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Predicting the Life Contact for Half Toroidal Continuously Variable Transmission

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Abstract: A contact fatigue life analysis for half toroidal Continuously Variable Transmission (CVT) has been developed which was based on a Lundberg-Palmgren theory. The analysis was used to predict the contact life of half toroidal CVT drive. Considering the elastic deformation of the contact points between the rolling elements. The formulas for calculation of the contact Hertz parameters between the rolling elements are introduced based on Hertz contact theory. By using the above-mentioned formulas, the life contact predication for the half toroidal CVT drives are obtained. Parametric results were compared for life as a function of speed ratio for different system parameters in the form of graphs. These graphs give useful information for designer in manufacture and design of such drives during its operation.

Key words: Half toroidal, elliptical contact, fatigue life, system parameters

INTRODUCTION

The traction Continuously Variable Transmission (CVT) drives has continued to be an object of considerable research interest within the mechanical design community, driven primarily by automotive industry's demands for more energy efficient and environmentally friendlier vehicles. An overview of the historical background of the Continuously Variable Transmission (CVT) have been introduced[1,2], among them, primarily two types that are of interest in the automotive area, viz., half toroidal traction drives and full-toroidal traction drives. The different characteristic in these systems is the shape of cavities in the power transmitting elements[3,4].

The half toroidal CVT transmits torque by means of shear resistance of the traction fluid film taking advantage of the property of the fluid solidification under high contact pressure[5]. The performance of the drive depends to a large extent on the maximum value of Hertzian stress in the elliptical contact area, where both design parameters and operating conditions must be take into consideration such a contact. The stress calculations of the contacts of the toroidal CVT for fatigue life analysis were performed by using Hamrock's method[6]. The stress-cycle data on traction rolling element of half toroidal CVT was proved that the extremely purified steel and carbonitriding heat treatment contributed for longer life under high contact pressure[7]. In this study, the formulation of Hertz parameters for half toroidal CVT contacts specifically for point contact. By using the mentioned formulation, the fatigue life predications for the drive are obtained based on a Lundberg-Palmgren theory. In addition, a parametric study on the contact life for the main system parameters is compared.

BASIC GEOMETRY AND SPEED RATIO

The heart of a half toroidal transmission is the variator, the basic function of which is shown in Fig. 1.

The power is transmitted from an input disk through a power roller to an output disk. Both the input dusk and the output toroidal disks rotate about AA′. The power rollers rotates about BB′ and the speed ratio is altered continuously by varying the angle ϕ of the rotated axis of the power roller, which changes the contact rolling circle R_1 radii of the input disk and R_3 of the output disk. The basic geometry of a half toroidal CVT are determined by four design parameters which are shown in Fig. 1, i.e. the cavity radius r_0, the aspect ratio of cavity, $k_0 = e_0/r_0$, the

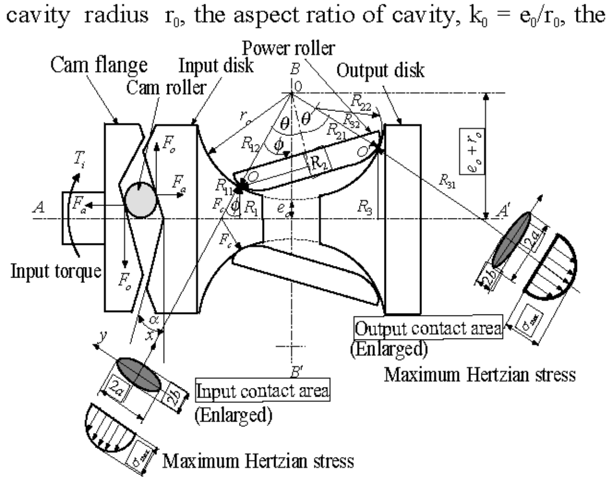

Fig. 1: Loading cam, basic geometry and principle curvatures of the variator of half toroidal CVT

half cone angle of the power roller, θ and the profile radius of the power roller R_{22}.

