A PROPOSAL FOR AN OPTIMIZATION OF A WILSON GEARBOX USING NUMERICAL SIMULATION

Bruno Rodrigues de Sunti, brunodesunti@hotmail.com
Geraldo Gonçalves Delgado Neto, geraneto@fem.unicamp.br
Marcelo Gennari Tanikawa , marcelo.tanikawa@gmail.com
Breno Raizer, breno.raizer@gmail.com
Franco Giuseppe Dedini, dedini@fem.unicamp.br
School of Mechanical Engineering - Unicamp; Mendeleyev Street, 200 - Barão Geraldo - Campinas - SP

Abstract. *The continuous increase in the number of automobiles, mainly in large cities, has increased the demand for compact cars with great internal space and that are simultaneously agile, efficient, economical and reliable. The gearbox has direct influence on these characteristics and it does motivate the study of the Wilson gearbox. The Wilson gearbox will supply the required by today's market, because among other features it is small in size with a high energy density, which reduces the vehicle's weight increasing its efficiency. Wilson gearbox uses epicycle gears, so it has the capacity to transmit high torque being silent enough it is a manual gearbox of ease automation and, operates on a simple way: to obtain each transmission ratio an external ring just need to be braked. During the First World War the Wilson gearbox was used in combat tanks and afterwards in England buses, but unfortunately due to its complex construction for the time it fell into disuse. An unfavorable characteristic of the Wilson gearbox is its difficulty in obtaning the desired transmission ratios. Searching for a solution to this problem, the kinematic modeling with four sets of epicycle gears and a clutch, providing four forward gears and one reverse gear, was made. This model will be used to optimize the construction parameters and the transmissions ratio of the kinetic pair of an epicycle gear through numerical simulation. The results will be close to an ideal torque curve.*

Keywords*: Wilson gearbox, automobiles, epicycle gears, transmission ratio, numerical simulation*

1. INTRODUCTION

The use of planetary gear is diversified within the automotive and aerospace engineering, and we can also highlight the use in machines as drills and washing machines. A gear train is called an ordinary gear train if all the rotating shafts are mounted on a common stationary frame, and a planetary gear train (PGT) or epicyclic gear train (EGT) if some gears not only rotate about their own joint axes, but also revolve around some other gears (Tsai, L.W., 2001). PGTs are transmission systems of high kinematic complexity and difficult visualization. However, their advantages are many: they are compact, light, allow high speed reductions, and multiple transmission ratios. Due to their constant engagement, bifurcating capacity and power addition they are considered highly reliable (Becker, M. et al., 2003). In addition, Chen, Y.F. et al., (2009) highlights many PGTs advantages including the large torque-toweight ratio and high efficiency, and emphasizes its widely used in the transmissions of helicopters, heavy machinery and marine vehicles. Besides its many uses, these transmissions have a large number of construction possibilities which makes them extremely versatile.

1.1. Wilson Gearbox History

According to Arias Paz (1958), the use of epicyclic gears in gearboxes began in the early days of automobiles in models like the Oldsmobile and Lanchester and later on Ford used them in his famous model T.

In 1929 the use of planetary gears in automotive transmissions starts in a model called Wilson Gearbox, created by Walter Gordon Wilson. Born in Blackrock in Ireland in 1874 and graduated in mechanical engineering, Walter Gordon Wilson got involved with the British military industry around 1904. During the First World War he incorporated planetary gears in the Mark V tank, allowing the tank to be operated by a single driver instead of the four previously required.

After the war, in 1928 Wilson improved the transmission used in the Mark V tank, giving rise the pre-selector gearbox. In a partnership with JD Siddeley he created the company Improved Gears Ltd to develop the commercial project of the Wilson Gearbox, the company later became Self-Changing Gears Ltd.

Afterwards in England the Wilson gearbox was applied in buses. However, due to its complex construction for the time it fell into disuse.

1.2. Types and operating mode of the Wilson gearbox

The coaxial layout of the elements in a planetary transmission makes it ideal for use of brakes and clutches which can be applied in such a way to select a transmission ratio without interrupting the torque flow. The operating mode of the Wilson gearbox is simple. To obtain each speed ratio it is necessary to break the corresponding external cylinder, so the rotation of the shaft coupled to the flywheel forces the system to rotate, obtaining the desired rotation in the transmission - output shaft. Besides a direct transmission ratio being obtained when the clutch engages making all the external cylinders to rotate freely on the same flywheel speed without any relative motion between the gears, resulting the same rotation in both - the output shaft and the input shaft.

