UTIMATE
TRANSMISSIONS

TECHNOLOGY 3

Double Roller Full Toroidal Variator

Contents

Power Density

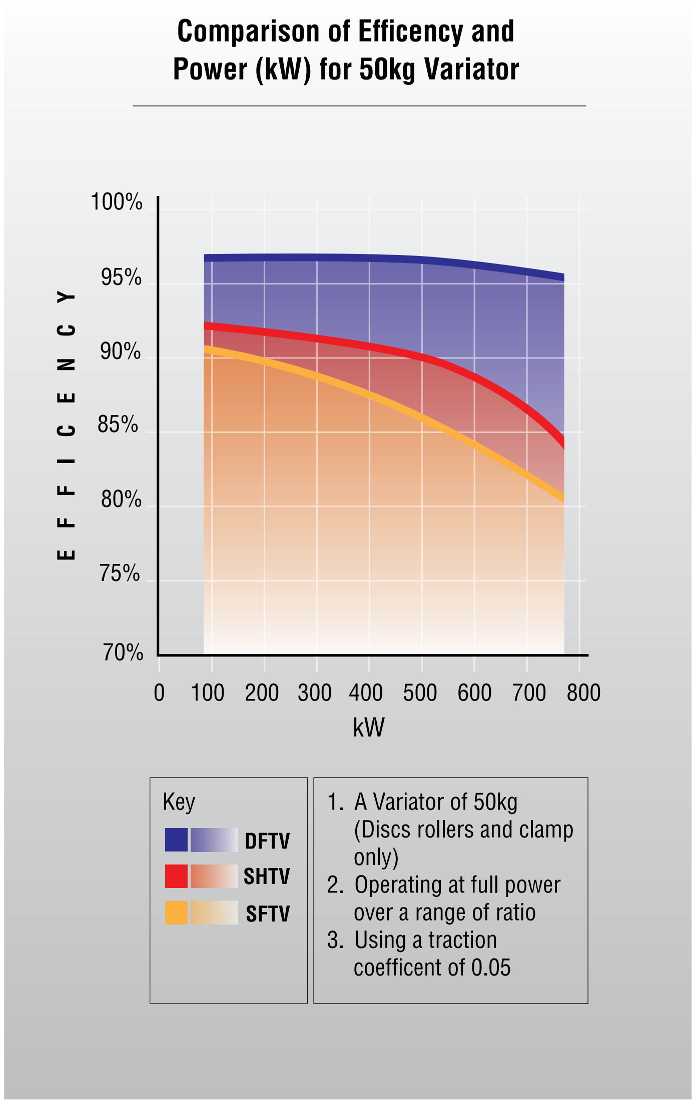
The Power density of a DFTV is significantly higher than either the SFTV or the SHTV. It is also significantly higher than a similar capacity Push belt or Pull Chain CVT.

The diagram (right) maps the efficiency and input power envelopes for the three toroidal mechanisms. The efficiency and power density of a typical push belt CVT is similar to that of a SHTV.

A mathematical study of the power density of the DFTV design has been carried out that forms the basis of design software that can also be used to design working CVT's.

This study is comparative in nature using known design parameters used in the design of SFTV's and SHTV's.

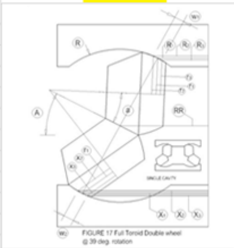
The reasons for this improvement are explained in detail later in this paper.



Power Density

The reason for the dramatic improvement in power density is associated with four main factors. These all stem simply from the geometry of the DFTV design and have nothing to do with any new form of mechanical system or mechanical action.

1. Much larger contact patch area, allowing higher clamping forces and more torque transfer
2. Larger lever-arms that deliver higher torque values for the same clamping pressures.
3. More rollers within the cavity made possible because of their smaller size.
4. Greater conformity of contact radii made possible because of the inclination of the rollers to the central axis.

CALCULATION OF SPIN LOSSES MAXIMUM INPUT TORQUE AND MECHANICAL VOLUME FOR TORQUE CONTROLLED VARIATOR DOUBLE ROLLER																		
The following model can be used to calculate 1) the maximum spin losses, 2) The maximum ratio, 3) Mechanical Volume 4) Maximum input torque. 5) Estimated weight																		
Primrary inputs are shown as						*Secondary inputs from Hertzian contact calculator are shown as*									*Calculated Outputs*			
INPUTS																		
Radius of centre of Toroid						RR	67.07	mm.		INPUT RPM					6000	RPM		
Transverse radius of Toroid						R	50.92	mm.	Width of contact point on input					W1	9.85	mm.		
Wheel inclination						a	21.70	deg	Width of contact point on output					W2	9.36	mm.		
Radius of roller						r1	24.80	mm.		**OUTPUT RPM**					**2,561**	RPM	*0.42675328*	
Primrary rotation angle						A	31.95	deg		**ROLLER RPM**					**9,707**	RPM	25.20	m/sec
Radius of crown of roller							38.70	mm.		**TOTAL RATIO CHANGE**					**5.49**			
Conformity of wheel to cavity							76.00	%	Area of inner contact point						**19.21**	sq.mm		
Length of input contact point						L1	2.48	mm.	Maximum Hertzian Stress						**3.85**	Gpa	check in	check out
Number of wheels							4	rollers	Maximum stress on inner point from Hertzian table						**49,514**	N	49,314	49,695
Number of cavities							1	cavity	Traction coefficient						**0.055**			
Length of output contact point						L2	3.2016		maximum reaction force on roller bearings						5,447	N		
	R1	R2	R3	r1	r2	r3		Sin A	Cos A	Sin a	Cos a	Size factor	Total clamping force		198,058	N		
	40.1211469	44.3014173	35.9408765	24.7986	26.6202	22.9770		0.529179	0.84851021	0.36974676	0.92913257	1.38			stress under output contact			3.17
	X1	X2	X3	r1	x2	x3		Sin A	Cos A	Sin a	Cos a		Overall radius of transmission without actuator					107.07
	94.0148531	97.9843536	90.0453526	24.7986	26.5283	23.0689		0.529179	0.84851021	0.36974676	0.92913257		Overall length of transmission without hydraulics					165.60
													Overall length between outside of discs					122.21
							Speed dif	% loss		Spin for this position			MAXIMUM INPUT TORQUE				spin	
Speed at	R1	1,511,765		r1	1,511,765	mm/minute	0						**437.05**			Nm		
Speed at	R2	1,669,277		r2	1,622,812		46,465	3.07%		**5.83%**								
Speed at	R3	1,354,252		r3	1,400,717		-46,465	3.07%		ESTIMATED MASS			ESTIMATED VOLUME					
										35.05		kg	**7.15**			litres		
Speed at	X1	1,511,765		r1	1,511,765		0			% of SFTV			4.40	Discs and rollers only		litres	torque density	99.340317
Speed at	X2	1,575,595		x2	1,617,213		-41,619	2.75%		**63.82%**			KW					
Speed at	X3	1,447,935		x3	1,406,316		41,619	2.75%					**274.58**					
										SPIN LOSSES								
CALCULATION OF APPARENT RADIUS OF CONTACTING POINTS										Spin loss in "L" position				5.83%		ENERGY LOST TO SPIN		
Angle of inclination of circle of contact of toroidal disc						A	31.95	deg		Spin loss in "LM" position				3.58%		2.23%	factor	0.675
Angle of inclination of circle of contact of roller						a	21.70	deg		Spin loss in "C" position				0.00%		ENERGY LOST TO BEARING		
Radius of input circle of contact on toroidal disc							75.82	mm.		Spin loss in "HM" position				3.19%		0.00%		
Radius of output circle of contact of toroidal disc							-177.66	mm.		Spin loss in "H" position				7.17%		TOTAL LOSS		
Radius of contact of (toroidal) roller surface							26.69	mm.		Average spin loss				**3.30%**		2.23%		
										VOLUME OF OPERATIONAL CAVITIES				4.45	litres			

Model of DFTV with input torque of 430Nm

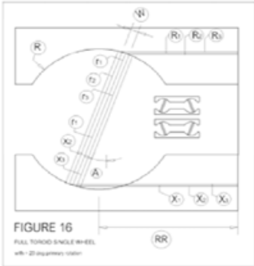
Above is an extract from the mathematical model that maps all of these geometrical interactions in a DFTV and compares them to the geometry of a SFTV.

