Design and Experiences of a Planetary Gear Box for Adaptive Drives: Proceedings of the 7th European Conference on Mechanism Science

Chapter · January 2019

DOI: 10.1007/978-3-319-98020-1_33

Konstantin Ivanov
Institute of Mechanics and Machine Science of Kazakhstan

Claudia Aide González-Cruz
Autonomous University of Queretaro

Marco Ceccarelli
Università degli studi di Roma Tor Vergata

Daniele Cafolla
IRCCS Istituto Neurologico Mediterraneo Neuromed

Design and experiences of a planetary gear box for adaptive drives

A.K. Ozhiken[1], K. Ivanov[2], M. Ceccarelli[3], C. A. Gonzalez-Cruz[4], D. Caffolla[3]

[1]Al-Farabi Kazakh National University, Kazakhstan
[2]Almaty University of Power Engineering and Telecommunications, Kazakhstan,
[3] LARM, University of Cassino and South Latium, Cassino 03043, Italy
[4]Univesidad Autónoma de Querétaro, Querétaro 76010, Mexico
ozhiken11@gmail.com, ivanovgreek@mail.ru, ceccarelli@unicas.it, claudia.aide.gonzalez@gmail.com, cafolla@unicas.it

Abstract. This paper presents a design of an adaptive gear variator together with lab experiences for its operation characterization. The proposed design is based on a solution with a differential planetary gear systems in which the second degree of freedom is activated when the input torque exceeds the possibility for the first degree of freedom. Both kinematical and mechanical designs are presented with the peculiarities of the design. Experimental validation is worked out with a proper test-bed and test results are discussed for a characterization of the presented variator with its peculiar functioning.

Keywords: gear system, planetary gears, gear variators, design, performance analysis.

1 Introduction

The concept “variator” means the frictional mechanism with the controllable transfer ratio [1]. The following frictional mechanisms are used as the variators: crown variator and conic variator with intermediate rollers and belt transmission with compound wedge pulleys. The control of the crown variator and conic variator is carried out by change of position of intermediate roller. While the control of the belt transmission is carried out by change of diameter of wedge pulley.

The main deficiency of a frictional variator is the low reliability and the complexity of its control. In this way, the more reliable mechanism is the hydro-mechanical transmission (CVT) uniting the fluid converter with a stepped gearing. In this transmission the fluid converter carries out the smooth change of the transfer ratio in narrow limits of each transfer step. However, the deficiencies that the hydro-mechanical CVT presents are: a) a complex design, b) a complex and not quite adequate control system of the gear change and c) ruptures of a transferred power stream leading to blows, to mention a few.

Attempts of the use of double coupling step transfers to soften the forward motion at the switching steps lead to the complication of the design [2].

The gear variator is a new wheelwork branch with constant engagement of toothed wheels and with variable transfer ratio. Theoretical preconditions of gear variators have been developed by Ivanov [3-5]. The construction of experimental prototypes of adaptive gear variator has been created on the basis of the execution of the necessary and the sufficient adaptation conditions (a conditions of existence of a toothed variator) [8-12].

The objective of the present work is the description of a pilot model of an adaptive gear variator at Almaty University of Power Engineering and Telecommunications and the experimental confirmation of its properties on a test-bed.

2 Description of Adaptive Gear Variator

The adaptive gear variator is intended for use in the module of lifting of loads. The working principle behhind the design is that a constant power of the gear variator, the lift velocity should be adapted to the variable weight of the freight. Thus, the engine capacity is defined according to the necessary work to reach the minimum lift velocity for the freight at its maximum weight and viceversa, the maximum lift velocity that can be reached at the minumum freight weight. In this way, it will be possible to profit all the engine capacity.

The design of a gear variator is developed on the basis of the theory of an adaptive gear variator [3-5]. The proposal design contains: a) a necessary adaptation condition, b) a sufficient adaptation condition and c) an independent start condition. The necessary adaptation condition is reduced to the equation of the force adaption effect

$$\omega_{H2} = M_{H1}\omega_{H1}/M_{H2}, \tag{1}$$

where ω_{H2} is the output angular velocity, ω_{H1} is the input angular velocity, M_{H1} is the input motive moment and M_{H2} is the output moment of resistance. This condition allows to define output angular velocity on the set of constant parameters for the input power and the resistance moment.