Historically were used two types of Wilson gearbox, one with two transmission ratio forward, one reverse and the direct drive and another type with three transmission ratio forward, one reverse and the direct drive, meanwhile Wilson gearboxes with more transmissions ratios can be obtained. The latter type is object of the present study as shown in Fig. 1.

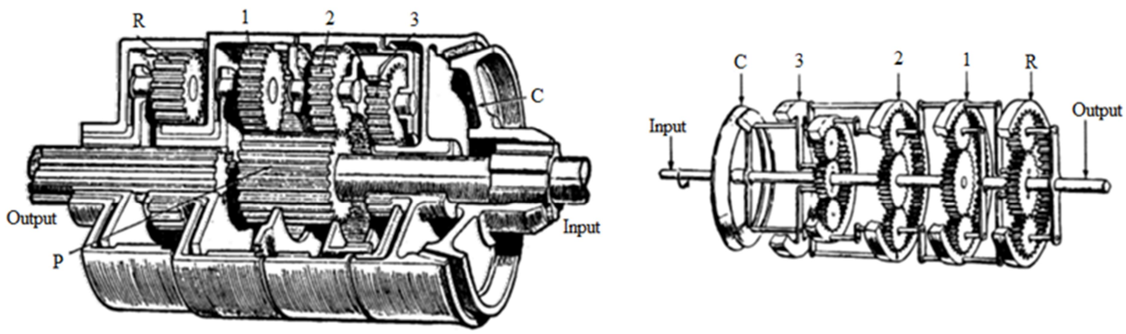


Figure 1. Wilson gearbox (Adapted from Arias Paz, 1958).

The Wilson gearbox of the Fig. 1 is composed of four external cylinder forming the first, second, third and the reverse speed and a clutch that when engaged makes the direct drive. Various modes of braking can be used, among others a belt brake, a brake shoe or even a disc brake plugged to the outer cylinder.

An important feature of the Wilson gearbox is that all the gears are constantly engaged enabling a noiseless operation and a easily automatization, besides the transmitted effort being distributed into a large number of teeth on gear enabling the transmission of high torques.

Robotized or automated gearboxes are deliberately derived from manual gearboxes, including the friction clutch. These cannot be defined as semiautomatic gearboxes because their control systems can handle all automatic functions. The decision to use a modified manual gearbox can be justified by other goals, such as reducing the cost of production, comparison with automatic gearboxes. We should not forget, finally, a positive impact on improving sports car performance, because shifting and start-up times can be reduced to figures not readily available even to professional drivers. This last category of gearboxes is attaining a significant market share and could have further development in coming years (Genta, G. and Morello, L., 2009).

2. TRANSMISSION RATIO CALCULATION

For the transmission ratios calculation of a Wilson gearbox it is necessary to analyze all the planetary simultaneously and it is possible if the operating mode of a single planetary is known. Aftermost a study in the relation between the planetary sets in a Wilson gearbox is required.

2.1. Planetary transmission ratio

Analysis of planetary gear ratios is more complex than the analysis of simple gear trains, but it proceeds straightforwardly from a basic kinematic analysis of the elements (Stone, R. and Ball, J.K., 2004).

An epicyclic transmission is composed of three parts: the sun gear in the middle of the system, the ring gear that is a wheel with inner teeth which compose the external part of the system and the planet gears assembled on a carrier which rotates around the central shaft of the set meshing between the sun gear and the ring gear. Sun, planetary carrier and ring gear are concentric organs (Mathis, R. and Remond, Y., 2008). This terminology is due to the fact that a point, which belongs to the planet gear, possessing a mobile shaft, describes an epicyclic curve.

A free planetary has two degrees of freedom. So a planetary gear set requires that one element must be held for occur torque transfer, turning it into a system of a single degree of freedom with only one input and one output. Six alternative assembly arrangements are possible in a simple PGT alternating the input and the output power element. Thereby the transmission ratio can be given by:

$$RT = \frac{\omega_{in}}{\omega_{out}} \tag{1}$$

Where $\boldsymbol{\omega_{out}}$ is the output shaft rotation and $\boldsymbol{\omega_{in}}$the input shaft rotation.

