

Kinematic and Dynamic Analysis of a Classic, Three-Axis Manual Gearbox, Without a Direct Socket

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Abstract: Fixed axis mechanical transmissions today have the largest spread on the entire planet, being practically used in almost all areas. From the gearboxes of the vehicles to the stationary reducers, used in the electrical, electronic and electro technical equipment, in the heavy industry but also in the light industry, in energy and in transport, practically the transmissions with fixed axes meet today everywhere, being part of our daily life. For this reason, we want to present the kinematic and dynamic analysis at a classic, manual, three-axis gearbox, with six speeds, without a direct socket. With the help of the presented method, one can analyze the dynamics of such a gearbox in order to design it optimally, so as to achieve the highest yields in operation, especially in the most used speeds, so that the transmission losses are limited to up.

Keywords: Automation, Mechatronic Systems, Mechanical Transmissions, Gearboxes, Gearing, Manual Gearbox, Three-axis Gearbox, Six Speeds Gearbox, Dynamics

Introduction

The beginning of the use of bar and gear mechanisms must be sought in ancient Egypt at least a thousand years before Christ. Here, for the first time, the "spiked" wheel transmissions were used for irrigation of crops as well as the snail gears for cotton processing (Fig. 1).

Such gears have been built and used since ancient times, initially for lifting heavy ship anchors as well as for pre-tensioning catapults used on battlefields. They were then introduced to machines with wind and water (as reducers or multipliers to pumps, windmills, or water).

230 years BC, in the city of Alexandria in Egypt, the multi-lever wheel and rack gear were used.

The transmission of the gear by means of gears with gears has seen substantial progress since 1364 AD when the Italian craftsman Giovanni da Dondi made an astronomical clock, whose composition included interior gears and elliptical gears.

The first gear adjustable gears were used in 1769 by Cugnot to equip the first vehicle powered by a steam engine (Fig. 2).

The first engineer (scientist), who actually designs such transmissions, is considered to be the Italian master Leonardo da Vinci (15th century) (Fig. 3).

The Benz engine (Fig. 4) had gears with gears but also with chain gears (patented after 1882). On the Fig. 5, you can see the sketch of the first gearbox with gears (gears) and chain gears made in 1870 by the British Starley and Hillman.

After 1912, in Cleveland (USA), specialized manufacturing of wheels, gears and gears (cylindrical, screw, conical, with straight, inclined or curved teeth) begins to be produced. Gearboxes with fixed axles have the widest spread on all types of vehicles (Fig. 6).

Gearboxes are independent gearboxes with permanent gear, mounted on shafts and enclosed in a watertight housing. They serve:

- Speed reduction
- Increase of the transmitted moment
- Modification of the direction of rotation or the plane of movement
- Sum the power flow from several motors to a work machine
- Distributes the power flow from one motor to several work machines.

In the case of speed reducers, the gear wheels are fixedly fixed to the shafts, permanently engage and realize a totally fixed transmission ratio, defined as the ratio between the speed at the input and the speed at the output of the gearbox, as opposed to the gearboxes at which some wheels are movable. On shafts (balancing wheels), intermittently gears and performs a total gear ratio. They are also different from the gear speed variants (used less frequently), where the total transmission ratio can be continuously varied.

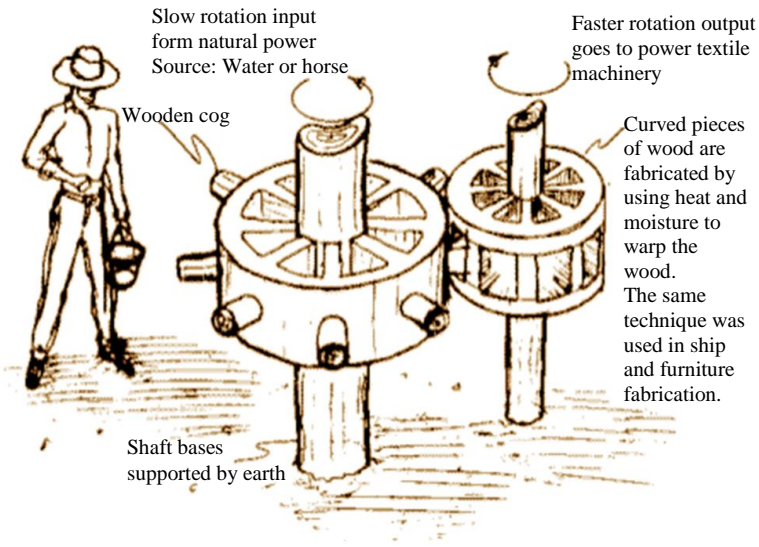


Fig. 1: Ancient Egypt at least a thousand years before Christ. Here, for the first time, the "spiked" wheel transmissions were used for irrigation of crops