Power Density

CALCULATION OF SPIN LOSSES MAXIMUM INPUT TORQUE AND MECHANICAL VOLUME FOR TORQUE CONTROLLED VARIATOR SINGLE ROLLER

The following model can be used to calculate 1) the maximum spin losses, 2) The maximum ratio, 3) Mechanical Volume 4) Maximum input torque. 5) Estimated weight

Primrary inputs are shown as (yellow) — *Secondary inputs from Hertzian contact calculator are shown as* (orange) — *Calculated Outputs* (green)



INPUTS									
Radius of centre of Toroid	RR	58.76	mm.	INPUT RPM		6000	RPM		
Transverse radius of Toroid	R	51.99	mm.	Width of contact point on input	W1	5.03	mm.		
Single roller	a			Width of contact point on output	W2	4.40	mm.		
Radius of roller	r1	51.99	mm.	**OUTPUT RPM**		**2,561**	RPM	*0.42686038*	
Primrary rotation angle	A	27.00	deg	**ROLLER RPM**		**4,057**	RPM	22.08 m/sec	
Radius of crown of roller		25.99	mm.	**TOTAL RATIO CHANGE**		**5.49**			
Conformity of wheel to cavity		50.00	%	Area of inner contact point		**14.13**	sq.mm		
Length of input contact point		3.58	mm.	Maximum Hertzian Stress		**3.95**	Gpa	check in	check out
Number of wheels		3	rollers	Maximum stress on input point from Hertzian table		**37,099**	N	37,206	37,127
Number of cavities		2	cavity	Traction coefficient		**0.055**			
Length of output contact point		5.46	mm.	maximum reaction force on roller bearings		4,081	N		

R1	R2	R3	r1	r2	r3	Sin A	Cos A	Sin a	Cos a	Size factor	Total clamping force	111,297
35.155796	37.396428	32.915164	51.9870	51.9870	51.9870	0.4539905	0.8910065	0	1	1.209	stress under output contact	2.95
X1	X2	X3	r1	x2	x3	Sin A	Cos A	Sin a	Cos a		Overall radius of transmission without actuator	93.80
82.359004	84.319557	80.398451	51.9870	51.9870	51.9870	0.4539905	0.8910065	0	1		Overall length of transmission without hydraulics	338.13

					Speed dif	% loss
Speed at	R1	1,324,670	r1	1,324,670	0	
Speed at	R2	1,409,097	r2	1,324,670	84,427	6.37%
Speed at	R3	1,240,243	r3	1,324,670	-84,427	6.37%
Speed at	X1	1,324,670	r1	1,324,670	0	
Speed at	X2	1,356,204	x2	1,324,670	31,534	2.38%
Speed at	X3	1,293,137	x3	1,324,670	-31,534	2.38%

Total spin loss	**8.75%**		
MAXIMUM INPUT TORQUE	**430.40**	Nm	9.34101705
ESTIMATED MASS	**54.93**	kg	
ESTIMATED VOLUME	**11.21**	litres	
KW	**270.41**		

CALCULATION OF APPARENT RADIUS OF CONTACTING POINTS			
Angle of inclination of circle of contact of toroidal disc	A	27.00	deg
Angle of inclination of circle of contact of roller	a	0.00	deg
Radius of input circle of contact on toroidal disc		77.44	mm.
Radius of output circle of contact of toroidal disc		-181.41	mm.
Radius of contact of (toroidal) roller surface		51.99	mm.

SPIN LOSSES	
Spin loss in "L" position	8.75%
Spin loss in "C" position	6.63%
Spin loss in "H" position	6.57%
Average spin loss	**7.32%**
VOLUME OF OPERATIONAL CAVITIES	8.01 litres

ENERGY LOST TO SPIN	4.94%	factor	0.675
ENERGY LOST TO BEARING	0.00%		
TOTAL LOSS	4.94%		

Model of DFTV with input torque of 430Nm

It can be seen that the predicted torque density of this model for a DFTV is 187Nm/litre.
This is compared to a SFTV at 38.4Nm/litre – Almost five times.

A similar analysis has been done for a SHTV.

Power Density

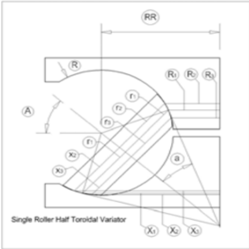
CALCULATION OF SPIN LOSSES MAXIMUM INPUT TORQUE AND MECHANICAL VOLUME FOR TORQUE CONTROLLED VARIATOR SINGLE ROLLER HALF TOROID

The following model can be used to calculate 1) the maximum spin losses, 2) The maximum ratio, 3) Mechanical Volume 4) Maximum input torque. 5) Estimated weight

Primrary inputs are shown as (yellow) *Secondary inputs from Hertzian contact calculator are shown as* (orange) *Calculated Outputs* (green)

*NOTE - It is possible to input without preparing a drawing provided the following rules are followed. The Roller radius is calculated after inputting the "included angle". The number of wheels can be inputted with some knoledge of the geometry. The number of cavities is a initial decision. Hertzian stress is a function of experience/life. The maxamium allowable "spin" is related to acceptable values and in the case of the Half Toroid occurrs in the centre position. The volume is calcutated as Transverse radius of Toroid * 2 * X1 * X1* π** a Constant developed from real designs. The contact elypse and maximum allowable stress are calculated using http://www.tribology-abc.com/calculators/e2_2.htm . The Mass is calculated as volume by a machine density of 4.9

INPUTS



Radius of centre of Toroid	57.60	mm.	INPUT RPM		6000	RPM		
Transverse radius of Toroid	35.04	mm.	Width of input contact point	W1	8.97	mm.		
Included angle	26.88	deg	Width of output contact point	W2	8.01	mm.		
Radius of roller	31.25	mm.	**OUTPUT RPM**		**2,240**	RPM		
Primrary rotation angle	42.00	deg	**WHEEL RPM**		**4,783**	RPM		
Radius of crown of roller	28.03	mm.	**TOTAL RATIO CHANGE**		**5.36**			
Conformity of roller to cavity	80.00	%	Area of input contact point		**14.73**	sq.mm		
Length of input contact point	2.09	mm.	Maximum Hertzian Stress		**4.20**	Gpa	check in	check out
Number of rollers	3	rollers	Maximum stress on input point from Hertzian table		**41,425**	N	41,245	41,523
Number of cavities	2	cavity	Traction coefficient		**0.070**			
Length of output contact point	3.552	mm.	maximum reaction force on roller bearings		5,799	N		

R1	R2	R3	r1	r2	r3	Sin A	Cos A	Sin a	Cos a	Size factor
24.914374	26.52999104	23.2988	31.2549	33.2817	29.2282	0.6691306	0.7431448	0.4520767	0.8919791	0.96

X1	X2	X3	r1	x2	x3	Sin A	Cos A	Sin a	Cos a
66.741658	70.60621864	62.8771	31.2549	33.0647	29.4452	0.6691306	0.7431448	0.4520767	0.8919791

Total clamping force	124,275	N
stress under outerput contact	2.79	
Reaction force on thrust bearing	37454.6	N
Overall radius of transmission without actuator	101.20	
Overall length of transmission without hydraulics	227.9024	
Overall length between outside of discs	168.192	

					Speed dif	% loss
Speed at	R1	938,774	r1	938,774	0	
Speed at	R2	999,650	r2	999,649	1	0.00%
Speed at	R3	877,897	r3	877,898	-1	0.00%
Speed at	X1	938,774	r1	938,774	0	
Speed at	X2	993,132	x2	993,131	0	0.00%
Speed at	X3	884,416	x3	884,416	0	0.00%

Total spin loss: **4.93%**

MAXIMUM INPUT TORQUE: **433.47** Nm

ESTIMATED MASS: **92.54** kg

ESTIMATED VOLUME: **18.89** litres

spin in this position: 0.00%

5.41 Discs and rollers only litres torque den: 80.14951

KW: **272.34**

CALCULATION OF APPARENT RADIUS OF CONTACTING POINTS		
Angle of inclination of circle of inner contact	21.12	deg
Angle of inclination of circle of outer contact	105.12	deg
Angle of inclination of roller	26.88	deg
Radius of roller circle of contact	35.04	mm.
Radius of inner circle of contact on toroidal disc	26.71	mm.
Radius of outer circle of contact on toroidal disc	-255.82	mm.

SPIN LOSSES		
Spin loss in "L" position	4.93%	ENERGY LOST TO SPIN
Spin loss in "LM" position	2.50%	5.20%
Spin loss in "C" position	0.00%	
Spin loss in "HM" position	3.33%	
Spin loss in "H" position	4.86%	
Average spin loss	**2.60%**	

Model of SHTV with input torque of 430Nm

In this case the predicted torque density is 94.25Nm, However the predicted energy losses are much higher.
Torque density, life, and efficiency are so interrelated that it is not possible to make conclusions about one of them without checking what is happening with the other two.

It is very clear that the DFTV will always be much smaller than either a SFTV or a SHTV, when similar efficiencies and life are being designed for.

