Design And Development Of Double Acting Friction (TAF) Transmission System Using Half Toroidal CVT

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Abstract

A continuously variable transmission (CVT) is a usual transmission that can change seamlessly through a continuous range of effective gear ratios and CVT also known as a single-speed transmission, stepless transmission, pulley transmission, or, in case of motorcycles, a twist-and-go. This contrasts with other mechanical transmissions that offer a set number of gear ratios. The suppleness of a CVT allows the input shaft to maintain a constant angular velocity. A continuously variable transmission (CVT) transfers power throughout a range of speed/torque ratios from engine input towards output, continuously exclusive of interruption. Contrast with either manual or conventional automatic transmissions that make use of discrete ratios as well as normally disengage when changing ratio. The CVT grouping includes infinitely variable transmissions (IVT) so that give a zero output speed within the operating range. The "Half Toroidal" type CVT scheme has basic component arrays. This work documents a successfully developed experiment of a "Half Toroidal" continuously variable transmission (CVT) by adjusting its geometrical arrangement of CVT design as well as compared the experimental results of torque, speed, and power delivered at the output disc with those obtained by a theoretical.

Keywords: "Toroidal drive", infinite speed ratios by toroidal drive, CVT (continuously variable transmission)

INTRODUCTION

The Half Toroidal CVT is an innovative transmission that executes smooth, continuous gear ratio changes by changing the angle of the power rollers between the input disk and output disk. As we know there are various problems with conventional transmission system in automobile industry. Also the efficiency of the system is max 70% and operation is not smooth due to engagement and disengagement of clutch while shifting the gears. Hence there is a need of automotive transmission system with better efficiency, smooth and simple changing of gear ratio, we are going to use a Continuous Variable Transmission System. From environment saving point of view the analysis performed seems to indicate that CVT adoption could produce a certain reduction of polluted emissions and a lower level of noise. All this features can be got because of the possibility offered by the CVT of changing the speed ratio in a continuous way under load conditions. The Half Toroidal is one of the major types of Continuous Variable Transmission System. The first toroidal drive is patented in 1877 by C. W. Hunt. In last 30 years there is a significant improvement in the fields of material, lubrication fluids, tribology and control

I.WORKING PRINCIPLE OF HALF TOROIDAL CVT

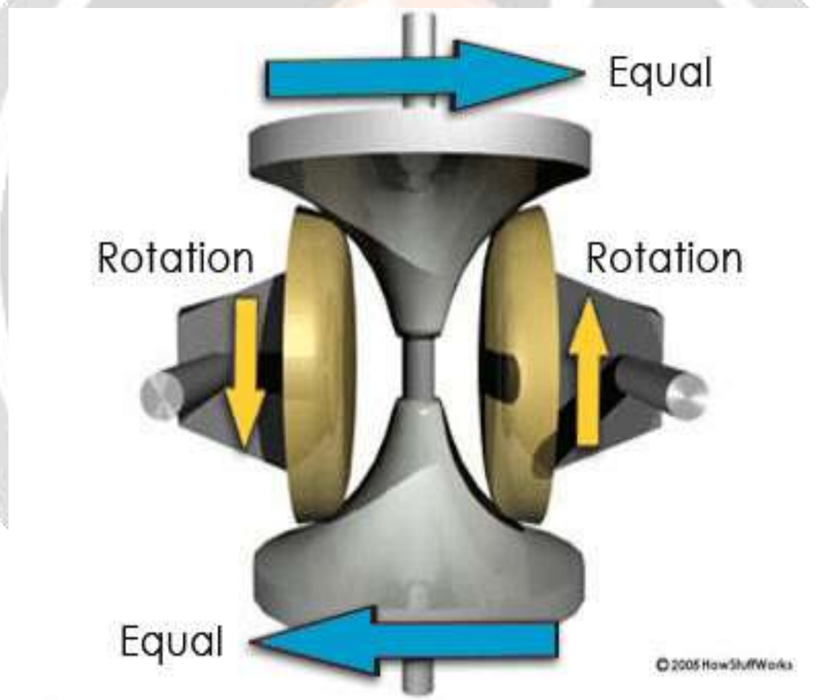
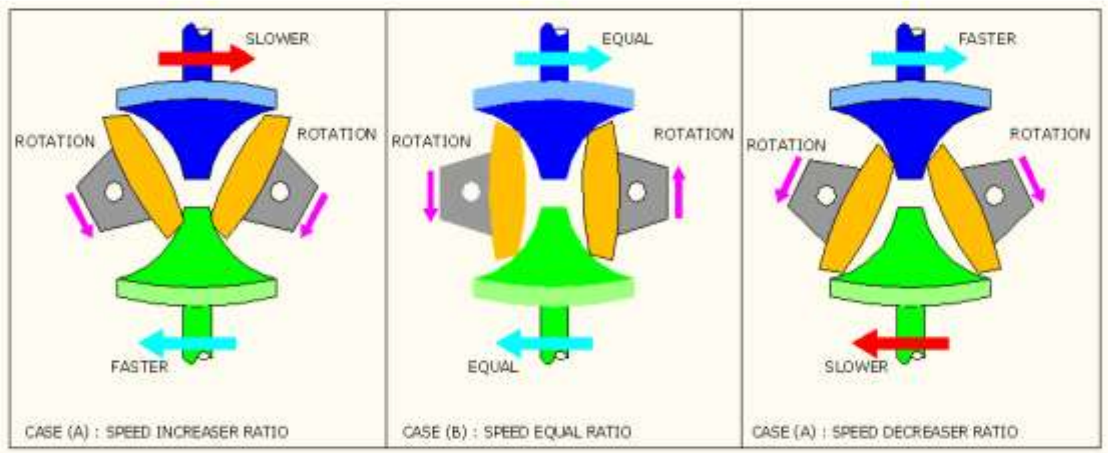