Fig. 2: The first adjustable gearboxes were used in 1769 by Cugnot to equip the first vehicle powered by a steam engine

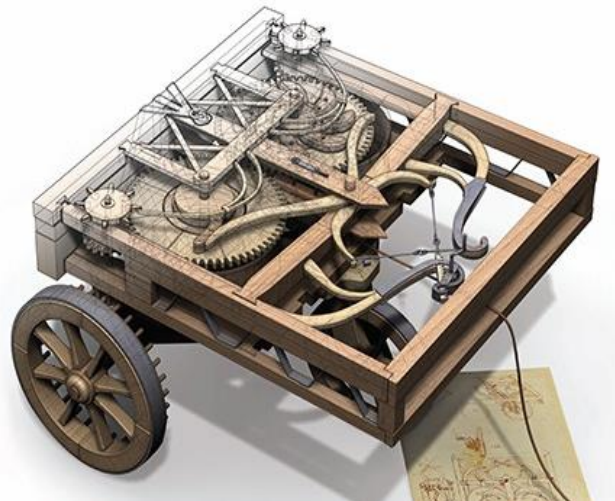


Fig. 3: Car of the Italian craftsman Leonardo da Vinci (15th century)

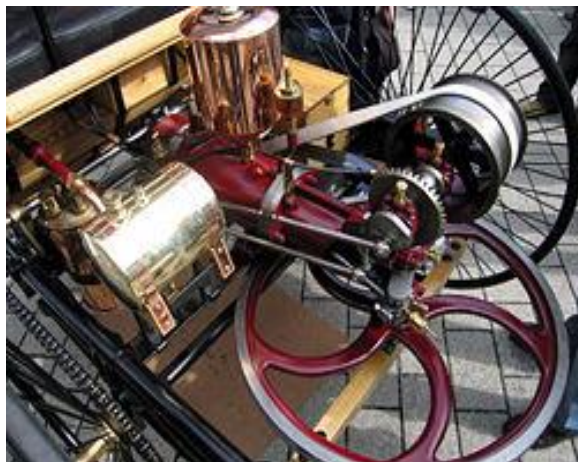


Fig. 4: The Benz engine had gears with gears

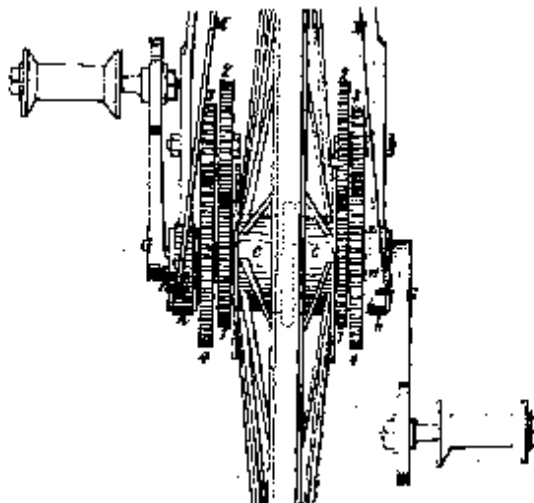


Fig. 5: Sketch of a first patent for cogwheels and cogwheels made in 1870 by the British Starley and Hillman

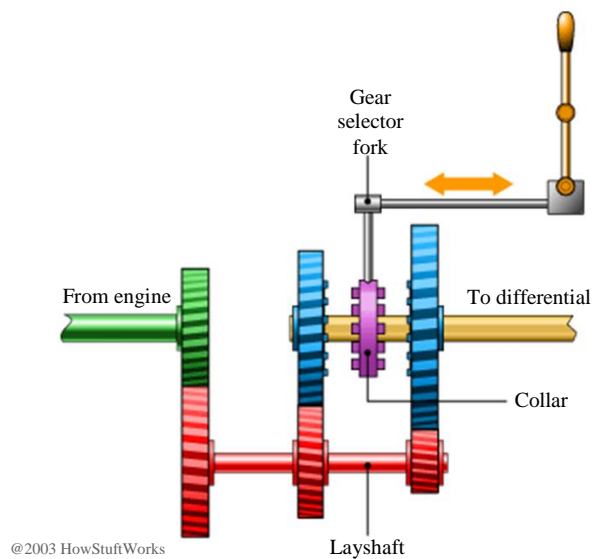


Fig. 6: Gearboxes with fixed axles have the widest spread on all types of vehicles

Gear speed reducers are used in all areas of machine construction (Brewer, 1991).