The sufficient adaptation condition is reduced to the relation among the teeth number, z_i $(i = 1..6)$, of the mechanism wheels

$$-\frac{z_8}{z_7} = \frac{z_3z_4z_5 - z_1z_5z_6}{z_3z_4z_6 - z_1z_4z_5} \tag{2}$$

From this condition it is possible to select a known teeth number for the toothed wheels 7 and 8 in the parallel gearing of the planetary kinematic chain.

The independent start condition is defined by the equality of the lengths of the input and output carriers, which is expressed by means of the teeth number of the wheels

$$z_1 + z_2 = z_4 + z_5 \tag{3}$$

This condition allows to select the teeth number of the toothed wheels of a planetary kinematic chain.

Considering the three conditions after mentioned, by means of Eqs. (1-3), an adaptive gear variator is developed. **Fig. 1** shows the kinematic design of the proposed gear variator. It contains the frame 0, the carrier H_1, the closed contour with toothed wheels 1-2-3-6-5-4 and the carrier H_2. The closed contour contains the satellite 2, the block of solar wheels 1-4, the block of ring wheels 3-6 and the satellite 5. The mechanism has additional doubling constraint in the form of a tooth gearing 8, 7, connecting the input carrier H_1 with the output satellite 5.

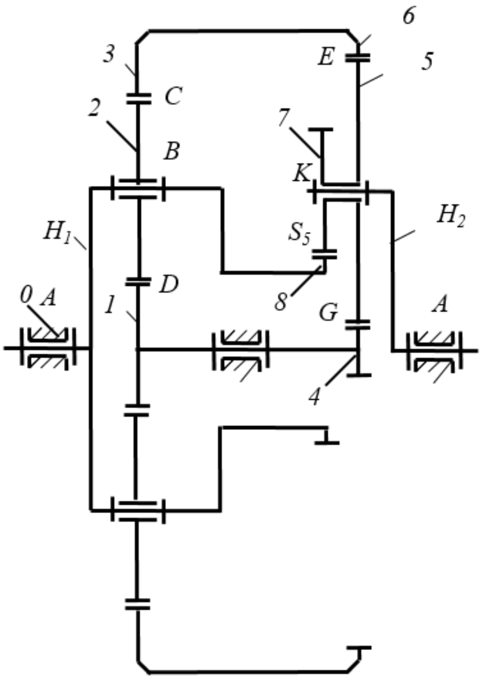

Fig. 1. Kinematic design of the proposed adaptive gear variator.

The mechanism presented on **Fig. 1** contains the mobile closed contour with additional constraint and the doubling transfer 8-7 with a speed coincidence center, creating a shifting bearing (carrying out function of the moment lever). This mechanism provides force adaptation to a variable load as it satisfies necessary and sufficient conditions of force adaptation [14, 15].

3 Design and work of adaptive gear variator

Design documentation under the developed circuit design is executed and an adaptive gear variator is designed. The knots design of the adaptive gear variator at the Almaty University of Power Engineering and Telecommunications (AUPET) is presented on **Fig. 2**. The CAD model in **Fig. 2** (b) shows the main components of the variator: the covers, the input and output carriers and the closed gear chain formed by the sun gear (1), the four satellites (2), the internal epicyclic gears (3-6), the two satellite gears (5) and the sun gear (4). It also shows the additional doubling constraint formed by the gears (8-7), which connect the input carrier with the output satellite (5).

The built prototype of the adaptive gear variator is shown in **Fig. 2** (b). It shows: 1) input half-coupling, 2) input cover with the input bearing of the case, 3) mobile case with gear rings and an output cover, 4) input carrier as an assembly, 5) output carrier as an assembly, 6) output bearing of the case, 7) output half-coupling.