The working rolling radii can be defined as follows:

$$R_1 = r_0\,(1+k_0-\cos\phi) \tag{1}$$

$$R_2 = r_0 \sin\theta \tag{2}$$

$$R_3 = r_0\,(1+k_0-\cos(2\theta-\phi)) \tag{3}$$

where, e_0 is the radial distance.

The theoretical speed ratio e_s, solely depends on the swing angle of the power roller θ and is represented by:

$$e_s = \frac{\omega_3}{\omega_1} = \frac{R_1}{R_3} = \frac{1+k_o-\cos\phi}{1+k_o-\cos(2\theta-\phi)} \tag{4}$$

LOADING GEBERATION BY LOADING CAM

To generate the loading force, Machida employs a loading cam for a half toroidal CVT, which is used cam rollers constrained in wedges to create axial load[3]. The schematic diagram in Fig. 1 shows the loading cam of a half toroidal CVT with α is the cam lead angle, F_a and F_0 are the axial and tangential (driving force of the cam flange) components of the resulting force applied at the cam rollers, L_c is the cam lead which is equal to $2\pi R \tan\alpha$, R is the profile pitch and μ is the coefficient of friction between the cam surface and cam rollers[8].

When a input torque T_i is applied to the cam flange which is equal to F_0R, the inclined cam face produces an axial force component that presented in terms of input torque and the loading cam parameters as:

$$F_a = \frac{2\pi T_i(\cos\alpha-\mu\sin\alpha)\tan\alpha}{L_c(\sin\alpha+\mu\cos\alpha)} \tag{5}$$

If the friction is neglected, i.e., then the equation of the axial load simplifies to:

$$F_a = 2\pi T_i/L_c \tag{6}$$

The axial load in Eq. 6 is directly applied on the input disk. If there are n power rollers in a cavity, the normal contact load is calculated as:

$$F_c = F_a/n\sin\theta \tag{7}$$

The traction force on the input disk is:

$$F_t = T_i/nR_1 \tag{8}$$

The traction coefficient μ_c at the contact point is:

$$\mu_c = F_t/F_c = (Lc \sin\phi)/(2\pi r_0(1+k_0-\cos\phi)) \tag{9}$$

As seen from Eq. 9 in which ϕ is contained, the effective traction coefficient, which depends on the loading cam, varies with the speed ratio. The rotation angle of the power roller ϕ_m can be determined when the effective traction coefficient is at a maximum and is given by:

$$d\mu_c/d\phi = 0 \tag{10}$$

and there by yields the following equation:

$$\phi_m = \cos^{-1}(1/(1+k_0)) \tag{11}$$

To avoid gross slippage between the power rollers and the input disk, the effective traction coefficient μ_c must not exceed the available maximum traction coefficient μ_{max} which depends on the properties of the traction oil to be used and can be determined by using twin disk machine[9].

Then, the necessary cam lead is calculated by:

$$Lc = 2\pi r_0\mu_{max}(1+k_0-\cos\phi_m)\;/\sin\phi_m \tag{12}$$

CONTACT PARAMETERS

Contact hertzian parameters: Due to the elastic deformation, the contact patch between the power roller and disks will take the shape of ellipse centered at the points of contact O and O′ as depicted in Fig. 1. The size and the orientation of the contact ellipse depend on the contact force F_c, the material property and the geometry of the power roller and disk surfaces.

By using the principal curvatures of the rotating elements which are given below:

$$R_{11} = R_1/\cos\phi \qquad R_{21} = r_o \qquad R_{31} = R_3/\cos(2\theta-\phi)$$

$$R_{12} = -r_o \qquad R_{22} = K_R \cdot R_{21} \qquad R_{32} = -r_o$$

The first suffix of the principal curvature radius R, 1, 2 or 3 designates the input, power roller or output disk, respectively. The second suffix 1 designates is the principle curvature radius, which is along the profile tangent in the cross section direction. The second suffix 2 designates the principal curvature radius, which is along the profile tangent in the axial direction, this value is negative since the profile in this direction is concave, K_R is the conformity ratio of power roller.