Fig.Half Toroidal CVT

The Half Toroidal CVT is an innovative transmission that executes smooth, continuous gear ratio changes by changing the angle of the power rollers between the input disk and output disk.

1. One disc connects to the engine. This is equivalent to the driving pulley.
2. Another disc connects to the drive shaft. This is equivalent to the driven pulley.
3. Rollers, or wheels, located between the discs act like the belt, transmitting power from one disc to the other.



The wheels can rotate along two axes. They spin around the horizontal axis and tilt in or out around the vertical axis, which allows the wheels to touch the discs in different areas. When the wheels are in contact with the driving disc near the center, they must contact the driven disc near the rim, resulting in a reduction in speed and an increase in torque (i.e. low gear). When the wheels touch the driving disc near the rim, they must contact the driven disc near the center, resulting in an increase in speed and a decrease in torque (i.e. overdrive gear). A simple tilt of the wheels, then, incrementally changes the gear ratio, providing for smooth, nearly instantaneous ratio changes.

II.DESIGN AND CALCULATION

1. *Motor Selection*

Thus selecting a motor of the following specifications:

Single phase AC motor

Commutator motor

TEFC construction

Power = 120 watt

Speed= 0-6000 rpm (variable)

2. *Design of Input Shaft*

Material – high grade steel (EN 24)

ASME CODE FOR DESIGN OF SHAFT

Sut= 800 N/mm^2

Syt=680 N/mm^2

T all=204 N/mm2

T all=144 N/mm2

consider 25% over load

0.75 *144=108 N/mm2

assume 100% efficiency

Tdesign =1019*10^3

select the diameter of the shaft =18mm

Td=3.14/16*T actual*d^3

T act=1.03 N/mm^2

T actual < T all.

inputshaft is safe under the torsional load.

3. *Design of spur gear*

power =120 watt

speed=6000rpm

b=8m

No.of teeth of pinion=18

No.of teeth of gear=90

i=5

gear speed=600/5=1200

material of pinion

and gear plain carbon steel (40c8)

Sut=600 N/mm^2

fos=1.5

considering 25% overload

Tdesign=1.25T =1.09*10^3 N/mm^2

Dg=90

Peff=219.38N

Selecting standard module=1.0mm

4. *Design and calculation of input toroidal hub*

Material high grade steel (EN24) (ref.v.b.bhandari)

Sut=800N/mm^2

Syt=680N/mm^2

Outside diameter =25mm

Inside diameter =20mm

Tall=204N/mm^2

Tact=144N/mm^2

T design=1.09*10^3

T actual =0.65 N/mm2

T actual<T allowable

Actual shear stress is less than allowable shear stress.