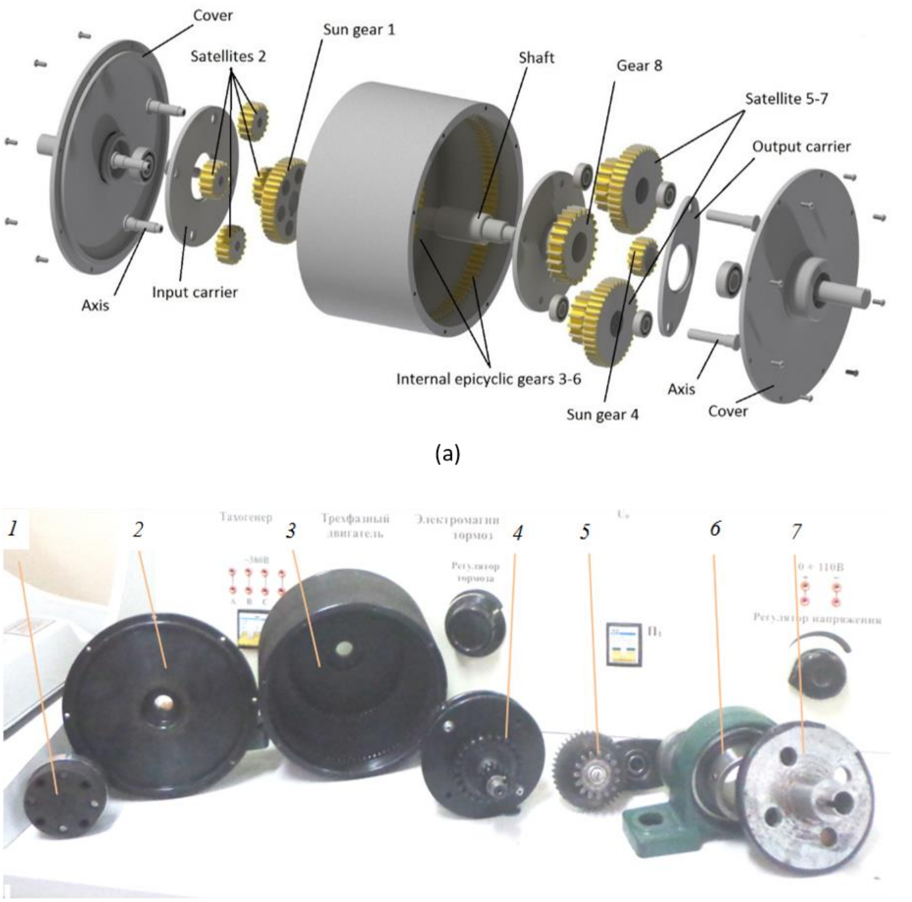

Fig. 2. Design solution for the AUPET gear adaptive variator: a) the CAD design with components; b) the built prototype: 1) input half-coupling, 2) input cover, 3) mobile case with gear rings, 4) input carrier as an assembly, 5) output carrier as an assembly, 6) output case bearing, 7) output half-coupling.

The adaptive gear train operates as follow: When the friction forces between gears is high, satellite gears (5) are locked to the sun gear (4) and the epicyclic internal gear (6), then the movement is transmitted from the input carrier to the output carrier. However, when the friction forces between gears are small, the satellite gears (5) run like output link holding the output carrier in a fixed position and activating the second degree of freedom of the adaptive gear variator. The design parameters of the variator gears are listed in **Table 1**.

Table 1. Parameters of the adaptive variator gears at AUPET.

Parameters	Number of teeth, z	Modulus [mm]	Radio [mm]
Input sun gear	40	2	400
Input satellite gear	6	2	160
Input epicyclic internal gear	72	2	720
Output sun gear	16	2	160
Ouput satellite gear	40	2	400
Output epicyclic internal gear	96	2	960
Output satellite	15	3	225
Additional transmission gear	24	3	360
Input and output carriers	-	-	675

4 Test bed description

Tests for the adaptive gear variator were performed in the AUPET test-bed, see **Fig. 3**. It consists of a basis (tractive) engine (1) that drive the adaptive gear variator (2), which leads the auxiliary (brake) engine (3), which produces a resistant loading torque. The instrumented panel (4) measures different physical variables, such as resistant torque M_R, angular velocity n_R, current strength I_R, voltage U_R and power P_R. While, the monitoring system (5) lets the system performance supervision.

In order to analyze the performance of the adaptive gear variator, a set of tests were conducted following two main objectives: 1) prove the achievement of the force adaptation effect under operating conditions and 2) evaluate the possibility of the motion starts in the presence of an initial resistance torque. Thus, by means of tests, the resistance torque M_R and the angular velocity ω_R of the output variator shaft can be experimentally defined.

The tests are accomplished at constant traction power and with a smooth increase of the resistance torque from null to maximum. While, the shaft variator parameters were measured at the different loading conditions.

