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Dynamic Synthesis of a Classic, Manual Gearbox

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Abstract: Today, various types of gearboxes have been introduced into the vehicles to change the way the classic manual gearboxes work, such as automatic gearboxes, semi-automatic, continuous variable, dual-clutch automatic gearboxes etc. However, most gear shifters for in-service vehicles are still classical manuals, which is why their optimal synthesis based on their dynamics and especially on optimal performance is now more than necessary. 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 classic, mechanical, manual gearshift can be achieved regardless of its operating status.

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

Introduction

The gearbox, or gearshift, is the centerpiece of a transmission.

The most common types of gearboxes have been and are still maintained, the classic gearboxes, initially the first gearboxes being built with spur gears, with straight teeth since they were permanently coupled with the secondary (output) shaft, rotating permanently with it, the various steps being practiced by coupling or decoupling such balancing wheel with its equivalent wheel located on the tertiary, intermediate or auxiliary shaft, all the wheels on the secondary shaft being also permanently connected in rotation. The tertiary (intermediate or auxiliary) shaft receives constant rotation from the input shaft by means of a permanent fixed gear made between the input shaft sprocket and the corresponding sprocket on the intermediate shaft, thus constantly driving the intermediate shaft together with all its wheels (Fig. 1).

At the initial solution (Fig. 1), the output shaft wheels were coupled in turn with one of the wheels of the intermediate shaft by moving them on the output shaft guided on some grooves. Obviously for this reason, they could only be riding straight teeth (which have many drawbacks compared to tilted or curved teeth).

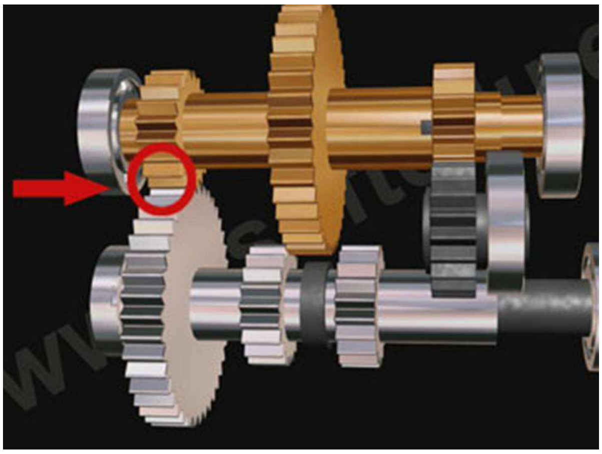

Fig. 1: Classic gearbox with sliding wheels

Since the 1970s, this initial solution has been quickly replaced with a top one with all the wheels (including those of the output shaft) designed with inclined or curved teeth, the output shaft wheels being normally free on their secondary (output shaft), i.e., rotating freely on it and being rotatably coupled to the secondary shaft, in turn, by means of syncrons, which are smaller toothed wheels with straight teeth permanently rotating with the secondary shaft and which can to balancing to engage with the respective gear, thus enabling it to (selectively) engage the secondary shaft in rotation. Such a solution, also used in trucks and buses, is shown in Fig. 2, the classic gearbox with syncrones representing the best solution that existed and which is still the most used today, even if the boxes appeared, refined and multiplied hybrid or automatic gears.

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 classic, mechanical, manual gearshift can be achieved regardless of its operating status (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-1910, 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).

