. 1 responsible for CVTs, Product Development Center No. 2 responsible for ATs, the R&D Administration Department and part of the Technical Cost Reduction Promotion Department. One corner of the design floor is also occupied by Sales and Marketing Department No. 2, which is in charge of MMC and routinely handles MMC's requests.

Similarly, testing personnel are assigned to Experiment Department No. 1 responsible for CVTs, Experiment Department No. 2 responsible for ATs and the Materials Development Group in charge of developing materials.

The principal products Okazaki is responsible for include front- and rear-drive ATs, front-drive CVTs and all front- and rear-drive ATs for minicars, which MMC had previously possessed. In addition, it is also responsible for adapting the transmissions developed at the Fuji and Atsugi centers for use on MMC vehicles.

Being near the proving ground makes it relatively easy to conduct confirmation tests using MMC's camouflaged test cars, which is especially helpful in improving testing work efficiency.

3. Surroundings

MCC Okazaki, the site of JATCO's Okazaki Product Development Department, is situated on the western edge of the city of Okazaki in Aichi Prefecture (Fig. 3). It is a semi-agricultural and semi-industrial area that is also close to the cities of Toyoda and Anjou. It is a relatively quiet area with an expansive pastoral landscape that includes nearby orchards.

Fig. 3 Location of Okazaki Product Development Department

Fig. 4 Okazaki Castle

Fig. 5 Mikawa Bay

特に，岡崎市は徳川家康の生誕地でもあることから，岡崎城(Fig.4)を始めとする家康ゆかりの社寺・史跡は至る所にあり，三河花火や八丁味噌など伝統を誇る郷土物産も多い．

また，当事業所から自動車で約1時間走れば，伊勢湾や三河湾(Fig.5)のサーフィンスポットにも行けるなど，海にも近いことからサーフィンをする若者も少なくない．湾岸付近に多く存在するナトリウム泉質の高い温泉も人気がある．

中京地区は自動車関連企業が多く，MMC岡崎の近隣にもトヨタ自動車はもとより，アイシン，デンソーなどの大手部品メーカが存在している．

その中でMMC岡崎は585,000m^{2}の敷地を有し，1970年にMMCの開発部門である乗用車技術センター発足以来，一貫してMMC車両の開発拠点になっている．

同一敷地内に名古屋製作所の岡崎工場もあり，コルト，グランディスなどの車両を生産，出荷している．

地理的には，交通の便は良いとは言えず，新幹線の最寄駅である三河安城駅からも自動車で約20分を要する．そのため，MMC殿も含め，ジヤトコ社員もほとんどの人が自家用車通勤をしており，社員の駐車場問題はMMC，ジヤトコにとっても重要な課題となっている．

MMC岡崎の周囲は，かつては全周に渡り田園地帯だったが，最近では高層マンションが建ち始めた．最上階からテストコースを視認できるところも出現し，秘匿車の機密保持も，MMC殿の課題になっている．最近，MMC殿はセキュリティシステムを全面的に見直す等，秘匿対策に更に力をいれている．

Because Okazaki is the birthplace of Tokugawa Ieyasu, shrines, temples and historical sites connected with him, especially Okazaki Castle (Fig. 4), are found everywhere. There are also many local products boasting long traditions such as Mikawa fireworks and Hatcho miso soybean paste.

Since the Okazaki Product Development Department is located close to the sea, many young employees enjoy surfing. Surfing spots along the Ise Bay and the Mikawa Bay (Fig. 5) can be reached in about one hour by car. Many popular hot springs rich in sodium chloride are also located near the coast.

Many companies related to the automotive industry are located in the Chukyou (i.e., Nagoya) area, notably Toyota Motor Corporation as well as large parts manufacturers like Aishin and Denso.

Amid these surroundings, MMC Okazaki occupies a site 585,000 m^2 in area. Ever since the Car Research & Development Center, which serves as MMC's product development base, was established here in 1970, it has consistently served as the heart of MMC's vehicle development activities. Also located on the same premises is the Okazaki Plant of the Nagoya Works that builds and ships the Colt, Grandis and other models.

Geographically speaking, it would be hard to say that transportation connections are good, as it takes about 20 minutes by car from the nearest Shinkansen bullet train station (Mikawa Anjou). For that reason, nearly all JATCO employees and MMC personnel commute to work by their private cars. Securing sufficient parking for everyone is a serious issue for both MMC and JATCO.