To obtain the PGT transmission ratio in each alternative its required to know the basic reason "b" of the mechanism, that is given by:

$$b = -\frac{Z_S}{Z_A} \tag{2}$$

Where **Zs** is the sun gear teeth number and **Za** the external gear teeth number.

The negative sign means that fixing the carrier the ring gear rotates in the opposite direction of the sun gear rotation. The basic reason establishes a relationship between the diameters of the sun and ring gears. For a plan planetary the value of "b" may range from -1, when both gears have the same diameter and the planets have null diameter and 0, that is the case where the sun gear diameter is null and the planets have the diameter equal to the radius of the external gear. For practical situations the value of "b" is usually between -1/3 and -2/3.

With the parameter "b" set and analyzing the relations of speeds of the planets and some dimensional parameters of the epicyclic transmission, it is easily demonstrated that the basic equation of a planetary is given by:

$$\omega_A - b.\omega_S + \omega_b.(b-1) = 0 \tag{3}$$

Through this equation it is possible to plot the graphic of the Fig. 2 that relates the transmission ratio with the basic reason "b" for the six alternatives PGT assembly. In this graphic it's possible to analyse in which regions will have reduction, multiplication and inversion.

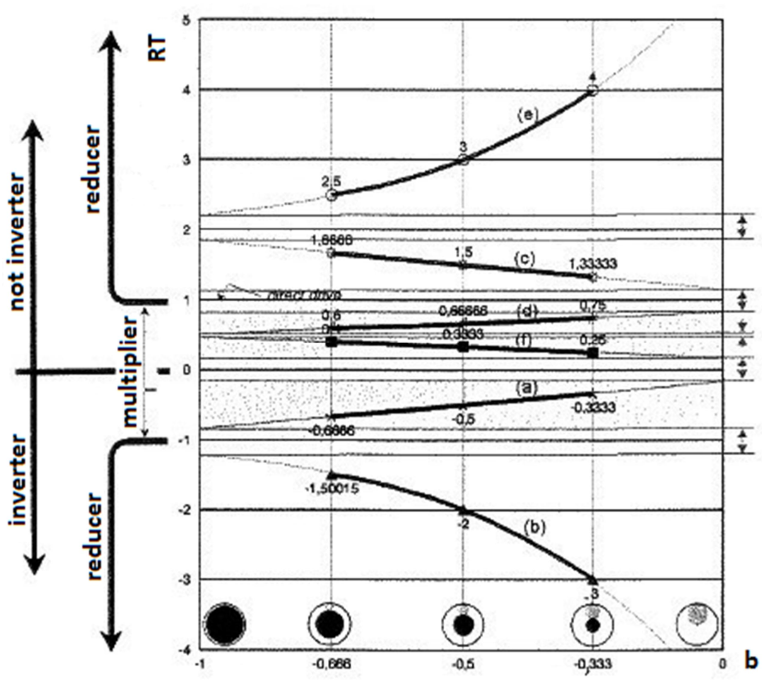


Figure 2. Transmission ratio in function of the basic reason "b" (Adapted from Amaral, D., 2000).

The graph shows the great variety of reductions and multiplications that can be obtained with a simple PGT. It is observed that there is some ratios impossible to obtain with only a PGT. So, due this large number of possibilities constructing, this kind of gearbox becomes extremely versatile.

2.2. Wilson gearbox map

One of the possible ways to represent the Wilson gearbox is through the use of block diagrams as shown in Fig. 3. Through this representation it is easy to understand its operation mode.

