

The Design of a Milner CVT using Simulation Based Design of Experiments

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ABSTRACT

The Milner continuously variable transmission (MCVT) is a traction drive transmission based on rolling contacts analogous to those found in angular contact roller bearings. The MCVT is applicable to very high power density applications or those with demanding packaging constraints, due primarily to the existence of multiple traction contacts on each of the rolling elements.

The transmission has been modelled analytically and then Design of Experiments (DoE) techniques have been applied to create response surface model sets describing performance criteria over a wide range of design input factors. The response surface models have then been interrogated to find optimal solutions. The results show that it is possible to create a transmission with a significantly higher ratio range (10:1), compared with 5:1 that is typical for existing designs [1]. The results also indicated that the traction patch spin factor could also be reduced over the existing designs and it is hypothesised that this should lead to an increase in efficiency.

Finally, research into the stresses set up by the rolling contact indicated that stress is highest at the greatest reductions in ratio and that maximum stress appears to be proportional to the ratio range of the transmission. Hence a design optimisation tool has been developed allowing engineers to rapidly optimise geometries of the MCVT to achieve desired trade-offs between ratio range, ratio position, transmission life/durability and transmission efficiency.

INTRODUCTION

The work presented in this paper describes ongoing research in to the performance optimization of the MCVT. In this work design of experiments techniques have been applied to a validated analytical model [3] of the MCVT to investigate its performance over a wide range of geometry variations, sizings and torque demands. Response surface models (RSMs) for transmission performance responses have been developed and reviewed. The use of these RSMs in a design optimization process is then explained, along with some example optimization outputs.

THE MILNER CVT

The Milner CVT is a rolling traction drive transmission consisting of 3 independent rolling elements: the inner races; the outer races and the planet/ carrier assembly. It is analogous to both an angular contact roller bearing of variable geometry and to an epicyclic transmission with variable ratio elements for the sun, planet and annulus gears.

The Milner CVT is shown schematically in Figure 1. Primarily it consists of two outer race halves, two inner race halves, and a number of spherical planets (typically 3), which drive a carrier assembly via planet followers.

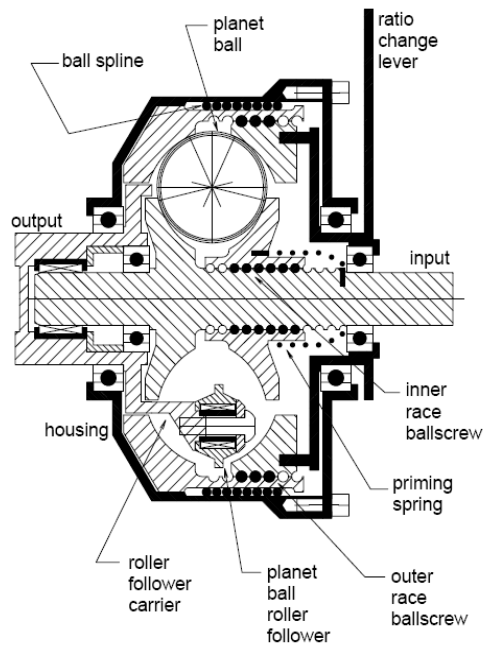
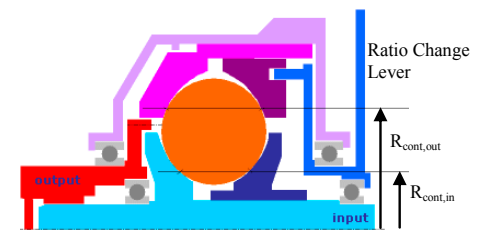
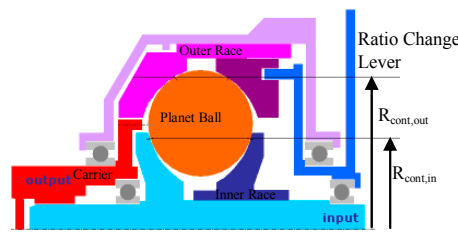
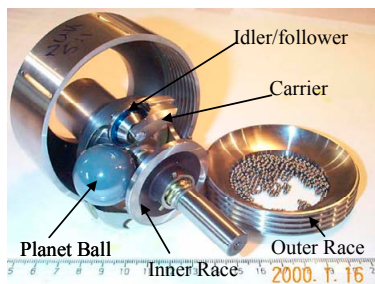