In the past, the area near MMC Okazaki was pastoral all around, but high-rise condominiums have begun to be built in recent years. There are now buildings from which the test course is visible from the top floor. Protecting the secrecy of camouflaged cars under development has also become an issue for MMC. Recently, MMC has stepped up its efforts to maintain secrecy, including making an overall review of its security system.

4. 岡崎事業所の今後

合併後半年が過ぎ，岡崎地区でもジヤトコの仕事のやり方が定着してきた．今後は従来の良い面を継承しつつ更に発展させて，合併によるシナジー効果を最大限創出できるよう，進めていく．

MMC殿の要求が益々多様化する中で，その要求にタイムリーに応えていくためには更に開発期間の短縮が必要となってくる．その意味で，最大効率が発揮できるジヤトコ岡崎のロケーションは最適であり，MMC殿向けユニット開発の主要拠点としてのジヤトコ岡崎の役割は，今後益々重要になっていくと予想される．

岡崎(京都)地区のジヤトコ社員は今後ともMMC殿と密接に連携し，より良いユニットをタイムリーに開発，供給していく．

4. Future of Okazaki Product Development Department

JATCO's ways of doing work took root about six months after the merger. Efforts will be made in the coming years to promote further development, while continuing the good aspects of before, so as to create maximum synergies through the merger.

Amid the increasing diversification of MMC's requests, it will be necessary to shorten development lead times further in order to respond to their requirements in a timely fashion. In that sense, JATCO Okazaki is optimally located for attaining maximum efficiencies. It is expected that JATCO Okazaki's role as a major development base of transmissions for MMC will continue to become even more important in the years ahead.

JATCO employees in Okazaki (and Kyoto) will be working closely with MMC in the coming yeas to develop and supply even better transmissions in a timely manner.

■ Author ■

Toyoji OHNISHI

事業所紹介

ジヤトコUSA(JUS)の紹介

Introduction of JATCO USA (JUS)

町田 雅昭*
Tsuneaki MACHIDA

1. 概要

社　名　JATCO USA, Inc. (Fig. 1 & Fig. 2)
所在地　米国ミシガン州ウィクソム市(デトロイト郊外)
設　立　1997年9月17日
資本金　60万ドル(約7千万円)
従業員　50名(出向者を含む　04年5月現在)

当社はジヤトコ(株)の100%子会社で，世界最大の自動車市場(年間1,600～1,700万台)である米国のBIG-3のお膝元であるデトロイトにある．位置的にはデトロイト市中心部から北西に30分の工業団地内にあり，建屋は約2,200m². BIG-3の本社やテクニカル・センターには車で30-40分，日産テクニカル・センター・ノースアメリカ(NTCNA)には15分の距離にある．営業機能(BIG-3向け)，開発実験，品質サービスの他，一昨年からリマン(ATの再生)を開始し，ジヤトコ本社及び北米顧客に対してサービスを提供している．

デトロイトと言うと一面の工業地帯を想像される方も多いと思うが，郊外は緑が多く，豊かな自然が残っており，小動物をよく見掛ける住みやすい環境である．又シカゴには車で4時間，ナイアガラの滝へは5時間，ニューヨークへは10-11時間の距離である．

1. Outline of the Company

Company Name: JATCO USA, Inc.
Location:Wixom, Michigan, USA (suburb of Detroit)
Establishment: September 17, 1997
Capitalization: $600,000
No. of Employees: 50 (as of May 2004, including dispatchees)

JUS is a 100% subsidiary of JATCO Ltd. and is located in suburban Detroit, the home of the "BIG-3" in the USA, the largest automotive market in the world. The company is located within an industrial park 30 minutes northwest from downtown Detroit and has 2,200 square meters of floor space. JUS is 30-40 minutes driving distance from any of the BIG-3 Headquarters and Technical Centers, and 15 minutes from Nissan Technical Center North America (NTCNA). Our functions are Sales & Marketing (for BIG-3), Engineering & Development, Quality/Service and (from 2002) Remanufacturing, and we are providing services to JATCO Ltd. and its customers in North America.

Many people may imagine an all-out industrial zone when they hear the name, "Detroit", but its suburbs are full of greenery and comfortable for living with lots of nature and you can see small animals frequently. We can drive 4 hours to Chicago, 5 hours to Niagara Falls, and 10-11 hours to New York city.

Fig. 1 JUS outside

Fig. 2 JUS inside

* JATCO USA, Inc.