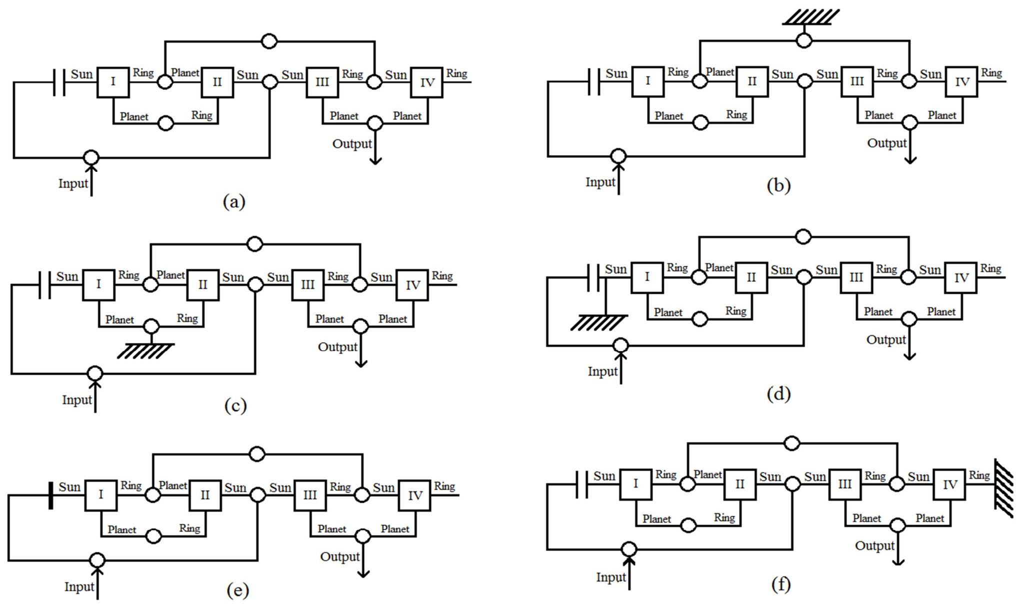


Figure 3. Block diagram of a four speed Wilson Gearbox

In Fig. 3 the first case (a) represents the gearbox when the vehicle is disengaged, (b) represents the first speed engaged, in other words with the ring gear of the planetary III braked (c) the second speed engaged, in other words with the ring gear of the planetary II braked, (d) the third speed engaged, in order words, the outlet portion of the clutch braked, which corresponds to stop the sun gear of the planetary I, - (e) direct drive, in other words, when the clutch is engaged and (f) the reverse speed which is obtained by braking the ring gear of the planetary IV.

2.3. Elements association in a Wilson gearbox

Another possible way of representing the Wilson gearbox is shown in the Fig. 4. Analyzing this representation the association between the planetary elements of the Wilson gearbox becomes clear. The association of an element of a planetary with an element of another planetary implies angular velocity restriction, in other words both elements have the same angular velocity.

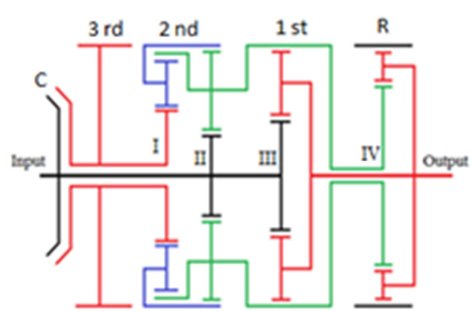


Figure 4. Relationship between the planetary elements

With the possible transmission ratios obtained in a planetary and knowing the rotation relations between different planetary elements that constitute the Wilson gearbox, it is possible through a kinematic description to model the final transmission ratios of the Wilson gearbox.

2.4. Wilson gearbox transmission ratio

To model each of the transmission ratio, first it is settled which rotations are used as the transmission generalized coordinates being the rotation of the input shaft ω_{in} known for all cases. (Morais, R. F., 2004).

All analyses were performed only for the permanent state, that is, after the rotations of the planetary are fully established.

For the first transmission ratio the second coordinate is the external gear rotation of the planetary 1, ω_{A1}, that is the component braked and the rotation to be determined is the output shaft rotation, ω_{out}.

Knowing the output shaft rotation it is possible to establish the first gear transmission ratio. The generic relationship between the input and output shaft rotations is:

$$\omega_{out} = D_1 . \omega_{in} \tag{4}$$

Adopting the case E of Tab. 1 as it is more appropriate for the situation we have:

$$\frac{\omega_{out} - \omega_{A1}}{\omega_{in} - \omega_{A1}} = \frac{b_1}{b_1 - 1} \tag{5}$$

Where ω_{A1}=0:

$$\omega_{out} = \frac{b_1}{b_1 - 1} \omega_{in} \tag{6}$$

The same procedure is carried out to the others transmission ratios as shows Morais, R. F. (2004) and the results are presented in Tab. 1.

