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Design and Numerical Characterization of a New Planetary Transmission

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Abstract— **This paper describes the design and characterization of a novel planetary transmission that can be used to adjust the transmission ratio according to the externally applied load. A basic modeling has been formulated to characterize both its design and operation. A detailed 3D CAD model has been proposed in order to investigate the operation feasibility of the proposed design solution. A proper dynamic model has been developed within MSC ADAMS software. Simulation tests have been carried out and results are discussed to validate the proposed design solution.**

Key words: **Mechanical Transmissions, Gears, Planetary Gears, Design, Simulation.**

I. INTRODUCTION

Gearboxes are used in various types of industrial machinery to provide suitable torque while reducing speed from a rotating power source by using gear ratios. Gearboxes are used in many applications, such as wind turbines, conveyors, draglines, bridges and many other machiners. Gears also are used in differential drives of automobiles, final drives of tractors and heavy machineries mainly as reducer. The efficiency of gear trains depends on many factors such as the type and profile of teeth profile, contact stresses, number and type of bearings. These factors have been studied with recent approaches in [3], [10] and [13]. Planetary gear transmissions are commonly used in applications where a large speed reduction is required as pointed out in [12] and [8]. Several design solutions have been proposed in the literature, like for example in [5], [14] and [1]. For example a cam-based infinitely variable transmission can be used for continuously variable transmission, which can also achieve any transmission ratio, [5]. This mechanism consists of two main parts, namely a cam mechanism and a planetary gear set. Cam-based CVT (continuously variable mechanism) can be more complex than others, [5]. In the case of a speed reducer, a gear box with conical gears consists of 8 conical gears. Two of them are horizontal and 6 pinions are located vertically. Each pairs of pinions are locked together. But in this mechanism design in order to obtain any speed ratios it is necessary to change value of pinion or horizontal gear, [14]. Magnetic planetary gears can be also a solution for gearboxes, when they consist of a sun gear, four planetary gears, and a ring gear. But each gear must have an axially permanent magnet that is sandwiched between two yokes made of electromagnetic soft iron. Magnetic gears have main advantage for a low mechanical loss, but the transmission torque is usually very low as pointed out in [11]. Planetary gears can be designed as a continuously variable transmission as proposed for example in [4]. This mechanism can change the gear ratio depending on the load through two degrees of freedom and eventually by using a brake, [4]. Theoretical and experimental study of pushing CVT dynamics is presented in [1], where the work is focused to design advanced CVT systems with improved efficiency. A mechanism with a planetary gear set and a torque converter is designed as a continuously variable transmission in [2]. This mechanism has two degrees of freedom and makes uses of an external torque to start the movement, [2].

Open issues can be still identified in the efficiency smoothly changing reduction ratio depending on the external load to output shaft. This paper describes a design of a new planetary transmission with two degrees of freedom. The main purpose of this new planetary transmission with two degrees of freedom is related to the capability at adapting the operation to variable loading conditions by preserving efficiency and input-output load ratio. Basic principles of this type of gear box are presented in [6] and [7] and this paper gives further developments in the efficiency of planetary transmission. The proposed design solution provides a motion of output link with a speed that is inversely proportional to shaft loading. These features are suitable for using the proposed design in practical applications such as differential planetary gear box in transmissions for vehicles, metal cutting tools, wind turbines and other transmission applications needing smooth control of ratio reduction but adaptation to a variable load. A proper dynamic model has been developed within MSC ADAMS software to provide information on the feasibility of the proposed design solution. Simulation tests have been carried out and results are discussed for validating the proposed design and characterizing its operation.

II. THE IDEA FOR A NEW PLANETARY TRANSMISSION

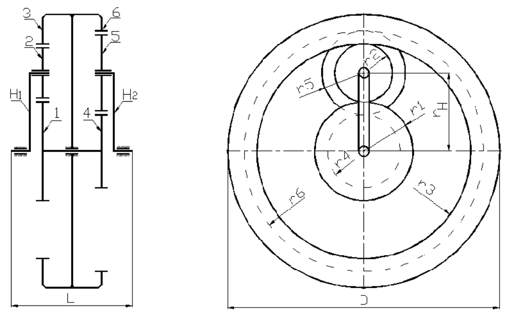
A planetary mechanism contains at least one rigid body which rotates about its own axis and at the same time revolves about another axis. Points of this body will generate epicycloids or hypocycloids trajectories. Therefore a planetary mechanism is often called as an epicyclic or cyclic mechanism. A planetary mechanism can be obtained by mounting a rigid body that is often referred to as a planet, on a crank pin. The crank is generally called the arm or carrier, [9].

