STUDY OF INFLUENTIAL PARAMETERS ON LIFETIME OF A TRACKBALL CVT WHICH IS UNDER SLIP-SPIN

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Abstract. *The purpose of this work is to introduce some concepts about parameters that influences the behaviour of traction CVTs under contact load. It studies the effects of parameters variation and consequences in lifetime of a selected CVT. Presentation of a trackball traction CVT is done as well as its geometry and way of operation. Posteriorly this part, the authors present a model to calculate lifetime based in four step: the rolling contact based in Hertz theory; Lubrication based in Reynolds equations; Mapping the spin and slip areas by means of FAST algorithm by Kalker; Calculating and displaying the stress distribution on the contact. Finally it is possible to calculate the lifetime of that CVT means of LundBerg Palmgreen for any condition (not necessarily this way). Lifetime and contact efficience change with the parameters variation (transmission ratio, torque, lubricant, contact geometry, slip and spin conditions) and it could be demonstrated that the best transmission ratio (for this type of CVT) is 1:1 to increase the system lifetime. On the other side, it does not provide the best contact efficiency.*

Keywords: *CVT, Transmission, Contact Theory, Lifetime*

1. INTRODUCTION

History have shown along years that relationship between human and machines have become more and more intensive. For example, the need of finding ways of transmit forces and movements demanded such study, creativity, strength and many attempts. In the beginning, the man discovered the handspike, studied effects of forces in free bodies and how transmit them in intention to generate movement, followed later by the development of engines, clucks and transmissions.

Transmissions are able to transmit mechanical energy and, by mean of its configuration, balance how that energy is transmitted. It can be as a high torque and reduced speed or as a low torque and high speed. Its main function is to allow engines, turbines or compressors to remain in its maximum torque or power state during a time. In other word, it can be explained there are a operation state which engines are better adapted to the load conditions.

Currently, there are many types and commercialized models in the world, mainly in developed countries, where the economy and researches are more stimulated. They are also used in military applications as long as it is known they are very compatible and powerful in this case (for power and speed).

Transmissions can be classified in 4 main groups : Convention Transmissions, Epicyclical Transmissions or Planetary, Continuously Variable Transmissions and Hybrid Transmissions.

Each one has its main properties but, in this paper, only the Track-Ball Continuously Variable Transmission will be discussed.

2. REVISION ABOUT CVT

Continuously Variable Transmissions had been used in the automotive industry principle, discarded after and now are coming back and occupying a small slice (but increasing) of the market.

CVT is a traction drive transmission characterized by the use of high friction materials on the contact spots, generally special steel alloys, and the operation in a viscous environment with no longer slips (so, it has good efficiency). The basic principle is the same of any CVT model, the changes on the contact spots geometry.

Among many advantages of using CVT in relation of other types, the efficiency with few powerless in low and mediums torques must be the most notable and required. Other qualities like a smoothness of the transmission ratio shifting, a easier control system and a more economic fuel consumption contribute too. Efficiency depends on the CVT model, its parameters, used alloys, lubrication and load conditions.

In spite of the great variety of available models, it is not so accepted by the third world companies, in not conventional applications and there are currently a diffidence felling about CVT capacities under its performance and efficiency. There seems be, although, a conflict: A great and fantastic variety of CVT models available to a restricted and fearful market. However, telling the truth, there are few companies which produces general traction drives transmissions whose lifetime characteristics and capacities are well defined.

3. GEOMETRIC CHARACTERISTICS OF A TRACK-BALL CVT

Traction CVTs are evidenced basically for its contact rolling elements' shape and size. The analyze of a Track-Ball TD-CVT was chosen because of its geometry form to be relatively simple. In addiction, it is a model whose data are easily available in other bibliographic sources.

The Figure 1 shows three possible configurations the CVT could be, only differentiate by the leaning of the sphere rotation axis over one side and, consequently, by the effective radius variation on the contact spot. It results on a changing of transmission ratio materials on the contact spots, generally special steel alloys, and the operation in a viscous environment with no longer slips (so, it has good efficiency). The basic principle is the same of any CVT model, the changes on the contact spots geometry.

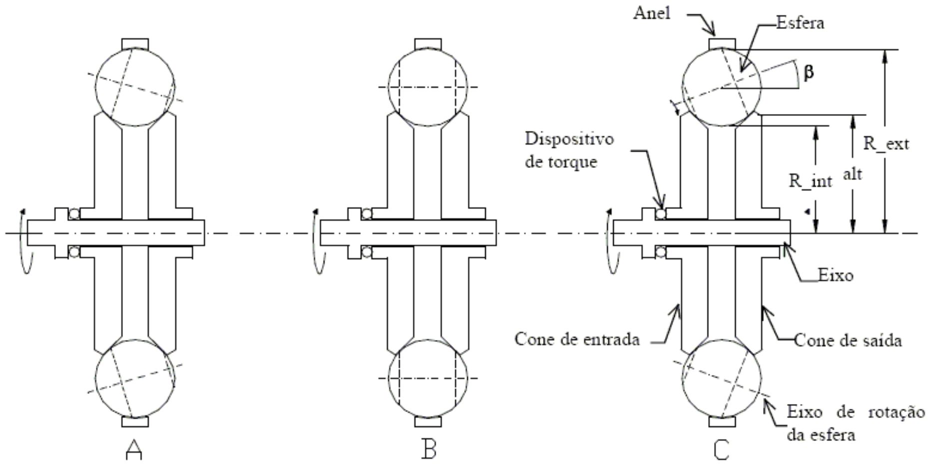


Figure 1 - CVT presenting (A) amplification (B) ratio 1:1 (C) reduction.