The Hertzian contact parameters for the contact patch (an ellipse) of the half toroidal CVT can be obtained by applying the following equations[10].

$$\sigma_{max} = \frac{1.5F_c}{\pi ab} \quad (13)$$

$$b = \beta\left[\frac{3F_c}{4(B+A)}\left(\frac{1-\nu_1^2}{E_1}+\frac{1-\nu_2^2}{E_2}\right)\right]^{1/3} \quad (14)$$

$$a = b/k \quad (15)$$

where, a and b are the half-length of the long and short axes, with a in the crosswise rolling direction and b in the traction rolling direction; v_1 and v_2 are the Poisson ratios of the disk and power roller materials respectively; E_1 and E_2 are the Young's moduli of the disk and roller materials.

The quantity (B+A) and (B-A) related to the surface principal curvatures of the disk and the power roller, can be determined by the following equations

$$B + A = 0.5\times(1/R_{12} + 1/R_{11} + 1/R_{22} + 1/R_{21}), \quad (16)$$

$$B - A = 0.5\left[(1/R_{12} - 1/R_{11})^2 + (1/R_{22} - 1/R_{21})^2 + 2(1/R_{12} - 1/R_{11})\cdot(1/R_{22} - 1/R_{21})\cos 2\psi\right]^{1/2} \quad (17)$$

In the case of half toroidal CVT, the curvatures $1/R_{11}$ and $1/R_{21}$ are in the same plane, i.e., Ψ equals to zero. The two coefficients β and k depend on the surface principal curvatures of the disk and the power roller can be approximated with enough accuracy by linear relations with respect to (B-A)/(B+A).

$$k = 0.95 - 0.89\left[(B-A)/(B+A)\right] \quad (18)$$

$$\beta = 1.02 - 0.60\left[(B-A)/(B+A)\right] \quad (19)$$

where, the angle ϕ changes from ϕ = 33°~87° i.e., from e_{min} to e_{max}.

Maximum shear stresses: The maximum reversing orthogonal shear stress τ_0 occurs at a depth z_0 underneath the surface of contact is given[6], as follows:

$$\tau_o = \frac{\sqrt{2t-1}}{2t(t+1)}\sigma_{max} \quad (20)$$

$$z_o = \frac{1}{(t+1)\sqrt{2t-1}}b \quad (21)$$

where, t is an auxiliary parameter which is related to the elliptic parameter K, which is equal to a/b.

$$K = \left[(t^2-1)(2t-1)\right]^{-1/2} \quad (22)$$

LIFE CONTACT PREDICATION METHOD

In the high contact stress regime, which is the only practical regime in which to operate most traction drives, the achievement of a given number of stress repetitions without failure can only be assigned a probability. The Lundberg-Palmgren method has been used successfully to predict the life of ball bearings. The toroidal CVT drives have elements that resemble parts of large ball bearings, so the transfer between disciplines is simple and natural. By applying the Lundberg Palmgren theory to rotating elements of half toroidal CVT drives[2]. The process begins with the calculations of the Hertz contact parameters, as has been demonstrated above.

The life, millions of revolutions calculation takes the form:

$$L = K_4 \cdot K_2^{0.9} \cdot F_c^{-3} \cdot \rho^{-6.3} \cdot R^{-0.9} \quad (23)$$

$$K_2 = (z_o/b)^{4/3} \cdot (\tau_o/\sigma_{max})^{-31/3} \cdot (a^*)^{28/3} \cdot (b^*)^{35/3} \quad (24)$$

$$a^* = \frac{a}{0.0045\sqrt[3]{F_c/\rho}} \quad (25)$$

$$b^* = \frac{b}{0.0045\sqrt[3]{F_c/\rho}} \quad (26)$$

$$\rho = 1/R_{12} + 1/R_{11} + 1/R_{22} + 1/R_{21}$$
$$K_4 = 6.43\times 10^8 \quad (27)$$

where, z_0/b is the ratio of the depth to the maximum orthogonal shear stress to the semi-minor diameter of the contact ellipse τ_0/σ_{max} is the ratio of the value of the maximum orthogonal shear stress to the maximum Hertz contact stress ρ is the inverse curvature sum R is the rolling radius of the input contact path.

