Research on planetary bevel gear CVT system based on contact force

Jing Jing Liang[1]*, Rui Qin Li[1], Jia Jun Ren[2]

[1]*School of Mechanical and Power Engineering, North University of China, No. 3 school road, Taiyuan, China*

[2]*School of Mechanical Engineering, Taiyuan University of Technology, No.79 West Yingze Street, Taiyuan, China*

Received 1 August 2014, www.cmnt.lv

Abstract

A virtual prototyping model for dynamical characteristic curves based on contact force was established, through the joint modelling (geometric modelling and constraint modelling) of planetary bevel gear CVT system in UG and ADAMS. The virtual prototyping experimental data proved that the system has the feasibility of over-zero variable speed. It is also verified that the model has advantage on continuously variable speed range, compared with planetary cone ring continuously variable transmission system. The main effect factors of the continuously variable speed performance and output torque are obtained. This model could be used to further study on such issues.

Keywords: planetary bevel gear, continuously variable transmission, virtual prototyping experimental, contact force

1 Introduction

Planetary-type continuously variable transmission device with the central roller in planetary motion transmits driving force through the traction force (friction) resulting from rolling. It realizes the transmission by altering either the working radius of sun wheel or planet wheel， which features the advantages in wide range of transmission and good output performance [1]. Nonetheless, some defects are also found like low transmission efficiency and friction loss because the driving force transmission is achieved through the friction force based upon rolling mechanism. Therefore it is of great practical significance and application value to explore a new type of planetary-gear-type continuously variable transmission system with better speed governing performance and lower friction loss.

Scholars both in China and abroad currently focus on the systems with belt-type transmission and V belt mixed with planetary gear transmission and multiple planetary gear grouping [2-6] making less research on the Planetary Bevel Gear continuously variable transmission system. Besides, owing to the complexity of the working conditions of the systems, it is found necessary to carry out the model building and simulation analysis in advance in order to give correct prediction and guidance. The author, in the light of dynamics, applies UG and ADAMS to modelling and simulation analysis so as to obtain a simulation result based upon relevant dynamics, which stands very meaningful in the further research of the subject.

2 Structure and principle of planetary bevel gear continuously variable transmission system

Planetary Bevel Gear Continuously variable transmission System is a bevel gear planetary system in which the structure improvement is made by replacing the friction wheel with bevel gear transmission that passes driving force and the speed governing realization is obtained by changing the contact radius between the speed ring and speed cone. Its structure is shown in Figure 1.

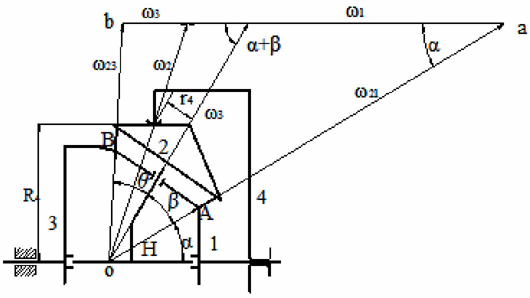

FIGURE 1 Structural diagram of Planetary Bevel Gear step-less speed transmission system

The speed ring keeps static and no turning at work but when the transmission is needed, the speed changing device will allow it to move horizontally along the conical surface of the speed cone. From the output axis different rotating speed and torque can be obtained by changing the contact position between the speed ring and the conical surface as well as the corresponding working radius. This paper aims at the exploration and study on the motion properties by applying virtual prototype technology to obtain the curve of the dynamics characteristics and make a good study of the step-less speed change performance of the system.

* ***Corresponding author*** e-mail: s2003126ljj@sina.com

3 Virtual prototype model establishment of main transmission

3.1 GEOMETRIC MODELING

In the previous study, the author has obtained the configuration parameters of the new type planetary step-less speed transmission system, which are optimized is shown in Table 1. Where: m – modulus, z_1 – teeth number of the bevel gear at input terminal, z_2 – teeth number of Planetary Bevel Gear, z_3 – teeth number of bevel gear at output terminal, δ_1 – reference cone angle of bevel gear at input terminal, δ_2 – reference cone angle of Planetary Bevel Gear, δ_3 – reference cone angle of bevel gear at output terminal, value is shown in Table 1.

