A NOVEL CAM-BASED INFINITLY VARIABLE TRANSMISSION

منظومة ناقل جديدة ذات تغيير لا متناهي تعتمد على أساس عمل الكامة

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الخلاصة

منظومة الناقل ذات التغيير اللا متناهي، هي المنظومة التي تسمح لتغيير مستمر(غير متقطع) في نسبة التحويل من ضمنها نسبة تحويل صفرية.يتضمن هذا البحث تصميم شكلي وتحليل كينماتيكي لمنظومة جديدة تعتمد على اساس عمل الكامة.المنظومة المقترحة تحتوي على عدد من الوحدات المتماثلة، كل وحدة تحتوي على كامة ثلاثية الابعاد مع تابعها، وعجلة ذات اخدود يشبه الكامة، وفاصل ذو اتجاه واحد.

خلال كل وحدة تتحول الحركة الدورانية الى حركة خطية ترددية ذات مدى متغير ومن ثم تنعكس الى حركة دورانية مرة اخرى. في البحث ، تم وضع الصيغ الرياضية للحركة وتمثيل المتغيرات في منحنيات. اضهر التمثيل، اثناء ثبات نسبة التحويل ان المنظومة تعطي مخرجات منتظمة (السرعة والتعجيل الدورانيان) نتيجة لمدخلات منتظمة. من جانب آخر،اثناء تغيير نسبة التحويل ظهر تذبذب قليل في قيم المخرجات وان هذا التذبذب يقل بزيادة عدد الوحدات في المنظومة الواحدة.

Abstract

Infinitely variable transmission system, is the system which allows for continuous (un discreet) variation of transmission ratio including zero transmission ratio. In this paper, the geometric design and kinematics analysis for a novel cam based infinitely variable transmission system are presented. The proposed system has a number of identical units, each unit contains three dimensional cam (camoid), cam follower, grooved wheel, and one way clutch(ratchet). Through each unit the rotational motion is converted to an oscillatory linear motion of variable amplitude and then rectified to rotational motion again. The mathematical expressions for the kinematics analysis are formulated and simulation is established according to this formulation. It was concluded that at fixed transmission ratio, the system produces uniform output for uniform input. On the other hand, during transmission ratio variation there is a little fluctuation in the values of the output angular velocity and acceleration. This fluctuation is decreased when the number of units is increased. Also it was concluded that the maximum transmission ratio of the system depends on the geometric design factors.

Keywords: infinitely variable transmission, cam, grooved wheel.

2 Introduction

Continuously variable transmission system (CVT) is the system which allows for continuous (un discrete) variation of transmission ratio between input and output rotational motion. On the other hand infinitely variable transmission (IVT) is a CVT system which can provide zero transmission ratio. CVT, IVT systems provide an infinite number of transmission ratios within a finite range, when the conventional transmission systems produce a discrete values of transmission ratios by gear shitting. In the automotive engineering there is a great wish to replace the traditional transmission systems (of discrete variation) by active CVT systems to produce a continuous matching between the engine and the driver wheels through transmission ratio variation. This allows for the engine to operate in its efficient speed and there is no power loss during the transmission ratio variation. Hence fuel consumption and gas emissions are minimized, whereas active and comfortable drive is achieved. In addition to automotive application, CVT systems would be applied in the wind turbine electric generator units [1]. This system maximizes the efficiency of the unit by controlling the velocity of the electric generator independently of the turbine velocity(which is variable because it depends on wind speed), so the electric power is produced at the desired frequency.

3 Background

There were early thoughts for the design of CVT systems, but actually until now their application are limited. The most common one is the traction type, in which the power is transmitted by traction force. The push V belt is the more applied traction CVT in which the power is transmitted by a V-belt between two pulleys, each one has a fixed sheave and movable sheave. The value of active radius of each pulley is controlled by changing the gap between the sheaves. Chen et al. [2] present an experimental investigation on the speed and torque loss components of a rubber V-belt CVT. Hyensoo Kim et al. [3] investigated analytically and experimentally the metal V-belt behavior of the V-belt CVT. Carbone et al. [4] study the influence of the clearance between plates in metal V-belt CVT. Also Carbone et al. [5] suggest a theoretical behavior during rapid ratio speed changes in the V- belt CVT. Another traction CVT is the toroidal type in which the power is transmitted purely by friction force between two contact surfaces or by the shear strength of a viscous fluid. Shinji et al. [6] present a new analytical model for determining the elastic deformation and clearance of a the half-toroidal CVT. Belfiore et al. [7] present a feasibility study of special kind of toroidal CVT. Traction based CVT systems have a well known limitations which reduced their usability, like, limited transmitted power and low reliability. Another CVT type is the ratcheting drive, in which the rotational input is converted to an oscillatory motion of varying amplitude then rectified to a rotational output through one way clutches. Benities et all [8,9] present a ratcheting IVT system includes one way clutches and two epicyclical gear system. The above ratcheting drive CVT systems produce non-uniform output for uniform input.