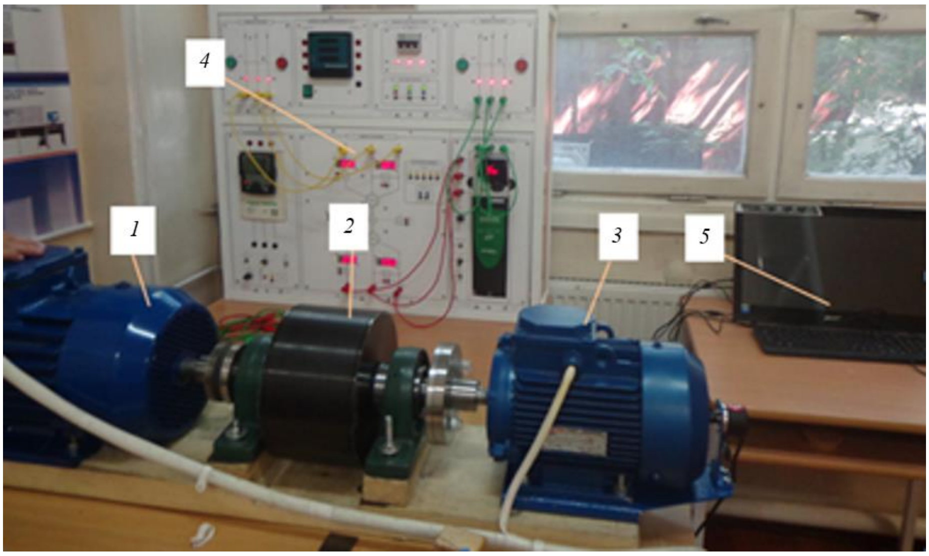

Fig. 3. AUPET test bed: 1-basic (tractive) engine, 2-gear adaptive variator, 3-auxiliary (brake) engine, 4-instrument panel, 5-monitoring system.

5 Experimental analysis

The received parameters of the brake engine and the variator at different loading conditions are listed in **Table 2**. It can be seen that at bigger resistance torque the angular velocity of the variator output shaft and its power decrease. Furthermore, when the resistance torque is minimum, the variator gear ratio is $u = 1$, which means that the first degree of freedom is operating and the movement is directly transmitted from the input carrier to the output carrier. However, as the the resistance torque increase, the output shaft velocity decrease and the gear ratio increases by the action of the second degree of freedom.

Table 2. Brake engine and variator parameters

Point	Resistance torque M_R [Nm]	Number of revolution n_R [rpm]	Angular velocity ω_R [s^{-1}]	Current strength I_R [A]	Voltage U_R [V]	Power P_R [W]	Gear ratio u
A	155.8	54	5.6	4.02	24.5	295.4	17.04
B	77.2	109	11.4	3.41	34.3	350.8	8.48
C	72.1	117	12.2	3.28	36.8	362.1	7.92
D	28.8	292	30.5	2.78	64.3	536.2	3.17
E	17.9	468	49.0	2.57	75.3	580.5	1.96
F	9.1	920	96.3	-	-	-	1.00

The effect of force adaptation or self-regulation characteristic of the gear variator can be seen in **Fig. 4**. A varying loading force demands a change in the angular velocity of the variator output shaft. The maximum freight weight 155.8 Nm is driven at the minimum operational velocity 5.6 cps (54 rpm), while the minimum freight weight 9.1 Nm is driven at 96.3 cps (920 rpm). It can be seen that as the freight weight decreases, the driven speed of the variator in the output shaft increases.

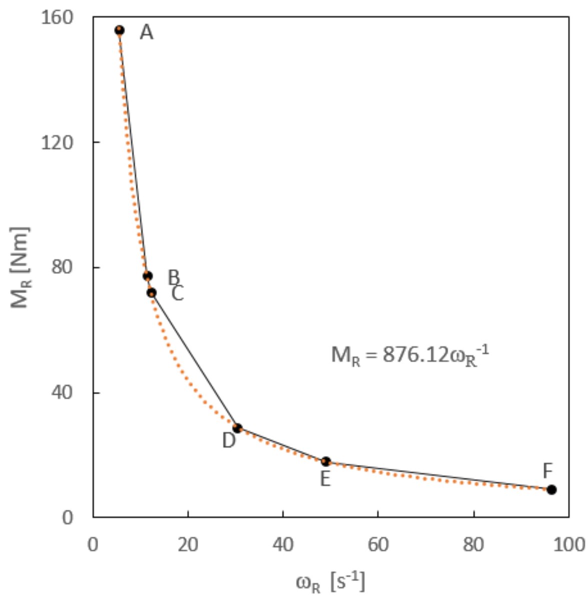

Fig. 4. Experimental tractive characteristic of gear variator.