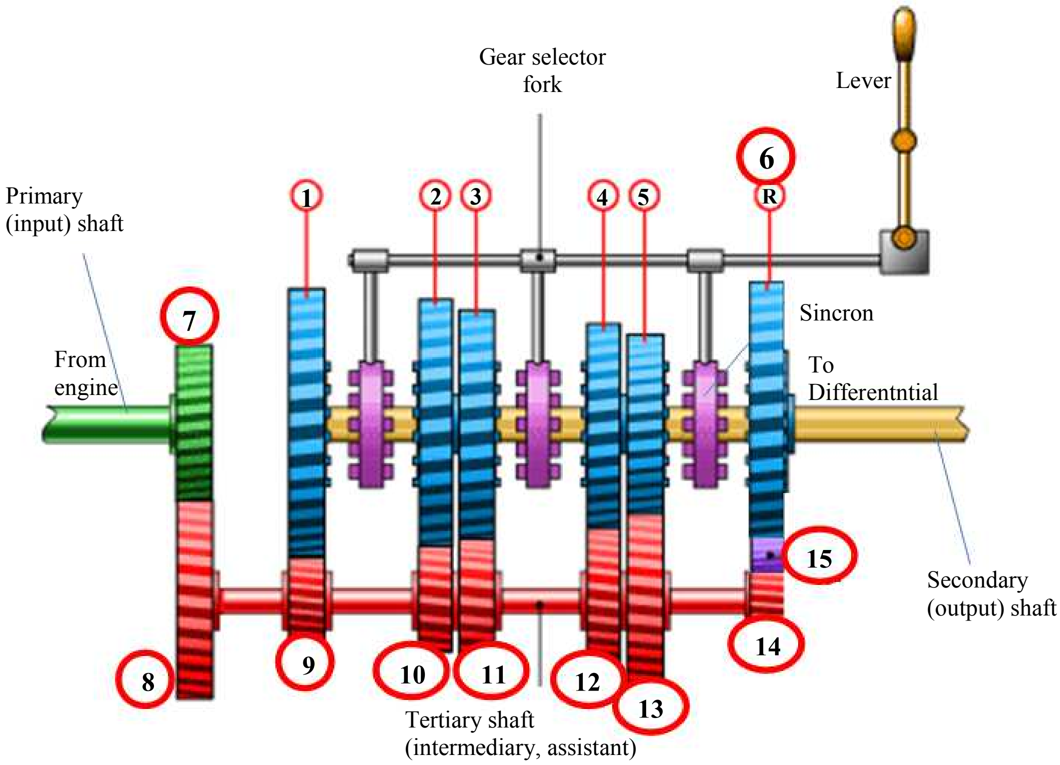

Fig. 2: Manual gearbox, synchronous (with curved or sloping teeth)

Materials and Methods

The input shaft (7) of the primary shaft (7) continuously transmits the rotational movement to the intermediate shaft by means of the wheel (8), the permanent gear 7-8 being the one that participates in all the obtained gears.

For stage I, the wheel 1 of the output shaft is rotated with it, via its synchronizer, supported by a sleeve actuated by the corresponding fork, which in turn is actuated by the gearshift lever (gearbox). Normally, the wheel 1 rotates permanently on the output shaft (free to pull it), being in permanent engagement with the wheel 9 of the intermediate shaft. The power flow is permanently transmitted to the wheel 1 by the wheels 7-8-9-1, by means of the two gears 7-8 and 9-1. When the wheel 9 is coupled to the output shaft, the power flow will also be transmitted to it. Latching mechanisms do not allow simultaneous coupling of two or more steps. The power flow of the first step can be expressed using relationships 1:

$$\begin{cases} i_{78} = -\dfrac{z_8}{z_7} \\ i_{91} = -\dfrac{z_1}{z_9} \\ i_I = i_{78} \cdot i_{91} = -\dfrac{z_8}{z_7} \cdot (-)\dfrac{z_1}{z_9} = \dfrac{z_8 \cdot z_1}{z_7 \cdot z_9} \end{cases} \tag{1}$$

The second step is achieved by coupling the wheel 2 in rotation with the output shaft, through its mechanism, with the corresponding synchronous. The power flow 7-8-10-2, will be given by system relationships (2):

$$\begin{cases} i_{78} = -\dfrac{z_8}{z_7} \\ i_{10,2} = -\dfrac{z_2}{z_{10}} \\ i_{II} = i_{78} \cdot i_{10,2} = -\dfrac{z_8}{z_7} \cdot (-)\dfrac{z_2}{z_{10}} = \dfrac{z_8 \cdot z_2}{z_7 \cdot z_{10}} \end{cases} \tag{2}$$

