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Configuration design of ten-speed automatic transmissions with twelve-link three-DOF Lepelletier gear mechanism†

Essam L. Esmail*

College of Engineering, University of Qadisiyah, Al Diwaniyah, Iraq

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Abstract

Many Lepelletier-type automatic transmissions are in production. Some of Lepelletier related configurations are still far from reaching maximum possible sequential speed ratios. This work focuses on analyzing and developing twelve-link three-DOF Lepelletier-type automatic transmissions to attain maximum speed ratios. Following the general trend of increased shift stages and a wider range of speed ratios, new ten-speed automatic transmissions are enumerated from the twelve-link three-DOF Lepelletier gear mechanisms. Furthermore, a ten-speed automatic transmission is modified from an existing eight-speed Lepelletier-type automatic transmission. The results show that the twelve-link three-DOF Lepelletier gear mechanism could reach eleven forward speeds at most. Nomographs are shown to be practical in detecting possible configurations and their clutching sequences.

Keywords: Automatic transmission; Clutching-sequences; Eleven-speed; Planetary gear train; Gear ratios; Lepelletier; Nomographs; Ten-speed; Speed ratios

1. Introduction

Automatic transmissions with planetary gear trains have been used in the automotive industry for a long time. Famous companies have proposed several kinds of multi-speed automatic transmissions. However, they always treat the development of the automatic transmissions as commercial secrets. Since research and development documents are mostly classified, information can only be obtained from patent documents, commercial catalogs, and non-academic magazines. The literature on planetary gear trains includes kinematic analysis [1-5], dynamic analysis [6], power flow and efficiency analysis [7-9], conceptual and configuration designs [10-21].

In 1992, Lepelletier [22] proposed three-DOF epicyclic gear mechanisms. These are now called Lepelletier gear mechanisms. In 2001, ZF used this gear set design to bring the first 6-speed passenger car transmission 6 HP 26 on the market [23].

The enumeration of clutching sequences has been the subject of several academic studies for the past several years. The selection of an optimal clutching sequence cannot be solved analytically. Nadel et al. [24-26] formulated the task as a constraint satisfaction problem. Hsieh and Tsai [13, 15], Hwang and Huang [16] and Hsu and Huang [17, 18] used algorithmic techniques. Ross and Route [10] and Esmail [19] introduced graphic techniques. Hattori et al. [27] used phase geometry method. Refs. [10, 16, 26-28] are restricted to constructing clutching sequences for specific types of automatic transmissions.

Hsieh and Tsai [15] used the concept of Fundamental gear entities (FGEs) proposed by Chatterjee and Tsai [12] in conjunction with their earlier kinematic study [2] to determine the most efficient clutching sequence associated with automatic transmissions [15]. A computer algorithm for the enumeration of clutching sequences is given by Hsieh [14]. Nevertheless, Hsieh and Tsai [15] did not develop an effective method for arranging clutching sequences. However, the elimination of invalid clutching sequences was conducted by inspection.

Esmail [19] proposed a methodology, based on nomographs, for the enumeration of clutching sequences of planetary gear mechanisms. This method simplifies the synthesis of the clutching sequence of a planetary gear mechanism efficiently.

Hsu and Huang [18] synthesized the number of teeth of all gears of six speed Ravigneaux-type automatic transmissions by assigning three (out of seven) desired speed ratios in the analytic method presented by Hsu [29]. The method presented by Hsu becomes a trial and error method when applied to the six-speed Ravigneaux-type automatic transmission: a method of reaching satisfactory results by trying out certain desired speed ratios until other speed ratios and the design constraints are satisfied. It is lengthy and tedious. They also proposed a planar-graph representation to arrange the desired clutches for

*Corresponding author. Tel.: +96 47804342187
E-mail address: dr.essamesmail@yahoo.com
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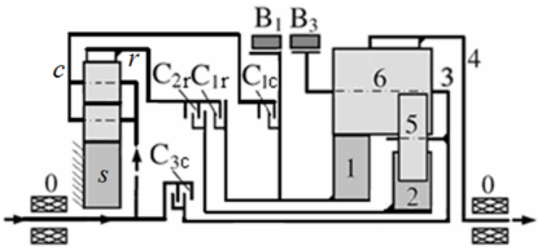


Fig. 1. Toyota 8-speed automatic transmission with double-planet, input GTE [31].

each possible clutching sequence into the Ravigneaux gear mechanism. Hwang and Huang [16] introduced coded sketches for connecting clutch elements to planetary gear trains for automotive automatic transmissions.