The figure below represent, from the left to right, (A) as a movement amplification. In other words, the outer cone rotation is higher than the inner one, once the ball effective radius on the inner cone contact spot is higher than the outer one. On the contrary, the configuration (C) shows a opposite case, known as a reduction ratio. Exactly in the centre of the figure, there is the (B) condition that represent a transmission ratio 1:1.

The transmission ratios is a relation between the outer effective radius and the inner effective radius, however, both are function of the CVT geometry. The next figure shows a ball of that transmission, being C and D spot contacts between ball and the conic rollers (the outer and the inner ones) respectively.

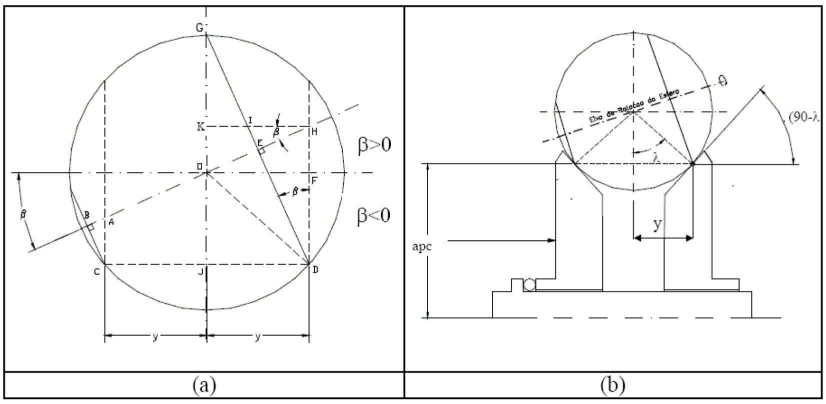


Figure 2 – Configuration between the spot contacts

The transmission ratio is given by:

$$rt = \frac{\overline{BC}}{\overline{DE}}\text{, being } \begin{cases} \overline{BC} = \left[\overline{FD} - y\tan(\beta)\right]\cos(\beta) \\ \overline{DE} = \left[\overline{FD} + y\tan(\beta)\right]\cos(\beta) \end{cases} \text{ e } \overline{FD} = \sqrt{(\overline{OD})^2 - y^2} = cte = c1 \qquad \text{eq 01}$$

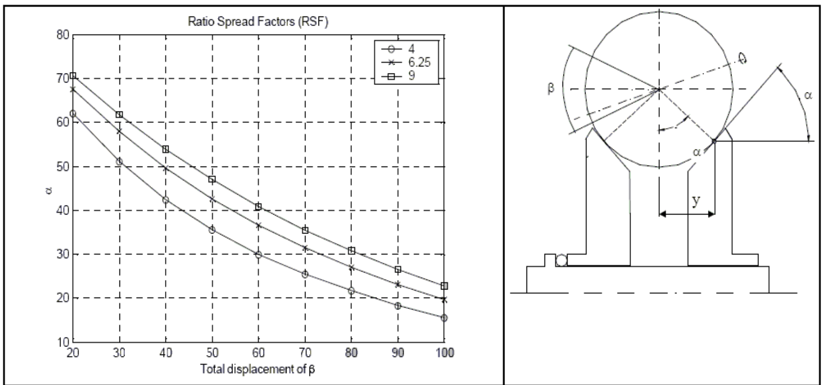
How OD and y are constant, the value of rt is only function of β.

$$rt = \frac{c1 - (y\tan(\beta))}{c1 + (y\tan(\beta))} \quad \text{which } \beta \text{ is the ball inclination angle} \qquad \text{eq 02}$$

By the equation 02, it is possible to observe the distance y is directly linked to the transmission ratio variation. To a well defined transmission ratio, varying from 1:2 to 2:1 (0,5 to 2), the overrange factor is defined to be a relation between the extreme transmission ratios. For this case, which the range is 1:2 to 2:1, the overrange factor is 4, that is, (2/0.5).

$$y = \sqrt{\frac{\overline{OD}^2}{9\tan(\beta_{\max}) + 1}}\text{, being the } \frac{y}{\overline{OD}} = \sin(\alpha) \text{ which } \alpha \text{ is the cone angle.} \qquad \text{eq 03}$$

The graphics above represent the relation between the angles α, β and the ovverrange factor. The analyzed CVT is characterized by : $2\beta_{max} = 60$, ovverrange = 4 and consequently, α = 30 (defined in its design). A control and adjustment tools is such important to set the wished condition of transmission.



(a) (b)

Figure 3 – Graphic of α and β as a function of the overrange factor.