TABLE 1 Configuration parameters

Parameter	Symbol	Number of Parameters
Modulus	m	$m=2$
Teeth	z_1, z_2, z_3	$z_1=19, z_2=21, z_3=41$
Sub-degree cone angle	$\delta_1, \delta_2, \delta_3$	$\delta_1=28^\circ, \delta_2=30^\circ, \delta_3=92^\circ$

All the above parameters are used for geometric modelling against various bevel gears, while doing it, each reference cone angle has to be adjusted according to the correct teeth clenching transmission to ensure an equal length of the reference cone generatrix for all teeth clenched bevel gears to satisfy that the teeth of three bevel gears are correctly clenched. In addition, various shared top clearance gear are to be selected for gear tip cones, gear root cones, and gear reference cones when the installation conditions for the bevel gears are taken into account. With a view of what has been stated above, a full bevel gear geometry model is obtained .once we put in the bevel gear geometrical dimensions calculation expression and involutes curve expression in UG, and through Boolean calculation from the curve to the curved surface and to a single tooth. No further description should be repeated here for other components like planetary frame, speed cone, speed ring and axis as well as their modelling based on their assembly relation and dimensions.

Upon the establishment of the model, the initial assembly can be made as per such conditions as the coincident constraint of the output terminal, input terminal bevel gear axes with output terminal, input terminal axis, planetary frame horizontal axes and speed ring axes; the coincident constraint of the planetary frame tilt axis with Planetary Bevel Gear and speed cone axes.

3.2 CONSTRAINT MODELING

Constraint modelling can guarantee the truth and reliability in the simulation analysis. During the process, necessary measures are simplified and the reliability of the result from the simulation analysis can also be ensured [7, 8]. The geometric models established in UG are introduced into ADAMS and based upon the transmission principle and motion relation demonstrated in the Planetary Bevel Gear step-less speed main transmission system, the following constraints are added to the system model.

Joint with fixture pair: input terminal bevel gear and input axis; output terminal bevel gear and output axis; speed cone and Planetary Bevel Gear.

Joint with rotation pair: input terminal bevel gear and planetary frame; connected components of speed cone and Planetary Bevel Gears with the planetary frame; output terminal bevel gear and the planetary frame; output terminal axes and earth.

A model based on Contact Force [9] has to be set up to realize the dynamic simulation for this system to simulate the real body contact. Therefore the constraints based upon Contact Force will be added between input terminal bevel gear and Planetary Bevel Gear; Planetary Bevel Gear and output terminal bevel gear; speed cone and speed ring.

In this paper Contact Force is added by using IMPACT function selected from ADAMS function library, which substantially has been simulated as a nonlinear spring damper. The contact force is composed of two parts: first the spring force which is generated from the mutual cut-in of the two components; the other the damper caused by the relevant velocity [10]. During the analysis of the Contact Force, the Contact Force array is used to define its characteristics. The parameters in the Contact Force array determines the related parameters in the IMPACT function and the main parameters include rigidity k, damped coefficient c, the force nonlinear index e, in-going depth d, static friction coefficient μ_s, dynamic friction coefficient μ_d, static Slip velocity v_s, dynamic Slip velocity v_d [11] and the selection of the parameter value is set by the empirical value is shown in Table 2.

TABLE 2 Empirical value table for contact force parameters

Parameter	Symbol	Number of parameters
Rigidity	$k(N/mm)$	10^5
Damped coefficient	$C(N\cdot S^{-1}/mm)$	50.000
Force nonlinear index	$d(mm)$	0.1
In-going depth	e	1.5
Static friction coefficient	μ_s	0.08
Dynamic friction coefficient	μ_d	0.05
Static slip velocity	$v_s(mm/s)$	0.1
Dynamic slip velocity	$v_d(mm/s)$	1

Given the speed ring is in translational motion, the motion sub-constraint is added between the speed ring and the earth.

Rotating motion stimulation (motion 1) is added to the revolute joint between the input terminal bevel gear and planetary frame. Translational motion stimulation (motion 2) is added to the flat pair between the speed ring and the earth.

Drive add-in: constant rotation speed drive is added to the system input axis and translation drive is added to the speed ring. The constant rotation speed in this paper is set for the system input axis: $1500r/min$, shift to $9000°/s$. Given work radius varying rate of the speed cone is $5mm$, the translation speed of the speed ring in the horizontal direction is converted as: $5/\sin 58° = 5.9mm/s$

Avoid in every way handling step signal in ADAMS resolver. After all, the two drives can be set in the step function form with the exorbitant curve, thus the system drives here are defined as the following: the input axis rotation speed is defined as: $STEP(time,0,0,0.05,9000d)$; the speed ring translation speed is set: $STEP(time,0,0,0.05,5.9)$. From the above, we get the virtue prototype simulation model is shown in Figure 2.