The ratcheting drive cam-based system [10] is differs from the above is that the output is uniform for uniform input, the system uses a 3 dimentional cam, number of cam follower, one way clutches, and epicyclical gear train.

The system presented here is a novel cam based ratcheting IVT which can provide a uniform output for uniform input, using components and configuration that are simpler than that used in [10].

4 Components of the proposed IVT System

The system presented here is a combination of multi identical units; each unit contains the following parts:

a-Three dimensional cam (camoid) fixed on the input shaft of the system. This camoid is a combination of infinite numbers of cam profiles along its axes of rotation each one of these profiles represents a uniform speed cam.

b- Grooved wheel coupled on the output shaft of the system by one way clutch(ratchet). The groove represents a cam profile of uniform speed.

c-Cam follower, which transmit the motion from the camoid to the grooved wheel.

d-Transmission ratio alterable, which changes the transmission ratio by shifting the camoid along the input shaft, this makes the follower to operates at different cam profiles on the camoid.

The IVT system of two units is shown the Fig.1.

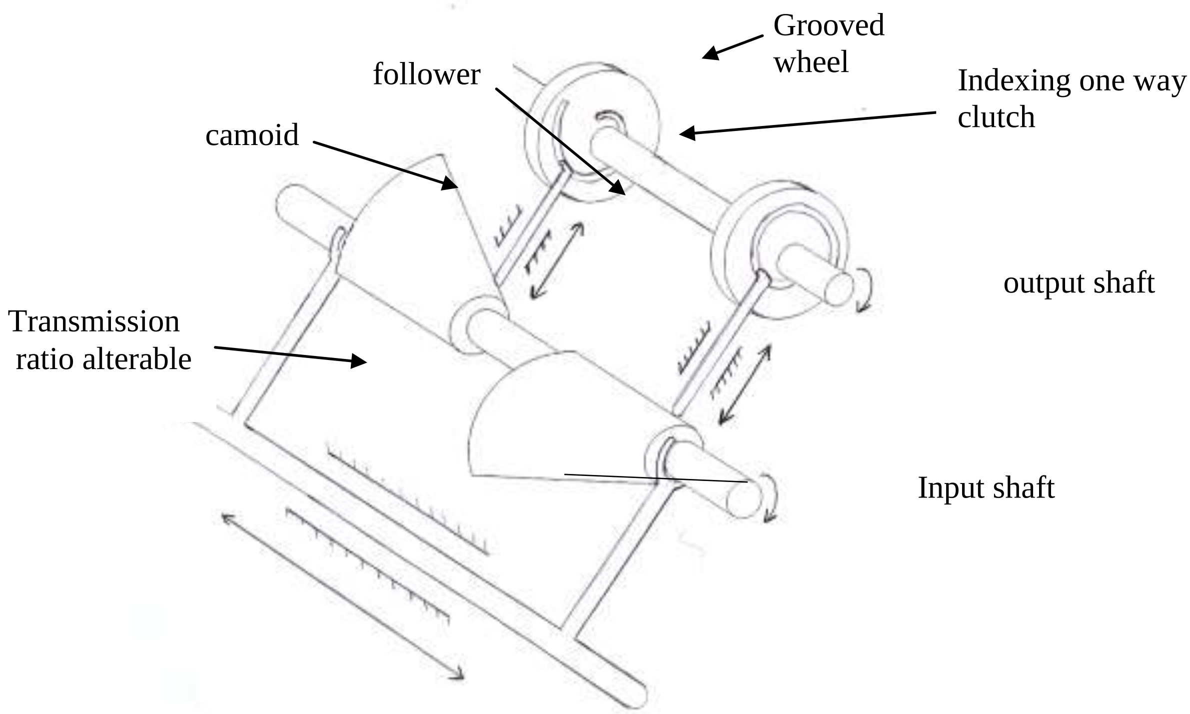

Fig. 1 A schematic diagram of the proposed IVT system contains two units

5 System Operation

During the system operation at specific unite of the system the angular motion of the camoid is converted to an oscillatory linear motion by the follower. The amplitude of this motion depends on the position at the camoid where the follower operates. Then, the oscillatory linear motion is rectified to an oscillatory angular motion of the grooved wheel, then to one way angular motion of the output shaft through the one way clutch (ratchet). The operation of the one way clutch is that it couples the wheel with the shaft in one direction of the angular motion and cancels this coupling in the opposite direction. During the un coupling period, the wheel revolves in opposite direction of the output shaft where the follower returns to its original position by a spring action, at this time the IVT unit doesn't transmit power, therefore multi units are used for keeping continuous power transmission in the system. These unites are configured in a away that at any time there is a follower is transmitting power.