The third step is achieved by coupling the wheel 3 in rotation with the output shaft, through its mechanism, with the corresponding synchronous. Power flow 7-8-11-3, will be given by system relationships (3):

$$\begin{cases} i_{78} = -\dfrac{z_8}{z_7} \\ i_{11,3} = -\dfrac{z_3}{z_{11}} \\ i_{III} = i_{78} \cdot i_{11,3} = -\dfrac{z_8}{z_7} \cdot (-)\dfrac{z_3}{z_{11}} = \dfrac{z_8 \cdot z_3}{z_7 \cdot z_{11}} \end{cases} \tag{3}$$

The fourth stage is achieved by coupling the wheel 4 in rotation with the output shaft, through its mechanism, with the corresponding synchronizer. The power flow 7-8-12-4 will be given by system relationships (4):

$$\begin{cases} i_{78} = -\dfrac{z_8}{z_7} \\ i_{12,4} = -\dfrac{z_4}{z_{12}} \\ i_{IV} = i_{78} \cdot i_{12,4} = -\dfrac{z_8}{z_7} \cdot (-)\dfrac{z_4}{z_{12}} = \dfrac{z_8 \cdot z_4}{z_7 \cdot z_{12}} \end{cases} \tag{4}$$

The fifth (last, forward) step is achieved by coupling the wheel 5 in rotation with the output shaft, through its mechanism, with the corresponding synchronous. The power flow 7-8-13-5 will be given by system relationships (5):

$$\begin{cases} i_{78} = -\dfrac{z_8}{z_7} \\ i_{13,5} = -\dfrac{z_5}{z_{13}} \\ i_V = i_{78} \cdot i_{13,5} = -\dfrac{z_8}{z_7} \cdot (-)\dfrac{z_5}{z_{13}} = \dfrac{z_8 \cdot z_5}{z_7 \cdot z_{13}} \end{cases} \tag{5}$$

The sixth step (reversing, or mars arriere) is accomplished by coupling the wheel 6 in rotation with the output shaft, through its mechanism, with the corresponding synchronous. The power flow 7-8-14-15-6, will be given by system relationships (6):

$$\begin{cases} i_{78} = -\dfrac{z_8}{z_7} \\ i_{14,15} = -\dfrac{z_{15}}{z_{14}} \\ i_{15,6} = -\dfrac{z_6}{z_{15}} \\ i_{VI} \equiv i_{MR} = i_{78} \cdot i_{14,15} \cdot i_{15,6} \\ = -\dfrac{z_8}{z_7} \cdot (-)\dfrac{z_{15}}{z_{14}} \cdot (-)\dfrac{z_6}{z_{15}} = -\dfrac{z_8 \cdot z_6}{z_7 \cdot z_{14}} \end{cases} \tag{6}$$

At each gear ratio of an outer gear, there is a minus sign showing that the direction of rotation from the input wheel of gear to the output wheel of the gear is changed. This sign is a conventional one, but it is important in calculations because the product of a even number of reports gives the sign plus to the end, while the product of an odd number of transmission reports generates the final sign minus.

From the final system relationship (6), written for backward travel, it can be noticed that with three gears instead of two (as in the forward steps), three minus signs multiplied finally give the minus sign so that the spindle the output changes its direction of rotation relative to the input shaft, causing the wheels of the bus to rotate in reverse and move it backwards, even if the internal combustion engine never changes its direction rotation.

Each forward gear shall have two corresponding transmission reports to two gears and three gears corresponding to three gears shall be taken at the reverse gear.

The main gear unit 7-8 is involved in achieving the power flow in each of the six gears achieved.

The flow is actually reversed by only four wheels 7-8-14-6 and the reversing wheel 15 does not actually participate in the final transmission ratio, i_{MR}, but has the essential role of changing the direction of rotation of the shaft output of the gearbox.