4. CONTACT WITH SLIP SPIN CONDITIONS

The study of contact between two elastic bodies is, generally, much extensive and many scientists, during the years, proposed solutions to diverse kinds of problems about contact between surfaces and any tribological interactions. The theory, at first moment, developed by HERTZ et al (1880-1890) concerned elastic bodies without friction forces. This one could be used in contact between two spheres, sphere and plan, two cylinders and cylinder and plan. After this theory was improved (generalized) to attend any bodies contact problem.

The pressure distribution is considered elliptical for the most general case, which the nearest surface to the contact is admitted to be a elliptical parabola. The next equation define the distance Z from a generic point M and the distance between two bodies can be solved like:

$$\frac{Z}{c} = \frac{x^2}{a^2} + \frac{y^2}{b^2} \quad \text{and} \quad Z_1 + Z_2 = Ax^2 + By^2 \qquad \text{eq 04}$$

By geometrical relations, there could be solutions for A and B, two equations that depends only on the contact surface's main curvatures and the angle between them. So it could be expressed like:

$$A + B = \frac{1}{2}\left[\frac{1}{R_{x1}} + \frac{1}{R_{y1}} + \frac{1}{R_{x2}} + \frac{1}{R_{y2}}\right] \qquad \text{eq 05}$$

$$A - B = \frac{1}{2}\left[\left(\frac{1}{R_{x1}} + \frac{1}{R_{y1}}\right)^2 + \left(\frac{1}{R_{x2}} + \frac{1}{R_{y2}}\right)^2 + 2\left(\frac{1}{R_{x1}} + \frac{1}{R_{y1}}\right)\left(\frac{1}{R_{x2}} + \frac{1}{R_{y2}}\right)\cos^2 \psi\right]^{1/2} \qquad \text{eq 06}$$

Considering R_{x1} = the curvature radius of the element 1 on the x plan
R_{x2} = the curvature radius of the element 2 on the x plan
R_{y1} = the curvature radius of the element 1 on the y plan
R_{y2} = the curvature radius of the element 2 on the y plan
ψ = the angle of leaning between the parabolas

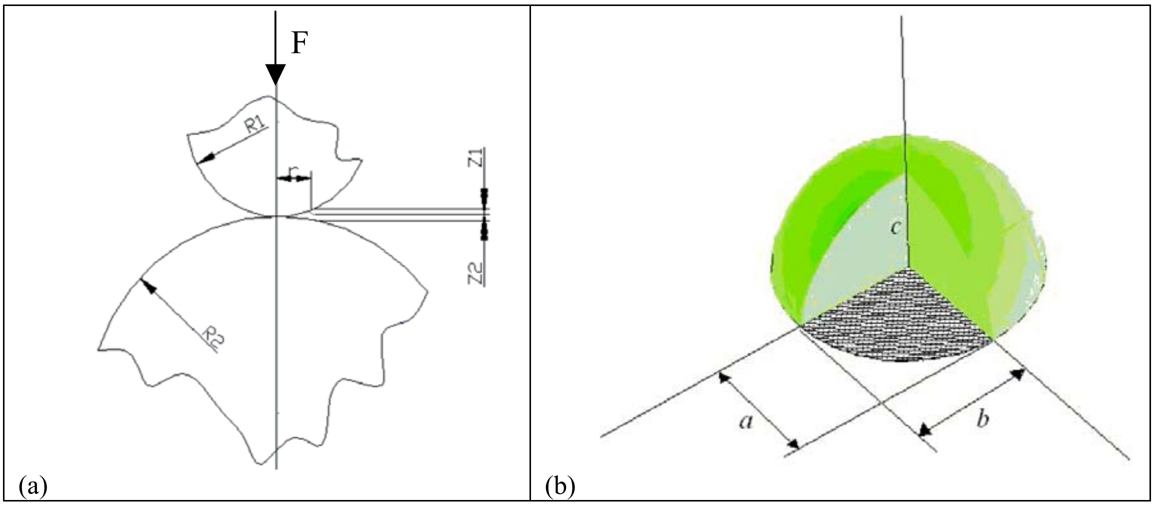


Figure 4 – Contact between two surfaces under force F (besides the pressure distribution)

The pressure distribution becomes only function from the geometry and force:

$$q = q_0\left(1 - \frac{x^2}{a^2} - \frac{y^2}{b^2}\right)^{1/2} \quad \text{which} \quad q_0 = \frac{3}{2}\frac{F}{\pi ab} \qquad \text{eq 07}$$

The variables *a* and *b* are result of the material deformation due the applied force F.

$$a = 1.926\left(\frac{3}{2}\left(\frac{F}{A+B}\right)\left(\frac{1-\upsilon}{E}\right)\right)^{1/3} \qquad b = 0.604\left(\frac{3\pi F}{(A-B)^2}\left(\frac{1-\upsilon^2}{E}\right)^2\right)^{1/3} \qquad \text{eq 08}$$

A simplified model to analize the lifetime under contact fatigue in traction drive transmissions is used. For a roller element made of steel, the number of stress cycles before failing is given by the next equations which is a way from the Lundberg Palmgreen to predict the lifetime of fatigue contact spots. This can be applied to gears, rollers and other elements. As the maximum shear cyclic stress that occurs in a deep Zo like a relative measure that the crack should go before reaching the surface and causing fails.