The variation of the transmission ratio is achieved by step less shifting of the camoid along the input shaft. This will change the amplitude of the oscillatory motion of the follower, and then the angular velocity of the output shaft is changed for a fixed value of the angular velocity of the input shaft. During the shifting process, the profile at which the follower operates is changed, and because of that the profiles are embodied continuously to make the camoid, therefore step less shifting is occures.

6 Camoid Profile Design

The three dimensional camoid geometry is a combination of infinite number of cam profiles embodied continuously along its axes of rotation, where uniform speed are presented at each profile. The geometry of the camoid is shown in Fig. 2 .

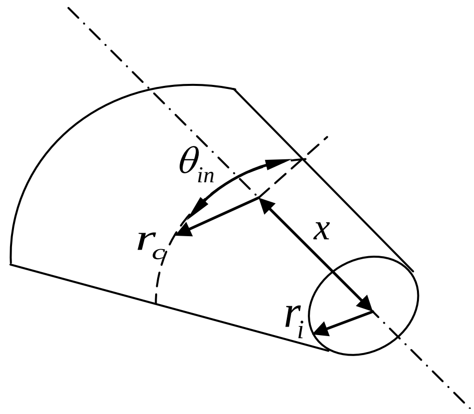

Fig. 2 The geometry of the camoid

Now, we may assume the surface function of the camoid as below

$$r_c = r_i + \theta_{in} \times f(x) \qquad (1)$$

Where

r_c is the radial distance from the axis of rotation of the input shaft

r_i is the radius of the base cylinder of the camoid

θ_{in} is the angle measured from the radial line of profile creation

x is the axial distance

In our work we assume that

$$f(x) = x^m$$

m is a geometric design factor which is constant for specific camoid

Now, the surface function of the camoid becomes

$$r_c = r_i + \theta_{in} x^m \qquad (2)$$

We now derive the above equation with respect to time, noting that r_i is constant

$$\frac{dr_c}{dt} = x^m \frac{d\theta_{in}}{dt} + \theta_{in} m x^{m-1} \frac{dx}{dt} \qquad (3)$$

Then we can write

$\frac{dr_c}{dt} = v_f$ = the velocity of the follower

$\frac{d\theta_{in}}{dt} = w_{in}$ = the angular velocity of the input shaft

$\frac{dx}{dt} = \dot{x}$ = the velocity of transmission ratio alterable

Now Eq. (3) becomes

$$v_f = x^m w_{in} + \theta_{in} m \dot{x} x^{m-1} \qquad (4)$$

Equation (4) represents the velocity of the follower during the operation of the system.

7 Grooved Wheel Design

For the design of the grooved wheel, we use a groove of uniform speed like in the camoid, but it differs from the camoid that it is of two dimensional geometry as shown in Fig. 3

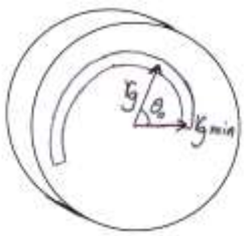

Fig. 3 the grooved wheel

Now the function of the groove at the wheel may be written as following

$$r_g = c\theta_o + r_{g\min} \qquad (5)$$

Where

r_g is the radial distance of the groove from the axis of rotation of the output shaft.

C is a geometric design factor which is constant for particular groove.

θ_o is the angle measured from the radial line of the groove creation.

$r_{g\min}$ is the minimum radial distance of the groove.

The derivation of Eq. (5) with respect to time gives

$$\frac{dr_g}{dt} = c\frac{d\theta_o}{dt}$$

It is clear that $\frac{dr_g}{dt} = v_f$ and $\frac{d\theta_o}{dt}$ is the angular speed of the output shaft, so that the above equation becomes

$$v_f = cw_o \qquad (6)$$

Where w_o is the angular speed of the output shaft.