Results and Discussion

The efficiency of each gearbox step is calculated individually, depending on the gears involved in the gear. Exact calculations are made using relations 7-9, where z_1 always represents the number of teeth at the input wheel of a gear and z_2 the number of teeth at the output wheel of a gear; α_0 is the normal engagement angle on the dividing circle, which typically has a standardized value, the most used being the value of 20°; β represents the tilting angle of the gear teeth (normally the gear teeth, but a gear uses compulsory wheels with the same taper angle β and the same engagement angle α_0); ε represents the degree of coverage of a gear (i.e., how many pairs of teeth are in the gear engaged). For external gearing, the degree of coverage ε is calculated first with relation (8) and then the mechanical efficiency of the gear using the relation (7). If we are dealing with an inner gear, first determines its degree of coverage by means of the relation (9), after which the mechanical efficiency of the gear is calculated by means of the relation (7). The calculations are repeated for each gear separately. Note: In relation (7) there is a plus or minus sign, where it will be taken + (plus) for all situations where the input gear wheel is externally toothed and the – (minus) sign will only be adopted if the input wheel in the gear will be one with internal teeth (a toothed crown). When determining the degree of coverage in the case of an inner gear, which uses instead of two external gear wheels, an external tooth wheel and another internal tooth, the z_e and z_i notations corresponding to the number of teeth of the external teeth wheel respectively of the one with internal teeth (instead of the z_1 and z_2 notations used for the external engagement):

$$\eta_m = \frac{z_1^2 \cdot \cos^2\beta}{z_1^2(tg^2\alpha_0 + \cos^2\beta) + \frac{2}{3}\pi^2\cos^4\beta(\varepsilon-1)(2\varepsilon-1) \pm 2\pi tg\alpha_0 z_1 \cos^2\beta(\varepsilon-1)} \tag{7}$$

$$\begin{aligned}\varepsilon^{a.e.} = \frac{1+tg^2\beta}{2\cdot\pi}\cdot\Big\{&\sqrt{\left[(z_1+2\cdot\cos\beta)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot(z_1+\cos\beta)}\\ &+\sqrt{\left[(z_2+2\cdot\cos\beta)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot(z_2+\cos\beta)}\\ &-(z_1+z_2)\cdot tg\alpha_0\Big\}\end{aligned} \tag{8}$$

$$\begin{aligned}\varepsilon^{a.i.} = \frac{1+tg^2\beta}{2\cdot\pi}\cdot\Big\{&\sqrt{\left[(z_e+2\cdot\cos\beta)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot(z_e+\cos\beta)}\\ &-\sqrt{\left[(z_i-2\cdot\cos\beta)\cdot tg\alpha_0\right]^2-4\cdot\cos^3\beta\cdot(z_i-\cos\beta)}\\ &-(z_e-z_i)\cdot tg\alpha_0\Big\}\end{aligned} \tag{9}$$

At each transmission stage of the forward gearbox (for the box in Fig. 2, being five forward strokes) we have two gears, so two separate mechanical returns (one for each gear) will be determined separately, the mechanical yield of the respective step being given by the product of the partial yields (belonging to the two gears participating in the respective gear).

For the gearbox in Fig. 2, the system relations (10) will be used to determine the first gear:

$$\begin{cases}\varepsilon_{78}^{a.e.} = \frac{1+tg^2\beta}{2\cdot\pi}\cdot\Big\{\sqrt{\left[(z_7+2\cdot\cos\beta)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot(z_7+\cos\beta)}\\ +\sqrt{\left[(z_8+2\cdot\cos\beta)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot(z_8+\cos\beta)}-(z_7+z_8)\cdot tg\alpha_0\Big\}\\ \eta_{78} = \frac{z_7^2\cdot\cos^2\beta}{z_7^2(tg^2\alpha_0+\cos^2\beta)+\frac{2}{3}\pi^2\cos^4\beta(\varepsilon-1)(2\varepsilon-1)\pm 2\pi tg\alpha_0 z_7\cos^2\beta(\varepsilon-1)}\\ \varepsilon_{91}^{a.e.} = \frac{1+tg^2\beta}{2\cdot\pi}\cdot\Big\{\sqrt{\left[(z_9+2\cdot\cos\beta)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot(z_9+\cos\beta)}\\ +\sqrt{\left[(z_1+2\cdot\cos\beta)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot(z_1+\cos\beta)}-(z_9+z_1)\cdot tg\alpha_0\Big\}\\ \eta_{91} = \frac{z_9^2\cdot\cos^2\beta}{z_9^2(tg^2\alpha_0+\cos^2\beta)+\frac{2}{3}\pi^2\cos^4\beta(\varepsilon-1)(2\varepsilon-1)\pm 2\pi tg\alpha_0 z_9\cos^2\beta(\varepsilon-1)}\\ \eta_I \equiv \eta_{71} = \eta_{78}\cdot\eta_{91}\end{cases} \tag{10}$$