Fig. 3 Downtown Detroit

Fig. 4 Detroit suburbs

2. 生い立ちと変遷

ジヤトコは1989年にデトロイトに情報収集を主目的とした駐在員事務所を設立し，その後，客先要請でロサンゼルスに品質調査要員を置いたが，駐在員事務所の性格から法律上営業活動は不可能だった．1997年に北米での営業活動，特にBIG-3よりの新規受注を狙った営業マーケッティング活動を開始することを目的に，この駐在員事務所を現地法人化し，JUSが発足した．

発足当初の機能は営業マーケッティングと品質サービスであり，日本人3名，米人2名で出発．営業ではBIG-3へのアプローチを開始し，品質サービス面では客先は日産，マツダ，いすゞ，三菱(ふそう)，日産ディーゼルの米国ディストリビュータ会社だった．その後，本社より本社の設計工数不足を補完する形で設計部隊を設置しないかとの話があり，1998年に小さいながら設計部隊を設置し，CADを入れた．FPシリーズの強化スケッチ検討の他，2次元図面の3次元図面化にも取り組んだ．また，同年米国マツダ向けに日本へコア返却，日本からリマン品を輸入販売するビジネスを開始した．

2001年にはオフィスが手狭となったため，現在地に新建屋をリースし，モーターベンチを設置すると共に2002年より小規模ながらリマンを開始した．品質サービス関係では新たにVW，ランドローバー，ジャガーが客先として加わった．又客先北米工場対応も開始した．

営業面ではBIG-3に対し粘り強くFPシリーズやCVTの売り込みを図り，単独ビジネスだけでなく，戦略パートナーとして認知，採用して貰えるように努力を続けた．米国では当初CVTについては懐疑的だったが，CAFE(企業平均燃費)の強化と共に燃費効率の高いCVTへの関心が高まっており，ビ

2. Its Birth and History

In 1989 JATCO established its Detroit Office with the main purpose of information gathering, and later placed quality investigation staff members in Los Angeles upon the request of a customer. Legally, however, we were not allowed to conduct any sales activity because of the nature of a representative office. In 1997 we modified this Office into a U.S. corporation with the objective of starting sales & marketing activities in North America, especially aiming at winning the BIG-3 business. This is how JUS was born.

The original functions of JUS were Sales & Marketing and Quality/Service, and we started with three (3) Japanese and two (2) American staff members. We immediately began our sales activities to the BIG-3 and continued our Quality/Service activities to our then customers; U.S. Distributors of Nissan, Mazda, Isuzu, Mitsubishi (Fuso) and Nissan Diesel. Soon after, JUS was asked to form a Design Team to alleviate the shortage of Product Engineering manpower at Headquarters and we started a small Design Team and installed CAD equipment. JUS studied sketch to fortify torque capacity for FP series (5 speed FWD AT) and also made 3D CAD files out of 2D drawings. The same year, we started our Reman business with Mazda North American Operations (MNAO) where JUS returns cores to Japan and imports/sells remanufactured ATs from Japan.

In 2001, as our then-office became small, we moved to our current leased facility at present site, and installed a motor bench for quality investigation, and from 2002 we started small Remanufacturing Operation. We added new customers in the Quality/Service area, such as VW, Land Rover and Jaguar. Also, we began to support our customers' North American plants.

As for Sales & Marketing, we made a persistent sales

ジネスチャンスは拡大している．またBIG-3や欧米部品メーカとの共同検討プロジェクト案件も複数あった．

この間，BIG-3の底力を見せつけられたり，グローバルな情報共有化のスピード，意志決定のスピード等，日本との彼我の差を感じることも多かったが，反面日本の自動車産業の強みも再評価できた．また日米ビジネス文化間の決定および実行プロセスの時間カーブの違いにも気が付いた．これらの相違は両者の忍耐と理解をまだ必要としている．

3. 現在の姿

JUSは現在財務・総務・人事からなる管理部門と営業マーケッティング，設計・実験，品質サービス，リマンと大きく5部門，合計50人の所帯となっている．その中でも2003年には特に実験と品質サービス部門が大幅に強化された．特に実験部隊は殆ど会社にいない状態で客先で日夜奮闘している．日本人出向者が19名(内ロス3名)と出向者比率が高いが，仕事の立ち上げ時点ではどうしてもある程度は仕方がない．今後現地人を育成し，徐々に人の現地化を図るのも大きな課題である．また日米文化の違いによる考え方や仕事の進め方の違い，コミュニケーション上のトラブルも日本人・米人がお互いに努力して乗り越えなければいけない．