Figure 1 MCVT Arrangement Schematic

The ratio of the MCVT is adjusted by moving the outer race halves axially, a motion achieved in this instance using an outer ballscrew to translate the rotational shift motion into axial separation. The planets move radially to vary the point of contact between the outer races and

planets. Conversely if the outer races are moved towards each other the planets are driven radially into the inner race. The inner race assembly is shown in this configuration with a ballscrew assembly that adjusts the clamping load on the planets in proportion to the torque being applied to the input shaft. The ratio of torque applied to normal force is essentially a function of the geometry of the inner ballscrew. Similarly the geometry of the outer ballscrew can be optimised in relation to the traction forces on the outer race such that the shift actuator force can be minimised. Figure 2 shows a photographic view of the disassembled 12kW MCVT described in [1]. Along with two schematic images of the MCVT at low and high ratio extremes.

The MCVT is designed as a rolling traction device, as such the system transmits traction through the contact patches that form between the planet balls and the inner and outer race. In these small contact patches traction fluid is squeezed to pressures of the order of 1-3GPa. This increase in pressure causes an exponential increase in viscosity of the traction fluid such that it behaves as an elastic solid. Thus traction is forces are transmitted at a slip rate that is typically less than 3% of the mean speed through the contact patch. A detailed review of current traction drive technology and the modeling of traction drives is given in [2].



(a) (b) (c)
Figure 2 The Milner CVT, (a) Photograph of the Milner CVT, (b) MCVT Schematic- Low Ratio (High Gear), (c) MCVT Schematic- High Ratio (Low Gear)

MODELLING THE MCVT

The MCVT has been modeled analytically using MATLAB, and validated with a range of experimental data. For brevity only a summary of the analytical modeling work is contained in this paper, further details of the models may be found in [3, 4 & 5].

MCVT GEOMETRY

The basic geometry of the inner and outer races and the planet balls is described by five variables. The radius of the planet balls (R_{planet}), the radius of the inner datum (R_{inner_datum}), the radius of the outer datum (R_{outer_datum}), the inner radius (curvature of the inner race, R_{inner}) and the outer radius (curvature of the outer race, R_{outer}). These variables are shown schematically in Figure 3.

Given an outer race contact angle θ , then the following geometry can be solved

$$R_{planet,out} = R_{planet} \cos \theta$$

and hence the outer contact radius can be defined as

$$R_{cont,out} = R_{out,datum} + R_{outer} \cos \theta$$

So the orbiting radius of the planet ball will be

$$R_{orbit} = R_{cont,out} - R_{planet} \cos \theta$$

Similarly simultaneous equations can be used to solve the contact angle at the inner race

$$\cos \phi = \frac{R_{in,datum} - R_{orbit}}{R_{inner} - R_{planet}}$$

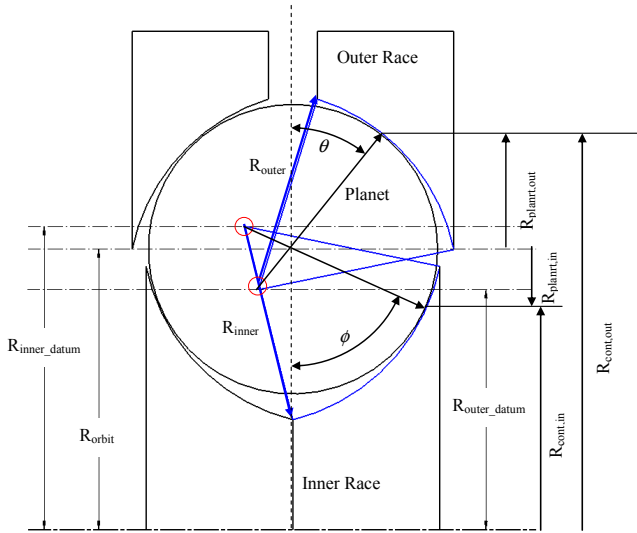


Figure 3 MCVT Major geometric values

The rolling radius of the planet ball relative to the inner race can be defined as

