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Modeling and theoretical analysis of a novel ratcheting-type cam-based infinitely variable transmission system

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ABSTRACT

An infinitely variable transmission (IVT) is a system that allows for a continuous (non-discrete) variation (including zero) in transmission ratio between two rotating elements. In this paper, a novel ratcheting-type IVT mechanism is presented and its geometrical design and kinematic analysis are studied in details. The proposed system contains two identical units. Each unit includes a cam with a follower, oscillatory slotted links pivoted at a shaft that can be moved vertically by a hydraulic ram (alterable transmission ratio), and a grooved wheel with an actuating rod. The input rotational motion is converted through each unit to an oscillatory angular motion of controlled amplitude. This resulting motion is rectified using a ratchet to get a unidirectional output rotational motion. Therefore, the system output motion will have a different velocity and acceleration than those of the system input. The kinematic analysis revealed that the transmission ratio can be varied continuously in a range from zero to infinity. The analysis also showed that, for particular transmission ratios, the system gives uniform output (angular velocity and acceleration) for a corresponding uniform input.

1. Introduction

Continuously variable transmission systems (CVT) are mechanisms in which the transmission ratio between input and output rotating elements can be changed continuously (stepless changing). In contrast, in traditional transmission systems such as gear boxes, the variation of the transmission ratio is carried out through discrete shifting. A more important type of such mechanisms is the infinitely variable transmission (IVT), which is a CVT system that involves a zero transmission ratio. The continuous transmission variation feature encourages the automotive engineers to use such systems as alternatives for the conventional transmission technique in automobiles. The CVT system achieves continuous matching between the engine and the wheels, securing smooth engine loading as well as maintaining a comfortable ride. In addition, the engine power loss during gear shifting is minimized, which means, in other words, reducing fuel consumption and gas emissions. Another

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Nomenclature

c_1	Constant of proportionality of the cam profile (m)
c_2	Constant of proportionality of the grove in the grooved wheel (m)
k	Transmission ratio (dimensionless)
L	Distance between the cam follower and the actuating rod (m)
x_1	Distance traveled by the cam follower (m)
x_2	Distance traveled by the rod (m)
y_1	Vertical distance from the center of rotation of the slotted link to the cam follower (m)
y_2	Vertical distance from the center of rotation of the slotted link to the rod that actuates the grooved wheel (m)
$\dot{y}_2$	Velocity of the transmission ratio alterable (m/s)
θ_1	Angular displacement of the cam (rad)
θ_2	Angular displacement of the grooved wheel (rad)
ω_1	Angular velocity of the input shaft (rad/s)
ω_2	Angular velocity of the output shaft (rad/s)
α_1	Angular acceleration input shaft (rad/s^2)
α_2	Angular acceleration of the output shaft (rad/s^2)

important advantage of CVT in automobiles is allowing the engine, to a good extent, to operate at its most efficient range of speed. A study of different types of transmissions, including CVT systems and their application in passenger cars, is presented in [1]. Continuously variable transmission systems may also be used in bicycles [2]; this allows for an infinite range of speed ratios and reduces the disadvantages of traditional shifting like lag chain slippage and derailment.

In addition to transportation applications, the CVT systems would also be used in wind turbine electric generator units [3]. This solution maximizes the efficiency of the unit by controlling the velocity of the electric generator independently of the turbine speed (which varies according to wind speed). Therefore, the electric power is produced at the desired voltage and frequency. More generally, there is an interest in applying CVT systems in a wide range of drive systems to control the torque and velocity at the driver and driven elements.

A substantial amount of prior work was introduced in the field of continuously variable transmission systems, and several types have been investigated. Nevertheless, their application is still limited due to practical considerations, as it will be explained later. The traction type is considered as the most common CVT system. In this type, the power is transmitted by traction force. The V belt type is the most applied traction CVT [4–7], in which the power is transmitted by a V belt between two pulleys, where each pulley has a fixed and movable sheave. The value of the active radius of each pulley is controlled by changing the gap between the sheaves. We refer the reader to reference [8] for an extensive review of the V-belt CVT.

Another traction CVT is the toroidal type [9–11], in which power is transmitted purely by the friction force between two contact surfaces or by the shear strength of a viscous fluid. The limited transmitted power and the low reliability are well-known limitations of the traction-based CVT systems, which reduce their usability.

There are many types of non-traction continuously variable transmission systems where the power transmission does not depend on friction, for instance the systems invented in [12,13], where ratchets are included in their combination. Ratcheting CVTs are very common non-traction systems, in which the rotational input motion is converted into an oscillatory motion of varying amplitude, then rectified into a rotational output motion through one-way clutches (ratchets).

Another type of ratcheting CVT is the inertial system that was introduced and studied in [14,15]. This type of CVT is based on the principle of inertia, which has many advantages, like compactness and minimum friction losses, in addition to its ability to protect the engine during overload. In the inertial CVT systems, rectifiers or ratchets are also needed to convert the oscillatory motion into one-way rotational motion. In general, the ratchets or rectifiers are the weakest links in the mechanism. Accordingly, Morales and Benitez [16] presented a numerical and experimental study on an inertial CVT system using different rectifiers.

