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Dynamic Synthesis of a Dual-Clutch Automatic Gearboxes

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Abstract: Relatively recent have occurred dual-clutch automated boxes, which promise a lot in the field and can be used successfully in cars and trains as well as in the aerospace industry. This paper wants not only to present such a gearbox but also to realize its dynamic synthesis, based on the engineering optimization of the mechanical efficiency achieved in each used speed gear. The presented method is original. In the paper are presented the original dynamic synthesis relations, based on the mechanical efficiency of the exchanger, determined separately for each of its speed gear used, for six forward and one on reverse. The applied method of calculation is based on a precise technology used to determine the mechanical efficiency of a gear, depending on the number of teeth of the two wheels in the gear, the angle of the pitch circle and the tilting angle of the wheel teeth in drive.

Keywords: Classic Manual Gearboxes, Automatic Gearboxes, Semi-Automatic Gearboxes, Continuous Variable Gearboxes, Dual-Clutch Automatic Gearboxes, Dynamic Synthesis

Introduction

The automatic gearbox has begun to be used on real-world vehicles in the 1980s. Since it was taken up quite a bit from the airplanes and adapted immediately, it initially presented more problems than those used in aerospace systems. As electronics and automation developed, issues related to hydraulic drives and controls (which were often blocked and easily) were eliminated. Response time was also a real issue, which is why rally drivers and formula I have consistently refused to change their old manual gearboxes with some automatic ones. Today, the response times of a hybrid or automatic gearbox are much shorter and for a normal car, using an automatic gearbox can be a huge simplification of its operation and a way to make it much easier for the driver of the vehicle, a maximum driving efficiency of the vehicle, which also brings a real fuel economy, regardless of the route and even greater in crowded urban areas. The most common are the automatic gearboxes installed on the buses.

Classic automated gearboxes have been made with the help of planetary gears being generally used in an automatic gearbox two or even three planetary gears mounted in series, in parallel or mixed to achieve the necessary gears, coupling and disengaging them automatically using hydraulic clutches and brakes.

The mode of operation of these gearboxes is much more complicated than manual ones, the transmission of power being carried out by one or more planetary mechanisms where the determination of the mechanical efficiency is a little more difficult. This is not why we will bypass the design, construction and use of such a gearbox, but because they generally have had a tough response to the actual gear changes, as it was noticed by the manufacturers car professionals. All of the improvements that have been brought to them could not actually achieve a modern automatic gearbox with a fast transmission response, changing a gear somewhat generally thick.

For this reason, another concept of dual-clutch automatic gearboxes (Fig. 1) has been designed, adapted and introduced in use, which is a symbiosis between the manual and the automatic box, which is extremely well designed so that the operation to make practical use of the classical principle of the manual gearbox with a very fast gear shift with the help of the synchronics already present in the manual gearbox, with the exception that they are no longer manually operated by the driver but automatically by a system computerized vehicle that feels the change in the vehicle's load and responds promptly to it by automatically changing gears by choosing the right one. Another major difference of the dual-clutch system is that two clutches are used in the construction and operation of this gearbox and at the same time two power transmission modules similar to the one in the manual gearbox.

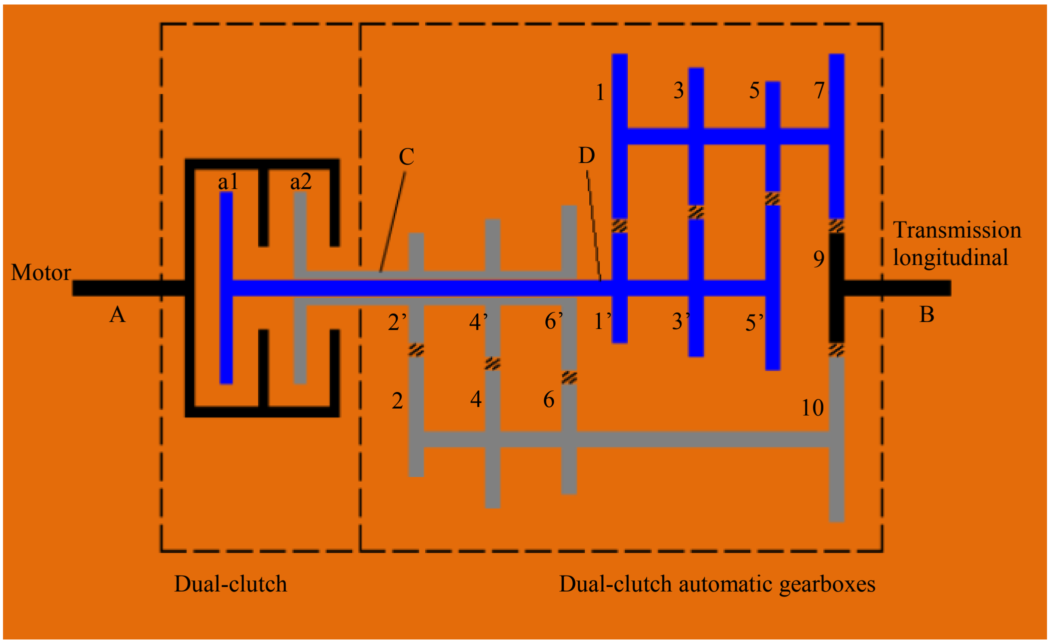

Fig. 1: A dual-clutch automatic gearboxes

The paper presents how to accurately determine the mechanical performance of a gearbox for passenger buses. Based on these relationships, an optimal synthesis of the performance of a dual-clutch automatic gearboxes can be achieved regardless of its operating status. The double clutch gearbox presented in the work can also be used in the aerospace industry.

Hain (1971) proposes a method of optimizing the cam mechanism to obtain an optimal (maximum) transmission angle and a minimum acceleration at the output.

Giordana *et al.* (1979) investigates the influence of measurement errors in the cinematic analysis of the cam.

Antonescu and Petrescu (1985), presented an analytical method for the synthesis of the mechanism with cam and flat thrust plate.

Angeles and Lopez-Cajun (1988) presented optimal synthesis of the cam mechanism and the oscillating flat stick.

Taraza *et al.* (2001) analyzes the influence of the cam profile, on the angular velocity variation of the shaft and on the internal combustion engine power, load, consumption and emissions.

Petrescu and Petrescu (2002b; 2005a; 2005b; 2005c) presented a method of synthesis of the rotary camshaft profile with rotary or rotatable cams, flat or roller, for obtaining high yields at the exit.