JUSでは毎年SAE大会の初日の夕方にオープンハウスを開催し，日頃お世話になっている客先の方々をお招きしてJUSの成長ぶりをご覧頂いている．これはなかなか好評で毎年顔を出して下さる方も多い．またGlobal Powertrain CongressやSAE Toptec等へのブース出展の他，SAE誌へ広告を出したり，広告塔を契約したり，SAEディナーへお客様をご招待したりしている．

4. JUSの今後

JUSは現在２大プロジェクトを抱えている．ひとつはJATCO Mexico S.A.製CVTビジネスの確実な立ち上げと成功であり，もうひとつはJUSでの品質調査体制の拡充とリマンの拡大である．このために一段と広いスペースが必要であり，新建屋の建築(リース)を計画している．人数も2007年には150人を超える計画であり，本当に会社としての形・実力が今迄以上に求められる．

また本社やJATCO Europe GmbH他との連繋に加え，メキシコに建設中のJATCO Mexico S.A.との連

pitch to the BIG-3 for our ATs and CVTs, and continued our efforts to let them consider JATCO not only as a single business supplier but also a strategic partner. In USA, initially there were doubts about CVTs among the BIG-3. However, with increasingly tougher US CAFE requirements, their interest for CVTs is growing and our business opportunities are expanding. Also, there were multiple joint study projects with BIG-3 and European/American auto parts suppliers.

During this time, I could feel the differences between Japan and USA. We witnessed the real power of BIG-3, the speed of their global communication, the speed of decision-making, etc. On the other hand, I re-evaluated some strong areas of the Japanese auto industry. We also noticed differences between the two business cultures on the time-curve lines in the decision and implementation processes. Those differences still require patience and understanding from both parties.

3. Current JUS

JUS now has 5 Departments: Administration (comprised of Finance/Accounting, General Affairs and Human Resources), Sales & Marketing, Engineering/Development, Quality/Service and Remanufacturing. We are now 50 in total. In 2003 we fortified especially our Development and Quality/Service teams. The Development staff members are hardly seen here in our Office, as they are working hard at our customer's sites. We now have 19 expatriates (including 3 in LA) and the ratio of expatriates is high, but it is necessary at these business launching stages. We have a big challenge of developing local employees and gradually localizing our human resources. Also we (Japanese and Americans) have to make our best efforts to overcome: the differences in our ways of thinking & our work the potential communication troubles arising from the differences of Japanese and US cultures.

JUS holds an annual Open House on the first day evening of SAE week and invites our customers to observe our growth. This event is well accepted, and we welcome many repeat visitors. Also JUS exhibits a booth at the Global Powertrain Congress, SAE Toptec, etc., puts an advertisement page in the SAE "Supercharger" magazines, billboards and invites our customers for SAE Detroit section dinners.

4. The Future of JUS

Currently, JUS has two (2) big projects. The smooth launch and success of JATCO Mexico S.A.-made CVT business, and the fortification/expansion of JUS Quality

繋もスタートする．本当の意味でグローバルJATCOの発展に貢献できる環境が整ってきた．この機会を十二分に活かして夢をひとつひとつ実現できるよう引き続き努力していきたい．本社や関係先の皆様には改めて御礼申し上げると共に，引き続き温かく見守って頂き，叱咤激励・ご支援の程を宜しくお願い申し上げます．

Investigation functions and Remanufacturing Operations. We need more space for these activities, and are planning to lease a larger building. In 2007, we envision more than 150 employees and JUS will be requested to be really in good shape and powerful as a corporation.

In addition to the current collaboration with Headquarters and JATCO Europe GmbH, we will need to begin collaborating more with JATCO Mexico S.A., presently under construction. Now that the circumstances are becoming mature enough, JUS will make a significant contribution to Global JATCO 's growth. We will continue to do our best to materialize our dreams making this opportunity as successful as possible. We thank our customers, Headquarters staff and other related people, and ask for your advice and support, while observing our growth from afar.