Benities et al. [17] presented a ratcheting IVT system that includes one-way clutches and two epicyclical gear systems. Dennis et al. [18] also presented a ratcheting IVT system including a three-dimension cam, a number of cam followers, one-way clutches, and an epicyclical gear train. The main advantage of the system described in [18] is that it can perform uniform output for uniform input. A ratcheting type cam-based IVT system was also presented in [19]. This system consists in a number of identical units, each unit containing a three-dimensional cam, a cam follower, and a grooved wheel.

In this paper, a novel ratcheting type cam-based infinitely variable transmission system is proposed. A full description of the new system, its components, operation and the kinematic analyses are presented.

2. System components and configuration

The proposed IVT system is a combination of two identical units assembled in one system as shown in Fig. 1, each unit containing the following parts:

(1) a cam fixed on the input shaft of the system;
(2) a cam follower;

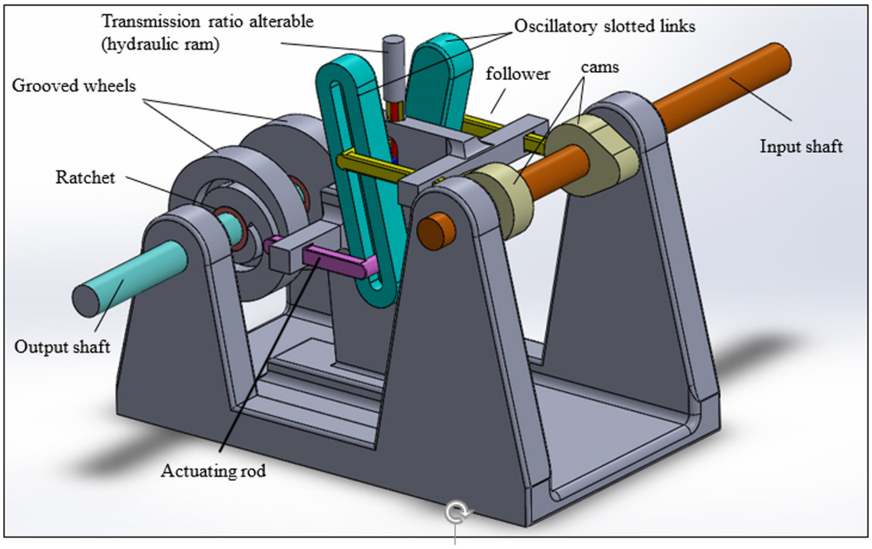

Fig. 1. Proposed infinitely variable transmission system (IVT).

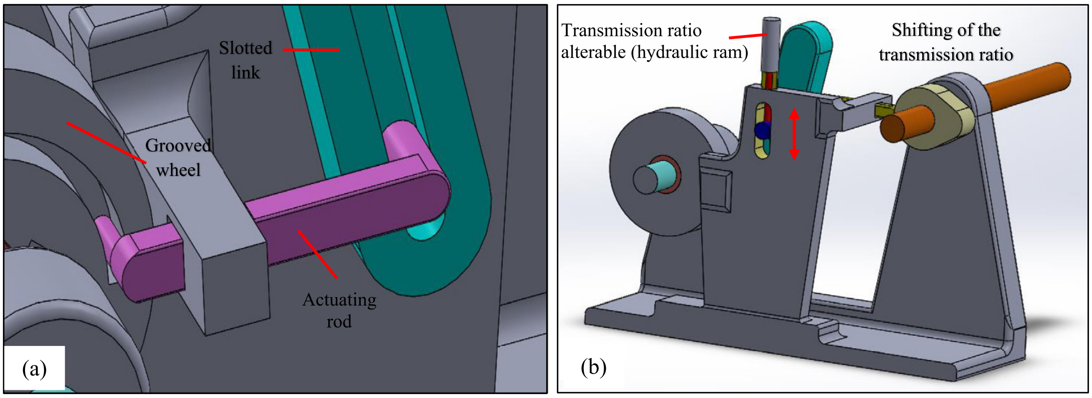

Fig. 2. A more detailed representation of the IVT system shows (a) the connection of the actuator with the slotted link and the grooved wheel, (b) the mechanism for changing the transmission ratio.

(3) an oscillatory slotted link pivoted with respect to the shaft connected to the alterable transmission ratio (hydraulic ram);
(4) a rod actuating a grooved wheel, this rod having a cylindrical transverse tip at both ends in order to connect with the slotted link at one extremity and with the grooved wheel at the other extremity (see Fig. 2a);
(5) a grooved wheel fitted to the output shaft via a one-way clutch (ratchet).

3. System operation

The principle of operation of the proposed IVT system can be described in terms of steps representing the form of motion at each part as follows.

(1) The angular motion of the cam at the input shaft is converted into a reciprocating motion via the cam follower.
(2) The reciprocating follower motion is converted into an oscillatory rotational motion at the slotted link. The amplitude of this oscillatory angular motion depends on the selected vertical position of the axis of rotation of the slotted link, which is controlled by the alterable transmission ratio.
(3) The oscillatory angular motion of the slotted link is rectified again into a reciprocating linear motion via the actuating rod.
(4) The motion of the actuating rod is converted into an oscillatory rotational motion at the grooved wheel on the output shaft. The angular amplitude of this motion, depending on the motion of the alterable, may differ from that at the cam for the same period of time. For high transmission ratios, the grooved wheel revolves with an angle larger than one

