Frictional Loss Mechanism Due to Polygonal Winding Slip of Chain Type CVT

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Abstract. The objective of this study is to investigate the influence of frictional loss due to polygonal winding slip on the gross energy loss of chain type CVTs. The frictional losses between rocker pins and link and that of rocker pins and pulley sheaves were investigated. It was found that the former was very low compared with gross energy loss, on the other hand, the later was relatively large compared with gross energy loss of CVT especially when driving torque was decreased, while the frictional loss was increased with increasing driving torque. This study found that remarkable about 26.5 % was remained as inevitable frictional loss due to polygonal winding slip of the current CVT.

1. Introduction

Continuously Variable Transmissions (CVTs) have been installed into many cars since they provide comfortable driver's feeling under any driving circumstance and low fuel consumption by adjusting the speed ratio continuously. There are many researches related with CVTs. CVTs installed to passenger cars are classified into two types, push belt type CVTs and tensile chain belt type CVTs. Chain belt type CVTs have a metal chain belt and two pairs of pulleys. Chain belt has rocker pins and link plates, while pulleys have movable sheave and fixed sheave. Thrust forces are applied to both pulleys, and traction is generated by frictional force between a chain belt and pulleys. Rocker pins are subjected to a compressive force due to thrust force from pulley sheaves, while link plates are subjected to a tensile force due to belt tension. Chain belt type CVTs transmit power by means of friction between the chain belt and pulley sheaves and relating with the difference in tensions. Winding radius can be small in pulley groove since chain belt is multibody and has high flexibility. However, it is inevitable that the chain belt is wrapped in polygonal shape in pulley grooves, because the unit length of the base component of the chain belt is relatively large, of which the unit length is coincident with distance between two pins. The length of chain belt required in pulley groove is reduced when chain belt is wrapped in polygonal shape, and chain belt slips in the circumferential
direction. Energy loss should be caused by this polygonal winding slip, and this loss have not been discussed clearly yet. The objective of this study is to investigate the influence of frictional loss due to polygonal winding slip on the gross energy loss of chain type CVTs. In this study, commercially available chain belt and its components were prepared to investigate three types of energy losses. The gross energy loss of the CVT was measured on the test bench, at first. For calculating frictional loss between rocker pins and link plates, tensile forces of the link plate was measured by strain gauges, and transparent sheave made with acryl resin was used for driving and driven pulley for observing change the rotational angle of the rocker pin. Compressive force of the rocker pins was measured by strain gauges attached on the leading and following pin, respectively for calculating frictional loss between rocker pins and pulley sheaves.

2. Test method

2.1. Test belt and testing system
Commercially available chain belt and its components were prepared to investigate three types of energy losses. Fig.1 shows the chain belt and its components. The numbers of rocker pins and that of link plates were 150 and 787, respectively. Overall length of the chain belt was 720mm. Leading and following rocker pin were inserted into the hole of a link plate. Two pins of the one set of pins were countered in the hole of link plate as shown in the Fig.2. That means that the number of sets of pins inserted into a link plate was two.

![Figure 1. Testing belt and its components.](image1)

![Figure 2. Definition of leading and following pin.](image2)

Fig.3 shows the illustration of our testing system. Test bench with testing system was also prepared to measure the rotational speed and transmitting torque. Pulley thrust force was applied to a chain belt by the oil pressure. The generated mechanical power on AC motor was transmitted from driving axis to driven axis through the CVT unit. The rotational speed and transmitting torque were simultaneously measured by conventional tachometers and torque meters, respectively. Driven torque was applied by
rotational frictional disc brake. CVT fluid was supplied to prevent wear of a circulated chain belt and pulleys during the test.

![Figure 3. Schematic view of testing system.](image)

2.2. Measurement of gross energy loss.

The gross energy loss of the CVT unit was measured on the test bench. Table 1 shows the test conditions for this test. Speed ratio was defined as equation (1).

\[ i_0 = \frac{N_{DR}}{N_{DN}} \]  

(1)

where, \( N_{DR} \) and \( N_{DN} \) denote the rotational speed of the driving shaft and that of the driven shaft, respectively. Initial speed ratio under no loading condition \( i_0 \) was set to 2.48. The pulley thrust force of the driven pulley \( Q_{DN} \) was kept constant at 32 kN, while driving torque \( T_{DR} \) was 12 Nm, 58 Nm, and 135 Nm, respectively. The displacement of movable sheave of driving pulley was constrained with the pulley-stopper to keep the speed ratio constant during the test. Power loss \( H \) was measured at test bench and calculated by equation (2).

\[ H = \frac{2\pi}{60} (T_{DR} N_{DR} - T_{DN} N_{DN}) \]  

(2)

where \( T_{DR} \) denotes driven torque. Energy loss per circulation of CVT unit \( E \) was also calculated by equation (3).