The total stressed volume (Vz), by Lundberg and Palmgreen theory and used by COY in his calculations, could be done like :

$$Vz = aZ_0 l = 2\pi aRZ_0 \qquad \text{eq 09}$$

The L_{10} Lifetime in only one elliptical contact can be calculate as:

$$L_{10} = \left(\frac{1{,}428.10^{95} Z_O{}^{7/3}}{\tau_0{}^{31/3}.Vz} \right)^{9/10}$$ eq 10

The stick-slip area in the contact could be calculated by the FASTSIM algorithm created by Kalker in his paper. With this, it is possible to determine by the behavior of the shear traction and slip, simplified to a elastic displacement of the material in the tangential direction. The main equations are solved by using this method that associates to the Coulomb's law :

$$\frac{w_x}{V} = \upsilon_x - \phi.y - \frac{dl_x}{dx}$$ eq 11

$$\frac{w_y}{V} = \upsilon_y + \phi.x - \frac{dl_y}{dx}$$ eq 12

which w is the slip,
V is the rolling velocity,
l is the deformation
υ is the creepage

Like in the most of the transmissions, the CVTs need of a special lubrication whose main function is increase the traction coefficient, with out increase the stress levels on the contact spot. The equations that drive the lubrication state on the contact are named by Reynolds equations and they determine the thickness of the lubrication layer .

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\eta}\frac{\partial P}{\partial x}\right) + \frac{\partial}{\partial y}\left(\frac{\rho h^3}{12\eta}\frac{\partial P}{\partial y}\right) = \frac{\partial}{\partial x}(\rho u h)$$ eq 13

considering P as the pressure
h as the lubrication layer's thickness
ρ as the lubricant density
η as the dynamic viscosity

As a understanding of the physical mechanism for the movement transmission on the lubricated contacts is very complex and it concerns many factors. So it was decided to established , after the 70's, a determined condition to analyze the characteristics of these contacts.

Associated to these contact conditions, under EHD lubrication, the term "Traction" is itself a measure derived from the lubricant proprieties (thickness) and involved forces on the contact spots (pressure).

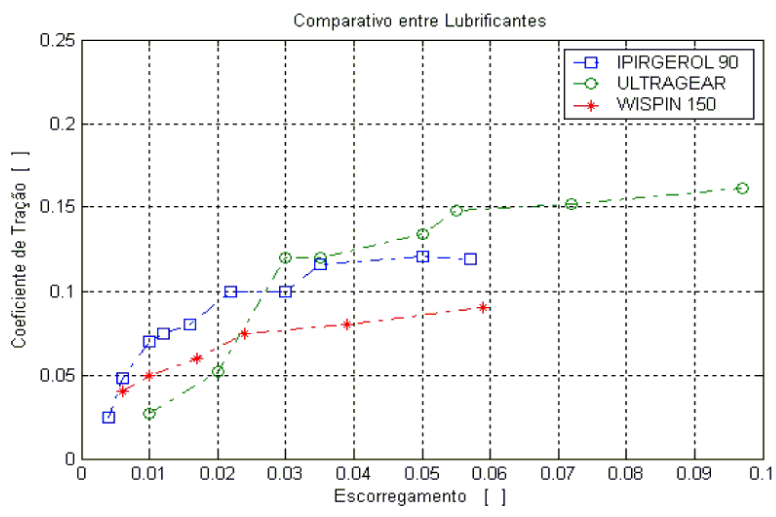


Figure 5 – Traction coefficient of conventional lubricant oils

The traction phenomenon involves many effects like the rolling friction, the slip friction and the pressure and temperature effects. To ensure the traction, it is usual to use the traction coefficient, given by the ratio between the tangential force and the normal force when applied in each contact. This coefficient is determined to specific conditions of temperature, velocity, pressure, and slip. With its use, distinct geometries can be compared, specified oils and the lubrication type too.

Considering two bodies under a normal and tangential load on the contact spot. Naturally it could be assumed a Hertzian load on this area and the normal pressure distribution can not be affected by the tangential forces. It the condition of slip occurs on the entire contact region, the maximum traction force T would be $C_t.P$, which C_t is the traction coefficient to this slip/spin condition. So, the shear stress τ_0 will be given by $C_t.q_0$.

Meanwhile the slip is dependent not only of the geometry of the contact but the load and the traction coefficient too. The spin is dependent only of the geometry contact. It occurs generally when the roller elements present rotation axis not parallel. This characteristic is unavoidable in every traction drives CVT that changing its configuration to vary velocity. The spin factor is introduced as:

$$\xi = \frac{\omega_s}{\omega_{con}} = \sin(\alpha) - (1+e)\frac{\omega_{esf}}{\omega_{con}}\sin(\beta - \alpha) \qquad \text{eq 12}$$

which $e = \frac{\Delta U}{\overline{U}}$ is the slip ratio.