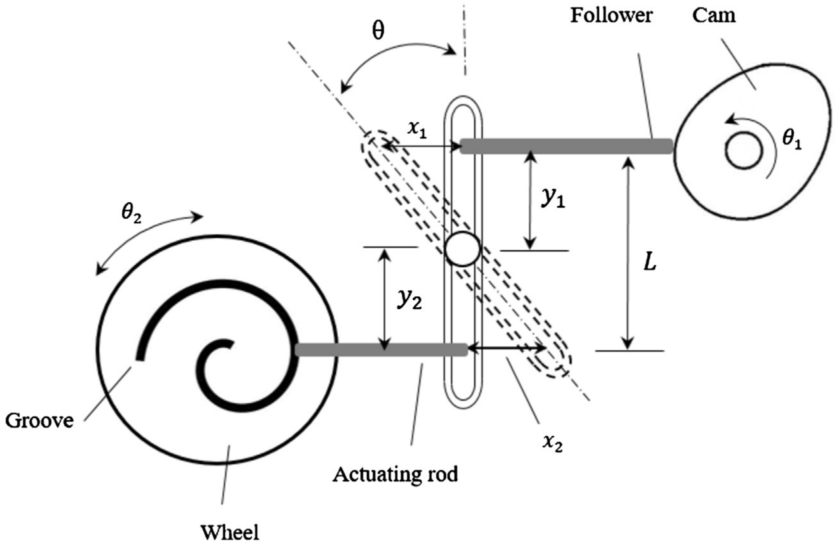

Fig. 3. Schematic diagram showing the operation of a single unit in the IVT system.

revolution, i.e. there is a need for a cam profile covering an angle larger than 2π. This is the reason behind using a groove on a wheel rather than a cam.

(5) By using a one-way clutch (ratchet), a selected one-way direction of the angular oscillatory motion of the grooved wheel will actuate the output shaft. This results in a one-directional rotational motion of the output shaft.

The description of the operation above illustrates the motion during the outward stroke of the cam follower. For the return stroke, the mechanism travels to its original position. This can be achieved by means of torsional springs fitted to the slotted links that keep the cam and the follower in contact continually during operation. During the outward stroke, by the effect of the ratchet, the grooved wheel and the output shaft are locked, while they slip relative to each other during the return stroke. This mean that the ratchet maintains a selected unidirectional rotational motion of the output shaft from oscillatory angular motion. Accordingly, during the return stroke, there will be a discontinuity in the transmission of the power if a single unit is used in the system. In order to achieve a continuous transmitted torque, the system must have at least two units oriented along the input and output shafts (see Fig. 1).

The variation in the transmission ratio is achieved by a stepless movement (shifting) of the axis of rotation of the slotted links by a alterable transmission ratio. A hydraulic ram can be used for this purpose, as illustrated in Fig. 2b. This shifting will change the amplitude of the angular oscillatory motion of the slotted link and then the angular displacement of the output shaft. In other words, the ratio between the input and the output angular displacements is controlled by the alterable position, which can be shifted continuously (stepless) along the vertical axis. This feature allows the proposed system to be considered as a continuously variable transmission system. The system also can be considered as an IVT, as it will be demonstrated by the kinematic analysis of the system that is presented in details in the next section.

4. Kinematics

In this section, the kinematics of the proposed system is presented in details. Fig. 3 is a schematic diagram that illustrates the motion of a single unit in the IVT system.

The cam profile is designed to deliver a constant speed during the outward stroke (the power stroke for each unit in the system). Accordingly, the linear displacement of the follower can be described as follows:

$$x_1 = c_1\theta_1 \tag{1}$$

where
x_1 is the distance traveled by the cam follower during cam rotation,
c_1 is a constant of proportionality,
θ_1 is the angle of rotation of the cam and the input shaft.

Eq. (1) is differentiated with respect to time to obtain the velocity of the follower; this gives $v_1 = c_1$, which is constant and independent of θ_1. Furthermore, the acceleration is obtained by differentiation of the velocity equation with respect to time. This gives zero acceleration during the motion of the follower. The displacement, velocity, and acceleration of the follower, having uniform speed profiles for the outward and return strokes, are displayed in Fig. 4.

It can be seen in this figure that the velocity of the follower is negative in the range $\pi \leq \underline{\theta}_1 \leq 2\pi$, which represents the return stroke of the follower. This problem can be easily overcome by adding another unit that performs an out stroke,

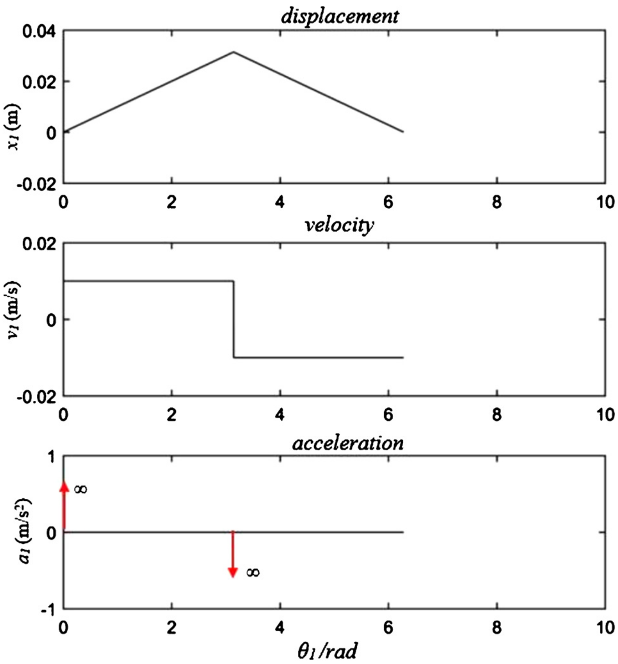

Fig. 4. Displacement, velocity, and acceleration for a cam with a constant velocity profile for outward and return strokes ($c_1 = 0.01$).