Design of inpur toroidal hub is safe under ther torsional load

5. *Design of output shaft*

DESIGNATION	ULTIMATE TENSILE STRENGTH N/mm^2	ULTIMATE TENSILE STRENGTH N/mm^2
EN 24	800	680

6.Design of key

Selecting parallel key from standard data book for given application. (ref. v.b.bhandari)

DESIGNATION	ULTIMATE TENSILE STRENGTH N/mm^2	ULTIMATE TENSILE STRENGTH N/mm^2
EN 24	*520*	*340*

FOR SHAFT DIAMETER	ABOVE UPTO	17 22
KEY CROSS SECTION	WIDTH HEIGHT LENGTH	6 6 30

T allowable =102N/mm^2

T actual=0.73 N/mm^2

Actual shear stress is less than allowable shear stress hence key is safe under shear stress.

T design = 1.19*10^3 N/mm (100% efficiency)

T C*d/2*w/2*t actual

T actual =1.46 N/mm^2

T actual < T allowable.

Key is safe under crushing load.

6. Design of output toroidal hub

Outside diameter =25mm

Inside diameter =18mm

DESIGNATION	ULTIMATE TENSILE STRENGTH N/mm^2	ULTIMATE TENSILE STRENGTH N/mm^2
EN 24	800	680

T allowable=144

T design =1019*10^3 N/mm

T actual= 0.53 N/mm^2

T actual< T allowable

Actual shear stress is less than allowable shear stress hence hub is safe under torsional load.

7. *Design of power roller*

DESIGNATION	ULTIMATE TENSILE STRENGTH N/mm^2	ULTIMATE TENSILE STRENGTH N/mm^2
EN 24	800	680

Outside diameter of roller (mm)	46
Inside diameter of roller(mm)	35
Contact cone angle (degree)	62.2
Number of power roller(n)	2
Swing angle of power roller(degree)	27.5 & 97.5
Maximum input torque T max(Nm)	1019*10^3
Maximum input speed N max (rpm)	6000
Speed ratio (G)	0.427 & 2.34

T allowable =144 N/mm^2

T actual = 0.09 N/mm^2

T actual < T allowable

Actual shear stress is less than allowable shear stress and hence power roller is safe under torsional load

8. *Selection of belt*

The most commonly used power-transmitting device in a belt-type CVT is either a steel V-belt or a rubber V-belt. A lot of relevant work on metal and rubber V-belts is subsequently cited as such CVTs are extensively researched by today's automobile manufacturers

and scientists. Most existing models of belt CVTs, with a few exceptions, are steady-state models that are based on
the principles of quasi-static equilibrium. Gerbert [17,18] extensively work on understanding the mechanics of traction belts,
especially metal pushing V-belts and rubber V-belts. The author used quasi-static equilibrium analysis to develop a set of
equations that capture the dynamic interactions between the belt and the pulley. Since the belt is capable of moving both
radially and tangentially, variable sliding angle approach was implemented to describe friction between the belt and the