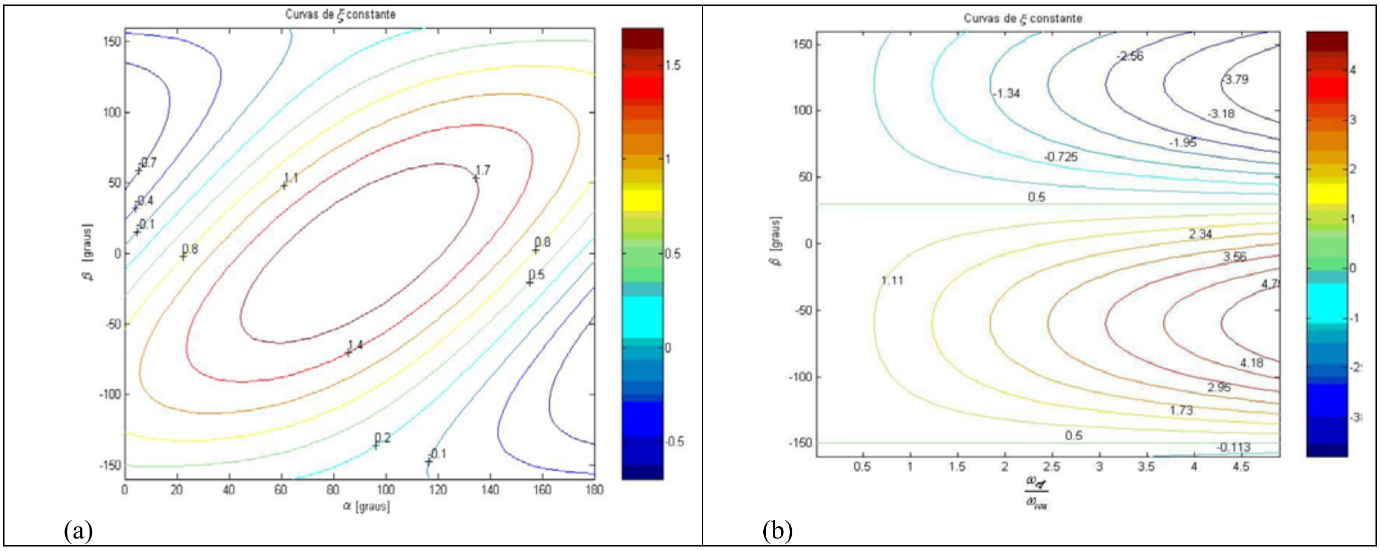


Figure 6 – Spin Factor on the contact spot for a Track-Ball CVT with overrange=4

5. LIFETIME CHARACTERIZATION

Considering a simple case for that transmission, whose the use is exclusive for bikes. What a cyclist with a drag coefficient of 0,4 at a speed of 36 km/h on a straight line, pedaling in a rotation of 72 rpm supplies 254 W to be transmitted from the inner cone to the outer one. So, for the case the transmission ratio varies from 1:2 to 2:1, the results according to the next table is:

Table 1 – Characterization of Contact Lifetime

Transmission Ratio	0,50	0,60	0,70	0,75	0,80	0,90	1,00	1,11	1,25	1,33	1,43	1,67	2,00
Slip Outer Cone [%]	0,50	1,12	1,47	1,75	2,00	2,23	2,42	2,60	2,75	2,90	3,10	3,22	3,38
Contact Efficience [%]	96,0	94,2	93,4	89,4	86,9	85,5	84,6	85,5	86,9	89,4	93,4	94,2	96,0
L10 Lifetime [h]	2020	2920	3470	3800	4030	4350	4500	4350	4030	3800	3470	2920	2020