Efficiency

A Double Roller Full Toroidal Variator (DFTV) is perhaps the most efficient form of CVT available.

Typically the overall efficiency advantage of a DFTV over a Torotrak SFTV design is of the order of 8% -10%.

The efficiency advantages are brought about by

1. the reduction in velocity differences at the roller and disc contacts
2. the elimination of roller thrust bearings
3. the reduction in size of the hydraulics required to control ratios.
4. The fact that in a DFTV the discs both rotate in the same direction

There is a tradeoff between efficiency and Power density. It is possible to improve the efficiency of either a DFTV or a SFTV by decreasing the ratio of the crown radius of the rollers to the toroidal cavity so that the contact patch becomes narrower. However if the same power or torque capabilities are to be maintained the components must be larger and the power density reduces.

In the case of a SHTV decreasing the crown radius does not have such a dramatic affect on the power density because most of the energy lost is associated with the thrust bearing, the efficiency of which is not affected by a change in crown radius of the roller.

NSK have published a great deal of information on this subject and the chart below is a good example.

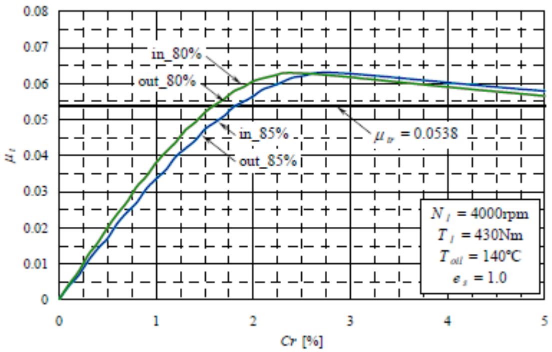


Fig.4 Comparison of power-roller curvature ratio of 80 and 85% on traction curves at $e_s = 1$ and $T_{oil} = 140\,°C$

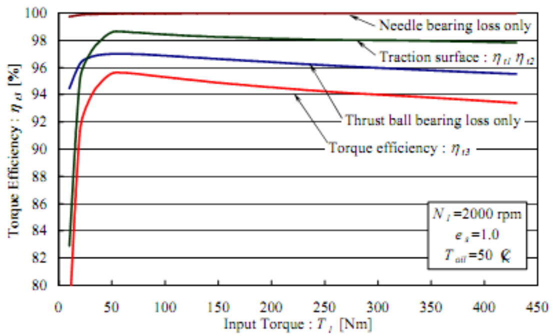


Fig.6 Calculated torque-transmission efficiency of the variator, and portion of torque losses of thrust ball-bearings, needle bearings and spin loss on the traction contact at $e_s = 1$, $N_i = 2000$rpm and $T_{oil} = 50\,°C$

Extract from Paper "Development of a 6 Power – Roller Half – Toroidal CVT – Mechanism and Efficiency – " Hirohisa Tanaka & Nozomi Yoyoda Yokohama National University & Hisashi Machida, & Takashi Imanishi NSK Ltd.

Efficiency

NSK have also studied the relative differences between a **SFTV** and a **SHTV** efficiencies. They have also studied varying the crown radius in a SFTV and its affect on efficiency and power throughput. The chart below is also a good example of this work

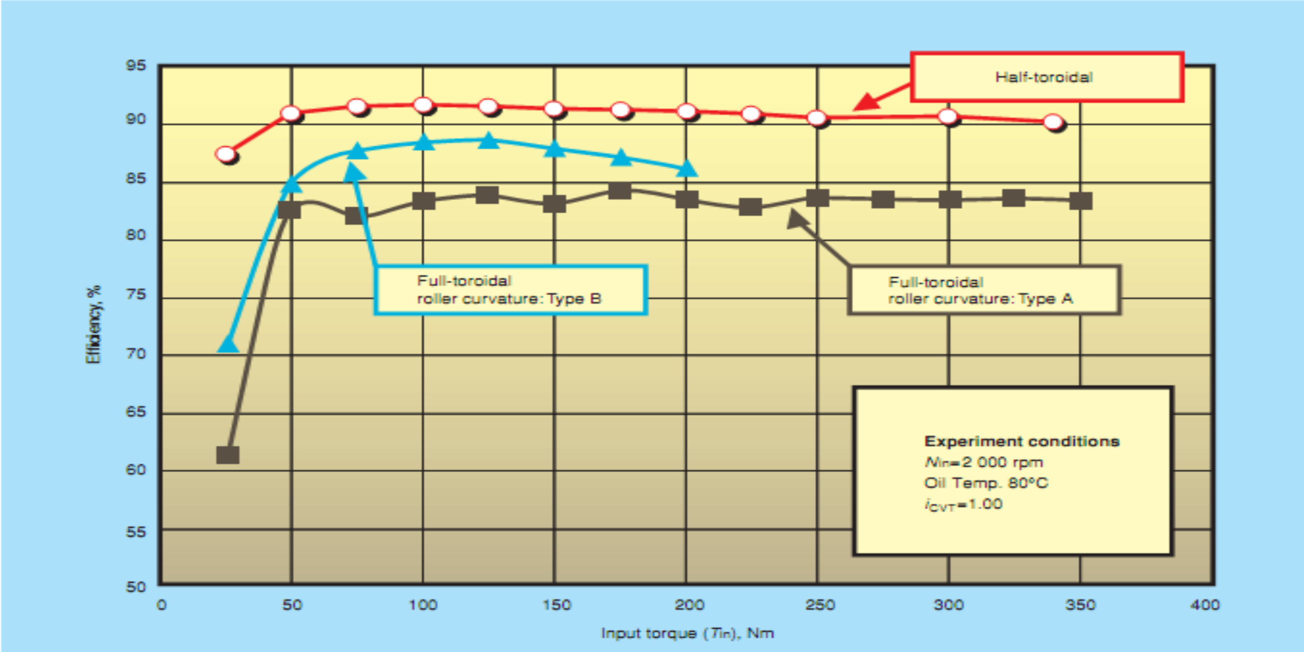


Fig. 9 Efficiency of half-toroidal and full-toroidal CVT

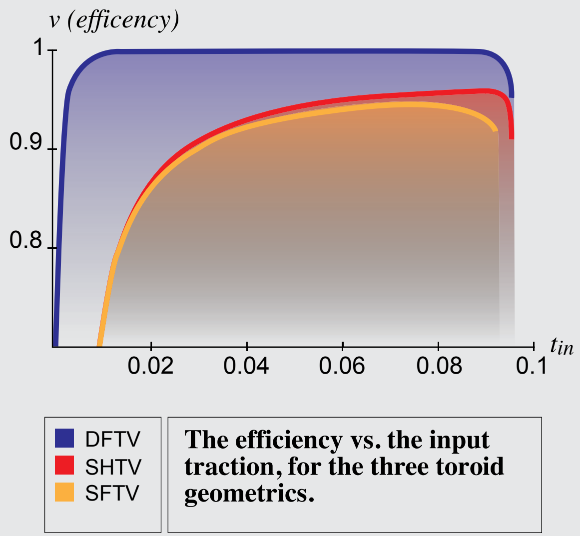
Extract from paper "Development of POWERTOROS Unit Half toroidal CVT – Comparison between Half-Toroidal and Full-Toroidal CVT's" Takashi Imanishi & Hishashi Machida Research and development Centre NSK.

Additional energy savings are associated with the fact that both the input and output discs rotate in the same direction for the **DFTV**.

With an input of 6,000RPM and a ratio spread of 4 the maximum differential rotational speeds of a DFTV are 6,000RPM in high gear with 3,000RPM in low , while that of a SFTV will be 18,000RPM in high gear and 9,000RPM in low.

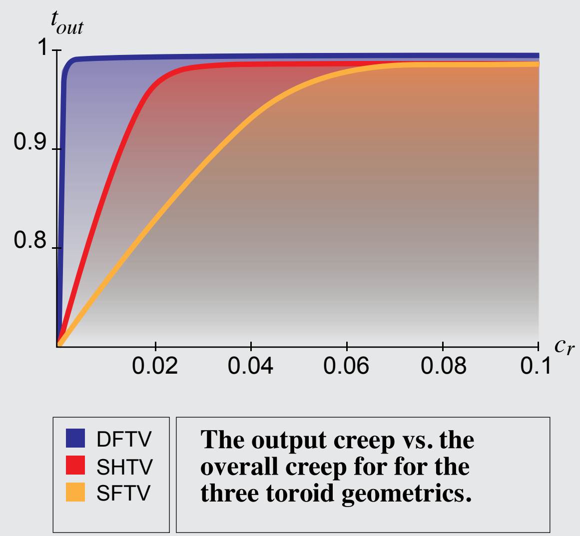
Efficiency

The DFTV exhibits a significant improvement in efficiency over the SFTV and SHTV and almost zero creep when in the 1:1 ratio position. This is despite the existance of three power contacts.