We now substitute for v_f from Eq. (6) into Eq. (4) to get the general equation of calculating the angular speed of the output shaft.

$$w_o = \frac{1}{c}\left(x^m w_{in} + m\theta_{in}\dot{x}x^{m-1}\right) \qquad (7)$$

We know that, fixing the transmission ratio alterable means $\dot{x}=0$ & x is constant, so the above equation becomes

$$w_o = \frac{x^m}{c} w_{in} \qquad (8)$$

According to the above formula, we conclude that uniform output w_o is achieved from uniform input w_{in} at specific values of *x, c,* and *m.*

By definition, the transmission ratio is the ratio of the output angular velocity of the system to its input angular velocity i.e.

$$k = \frac{w_o}{w_{in}}$$

Where *k* is the transmission ratio

From Eq. (8) and the above equation we can get

$$k = \frac{w_o}{w_{in}} = \frac{x^m}{c} \qquad (9)$$

We can note that putting the value of *x=0* in the above equation gives *k=0* which is the zero transmission ratio of the system.

To evaluate the output angular acceleration, we derive Eq.(7) with respect to time

$$\dot{w}_o = \frac{1}{c}\left[2mw_{in}\dot{x}x^{m-1} + mx^{m-1}\left(\frac{(m-1)\dot{x}^2}{x} + \ddot{x}\right)\theta_{in} + x^m\dot{w}_{in}\right] \qquad (10)$$

Where

$\dot{w}_o$ is the angular acceleration of the output shaft

$\dot{w}_{in}$ is the angular acceleration of the input shaft

$\ddot{x}$ is the acceleration of the alterable.

Equation (10) gives the output angular acceleration of the IVT system

8 Geometric Design Considerations

The geometric design of the IVT system is mainly based on the maximum desired transmission ratio and the space of operation. In the design of this system, the maximum transmission ratio depends on the value of the maximum distance that the followers move, whereas this distance must be compatible with the design of the camoid and the grooved wheel i.e.

$$\text{maximum distance} = (r_c)_{\max} - r_i = (r_g)_{\max} - r_{g\min}$$

Where

$(r_c)_{\max}$ is the maximum radial distance of the camiod.
$(r_g)_{\max}$ is the maximum radial distance of the groove.
For multi units system the above relation becomes

$$\frac{\text{maximum distance}}{n} = \frac{(r_c)_{\max} - r_i}{n} = \frac{(r_g)_{\max} - r_{g\min}}{n}$$

Where *n* is the number of the units

Hence, we can conclude that the increasing of the number of units decreases the transverse space of system for the same value of the maximum transmission ratio.
Groove interference is another limitation in the geometric design of the grooved wheel. This limitation depends on the width of the groove and the value of the geometric constant *C*.

9 Simulation

In this article we will plot the output variables of the system according to the theoretical formula derived for design and operation of the proposed IVT system. The numerical values of the variables that used in the simulation are nearly compatible if this proposed system is used in the small automobiles. The values are given in the appendix.

9-1 Fixed transmission ratio

Fixed transmission ratio means that the alterable has zero velocity and acceleration. For this case, according to Eq.(7) Fig. 4 shows the output angular velocity versus time at constant input angular velocity.

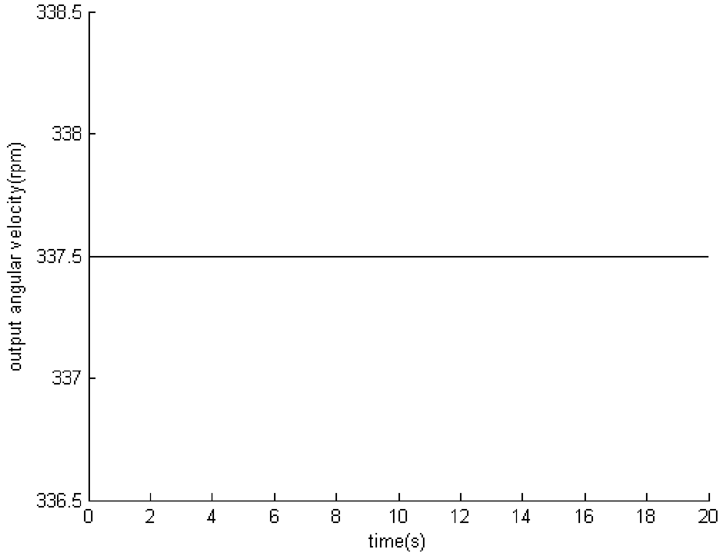

Fig. 4 The output angular velocity versus time at fixed transmission ratio

According to Eq.(10), the output angular acceleration versus time is plotted in Fig. 5 at constant input angular acceleration