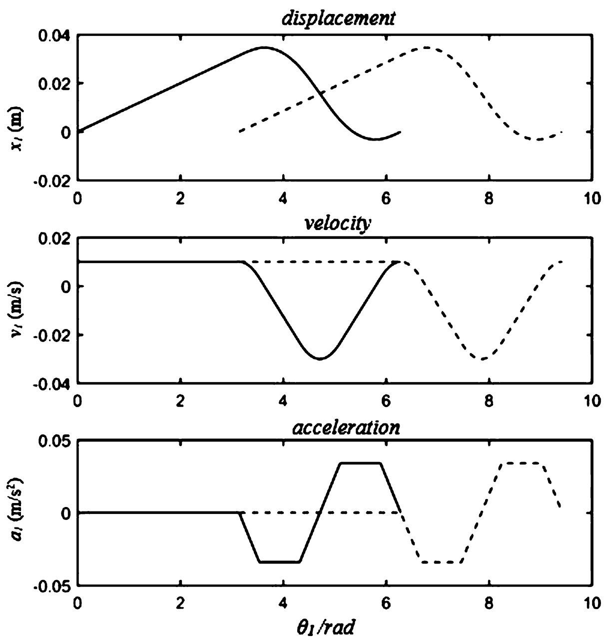

Fig. 5. Displacement, velocity, and acceleration of the followers for two units against the input shaft angle of rotation, θ_1. Solid line for the first unit, and dashed line for the second unit.

while the first unit is in its return stroke. Furthermore, the acceleration moves toward $+\infty$ during the step change in velocity when $\theta_1 = 0$, and $-\infty$ when $\theta_1 = \pi$. This means that an infinite value of force is required at the contact between the cam and the follower; accordingly, this will lead to damage to the contact surfaces. This issue can be resolved by designing the return stroke profile of a trapezoidal type. Combining the constant velocity and the trapezoidal profiles leads to continuous and smooth behavior for the displacement, velocity, and acceleration of the follower. This can be seen in Fig. 5 for a dual unit in the system; the equation representing the profiles at all intervals are illustrated in Table 1.

Table 1
Kinematic equations of the motion of the cam follower through one revolution.

θ_1	Acceleration	Velocity	Displacement
$[0,\ \pi]$	0	c_1	$c_1\theta_1$
$[\pi,\ (9/8)\pi]$	$\frac{-8A}{\pi}(\theta_1-\pi)$	$\frac{-4A}{\pi}\theta_1^2+8A\theta_1-4A\pi+c_1$	$\frac{-4A}{3\pi}\theta_1^3+4A\theta_1^2+(-4A\pi+c_1)\theta_1+\frac{4}{3}A\pi^2$
$[(9/8)\pi,\ (11/8)\pi]$	$-A$	$-A\theta_1+\frac{17}{16}A\pi+c_1$	$-\frac{A}{2}\theta_1^2+(\frac{17}{16}A\pi+c_1)\theta_1-0.56A\pi^2$
$[(11/8)\pi,\ (13/8)\pi]$	$\frac{8A}{\pi}\theta_1-12A$	$\frac{4A}{\pi}\theta_1^2-12A\theta_1+\frac{69}{8}A\pi+c_1$	$\frac{4A}{3\pi}\theta_1^3-6A\theta_1^2+\left(\frac{69}{8}A\pi+c_1\right)\theta_1-4A\pi^2$
$[(13/8)\pi,\ (15/8)\pi]$	A	$A\theta_1-\frac{31}{16}A\pi+c_1$	$\frac{A}{2}\theta_1^2-\left(\frac{31}{16}A\pi-c_1\right)\theta_1+1.69A\pi^2$
$[(15/8)\pi,\ 2\pi]$	$\frac{-8A}{\pi}(\theta_1-2\pi)$	$\frac{-4A}{\pi}\theta_1^2+16A\theta_1-16A\pi+c_1$	$\frac{-4A}{3\pi}\theta_1^3+8A\theta_1^2-(16A\pi-c_1)\theta_1+\frac{32}{3}A\pi^2-2\pi c_1$

In order to construct the full kinematic model of the system, the motion of the cam follower for the outward stroke is only considered as it represents the power stroke that is characterized by Eq. (1). Now, in order to relate the motions of each paired element in the system (see Fig. 3), the angle turned by the slotted link can be written as:

$$\tan\theta=\frac{x_1}{y_1} \tag{2}$$

Similarly, the relation between the motion of the actuating rod and the slotted link is:

$$\tan\theta=\frac{x_2}{y_2} \tag{3}$$

Regarding the grooved wheel, the groove is designed to have a constant speed profile. This is represented as following:

$$x_2=c_2\theta_2 \tag{4}$$

where
x_2 is the displacement of the rod,
c_2 is a constant of proportionality,
θ_2 is the angle of rotation of the grooved wheel.