For the gearbox in Fig. 2, the system relations (11) will be used to determine the gearing of the second gear:

$$\left\{\begin{array}{l}
\varepsilon_{78}^{a.e.}=\dfrac{1+tg^2\beta}{2\cdot\pi}\cdot\left\{\sqrt{\left[\left(z_7+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_7+\cos\beta\right)}\right.\\
\left.+\sqrt{\left[\left(z_8+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_8+\cos\beta\right)}-\left(z_7+z_8\right)\cdot tg\alpha_0\right\}\\
\eta_{78}=\dfrac{z_7^2\cdot\cos^2\beta}{z_7^2(tg^2\alpha_0+\cos^2\beta)+\dfrac{2}{3}\pi^2\cos^4\beta(\varepsilon-1)(2\varepsilon-1)\pm 2\pi tg\alpha_0 z_7\cos^2\beta(\varepsilon-1)}\\
\varepsilon_{10,2}^{a.e.}=\dfrac{1+tg^2\beta}{2\cdot\pi}\cdot\left\{\sqrt{\left[\left(z_{10}+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_{10}+\cos\beta\right)}\right.\\
\left.+\sqrt{\left[\left(z_2+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_2+\cos\beta\right)}-\left(z_{10}+z_2\right)\cdot tg\alpha_0\right\}\\
\eta_{10,2}=\dfrac{z_{10}^2\cdot\cos^2\beta}{z_{10}^2(tg^2\alpha_0+\cos^2\beta)+\dfrac{2}{3}\pi^2\cos^4\beta(\varepsilon-1)(2\varepsilon-1)\pm 2\pi tg\alpha_0 z_{10}\cos^2\beta(\varepsilon-1)}\\
\eta_{II}\equiv\eta_{72}=\eta_{78}\cdot\eta_{10,2}
\end{array}\right. \tag{11}$$

For the gearbox of Fig. 2, the system relations (12) will be used to determine the gearing of the third gear:

$$\left\{\begin{array}{l}
\varepsilon_{78}^{a.e.}=\dfrac{1+tg^2\beta}{2\cdot\pi}\cdot\left\{\sqrt{\left[\left(z_7+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_7+\cos\beta\right)}\right.\\
\left.+\sqrt{\left[\left(z_8+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_8+\cos\beta\right)}-\left(z_7+z_8\right)\cdot tg\alpha_0\right\}\\
\eta_{78}=\dfrac{z_7^2\cdot\cos^2\beta}{z_7^2(tg^2\alpha_0+\cos^2\beta)+\dfrac{2}{3}\pi^2\cos^4\beta(\varepsilon-1)(2\varepsilon-1)\pm 2\pi tg\alpha_0 z_7\cos^2\beta(\varepsilon-1)}\\
\varepsilon_{11,3}^{a.e.}=\dfrac{1+tg^2\beta}{2\cdot\pi}\cdot\left\{\sqrt{\left[\left(z_{11}+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_{11}+\cos\beta\right)}\right.\\
\left.+\sqrt{\left[\left(z_3+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_3+\cos\beta\right)}-\left(z_{11}+z_3\right)\cdot tg\alpha_0\right\}\\
\eta_{11,3}=\dfrac{z_{11}^2\cdot\cos^2\beta}{z_{11}^2(tg^2\alpha_0+\cos^2\beta)+\dfrac{2}{3}\pi^2\cos^4\beta(\varepsilon-1)(2\varepsilon-1)\pm 2\pi tg\alpha_0 z_{11}\cos^2\beta(\varepsilon-1)}\\
\eta_{III}\equiv\eta_{73}=\eta_{78}\cdot\eta_{11,3}
\end{array}\right. \tag{12}$$