There is a great variety of speed reducers. They are classified according to the following criteria:

1. After the transmission report
 - a. Reducers with a speed reduction step
 - b. Gearboxes with two or more speed reducers
2. According to the relative position of the input shaft (motor) and the output shaft
 - a. Coaxial reducers (with return), in which the input shaft is coaxial with the output one
 - b. Parallel reducers, in which the input and output shafts are parallel
3. According to the position of the shafts
 - a. Reducers with horizontal axes
 - b. Reducers with vertical axes
 - c. Reducers with inclined axes
4. By gear type
 - a. Cylindrical reducers
 - b. Tapered reducers
 - c. Hypoid reducers
 - d. Gear reducers
 - e. Combined reducers (cylinder-conical, cylinder-snails, etc.)
 - f. Planetary reducers
5. By type of axes
 - a. Fixed axis reducers
 - b. Reducers with movable axles

If the gear unit together with the motor constitutes a single aggregate (the motor is driven directly to the input shaft by a flange) then the unit is called a Gearmotor.

In many constructive solutions, gear reducers with gear wheels are used in kinematic schemes along with other types of transmissions: By belts, chains, friction, screw-nut, speed variator, gearboxes, etc.

The advantages of using reducers in the kinematic diagrams of machines and mechanisms are:

- Constant transmission report
- Ensures a wide range of powers
- Relatively small gauge
- High efficiency (with the exception of gear reducers)
- Simple and cheap maintenance

Disadvantages include:

- High-cost price
- The need for precision machining and assembly
- Their operation is accompanied by noise and vibration

The main parameters of a gearbox are:

- Nominal power
- The transmission report i_T
- Input shaft speed
- Distance between axes (standardized)

Due to the multiple uses in the machine-building industry and various devices, the gear speed parameters are standardized.

The choice of gearbox type in a kinematic scheme is made according to:

- The required transmission report
- The nominal power required
- Average load required
- The average working speed required
- Available size
- The relative position of the axes of the motor and of the working organ (machine)
- The overall efficiency of the kinematic scheme

Depending on these requirements, the following types of gear units can be used: Cylindrical, conical, conical-cylindrical, screw, cylinder-screw, planetary.

Gearboxes with cylindrical gears. These are the most used types of gear reducers because:

- They are produced in a wide range of powers: From very small installed powers (of the order of the Watts) to 900000 W (900 kW)
- Total transmission ratios, $i_T \max = 200$ ($i_T \max = 6.3$, for one-stage reducers; $i_T = 60$, for 2-speed reducers, $i_T = 200$, for 3-speed reducers)
- High peripheral speeds, $V_{\max} = 200$ m/s
- The possibility of typing and standardized or standardized execution

They are constructed in versions with 1, 2 and 3 reduction stages, with straight or inclined teeth. The notations in the figure are:

- Entry into the gearbox, with the letter I
- Exit from the gearbox, with the letter E
- The Fig. 1 to 6, represent the wheels that make up the gears of the reduction gears

From the point of view of the inclination of the gear, when choosing the type of gearbox with cylindrical gears, the following recommendations are taken into account:

- Gearboxes with straight cylindrical gears, for small and medium installed powers, small and medium peripheral speeds and to the ball wheels from the gearboxes

- Gear units with inclined cylindrical gear wheels, for small and medium installed powers, high peripheral speeds, silent gears
- Gearboxes with cylindrical V-toothed gears, for high installed powers and low peripheral speeds (Fig. 7)

The classic gearboxes, manual, the most widespread and most used, are mainly of three types: With two axes (Fig. 8), with three axes with welding the input and output shafts in the direct socket (Fig. 9), with three axes without welding shafts (Fig. 10).