Gathering Eqs. (1) to (4) gives the angular relation between the input and output shaft as follows:

$$\frac{c_1\theta_1}{y_1}=\frac{c_2\theta_2}{y_2} \tag{5}$$

Rearranging Eq. (5) using the relation $y_1+y_2=L$, where L is the vertical distance between the follower and the actuating rod (which is constant in the system), see Fig. 3, Eq. (5) can be written as:

$$\frac{c_1\theta_1}{L-y_2}=\frac{c_2\theta_2}{y_2} \tag{6}$$

Now Eq. (6) is differentiated with respect to time to evaluate the angular velocity equation of the system, which gives:

$$\omega_2=\frac{c_1}{c_2}\left[\frac{L\dot{y}_2}{(L-y_2)^2}\theta_1+\left(\frac{y_2}{L-y_2}\right)\omega_1\right] \tag{7}$$

Furthermore, Eq. (7) is also differentiated with respect to time to evaluate the angular acceleration of the system, which gives:

$$\alpha_2=\frac{c_1}{c_2}\left\{\left(\frac{L\ddot{y}_2}{(L-y_2)^2}+\frac{2L\dot{y}_2^2}{(L-y_2)^3}\right)\theta_1+\frac{2L\dot{y}_2}{(L-y_2)^2}\omega_1+\frac{y_2}{L-y_2}\alpha_1\right\} \tag{8}$$

where
ω_1,α_1 are the angular velocity and acceleration of the input shaft, respectively,
ω_2,α_2 are the angular velocity and acceleration of the output shaft, respectively,
$\dot{y}_2,\ \ddot{y}_2=$ are the velocity and acceleration of the arm of the alterable transmission ratio respectively.

Eq. (7) as well as Eq. (8) include a term that is function of the angle of rotation of the cam, θ_1. As described previously, the value of this angle changes periodically in the range of $(0\le\ \theta_1\le 2\pi)$. Consequently, a fluctuation occurs in the output angular velocity and acceleration due to this change. However, these terms vanish when $\dot{y}_2,\ \ddot{y}_2$ are equal to zero, i.e. the alterable transmission ratio is adjusted at a particular position to give a specific transmission ratio. This will be illustrated and discussed in the next section.

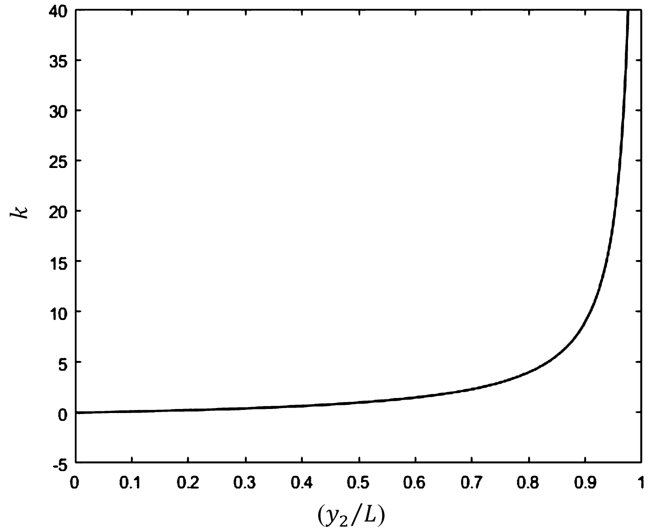

Fig. 6. Variation of the transmission ratio with the dimensionless position of the alterable transmission ratio.

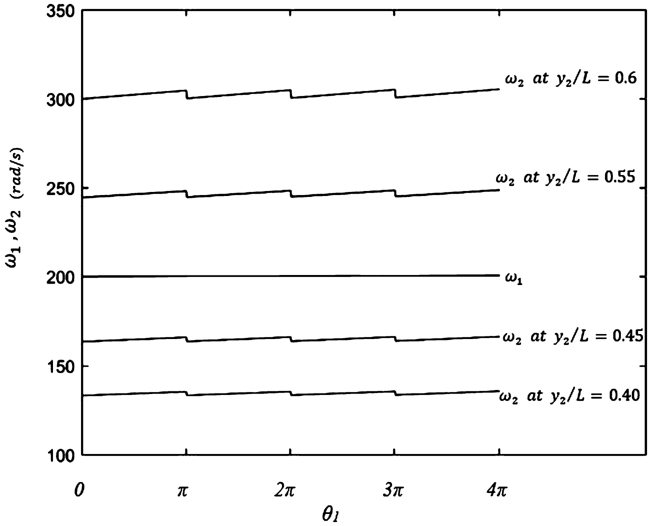

Fig. 7. The input and output angular velocity versus the angular displacement of the input shaft ($\dot{y}_2 = 0.045$).

5. Results and discussion

In the previous section, the equations representing the kinematics of the system were formulated. These equations are used in this section to illustrate the trend of the transmission ratio, the output angular velocity and acceleration against the input variables of the system. The transmission ratio (the ratio between the output and the input angular velocities) is considered as an important parameter in assessing the power transmission systems. In the current system, the transmission ratio k can be formulated from Eq. (7) for a given alterable static position (i.e. $\dot{y}_2 = 0$) as follows:

$$k = \frac{\omega_2}{\omega_1} = \frac{c_1}{c_2}\left(\frac{y_2}{L - y_2}\right) \tag{9}$$

Fig. 6 shows the transmission ratio versus the dimensionless ratio y_2/L.