■ Author ■

Tsuneaki MACHIDA

特　許　紹　介

Patents

1. 車両用油圧装置の制御装置

(Fig. 1)

出　願：出願日　1989.6.7　特願平1－143001
登　録：登録日　1998.9.18　特許登録第2829031号
名　称：車両用油圧装置の制御装置
発明者：佐野　邦彦

【目的】
エンジンアイドリング状態で，かつ電源電圧が一定値以上の場合に，電子制御装置からソレノイドに指令信号が出力されるとソレノイドが作動してしまうため，空打ちが発生するという問題がある．本発明はこのような従来技術の問題を解決し，空打ちをなくしソレノイドの耐久性を向上させることを目的としている．

【発明の構成】
エンジン回転速度が300rpm以上であるかどうかを判断し，300rpm以下，すなわち実質的にエンジンが停止している状態では，他の条件にかからずライン圧ソレノイドの作動を禁止する．

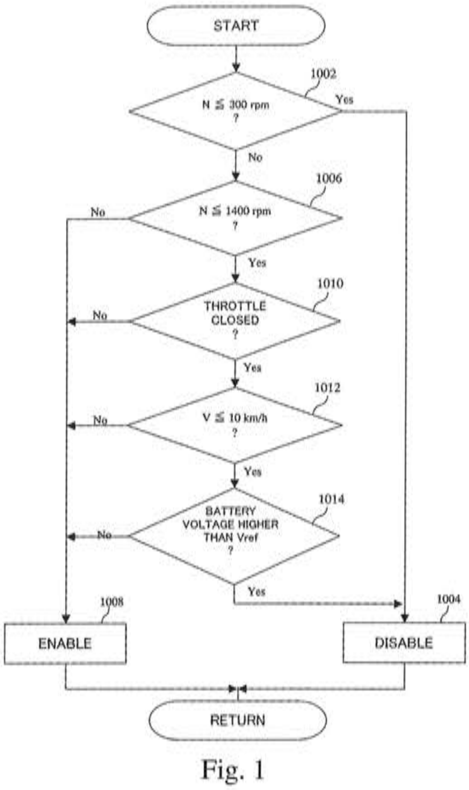

Fig. 1

1.Pressure control system for automotive automatic power transmission

(Fig. 1)

Application Number: 1-143001
Application Date: 6.7.1989
Patent Number: 2829031
Registration Date: 9.18.1998
Title: Pressure control system for automotive automatic power
Inventor: Kunihiko Sano

【Summary of the invention】
It is an object of the present invention to provide a pressure control system which can solve the problem in the prior art and thus can provide a longer life for the solenoid.

The process as shown in Fig.1, at the step 1002, the engine speed indicative signal value N is checked whether it represents an engine speed lower than or equal to 300 rpm. When the engine speed is lower than or equal to 300 rpm, judgement can be made that the engine is stopping.
In such case, the line pressure solenoid is disabled at a step 1004.

On the other hand, when the engine speed indicative signal value N represents an engine speed higher than 300 rpm as checked at the step 1002, the engine speed indicative signal value N is checked whether it represents 1400 rpm which serves as a low engine speed criterion. The low engine speed criterion may be set at a possible highest engine idling speed at a no load condition, at a step 1006. If the engine speed N as checked at the step 1006 is lower than or equal to 1400 rpm, a check is performed whether the throttle valve is fully closed or the open angle thereof is substantially small to be judged that the throttle valve is nearly fully closed, at a step 1010.
In practice, at the step 1010, the throttle angle indicative signal value TVO is compared with a throttle open angle criterion to make judgement when the throttle angle indicative signal value TVO is smaller than the throttle open angle criterion. If the throttle valve open angle is smaller than the throttle open angle criterion as checked at the step 1010, the vehicle speed is checked whether it is lower than a predetermined low vehicle speed

また，エンジン回転速度が300rpmよりも大きくエンジンが動作している場合であっても，ほぼ車両が停止しているエンジンアイドリング状態（エンジン回転速度が300rpmよりも大きく1400rpm以下であり，スロットル開度が全閉に近い所定値以下であり，車速が10Km/h以下であり），かつコントロールユニット300の電源電圧が一定値以上である場合にライン圧ソレノイドの作動を禁止する．これ以外の場合にはコントロールユニット300から指令される信号に応じてライン圧ソレノイドの作動を許可する．

【作用・効果】

エンジンアイドリング状態で，かつ電源電圧が一定値以上の場合にライン圧ソレノイドの作動を禁止することにより，ほぼ車両が停止したエンジンのアイドリング状態の比較的騒音の小さい状態でライン圧ソレノイドの作動による大きい音が発生するのを防止し，車室内に大きい音が入ってしまうことによる不快感を与えるのを防止する．

