Half-Toroidal Traction Drive Continuously Variable Power Transmission for Automobiles*

(Traction Drive Materials, Transmission Design and Efficiency)

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A continuously variable power transmission, CVT, has the potential to provide comfortable cruising, powerful acceleration, fuel economy and improvement of exhaust emission as an automotive propulsion system. The authors have been engaged in research and development of a half-toroidal traction drive CVT since 1980. In practical use, reliability and compactness are important issues, which motivate ceaseless improvement of traction drive materials, synthetic traction fluids and transmission efficiency. This paper focuses on the fatigue of traction rolling elements, endurance of high-temperature traction fluids and efficiencies of single- and dual-cavity CVTs for 1.5 L and 3 L passenger cars.

Key Words: Power Transmission, Traction Drive, Continuously Variable Transmission, Fatigue Life, Traction Fluid, Automobiles

1. Introduction

Environmental issues such as exhaust emission and fuel economy have stimulated the development of a continuously variable power transmission for an engine-vehicle propulsion system. The authors have been developing rolling-type CVTs, continuously variable transmissions, of a half-toroidal traction drive since 1980⁽¹⁾⁻⁽³⁾. In practical use, reliability and compactness are important issues, which motivate ceaseless improvement of traction drive materials and

synthetic traction fluids. This paper focuses on the fatigue of traction rolling elements, endurance of high-temperature traction fluids and efficiencies of single- and dual-cavity CVTs for 1.5 L and 3 L passenger cars.

2. Rolling Materials and Traction Fluids

2.1 Surface-originated rolling contact fatigue

A traction drive transmits shear force through a highly pressurized thin EHL film. Fatigue usually starts to occur on the driven output disk which is subjected to large tensile stress. Carburized heat treatment is effective in increasing the life of the rolling materials. Recent investigation has clarified the reason for surface- or subsurface-originated flaking^{(4),(5)}. Both of them are primarily caused by deposition of hard nonmetallic oxides such as Al_2O_3 and $\text{CaO} \cdot \text{Al}_2\text{O}_3$. However, even after removing these inclusions, surface-originated flaking starts to occur before subsurface flaking. One of the reason for this is a gradual decrease of film thickness due to fluid viscosity change caused by high-temperature shear, and another is wear of minute projections of the rolling materials. Both factors are examined by using

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Fig. 1 Half-toroidal CVT unit

Fig. 2 Viscosity changes of early-stage oil and new type traction oil after endurance tests

a single-cavity CVT shown in Fig. 1. The CVT is tested at a maximum Hertzian pressure of $P_{\max}=2.9$ GPa, contact force of $F_c=1.23 \times 10^4$ N and velocity of $U=15$ m/s. Geometric parameters of the CVT are $r_0=30$ mm, $R_{22}=24$ mm and $\theta_0=55^\circ$ which are radius of disk, major radius of curvature of power roller and half-cone-angle of power roller, respectively. The major and minor radii of the contact ellipse are 2.812 $\times$ 0.712 mm.

The above-mentioned fatigue is caused by the decrease of viscosity due to damage to polymer additives under high-pressure, high-temperature shear drive. Figure 2 shows a comparison of change in viscosity of traction fluids measured at 40°C after endurance traction tests at the bulk temperatures of 80°C and 120°C. The early-stage oil is a conventional traction fluid of cyclo-aliphatic hydrocarbon containing zinc and sulphur. The viscosity decreases from 3.1×10^{-2} Pa·s to 2.5×10^{-2} Pa·s after 100 hours in the 80°C test. However a new type of alicyclic compound with a small amount of additive withstands testing with

Fig. 3 Life and fatigue index of rolling bearings measured by X-ray diffraction (● : surface-originated, ○ : subsurface-originated)

high-temperature shear of 120°C⁽⁴⁾. A calculated film thickness parameter, thickness divided by surface roughness, of the early-stage oil changes from 1.3 to 1.1 after 100 hours; however the latter remains at 1.4 ~ 1.3 after 300 hours. The issue of early damage to the traction surface is almost solved by using the newly developed traction fluid.

The progression of fatigue life of the traction surface is evaluated by X-ray diffraction measurement. A fatigue index⁽⁵⁾ is defined by the decrease of half-breadth value related to the amount of retained austenite and residual stress on and beneath the raceway. Figure 3 shows empirical relations of typical ball bearing fatigue index and remaining life of subsurface- and surface-originated fatigue. According to the figure the surface-originated fatigue life ends at the index of 2 (filled circles in Fig. 3) and subsurface fatigue life at 3. We apply this measurement to the life estimation of the traction drive materials of carburized steel with hardness of $H_{RC} 60$ after a test of 10^8 traction cycles at 110~120°C. Figure 4 shows that the fatigue index (FI) decreases exponentially from the surface and it does not spread beyond the depth of 0.3 mm. If we apply the ball bearing index, the life of the surface with $FI=3.37$ should be finished; however actually no damage is observable. The reason for this is primarily the difference of heat treatment.