$$R_{planet,in} = R_{planet} \cos \phi$$

In the absence of slip the transmission ratio is defined as

$$\frac{\omega_{input}}{\omega_{output}} = \frac{\omega_{input}}{\omega_{orbit}} = \frac{R_{cont,in} R_{planet,out} + R_{cont,out} R_{planet,in}}{R_{cont,in} R_{planet,out}}$$

TORQUE EFFECTS

For a given input torque on the transmission the traction force at the inner contact patch is given by

$$Ft_{in} = Tq_{in} / (R_{cont,in} 2N_{planets})$$

Assuming an ideal clamping system exists to control the traction coefficient to a desired value then

$$N_{in} = Ft_{in} / \mu_{des}$$

Similarly the normal force at the outer contact becomes

$$N_{out} = \frac{N_{in} \cos \phi + 0.5 M_{planet} R_{orbit} \omega_{orbit}^2}{\cos \theta}$$

When the centripetal force of the orbiting planets is taken in to account.

CONTACT CONDITIONS

The contact conditions are crucial in modelling the traction conditions at each contact patch and consequently any losses that occur. The Matlab model assumes that the pressure distribution within each contact patch is purely Hertzian [7-9]. Each contact patch is formed by elastic deformation of the surfaces of the planet ball and the races, the level of deformation and hence the size of the contact patch is determined by the magnitude of the mean force between the components. The contact patch will be elliptical in shape, the exact shape is dependent on the geometry of the planet and races and the ratio position at anyone time. A more detailed description of the contact mechanics theory can be found in [6 & 7]

The Inner Race to Ball Contact

In the following analysis, convex curves have a positive value of radius and concave curves a negative value. Hence, if component 1 is the planet ball and component 2 is the raceway and their respective radii in the x direction are $r_{x,1,inner}$ and $r_{x,2,inner}$

$$r_{x,1,inner} = R_{planet} \text{ and } r_{x,2,inner} = \frac{R_{cont,in}}{\cos \phi}$$

$$\text{Hence, } R'_{x,inner} = \frac{r_{x,1} \cdot r_{x,2}}{r_{x,1} + r_{x,2}} = \frac{R_{planet} \cdot \frac{R_{cont,in}}{\cos \phi}}{\left(R_{planet} + \frac{R_{cont,in}}{\cos \phi} \right)}$$

Here it is important to note the + sign assigned to $r_{x,2}$. This is because as the ball rolls around the inner race it is rolling around a curve which is convex in the x direction.

$$\text{Rearranging } R'_{x,inner} = \frac{R_{planet} \cdot R_{cont,in}}{(R_{planet} \cos \phi + R_{cont,in})}$$

Similarly in the y direction, $r_{y,1,inner} = R_{planet}$ and $r_{y,2,inner} = -R_{inner}$

$$\text{Hence, } R'_{y,inner} = \frac{R_{planet} \cdot R_{inner}}{R_{inner} - R_{planet}}$$

The Outer Race to Ball Contact

The situation at the outer race to ball contact is similar. The corresponding equations for the outer contacts are

$$r_{x,1,outer} = R_{planet} \text{ and } r_{x,2,outer} = -\frac{R_{cont,out}}{\cos \theta}$$

$$\text{Such that } R'_{x,outer} = \frac{R_{planet} \cdot R_{cont,out}}{(R_{cont,out} - R_{planet} \cos \theta)}$$

$$\text{While in the y direction } r_{y,1,outer} = R_{planet} \text{ and } r_{y,2,outer} = -R_{outer}$$

$$\text{Hence, } R'_{y,outer} = \frac{R_{planet} \cdot R_{outer}}{(R_{outer} - R_{planet})}$$

$$\text{Combining the above equations, } \frac{R'_x}{R'_y} \text{ and hence } a/b$$

may be obtained for the inner and outer contacts respectively. A further expression is now required to take account of the total load N and the material properties of the two contacting bodies. This is

$$a = \left\{ \frac{3N \sqrt{R'_x R'_y \left[\frac{E(e)}{1-e^2} - K(e) \right] [K(e) - E(e)]}}{\pi E' e^2} \right\}^{1/3}$$