NUMERICAL RESULTS AND DISCUSSIONS

The analytical results obtained for the life calculation depends on the Hertzian contact parameters and the maximum reversing orthogonal shear stress, which occurs at a certain depth underneath the surface of contact. The numerical results can be evaluated for the most important influence variables (input torque T_i, material property of rolling elements E, half cone angle of power roller θ, number of power roller n and maximum traction coefficient, μ_{max}), which are varied to have clear merits of the life of contact.

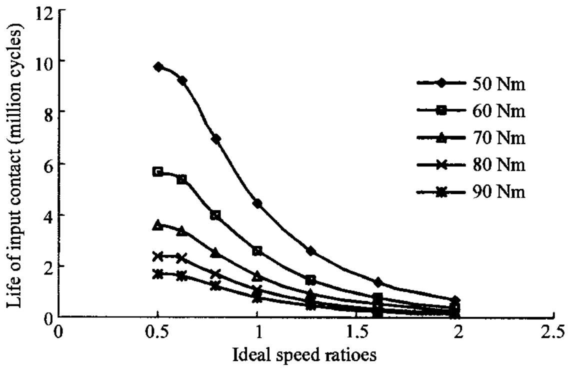

Fig. 2: Life of input contact as a function of ideal speed ratio for different input torques; $k_0 = 0.625$, $K_R = 0.8$, $n = 2$, $\mu_{max} = 0.095$, $\theta = 60°$, $E_1 = E_2 = 210GPa$

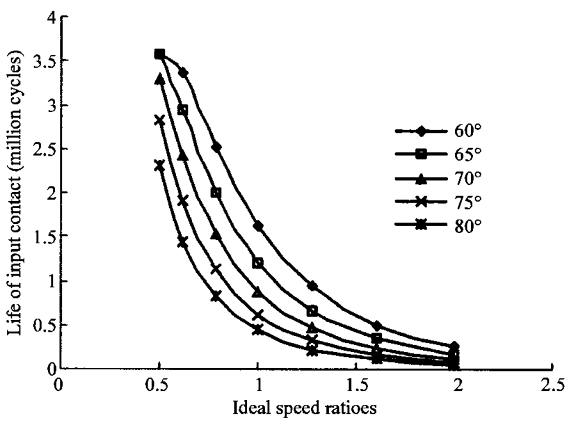

Fig. 3: Life of input contact as a function of ideal speed ratio for different half cone angles of power roller; $k_0 = 0.625$, $K_R = 0.8$, $n = 2$, $\mu_{max} = 0.095$, $T_i = 70$ Nm, $E_1 = E_2 = 210GPa$

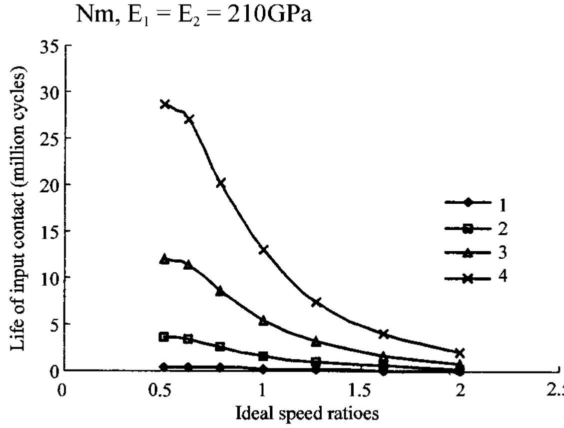

Fig. 4: Life of input contact as a function of ideal speed ratio for different number of power roller; $k_0 = 0.625$, $K_R = 0.8$, $\theta = 60°$, $\mu_{max} = 0.095$, $T_i = 70$ Nm, $E_1 = E_2 = 210GPa$