2. 自動変速機の制御装置

(Fig. 2 & 3)

出　願：1992.8.24　特願平4−247327
登　録：2001.12.14　特許第3260438号
名　称：自動変速機の制御装置
発明者：飯塚尚典

【目的】

自動変速機が変速を指令しない状態においてライン圧の学習制御が行われることを防止する．

【発明の構成】

自動変速機の入力軸回転速度を検出する入力軸回転速度センサと，自動変速機の出力軸の回転速度を検出する出力軸回転速度センサと，両センサからの信号に基づいて変速比を算出する変速比算出手段と，変速に際して変速比の変化が開始されてから終了するまでの時間を計測するイナーシャフェーズ時間計測手段と，これによって計測されるイナーシャフェーズ時間があらかじめ設定された目標値と一致するように変速中に摩擦締結要素に作用させるライン圧を調整するライン圧調整手段と，変速が指令されてから変速比の変化が開始されるまでのイナーシャフェーズ開始前時間を計測するイナーシャフェーズ開始前時間計測手段と，を有しており，前記イナーシャフェーズ開始前時

criterion, e.g. 10 km/h, at a step 1012. When the vehicle speed is lower than or equal to the low vehicle speed criterion, the battery voltage is checked if it is higher than or equal to the predetermined value Vref. If battery voltage is higher than or equal to the predetermined value, the process goes to the step 1004 to disable the line pressure solenoid. Otherwise, the process goes to the step 1008. After processing process at one of the steps 1004 and 1008 the process goes to an END.

2. Control system for automotive automatic transmission

(Fig. 2 & 3)

Application Number: 4-247327
Application Date: 24.8.1992
Patent Number: 3260438
Registration Date: 14.12.2001
Title: Control system for automotive automatic transmission
Inventor: Naonori Iizuka

【Summary of the invention】

It is an object of the present invention to preventing the learning control of the line pressure in the condition that the transmission does not instruct the gear change.

line pressure increase-inhibitingmeans

In order to accomplish the object, a control system for an automotive automatic transmission including an input shaft rotation speed sensor for sensing the rotation speed of an input shaft of the transmission, an output shaft rotation speed sensor for sensing the rotation speed of an output shaft of the transmission, a gear ratio deriving means for driving a gear ratio based on the information signals from said input and output shaft rotation speed sensors, an inertial phase keeping time measuring means for measuring the time elapsed from the time when a change of said gear ratio starts to the time when said change ends, a line pressure adjusting means for adjusting the line pressure in a manner to harmonize the inertial phase keeping time with a predetermined target value, an inertial phase starting time measuring means for measuring the time elapsed from issuance of a gear change instruction to the time when the change of said gear ratio starts, and a line pressure increasing means for increasing the line pressure irrespective of condition of said line pressure adjusting means when said inertial phase starting time exceeds a reference value, which is characterized in that an inhibiting means is employed which inhibits operation of said line pressure increasing means in case wherein even when said inertial phase starting time exceeds a given value longer than said reference value, the change of said gear ratio fails to occur.

間があらかじめ設定された基準値を越えた場合には，前記ライン圧調整手段にかかわらずライン圧上昇手段により変速中のライン圧を上昇させるが，前記イナーシャフェーズ開始前時間が前記基準値よりも長い時間である所定時間を経過した場合には，ライン圧上昇禁止手段により前記ライン圧上昇手段の作動を禁止する構成とした．

【作用・効果】

本発明によると，イナーシャフェーズ開始前時間が所定時間よりも長い場合には変速中のライン圧を急速に上昇させる制御を行わないようにしたので，セレクトポジションスイッチが故障して，例えばDレンジの信号が出力されているが実際には自動変速機がNレンジの状態となっているような場合にはライン圧の学習制御が行われない．したがって，セレクトポジションスイッチが正常に復帰した場合にライン圧は通常の状態となっており，大きい変速ショックを生ずることはない．

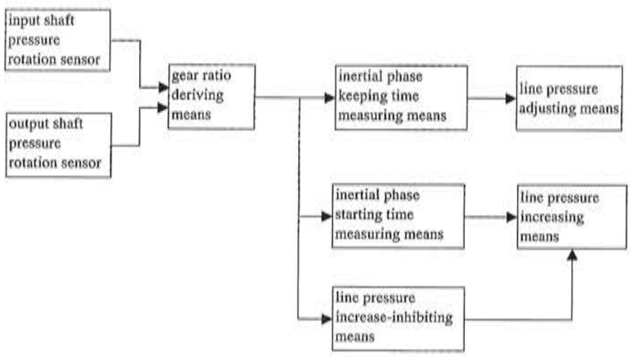

Fig. 2

According to the invention, the inertial phase starting time exceeds a predetermined time, the control for quickly increasing the line pressure is suppressed. Thus, when, due to for example a failure of the selection position switch 304, the transmission is kept in N-range irrespective of issuance of D-range instruction signal, the learning control to the line pressure is not carried out. Accordingly, when the select position switch thereafter returns to work, the line pressure assumes a normal condition and thus undesired shift shock is not produced.