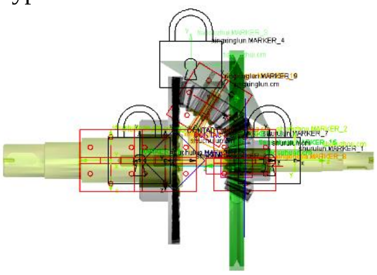

FIGURE 2 Virtue prototype simulation model diagram

4 Performance verification of speed modulation in virtue prototype stimulation model no-load operate

According to the above-set constraint and drive value, we have to ensure the system input axis rotates at a constant angle speed and the speed ring makes parallel move at a constant speed in order to find out the curve patterns in the output angle speed variation during the simulation of the moving process in which the system allows the speed ring to touch initially with the speed cone and then move away. After setting the measuring and simulating environment, we may start simulation and allow the after-treatment module to output the diagrams of the curves is shown in Figure 3.

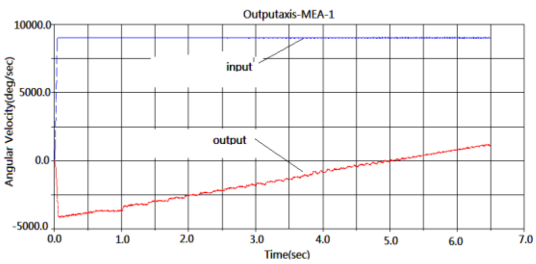

FIGURE 3 under no-load torque input & output rotation speed

4.1 CONSTANT LOAD TORQUE AT WORK

When the input axis reaches the constant speed: $1500r/min$, to the output axis we add constant load torque $6000N \cdot mm$ and with step function definition we add $STEP(time,0,0,0.05,6000)$, thus the output angle speed curve is shown in Figure 4. From the result of the above simulation, we perceive the continuous change within a certain range done by the output rotation speed in the Planetary Bevel Gear continuously variable transmission system and also the zero cross speed modulation enablement, which proves the feasibility of the continuously variable transmission, however, in the process of the simulation, the system occasionally shows the phenomenon that the transmission is done at a fixed transmission ratio. Through our study and research, we presume it is caused by the inertia of the Planetary Bevel Gear, for which an improvement of the system model will be made by setting Planetary Bevel Gear and speed cone linkages to a symmetric layout structure is shown in Figure 5.

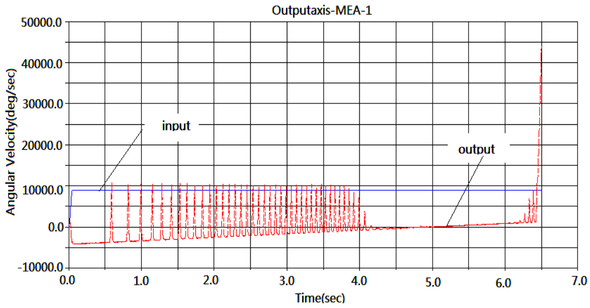

FIGURE 4 Under constant load torque input & output rotation

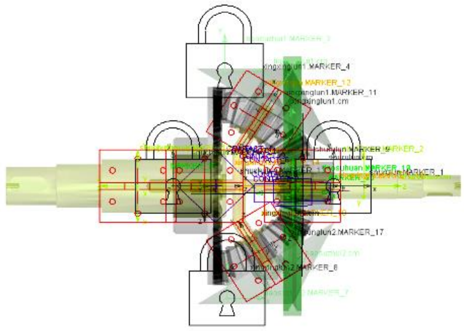

FIGURE 5 Virtue prototype simulation model after modification

4.2 ANALYSIS ON THE MODIFIED MODEL SIMULATION

By giving the simulation analysis to both no-load running of the modified model and constant-load torque, we get the simulation curves is shown in Figures 6 and 7. From the above simulation curve analysis, we find the system model can realize a stable continuously variable transmission upon the improvement, with the transmission rate in the range of $-0.45 \sim 0.14$.

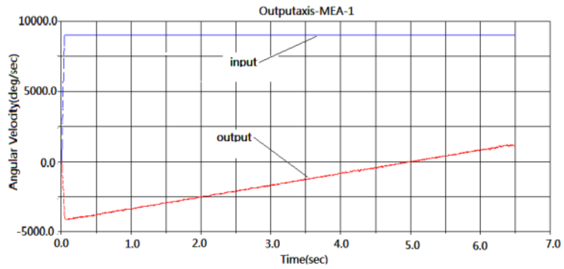

FIGURE 6 Under no-load torque input & output rotation speed diagram of the modified model when the input rotating speed = 9000d/sec