The efficiency vs. the input traction, for the three toroid geometrics.

Calculations are done at a ratio S_{rID}=1 and clamping force F_D = 22kN



The output creep vs. the overall creep for for the three toroid geometrics.

Calculations are done at a ratio S_{rID}=1 and clamping force F_D = 22kN

These graphs depict the DFTV in its most efficient state which occurs over almost 60% of its ratio range. The DFTV exhibits an efficiency improvement over the SFTV of between 6% and 4.5% at all times.

In a DFTV and a SFTV no thrust bearings are required to support the rollers. In a SHTV although the geometric arrangement reduces the differential velocities it necessitates the thrust bearings, and these consume a great deal of energy.

In a DFTV the geometry reduces the differential velocities but leaves the system balanced with no need for thrust bearings.

In a SFTV the hydraulic system must move the rollers against the Torque reaction Force (TRF) a reasonable distance during a ratio change. In order for the ratio change to be executed quickly the hydraulic system must be reasonably large.

Preliminary results of an indpendant academic study, being undertaken at a leading tribology research centre.

The details of this study can be made available by request.

Efficiency

In a SHTV the arrangement is the same but the rollers only move a very small distance although this movement must be extremely fast or a ratio overrun will occur.

Calculations carried out by Ultimate Transmissions indicate that the energy consumption of a Torotrak control system will be of the order of 5%. Not enough information is available to study the SHTV. Although the ratio changing itself occurs over only small time frames, the hydraulic system is operating all the time. The system must also be sized so as to function properly at low RPM which means that at high RPM it continues to consume comparatively large amounts of power.

The ratio control of a DFTV can be executed by a very small stepper motor requiring an energy input of less than 0.1% of the input power. Virtually no force is required to execute a ratio change.

In a DFTV the input and output discs rotate in the same direction. In a SFTV and a SHTV they rotate in opposite directions.

This means that a simpler variator architecture can be adopted and far less parasitic losses associated with windage and oil churning are experienced.

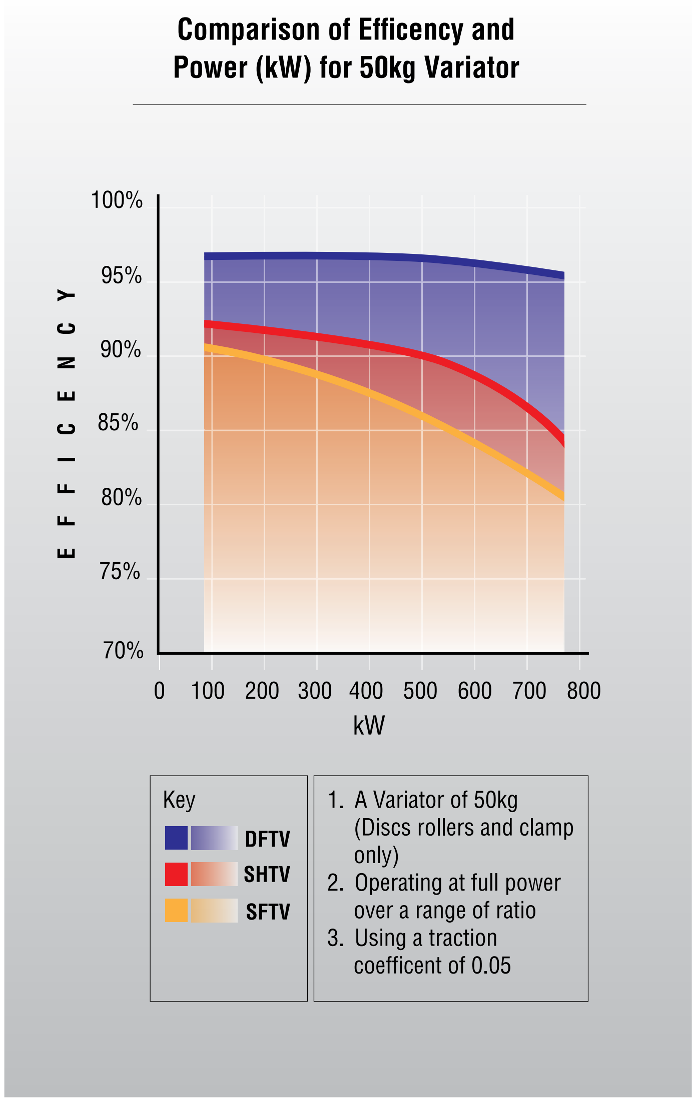
The diagram right, maps the efficiency and power capabilities for the three mechanisms.

The typical power range for a SFTV or SHTV of this size is 300kW to 400kW or 400Nm at 6000RPM with a vehicle design life of 400,000kms.

This capability can be expanded beyond this by increasing the conformity of the roller to the toroidal discs.

The design range for a DFTV of 50 kg. is 800 kW. It cannot be expanded beyond this because the support bearings for the rollers become over stressed for this design life.

All variators can be increased in power by increasing the traction coefficient. The 0.05 used here is considered the most likely upper limit in real CVT's.



In order to design transmissions using the DFTV the operating parameters must first be established.

A relatively simple software has been produced by Ultimate Transmissions that enables these parameters to be established for all three toroidal variator types. By comparing the output parameters of all three some simple reverse engineering can be undertaken using working examples of the SFTV and the SHTV to fine tune or confirm the parameters predicted for the DFTV.

The software is based on this description and can be made available to any parties interested in more detailed evaluation.

Designing Toroidal Variators using a Comparative Approach

Single Roller Full Toroidal Variator (SFTV) Single Roller Half Toroidal Variator (SHTV) and Double Roller Full Toroidal Variator (DFTV)

PREPARED BY MICHAEL DURACK FOR ULTIMATE TRANSMISSIONS

Overview

Traction drives are one of the oldest forms of transmissions. They preceded gears as they were much simpler to make.

The first forms of traction drives used “dry” friction to transfer force from what was typically a roller on a disc that could change its position on the disc and so change ratio. Unfortunately dry friction is very prone to wear and these early drives required high levels of maintenance. Many smaller machines continue to use friction drives (as they were called) when gears, present particular mechanical problems.

cont >

Design Software

Modern high power traction drives use a fluid called a “Traction Fluid” to separate the rolling components so that no metal to metal contact occurs and wear is more or less eliminated. The traction fluid increases in viscosity by billions of times when subject to very high pressures and behaves like a glassy solid as the rollers roll over the disc.

The most successful of these are called Toroidal Variators. There are the three types - SFTV, SHTV, and DFTV.

The SFTV is promoted by companies like Torotrak (www.torotrak.com) CVTcorp (www.cvtcorp.com) while the SHTV by companies like Nissan and NSK (www.nsk.com under the brand name Powertoros).

The DFTV has been recently developed by Ultimate Transmissions and exhibits superior performance and greater power density.

Excel spreadsheets have been developed by Michael Durack for Ultimate Transmissions that can be used to calculate the key performance characteristics of Double Roller Full Toroidal Variators (DFTV) Single Roller Full Toroidal Variators (SFTV) and Single Roller Half Toroidal Variators (SHTV).

The key factors that control the ability of a Toroidal based CVT to transmit torque are

- The maximum stress that the contact points on the rollers and discs can withstand.
- The Traction coefficient of the Traction Fluid
- The degree of differential velocities present at the contact points.

These excel spread sheets calculate with a reasonable level of certainty the relationship of these three variables within the three different mechanisms, and allow designs to be produced rapidly.

Basic Inputs

In all three cases these spreadsheets rely on the following basic inputs. These inputs are shown as yellow in the spread sheets. They are inputted into the sheet referred to as DFTV L, SFTV L and SHTV L. It should be noted that the DFTV and SFTV models are in the same work book.