The proposed new planetary transmission can be considered a CVT with a planetary gear set. In this paper a new solution is considered for improving the efficiency of planetary transmissions. This mechanism has two mobile planetary gear sets with an asymmetrical design. The asymmetrical design gives special operation features. The special operation features can be recognized in smoothly changing reduction ratio as depending on the load of output shaft.

Referring to Fig.1 the new planetary transmission is conceived with two degrees of freedom with a mechanism consisting of an input carrier H_1, an output carrier H_2, central (sun) gears 1 and 4, which are fixed on a shaft, satellites 2 and 5, central internal gears 3 and 6 which are fixed together. Gears 2-3-6-5-4-1 form a closed mechanical chain with a differential operation. Carrier H_1 transfers input driving force to the closed mechanical chain and carrier H_2 transfers output resistance force. Motion starts at fixed output carrier with one degree of freedom. At this time satellite 5 is the output link. To transmit motion from input carrier H_1 to output carrier H_2 satellite 5 must be locked and this can be obtained thanks also to friction at gear contacts. This is the peculiarity of the proposed system. The input carrier H_1 moves gear 2 that pushes both gears 1 and 3 that transmit different forces to gears 6 and 4 correspondingly. Thus, gear 5 moves by different forces coming from its contacts with gears 6 and 4, and therefore carrier H_2 moves. In addition the mechanism will be able to work with two degrees of freedom because of the possibility of activating a second degree of freedom when satellite 5 will be unlocked by overcoming frictions at gear teeth contacts. Because of its functioning this mechanism can be applied as gearbox of cars, metal cutting machines and where is necessary smoothly to change reduction ratio of transmissions. This mechanism can start movement without using additional device, when force can overcome friction on the satellites.

This planetary transmission can change reduction ratio like CVT as depending on an external load of output carrier.

Main design characteristics of the proposed design related to two input mobile links, namely two degrees of freedom, stepless operations, smoothly and automatically changing reduction ratio depending on the load at the output shaft. The operation advantages of this mechanism are in smoothly and automatically changing reduction ratio depending on the load of the output link and the possibility to start movement without using any additional device. This mechanism can be a suitable transmission solution for any non constant operation, since it is able to adapt its operation to variable load.



(a) (b)

Figure 1 A kinematic scheme for a new planetary gear box with design parameters: (a) longitudinal view; (b) cross-section view.

A kinematic characterization of the mechanism can be expressed as function of parameters of external torques on the carriers M_{H1}, M_{H2} and input angular velocity ω_{H1}. Referring to Fig.1, the kinematic relations among the angular velocities of the gears with z_1, z_2, z_3, z_4, z_5, z_6 teeth can be expressed in the form

$$\frac{\omega_1 - \omega_{H1}}{\omega_3 - \omega_{H1}} = u_{13}^{(H1)} \tag{1}$$

$$\frac{\omega_1 - \omega_{H2}}{\omega_3 - \omega_{H2}} = u_{46}^{(H2)} \tag{2}$$

Where $u_{46}^{(H2)} = -\frac{z_6}{z_4}$ (3)

$$\omega_{H2} = M_{H1}\omega_{H1} / M_{H2} \tag{4}$$

When z_i are the number of teeth in the gear (i=1,..,6). From Eqs. (1) and (2) angular velocities ω_3, ω_1 of gears 3 and 1 can be obtained as

$$\omega_3 = \frac{(u_{13}^{(H1)} - 1)\omega_{H1} - (u_{46}^{(H2)} - 1)\omega_{H2}}{u_{13}^{(H1)} - u_{46}^{(H2)}} \tag{5}$$