For the gearbox of Fig. 2, the system relationships (13) will be used to determine the fourth gear gear:

$$\left\{\begin{array}{l}
\varepsilon_{78}^{a.e.}=\dfrac{1+tg^2\beta}{2\cdot\pi}\cdot\left\{\sqrt{\left[\left(z_7+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_7+\cos\beta\right)}\right.\\
\left.+\sqrt{\left[\left(z_8+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_8+\cos\beta\right)}-\left(z_7+z_8\right)\cdot tg\alpha_0\right\}\\
\eta_{78}=\dfrac{z_7^2\cdot\cos^2\beta}{z_7^2(tg^2\alpha_0+\cos^2\beta)+\dfrac{2}{3}\pi^2\cos^4\beta(\varepsilon-1)(2\varepsilon-1)\pm 2\pi tg\alpha_0 z_7\cos^2\beta(\varepsilon-1)}\\
\varepsilon_{12,4}^{a.e.}=\dfrac{1+tg^2\beta}{2\cdot\pi}\cdot\left\{\sqrt{\left[\left(z_{12}+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot 2\left(z_{12}+\cos\beta\right)}\right.\\
\left.+\sqrt{\left[\left(z_4+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_4+\cos\beta\right)}-\left(z_{12}+z_4\right)\cdot tg\alpha_0\right\}\\
\eta_{12,4}=\dfrac{z_{12}^2\cdot\cos^2\beta}{z_{12}^2(tg^2\alpha_0+\cos^2\beta)+\dfrac{2}{3}\pi^2\cos^4\beta(\varepsilon-1)(2\varepsilon-1)\pm 2\pi tg\alpha_0 z_{12}\cos^2\beta(\varepsilon-1)}\\
\eta_{IV}\equiv\eta_{74}=\eta_{78}\cdot\eta_{12,4}
\end{array}\right. \tag{13}$$

For the gearbox of Fig. 2, the system relationships (14) will be used to determine the fifth gear gear:

$$\begin{cases} \varepsilon_{78}^{a.e.} = \frac{1+tg^2\beta}{2\cdot\pi}\cdot\left\{\sqrt{\left[\left(z_7+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_7+\cos\beta\right)}\right. \\ \left.+\sqrt{\left[\left(z_8+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_8+\cos\beta\right)}-\left(z_7+z_8\right)\cdot tg\alpha_0\right\} \\ \eta_{78} = \frac{z_7^2\cdot\cos^2\beta}{z_7^2(tg^2\alpha_0+\cos^2\beta)+\frac{2}{3}\pi^2\cos^4\beta(\varepsilon-1)(2\varepsilon-1)\pm 2\pi tg\alpha_0 z_7\cos^2\beta(\varepsilon-1)} \\ \varepsilon_{13,5}^{a.e.} = \frac{1+tg^2\beta}{2\cdot\pi}\cdot\left\{\sqrt{\left[\left(z_{13}+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot 2\left(z_{13}+\cos\beta\right)}\right. \\ \left.+\sqrt{\left[\left(z_5+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_5+\cos\beta\right)}-\left(z_{13}+z_5\right)\cdot tg\alpha_0\right\} \\ \eta_{13,5} = \frac{z_{13}^2\cdot\cos^2\beta}{z_{13}^2(tg^2\alpha_0+\cos^2\beta)+\frac{2}{3}\pi^2\cos^4\beta(\varepsilon-1)(2\varepsilon-1)\pm 2\pi tg\alpha_0 z_{13}\cos^2\beta(\varepsilon-1)} \\ \eta_V \equiv \eta_{75} = \eta_{78}\cdot\eta_{13,5} \end{cases} \tag{14}$$

For the gearbox of Fig. 2, the system relationships (15) will be used to determine the sixth gear (reverse).