An original method of calculating the gearing efficiency is presented and developed successively in the following works: (Petrescu and Petrescu, 2002c, 2003a, 2005f; Petrescu *et al.*, 2006; 2007; 2008a; 2008b, 2009; 2016a, 2017x).

Materials and Methods

From a cinematic point of view, a double clutch gearbox is actually composed of two manual gearboxes arranged in parallel (one with blue and one with gray). Practically in the same case, we have two gearboxes, each with its own clutch, the first box containing the odd steps (1, 3, 5) and the second one containing the stairs (2, 4, 6):

- A - transaxle input shaft (from engine, crankshaft)
- B - transaxle output shaft (to longitudinal or cardan transmission)
- a1 - clutch 1 for odd steps 1, 3 and 5
- a2 - clutch 2, for pitch steps 2, 4 and 6
- D - Input shaft 1 (tubular)
- C - input shaft 2

This configuration has the great advantage of allowing pre-selection of the gears. For example, when the car moves in the 1st gear, the power flow is transmitted from the engine to the wheels via the clutch 1 which is coupled. After a certain speed threshold, stage 2 is selected, but the power is also transmitted through stage 1 and clutch 1, as the clutch 2 remains off. In this phase, we have two selected stages, 1 and 2, with the power transmitted through the clutch 1. When switching to gear 2, clutch 1 opens (disengages) and the clutch 2 closes (engages). Wheels 1, 3, 5, 2, 4, 6 rotate freely on their shafts and transmit power one by one only when coupled by their synchronized actuators (electric motors). The output shaft receives its power permanently through the wheel 9 or from the wheel 7 times from 10. It is interesting that two power streams can be coupled simultaneously, one by the wheel 7 and the other by the wheel 10, without both transmitting the power at the same time, this is virtually dictated by the two clutches a1 and a2 that will never be coupled simultaneously, but only one of them; if none is engaged then the gearbox no longer transmits anything being set to zero. The real advantage of the system is that two power streams can be prepared simultaneously in two different stages, but coupling is done only for one of them; this ensures a rapid shift of gears, the box being not just faster than a classic automatic but even faster than manual ones.

Due to the possibility to preselect the gear to be used, the gearing time can be reduced to 0.2 sec (two tenths of a second), without causing shock and vibration in the transmission.

This mode of operation of the double clutch gearbox has the following advantages.

Compared to a manual gearbox:

- Very fast gear changes due to preselection
- The gear change is done without interruption of the power flow without shock, noise, vibration, kick-in (the driver must no longer have the experience, changing the steps automatically); smooth and shock-free change of gears, thanks in particular to controlled clutch slippage

Compared to a classic automated box with hydrotransformer:

- faster gear shift due to preselection;
- better mechanical performance due to lack of hydrotransformer and complex planetary mechanisms

The gear units of a double clutch box are similar to those of a manual gearbox. Coupling of gears is also done by synchronization, the only difference being that in a double clutch box the coupling of the steps is done with electro-hydraulic or electric actuators controlled by a control computer and not directly by the driver.

Depending on the type of clutch and the mode of operation, the double clutch boxes are classified into:

1. box multi-disc clutches, wet, hydraulically operated
2. box, single-clutch, dry, electrically operated clutches

The double wet hydraulic clutch (Fig. 2) is used when the maximum torque exceeds 250 Nm. Besides the advantage of the larger transmitted torque, a wet multidisc clutch dissipates the heat more easily, which is why it does not wear off and the coupling and uncoupling is smoother and more shock free.

In automobiles where the propellant develops below 200-250 Nm, the use of dipped, monodisc clutches with electric drive has the advantage of lower fuel consumption due to the lack of oil pump, hydraulic system and lower friction losses (Fig. 3).

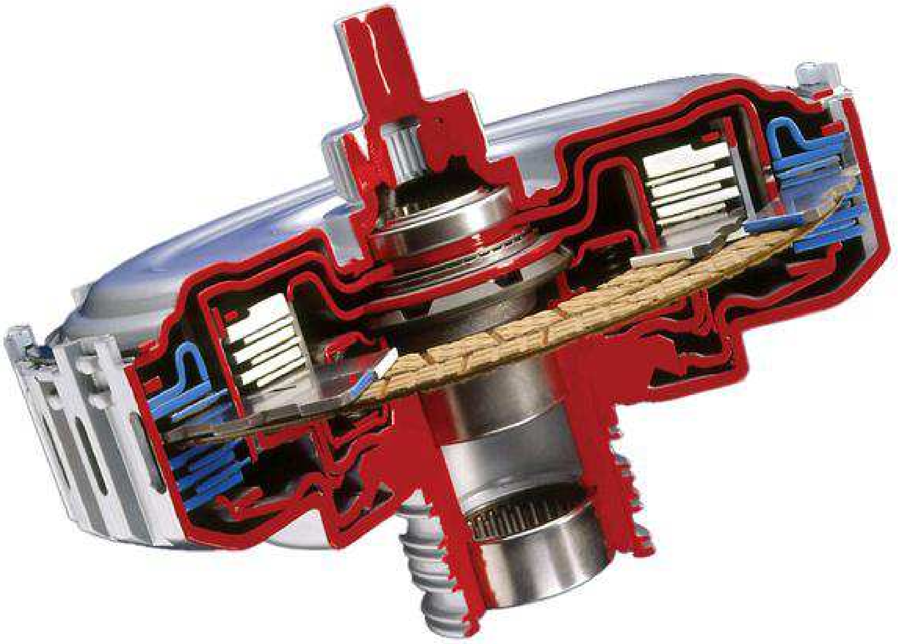

Fig. 2: Double, "DualTronic" clutch, humid, hydraulically operated

Fig. 3: Double clutch dry

Coupling and uncoupling to a dry clutch is less smooth but direct, mechanical and low in fuel consumption.

Option 2 for small cars may be indicated for use, but not for trucks and buses, where version 1 is available with a wet DualTronic double hydraulic clutch, hydraulically powered, because the carmakers on these types of cars develop very high powers and couples elevated, with values exceeding frequently 250 Nm.

The decision to equip the car with a wet or dry clutch double cluster is taken into consideration for several reasons. Table 1 below shows the characteristics of the two clutch solutions.

The double clutch gearbox can be used for any powertrain architecture: front, rear, or full throttle. Also this type of gearbox can transmit an engine torque of up to 1250 Nm.

Due to its major advantages, the dual-clutch automatic gearbox penetrated rapidly, replacing the classic automatic gearbox, but also the manual gearbox.