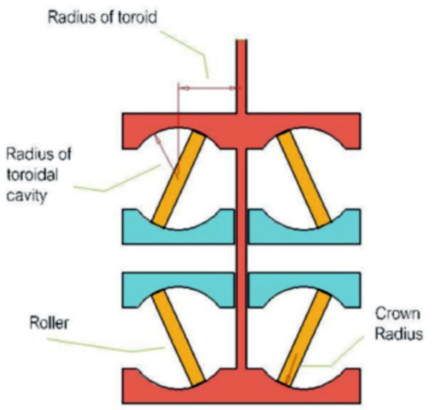
cont >

Design Software

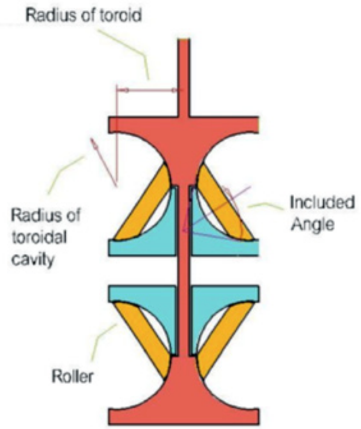
DESIGNING TOROIDAL VARIATORS USING A COMPARATIVE APPROACH

1. The Radius of the Toroidal cavity (R)
2. The Radius of the Toroid (RR)
3. The Radius of the roller (r1). This number applies only to the DFTV and the SFTV. The roller radius for the SHTV is derived from R and Cos of the “included angle” a.
4. The relationship of the crown radius of the roller to the toroidal cavity radius as a percentage. This relationship is often referred to as the crown radius conformity. (c)
5. The Number of cavities. (NC)
6. The number of rollers (or roller pairs DFTV) in each cavity. (NR)
7. The (Design) Traction coefficient. (TC)
8. The maximum contact pressure allowed at the contact of the roller with the discs. (MH)
9. The maximum input RPM. (IRPM)
10. The Primary Rotational Angle or the angle of rotation that determines the gear ratio operating inside the Variator. (A). This angle is zero in the centre and either positive or negative on either side of this.

    In the case of the DFTV and the SHTV there are two secondary geometric inputs that are designed to generate zero (or close to zero) spin, or zero differential velocities at the roller contacts for some positions. It is not possible with the SFTV to arrange this in any position.



TYPICAL SFTV

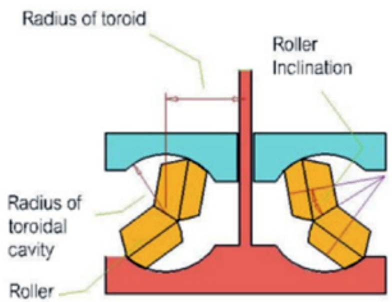


TYPICAL SFTV

cont >

11. The Roller Inclination (a). For DFTV only. This represents the angle formed between the roller axis and the centre line or conical contact line, where the two cones contact each other. At a certain angle it is possible to ensure no spin or differential velocities occur at the contact patch in one position . This position is usually arranged to happen in the 1:1 ratio or centre position.

12. The Included angle (a). For SHTV only. This represents the angle formed by the tangent line at the contact point and the centre of rotation of the roller. At a certain angle it is possible to ensure no spin or differential velocities occur at the contact patches in at least one position . This position is usually arranged at the extreme speedup or slow down positions.



TYPICAL SFTV

All of the lengths inputted in this first step can be scaled up using a “Size factor”. This size factor can be entered manually to automatically scale up all lengths areas and performance outputs. This is shown in blue in the spread sheet.

Secondary Geometric Outputs

A number of geometric outputs are calculated (using the Basic Inputs) that determine the contact radii of the points of contact. It is necessary to calculate these radii accurately so that the properties of the elliptical Hertzian contact points can be established accurately.

These are shown as an olive colour in the spread sheets.

Firstly the contact radius of the roller is calculated (in the section CALCULATION OF APPARENT RADIUS OF CONTACTING POINTS) in the spread sheet. Using the following formulae.

cont >

rr = r1/COS aa where

- rr is the Actual Radius of contact of (toroidal) roller surface
- r1 is the radius of the roller where it contacts the disc
- aa (in the case of the SFTV) is the Angle of inclination of the roller plane of rotation to the line connecting the contact point to thecentre of the toroidal cavity in the case of the SHTV it is the angle (also aa) between the plane of rotation of the roller and the plane containing the tangents of the contact patch

In the case of the SFTV the angle aa is zero so the contact radius is the same as the actual radius, whereas for the other two the contact radius is larger than the actual radius. rr is the same for either the input and output contact and remains the same for any ratio.

Secondly the radius of the input and output contacts on the discs are calculated (again located in the section CALCULATION OF APPARENT RADIUS OF CONTACTING POINTS) in the spread sheet, using the following formulae. The input and output contacts are different except for the centre position where they are both the same.

RI = R1/SIN A where

- RI is the Actual Radius of the contact circle of the disc to the roller at the input contact point.
- R1 is the distance of the input contact patch to the centre of rotation of the discs.
- A is the angle of rotation (ratio) of the roller or rollers.

RO = X1/SIN A where

- RO is the Actual Radius of the contact circle of the disc to the roller at the output contact point.
- X1 is the distance of the ioutput contact patch to the centre of rotation of the discs.
- A is the angle of rotation (ratio) of the roller or rollers.

For both the SFTV and the DFTV at the centre position (1:1) the radius is infinite. Generally the outside contact point in both the SFTV and DFTV is a negative number. In the case of the SHTV only the extreme outer point may be negative.

For the SHTV the following formulae are used. This requires two steps first to calculate the angles of inclination of the circle of contact for the input and output and to then apply it to the contact radii on the discs.

cont >

DESIGNING TOROIDAL VARIATORS USING A COMPARATIVE APPROACH

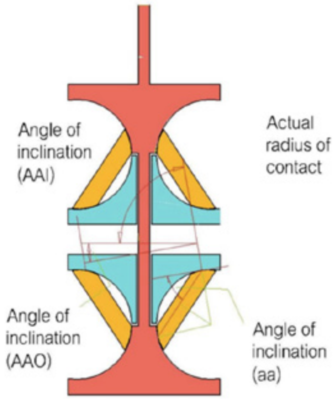
Step 1

AAI = 90 – aa – A where

- AAI is the angle of inclination of the input contact circle to the direction of the roller reaction
- aa is the included angle
- A is the primary rotational angle

AAO = 90 – aa + A where

- AAO is the angle of inclination of the output contact circle to the direction of the roller reaction
- aa is the included angle
- A is the primary rotational angle



TYPICAL SFTV

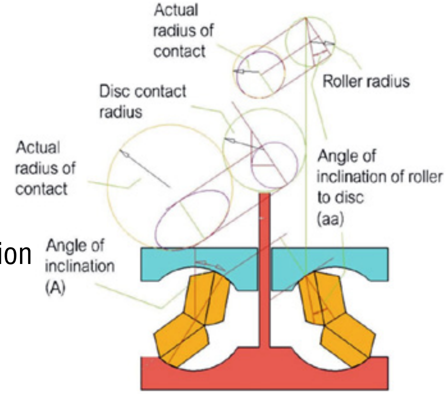
Step 2

RI = R1/COS AAI where

- RI is the radius of the contact circle of the disc to the roller at the input contact point.
- R1 is the distance of the input contact patch to the centre of rotation of the discs.
- AAI is the angle of inclination of the input contact circle to the direction of the roller reaction

RO = X1/COS AAO where

- RI is the radius of the contact circle of the disc to the roller at the input contact point.
- R1 is the distance of the input contact patch to the centre of rotation of the discs.
- AAO is the angle of inclination of the output contact circle to the direction of the roller reaction



TYPICAL SFTV

cont >

The actual crown radius is calculated using this formula
cr = R x c where

- cr is the actual crown radius of the roller
- R is the radius of the toroidal cavity
- c is the crown radius conformity

Secondary Inputs

Once all of the contact radii have been calculated it is possible to calculate the maximum force that the rollers can exert on the discs. This is done using a Hertzian contact stress calculator ensuring that the maximum Hertzian stress (inputted earlier) is not exceeded.

Using this calculator the following inputs are derived. These are then manually entered into the appropriate field shown brown in the spread sheet.

- Maximum force for the input point (MF)
- Width of the contact point at the input point (W1)
- Length of the contact point at the input point (L1)

The calculation is repeated for the outer contact using the same maximum force but the new contact radii appropriate for that point, and the following inputs are derived

- Maximum stress at the output point (MHS)
- Width of contact point at the output point (W2)
- Length of contact point at the output point (L2)

Generally the stress is lower at the output point and the contact patch longer and narrower because of the different contact radii.

It is important when calculating the patch width and length of the SFTV that the correct number is applied to the contact patch. Generally the SFTV contact patch is longer than it is wide, whereas the contact patch for the other two is almost always wider than it is long. This is simply the result of using closer conformity rollers in the SHTV and the DFTV than in the SFTV.

cont >

Calculated Torque Rpm & Spin (Low Gear)

The model now calculates the maximum input torque. It is always assumed that this occurs at extreme low gear and will be the ultimate limitation placed on the CVT. The maximum input torque is calculated using this formula

Maximum input torque = NC x NR x MF x TC x R1 where

- NC is the number of cavities
- NR is the number of rollers or roller pairs in the CVT
- MF is the maximum reaction force on the roller disc contact
- TC is the traction coefficient
- R1 is the radius of the input contact to the centre of rotation of the discs.