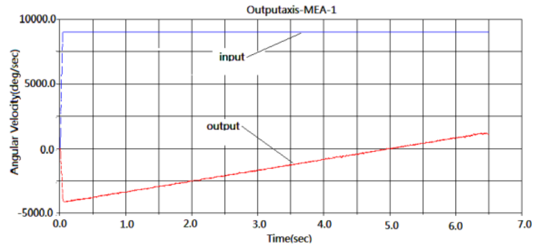

FIGURE 7 Under constant load torque input & output rotation speeddiagram of the modified model when the input rotating

5 Simulation analysis on the system output torque

The output torque presents an important indicator to judge the performance of the continuously variable transmission, of which the parameter simulation is carried out by configuring at the system terminal the input power and constant input rotation speed, meanwhile ensuring a stable running of the whole system with the condition that a certain load torque is given to the output axis, thus the output torque curves are recorded from the output axis. The simulation analysis in this paper is conducted according to the situations provided as following.

5.1 SIMULATION ANALYSIS OF THE OUTPUT TORQUE UNDER THE CONSTANT POWER AND CONSTANT LOAD TORQUE

Setting input power $P = 2.2kW$, load torque $60N \cdot m$, input axis rotation speed $n_1 = 1500r/\min$, $n_1 = 1000r/\min$, $n_1 = 750r/\min$ respectively, we give the simulation analysis to the output torque on the output axis, and the curves can be obtained is shown in Figures 8-10.

From the simulation result analysis, we know that, once the input power and load torque have been rated, the output axis' output torque makes a continual change if the rotation speed of the input axis is changed. The position and scale of the maximum output torque also change: with the continual decrease of the input rotation speed, the maximum output torque continues to increase.

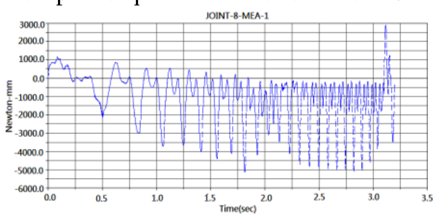

FIGURE 8 Output torque diagram when input rotation speed = 1500*r*/*min*

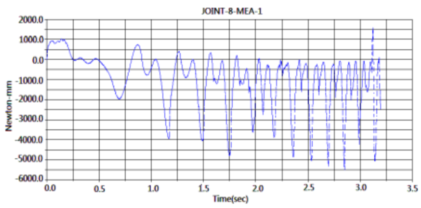

FIGURE 9 Output torque diagram when input rotation speed = 1000*r*/*min*

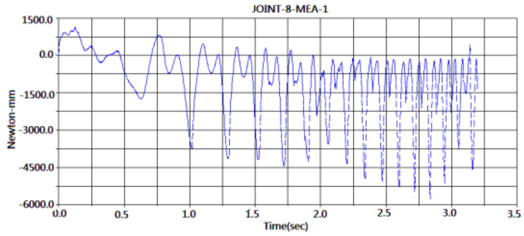

FIGURE 10 Output torque diagram when input rotation speed = 750*r*/*min*

5.2 OUTPUT TORQUE SIMULATION ANALYSIS UNDER CONSTANT POWER AND CONSTANT INPUT ROTATE SPEED

A simulation analysis is made on the output torque over output axis while keeping the input power and input rotation speed at a fixed rate with variable load torque and system stable running. We set input power $P = 2.2kW$, input rotation speed $n_1 = 1500r/\min$, load torque respectively $0.5N \cdot m$, $5N \cdot m$, $50N \cdot m$, in each case, the output torques over the output axes are recorded and shown in Figures 11-13. From the simulation result analysis, we know that, once the input power and input rotation speed are kept unchanged given any change in load torque, no obvious change is witnessed from the output torque over the output axis, which shows that the load torque does not exercise a clear impact on the system output torque.