The literature survey reveals that although several studies have been made on the clutching sequence synthesis, they are tedious and focus on the use of optimization method [30]. A simple methodology for the design of Lepelletier-type planetary automatic transmissions needs to be developed. Furthermore, the practicality of ten-speed Lepelletier gear mechanisms has not been investigated. Therefore, the purpose of this paper is to present a simple methodology for the design of all feasible and practical automatic transmissions based on the twelve-link three-DOF Lepelletier gear mechanism with the aid of nomography.

2. Lepelletier-type automatic transmission

Fig. 1 shows the Lepelletier automatic transmission [31] which can provide eight forward speeds and one reverse speed. Table 1 shows the corresponding clutching sequence. An examination of existing Lepelletier automatic transmissions reveals that all of them adopt seven-link two-DOF Ravigneaux gear mechanism integrated with a single- or double-planet, simple Gear train entity (GTE) to form three-DOF planetary gear mechanisms. Almost all the automotive manufacturers have developed the three-DOF Lepelletier gear mechanism as six- or eight-speed automatic transmission such as Audi and Toyota [31-34].

In Fig. 1, the carrier, link 3, can be clutched either to the input shaft by a rotating clutch, C_3, or to the casing of the transmission by a brake, B_3. Similarly, the large sun gear, link 1, can be clutched either to the input shaft of the simple GTE by rotating clutch, C_{1c}, or to the casing by a brake, B_1. The large sun gear can also be clutched to the output shaft of the ring gear of the double-planet simple GTE by rotating clutch, C_{1r}. The small sun gear, link 2, can be clutched to the output shaft of the simple GTE by rotating clutch, C_2. The carrier, link 4, which is permanently attached to the final reduction unit, is designated as the output of the gear train. Four multi-plate clutches, two multi-disc brakes and a free-wheel (not shown in the Figure) are used to operate the eight forward speeds and one reverse drive.

In Table 1, the ranges of output velocities are classified into Under drive (UD), Over drive (OD), Direct drive (DD), and Reverse drive (RD), according to whether the speed is between zero and the input speed, more than the input speed, equal to the input velocity, or less than zero.

Generally, only coaxial links can be selected as the input link, the output link, the common link or the ground link to obtain the desired speed ratios. To obtain an automatic transmission consisting of more than eight speeds, three-DOF Lepelletier gear mechanisms having twelve-links, are considered. The two-ring eight-link Ravigneaux gear mechanisms, are shown in Fig. 3.

Lepelletier PGTs are analyzed as a compound of two subsystems. The first subsystem is an input set of one degree of

Table 1. Clutching sequence for Eight-speed Lepelletier automatic transmission [31].

	C_{1r}	C_{1c}	C_2	C_3	B_1	B_3
UD_1			X			X
UD_2			X		X	
UD_3	X		X			
UD_4		X	X			
UD_5			X	X		
DD		X		X		
OD_1	X			X		
OD_2				X	X	
RD	X					X

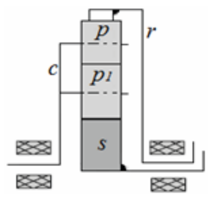
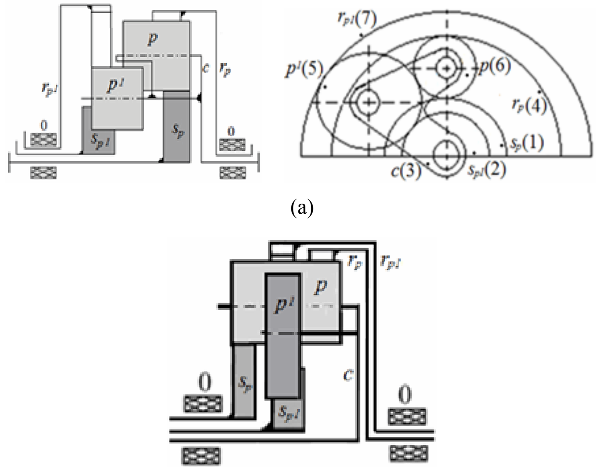


Fig. 2. Double-planet simple GTE.



(a)

(b)

Fig. 3. Eight-link Ravigneaux gear mechanisms.

freedom given by a single- or double-planet simple GTE supplying two angular velocities. The second subsystem is an output set of two degrees of freedom given by a Ravigneaux GTE [35-38]. The simple GTE, shown in Fig. 2, consists of a sun gear, a ring gear, planet gear(s), and a carrier. The eight-link two-DOF Ravigneaux gear mechanism, shown in Fig. 3, consist of gearbox casing (0), sun gear s_p(1), sun gear s_p^1(2), carrier c(3), ring gear r_p(4), planet gear p^1(5), planet gear p(6), and ring gear r_p^1 (7).