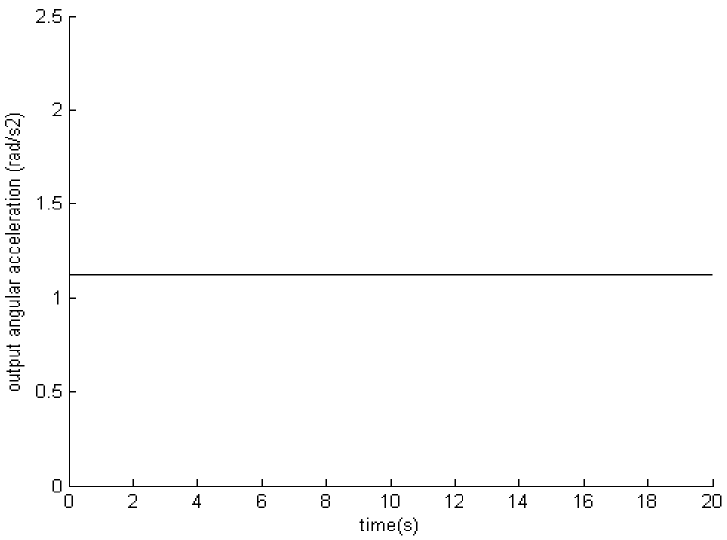

Fig. 5 The output angular acceleration versus time at fixed transmission ratio

9-2 Variable transmission ratio

According to equation (7) the output angular velocity versus the position of the alterable during transmission ratio variation is shown in fig. 6

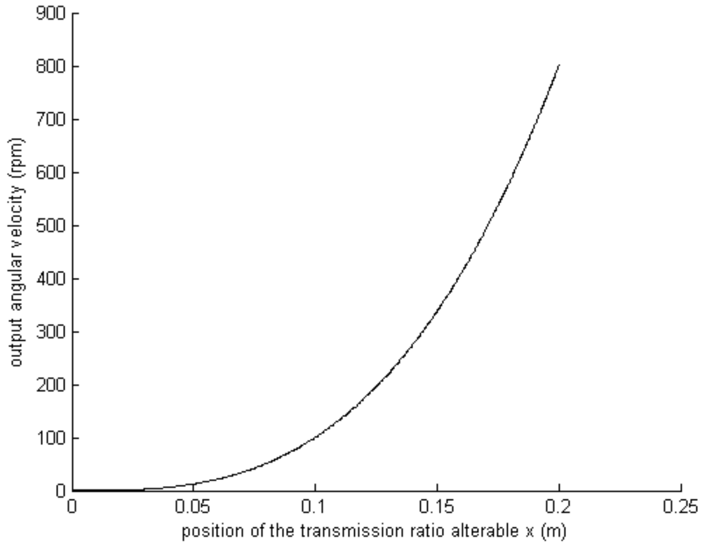

Fig. 6 The output angular velocity versus the position of the transmission ratio alterable

According to Eq.(7), Fig.7 shows the output angular velocity versus time during the transmission ratio variation.

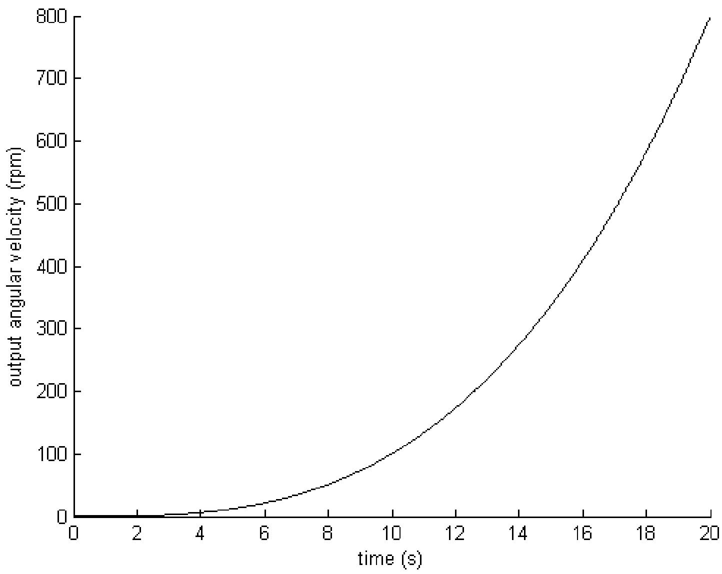

Fig.7 The output angular velocity versus time during transmission ratio variation