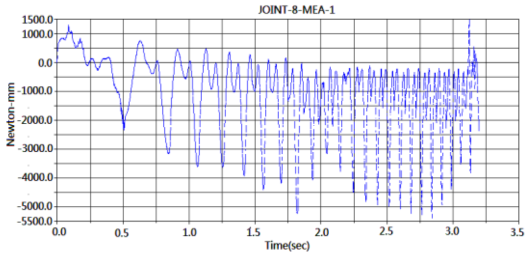

FIGURE 11 Output torque diagram when the load torque = 0.5*N*·*m*

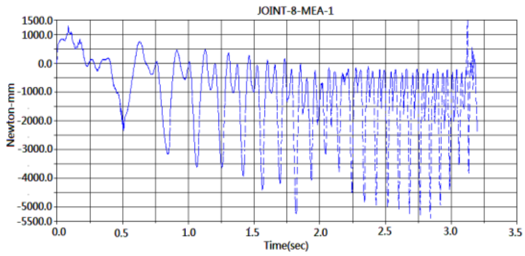

FIGURE 12 Output torque diagram when the load torque = 5*N*·*m*

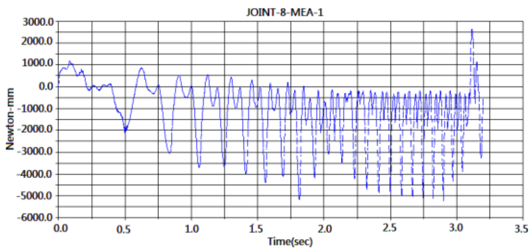

FIGURE 13 Output torque diagram when the load torque = 50*N*·*m*

6 Conclusions

The output rotation speed under the new type Planetary Bevel Gear continuously variable transmission is able to realize a continual change within a certain range, i.e. continuously variable transmission, and also zero cross speed modulation. This proves the feasibility of the system, and its transmission rate is $-0.45 \sim 0.14$. While the original ring cone transmission system provides a transmission rate. The system model after its improvement conducts a stable and continuously variable transmission, which testifies that there is some influence on the performance of the whole main transmission system's continuously variable transmission imposed by the inertia of both the Planetary Bevel Gears and speed cone linkages, of which the amount can be changed to improve the performance of the system continuously variable transmission.

The input rotation speed becomes a vital element that affects the output torque. The maximum output torque continues to increase with the continual decrease of the input rotation speed. On the other hand, the influence on the output torque resulted from the load torque is comparatively small.

Acknowledgements

This research is sponsored by the National Natural Science Foundation of China (Grant No. 51275486) and the Specialized Research Fund for the Doctoral Program of Higher Education (Grant No. 20111420110005).

References

[1] Ruan Z 1999 A Guide to Mechanical Continuously variable transmission Design and Choice *Chemical Industry Press* Beijing Chapter 5
[2] Sun D, Qin D 2008 *Journal of Nanjing Engineering Academy* **6**(4) 9-22 *(in Chinese)*
[3] Duan Q, Yang S 2001 *Journal of Mechanical Design and Research* **17**(3) 40-3 *(in Chinese)*
[4] Zou Z, Gu X, Yang X 2000 *Journal of Agricultural Mechanical Journal* **31**(2) 9-11 *(in Chinese)*
[5] Kim Y S, Park J M , Choi S H 2006 *Journal of Mechanical Science and Technology* **20**(6) 770-81
[6] Bullinger M, Pfeiffer F, Ulbrich H 2005 *Journal of Multibody System Dynamics* **13**(2) 175-194
[7] Huang H, Zou F 2009 *Journal of Mechanical Design and Research* **6**(3) 114-116 *(in Chinese)*
[8] Feng W, Chen X, Ma Z 2007 *Journal of Liaoning Engineering Technology University* **1**(3) 120-2 *(in Chinese)*
[9] Zhu C, Liu T, Zhu W 2010 *Journal of Hydraumatic Aeromechanics and Seal* **4**(5) 29-31 *(in Chinese)*
[10] Li Z 2006 ADAMS Introduction with Detailed Annotation and Examples *National Defense Industry Press* Beijing 56-96
[11] Shu Z, Shu G 2009 *Journal of Mechanical Transmission* **32**(1) 15-19 *(in Chinese)*

Authors

Jing Jing Liang, born in January, 1975, Taiyuan, China

Current position, grades: lecturer of Mechanical and power Engineering.
University studies: Master's degree at Taiyuan Institute of Technology in 2006, Bachelor's degree at Baotou Iron and Steel Institute in 1997.
Scientific interests: mechanism and robot theory and simulation technology.
Publications: 1 patent, 5 academic papers, 2 textbooks.

Ruiqin Li, born in November, 1964, Taiyuan, China

Current position, grades: professor, PhD supervisor, North University of China.
University studies: PhD degree at Shanghai Jiao Tong University in 2004.
Scientific interests: mechanism and robot theory and simulation technology
Publications: 1 monograph, 7 textbooks, 60 academic papers.

Jiajun Ren, born in May, 1958, Xiaoyi, China

Current position, grades: professor, Vice President of School of Mechanical Engineering of Taiyuan University of Technology.
University studies: Master's degree and Bachelor's degree from Taiyuan University of Technology in 1990/1982.
Scientific interests: optimization of mechanical design and CAD, mechanical transmission, ergonomics and design.
Publications: 2 monograph, 2 textbooks, 30 academic papers.