Table 1. Wilson gearbox transmission ratio

Element	Relation	D
ω_{out}	$D_1 . \omega_{in}$	$\frac{b_1}{b_1 - 1}$
ω_{out}	$D_2 . \omega_{in}$	$\frac{\frac{b_2}{b_2 - 1} - b_1}{1 - b_1}$
ω_{out}	$D_3 . \omega_{in}$	$\frac{\frac{\frac{b_2}{b_A - b_A . b_2 + b_2} - b_2}{1 - b_2} - b_1}{1 - b_1}$
ω_{out}	$D_4 . \omega_{in}$	1
ω_{out}	$D_R . \omega_{in}$	$\frac{b_1 . b_R}{(b_R - 1) - b_R . (1 - b_1)}$

3. VEHICULAR APPLICATION

Achieved the Wilson gearbox synthesis was simulated its application in the vehicle performance by changing the basic reasons "b" value of the planetary sets by varying the gears teeth numbers.

3.1. Vehicular performance

Vehicle performance analysis is possible only if a study of their mechanical and geometrical characteristics is carried out. With this knowledge it is possible to use models to calculate performance values.

The inertia involved in the vehicle movement is essential to its performance and consequently in their energy consumption. The total vehicle inertia consists of the translational inertia and the rotational inertia, in other words the inertia of the engine, transmission and wheels. The combination one inertia with the other can be done using the sum of the vehicle mass and the equivalent moving parts mass. This is done by calculating the mass factor given by Eq. (7) and (8) - (Gillespie, T. D., 1992).

$$mass\ factor = \frac{M_V + M_r}{M_V} \tag{7}$$

$$mass\ factor = 1 + 0{,}04 + 0{,}0025.N_{tf}^2 \tag{8}$$

Where M_V is the vehicle mass, M_r the equivalent moving parts mass and N_{tf} the transmission ratio considering the differential.

The vehicle capacity to overcome the resistive efforts and the motion contrary forces determines its performance. Figure 5 represents the main forces opposed to the vehicle motion.

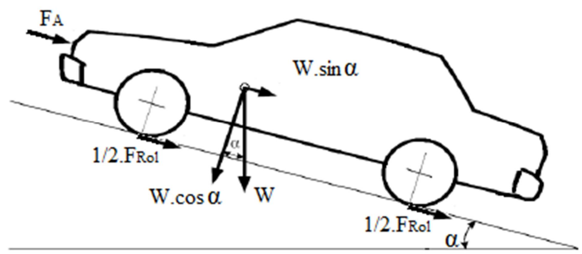


Figure 5. Main resistive efforts of the vehicle, adopted of Ehsani, G., et al. (2005).

The rolling resistance force, F_{Rol}, arises from the interaction of the tires with the ground and may be given by Eq. (9).

$$F_{Rol} = W.f_r.\cos(\alpha) \tag{9}$$

Where W is the vehicle weight, f_r the rolling resistance coefficient and α the road angle.

The rolling resistance coefficient is influenced by many factors being difficult an equation that lists all them. Therefore, one of the most accepted approaches is given by Eq. (10) described by (Ehsani, G., et al., 2005).

$$f_r = 0{,}01.(1 + \frac{V}{100}) \tag{10}$$

Where V is the vehicle velocity in km/h.

The aerodynamic forces, $F_A$resulting from the interaction of the vehicle body with the fluid comes from two sources: viscous friction and pressure differential due to the shape of the vehicle and may be described by Eq. (11).

$$F_A = \frac{1}{2}A.C_d.\rho.V^2 \tag{11}$$

Where A is the vehicle frontal area, C_d the drag coefficient, ρ the fluid mass density and V the vehicle velocity.