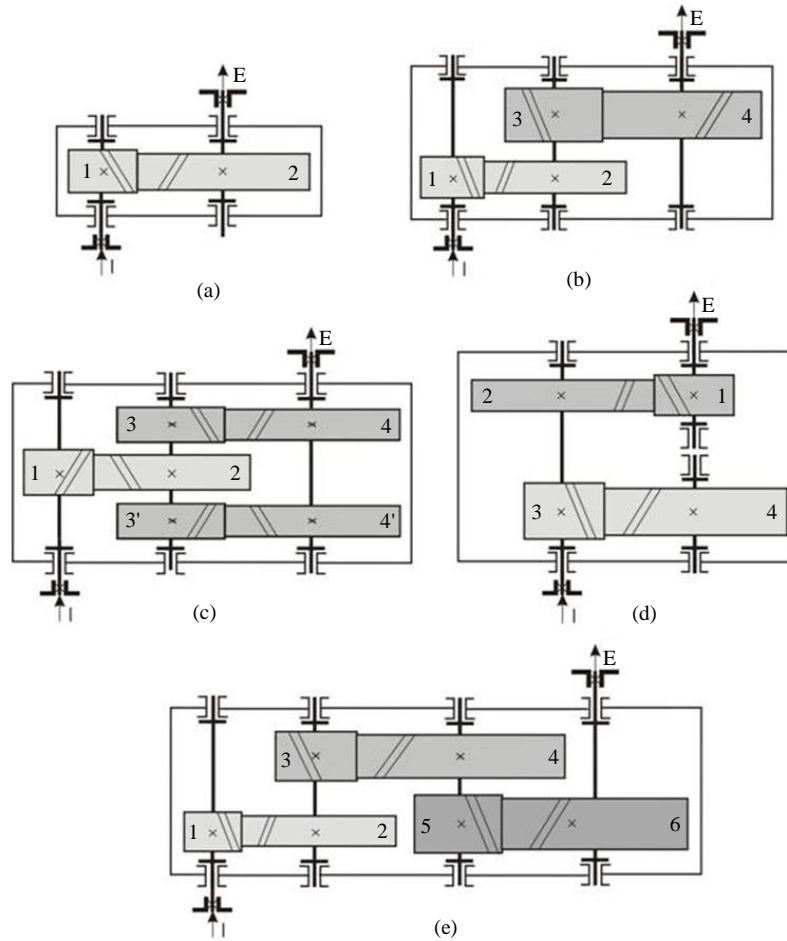


Fig. 7: Kinematic diagrams for cylindrical gearboxes

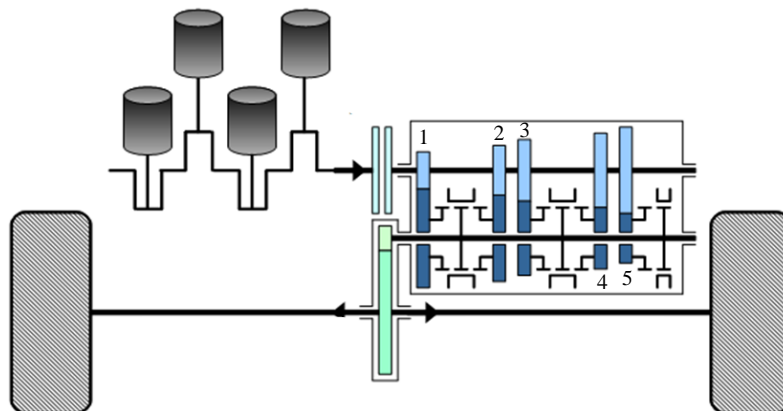


Fig. 8: The classic gearboxes, manual, with two axes

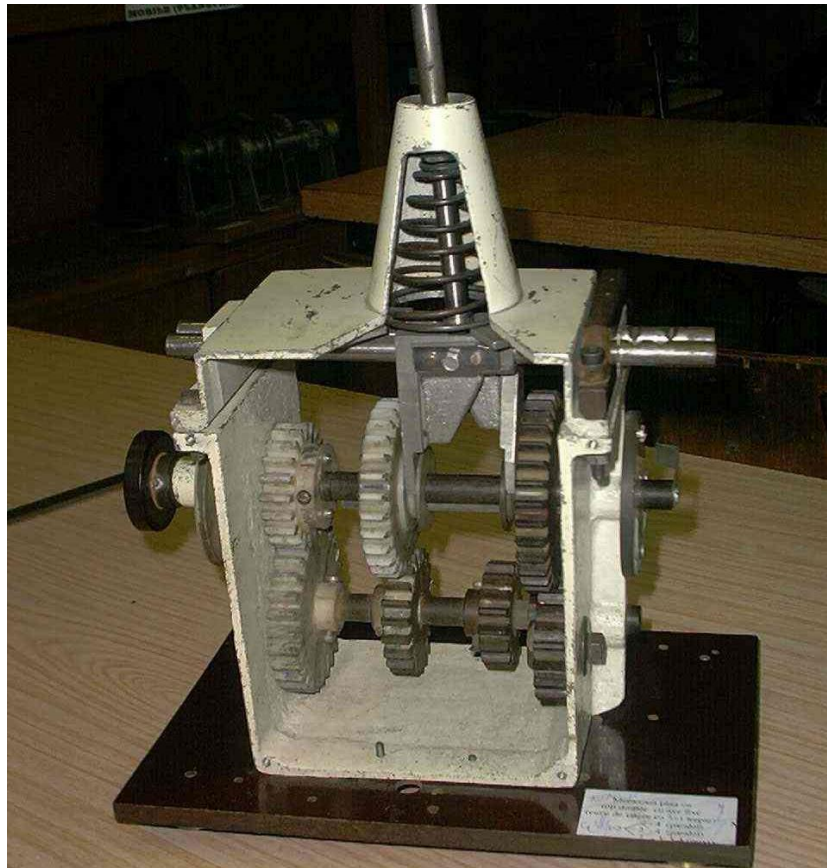


Fig. 9: The classic gearboxes, manual, with three axes with welding the input and output shafts in the direct socket