R1 is calculated by using this formula

R1 = RR - R x SIN A where

- R1 is the input radii
- RR is the radius of the toroid
- R is the radius of the toroidal cavity
- A is the angle of rotation of the roller

The output RPM is calculated using this formula

ORPM = IRPM x R1/X1 where

- IRPM is the input RPM
- ORPM is the output RPM
- R1 is the input radii
- X1 is the output radii

cont >

DESIGNING TOROIDAL VARIATORS USING A COMPARATIVE APPROACH

The model now calculates the surface velocities of the extreme outside edges of the contact patch.

It firstly calculates the rotational radii on the disc at the contact patch and the roller at the contact patch using the basic formulae for the disc and roller

R2 = R1 + W1/2 x COS A where

- R2 is the outside edge (or faster edge) of the patch on the disc
- W1 is the width of the contact patch
- A is the angle of rotation

r2 = r1 + W1/2 x SIN a where

- r2 is the matching edge to R2
- W1 is the width of the contact
- a is the angle of inclination of the rollers

It now calculates the speed of the surfaces at the edges and centre of both the input and output the contact patches. Firstly it calculates the speed of the centre of all of the patches using this formula.

Speed at R1 = R1 x 2 x   x IRPM where

- R1 is the input contact radius
- IRPM is the Input RPM

The Roller RPM (RRPM)can now be calculated using this formula

RRPM = Speed at R1/r1/ /2 where

- r1 is the radius of the roller

It now uses the other contact radii to calculate the speeds at the outer most points of the contacts.

It calculates the difference in speed for the outer edges of the input and output points and expresses it as an (absolute) percentage of the average speed. The four percentages are summed and divided by two. The division by two is simply because this method causes four numbers to be summed when in reality there are only two.

cont >

Calculation of Actual Energy Loss Associated with Calculated Spin Percentage for Low Gear

A study of energy losses disclosed by NSK in published data of the SHTV and this model indicates that the percentage should be factored by 0.675 to have it align with actual efficiency losses quoted by NSK for the "spin" only.

This factor is approximately 1/2 which would represent the relationship of the average pressures under the patch to the maximum pressure.

This constant also relates closely to the efficiency losses calculated by Professor Carbone for the DFTV and the SFTV.

Calculation of Output Data for Other Gear Ratios

The models also create the performance characteristics for other gear ratios.

For the DFTV and SHTV these ratios are

- High gear which is set as the mirror of Low gear with A simply the negative value of the value of A for low gear. Referred to as DFTV H, SFTV H, and SHTV H
- The 1:1 ratio or central position where A equals zero. Referred to as DFTV C, SFTV C, SHTV C.
- A ratio called DFTV LM and SHTV LM – meaning Medium Low. There is no ratio for the SFTV. This ratio is such that the output speed is approximately half way between the Low Gear position and the central position.
- A ratio called DFTV HM and SHTV HM. There is no ratio for the SFTV. This ratio is such that the output speed is approximately half way between the High Gear position and the central position. It is the mirror of the Low Medium gears.

The Alternative ratios first calculate the amount of force that the roller must be subjected to in order to create the same level of input torque but inputted to the new radius for that ratio. The following formula is used to calculate this force.

cont >

MF = Maximum Input torque / NR / NC / TC / R1 where

- MF is the force required for that position
- NR is the number of rollers
- NC is the number of cavities
- TC is the traction coefficient used for low gear.
- R1 is the input radius for that gear ratio

This force is then used to calculate the maximum Hertzian stress and size of the input and output contact patches.

The spin losses can then be calculated as for Low Gear.

The process is repeated for the other gear ratios and a complete picture of stresses, energy losses, and output RPM's is created.

The Total Ratio Change (or spread) TRC is calculated simply by dividing the output RPM at High Gear by the output RPM at low gear.

Calculation of Overall Energy Losses Due to Spin

With the spin loss for each ratio position calculated it is possible to create an average loss assuming that the CVT duty cycle uses each ratio an equal amount of time.

The SFTV has only three ratios calculated because the energy losses are similar for each ratio so the formula is simply:

Average spin loss = (Spin loss in L position + (Spin loss in C position x 2) + Spin loss in H position)/4

For the other the formula is as follows

Average spin loss = (Spin loss in L position + loss in LM position + (loss in C position x 2) + loss in HM position + loss in H position)/6

From this the actual energy loss due to spin can be calculated using the factor described earlier.

cont >

DESIGNING TOROIDAL VARIATORS USING A COMPARATIVE APPROACH

Many other factors affect the final physical size of a working CVT and this figure can only be used as a guide to determine likely outcomes as a theoretical design is brought up to the status of a working mechanism.

Other Volumetric & Weight Outputs of the Models.

Conceptual autocad drawings of working SFTVs SHTVs and DFTVs have been prepared. These have produced fractional relationships to the fundamental geometries of the three mechanisms. These are then used to create real volumetric and weight comparison, for mechanisms with the same maximum torque and similar efficiencies.

- Overall radius of the transmission without actuator
- Overall length of the transmission without hydraulics
- Overall length between the outside of the discs
- Estimated volume of the operational transmission
- Estimated weight of the transmission (a standard density of 4.9 is used to convert volume to density in all cases.)
- The fractional weight of the DFTV when compared to the other two

<u>Generally the DFTV remains around 30% of the weight and volume of the other two, while also remaining a few percentage points more efficient.</u>

Other Operational Outputs of the Models.

The models also produce other operational numbers that can be used to assist with design such as –

- The torque reaction force on the rollers that must be supported by the roller bearing system.
- The maximum roller RPM
- The Maximum stress cycles on the rollers per minute
- The maximum differential speeds over the contact points.
- The area over which the heat generated by spin is distributed over the rollers.

cont >

- The area over which the heat generated by spin is distributed over the discs
- The worst case energy loss at the contact patch
- The time spent by a point on the contact patch while being heated (assumes maximum torque input in low gear position)

Generally the SHTV contact patches experience around 25%- 50% of the heating intensity as that experienced by the SFTV. The area over which the heat is generated is larger and the spin intensity less than half. The DFTV is somewhere in the middle with greater area on the rollers and less time spent by the patch in the heating zone, but with more than twice the spin.

A separate calculation is carried out to determine the maximum Hertzian stress on the conical contact patches for the DFTV only.

A separate Sheet is included in the SHTV work book that calculates some of the operational characteristics of the thrust bearing. Although the detailed geometry of this thrust bearing is not known the calculations can be used to develop an understanding of the correlation of the likely energy loss on these bearings and the efficiency losses reported by NSK and Carbone.

Calculation Of Actual Traction Coefficient (For DFTV Only)

A separate set of calculations is added to the DFTV calculations that take account of the ability of the DFTV to be designed using a direct relationship between the Torque reaction experienced by the rollers and the clamping force.

It is intended that the design of the DFTV includes a clamping system that is generated as a direct multiple of the torque reaction on the rollers.

The SFTV and the SHTV use other forms of clamping control that attempt to create a situation where “over clamping” is avoided.

The DFTV is unusual in that the areas of greatest spin occur at the high and low ends of the gear ratio range. The SHTV is the opposite and the SFTV remains with levels of spin that are similar over the whole range of ratios.

A simple clamping system that simply multiplies the roller torque reaction by the inverse of the traction coefficient can be used by all three mechanisms. In the centre position this relationship holds true, however as the rollers move away from the centre position the “wedging” action of the toroidal cavity will cause the actual roller reactions on the disc to increase by the inverse of the COS of the rotated angle.

cont >

This increase in clamping force along with the reduction in Traction Coefficient suits the DFTV but not the other two. As the rollers move away from the centre (zero spin) position while the clamping force remains a multiple of the torque reaction on the rollers the actual force of the roller onto the disc increases by a factor equal to the inverse of the COS of how much the rollers are rotated.

Conversely the designer can select a traction coefficient that is selected as the highest possible (less a safety factor) for a contact patch experiencing the worst case spin, and using a particular traction fluid and operating at a predicted temperature. The model will then check how much the traction coefficient will need to increase as the rollers rotate torwards the centre position.

A great deal of information is readily available on the “safe” traction coefficients for different levels of spin and temperature. The “unclamping” traction coefficient is thus reported as the “Actual clamping coefficient” In a typical DFTV this may move from 0.055 in the worst spin condition (>6%) to 0.065 when in the centre position with zero spin.

Although this relationship of spin and 1/COSA may not give a perfect platform for minimizing over-clamping it is simple and robust.

Selection Of Key Operational Limits When Designing A Dftv

The two key factors in setting the operational limits when designing a DFTV are

1. **Maximum Hertzian Stress**
2. **Traction Coefficient**

The most successful toroidal variator is undoubtedly the NSK SHTV. These transmissions have generally given reliable service in real vehicles of over 400,000 kms. It is believed that the most common failures were associated with the thrust bearings not the rollers and discs.