If we concentrate our attention on a short period of time at the above figure, we will see (in Fig. 8) small fluctuation of w_o as an effect of θ_{in} fluctuation (from 0 to $\frac{2\pi}{n}$), hence we will see that, increasing the number of units in the system will decrease the amplitude of w_o fluctuation. The output angular velocity versus short period of time during transmission ratio variation is shown in Fig. 8

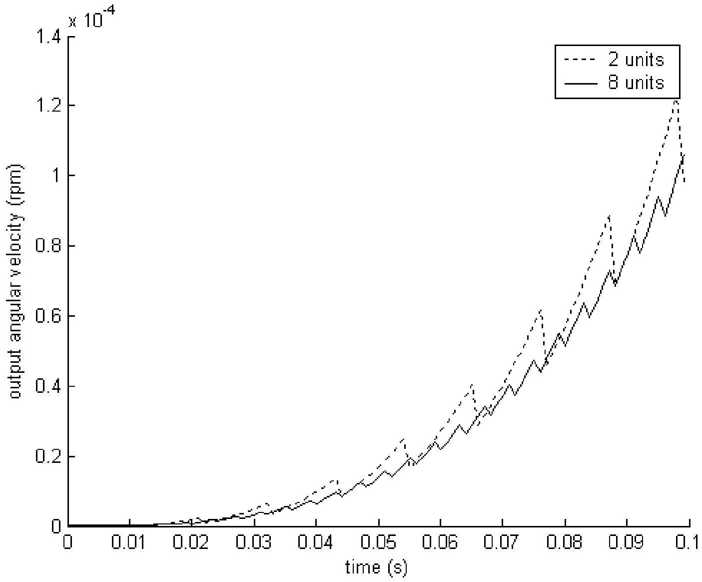

Fig. 8 The output angular velocity for two systems of different number of units during a short period of transmission ratio variation

The output angular acceleration versus the position of the alterable according to eq. (10) is shown in fig. (9).

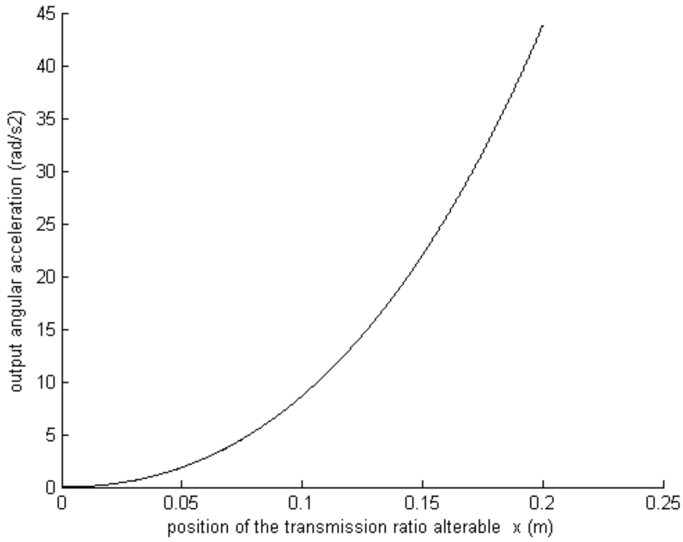

Fig. 9 The output angular acceleration versus the position of the transmission ratio alterable

The output angular acceleration versus time during transmission ratio variation is shown in Fig. 10 (according to Eq. (10))

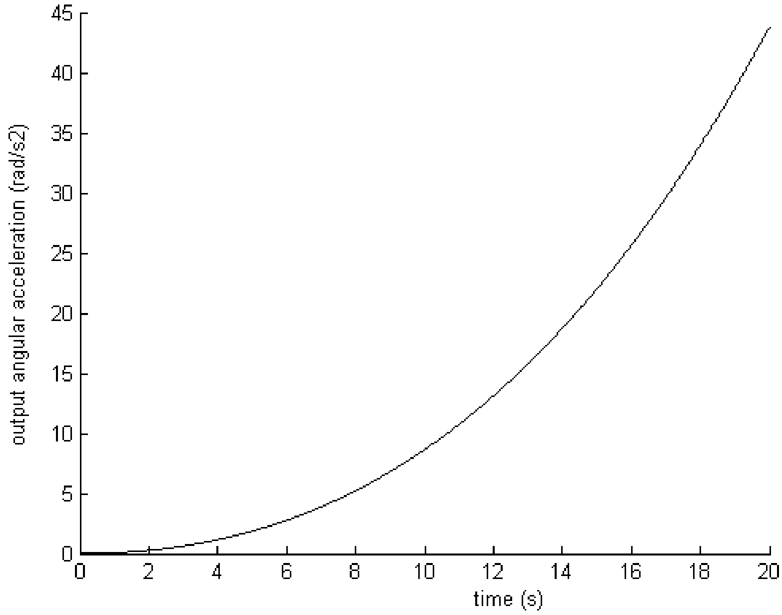

Fig.10 The output angular acceleration versus time during transmission ratio variation

As in the output angular velocity, the concentration on a short period of time will show small fluctuation of $\dot{w}_o$ as a result of θ_{in} fluctuation too. This shown in Fig.11, also the effect of increasing the number of units on the amplitude of fluctuation is shown in this figure.