It can be seen in this figure that the variation of the alterable position along its full range makes the transmission ratio vary from zero to infinity in a continuous manner. This trend does not only satisfy the requirement of the CVT systems, it also satisfies the infinitely variable transmission system requirements, which includes the zero ratio. When $k = 0$, the input shaft is capable of rotating freely, and the cams, the followers, and the slotted links continue to move due to the input shaft rotation. In this case, the actuating rods and the grooved wheels are blocked, while the output shaft is free to rotate in the direction of the ratchet disengagement. The ability of the input shaft to rotate at a transmission ratio of zero

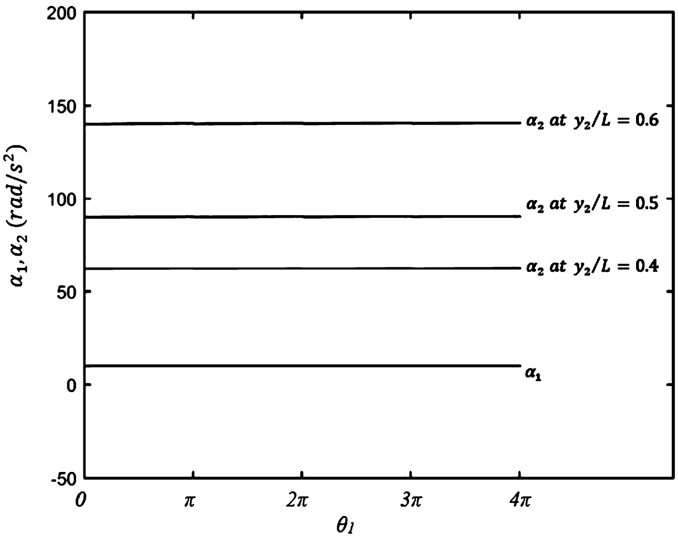

Fig. 8. Input and output angular acceleration versus the angular displacement of the input shaft ($\dot{y}_2 = 0.01$).

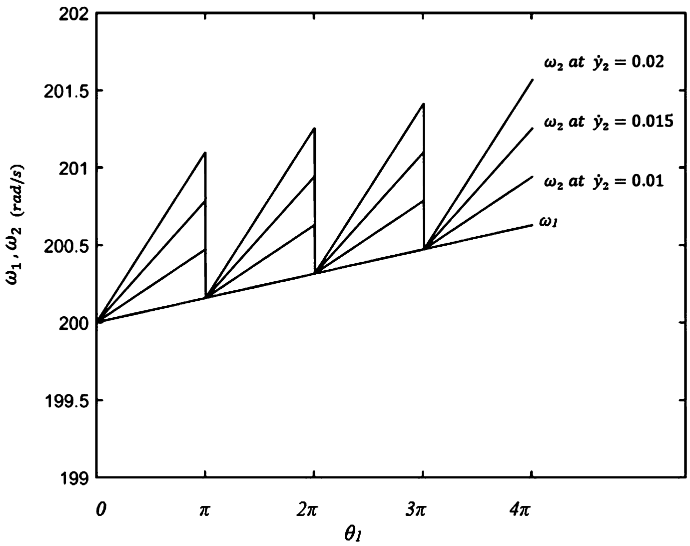

Fig. 9. Fluctuation of the output angular velocity with the angular displacement of the input shaft ($y_2/L = 0.5$).

($k = 0$) is a useful feature in a power transmission system, as it allows the driver motor or the engine to run continuously without disengagement by means of additional mechanism like clutches. On the other hand, when the value of $y_2 = L$, k goes towards infinity, the followers are blocked and lead to the input shaft blockage. In fact, increasing k to a value close to infinity needs a very large length of the slotted links and a large space for their movements. Thus, practically, this case is not applicable.

In order to illustrate the output angular velocity and acceleration of the system, typical inputs of 200 rad/s (angular velocity) and 10 rad/s^2 (acceleration) were considered. Fig. 7 shows the output angular velocity and Fig. 8 shows the output angular acceleration, both against the angle of rotation of the input shaft. In these two figures, the effect of the position of the alterable transmission ratio is considered at particular constant velocities (mentioned in each figure).

Figs. 7 and 8 illustrate the fact that the output angular velocity and acceleration are increased with increasing the value of y_2/L. Furthermore, the output velocity and acceleration have a trend similar to that of the input, but with some fluctuation in the velocity. This fluctuation comes from the fluctuation of θ_1 in the range of $0 \le \theta_1 \le \pi$ in Eqs. (7) and (8) as mentioned previously. Practically, this amount of fluctuation (about 1% for the velocity and 0.1% for the acceleration) might not affect strongly the performance of the system, as it can be eliminated by the effect of the inertia of the moving parts of the system.

The effect of the alterable velocity on the output angular velocity and acceleration is shown in Figs. 9 and 10, respectively for particular values of the alterable position.