On buses and trolleybuses, this gearshift system is now the most widely used optimal option.

All double clutch gearboxes are electronically controlled via a mini computer. Clutch drive and gear clutch are hydraulically operated via electro-hydraulic or electric valves using DC motors.

From the point of view of the interaction with the driver there is no difference between a double clutch box and a classic automatic gearbox. Cars are equipped with program selectors (P, R, N, D) and the gearbox can operate both automatically and manually ("sequentially").

Important Note: So, as with automatic gearboxes, the driver can switch the system over to the automatic pilot, or he can assign the right and obligation to drive the vehicle manually by manually changing the steps by manual (sequential) control. This is simple, from a program selector.

Table 1: Comparative characteristics of the two clutch solutions

Criterion	Double clutch dry (mechanical)	Double clutch wet (hydraulic)
Controllable coupling/decoupling	-	+
Wear	-	+
Heat dissipation	-	+
Low temperature behaviors	+	-
Dimensions	-	+
Mechanical inertia	-	+
Coupling transmission capacity	-	+
Mass	+	-
Comfort to the speed gear change	-	+
Fuel consumption	+	-
The total cost	+	-

Table 2: Name of the double clutch gearbox, according to the manufacturer

Car manufacturer (transmission)	The name of the technology DCT
Volkswagen (Borg Warner)	Direct Shift Gearbox (DSG)
Porsche (ZF)	Porsche Doppel Kupplunggetriebe (PDK)
Ford (Getrag)	PowerShift
Fiat (FPT)	Dual Dry Clutch Transmission (DDCT)
Mercedes-Benz (Getrag)	SpeedShift
Mitsubishi (Getrag)	Twin Clutch Sport Shift Transmission (TC-SST)
Renault (Getrag)	Efficient Dual Clutch (EDC)

Depending on the transmission or car manufacturer, double clutch boxes are named differently (Table 2), even if the operating principle is the same (similar).

Due to the advantages of this gearbox, it is becoming an option for most car manufacturers and probably in ten years if something new is not coming, it will occupy a big market share.

Even though today, both in the European and the global market, the share of dual clutch gearboxes is steadily increasing, it is right to present the situation as well (Fig. 4).

A CVT is a combination of manual and classic automated planetary gearboxes, plus other mechanisms with rigid memory (cam-tap, cross of Malta or Geneva drive), but also other elements (two driven and other driven pulleys driven by a metal strap, varying the transmission ratio by means of an electro-hydraulic control module that controls the pressure in the cylinders of the two pulleys) and other systems, a CVT becoming constructively complex, hybrid variant with low market opportunities. Initially, its share was higher, reaching just 3.5% today.

The Double Clutch (DCT), although relatively new, has already managed to surpass a CVT with a current 4% share.

An AMT, that is, an automated, robotized gearbox, although it currently has a market share of 7% due to the high manufacturing costs and the high technology (difficult to access) technology and the complex system, is not increasing and even it is likely to start falling with the appearance of DCTs.

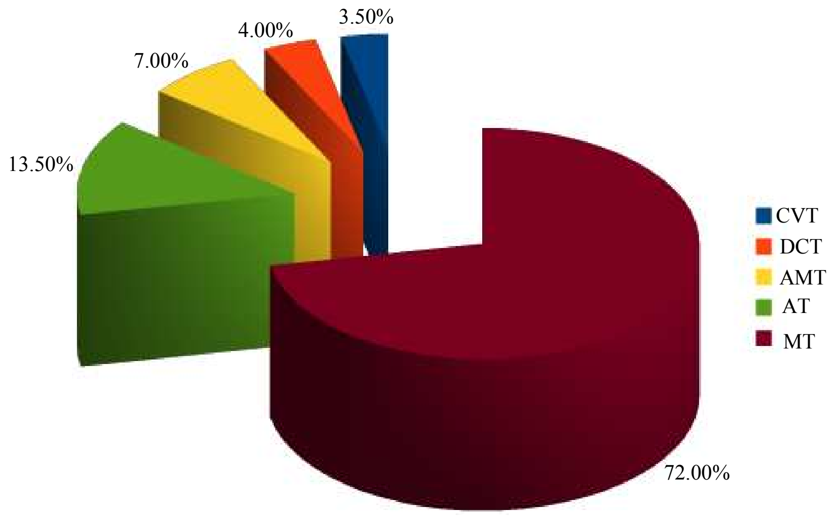

Fig. 4: Speed gearbox quotations up to date; CVT - Continuous variation box; DCT-dual clutch box; AMT - automated box (robotic) AT - classic automated hydro transformer box; MT - manual gearbox; AT - classic automated hydro transformer box; MT - manual gearbox

The classic automatic AT gearbox with hydrotransformer and planetary gears has a large market share of 13.5% due to the fact that it represents the oldest change (the first gearbox other than the manual gearbox), but not it is known how long it will be able to keep this quotation on the market because the new DCT chia has just emerged, is fast growing and eats permanently not only from the hand box but also from ATs.

Even though MTs, the oldest gearboxes (obviously the synchronous models, not those with wheel-ridden wheels that have faded away) still retain a 72% market relaxation rate, they are still in a permanent decline and the next 20-30 years will suffer a dramatic drop, losing priority in the auto market.

Results

The power transmission in Stage I of a dual clutch automatic gearbox can be seen in Fig. 5: a_1-$z_{1'}$-z_1-z_7-z_9.

For the dual clutch automatic gearbox (Fig. 1), the system relations (1; Fig. 5) will be used to determine the first gear:

$$\begin{cases} \varepsilon_{1'1}^{a.e.} = \dfrac{1+tg^2\beta}{2.\pi} \cdot \left\{ \sqrt{\left[\left(z_{1'} + 2\cdot\cos\beta\right).tg\alpha_0\right]^2 + 4\cdot\cos^3\beta\cdot\left(z_{1'}+\cos\beta\right)} \right. \\ \left. + \sqrt{\left[\left(z_1 + 2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2 + 4\cdot\cos^3\beta\cdot\left(z_1+\cos\beta\right)} - \left(z_{1'}+z_1\right)\cdot tg\alpha_0 \right\} \\ \eta_{1'1} = \dfrac{z_{1'}^2\cdot\cos^2\beta}{z_{1'}^2\left(tg^2a_0+\cos^2\beta\right)+\dfrac{2}{3}\pi^2\cos^4\beta\left(\varepsilon-1\right)\left(2\varepsilon-1\right)\pm 2\pi tga_0 z_{1'}\cos^2\beta\left(\varepsilon-1\right)} \\ \varepsilon_{1'1}^{a.e.} = \dfrac{1+tg^2\beta}{2\cdot\pi} \cdot \left\{ \sqrt{\left[\left(z_7 + 2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2 + 4\cdot\cos^3\beta\cdot\left(z_7+\beta\right)} \right. \\ \left. + \sqrt{\left[\left(z_9 + 2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2 + 4\cdot\cos^3\beta\cdot\left(z_9+\cos\beta\right)} - \left(z_7+z_9\right)\cdot tg\alpha_0 \right\} \\ \eta_{79} = \dfrac{z_7^2\cdot\cos^2\beta}{z_7^2\left(tg^2a_0+\cos^2\beta\right)+\dfrac{2}{3}\pi^2\cos^4\beta\left(\varepsilon-1\right)\left(2\varepsilon-1\right)\pm 2\pi tga_0 z_7\cos^2\beta\left(\varepsilon-1\right)} \\ i_I = i_{1'9} = i_{1'1}\cdot i_{79} = -\dfrac{z_1}{z_{1'}}\cdot(-)\dfrac{z_9}{z_7} = \dfrac{z_1\cdot z_9}{z_{1'}\cdot z_7} \\ \eta_I \equiv \eta_{1'9} = \eta_{1'1}\cdot\eta_{79} \end{cases} \tag{1}$$

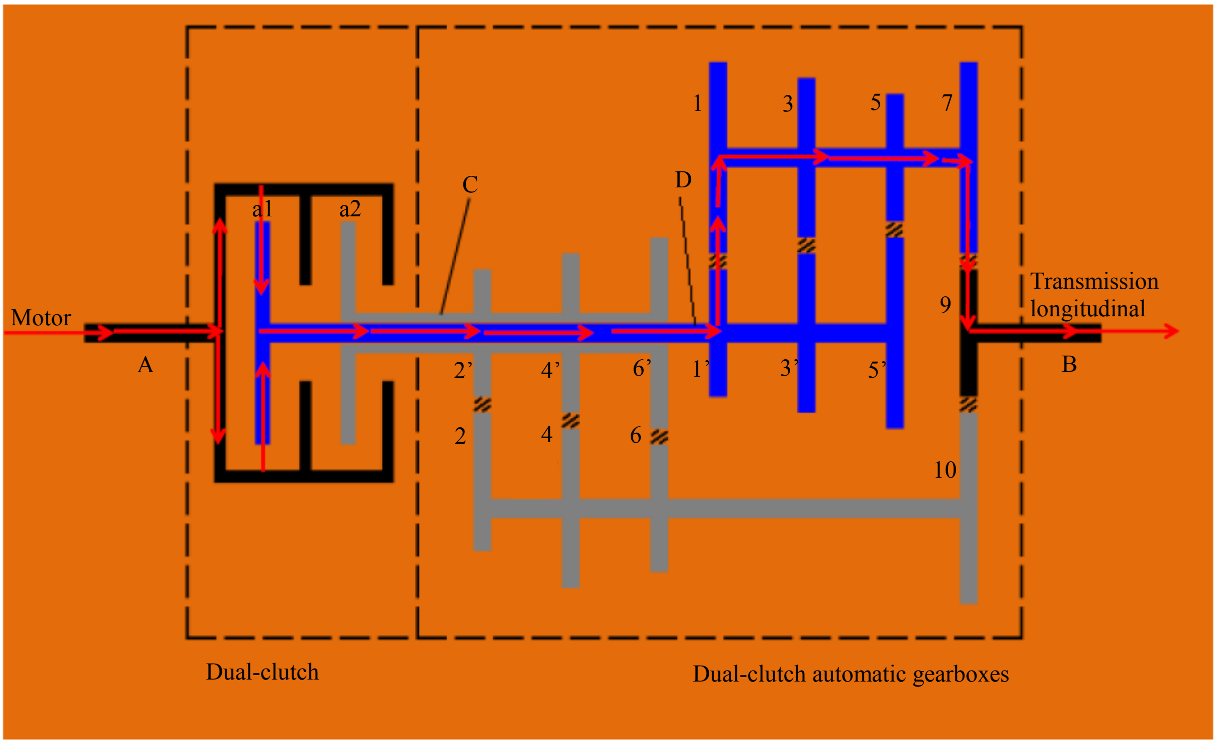

Fig. 5: The power transmission in stage I of a dual clutch automatic gearbox

The power transmission in Stage II of a dual clutch automatic gearbox can be seen in Fig. 6: a_2-$z_{2'}$-z_2-z_{10}-z_9. For the dual clutch automatic gearbox, system relationships (2; Fig. 6) will be used to determine the mechanical efficiency of the second gear:

$$\begin{cases} \varepsilon_{2'2}^{a.e.} = \dfrac{1+tg^2\beta}{2\cdot\pi}\cdot\Big\{\sqrt{\left[2\left(z_{2'}+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_{2'}+\cos\beta\right)} \\ +\sqrt{\left[\left(z_2+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_2+\cos\beta\right)}-\left(z_{2'}+z_2\right)\cdot tg\alpha_0\Big\} \\ \eta_{2'2} = \dfrac{z_{2'}^2\cdot\cos^2\beta}{z_{2'}^2\left(tg^2a_0+\cos^2\beta\right)+\dfrac{2}{3}\pi^2\cos^4\beta\left(\varepsilon-1\right)\left(2\varepsilon-1\right)\pm 2\pi tga_0 z_{2'}\cos^2\beta\left(\varepsilon-1\right)} \\ \varepsilon_{10,9}^{a.e.} = \dfrac{1+tg^2\beta}{2\cdot\pi}\cdot\Big\{\sqrt{\left[\left(z_{10}+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_{10}+\beta\right)} \\ +\sqrt{\left[\left(z_9+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_9+\cos\beta\right)}-\left(z_{10}+z_9\right)\cdot tg\alpha_0\Big\} \\ \eta_{10,9} = \dfrac{z_{10}^2\cdot\cos^2\beta}{z_{10}^2\left(tg^2a_0+\cos^2\beta\right)+\dfrac{2}{3}\pi^2\cos^4\beta\left(\varepsilon-1\right)\left(2\varepsilon-1\right)\pm 2\pi tga_0 z_{10}\cos^2\beta\left(\varepsilon-1\right)} \\ i_{II} = i_{2'9} = i_{2'2}.i_{10,9} = -\dfrac{z_2}{z_{2'}}\cdot(-)\dfrac{z_9}{z_{10}} = \dfrac{z_2\cdot z_9}{z_{2'}\cdot z_{10}} \\ \eta_{II} \equiv \eta_{2'9} = \eta_{2'2}\cdot\eta_{10,9} \end{cases} \tag{2}$$