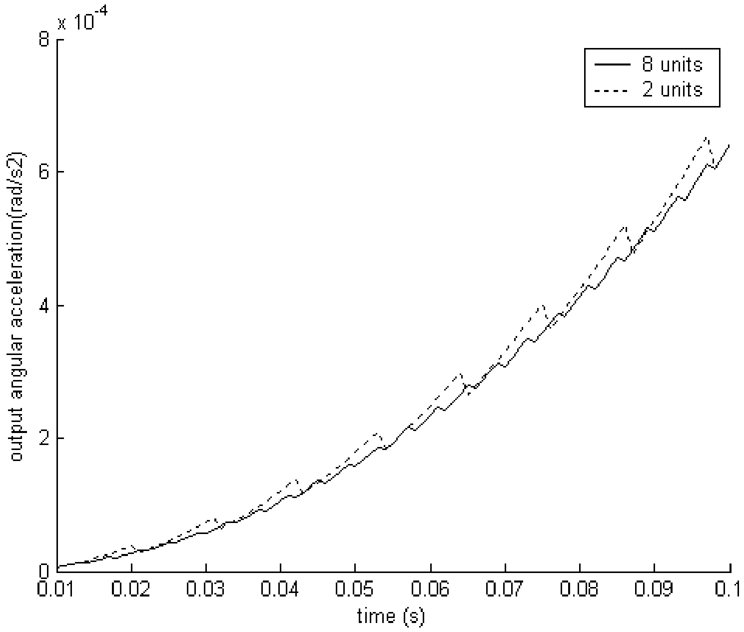

Fig. 11 The output angular acceleration for two systems of different number of units during a short period of transmission ratio variation

9-3 Maximum transmission ratio

The maximum transmission ratio of the system is a design property, which depends on the geometric constants *m*, *c*, and the maximum value of *x* at which the follower reaches on the camoid. According to Eq.(9), Fig. (12) shows the effect of *c* and *m* on the maximum transmission ratio .

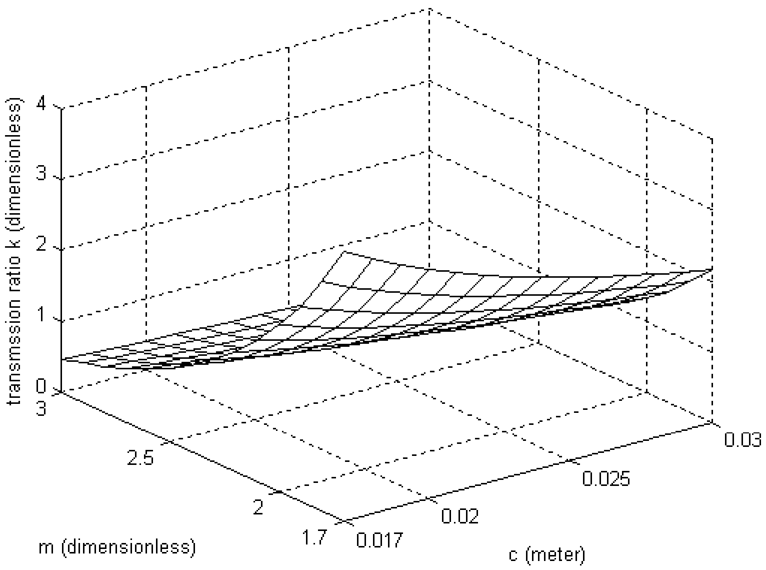

Fig. 12 The maximum transmission ratio versus the geometric constants *c* and *m*