This work, as an extension to previous works [21, 38], focuses on ten-speed automatic transmissions with twelve-link three-DOF Lepelletier gear mechanisms.

The objectives of this paper are to

- Identify the speed ratio characteristics of Lepelletier gear mechanisms, and consequently simplifying their kinematic analysis.
- Develop and analyze twelve-link three-DOF Lepelletier gear mechanisms.
- Develop an efficient method for the enumeration of all feasible and practical automatic transmissions based on the twelve-link three-DOF Lepelletier gear mechanism.
- Enumerate new ten-speed Lepelletier automatic transmissions from twelve-link three-DOF Lepelletier gear mechanisms.
- Modify an existing eight-speed Lepelletier automatic transmission to add two more forward speeds.
- Enumerate a novel eleven-speed Lepelletier automatic transmission from twelve-link three-DOF Lepelletier gear mechanism.

Nomographs are found practical in detecting possible configurations and clutching sequences without knowing the exact dimensions of the gears.

3. Synthesis of kinematic nomographs

The kinematic design, and the clutching sequence arrangement are two of the most important factors in transmission design . Since the Lepelletier gear mechanism can be decomposed into simple and Ravigneaux GTEs, the kinematics of the Lepelletier gear mechanism are closely related to the kinematics of its individual GTEs. Since a GTE is composed of several ring gears, sun gears, planets, and one carrier, it is possible to estimate the speed ratios without specifying the exact size of each gear.

The term "gear ratio" ($N_{p,x}$) is used to denote the speed ratio of a meshing gear pair, while "speed ratio" is used to denote the speed ratio between the input link and the output link of an Epicyclic gear mechanism (EGM). Let p and x be a gear pair and c be a carrier, then links p, x and c form a GPE. The GPE equation can be written as

$$N_{p,x} = \pm\frac{Z_p}{Z_x} = \frac{\omega_x - \omega_c}{\omega_p - \omega_c} \tag{1}$$

where ω_p , ω_x and ω_c denote the speeds of a gear pair p and x, and its carrier c, respectively, $N_{p,x}$ represent the planet gear ratio between gears p and x, and Z_p and Z_x denote the number of teeth on gears p and x, respectively, with the plus sign corresponding to the internal gear pair and minus sign to the external gear pair, respectively.

A nomograph is a graph containing several scales graduated for different variables such that an intersecting straight line enables related values to be read off. Planetary gear trains have two degrees of freedom, which means that the rotations of two of the elements must be specified to completely define the motion of the gear system.

The kinematic nomograph for double-planet Ravigneaux gear train, shown in Fig. 3, will be constructed.

Eq. (1) can be rewritten for the links of the double -planet GTE as follows

$$N_{p,c} = \frac{\omega_c - \omega_c}{\omega_p - \omega_c} = 0\,, \tag{2}$$

$$N_{p,p} = \frac{\omega_p - \omega_c}{\omega_p - \omega_c} = 1\,, \tag{3}$$

$$N_{p,r_p} = \frac{Z_p}{Z_{r_p}}\,, \tag{4}$$

$$N_{p,s_p} = -\frac{Z_p}{Z_{s_p}}\,, \tag{5}$$

$$N_{p,p^1} = -\frac{Z_p}{Z_{p^1}}\,, \tag{6}$$

$$N_{p^1,s_{p^1}} = -\frac{Z_{p^1}}{Z_{s_{p^1}}}\,, \tag{7}$$

$$N_{p^1,r_{p^1}} = \frac{Z_{p^1}}{Z_{r_{p^1}}}\,. \tag{8}$$

In double planet FGE, the planet gear ratio for the gears that are not meshing directly with the planet link on which the gear ratio is calculated (p), and are meshing with the other planet link (p^1), can be found in terms of the planet gear ratio of the two planets ($N_{p,p1}$) as

$$N_{p,s_{p^1}} = N_{p,p^1} \cdot N_{p^1,s_{p^1}} = \frac{Z_p}{Z_{s_{p^1}}}\,, \tag{9}$$

$$N_{p,r_{p^1}} = N_{p,p^1} \cdot N_{p^1,r_{p^1}} = -\frac{Z_p}{Z_{r_{p^1}}}\,. \tag{10}$$