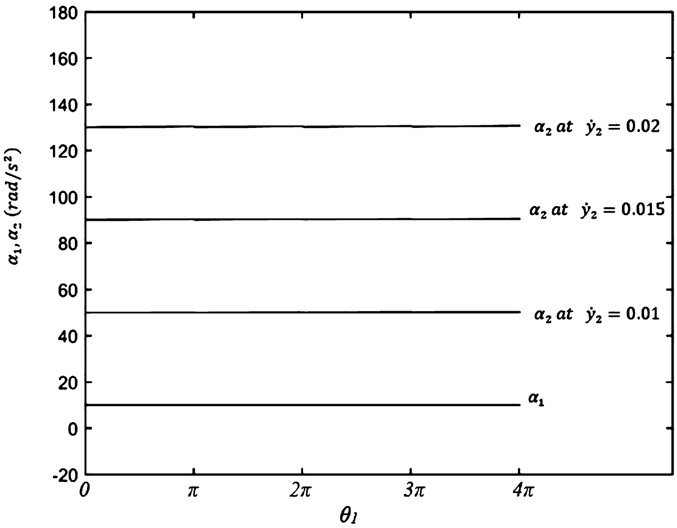

Fig. 10. Input and output angular acceleration for a range of alterable velocity ($y_2/L = 0.5$).

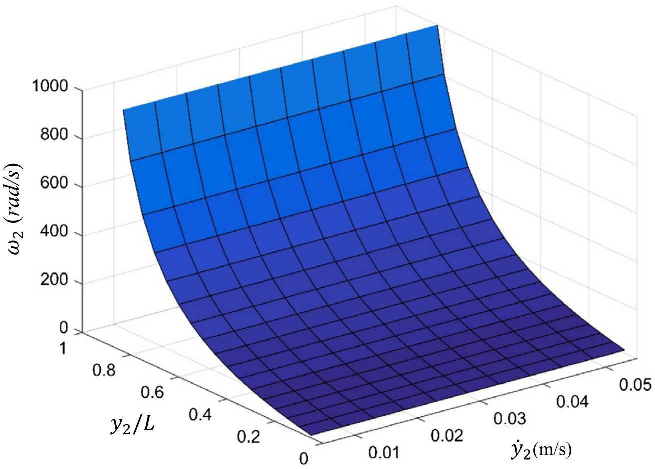

Fig. 11. 3D Illustration of the output angular velocity for a range of alterable position and velocity.

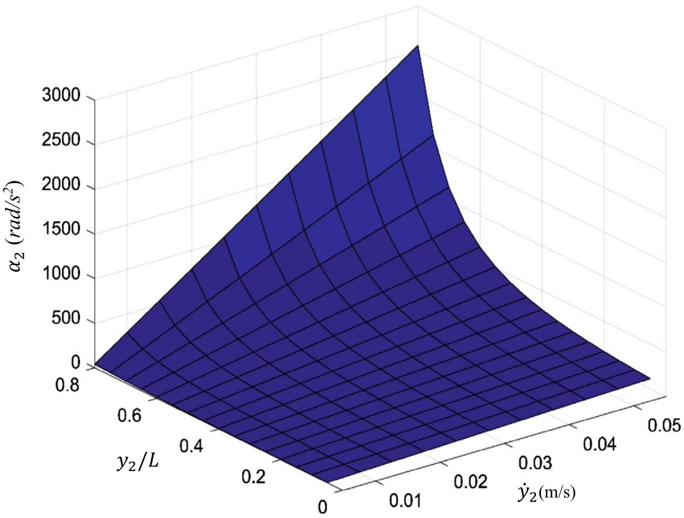

Fig. 12. 3D illustration of the output angular acceleration versus the alterable position and velocity.

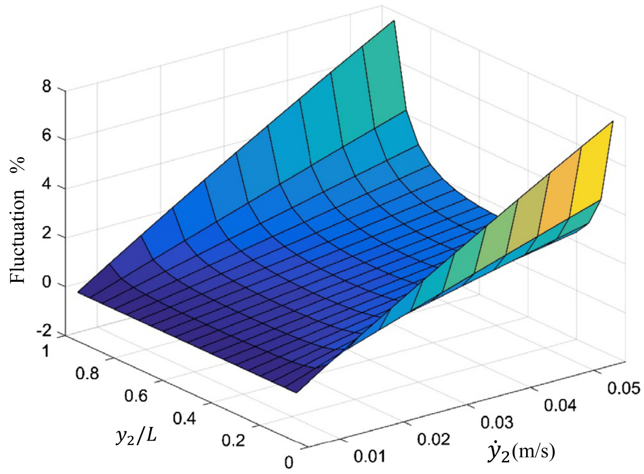

Fig. 13. Percentage fluctuation in the output angular velocity versus the alterable position and velocity.

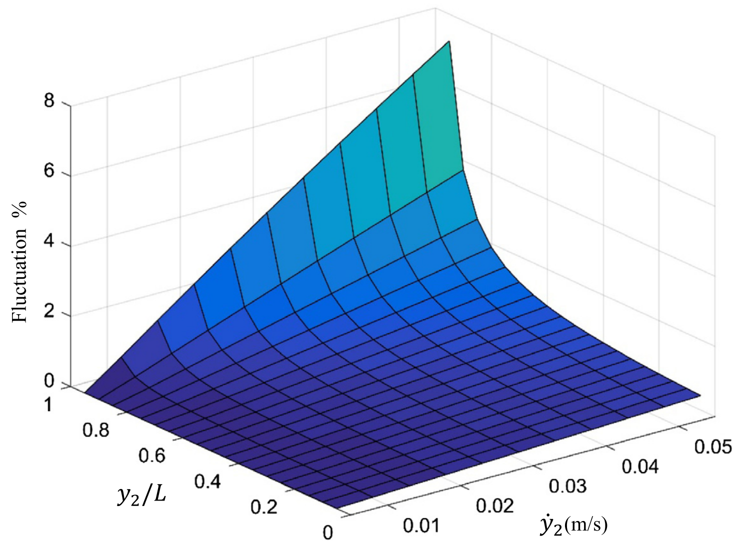

Fig. 14. Percentage fluctuation in the output angular acceleration versus the alterable position and velocity.