The vehicle mass interaction with the road angle produces a component in the direction of motion, F_I, which is in favor of the motion when the downhill and against the motion when uphill (Ehsani, G., et al., 2005). This component is given by Eq. (12).

$$F_I = M_V.g.\sin(\alpha) \tag{12}$$

With the resistive forces defined with Newton's second law it is possible to reach the equation of vehicle longitudinal motion which consists of the traction and resistance forces to motion, Eq. (13).

$$(M_V + M_r).\frac{dV}{dt} = F_{traction} - F_{res} = \left(F_{tf} + F_{tb}\right) - (F_{Rol} + F_A + F_i) \tag{13}$$

Where F_{tf} is the front force traction, F_{tt} the back force traction and $\frac{dV}{dt}$ the vehicle acceleration.

Manipulating Eq. (13) it is possible to obtain Eq. (14):

$$F_{traction} = (M_V + M_r)\frac{dV}{dt} + 0{,}01.\left(1 + \frac{V}{100}\right) + \frac{1}{2}A.C_d.\rho.V^2 + M_V.g.\sin(\alpha) \tag{14}$$

Thus, besides the speed, acceleration is one of the factors that significantly increases the traction force required in the vehicle.

The speed variation at every moment creates different forces required for traction, which multiplied by the speed result in the power required to move the vehicle.

$$P_{required} = (M_V + M_r)\frac{dV}{dt}V + 0{,}01.\left(V + \frac{V^2}{100}\right) + \frac{1}{2}A.C_d.\rho.V^3 + M_V.g.\sin(\alpha).V \quad (15)$$

3.2. Engine characteristic curves

The torque demand presented is in the wheel and need a fix for the application of the equations in the engine. This fix will consider the overall efficiency of transmission, the transmission ratios and the differential transmission ratio as shown in Eq. (16).

$$F_{tengine} = \frac{F_{traction}}{i_T.i_D.\eta_T} \quad (16)$$

Where i_T is the gearbox transmission ratio, i_D the differential transmission ratio and η_T the transmission efficiency.

The internal combustion engines are limited by the range of rotation speed between slow gear and maximum rotation. The maximum power and torque values are not uniformly offered and are available in specific operation bands. The transmission ratios suit the available torque to the traction force required (Bosch, R., 2005).

An ideal engine would be one that produces constant maximum power regardless of the speed. In vehicle dynamics to achieve results close to this the transmission is introduced making the engine work in a region near the maximum power, therefore the power in the wheels remains nearly constant at the maximum value. Similarly, the torque at the wheels should be similar to an ideal engine torque, according to the envelope curve (Jazar, R. N., 2008). Figure 6 shows a graph of torque at the wheels in a vehicle with six transmission ratio gearbox.

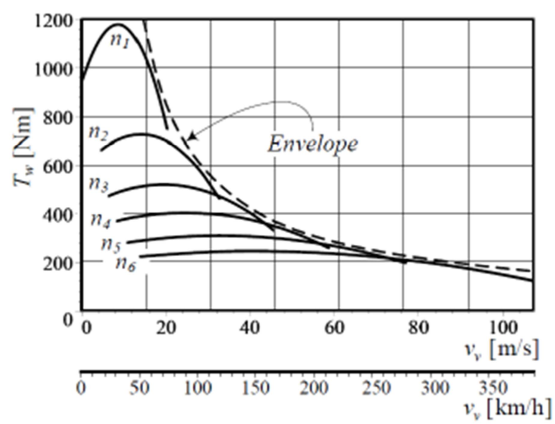


Figure 6. Graph of torque at the wheels in a vehicle with six transmission ratio gearbox (Jazar, R. N., 2008).