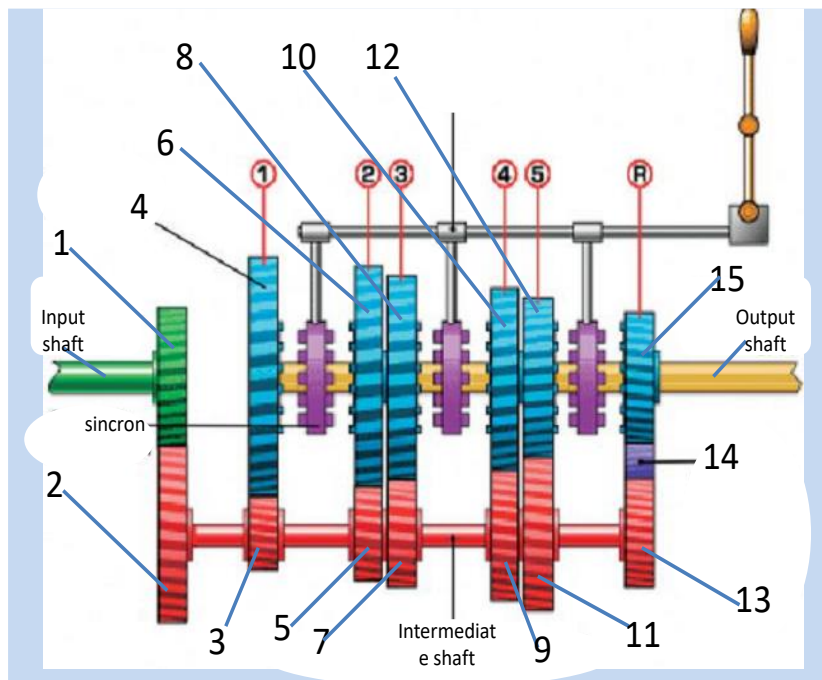


Fig. 10: Classic, manual, three-axis gearboxes without welding the input and output shafts directly

The great advantage of such a system is that the input and output gears are practically welded, i.e., the input shaft is attached to the output shaft in the main direct socket, the transmission ratio becomes 1 and the power lost in this step is very low. Intensely used on the long road being zero, so 100% practical efficiency.

Other gears are not very convenient, however, with lower efficiency and when walking in the city, where the lower gears are used, the gearbox in Fig. 9 with welding shafts, it is no longer advantageous (Antonescu and Petrescu, 1985; 1989; Antonescu *et al.*, 1985a; 1985b; 1986; 1987; 1988; 1994; 1997; 2000a; 2000b; 2001; Aversa *et al.*, 2017a; 2017b; 2017c; 2017d; 2017e; 2016a; 2016b; 2016c; 2016d; Cao *et al.*, 2013; Dong *et al.*, 2013; Franklin, 1930; He *et al.*, 2013; Lee, 2013; Lin *et al.*, 2013; Liu *et al.*, 2013; Padula and Perdereau, 2013; Perumaal and Jawahar, 2013; Petrescu, 2011; 2012; 2019; Petrescu and Petrescu, 1995a; 1995b; 1997a; 1997b; 1997c; 2000a; 2000b; 2002a; 2002b; 2003; 2005a; 2005b; 2005c; 2005d; 2005e; 2011a-c; 2012a; 2012b; 2013a-e; 2014a-h; 2016a; 2016b; 2016c; Petrescu *et al.*, 2007; 2009; 2016; 2017a; 2017b; 2017c; 2017d; 2017e; 2017f; 2017g; 2017h; 2017i; 2017j; 2017k; 2017l; 2017m; 2017n; 2017o; 2017p; 2017q; 2017r; 2017s; 2017t; 2017u; 2017v; 2017w; 2017x; 2018a; 2018b; 2018c; 2018d; 2018e; 2018f; 2018g; Petrescu and Petrescu, 2019a-c; Petrescu and Calautit, 2016a-b; Svensson *et al.*, 2004).

Materials and Methods

A more convenient solution is the one presented in Fig. 10, with three axes without welding shafts, a solution that will be carried out in this work both the kinematic analysis of the gearbox and the dynamic analysis (determination of yields) for the main city steps.

One has 15 gear wheels from 1 to 15, whose tooth numbers are given in Table 1.

The input shaft 1 is a sprocket with gear 1 which is therefore fixed to the shaft and rotates permanently with it, the intermediate or auxiliary shaft 2 has on it 7 gear wheels, 3, 5, 7, 9, 11, 13, which are fixed on the shaft and rotate permanently with the shaft 2 helps permanently transmitting the movement and to the wheels on the output shaft 3 which rotate all the time and have permanent power but transmit it to the output shaft only when they are coupled and the output shaft 3 has freely mounted on it all its gear wheels 4, 6, 8, 10, 12, 15, which will be coupled with the output shaft only one at a time depending on the trap chosen by means of synchronous wheels. There are also ballast mechanisms that prevent the coupling of two steps simultaneously.

Table 1: Tooth numbers

Z1	Z2	Z3	Z4	Z5
40	48	16	50	18
Z6	Z7	Z8	Z9	Z10
20	20	16	24	16
Z11	Z12	Z13	Z14	Z15
36	18	20	16	50

Dynamic analysis of the classic gearbox mechanism, manually, with three axes without solidification steps.