$$\omega_1 = u_{13}^{(H1)}(\omega_3 - \omega_{H1}) + \omega_{H1} \tag{6}$$

A fairly easy numerical example can be carried out for an application for a wind turbine, Fig.2. Assuming from wind flow ω_{H1}=100 rpm and M_{H1} = 15 Nm; M_{H2} = 14 Nm, (Fig.1), the output and intermediate angular velocities ω_{H2}, ω_1, ω_3 and internal forces can be computed with the proposed model through Eqs. (1) to (6) by considering $\omega_4 = \omega_1$, $\omega_6 = \omega_3$. From Eq. (4) angular velocity of output carrier is computed as ω_{H2} = 75 rpm. From Eqs. (5) and (6) angular velocities of gears 1 and 3 are computed as ω_1=250 rpm and ω_3=50 rpm, respectively.

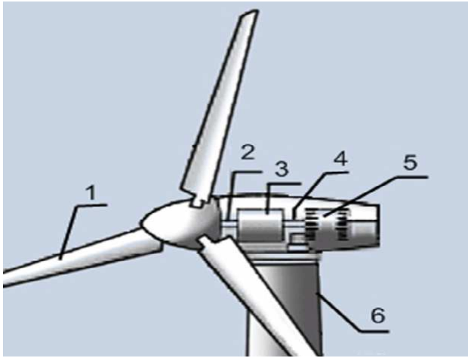


Figure 2 A wind turbine with a proposed planetary gear box: (1-blades; 2-input shaft; 3-planetary gear box transmission 4-output shaft; 5-generator; 6-tower).

III. A MECHANICAL DESIGN AND VIRTUAL MODEL

A CAD design of a gearbox with planetary gear set has been worked out in Solid Works software. Fig.3 shows an exploded CAD design of planetary gear box with the following main components, referring to Fig.1: 1-output carrier; 2-bearing; 3-output satellite; 4- spindle of output satellite; 5-bearing; 6-bearing of internal gears; 7-gearshuft; 8-sun gear; 9-epicyclic gears; 10-input satellite; 11-spindle of input satellite; 12-input carrier. The full mechanical design of the mechanism with housing is shown in Fig.4.

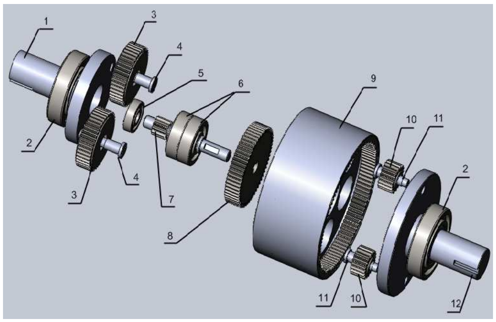


Figure 3 A CAD exploded assembly of a new gear box design.

The proposed planetary transmission consists of a mechanical planetary gear set without additional devices such as torque converters or electronic parts. General design characteristics have been selected for practical applications of the transmission, like for example, in wind turbine installations. A wind turbine installation can be identified, for example, by referring to a small wind turbine of 5 kW power and with average wind speeds of 15-20 m/s, Fig.2.

All the geometrical parameters have been defined within the CAD model in Figs.3 and 4. The design parameters can be sized for the wind application in Fig.2 with a maximum high D (in Fig.1b) of 180 mm and a maximum longitudinal size L (in Fig.1a) of 110 mm. The input and output shafts have a diameters of 32 mm and 30 mm, respectively. The overall weight is 5 kg if made of steel.

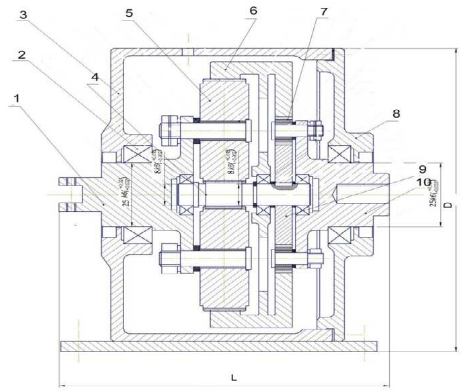


Figure 4 Mechanical design of a new planetary gear box in Fig.2: 1-output carrier; 2-bearing; 3-housing; 4-gear shaft; 5-output satellite; 6-epicyclic gears; 7-input satellite; 8-cover; 9-sun gear; 10-input carrier.