The power transmission in stage III of a dual clutch automatic gearbox can be seen in Fig. 7: a_1-$z_{3'}$-z_3-z_7-z_9. The system relationships (3) will be used to determine the third step mechanical efficiency:

$$\begin{cases} \varepsilon_{3'3}^{a.e.} = \dfrac{1+tg^2\beta}{2\cdot\pi}\cdot\left\{\sqrt{\left[\left(z_{3'}+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_{3'}+\cos\beta\right)}\right. \\ \left.+\sqrt{\left[\left(z_3+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_3+\cos\beta\right)}-\left(z_{3'}+z_3\right)\cdot tg\alpha_0\right\} \\ \eta_{3'3} = \dfrac{z_{3'}^2\cdot\cos^2\beta}{z_{3'}^2\left(tg^2a_0+\cos^2\beta\right)+\dfrac{2}{3}\pi^2\cos^4\beta\left(\varepsilon-1\right)\left(2\varepsilon-1\right)\pm 2\pi tga_0 z_{3'}\cos^2\beta\left(\varepsilon-1\right)} \\ \varepsilon_{79}^{a.e.} = \dfrac{1+tg^2\beta}{2\cdot\pi}\cdot\left\{\sqrt{\left[\left(z_7+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_7+\cos\beta\right)}\right. \\ \left.+\sqrt{\left[\left(z_9+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_9+\cos\beta\right)}-\left(z_7+z_9\right)\cdot tg\alpha_0\right\} \\ \eta_{79} = \dfrac{z_7^2\cdot\cos^2\beta}{z_7^2\left(tg^2a_0+\cos^2\beta\right)+\dfrac{2}{3}\pi^2\cos^4\beta\left(\varepsilon-1\right)\left(2\varepsilon-1\right)\pm 2\pi tga_0 z_7\cos^2\beta\left(\varepsilon-1\right)} \\ i_{III} = i_{3'9} = i_{3'3}\cdot i_{79} = -\dfrac{z_3}{z_{3'}}\cdot(-)\dfrac{z_9}{z_7} = \dfrac{z_3\cdot z_9}{z_{3'}\cdot z_7} \\ \eta_{III} \equiv \eta_{3'9} = \eta_{3'3}\cdot\eta_{79} \end{cases} \tag{3}$$