The gearbox efficiency analyzed in the three frequently used steps, 3, 4 and 5, is further determined.

The calculation relationships used in succession are the two below, (I) and (II), only for external gears:

$$\varepsilon^{\alpha,\varepsilon} = \frac{1 + \operatorname{tg}^2 \beta}{2 \cdot \pi} \cdot \left\{ \sqrt{\left[(z_1 + 2 \cdot \cos \beta) \cdot \operatorname{tg} \alpha_0 \right]^2 + 4 \cdot \cos^3 \beta \cdot (z_1 + \cos \beta)} + \sqrt{\left[(z_2 + 2 \cdot \cos \beta) \cdot \operatorname{tg} \alpha_0 \right]^2 + 4 \cdot \cos^3 \beta \cdot (z_2 + \cos \beta)} - (z_1 + z_2) \cdot \operatorname{tg} \alpha_0 \right\} \quad (\text{I})$$

$$\eta_m = \frac{z_1^2 \cdot \cos^2 \beta}{z_1^2 (\operatorname{tg}^2 \alpha_0 + \cos^2 \beta) + \frac{2}{3} \pi^2 \cos^4 \beta (\varepsilon - 1) (2\varepsilon - 1) \pm 2\pi \operatorname{tg} \alpha_0 z_1 \cos^2 \beta (\varepsilon - 1)} \quad (\text{II})$$

Results and Discussion

For the reverse gear, there is an additional gear wheel 14 which reverses the direction of rotation of the output shaft but only in the reverse gear 6.

Kinematic analysis of the classic gearbox mechanism, manually, with three axes without solidification steps.

The transmission reports made by this classic (manual) gearbox are determined (Table 2):

- 1) η_0 is determined for gear 1.2 directly with relations I and II:

$$\eta_0 = 0,853922$$

- 2) η_{78} is determined for gear 7.8 with relations I and II, where 1 is replaced by 7 and 2 is replaced by 8. The yield in step 3 is then obtained through the product between the yield 0 and that of gear 7.8; $\eta_3 = \eta_0 \cdot \eta_{78}$:

$$\eta_{78} = 0,833237$$

$$\eta_3 = 0,711519$$

- 3) $\eta_{9,10}$ is determined for the gear 9.10 with relations I and II, where 1 is replaced by 9 and 2 is replaced by 10. The yield in step 4 is then obtained by the product between the yield 0 and that of the gear 9.10; $\eta_4 = \eta_0 \cdot \eta_{9,10}$:

$$\eta_{9,10} = 0,842036$$

$$\eta_4 = 0,719033$$

- 4) $\eta_{11,12}$ is determined for the gear 11.12 with relations I and II, where 1 is replaced by 11 and 2 is replaced by 12. The yield in step 5 is then obtained by the product between the yield 0 and that of the gear 11.12; $\eta_5 = \eta_0 \cdot \eta_{11,12}$:

$$\eta_{11,12} = 0,856361$$

$$\eta_5 = 0,731265$$

The angles α_0, β are constant for all the gears of this gearbox as follows: $\alpha_0 = 18,6, \beta = 16$.

The efficiency results in the main gears, which are used more often by the driver of the vehicle, are generally high and could be optimized so that they are even higher with the optimum design of the gearbox, with gears having a larger number of teeth and possibly with decreasing the pressure angle normal on the division circle to an optimum value.

Next, another optimized constructive variant will be presented (Table 3 and 4).

- 1) η_0 is determined for gear 1.2 directly with relations I and II:

$$\eta_0 = 0,82513$$

- 2) η_{78} is determined for gear 7.8 with relations I and II, where 1 is replaced by 7 and 2 is replaced by 8. The yield in step 3 is then obtained through the product between the yield 0 and that of gear 7.8; $\eta_3 = \eta_0 \cdot \eta_{78}$:

$$\eta_{78} = 0,80379$$

$$\eta_3 = 0,66323$$

- 3) $\eta_{9,10}$ is determined for the gear 9.10 with relations I and II, where 1 is replaced by 9 and 2 is replaced by 10. The yield in step 4 is then obtained by the product between the yield 0 and that of the gear 9.10; $\eta_4 = \eta_0 \cdot \eta_{9,10}$:

$$\eta_{9,10} = 0,81154$$

$$\eta_4 = 0,66962$$

- 4) $\eta_{11,12}$ is determined for the gear 11.12 with relations I and II, where 1 is replaced by 11 and 2 is replaced by 12. The yield in step 5 is then obtained by the product between the yield 0 and that of the gear 11.12; $\eta_5 = \eta_0 \cdot \eta_{11,12}$:

$$\eta_{11,12} = 0,82433$$

$$\eta_5 = 0,68018$$

The angles α_0, β are constant for all the gears of this gearbox, as follows: $\alpha_0 = 19,9, \beta = 29$:

As can be seen from the new values, the transmission ratios in the five forward gears of the gearbox are much optimized, even if their output yield is slightly lower. This efficiency can be increased significantly, especially for the most used steps by drivers, 3, 4, 5, through a new constructive scheme in which the number of teeth of the wheels will be much higher and eventually the pressure angle of the wheels will be reduced (alpha 0).