Without all of the design details it is not possible to understand exactly how this was done. However a detailed study of a (real) NSK SFTV transmission would allow reverse engineering that would uncover more accurately what limits of these two factors were used in the design.

A study of published NSK academic papers lead to the conclusion that they were as follows

1. 3.95GPa maximum stress or 2.63 GPa average
2. 0.055 in the centre position (4% spin) with over-clamping occurring in most other positions.

cont >

Design Software

DESIGNING TOROIDAL VARIATORS USING A COMPARATIVE APPROACH

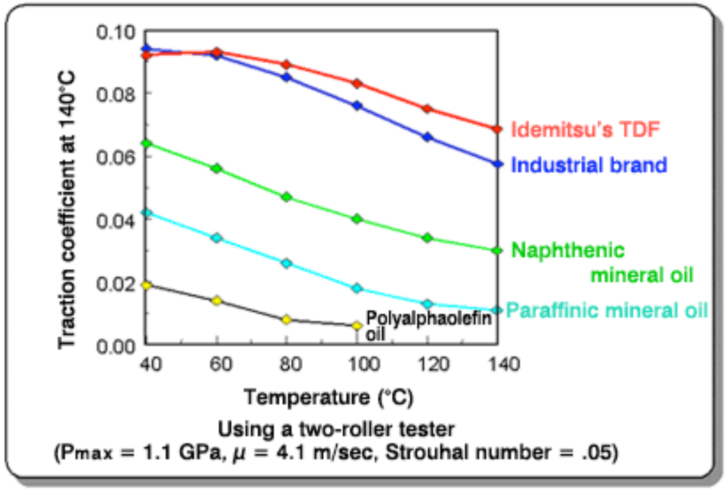
A study of Torotrak information is very confusing but it seems more like

1. 4.2 GPa or 2.8 average
2. 0.07 in the centre position (>6% spin) with all other positions less than 0.07.

It is highly likely that the 0.07 coefficient quoted by Torotrak is unrealistically high in order to reduce the apparent size and weight of their transmissions.

This is particularly true when you consider that the contact patch in a Torotrak Variator may be 20 -30 degrees hotter than the patch in a NSK variator.

Traction Coefficient Comparison



NSK use an Idemitsu Traction fluid that states its traction coefficients according to the above chart.

Torotrak use Santotrak fluids that are nowhere as well defined as Idemitsu when it comes to temperature affects.

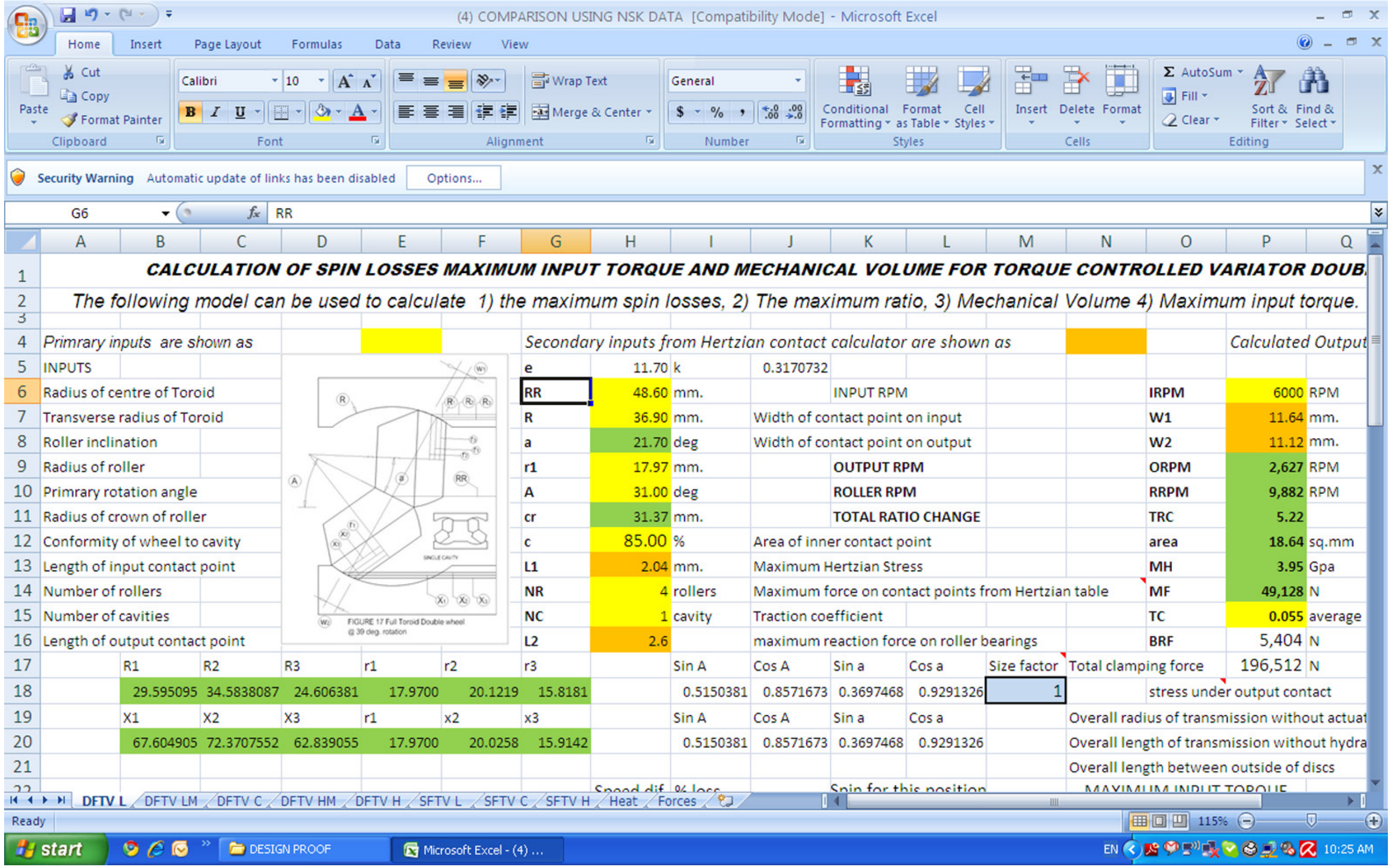
If the traction fluid temperature is running at 160 degrees then using 0.07 would be very unsafe, particularly with a fluid with less quality than the Idemitsu fluid.

Any gross slip that might occur in Low gear will immediately damage the roller surface and the disc in the region of low gear.

Torotrak's figure (differential velocity) in the worst case at 6,000RPM is 2.47 m/sec while NSK is 0.52 m/sec and the DFTV is 1.79 m/sec.