\[ E = H \frac{60L}{2\pi R_{DN} N_{DN}} \]  

(3)

where \( L \) denotes the overall length of the chain belt.

| Table 1. Test condition for measuring energy loss. |
|---------------------------------------------------|
| Initial driving rotational speed \( N_{DR} \) [rpm] | 600  |
| Distance between driving and driven shafts \( C \) [mm] | 173  |
| Pulley thrust force of driven pulley \( Q_{DN} \) [kN] | 32   |
| Pulley thrust force of driving pulley \( Q_{DR} \) [kN] | Resultant reaction force |
| Initial speed ratio \( i_0 \) [-] | 2.48 |
| Driving torque \( T_{DR} \) [Nm] | 12, 58, 135 |
2.3. Measurement of tensile force of link plate.

Tensile forces of the link plate were measured by strain gauges. Fig. 4 shows the locations of strain gauges set on the surfaces of link plate. Four strain gauges were attached on inner and outer part on both side surfaces of a specific link plate to cancel the bending strain. Here, tensile strain $\varepsilon_t$ was calculated by equation (4).

$$\varepsilon_t = \frac{\varepsilon_1 + \varepsilon_2}{2}$$  \hspace{1cm} (4)

where, $\varepsilon_1$ and $\varepsilon_2$ denotes the strain on each side. Table 2 shows the test conditions for this test.

**Table 2. Test condition for measuring tensile force of link plate.**

| Initial driving rotational speed $N_{DR}$ [rpm] | 50 |
|------------------------------------------------|----|
| Distance between driving and driven shafts C [mm] | 173 |
| Pulley thrust force of driven pulley $Q_{DN}$ [kN] | 32 |
| Pulley thrust force of driving pulley $Q_{DR}$ [kN] | Resultant reaction force |
| Initial speed ratio $i_0$ [-] | 2.48 |
| Driving torque $T_{DR}$ [Nm] | 12, 58, 135 |

Figure 4. Locations of strain gauges set on surface of link plate.

2.4. Measurement of rotation angle of rocker pin

Fig. 5 shows definition of rotation angle $\alpha$ of the rocker pin. In this study, transparent sheave made with acryl resin was used for driving and driven pulley for observing change of the rotational angle of the rocker pin, alternatively. The pulley groove width was adjusted so that speed ratio was 2.48. Rotational speeds of driving and driven pulley were manually adjusted under no torque condition. The displacements of movable sheave of driving and driven pulley were constrained with the pulley-stopper. Fig. 6 shows examples of observed rotations of the rocker pins in driving and driven transparent pulley groove. Change of the rotational angles of rocker pins were observed at every ten degrees of the angle from entrance to exit of each pulley.
Fig. 5. Definition of rotation angle of the rocker pin.

Fig. 6. Observed rotations of pins in driving and driven pulley groove.

2.5. Measurement of compressive force of rocker pin.
Compressive forces of the rocker pin was measured by strain gauges attached on the leading and following pin, respectively. Conventional steel pulley was used for this test. Fig.7 shows the locations of strain gauges set on the thin milled faces of the rocker pin. Two strain gauges were attached on both faces of the edge of each rocker pin in the width direction to cancel the bending strain. Test conditions were followed as same with those on Table 3. Here, compressive strain $\varepsilon_c$ was also calculated by equation (4).

Fig. 7. Locations of strain gauges set on surface of rocker pin.
3. Result and discussion

3.1. Frictional loss between rocker pins and link plates.

Fig. 8 shows the change of tensile force of link plate with respect to the non-dimensional location of the link plate, when the driving torque $T_{DR}$ was 58 Nm. In this test, non-dimensional location in a circulation was defined as the passing distance divided by the belt length. The notation DR and DN denote at driving and driven pulley groove, respectively.

![Graph of Fig. 8](image)

(a) Outer part.  
(b) Inner part.

**Figure 8.** Example data of changes of tensile forces of the link plate.

Fig. 9 shows the change of the rotation angle of rocker pins with respect to non-dimensional location of rocker pin at pulley groove. In this test, non-dimensional location in pulley groove was defined as the passing distance from the entrance to exit divided by the belt length in each pulley groove. Rotation angle of rocker pin were about 10.7 deg and 8.81 deg in the driving and driven pulley groove in average, while those are almost 0 when the belt was traveled during strings.

![Graph of Fig. 9](image)

(a) Driving pulley.  
(b) Driven pulley.