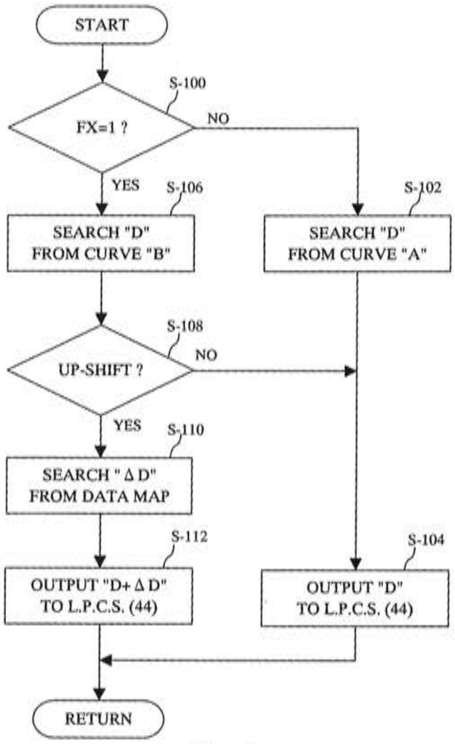

Fig. 3

社外技術発表一覧（2003年1月1日～2003年12月31日）

発表日	発表先	表題	発表者
2003.2.6	自動車技術会関東支部主催 講演講習会	CVTの現状と将来技術動向	第一性能設計部　柴山　尚士 第二商品開発室　菅野　一彦 第一商品開発室　安保　佳寿
2003.2.18-19	Mechatronische Getriebesysteme	Development of toroidal CVT	先行ユニット開発室　日比　利文 先行ユニット開発室　竹内　徹 先行ユニット開発室　住　泰夫 日産自動車（株）　山本　建 第二商品開発室　木島　正道
2003.3.3～6	SAE (Detroit, USA)	Development of New-Generation Belt CVTs With High Torque Capacity for Front-Drive Cars	第一商品開発室　安保　佳寿 第二商品開発室　菅野　一彦 第一性能設計部　柴山　尚士 日産自動車（株）　早崎　康市
2003.3.14	日本機械学会関東支部 第9期　総会講演会（横浜）	トラクションドライブを応用したトロイダルCVTの開発	先行ユニット開発室　竹内　徹
2003.4	自動車技術会誌 '03/4月号	トルクコンバータの流体技術動向と将来展望	第一機能部品開発部　岡田　克彦 第一機能部品開発部　立脇　敬一
2003.5.19～22	2003 JSAE/SAE International Spring Fuels & Lubricants Meeting (Yokohama)	Development of a Multi-purpose ATF Meeting DEXRON III, MERCON and JASO M315 specifications	第一機能部品開発部　荒川　慶江 第一機能部品開発部　矢内原梨花 第一機能部品開発部　村上　靖宏
2003.5.19～22	2003 JSAE/SAE International Spring Fuels & Lubricants Meeting (Yokohama)	Testing Method and Effect of ATF Performance on Degradation of Wet Friction Material	第一機能部品開発部　前田　誠 第一機能部品開発部　村上　靖宏
2003.5.21～23	(社)自動車技術会 学術講演会 春季大会（横浜）	薄型トルクコンバータの開発	第一機能部品開発部　岡田　克彦 第一機能部品開発部　立脇　敬一
2003.5.21～23	(社)自動車技術会 学術講演会 春季大会（横浜）	高トルク容量ベルトCVTの開発	第一商品開発室　天野　宏 第一商品開発室　安保　佳寿 第二商品開発室　菅野　一彦 日産自動車（株）　早崎　康市 日産自動車（株）　小林　昌之 日産自動車（株）　水口　賢
2003.5.21～23	(社)自動車技術会 学術講演会 春季大会（横浜）	高性能ベルトCVTフルードの開発	第一機能部品開発部　茂木　靖裕 第一機能部品開発部　荒川　慶江 第一機能部品開発部　村上　靖宏
2003.5.28～30	第3回制御部門大会（神戸）	自動変速機制御装置の機能検査システムの開発とその応用	先行技術開発部　松村　利夫 第二機能部品開発部　市川　修二 第一機能部品開発部　佐藤　雅行
2003.6.4	(社)自動車技術会中部支部 研究発表会（名古屋）	新高トルク容量CVTの開発	第二商品開発室　菅野　一彦 第一商品開発室　安保　佳寿 第一性能設計部　柴山　尚士 第一性能設計部　岡原　博文 第一性能設計部　落合　辰夫
2003.8.5～8	日本機械学会 2003年度年次大会（徳島）	各種添加剤が金属間の摩擦係数にあたえる影響とその評価方法に関する考察	第一機能部品開発部　村上　靖宏 第一機能部品開発部　荒川　慶江 第一機能部品開発部　茂木　靖裕
2003.8.5～8	日本機械学会 2003年度年次大会（徳島）	自動車用扁平トルクコンバータの内部流れ解析	第一機能部品開発部　岡田　克彦 第一機能部品開発部　立脇　敬一
2003.8.5～8	日本機械学会 2003年度年次大会（徳島）	3次元数値シミュレーションによるベルトCVTトルク伝達メカニズム	先行技術開発部　加藤　芳章 先行技術開発部　河野　義裕
2003.9.2～5	2003 ASME/AGMA 2003 (Chicago, USA)	Investigation of the Noise and Vibration of Planetary Gear Drives	第二構造部品設計部　陳　勇