10 R

In studying the performance of any CVT system, the characteristics of two cases of operation are important, fixed transmission ratio, and variable transmission ratio. In this paper the simulation shows, at fixed transmission ratio, uniform input velocity gives uniform output velocity, and uniform acceleration of the input shaft gives uniform output of the input shaft. Actually, fixed transmission ratio means that the alterable is fixed at a constant value of x , hence $\dot{x}$ and $\ddot{x}$ equals to zero. Substitution of these values in equations (7), and (10) gives the uniformity of the output of the system. At variable transmission ratio operation the position of the alterable x changes with time according to its velocity $\dot{x}$ which is taken constant in the simulation, so that $\ddot{x} = 0$. The simulation shows that the main trends of the output velocity and acceleration are increased exponentially during increasing the transmission ratio. Figure (6) shows that the output angular velocity equal to zero at *x=0* where the input velocity is not zero. This point represents zero transmission ratio which is very useful ability of the CVT. Studying small interval of the system outputs shows a small amplitude fluctuation. This fluctuation is an effect of fluctuation of θ_{in} from its maximum value to zero in equations (7), and (10) during switching the active unit. Therefore increasing number of units decreases the maximum value of θ_{in}, hence the amplitude of the output fluctuation is decreased. The output fluctuation did not appear at fixed transmission ratio because that the values of $\dot{x}$, and $\ddot{x}$ equal to zero which made the terms of θ_{in} are equal to zero in equations (7), and (10).

For the system geometric design the maximum transmission ratio is an important category in the CVT systems, which in this system depends on the geometric constants *c*, and *m*. This relation is shown in fig. (12) which is plotted at maximum value of *x*. Figure (12) show that increasing *m* increases the maximum transmission ratio, but increasing c decreases it. The effects of geometric constants on the transmission ratio can be indicated from eq.(9).

11 Conclusions

In this paper a novel cam based ratcheting drive IVT system was presented. A mathematical formulation for the geometric design and kinematics analysis were derived and a simulation was established for the main output variables based on this formulation. The simulation demonstrate that, for fixed transmission ratio, uniform output angular velocity and acceleration was produced at uniform input angular velocity and acceleration respectively. The simulation also demonstrate small fluctuation in the values of the output angular velocity and acceleration during the transmission ratio variation, this fluctuation is a result of fluctuation of the camoid angular position value, so that, increasing the number of units of the system will decrease this fluctuation. Also it was concluded that, increasing of the number of units decreases the transverse space of system for the same value of the maximum transmission ratio.

. In the simulation, it was appeared that the maximum transmission ratio of the system is proportional inversely to the values of the geometric constant *m* and *c* for the range of numerical values used in this paper.

12 Nomenclature

c	a geometric design factor which is constant for specific groove
k	the transmission ratio
m	a geometric design factor which is constant for specific camoid
n	the number of the units
x	the axial distance at the camoid
$\dot{x}$	the velocity of transmission ratio alterable
$\ddot{x}$	the acceleration of the alterable.
r_c	the radial distance of point p from the axis of rotation of the input shaft
r_g	radial distance of the groove from the axis of rotation of the output shaft.
r_i	the radius of the base cylinder of the camoid
$r_{g\min}$	the minimum radial distance of the groove.
$(r_c)_{\max}$	the maximum radial distance of the camiod.
$(r_g)_{\max}$	the maximum radial distance of the groove
v_f	the velocity of the follower
w_{in}	the angular of the input shaft
w_o	the angular speed of the output shaft.
$\dot{w}_{in}$	the angular acceleration of the input shaft
$\dot{w}_o$	the angular acceleration of the output shaft
θ_{in}	The angle of point *p* measured from the radial line of profile creation
θ_o	angle measured from the radial line of the groove creation

13 Appendix

The numerical values of the variables which used in the simulation are assumed as in the table below

Fig. 4	w_{in} =3000 rpm , $\dot{w}_{in}$ =0 , $x=.15m$, $m=3$,$c=.03$ m
Fig.5	$\dot{w}_{in}$ =10 rad/s^2 , $x=.15m$, $m=3$,$c=.03$ m
Fig. 6	w_{in} =3000 rpm , $\dot{w}_{in}$ =0 , $\dot{x}$ =.01 m/s , $m=3$,$c=.03$ m
Fig.7	w_{in} =3000 rpm , $\dot{w}_{in}$ =0 , $\dot{x}$ =.01 m/s , $m=3$,$c=.03$ m
Fig.8	w_{in} (at $t=0$) =3000 rpm , $\dot{w}_{in}$ =10 rad/s^2 , $\dot{x}$ =.01 m/s, $m=3$,$c=.03$ m
Fig.9	w_{in} (at $t=0$) =3000 rpm , $\dot{w}_{in}$ =10 rad/s^2 , $\dot{x}$ =.01 m/s, $m=3$,$c=.03$ m
Fig.10	$x_{\max} = 0.2m$

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