**Figure 9.** Change of rotation angle of rocker pin with respect to non-dimensional location of rocker pin.
The frictional loss between rocker pins and link plates per circulation \( E_{pl} \) was calculated by equation (5).

\[
E_{pl} = n \mu_{pl} (\alpha_{DR} + \alpha_{DN}) \frac{h}{2} \sum (T_{i,tight} + T_{i,slack})
\]

where \( n \), \( h \), \( T_{i,tight} \) and \( T_{i,slack} \) denote the numbers of rocker pin of a chain belt, height of the rocker pin, the tight part tension of link plate and the slack part tension of link plate, respectively. In the calculation, sliding distances between rocker pin and link plate was determined by multiplying half height of the rocker pin by the rotation angle. It was assumed that mechanical energy was lost at only entrance and exit of driving and driven pulley for four times during a circulation by the rolling frictions between rocker pins and link plates, and rolling frictional coefficient \( \mu_{pl} \) was assumed to be \( 2.5 \times 10^{-3} \). Fig.10 shows the change of gross energy loss measured at test bench \( E \) and calculated frictional loss between rocker pins and link plates \( E_{pl} \) with respect to driving torque \( T_{DR} \). It was found that frictional losses between rocker pins and link plates was very low compared with gross energy loss \( E \), while the gross energy loss \( E \) was increased with increasing driving torque.

**Figure 10.** Change of gross energy loss measured at test bench \( E \) and calculated frictional loss between rocker pins and link plates \( E_{pl} \) with respect to driving torque \( T_{DR} \).

### 3.2. Frictional loss between rocker pins and pulley sheaves

Fig.11 shows the change of the compressive force of the rocker pin with respect to the non-dimensional location of the rocker pin, when the driving torque \( T_{DR} \) was 58 Nm.
Figure 11. Example data of changes of compressive forces of the rocker pin with respect to non-dimensional location of rocker pin.

Pitch radius of chain belt at driving and driven pulley $R_{DR}$, $R_{DN}$ were calculated by equation (6) and (7), respectively.

$$ R_{DR} = \frac{-C\pi(i + 1)^2 + \sqrt{(C\pi(i + 1)^2 - 4C(i - 1)^2(L - 2C)^2)}}{2(i - 1)^2} $$

(6)

$$ R_{DN} = iR_{DR} $$

(7)

where $L$ denotes the length of the chain belt. Contact arc of the chain belt at driving and driven pulley $\theta_{DR}$, $\theta_{DN}$ were calculated by equation (8) and (9), respectively.

$$ \theta_{DR} = \pi - 2(R_{DN} - R_{DR})C $$

(8)

$$ \theta_{DN} = \pi + 2(R_{DN} - R_{DR})C $$

(9)

Numbers of the rocker pins sets at driving and driven pulley groove $N_p$ was calculated by equation (10).

$$ N_p = \frac{nL}{R\theta} $$

(10)

Numbers of pairs of rocker pin at driving and driven pulley calculated by equation (10) were about 8 and about 31, and then, chain belt was wound around the driving and driven pulley in octagon and 31-square shape. The frictional loss between rocker pins and pulley sheaves due to polygonal winding slip is related with the difference between the length for polygonal wrapping and that of perfect circle in driving and driven pulley groove. The slip distances $s$ due to polygonal winding was calculated by equation (11).

$$ s = (R\theta - NL) $$

(11)

where $l$ denotes the length of the pitch between rocker pins. The frictional loss between rocker pins and pulley sheaves per circulation $E_{pp}$ was calculated by equation (12).

$$ E_{pp} = 2n\mu_p(s_{DR} + s_{DN})\sum(P_{Leading\ pin} + P_{Following\ pin}) $$

(12)

where $P_{Leading\ pin}$ and $P_{Following\ pin}$ denote the compressive force of leading pin and that of following pin, respectively. Frictional coefficient between rocker pin and pulley sheave $\mu_p$ was assumed to be 0.1. Fig.12 shows the change of the gross energy loss measured at test bench $E$ and calculated frictional loss between rocker pins and pulley sheaves $E_{pp}$ with respect to driving torque $T_{DR}$. The frictional loss between rocker pins and pulley sheaves was relatively large compared with gross energy loss of CVT especially when driving torque was decreased, while the frictional loss was increased with increasing driving torque. The rate of the frictional losses were about 26.2 %, 22.0 %, and 20.3 % of the gross energy when driving torques were 12 Nm, 58 Nm, and 135 Nm, respectively. The experimental data of the frictional losses were linearly approximated and extrapolated. This study found that remarkable about 26.5 % was remained as inevitable frictional loss due to polygonal winding slip of the current CVT.

4. Conclusion

(1) The frictional losses between rocker pins and link plates was very low compared with gross energy loss.

(2) The frictional losses between rocker pins and pulley sheaves was relatively large compared with gross energy loss of CVT especially when driving torque was decreased, while the frictional loss was increased with increasing driving torque.
This study found that remarkable about 26.5% was remained as inevitable frictional loss due to polygonal winding slip of the current CVT.

![Figure 12. Change of gross energy loss measured at test bench $E$ and calculated frictional loss between rocker pins and pulley sheaves $E_{\text{p}}$ with respect to driving torque $T_{\text{DR}}$.](image)

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