Table 2: The transmission reports made by this classic (manual) gearbox

$i_0 = i_{12}$	i_{34}	i_{56}	i_{78}	$i_{9,10}$
$= -z_2/z_1$	$= -z_4/z_3$	$= -z_6/z_5$	$= -z_8/z_7$	$= -z_{10}/z_9$
-1,2	-3,125	-1,11111	-0,8	-0,66667
$i_{11,12}$	$i_{13,14}$	$i_{14,15}$	i_1	i_2
$= -z_{12}/z_{11}$	$= -z_{14}/z_{13}$	$= -z_{15}/z_{14}$	$= i_0 i_{34}$	$= i_0 i_{56}$
-0,5	-0,8	-3,125	3,75	1,333333
i_3	i_4	i_5	$i_6 = i_{MR}$	
$= i_0 i_{78}$	$= i_0 i_{9,10}$	$= i_0 i_{11,12}$	$= i_0 i_{13,14} i_{14,15}$	
0,96	0,8	0,6	-3	

Table 3: Another optimized constructive variant

Z1	Z2	Z3	Z4	Z5
40	48	16	50	18
Z6	Z7	Z8	Z9	Z10
30	20	24	24	24
Z11	Z12	Z13	Z14	Z15
36	27	20	16	50

Table 4: Another optimized constructive variant

$i_0 = i_{12}$	i_{34}	i_{56}	i_{78}	$i_{9,10}$
$= -z_2/z_1$	$= -z_4/z_3$	$= -z_6/z_5$	$= -z_8/z_7$	$= -z_{10}/z_9$
-1,2	-3,125	-1,667	-1,2	-1
$i_{11,12}$	$i_{13,14}$	$i_{14,15}$	i_1	i_2
$= -z_{12}/z_{11}$	$= -z_{14}/z_{13}$	$= -z_{15}/z_{14}$	$= i_0 i_{34}$	$= i_0 i_{56}$
-0,75	-0,8	-3,125	3,75	2
i_3	i_4	i_5	$i_6 = i_{MR}$	
$= i_0 i_{78}$	$= i_0 i_{9,10}$	$= i_0 i_{11,12}$	$= i_0 i_{13,14} i_{14,15}$	
1,44	1,2	0,9	-3	

To be able to observe if the decrease of the efficiency of the gears is due to the optimum arrangement of the gears or the fact that the value of the angle of pressure α_0 has changed by increasing, then the recalculation of the new stages optimized with the same values for the angles of pressure and inclination of the gear will proceed as in the case used initially and it will be observed that the gears' returns go back to the initial values, so their decrease was mainly due not to the fact that the tooth numbers of the wheels were optimized in order to obtain optimum transmission ratios but due to the fact that the value of the pressure angle had been raised to close to the known standard value of 20 [deg]:

- 1) η_0 is determined for gear 1.2 directly with relations I and I:

$$\begin{array}{c} \eta_0 \\ 0,85392 \end{array}$$

- 2) η_{78} is determined for gear 7.8 with relations I and II, where 1 is replaced by 7 and 2 is replaced by 8. The yield in step 3 is then obtained through the product between the yield 0 and that of gear 7.8; $\eta_3 = \eta_0 \cdot \eta_{78}$:

$$\begin{array}{c} \eta_{78} \\ 0,82739 \\ \eta_3 \\ 0,70653 \end{array}$$

- 3) $\eta_{9,10}$ is determined for the gear 9.10 with relations I and II, where 1 is replaced by 9 and 2 is replaced by 10. The yield in step 4 is then obtained by the product between the yield 0 and that of the gear 9.10; $\eta_4 = \eta_0 \cdot \eta_{9,10}$:

$$\begin{array}{c} \eta_{9,10} \\ 0,83729 \\ \eta_4 \\ 0,71498 \end{array}$$

- 4) $\eta_{11,12}$ is determined for the gear 11.12 with relations I and II, where 1 is replaced by 11 and 2 is replaced by 12. The yield in step 5 is then obtained by the product between the yield 0 and that of the gear 11.12; $\eta_5 = \eta_0 \cdot \eta_{11,12}$:

$$\begin{array}{c} \eta_{11,12} \\ 0,85344 \\ \eta_5 \\ 0,72877 \end{array}$$

The angles α_0, β are constant for all the gears of this gearbox as follows: $\alpha_0 = 18.6, \beta = 16$.