Fig. 4 shows the basic nomograph to be created, where the axes labeled ω_{rp}, ω_{rp1}, ω_{sp}, ω_{sp1}, ω_p, ω_{p1}, and ω_c represent the rotational speed of the ring gear r_p , ring gear r_{p1}, sun gear s_p, sun gear s_{p1}, planet gear p, planet gear $p1$ and the planet gear carrier c. The ω_c axis is placed at the origin. The distance between the ω_c axis and the ω_p axis is set equal to 1. The ring gear r_p is always larger than the planet gear so that the value of

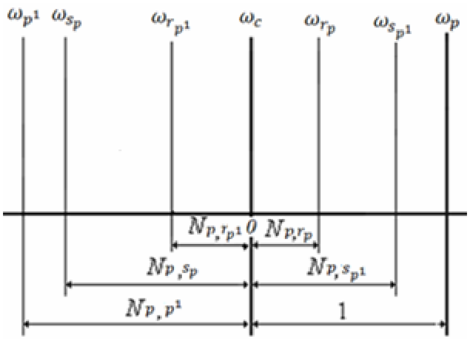


Fig. 4. Basic nomograph for the double-planet GTE shown in Fig. 3, in terms of planet gear p.

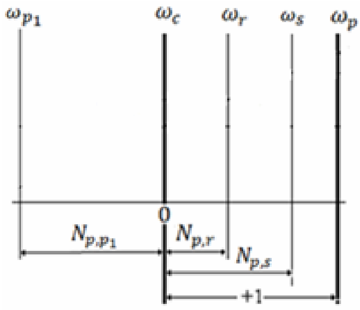


Fig. 5. Nomograph for the double-planet gear train entity shown in Fig. 2.

$N_{p,r}$ is smaller than one but larger than zero. The location of the ω_r axis is actually between the ω_c and the ω_p axes and the distance between the ω_c axis and the ω_r axis is set equal to $N_{p,r}$. Since the value of the first planet gear to the second planet gear ratio $N_{p,p1}$, is always smaller than zero, the location of the ω_{p1} axis is actually be to the left of the ω_c axis and the distance between the ω_c axis and the ω_{p1} axis is set equal to $N_{p,p1}$. Since the value of $N_{p,sp}$ is always smaller than zero, the location of the ω_{sp} axis is actually be to the left of the ω_c axis and the distance between the ω_c axis and the ω_{sp} axis is set equal to $N_{p,sp}$.

The ring gear r_p is always larger than the sun gear s_{p1}, $Z_{rp} > Z_{sp1}$ and $(1/Z_{rp}) < (1/Z_{sp1})$ then $(Z_p/Z_{rp}) < (Z_p/Z_{sp1})$ or $N_{p,rp} < N_{p,sp1}$, which implies that the value of $N_{p,rp}$ is always smaller than $N_{p,sp1}$. The location of the ω_{sp1} axis is actually to the right of the ω_{rp} axis, and the distance between the ω_c axis and the ω_{sp1} axis is set equal to $N_{p,sp1}$. The ring gear r_{p1} is always larger than the sun gear s_p, $Z_{rp1}>Z_{sp}$ and $(1/Z_{rp1})<(1/Z_{sp})$ then $(Z_p/Z_{rp1})<(Z_p/Z_{sp})$ and $(-Z_p/Z_{rp1})>(-Z_p/Z_{sp})$ or $N_{p,rp1}>N_{p,sp}$, which implies that the value of $N_{p,rp1}$ is always larger than $N_{p,sp}$. The location of the ω_{rp1} axis is actually be to the left of the ω_p axis. Its location is actually between the ω_{sp} axis and the ω_p axis and the distance between the ω_c axis and the ω_{rp1} axis equal $N_{p,rp1}$ as shown in Fig. 4.

Fig. 5 shows the nomograph for the double-planet gear train entity shown in Fig. 2.

Let $R^z_{x,y}$ denote the velocity ratio between links x and y with reference to link z, where x, y and z are any three links in the gear train.

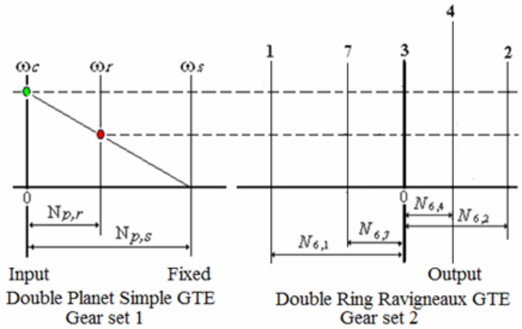


Fig. 6. Clutching sequence nomograph for the twelve-link Lepelletier gear mechanism.