Where $b = a\sqrt{1-e^2}$ and where E' is the effective modulus of elasticity of the two contacting bodies, a function of their elastic moduli E_1 and E_2 and their values of Poisson's ratio ν_1 and ν_2 such that

$$E' = \frac{2}{\left(\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2} \right)}$$

The dimensions of each contact may be calculated by assuming a Hertzian distribution of contact pressure. The total force supported by the contact is then

$$N = \frac{2}{3} P_{\max} \pi ab$$

Since the area of the contact patch is πab , the mean contact pressure is

$$P_{\text{mean}} = \frac{2}{3} P_{\max}$$

Conveniently the elliptical integrals $K(e)$ and $E(e)$ from can readily be solved numerically. This was performed in MATLAB as a separate routine to populate a look up table

SPIN

The contact patches are at an angle to the relative rotation of the components, so within each contact patch there will be a spin motion. For example, if the contact angle, Φ , between the inner race and the planet balls was 90° , then there would be purely spin, similarly if the angle is 0° , the motion is purely rolling with no spin, so for any contact angle other than 0° a component of spin will exist. Spin causes power loss and heat generation so has a negative effect on efficiency. It is planned in future developments to extend the spin model to a full efficiency prediction based on an elastohydrodynamic traction model, although this adds significantly to the computational time of the analytical model.

Figure 4 shows the spin kinematics diagram for the derivation of spin speeds in the MCVT. For the applications considered in this paper the outer race is static and thus $\omega_{outer} = 0$.

The spin velocities at the inner and outer contacts are respectively

$$\omega_{spin,in} = (\omega_{in} - \omega_{orbit}) \sin \phi - \omega_{planet} \sin \phi \text{ and}$$

$$\omega_{spin,out} = (\omega_{outer} - \omega_{orbit}) \sin \theta - \omega_{planet} \sin \theta$$

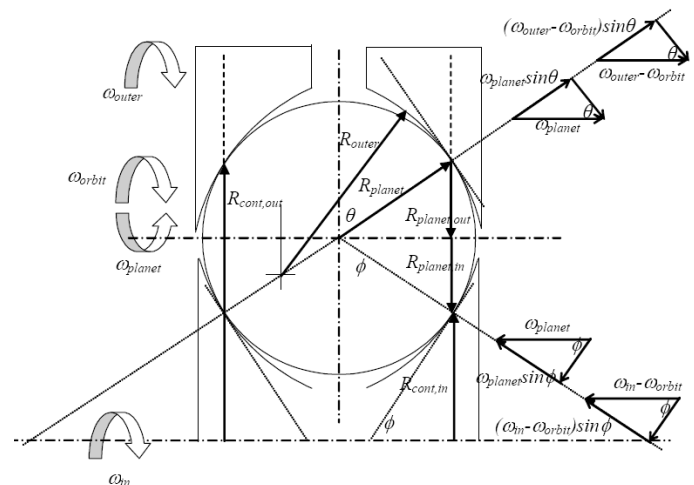


Figure 4 Spin kinematics

DESIGN OF EXPERIMENTS APPROACH

DESIGN FACTORS

As highlighted in the introduction, for the spherical planet design MCVT there are 5 variables that describe the geometry of the transmission and thus the ratio definitions and spin kinematics. There are a further 4 inputs that effect the power density of the transmission, these are the scaling factor, design traction coefficient, design torque and number of planets. To reduce the number of inputs in the first instance the planet radius was considered to be unity, and could thus be removed from the design. Table 1 shows the geometry of one of the existing MCVT prototypes and the corresponding values that these equate to.