We will continue to repeat the final experiment, with optimized transmission ratios, but using a much lower pressure angle, in an attempt to achieve higher efficiencies at the main gear gears:

- 1) η_0 is determined for gear 1.2 directly with relations I and II:

$$\begin{array}{c} \eta_0 \\ 0,93314 \end{array}$$

- 2) η_{78} is determined for gear 7.8 with relations I and II, where 1 is replaced by 7 and 2 is replaced by 8. The yield in step 3 is then obtained through the product between the yield 0 and that of gear 7.8; $\eta_3 = \eta_0 \cdot \eta_{78}$:

$$\begin{array}{c} \eta_{78} \\ 0,89096 \\ \eta_3 \\ 0,83139 \end{array}$$

- 3) $\eta_{9,10}$ is determined for the gear 9.10 with relations I and II, where 1 is replaced by 9 and 2 is replaced by 10. The yield in step 4 is then obtained by the product between the yield 0 and that of the gear 9.10; $\eta_4 = \eta_0 \cdot \eta_{9,10}$:

$$\begin{array}{c} \eta_{9,10} \\ 0,90950 \\ \eta_4 \\ 0,84869 \end{array}$$

- 4) $\eta_{11,12}$ is determined for the gear 11.12 with relations I and II, where 1 is replaced by 11 and 2 is replaced by 12. The yield in step 5 is then obtained by the product between the yield 0 and that of the gear 11.12; $\eta_5 = \eta_0 \cdot \eta_{11,12}$:

$$\begin{array}{c} \eta_{11,12} \\ 0,93716 \\ \eta_5 \\ 0,87450 \end{array}$$

The angles α_0, β are constant for all the gears of this gearbox as follows: $\alpha_0 = 4, \beta = 40$.

It is observed the increase of the efficiency in all the gears due to the decrease of the angle of pressure to gears to the value 4 [deg], in the conditions in which the numbers of optimized teeth were used to have some ratios of optimal transmission and power in all the gears speeds.

In fact, this is the main innovation brought about by the paper and by the original relations of calculation of the yield (I and II) used in the paper.

From a technological point of view, it is not too easy to lower the alpha angle of pressure to such a low value, 4

[deg]. Initially following the new theory proposed by the authors of this paper, since 2002, in France, the gear pressure angle has been reduced by only 4 [deg] from 20 [deg] to 16 [deg], but in the United States a little later, in laboratories of aerospace including NASA managed to reach values of 10-12 [deg], but did not go below 10 [deg], in the years 2010-2014. The authors of the paper verified various possible scenarios with the INVENTOR software, in 2014 (Petrescu and Petrescu, 2014a) and all the values obtained with the help of the new calculation relationships, I and II, were fully verified with those obtained with the help of the Inventor 2014 software, from $\alpha_0 = 10$ [deg] the $\alpha_0 = 30$ [deg]. Undervalued 10 [deg] the software said that it has no experimental data and cannot calculate the values proposed by the authors, on that date 2014. Even so with only $\alpha_0 = 10$ [deg] the values of the gears can be greatly optimized if the number is increased the teeth of the wheels used in gears, with the possible increase of the gauge, but with the great advantage of obtaining very high efficiency in the mechanical transmissions with fixed gears used.

Conclusion

Fixed axis mechanical transmissions today have the largest spread on the entire planet, being practically used in almost all areas. From the gearboxes of the vehicles to the stationary reducers, used in the electrical, electronic and electrotechnical equipment, in the heavy industry but also in the light industry, in energy and in transport, practically the transmissions with fixed axes meet today everywhere, being part of our daily life. For this reason, we want to present the kinematic and dynamic analysis at a classic, manual, three-axis gearbox, with six speeds, without a direct socket. With the help of the presented method, one can analyze the dynamics of such a gearbox in order to design it optimally, so as to achieve the highest yields in operation, especially in the most used speeds, so that the transmission losses are limited to up.

The classic gearboxes, manual, the most widespread and most used, are mainly of three types: With two axes (Fig. 8), with three axes with welding the input and output shafts in the direct socket (Fig. 9), with three axles without welding shafts (Fig. 10).

The paper examines how the geometrical-kinematic and dynamic analysis based on its yield and is performed based on a classic, manual, three-axis gearbox, without a direct socket, a model that is widely used in the automotive industry worldwide, from the beginnings of the car and until today, even if in the meantime there have been automatic gearboxes, or other with two gearboxes classics, to which a double-clutch has been added.

The efficiency results in the main gears, which are used more often by the driver of the vehicle, are generally high and could be optimized so that they are

even higher with the optimum design of the gearbox, with gears having a larger number of teeth and possibly with decreasing the pressure angle normal on the division circle to an optimum value.

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Author's Contributions

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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Figure 01-09:

Petrescu and Petrescu, 2011b, 2014; Petrescu, 2011

Figure 10:

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