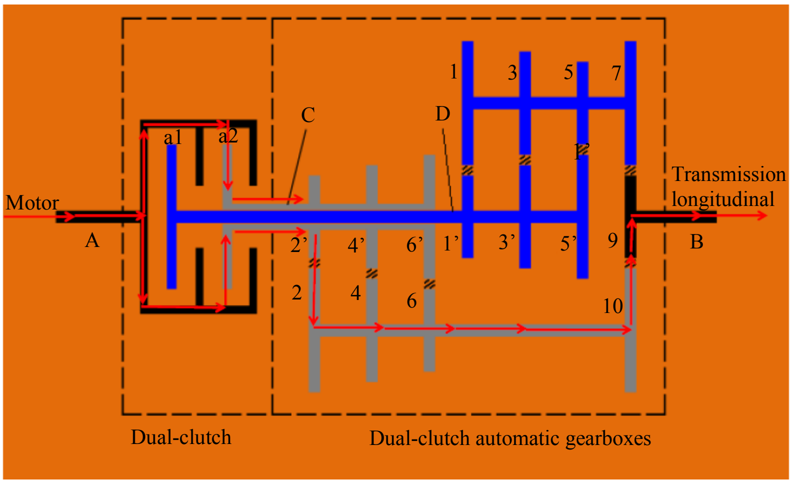

Fig. 6: The power transmission in stage II of a dual clutch automatic gearbox

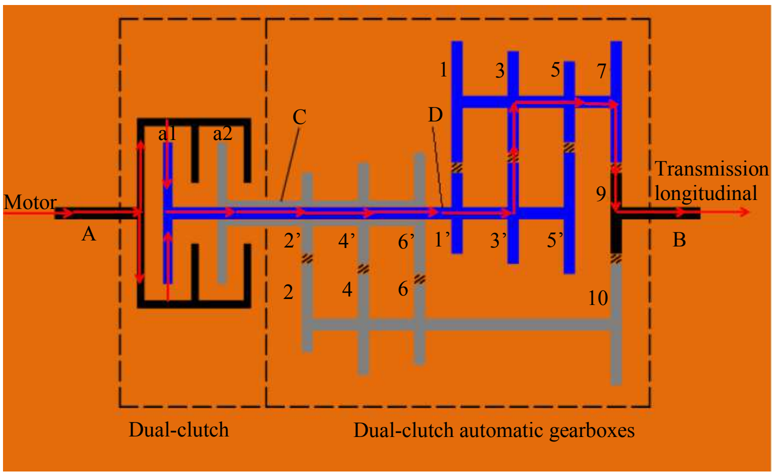

Fig. 7: The power transmission in stage III of a dual clutch automatic gearbox

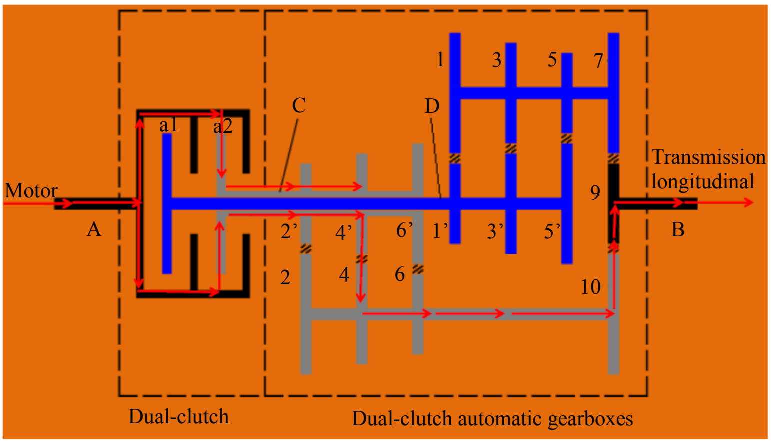

Fig. 8: The power transmission in stage IV of a dual clutch automatic gearbox

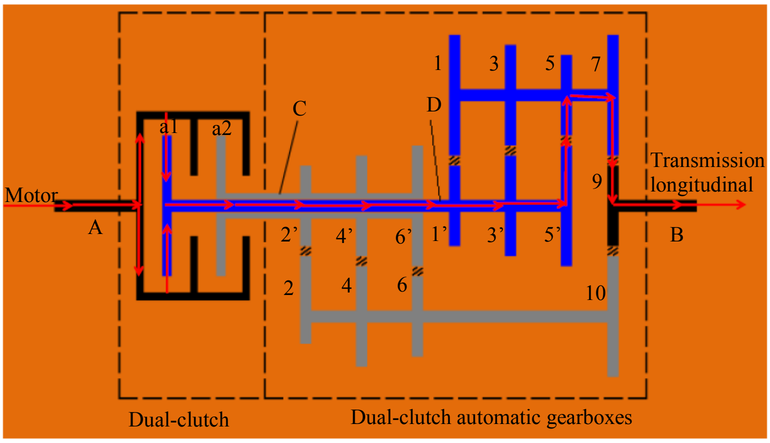

Fig. 9: The power transmission in stage V of a dual clutch automatic gearbox

The power transmission in stage IV of a dual clutch automatic gearbox can be seen in Fig. 8: a_2-$z_{4'}$-4-z_{10}-z_9. The system relationships (4) will be used in determining the fourth step mechanical efficiency:

$$\begin{cases} \varepsilon_{4'4}^{a.e.} = \dfrac{1+tg^2\beta}{2\cdot\pi}\cdot\left\{\sqrt{\left[2\left(z_{4'}+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_{4'}+\cos\beta\right)}\right. \\ \left.+\sqrt{\left[\left(z_4+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_4+\cos\beta\right)}-\left(z_{4'}+z_4\right)\cdot tg\alpha_0\right\} \\ \eta_{4'4} = \dfrac{z_{4'}^2\cdot\cos^2\beta}{z_{4'}^2\left(tg^2a_0+\cos^2\beta\right)+\dfrac{2}{3}\pi^2\cos^4\beta\left(\varepsilon-1\right)\left(2\varepsilon-1\right)\pm 2\pi tga_0 z_{4'}\cos^2\beta\left(\varepsilon-1\right)} \\ \varepsilon_{10,9}^{a.e.} = \dfrac{1+tg^2\beta}{2\cdot\pi}\cdot\left\{\sqrt{\left[\left(z_{10}+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_{10}+\cos\beta\right)}\right. \\ \left.+\sqrt{\left[\left(z_9+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_9+\cos\beta\right)}-\left(z_{10}+z_9\right)\cdot tg\alpha_0\right\} \\ \eta_{10,9} = \dfrac{z_{10}^2\cdot\cos^2\beta}{z_{10}^2\left(tg^2a_0+\cos^2\beta\right)+\dfrac{2}{3}\pi^2\cos^4\beta\left(\varepsilon-1\right)\left(2\varepsilon-1\right)\pm 2\pi tga_0 z_{10}\cos^2\beta\left(\varepsilon-1\right)} \\ i_{IV} = i_{4'9} = i_{4'4}\cdot i_{10,9} = -\dfrac{z_4}{z_{4'}}\cdot\left(-\right)\dfrac{z_9}{z_{10}} = \dfrac{z_4\cdot z_9}{z_{4'}\cdot z_{10}} \\ \eta_{IV} \equiv \eta_{4'9} = \eta_{4'4}\cdot\eta_{10,9} \end{cases} \quad (4)$$

The power transmission in step V of a dual clutch automatic gearbox can be seen in Fig. 9: a_1-$z_{5'}$-z_5-z_7-z_9. The system relationships (5) will be used to determine the fifth-gear mechanical efficiency:

$$\begin{cases}\varepsilon_{5'5}^{a.e.}=\dfrac{1+tg^2\beta}{2\cdot\pi}\cdot\Big\{\sqrt{\left[\left(z_{5'}+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_{5'}+\cos\beta\right)} \\ +\sqrt{\left[\left(z_5+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_5+\cos\beta\right)}-\left(z_{5'}+z_5\right)\cdot tg\alpha_0\Big\} \\ \eta_{5'5}=\dfrac{z_{5'}^2\cdot\cos^2\beta}{z_{5'}^2\left(tg^2a_0+\cos^2\beta\right)+\dfrac{2}{3}\pi^2\cos^4\beta(\varepsilon-1)(2\varepsilon-1)\pm2\pi tga_0z_{5'}\cos^2\beta(\varepsilon-1)} \\ \varepsilon_{79}^{a.e.}=\dfrac{1+tg^2\beta}{2\cdot\pi}\cdot\Big\{\sqrt{\left[\left(z_7+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_7+\cos\beta\right)} \\ +\sqrt{\left[\left(z_9+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_9+\cos\beta\right)}-\left(z_7+z_9\right)\cdot tg\alpha_0\Big\} \\ \eta_{79}=\dfrac{z_7^2\cdot\cos^2\beta}{z_7^2\left(tg^2a_0+\cos^2\beta\right)+\dfrac{2}{3}\pi^2\cos^4\beta(\varepsilon-1)(2\varepsilon-1)\pm2\pi tga_0z_7\cos^2\beta(\varepsilon-1)} \\ i_V=i_{5'9}=i_{5'5}\cdot i_{79}=-\dfrac{z_5}{z_{5'}}\cdot(-)\dfrac{z_9}{z_7}=\dfrac{z_5\cdot z_9}{z_{5'}\cdot z_7} \\ \eta_V\equiv\eta_{5'9}=\eta_{5'5}\cdot\eta_{79}\end{cases}\tag{5}$$