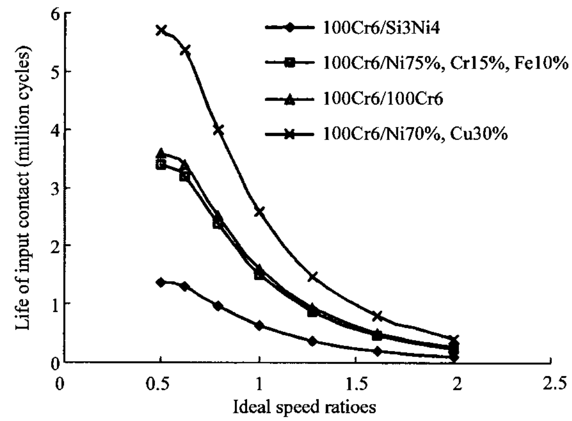

Fig. 5: Life of input contact as a function of ideal speed ratio for different combination of material; $k_0 = 0.625$, $K_R = 0.8$, $n = 2$, $\mu_{max} = 0.095$, $\theta = 60°$, $T_i = 70$ Nm

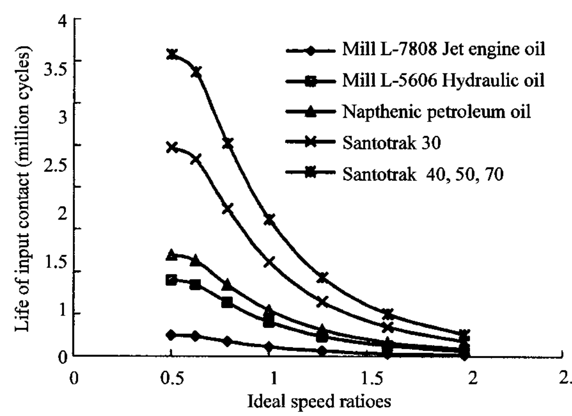

Fig. 6: Life of input contact as a function of ideal speed ratio for different maximum traction coefficients; $k_0 = 0.625$, $K_R = 0.8$, $n = 2$, $T_i = 70$ Nm, $E_1 = E_2 = 210GPa$

The comparative graphical presentations of the numerical results are shown in Fig. 2-6.

Effect of the input torque: Referring to Fig. 2, the input contact life varies non-linearly with the ideal speed ratio for different input torques. It can be noticed that the contact life decreases with any increase in the input torque and increases in the ideal speed ratio. The reason is that an increase in the input torque leads to an increase in the generated contact load (direct proportional). Consequently, high maximum Hertzian stress is expected

In fact, if high cyclic compressive loads are applied, life tends to be decreased due to the increase rate of the subsurface fatigue (flaking), while low values of

Table 1: Properties of four material combinations

Combinations	Properties
100Cr6/Si3Ni4	$E_1 = 210GPa$, $E_2 = 315GPa$ $v_1 = 0.3$, $v_2 = 0.26$
100Cr6/Ni75%, Cr 14%, Fe 10%	$E_1 = 210GPa$, $E_2 = 214GPa$ $v_1 = 0.3$, $v_2 = 0.306$
100Cr6/100Cr6	$E_1 = E_2 = 210GPa$ $v_1 = v_2 = 0.306$
100Cr6/Ni70 %, Cu 30 %	$E_1 = 210GPa$, $E_2 = 184GPa$ $v_1 = 0.3$, $v_2 = 0.239$

Table 2: Maximum traction coefficients by Monsanto

Lubricant	Maximum traction coefficient
Mill L-7808 Jet Oil	0.039
Mill H-5606 Hydraulic Oil	0.060
Napthenic Petroleum Oil	0.066
Santotrac 30	0.084
Santotrac 40,50,70	0.095

compressive loads are not sufficient to generate proper traction forces and consequently high torque capacity cannot be attained.

Effect of the half cone angle of power roller: The effects of the half cone angle of the power roller on the input contact life are casted in nonlinear relationships as shown in Fig. 3. For low values of the half power roller cone angle, the life of input contact increases due to the small resultant traction force. However, making the half cone angle of power roller smaller would naturally increase the speed ratio range which causing problems with the supporting bearings.

Effect of the number of power roller: It can be demonstrated that the input contact life decreases with any increase in the ideal speed ratio or by using small number of power rollers due to the increase in the generated traction force (Fig. 4). This owing to the contact compressive load is inversely proportional with the number of power roller. However, if the number of power rollers is too large, would naturally effective to increase the transmitted torque but the suspension system is very complex (the synchronizing between them cannot be obtained easily) and the speed ratio range will decease drastically[11].