Figure 5 shows the fatigue test results of four models of CVTs with input power capacity from 9 kW to 73.5 kW. In the figure the model numbers (TX68 ~ TX116) express the distance (mm) between the swinging center of a pair of power rollers, and filled symbols are failed results of testing with early-stage traction oil. The solid line is a calculated fatigue life estimated from the stress volume principle, parameters of which are used according to Coy's⁽⁶⁾ helical gear test of AISI 52100 steel with $H_{RC}=62$ hardness.

Fig. 4 Fatigue index of traction surface measured on and beneath the raceway (after 10^8 cycles with oil pan temperature of $110\sim 120^\circ\text{C}$)

Fig. 5 Fatigue life tests of four sizes of CVTs

Fig. 6 Traction coefficients of aromatic (Fluid 1) and alicyclic (Fluid 2) compounds

In our case it is possible to take the proportional constant to be four times larger and the fatigue issue is almost solved with the traction design of the maximum Hertzian pressure of 2.7 GPa.

2.2 Alicyclic Compounds Traction Fluid

It is well known that alicyclic compounds has higher traction coefficient than aromatic compounds⁽⁷⁾. Figure 6 shows a comparison of trac-

aromatic compounds

alicyclic compounds

Fig. 7 Photos of stable conformation of molecules of aromatic and alicyclic compounds

tion coefficients of aromatic (Fluid 1) and alicyclic (Fluid 2) compounds tested by the two disks rolling machine of SUJ2 bearing steel with $H_R C=62$ hardness. The Hertzian contact pressure is $P_{\max}=1.1$ GPa and rolling speed and slip are $U=4.1$ m/s and 5%. The traction property is related to the strength of shear resistance of fluid molecules which is controlled by the molecular shape and its stiffness. Figure 7 shows a comparison of the most stable conformation of both fluid molecules. We can see the alicyclic compounds (Fluid 2) hooked each other toughly than the aromatic compounds (Fluid 1). The stiffness of molecules is evaluated from the rotational barrier energy of C-C bonds substituting to the ring⁽⁸⁾. The alicyclic compounds (Fluid 2) have the energy of 3.43 kJ/mol, while the aromatic (Fluid 1) of 2.39, which means the former has higher stiffness of C-C bond than the latter. The array model of alicyclic compounds nipped by the traction surfaces is shown in Fig. 8. The newly designed traction fluid named Daphne 7074, viscosity of $0.458\text{ cm}^2/\text{s}$ (40°C) and $0.0652\text{ cm}^2/\text{s}$ (100°C) with viscosity index of $VI=90$, keeps the initial traction property as shown in Fig. 6. It is measured after the endurance test of 10^8 cycles by 1.3 L engine driven CVT⁽⁹⁾ at the working condition of $P_{\max}=2.15$ GPa, $U=24.2$ m/s, $T=120^\circ\text{C}$.

3. Efficiencies of Single- and Dual-Cavity CVTs

3.1 Single-cavity CVT

In 1988 a single-cavity half-toroidal CVT was designed for a 1.5 L turbo-charged FF drive passenger

Fig. 8 Molecular array model of alicyclic compounds

Fig. 9 Automatic power transmission with a single-cavity CVT for a 1.5 L turbo-charged passenger car

car. Figure 9 shows the transmission layout having a torque converter. Forward or reverse gears are selected by wet clutches after the output disk of CVT⁽³⁾. The specifications of the CVT are listed in Table 1. Basic characteristics of the speed ratio changing mechanism, vehicle acceleration and power transmission performance (rated input torque of 157 N·m rotational speed of 6000 rpm) were tested. The torque transmission efficiency is poor at 75%, whereas the speed transmission efficiency is 98%. There are three major causes of torque loss: spin moment at traction surface⁽¹⁾, thrust ball bearing of PCD=58 mm and two large angular ball bearings of PCD=80 mm. Figure 10 shows measured torque losses of these parts. The spin moment of the traction surface is reduced by optimum design of the aspect ratio of the toroidal disks and speed ratios, and by reducing the contact ellipsoid to a circle within the fatigue limits. The drag moment of thrust ball bearings is drastically reduced by using the shallow groove bearing⁽¹⁰⁾. For the angular bearings, the larger the contact angle, the

Table 1 Specifications of the transmission with a single-cavity CVT unit and the prototype dual-cavity CVT unit