発表日	発表先	表題	発表者
2003.9.17~19	(社)自動車技術会 学術講演会 秋季大会(名古屋)	歯面強度に優れた浸炭窒化歯車用鋼の開発	第一機能部品開発部 吉田 誠 (株)神戸製鋼所 永濱 睦久 ユニット技術部 田中 敏行 第一構造部品設計部 新明 正弘 第一実験部 清田 祥司 第一FRA/T事業所 加藤 直樹 (株)神戸製鋼所 岩崎 活浩 日産自動車(株) 渡辺 陽一
2003.9.17~19	(社)自動車技術会 学術講演会 秋季大会(名古屋)	AT制御ソフトウェアにおける品質開発の体系化と適用	第一機能部品開発部 佐藤 雅行
2003.9.17~19	(社)自動車技術会 学術講演会 秋季大会(名古屋)	3.5LクラスベルトCVTのトルク容量解析	先行技術開発部 加藤 芳章 第一実験部 山下 弘 先行技術開発部 河野 義裕
2003.9.23~25	Global Powertrain Congress (Detroit, USA)	Development of New Belt CVTs with High Torque Caoacity	第一性能設計部 岡原 博文 第二商品開発室 菅野 一彦 第一商品開発室 安保 佳寿
2003.11.11~13	トライボロジー会議2003秋 (新潟)	自動車用ベルトCVTの摩擦特性解析	先行技術開発部 加藤 芳章 先行技術開発部 河野 義裕 第一機能部品開発部 伊藤 靖朗 第二実験部 水科 文男
2003.11.21	JALOS環境フォーラム 経済産業省資源エネルギー庁潤滑油環境対策事業	拡大する自動車用自動変速機特需に応じる環境対応技術	第一機能部品開発部 村上 靖宏
2003.11.27	(社)自動車技術会 シンポジウム『動力伝達系の最新技術2002』(東京)	高トルク容量ベルトCVTの開発	日産自動車(株) 浦沢 徹 第一機能部品開発部 若原 龍雄 第一機能部品開発部 山本 雅弘
2003.12.2~3	2nd International IIR-Symposium Innovative Automotive Transmission (Offenbach, Germany)	Development of NewGeneration High Torque Belt-CVT for 3.5-liter Engine Class Cars	第一商品開発室 安保 佳寿

編　集　後　記

ジヤトコの技術論文集であるジヤトコ・テクニカル・レビューが誕生して、早いものでもう第5号の発行となりました。今回の第5号を含めますと、掲載した論文は95件、誌面に登場していただいた執筆者は170名を超えました。当社の技術が確実に蓄積されているという実感があります。

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－ ジヤトコ・テクニカル・レビュー事務局　佐藤真琴 －

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