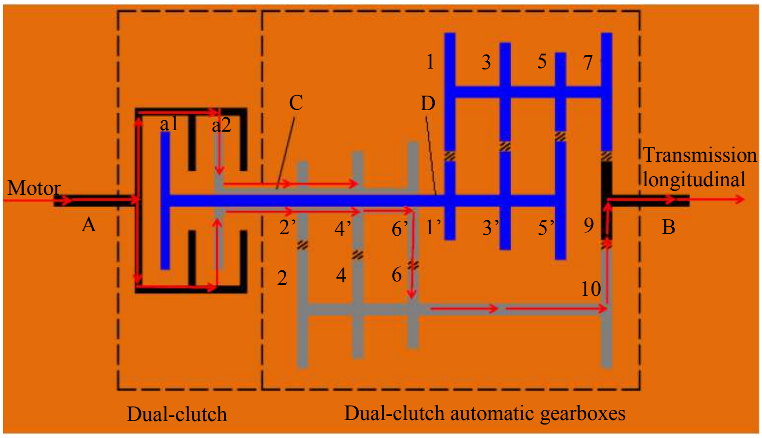

Fig. 10: The power transmission in stage VI of a dual clutch automatic gearbox

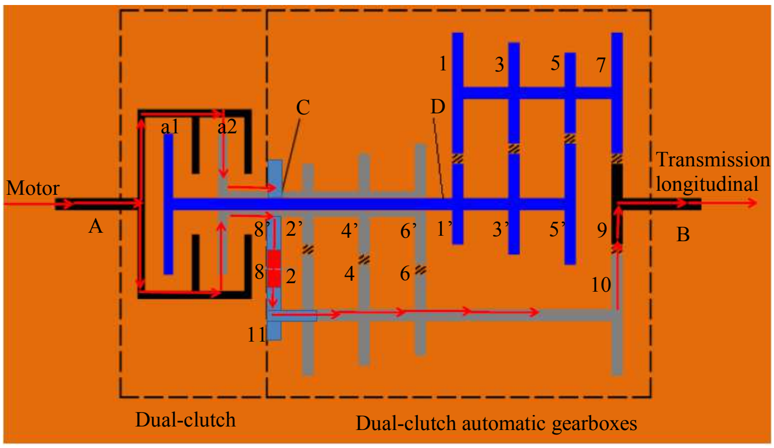

Fig. 11: The power transmission in stage VII (M.R.) of a dual clutch automatic gearbox

The power transmission in stage VI of a dual clutch automatic gearbox can be seen in Fig. 10: a_2-$z_{6'}$-z_6-z_{10}-z_9. The system relationships (6) will be used to determine the sixth step mechanical efficiency:

$$\begin{cases} \varepsilon_{6'6}^{a.e.} = \dfrac{1+tg^2\beta}{2\cdot\pi}\cdot\left\{\sqrt{\left[2\left(z_{6'}+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_{6'}+\cos\beta\right)}\right. \\ \left.+\sqrt{\left[\left(z_6+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_6+\cos\beta\right)}-\left(z_{6'}+z_6\right)\cdot tg\alpha_0\right\} \\ \eta_{6'6} = \dfrac{z_{6'}^2\cdot\cos^2\beta}{z_{6'}^2\left(tg^2a_0+\cos^2\beta\right)+\dfrac{2}{3}\pi^2\cos^4\beta\left(\varepsilon-1\right)\left(2\varepsilon-1\right)\pm 2\pi tga_0 z_{6'}\cos^2\beta\left(\varepsilon-1\right)} \\ \varepsilon_{10,9}^{a.e.} = \dfrac{1+tg^2\beta}{2\cdot\pi}\cdot\left\{\sqrt{\left[\left(z_{10}+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_{10}+\cos\beta\right)}\right. \\ \left.+\sqrt{\left[\left(z_9+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_9+\cos\beta\right)}-\left(z_{10}+z_9\right)\cdot tg\alpha_0\right\} \\ \eta_{10,9} = \dfrac{z_{10}^2\cdot\cos^2\beta}{z_{10}^2\left(tg^2a_0+\cos^2\beta\right)+\dfrac{2}{3}\pi^2\cos^4\beta\left(\varepsilon-1\right)\left(2\varepsilon-1\right)\pm 2\pi tga_0 z_{10}\cos^2\beta\left(\varepsilon-1\right)} \\ i_{VI} = i_{6'9} = i_{6'6}\cdot i_{10,9} = -\dfrac{z_6}{z_{6'}}\cdot\left(-\right)\dfrac{z_9}{z_{10}} = \dfrac{z_6\cdot z_9}{z_{6'}\cdot z_{10}} \\ \eta_{VI} \equiv \eta_{6'9} = \eta_{6'6}\cdot\eta_{10,9} \end{cases} \tag{6}$$

To achieve a reverse gear, a forward gear (such as the one shown in Fig. 1 in which the forward gear step 2 has been converted to the reverse gear) is introduced, introducing an additional sprocket 8 reverses the direction of rotation at the exit of the box compared to the input), or more normally, another stage is built, for example on the first flow that can reverse the direction of rotation (Fig. 11).

The reverse-flow power transmission in the reverse gear of a dual-clutch automatic gearbox can be seen in Fig. 11: a_2-$z_{8'}$-z_8-z_{11}-z_{10}-z_9.

When determining the mechanical efficiency of the seventh step, system relationships (7) will be used:

$$\begin{cases} \varepsilon_{8'8}^{a.e.} = \dfrac{1+tg^2\beta}{2\cdot\pi}\cdot\left\{\sqrt{\left[\left(z_{8'}+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_{8'}+\cos\beta\right)}\right. \\ \left.+\sqrt{\left[\left(z_8+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_8+\cos\beta\right)}-\left(z_{8'}+z_8\right)\cdot tg\alpha_0\right\} \\ \eta_{8'8} = \dfrac{z_{8'}^2\cdot\cos^2\beta}{z_{8'}^2\left(tg^2a_0+\cos^2\beta\right)+\dfrac{2}{3}\pi^2\cos^4\beta\left(\varepsilon-1\right)\left(2\varepsilon-1\right)\pm 2\pi tga_0 z_{8'}\cos^2\beta\left(\varepsilon-1\right)} \\ \varepsilon_{8,11}^{a.e.} = \dfrac{1+tg^2\beta}{2\cdot\pi}\cdot\left\{\sqrt{\left[\left(z_8+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot 2\left(z_8+\cos\beta\right)}\right. \\ \left.+\sqrt{\left[\left(z_{11}+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_{11}+\cos\beta\right)}-\left(z_8+z_{11}\right)\cdot tg\alpha_0\right\} \\ \eta_{8,11} = \dfrac{z_8^2\cdot\cos^2\beta}{z_8^2\left(tg^2a_0+\cos^2\beta\right)+\dfrac{2}{3}\pi^2\cos^4\beta\left(\varepsilon-1\right)\left(2\varepsilon-1\right)\pm 2\pi tga_0 z_8\cos^2\beta\left(\varepsilon-1\right)} \\ \varepsilon_{10,9}^{a.e.} = \dfrac{1+tg^2\beta}{2\cdot\pi}\cdot\left\{\sqrt{\left[\left(z_{10}+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot 25\left(z_{10}+\cos\beta\right)}\right. \\ \left.+\sqrt{\left[\left(z_9+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_9+\cos\beta\right)}-\left(z_{10}+z_9\right)\cdot tg\alpha_0\right\} \\ \eta_{10,9} = \dfrac{z_{10}^2\cdot\cos^2\beta}{z_{10}^2\left(tg^2a_0+\cos^2\beta\right)+\dfrac{2}{3}\pi^2\cos^4\beta\left(\varepsilon-1\right)\left(2\varepsilon-1\right)\pm 2\pi tga_0 z_{10}\cos^2\beta\left(\varepsilon-1\right)} \\ i_{VII} = i_{MR} = i_{8'9} = i_{8'8}\cdot i_{8,11}\cdot i_{10,9} = -\dfrac{z_8}{z_{8'}}\cdot\left(-\right)\dfrac{z_{11}}{z_8}\cdot\left(-\right)\dfrac{z_9}{z_{10}} = -\dfrac{z_9\cdot z_{11}}{z_{8'}\cdot z_{10}} \\ \eta_{VII} \equiv \eta_{MR} \equiv \eta_{8'9} = \eta_{8'8}\cdot\eta_{8,11}\cdot\eta_{10,9} \end{cases} \tag{7}$$