Effect of different combinations of materials: Figure 5 presents the typical results concerning the variation of life of the input contact for different ideal speed ratios and for the four material combinations, whose properties are listed in Table 1. Nonlinear relations are obtained which can help the designer in evaluating the cyclic input contact life. It can be observed that the first combination has a lower contact life compared with the other combinations due to the higher maximum Hertzian stress. Because of the material of power roller is the stiffest one (higher modulus of elasticity).

Effect of the maximum traction coefficient: Figure 6 present the relationships of the life of input contact and ideal speed ratio for different traction lubricants, whose values of maximum traction coefficient are listed in Table 2, as given by Monsanto[9]. It can be pronounced that Mill L-7808 Jet Oil (low values of the maximum traction coefficient) will leads to a low values of the life of input contact due to an increase in the applied contact load. This increasing in the contact load is related to severe rubbing action between the contacting surfaces, which leads to a high rate of wear and consequently a shorter service life.

However, on the half toroidal CVT, the life of contact is composed of five important parameters E, T_i, k_0, θ and μ_{max}. The basis for the design of the half toroidal CVT is to optimize the contact life with material, operating and geometrical parameters efficiently.

CONCLUSIONS

From the analysis presented in this study, the following conclusions can be drawn:

1. The most severe service condition where the minimum contact life appears at the maximum speed ratio for any given parameter.
2. Useful graphs in nonlinear relationships are observed for material, operating and geometrical parameters, which help the designer to predict the contact life within the speed ratio range in such systems.
3. The different system parameters play the major role in the life prediction in the contacts of the half toroidal CVT.

For systems with low values of the input torque and half cone angle of power roller, the life contact showed higher values than those having higher input torques and half cone angles of power roller. On the other hand, the life of contact show lower values for systems with low values of number of power roller and maximum traction coefficient.

In addition, the selection of the combination material with a moderate modulus of elasticity is preferable which predict a reasonable contact life to avoid fatigue failure.

REFERENCES

1. Zhongtang, R. *et al.*, 1999. Design and Selection Guidelines of Mechanical Continuously Variable Transmission. Beijing: Published by Chemical Industry Press, pp: 2-10 (In Chinese).

2. Frederick, W. III. *et al.*, 1983. Traction Drives: Selection and Application. New York: Published by Marcel Dekker Inc., pp: 28-119.
3. Imanishi, T., H. Machida and H. Tanaka, 1996. A geometrical Study of Toroidal CVT, Comparison between Half Toroidals and Full Toroidals. SAE Paper No. 9636411:107-111.
4. Imanishi, T. and H. Machida, 2001. Development of powertoros unit half-toroidal CVT (2): Comparison between half-toroidal and full-toroidal CVTs. NSK Technical J., 10: 1-8.
5. Kumura, H., J. Sugihara, Y. Nartita, Y. Arakawa, M. Nakano and N. Maruyama, 1999. Development of a Dual-Cavity Half-Toroidal CVT. In: Proc. CVT99. Eidnhoven, pp: 65-70.
6. Coy, J.J., S.H. Loewenthal and E.V. Zartsky, 1975. Fatigue life analysis for traction drives with application to a toroidal type geometry. Nasa Technical Note, D-8362, pp: 1-31.
7. Machida, H. and T. Abe, 1996. Fatigue life analysis of a traction drive CVT. SAE Paper No. 9636402, pp: 101-106.
8. Yamamoto, T., T. Osidari and M. Nakano, 2003. Improvement of Loading Cam Performance in a Toroidal CVT. JSAE Rev., 23: 481-487.
9. Machida, H., 1999. Traction drive CVT up to date. In: Proc. CVT99. Eidnhoven, pp: 71-76.
10. Nabil, A.A., Q. Datong, S. Wankai and L. Huaying, 2003. A parametric study on the contact stress of half toroidal continuously variable transmission. J. Chongqing University-Eng. Ed., 2: 6-12.
11. Imanish, T., H. Machida and H. Tanaka, 2001. Development of High Power-Density Double Cavity Half-Toroidal CVT. In: Proc. JSME (MPT'2001), Fukuoka, pp: 860-863.