The results in **Fig. 4** allow to characterize the traction of the gear variator as a function of the angular velocity of the resistance load $M_R = f(\alpha, \omega_R)$. From a numeric regression the gear variator traction can be represented as

$$M_R = \alpha \cdot \omega_R^{-1} \tag{4}$$

which means that the load M_R that can be driven is inversely proportional to the angular operational speed with a gain $\alpha = 876.12$. It is assumed that the value of the constant gain α is given by the engine parameters and the driven load.

6 Conclusions

The paper presents an experimental characterization of a new adaptive planetary gear variator by using a specific test-bed. The peculiarities of the variator are presented in the design structure and operation performance test results show the feasibility of the proposed gear variator

The design of a variator with two degree of freedom and with performance of necessary and sufficient conditions of existence provides transfer of all energy from the traction engine on the output shaft of a variator both at start-up, and in operating conditions. The variator admits an output shaft dead stop at the working traction engine, it also provides force adaptation to a variable load in work operating conditions. In motion operating conditions smooth change of speed of the output shaft occurs at smooth change of a moment of resistance, both at increase, and at decrease of a moment of resistance.

The adaptive gear variator is the highly effective self-controlled connecting gear which can be used for machines with variable technological resistance in all branches of engineering from motor industry to a robotics.

Acknowledgments

The first author wishes to gratefully acknowledge the Mexican Government Foundation CONACYT for the fellowship to pursue her postdoctoral studies at LARM in the academic year 2017-2018.

Reference

1. Lang, K.R.: Continuously Variable Transmissions. An Overview of CVT Research Past, Present, and Future (2000). http://www.cs.montana.edu/~halla/egen310/cvt/cvt.pdf
2. Kim, S., Oh, J., Choi, S.: Gear shift control of a dual-clutch transmission using optimal control allocation. Mechanism and Machine Theory 113, 109-125 (2017).
3. Ivanov, K. S.: Theory of Continuously Variable Transmission With Two Degrees of Freedom: Paradox of Mechanics. In ASME 2012 International Mechanical Engineering Congress and Exposition, Vol. 3: Design, Materials and Manufacturing, Parts A, B, and C, pp. 933-942. Houston, Texas, USA (2012). doi: 10.1115/IMECE2012-85762
4. Ivanov, K.S., Jilisbaeva, K.: Paradox in the Mechanism Science. In New Trends in Educational Activity in the Field of Mechanism and Machine Theory, pp. 132-138. Springer, Cham (2014).
5. Ivanov K.S., Ceccarelli M., Tultaev B.: Force Adaptation in Robot Transmissions. Symbiosis. International Journal of Advanced Robot and Automatization, 2(2), 1-8 (2017).
6. Ceccarelli M., Balbaev G., Ivanov K.: Experimental Test Relation of a New Planetary Transmission. International Journal of Mechanics and Control, 15 (2), 3-7 (2014).
7. Balbayev G., Ceccarelli M.: Design and Characterization of a New Planetary Gear Box. In: New Advances in Mechanisms, Transmissions and Applications, pp. 91-98. Springer, Dordrecht (2014).
8. Ivanov K.S.: Discovery of the Force Adaptation Effect. In proceedings of 11th World Congress in Mechanism and Machine Science, pp. 581 – 585. Tianjin, China (2004).
9. Ivanov K.S.: Gear Automatic Adaptive Variator with Constant Engagement of Gears. Proceedings of the 12th World Congress in Mechanism and Machine Science, pp. 182 - 188. Besancon, France (2007).
10. Ivanov K.S.: The simplest automatic transfer box. World Congress on Engineering (ICME), pp. 1179-1184. London, UK (2010).

11. Ivanov, K. S., Yaroslavceva, E. K.: Device of automatic and continuous change of a twisting moment-and changes of a corrected speed of output shaft depending on a tractive resistance. Patent No. 20 2012 101 273.1. Germany (2012).
12. Ivanov, K. S., Dinasilov, A.D., Yaroslavceva, E.K.: Gear Variator - Scientific Reality. In Mechanisms, Transmissions and Applications, pp. 169-176. Springer, Cham. Switzerland (2015).