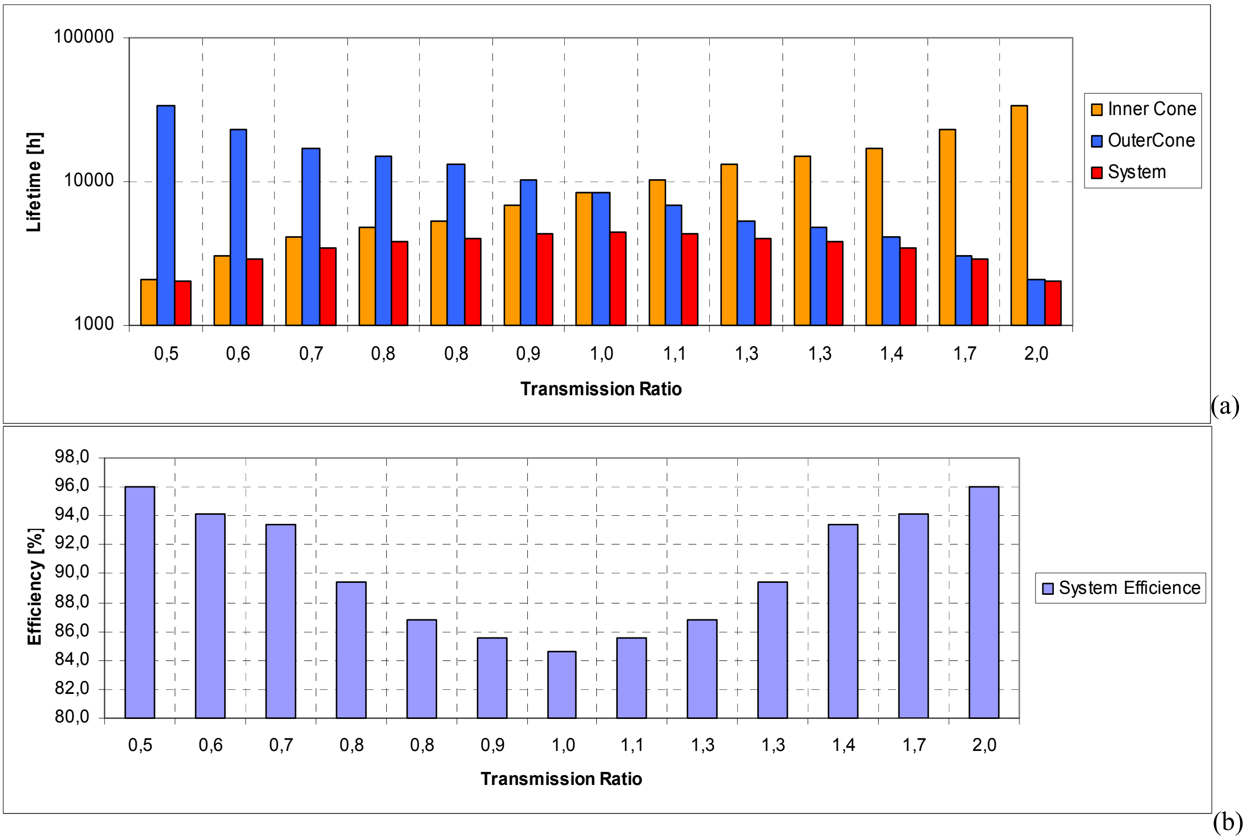


Figure 7 – Graphic from the table 1,(a) Lifetime and (b) efficiency of the CVT transmission

The load depends on the involved forces, with the mentioned data, it is possible to calculate the pressure distribution. The next figures show the stress level associated to the reduction 1:2 condition, for the outer cone.

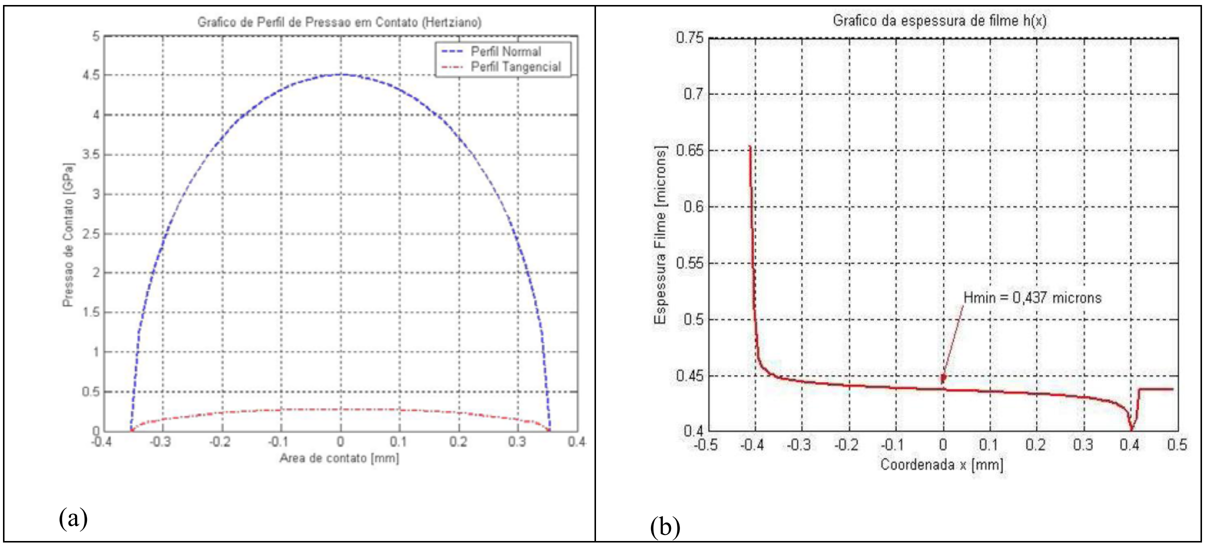


Figure 8 – Pressure distribution (Normal and Max Tangential) and lubricant thickness

The contact spot behavior, related to slip and spin phenomena, could be solved by the Kalker method. As it can be observed in the next figures, the slip in the contact area is function of the load, velocity and the material proprieties. The figure (A) show how the contact spot changes with the transmission ratio, keeping constant the load and the creepage proprieties. It could be seen slip condition increase in the contact spot of the outer cone when the transmission ratio also increase. The second one (B) show the contact spot as a function of the load, when the normal load increases, also the stick region increases, but not necessarily proportional as it could be seen in the equation 11 and 12.