Observation. The reverse gear is made up of three different gears, which is also reflected in the dynamics of the gearshift mechanism, so that gearbox performance in the reverse gear is determined by three partial yields calculated each separately so that total efficiency of the sixth step (its mechanical yield) is then determined as the product of the three partial yields:

$$\begin{cases} \varepsilon_{78}^{a.e.} = \frac{1+tg^2\beta}{2\cdot\pi}\cdot\left\{\sqrt{\left[\left(z_7+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_7+\cos\beta\right)}\right. \\ \left.+\sqrt{\left[\left(z_8+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_8+\cos\beta\right)}-\left(z_7+z_8\right)\cdot tg\alpha_0\right\} \\ \eta_{78} = \frac{z_7^2\cdot\cos^2\beta}{z_7^2(tg^2\alpha_0+\cos^2\beta)+\frac{2}{3}\pi^2\cos^4\beta(\varepsilon-1)(2\varepsilon-1)\pm 2\pi tg\alpha_0 z_7\cos^2\beta(\varepsilon-1)} \\ \varepsilon_{14,15}^{a.e.} = \frac{1+tg^2\beta}{2\cdot\pi}\cdot\left\{\sqrt{\left[\left(z_{14}+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot 2\left(z_{14}+\cos\beta\right)}\right. \\ \left.+\sqrt{\left[\left(z_{15}+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_{15}+\cos\beta\right)}-\left(z_{14}+z_{15}\right)\cdot tg\alpha_0\right\} \\ \eta_{14,15} = \frac{z_{14}^2\cdot\cos^2\beta}{z_{14}^2(tg^2\alpha_0+\cos^2\beta)+\frac{2}{3}\pi^2\cos^4\beta(\varepsilon-1)(2\varepsilon-1)\pm 2\pi tg\alpha_0 z_{14}\cos^2\beta(\varepsilon-1)} \\ \varepsilon_{15,6}^{a.e.} = \frac{1+tg^2\beta}{2\cdot\pi}\cdot\left\{\sqrt{\left[\left(z_{15}+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot 25\left(z_{15}+\cos\beta\right)}+\right. \\ \left.+\sqrt{\left[\left(z_6+2\cdot\cos\beta\right)\cdot tg\alpha_0\right]^2+4\cdot\cos^3\beta\cdot\left(z_6+\cos\beta\right)}-\left(z_{15}+z_6\right)\cdot tg\alpha_0\right\} \\ \eta_{15,6} = \frac{z_{15}^2\cdot\cos^2\beta}{z_{15}^2(tg^2\alpha_0+\cos^2\beta)+\frac{2}{3}\pi^2\cos^4\beta(\varepsilon-1)(2\varepsilon-1)\pm 2\pi tg\alpha_0 z_{15}\cos^2\beta(\varepsilon-1)} \\ \eta_{VI} \equiv \eta_{MR} \equiv \eta_{76} = \eta_{78}\cdot\eta_{14,15}\cdot\eta_{15,6} \end{cases} \tag{15}$$

Conclusion

Since the 1970s, this initial solution has been quickly replaced with a top one with all the wheels (including those of the output shaft) designed with inclined or curved teeth, the output shaft wheels being normally free on their secondary (output shaft), i.e., rotating freely on it and being rotatably coupled to the secondary shaft, in turn, by means of syncrons, which are smaller toothed wheels with straight teeth permanently rotating with the secondary shaft and which can to balancing to engage with the respective gear, thus enabling it to (selectively) engage the secondary shaft in rotation. Such a solution, also used in trucks and buses, is shown in Fig. 2, the classic gearbox with synchronous representing the best solution that existed and which is still the most used today, even if the boxes appeared, refined and multiplied hybrid or automatic gears.

Today, various types of gearboxes have been introduced into the vehicles to change the way the classic manual gearboxes work, such as automatic gearboxes, semi-automatic, continuous variable, dual-clutch automatic gearboxes etc. However, most gear shifters for in-service vehicles are still classical manuals, which is why their optimal synthesis based on their dynamics and especially on optimal performance is now more than necessary. 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 classic, mechanical, manual gearshift can be achieved regardless of its operating status.

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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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