4. SIMULATION RESULTS

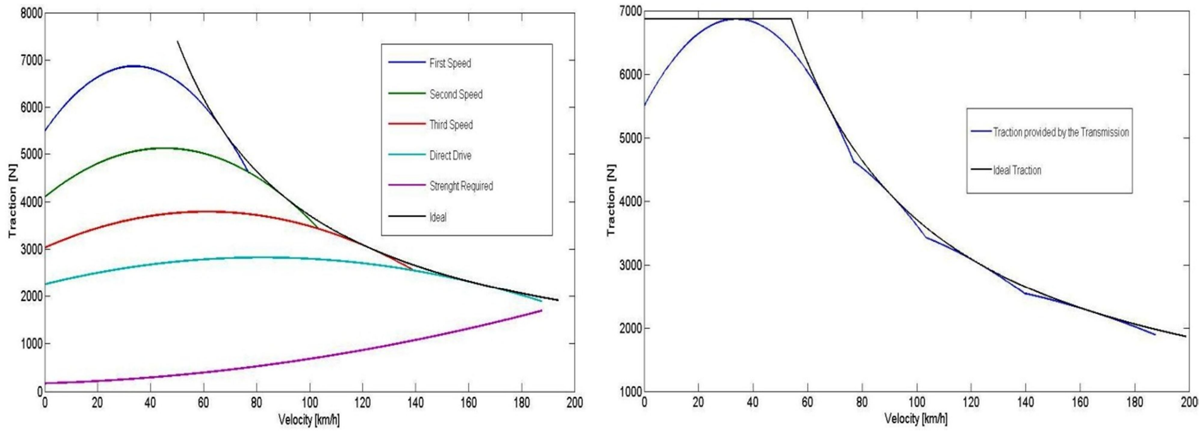
In order to obtain the Wilson gearbox efficiency for many transmission ratios its use has been simulate with the parameters of two vehicles on the market today a 2000 cc vehicle X and a vehicle Y with 1000 cc both originally with a commercial gearbox with 5 speed.

The planetary transmission efficiency or losses are made up of three main parts: Gear meshing motion pair of friction losses, bearing running friction losses and stir oil loss (Mao, J. and Hao, Z. Y., 2011).

The gearbox efficiency can be available by the reason of the ideal traction hyperbola area and the effective traction hyperbola area (Lechner, G. and Naunheimer, H., 1999).

This method of comparison of the available torque curves at the wheels with the ideal curve of maximum torque at the wheels will be used to calculate the efficiency of the Wilson gearbox using the software MATLAB. The result of this procedure depends on the scheduling of the transmission ratios. The ideal curve indicates the maximum torque possible if the scaling allowed a continuous acceleration, with use of maximum torque for each speed.

First it is required to trace the curves of available torque at the wheels, showing the torque for each speed ratio in terms of vehicle speed and also the ideal torque curve, as illustrated in Fig. 7a. Subsequently, it is necessary to trace on the available curve a real torque curve, which hopes to use the maximum power for each speed, as shown in Fig. 7b. This curve simulates a vehicle accelerating from 0 km/h to its maximum speed through all the speed ratios while maintaining the highest torque level possible to achieve the maximum acceleration.



(a) (b)

Figure 7. Graphic obtained in a simulation for vehicle X

Finally raised these curves, we can compare their areas, getting an efficiency ratio that indicates how much the actual torque curve approaches the ideal torque curve. A suitable domain is used to calculate the area, considering the ideal torque curve only between the maximum torque in first gear on the vertical axis and the maximum vehicle speed on the horizontal axis, as illustrated in Fig. 7b. How better the use of the torque provided by the engine higher is the efficiency.

Several transmission ratios were simulated aiming to achieve maximum transmission efficiency both for vehicle X and Y, the results are showed in Tab. 3. Figure 7 shown the graphic of the vehicle X with the parameters of the Tab. 3 first case which gives the maximum efficiency.

Table 3. Teeth number of gears sets

Vehicle X - 2000 cc							
Planetary				I	II	III	Efficiency (%)
Case	1 st	Ring	Teeth Number	61	61	33	94,42
		Sun		19	19	23	
	2 nd	Ring		67	67	51	94,04
		Sun		19	19	19	
	3 rd	Ring		49	67	57	93,94
		Sun		35	23	25	

Vehicle Y - 1000 cc							
Planetary				I	II	III	Efficiency (%)
Case	1 st	Ring	Teeth Number	59	59	35	92,58
		Sun		21	19	21	
	2 nd	Ring		43	43	43	91,39
		Sun		21	21	21	
	3 rd	Ring		57	43	51	89,79
		Sun		31	27	19	

In the case of vehicle X with 2000 cc the maximum efficiency obtained was 94,42% for the parameters shown in Tab. 3 first case. The original transmission efficiency is 96,82%. This is slightly smaller than the original but is rewarded by the smaller size and lower weight making this difference not significant.

In the case of vehicle Y with 1000 cc the maximum efficiency obtained was 92,58% for the parameters shown in Tab. 3 first situation, being the original commercial transmission efficiency 91,86%.