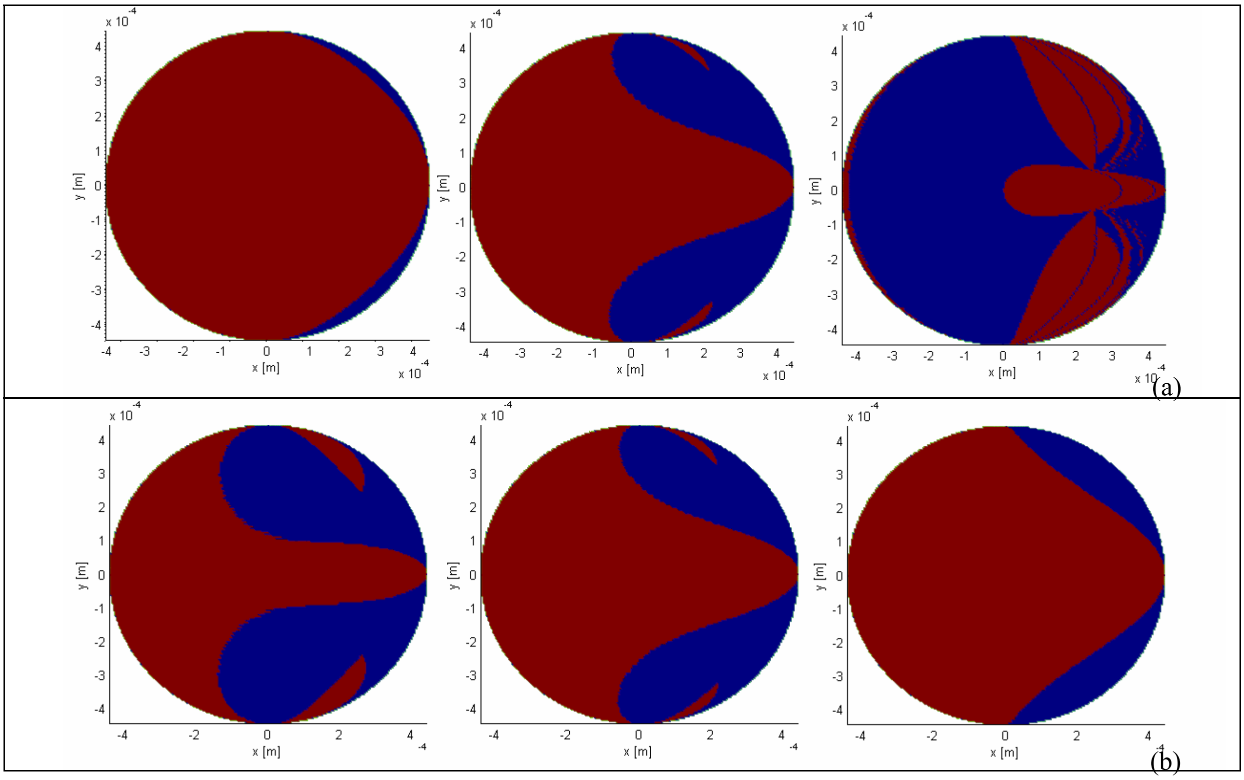


Figure 9 – Behavior of contact spot, the blue color represent the slip (creeping region boundary), the red one the stick . (For the outer cone)

It can be possible to see in the next graphics that even a transmission ratio different from other, the shear stress gradient in the cone is not such modified in its shape but the absolute value. It is true because the normal load is more intense than the tangential forces, even with slip and spin. The same happens to the Von Misses stress.

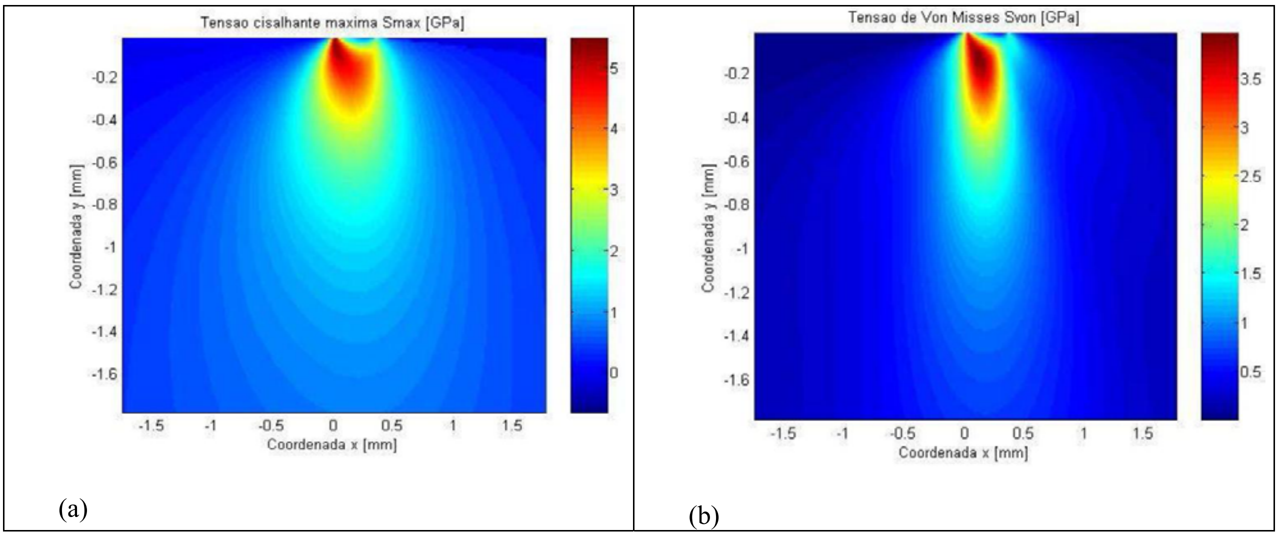


Figure 10 – (a) Max shear Stress distribution and (b) Von Misses Stress distribution of the inner cone.