Fig. 9 shows that the general trend of the output velocity is not affected by the alterable velocity ($\dot{y}_2$) and only the amount of fluctuation is increased when $\dot{y}_2$ is increased. In this figure, it can be observed that a small amount of fluctuation of the output in a range of approximately 0.15–0.5% is produced for the range of $\dot{y}_2 = 0.01$–0.02 m/s.

Fig. 10 shows that increasing the values of the alterable velocity will increase significantly the amount of the output angular acceleration in comparison with the input angular acceleration. However, the highest amount of fluctuation in the output acceleration is only about 0.23% when $\dot{y}_2 = 0.02$ m/s.

Now, the effects of the position and velocity of the alterable on both the output angular velocity and acceleration are illustrated in a three-dimensional representation, as can be seen in Figs. 11 and 12, respectively. Typical values of input angular velocity of 200 rad/s and angular acceleration of 10 rad/s^2 were considered in these figures.

Figs. 11 and 12 emphasize the results displayed in Figs. 7–10, where the velocity of the alterable has limited effect on the output angular velocity if compared with the alterable position. On the other hand, both position and velocity of the alterable affect significantly the output angular acceleration for the range considered in this study; as can be seen in Fig. 12.

The percentage fluctuation in output velocity and acceleration versus the alterable position and velocity are shown in Figs. 13 and 14 respectively.

These two figures illustrate that the amount of fluctuation in the output velocity and acceleration are equal to zero for an alterable velocity of zero. This behavior results from the fact that the terms including θ_1 (which has a periodical fluctuation) are eliminated in Eqs. (7) and (8) when $\dot{y}_2 = 0$.

It can also be noted in Fig. 13 that the fluctuation is minimum at $y_2/L = 0.5$ along the full range of $\dot{y}_2$. In general, the maximum percentage of fluctuation in both figures does not exceed 8% for the full range of variables considered in these figures.

6. Practical considerations

It is clear that a large number of CVT types have already been invented, designed, and analyzed. However, only theoretical analyses have often been considered, and practical applications of these systems are still limited due to their features and disadvantages. The ultimate friction force is considered as the major limitation in the traction type CVTs (V belt type for example). When this force is exceeded, slippage occurs, and the system does not work anymore.

The CVT system proposed in this paper represents a none-traction type that does not depend on the friction force for power transmission, which is considered as an advantage. Despite this feature, many issues should still be considered and solved before building a prototype. The performance and efficiency of the one-way clutch (ratchet) need to be investigated in order to transmit the required power. Furthermore, the friction losses during the return stroke should be minimized. Additionally, during the change of rotational direction, there will be a small play angle until the ratchet is fully engaged. This angle should also be minimized in order to eliminate the impact forces, which may produce undesired vibrations and unbalancing in the system.

A further major issue that should be considered in the design process is the value of the pressure angle at the contact between the actuating rod and the spiral groove. As the rotational motion of the grooved wheel is initiated from the translating motion of the rod, the slope of the spiral and hence the pressure angle should be as large as possible to maximize the force component that rotates the wheel. In addition, a roller can be fitted at the rod end to reduce the friction force between the rod and the groove profile. This will result in an efficient wheel rotation and will prevent the blocking of the mechanism.

It is worth mentioning that the system involves many connections where sliding relative motion arises during operation. This will increase the power loss due to friction forces in addition to wear, which contributes in parts shape distorting. Another source of geometrical distortion can be related to the elastic deformation due to high stresses during the loading of the system. This can also modify the kinematics of the system. Thus it should be taken into account when designing a real prototype. The analysis presented in this study is mainly focused on the theoretical design and analysis of the operation of the system. Future work will focus on the practical implementation of this system. Therefore, the practical considerations mentioned in this section will be taken into account.

7. Conclusions and remarks

In this paper, a novel ratcheting-type Infinitely Variable Transmission system is presented, the kinematics and operation of the system were modeled. The mathematical modeling of the system shows that the transmission ratio can be changed continuously from zero to infinity by changing the position of the alterable along its full range. The system produces uniform output velocity and acceleration for uniform inputs at particular values of the transmission ratios. On the other hand, during the motion of the alterable transmission ratio i.e. when $\dot{y}_2 \neq 0$, the angular velocity and acceleration are fluctuated as a result of the periodical value of the angular position of the input shaft. However, this fluctuation is small for the typical case considered in this study and might be practically eliminated by the effect of the inertia of the system. In general, the output angular velocity and acceleration are increased when the values of y_2/L is increased. However, the alterable velocity has a very small effect on the output velocity if compared with its effect on the output acceleration. One of the advantages of the proposed system is that it can transfer a large amount of torque as it is of the non-traction type. Furthermore, using a ratchet of selectable direction, the system can produce a reverse rotational motion of the output shaft in addition to forward and zero velocity. Finally, as this study is mainly theoretical, there are many practical considerations that should be studied carefully in order to implement this IVT system.

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