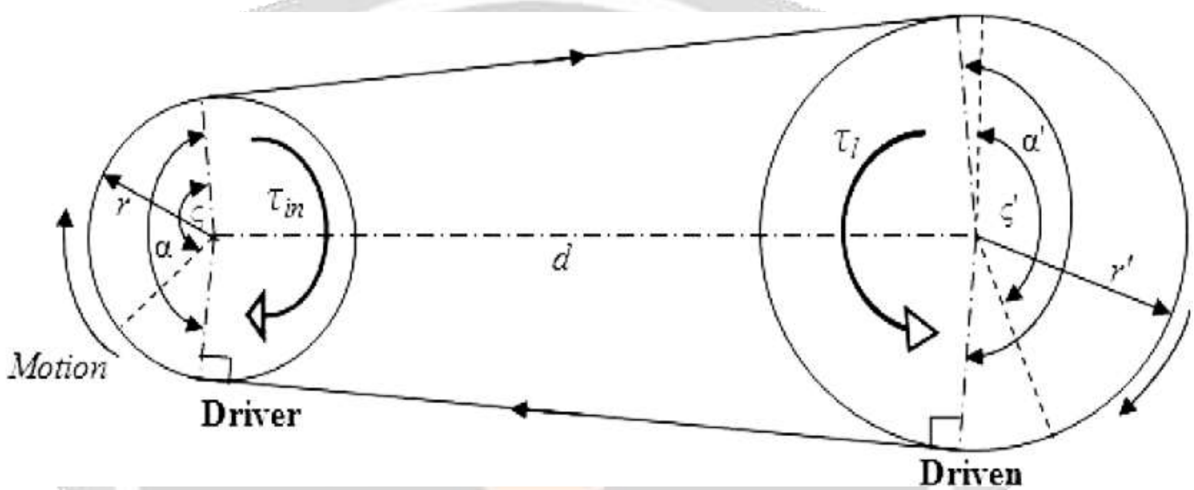


Fig.Geometrical Desciption Of Belt CVT Drive.

III.TEST AND TRIAL

To conduct trial
a) Torque Vs Speed Characteristics
b) Power Vs Speed Characteristics
c) Efficiency Vs speed characteristic

In order to conduct trial, an dynobrake pulley cord, weight pan are provided on the output shaft.
A. Input Data
1) Drive Motor
Power ,120 watt
0.5 Amp,
0 TO 6000 RPM
TEFC MOTOR
2) Diameter (Effective) of Dynobrake pulley = 65 mm.

B. Procedure
1) Start motor by turning electronic speed variator knob.
2) Let mechanism run & stabilize at certain speed (say 660 rpm)
3) Place the pulley cord on dynobrake pulley and add 100 gm weight into, the pan , note down the output speed for this load by means of tachometer.
4) Add another 10 gm weight & take reading .
5) Tabulate the readings in the observation table.

IV.CALCULATION AND RESULT

1.Model Calculations:

1 .For load = 0.1kg, P=2.20 watt N= 660

$P = 2\prod NT_{inp}/60$

$2.20 = 2\pi \times 660 \times T_{inp} /60$

$T_{inp} = 0.031$ N-m

Input Power ($P_{i/p}$) = $2\prod NT_{inp} / 60$

$= 2\pi \times 660 \times 0.031 /60$

Input Power **(**$P_{i/p}$) = 2.14 watt

Output Torque ($T_{o/p}$) = Weight pan x Radius of pulley

$= (0.1 \times 9.81) \times 40$

= 39.25 N-mm

$T_{o/p} = 0.03925$ N-m

Output Power ($P_{0/p}$)= $2\prod NT_{o/p}/60$

$=2\prod \times 500 \times 0.03925/60$

($P_{o/p}$) =2.05 watt

Efficiency = (Output power / Input power) x 100

η = (2.05 /2.14)

η =93.%

V.RESULT TABLE

Sr.no	Load (kg)	Power (watt)	Speed (rpm)	Input Torque (N.m)	Input power (watt)	Output speed (rpm)	Output torque (N.m)	Output Power (watt)	Efficiency
1.	0.1	2.20	660	0.031	2.14	500	0.03925	2.05	93%
2.	0.2	4.34	651	0.063	4.29	490	0.078	4.0023	93%
3.	0.3	6.41	640	0.095	6.36	480	0.117	5.88	92%
4.	0.4	8.14	610	0.1224	8.13	460	0.156	7.51	92%
5.	0.5	10.50	629	0.159	10.4	440	0.196	9.031	86%

VI.CONCLUSION

For contacts of the half toroidal CVT, the critical point of the maximum Hertzian stress is at the maximum deceleration position, i.e., at the initial rotation angle of power roller. The different system parameters play the major role in developing the Hertzian stress in the contacts of the half toroidal CVT. For systems with high input torques, the average maximum Hertzian stresses show higher values than those having low input torques within the speed ratio range. While, the average maximum Hertzian stresses show lower values for systems with high values of aspect ratios, curvature ratios, power roller cone angles and maximum traction coefficient. In addition, the selection of the combination material with a moderate modulus of elasticity is preferable which reduce the maximum Hertzian stress to avoid fatigue failure. Useful graphs relating with material, operating and geometrical parameters are of help for a designer to determine the maximum Hertzian stresses within the speed ratio range in such systems.

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