IV. SIMULATION RESULTS

A dynamic simulation of the planetary gear box has been carried out by using MSC ADAMS software. The MSC ADAMS model of the proposed planetary gear box is presented in Fig.5. Input values such as angular velocity, input and output torque, stiffness, dumping coefficients, and friction forces have been defined accordingly as listed in Table 1. Input angular velocity and torque have been set as a constant values of 100 rpm and 15 Nm. Output torque is variable. All gears are spur gears with module 1 mm. Friction coefficient of gears has been set as equal to 0.2 by referring to a contact of steep surfaces. Table 1 summarizes main other parameters that have been assumed by referring to feasible values for a real case of study considering material, penetration depth and force exponent. All the geometrical dimensions have been set as by referring to the models in Figs. 3 and 4. After setting the above-mentioned parameters significant attention has been addressed in properly modeling all the constraints and joints in order to achieve a reliable operation of the proposed model as in a feasible mechanical design.

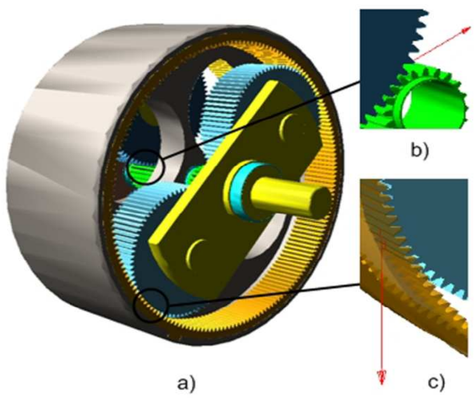


Figure 5 ADAMS model of gear box design in Fig. 3 and 4: a) ADAMS model; b) contacts between internal gear and satellite; c) contacts between sun gear and satellite.

TABLE 1 Input parameters for simulation of model in Fig. 4

Parameter	Value	Units
Input angular velocity	100	rpm
Input torque	15	Nm
Output angular velocity	Plots in Fig. 6	rpm
Output torque	Variable (14-15)	Nm
Damping	40	N*sec/mm
Young's modulus	2.07 E+005	N/mm**2
Density	7.801 E-006	kg/mm**3
Penetration depth	0.1	mm
Force exponent	1.8	

Several cases of study have been computed in order to investigate the dynamic behavior of the proposed transmission. In particular, preliminary tests have been carried out by considering a constant input speed as 100 rpm and torque as 15 Nm and variable output torque as 14-15 Nm. The output torque is prescribed by the alternative suitable operation then, they have been verified that the output speed, in accordance with the expected smooth variable transmission ratio and the constant output power. Examples of the results that have been obtained are reported in the plots of Figs.6 to 12. Angular velocities of the input and output epicyclic internal gears are presented in Fig. 6. The epicyclic internal gears rotate with the same speed approximately of 108 rpm. Fig. 7 shows the plots of the computed angular velocities of the sun gears. The input and output sun gears rotate with same speeds. Angular velocities of the input and output planet gears are presented in Fig. 8. Angular velocity of the input planet gear approximately is 250 rpm. Angular velocity of the output planet gear approximately is 125 rpm. Fig. 9 shows the plots of the computed torques of the epicyclic internal gears. Torque reaches 32 Nm at 2.5 second, after this time the system works properly with approximately value of 29 Nm. Fig. 10 shows the plot of the computed torques of the sun gears and torque value approximately is 75 Nm. Computed results of contact forces between gears are presented in Fig. 11 and 12. In Fig. 11 contact forces are plotted as during the simulated motion for a full rotation of the output shaft. Fig.11 shows the plots of the computed contact forces between input satellite and internal gear, which is related to Fig. 5 (c). Considering the curve in Fig. 11, the approximately highest contact force of 1.1 N appears at 2.5 second and the contact force decreases to 0.2 N at 7.5 second. This suddenly change of contact forces at 2.5 second can be thought as due mainly to friction at gear teeth contacts, while the system is starting a motion. Next changing of contact forces at 7.5 second can be thought due to variable applied load to the output shaft. Computed results of contact forces of the output planet gear and output internal gear are presented in Fig. 12, which is related to Fig. 5 (b). The highest contact force of 1.1 N appeares at 2.5 second and contact force of 0.5 N is computed at 7.5 second. The values of contact forces increase by increasing the values of the external loads on the output shaft.