A good condition that facilitated and cheapened the production was obtained with the number of teeth shown in Tab. 3 second case. With these parameters, 91,4% of efficiency was obtained, which is an excellent value slightly smaller than the original commercial gearbox but with high energy density and smaller size been adapted to a compact vehicle.

It is important to appreciate that the choice of gear ratios in a transmission is often dictated in practice by what is available or what is already in production. This situation occurs because of the large expense involved in engineering new gear sets, and installing or modifying the manufacturing plant to make the new parts. In theory the ratios are often chosen to give constant speed or varying speed increments between the gears (Happian-Smith, J., 2002).

5. CONCLUSIONS AND DISCUSSION

Ranging the teeth number of planetary gears a Wilson gearbox optimization was possible, in other words, it was obtained a Wilson gearbox with high efficiency. Simulated under the same conditions, the original vehicle gearbox efficiency in the case of the vehicle Y was minor to those of Wilson gearbox demonstrating a promising study.

In all cases the maximum speed obtained by the vehicle is limited by rotation cut because the force provided by the transmission is always greater than the vehicle resistive force. If a ratio transmission less than a unity was possible the maximum vehicle speed would likely be limited by the resistive force, but this condition is not possible to be obtained by physical conditions. Maybe the study involving a decrease in the differential transmission was viable, which would compensate this impossibility constructive. Besides that a study of a case associating a Wilson gearbox with two output shafts increasing the number of transmission ratios could be done.

Planetary gearboxes have many advantages: they are compact, strong and allow high-speed reductions because the load is distributed over many teeth. Besides that they possess high reliability because they have constant engagement, which eliminates the risk of damage due to engagement or disengagement allowing a high service life in addition to transmission ratios can be changed with no interruption in torque transfer. They also have bifurcating capacity and power addition, and allow multiple transmission ratios.

Compared with a conventional manual gearbox, this type of planetary gearbox has several advantages such as the gears widths which provide less weight, all elements rotate around the same central axis what provides advantages in packaging, choice of output element, lubrication, and control, beyond what can be easily automated since the system can incorporate the functions of clutch and brake. Advantages such as not requiring some driver skill, clutch operation and changing gears simultaneously can be tiring, especially when in heavy traffic; controls on larger vehicles can be heavy and most require some dexterity during operation.

Conventional manual gearboxes are easy to build up to 5 or 6 speeds, currently up to 8 speeds, but the relations of transmission are restricted by the physical size. Already the Wilson gearbox that has a constructive complexity and before depended on the manufacturing cost of the planetary today shows more interesting by the ease of construction processes and high flexibility of transmission ratios that can be obtained from the system. It is important to emphasize that the operation of the Wilson gearbox is simple despite the construction complexity.

Automatic and CVT transmissions have a hydraulic control fluid so fluidity is important. Ideally, the transmission oil temperature will warm up quickly on startup of the vehicle. This will enable the transmission to operate efficiently for as much of the time as possible. The transmission oil will obviously start near to the ambient air temperature and at low temperatures can be extremely viscous. Particularly if the ambient temperatures are near or below freezing it can be a severe problem. At the other operating extreme we will want to prevent the oil getting too hot and compromising the durability of the oil itself or the components in the unit. It can sometimes be very difficult to control the lubricant to a reasonable upper operating limit when the vehicle is operating in high ambient temperatures and working hard. This is not a problem for the Wilson gearbox since it has no hydraulic control, but rather mechanical control through brakes and clutch. In addition the traditional automatic gearboxes as Simpson and Ravigneaux have complex ratio transmissions synthesis. Almost all CVTs gearboxes also incur some speed ‘slip’ between input and output as well as torque losses, and both output torque and speed will be less than the ideal. Problem that will certainly not occur in this type of gearbox.

Concluding, any specification for a product contains the following requirements: it must be suited to its intended purpose, affordable (economic) and acceptable (environment friendly, easy to operate, and functional). These superordinate development goals define the fundamental performance features of a transmission. Vehicle transmissions have to be designed to provide torque conversion suited to operating conditions, adapted to the consumers requirements and at a competitive price. Thus means specifically service life appropriate to the intended use, low noise level, low weight, high efficiency and ease of operation.

6. ACKNOWLEDGEMENTS

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