Discussion

The computational relationships presented can successfully synthesize a modern double-clutch gearshift because the calculations are very accurate. This method s*et al*l tooth pairs in simultaneous engagement, which is why the theory is not just precise but also adapted to reality, describing successfully the operation of a gear with inclined teeth, whether cylindrical or tapered.

As one mentioned before, although this interesting gearbox has been introduced relatively recently to cars, it can be successfully designed in aviation, replacing the classic automated box (Frăţilă *et al*., 2011; Pelecudi, 1967; Antonescu, 2000; Comănescu *et al*., 2010; Aversa *et al*., 2016a; 2016b; 2016c; 2016d; 2017a; 2017b; 2017c; 2017d; 2017e; Mirsayar *et al*., 2017; Cao *et al*., 2013; Dong *et al*., 2013; De Melo *et al*., 2012; Garcia *et al*., 2007; Garcia-Murillo *et al*., 2013; He *et al*., 2013; Lee, 2013; Lin *et al*., 2013; Liu *et al*., 2013; Padula and Perdereau, 2013; Perumaal and Jawahar, 2013; Petrescu and Petrescu, 1995a; 1995b; 1997a; 1997b; 1997c; 2000a; 2000b; 2002a; 2002b; 2003; 2005a; 2005b; 2005c; 2005d; 2005e, 2016a; 2016b; 2016c; 2016d; 2016e; 2013; 2012a; 2012b; 2011; Petrescu *et al*., 2009; 2016a; 2016b; 2016c; 2016d; 2016e; 2017a; 2017b; 2017c; 2017d; 2017e; 2017f; 2017g; 2017h; 2017i; 2017j; 2017k; 2017l; 2017m; 2017n; 2017o; 2017p; 2017q; 2017r; 2017s; 2017t; 2017u; 2017v; 2017w; 2017x; 2017y; 2017z; 2017aa; 2017ab; 2017ac; 2017ad; 2017ae; Petrescu and Calautit, 2016a; 2016b; Reddy *et al*., 2012; Tabaković *et al*., 2013; Tang *et al*., 2013; Tong *et al*., 2013; Wang *et al*., 2013; Wen *et al*., 2012; Antonescu and Petrescu, 1985; 1989; Antonescu *et al*., 1985a; 1985b; 1986; 1987; 1988; 1994; 1997; 2000a; 2000b; 2001; List the first flights, From Wikipedia; Chen and Patton, 1999; Fernandez *et al*., 2005; Fonod *et al*., 2015; Lu *et al*., 2015-2016; Murray *et al*., 2010; Palumbo *et al*., 2012; Patre and Joshi, 2011; Sevil and Dogan, 2015; Sun and Joshi, 2009; Crickmore, 1997; Donald, 2003; Goodall, 2003; Graham, 2002; Jenkins, 2001; Landis and Dennis, 2005; Clément, Wikipedia; Cayley, Wikipedia; Coandă, Wikipedia; Gunston, 2010; Laming, 2000; Norris, 2010; Goddard, 1916; Kaufman, 1959; Oberth, 1955; Cataldo, 2006; Gruener, 2006; Sherson *et al*., 2006; Williams, 1995; Venkataraman, 1992; Oppenheimer and Volkoff, 1939; Michell, 1784; Droste, 1915; Finkelstein, 1958; Gorder, 2015; Hewish, 1970).

Conclusion

Although most of the gearboxes in operation are all classic, manual, for about 40 years have entered serial production and automatic gearboxes of various types.

The most successful were those with a continuous variable transmission, those with planetary mechanisms and more recently those with double clutch.

Automatic boxes have had a lot of trouble at the hydraulic installations that were used for both shareholders and automation and command. Today the command was taken over by electronics, chips and automation are also modern. Often, even action is no longer hydraulic but electric by actuators (electric motors step by step).

The planetary mechanisms are also modern, but even in that conditions the automatic boxes still have problems with the still high response time compared to the manual cans.

This is why relatively recent have occurred dual-clutch automated boxes, which promise a lot in the field, can be used successfully in cars and trains as well as in the aerospace industry.

This paper wants not only to present such a gearbox but also to realize its dynamic synthesis, based on the engineering optimization of the mechanical efficiency achieved in each used speed gear. The presented method is original.

Whether to be used in aerospace or road vehicles, the gearbox presented in the work has many advantages. It was built from two manual gearboxes operated with a double clutch.

The big advantage is symmetry and good balance. For this reason, it is not right to sacrifice the gear II gear to introduce an extra pinion on its circuit to achieve the reverse gear. The correct variant is the one chosen at the end with a separate step for the reverse, a seventh gear. In this way, the balance of the system is maintained and in addition for all the steps before it is possible to prepare them in advance, that is, two speed gears can be coupled at the same time, one being the one in operation and the other the one in which we want to switch system, thus achieving a preselection of the necessary gear, with a previous training, which allows the system a very fast coupling with extremely low response time, making this box a very modern, fast and special one.

For this reason, the coupling will be quiet, no noise or vibration, no risk of rupture, unmatched advantages to other gearbox models.

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Author's Contribution

All the authors contributed equally to prepare, develop and carry out this manuscript.

Ethics

This article is original and contains unpublished material. Authors declare that are not ethical issues and no conflict of interest that may arise after the publication of this manuscript.

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