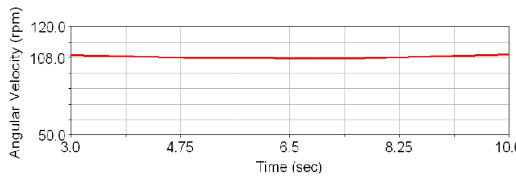


Figure 6 Computed plot of the angular speed of the input and output internal gears.

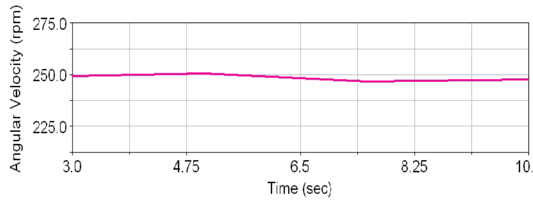


Figure 7 Computed plot of the angular speed of the sun gears.

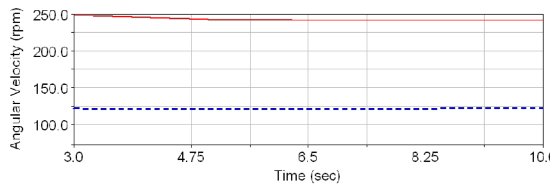


Figure 8 Computed plot of the angular speed of the input (continuous line) and output (dot line) planet gears.

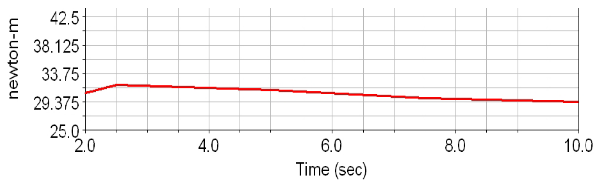


Figure 9 Computed plot of the torque of the internal gears.

Figure 10 Computed plot of the torque of the sun gears.

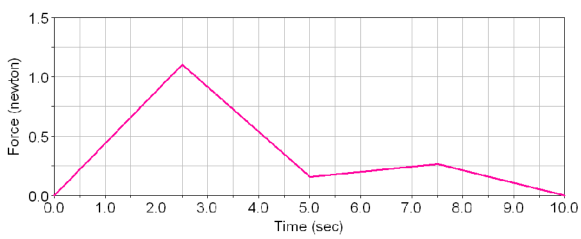


Figure 11 Computed plot of the contact forces between planet gears and internal gears in Fig.5 (c).

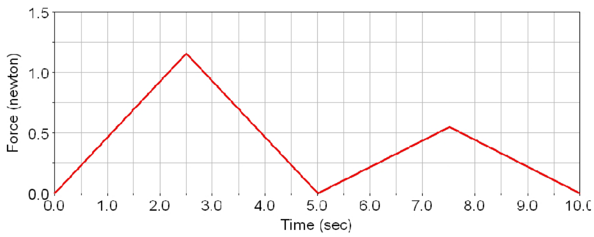


Figure 12 Computed plot of the contact forces between planet gears and sun gears in Fig.5 (b).

V. CONCLUSION

A planetary gear box with two degrees of freedom has been studied from aspects of mechanical design and kinematic modeling. A mechanical design and 3D CAD model of a planetary gear box with two degrees of freedom have been proposed in order to adapt the operation to variable loading. Design of the planetary gearbox is shown in kinematic scheme. The formulated equations are tested by numerical examples. A proper dynamic model and simulations have been carried out in MSC ADAMS environment. Simulation results show that the proposed planetary gear box has suitably constant output values both in terms of speed and torque. The simulation results also show that the proposed gear box smoothly changes the reduction ratio at constant input speed. Contact forces between gears are small enough to use the proposed system under the expected loading conditions.

VI. ACKNOWLEDGMENTS

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