Design Software

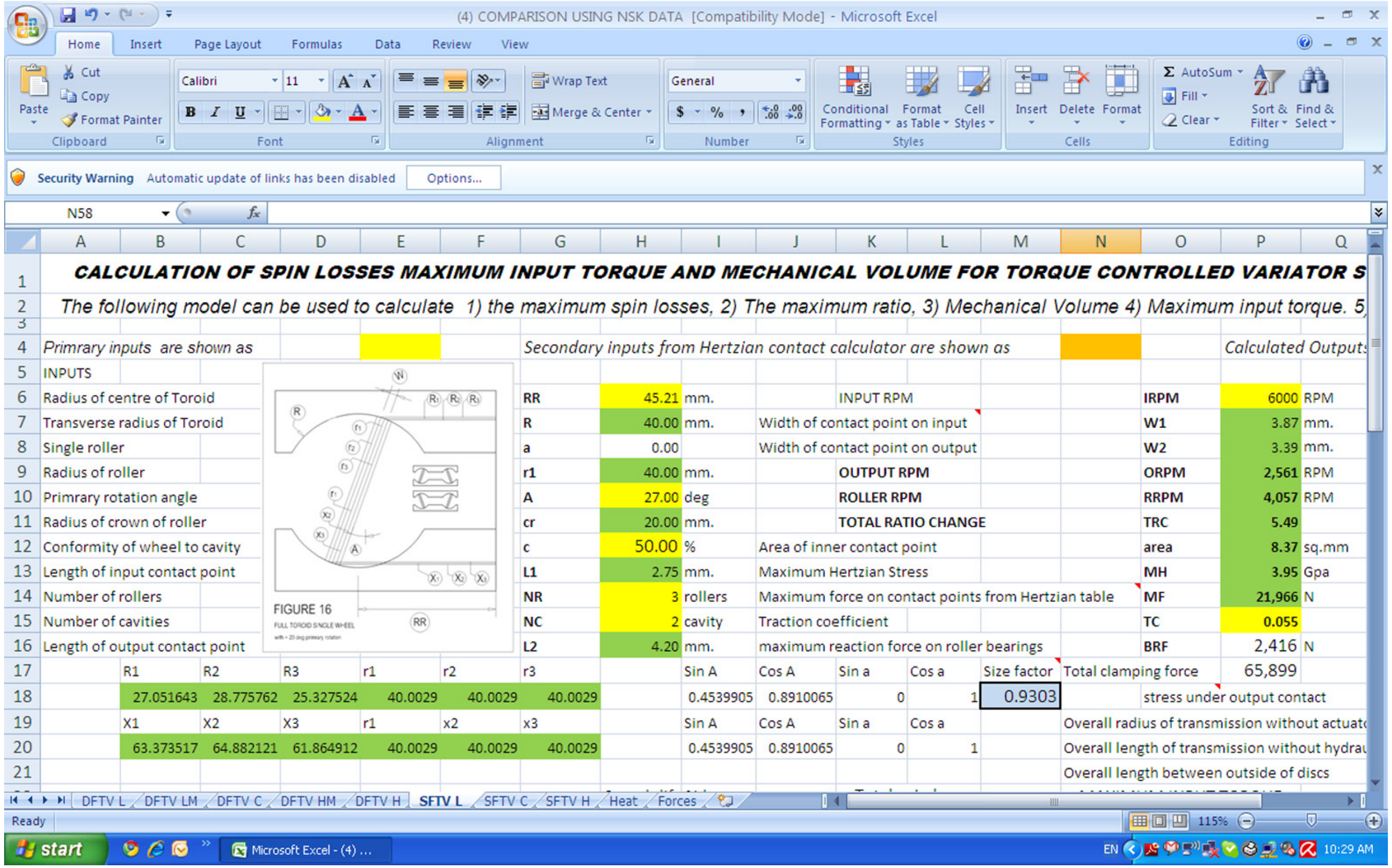
Below is a typical output pages derived from this software for the three toroidal types.



	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q
1	CALCULATION OF SPIN LOSSES MAXIMUM INPUT TORQUE AND MECHANICAL VOLUME FOR TORQUE CONTROLLED VARIATOR DOUB																
2	The following model can be used to calculate 1) the maximum spin losses, 2) The maximum ratio, 3) Mechanical Volume 4) Maximum input torque.																
3																	
4	Primrary inputs are shown as						Secondary inputs from Hertzian contact calculator are shown as									Calculated Output	
5	INPUTS						e	11.70	k	0.3170732							
6	Radius of centre of Toroid						RR	48.60	mm.		INPUT RPM				IRPM	6000	RPM
7	Transverse radius of Toroid						R	36.90	mm.	Width of contact point on input					W1	11.64	mm.
8	Roller inclination						a	21.70	deg	Width of contact point on output					W2	11.12	mm.
9	Radius of roller						r1	17.97	mm.		OUTPUT RPM				ORPM	2,627	RPM
10	Primrary rotation angle						A	31.00	deg		ROLLER RPM				RRPM	9,882	RPM
11	Radius of crown of roller						cr	31.37	mm.		TOTAL RATIO CHANGE				TRC	5.22	
12	Conformity of wheel to cavity						c	85.00	%	Area of inner contact point					area	18.64	sq.mm
13	Length of input contact point						L1	2.04	mm.	Maximum Hertzian Stress					MH	3.95	Gpa
14	Number of rollers						NR	4	rollers	Maximum force on contact points from Hertzian table					MF	49,128	N
15	Number of cavities						NC	1	cavity	Traction coefficient					TC	0.055	average
16	Length of output contact point						L2	2.6		maximum reaction force on roller bearings					BRF	5,404	N
17		R1	R2	R3	r1	r2	r3		Sin A	Cos A	Sin a	Cos a	Size factor	Total clamping force		196,512	N
18		29.595095	34.5838087	24.606381	17.9700	20.1219	15.8181		0.5150381	0.8571673	0.3697468	0.9291326	1		stress under output contact		
19		X1	X2	X3	r1	x2	x3		Sin A	Cos A	Sin a	Cos a		Overall radius of transmission without actuat			
20		67.604905	72.3707552	62.839055	17.9700	20.0258	15.9142		0.5150381	0.8571673	0.3697468	0.9291326		Overall length of transmission without hydra			
21														Overall length between outside of discs			
22								Speed dif	% loss		Spin for this position			MAXIMUM INPUT TORQUE			

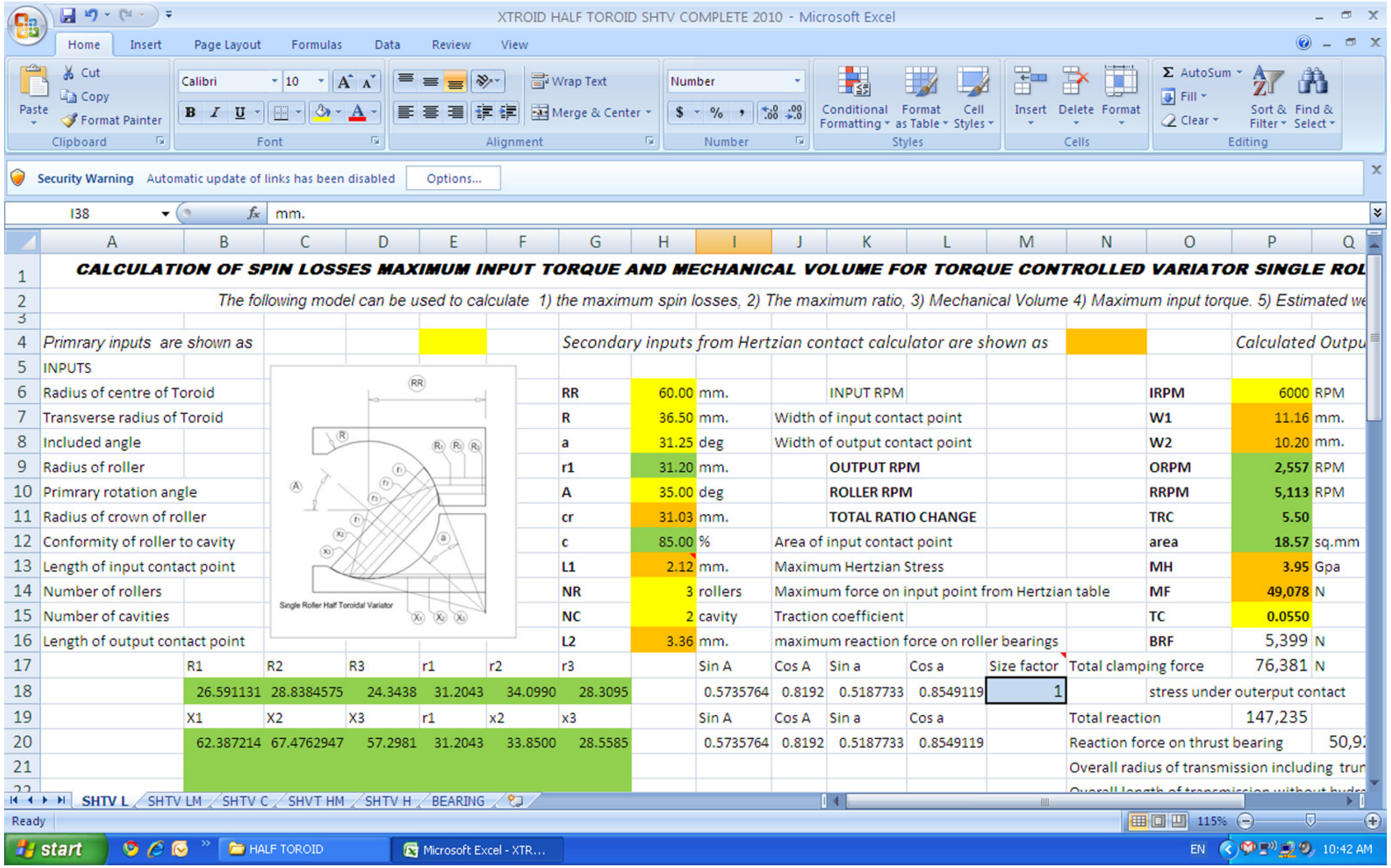
DESIGN PARAMETERS FOR DFTV WITH MAXIMUM POWER THROUGHPUT OF 270Kw

Design Software



DESIGN PARAMETERS FOR SFTV WITH MAXIMUM POWER THROUGHPUT OF 270Kw

Design Software



DESIGN PARAMETERS FOR SHTV WITH MAXIMUM POWER THROUGHPUT OF 270Kw