Cavity Formation		Single(Trans.)	Dual(Unit)
Toroid Radius (m)		5.0×10^{-2}	5.0×10^{-2}
CVT	Reduction	2.366	2.366
	Overdrive	0.422	0.422
Transmission	Reduction	12.62	2.503
	Overdrive	2.25	0.446
Disk Diameter(m)		1.65×10^{-1}	1.65×10^{-1}
Trunnion Shaft Distance(m)		1.50×10^{-1}	1.50×10^{-1}
Torque Converter Diameter(m)		2.36×10^{-1}	—
Axle Drive		FF	Gearbox
Ratio Control		Electric-Hydraulic	
Engine		1.5Liter Turbo	3.0Liter Turbo
Max. Input Speed		6000 rpm	7000 rpm
Max.Input Torque		157 N·m	333 N·m

Fig. 10 Experimental analysis of torque losses of the single-cavity CVT

Fig. 11 Prototype of a dual-cavity CVT

better⁽¹¹⁾, but it is much better to eliminate these bearings by changing the configuration from single- to dual-cavity CVT.

3.2 Dual-cavity CVT

Fundamental tests on efficiency and speed ratio synchronization of a dual-cavity CVT are executed by a prototype shown in Fig. 11. The input torque is transmitted from the input disks through the speed ratio changing power rollers to the output disks. An innovation of this transmission is to have a fixed wall⁽³⁾ for supporting the output disks, while the

Fig. 12 Efficiency test result of the prototype dual-cavity CVT

Fig. 13 Theoretical analysis of slip and torque losses of the prototype dual-cavity CVT

Fig. 14 Automatic power transmission with a dual-cavity CVT for a 3 L turbo-charged passenger car

conventional designs of dual-cavity CVTs such as Perbury's fully toroidal⁽¹²⁾ or Kraus's half-toroidal⁽¹³⁾ have no wall, which causes unequal torque transmission of rolling elements due to the reaction force from output gears or chains. Measured efficiency of the prototype is shown in Fig. 12. We can see that it remains 90% in the torque range of 50 N·m to 333 N·m. Figure 13 shows theoretical analyses of efficiencies^{(11),(10)} where the total efficiency (solid line) is expressed by multiplying the speed transmission efficiency (two-dotted line) by torque transmission efficiency (broken line). Following the prototype test,

Table 2 Specifications of the automatic power transmission with the dual-cavity CVT unit

Cavity Formation		Dual
Toroid Radius (m)		4.5×10^{-2}
CVT Ratio	Reduction	2.302
	Overdrive	0.434
Transmission Ratio	Reduction	3.055
	Overdrive	0.576
Disk Diameter(m)		1.65×10^{-1}
Trunnion Shaft Distance(m)		1.44×10^{-1}
Torque Converter Diameter(m)		2.5×10^{-1}
Axle Drive		Rear Wheel Drive
Ratio Control		Electric-Hydraulic
Engine		3.0 Liter Turbo
Max. Input Speed		7000 rpm
Max. Input Torque		392 N·m

Fig. 15 Efficiency measurements of the transmissions with the single- and dual-cavity CVTs

a new type of dual-cavity CVT shown in Fig. 14 with a torque converter and a planetary gear set was designed in 1990⁽¹⁴⁾. Acceleration, fuel consumption, noise and speed ratio synchronization have been tested in a 3 L passenger car. Its specifications are listed in Table 2. The dual-cavity CVT succeeds in achieving a substantial efficiency, as shown in Fig. 15, and cruising comfort.

4. Conclusions

Design criteria of half-toroidal traction drive CVTs for automobile use are considered. Three fundamental investigations of rolling fatigue, traction fluid molecular structures and efficiency are performed. Following the prototype test, a dual-cavity CVT for a 3 L passenger car was designed and its performance was tested. The results are summarized below.

(1) The fatigue index measurement on and beneath the traction surface by means of X-ray diffraction clarifies that the fatigue starts to occur from the surface when the maximum Hertzian pressure is designed to be lower than 2.7 GPa. Traction

Fig. 16 Photo of the dual-cavity CVT unit

Fig. 17 Photo of the automatic power transmission with the dual-cavity CVT

life of carburized steel is shown in the $P_{\max}$ - N chart of Fig. 5.

(2) Traction fluid consisting of alicyclic compounds maintains its initial property after traction tests of 10^8 cycles at a bulk temperature of 120°C as shown in Fig. 2.

(3) The single-cavity CVT for a 1.5 L passenger car has the maximum efficiency of only 75%. The analysis of torque loss clarifies the quantitative improvements of the deep groove thrust ball bearings and the balance mechanism of the loading force.

(4) A newly designed dual-cavity CVT with speed ratio range of 0.434 to 2.303 and transmission capacity of 200 kW succeeds in achieving a total efficiency of 85% in the transmitting torque range of $100\text{ N}\cdot\text{m}$ to $300\text{ N}\cdot\text{m}$. Photos of the dual-cavity unit and transmission are shown in Fig. 16 and Fig. 17, respectively.

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