It can be observed the transmission ratio 1:1 is the condition whose Lifetime is higher, however the sum of spin factor in both cones is high, making the efficiency to decrease. It is important to emphasize when the transmission is being more used in the better Lifetime prediction range (ratio 1:1), higher the cumulative Lifetime in a period.

If it was possible to increase the overrange of the transmission, supposing it has the same constructive shape but the overrange to be 6.25 and also 9. To calculate the Lifetime for these initial conditions of load, it is necessary that the geometry must be modified, preferably the coning angle α. The other geometric proprieties should be constant to make it easier to calculate. Next figure show how the Lifetime is according the power and ovverange for a 1:1 transmission ratio.

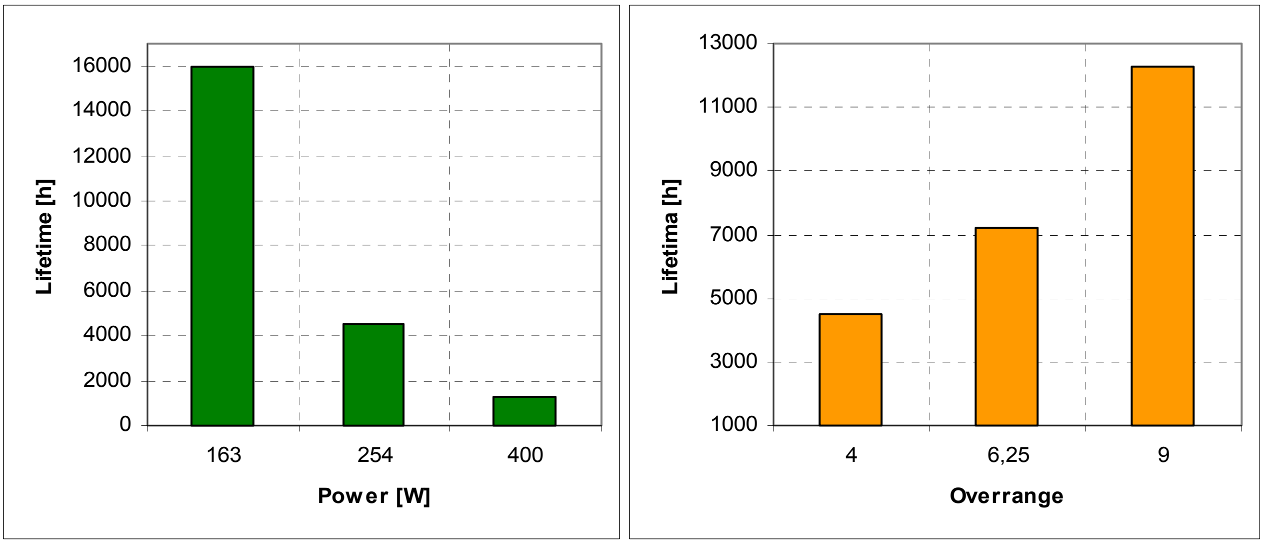


Figure 11 – The influence of load and the overrange in Lifetime calculation (Only for ratio 1:1)

It can be concluded that the overrange is also a factor which influents such in the conditions of a CVT transmission like the Track-Ball. It was noted the bigger overrange has a bigger Lifetime, it is due to the minimization of forces and stresses. In the opposite, when the load is higher, the Lifetime is lower than the other cases because the load is directly influent into the Lifetime calculation. So, it is more convenient to chose a CVT that operates in a optimized condition like a transmission ratio near 1:1 and a big overrange.

If a summary could be done about this CVT, it would have been concluded that the inclination, the sphere radius and the inclination are determinant into the conditions of spin and slip. The contact spot area decrease, but it was not computed, which there is not a more completed and complex model currently.

6. CONCLUSIONS

The efficiency is determined by the balance of the cyclist power, slip and spin losses and it is related to power losses due to these factors. It is higher in the contact whose ball inclination is near the coning angle, however it is impossible to cancel it due the CVT transmission and actual power losses.

As there are two cones, one will operate with a high spin factor while the other one will operate with a lower. So when the CVT operates in a transmission ratio 1:1, the efficiency is the best, or the highest.

It was possible to see how the contact between two surfaces is and how variable is influential to this. The contact area changes according the load and stick-slip condition.

The lifetime is dependent only the physical characteristics like material proprieties and the external characteristics like loads, slip-spin condition, which is clearly shown in this paper. The load conditions and the general results (lubrication thickness, contact spot area, shear stress) is near the result obtained by the other authors in their papers.

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