Applying Eq. (2) twice for links *y* and *z* and after simplification yields:

$$R^z_{x,y} = \frac{\omega_x - \omega_z}{\omega_y - \omega_z} = \frac{N_{p,x} - N_{p,z}}{N_{p,y} - N_{p,z}} . \quad (11)$$

If the *x* and *z* links are exchanged, then the two speed ratios are related by

$$\frac{1}{R^z_{x,y}} = 1 - \frac{1}{R^x_{z,y}} . \quad (12)$$

Typically, a rotating clutch is used for connecting two rotating links and a brake is used to fix a link to the casing. The rotating clutches and brakes are denoted by C and B, respectively.

In arranging a clutching sequence, it is highly desirable to achieve a single-shift transition, i.e., only one clutch is turned on while another is simultaneously turned off between two successive speed ratios.

An EGM contains several coaxial links: one coaxial link (called the input link) is clutched to the power source, another coaxial link (called the fixed link) is connected to the casing, a third coaxial link (called the output link) is attached to the final reduction unit, while several links belonging to different GTEs (called the common links) are connected to the casing or clutched to the corresponding coaxial links. An EGM can, therefore, provide several speed ratios depending on clutching conditions.

4. Synthesis of clutching sequences

Kinematic inversion is defined as the process of selecting the input, output, fixed and common links for the Lepelletier gear mechanism. By applying kinematic inversion, the speed ratios of Lepelletier gear mechanism can be identified without specifying the gear sizes, and a descending sequence of clutching conditions can be roughly configured. This will effectively facilitate the enumeration of all the feasible clutching sequences of the Lepelletier gear mechanism [39]. When a

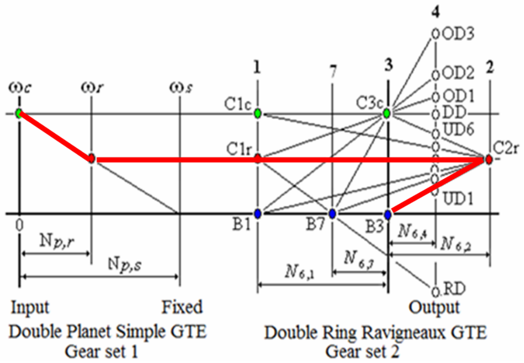


Fig. 7. Clutching sequence nomograph for the 10-speed automatic transmission with 12-link Lepelletier gear mechanism.

double-planet simple GTE is used with an eight-link Ravigneaux gear train, the clutching sequence nomograph, shown in Fig. 6, can be obtained.

The output shaft of the torque converter is the input shaft for the simple planetary gear set with the sun gear splined to the pump stator. The torque converter is obviously driven by the engine. From the simple planetary gear set, the power is transferred to the Ravigneaux gear set.

Ring gears 4 and 7 can be selected as the output links. Because of their symmetrical structure, both design results will be the same. The ring gear 4 of the Ravigneaux gear set is selected as the output link to the final drive of the transmission.

In this configuration gear sets 1 and 2 are used in series.

The sun gear of gear set 1 is grounded, which leads the ring gear to rotate slower than the carrier; therefore, carrier c of gear set 1 is selected as the input to the Lepelletier gear mechanism.

Since the sun gear of gear set 1 is grounded, the carrier is the input and the ring gear is the output, the ratio of this arrangement is $R^{s}_{c,r} = \dfrac{N_{p,c} - N_{p,s}}{N_{p,r} - N_{p,s}} = \dfrac{Z_r}{Z_r - Z_s}$, i.e., the ring gear is rotating $\dfrac{Z_r - Z_s}{Z_r}$ times slower than the carrier.

The most interesting peculiarity of the Lepelletier gear mechanism is that the Ravigneaux sun gears and planet carrier can be driven at two different rotational speeds; one of them is the output rotational speed of the double-planet simple GTE (gear set 1). These two different input speeds create a large number of possible speed ratios.

The two gear sets must share two common links so that the overall speed ratio can be expressed in terms of the speed ratios of the two gear sets. The common links transmit power from gear set 1 to gear set 2. One of the common links may be fixed link. The overall speed ratio can be expressed as a product of two speed ratios as:

$$R^{z}_{x,y} = \frac{\omega_x - \omega_z}{\omega_y - \omega_z} = \frac{\omega_x - \omega_z}{\omega_b - \omega_z} \times \frac{\omega_b - \omega_z}{\omega_y - \omega_z} = R^{z}_{x,b} \times R^{z}_{b,y}\,, \tag{13}$$

where b and z are the two common links and $R^{z}_{x,b}$ is associated with gear set 1 and $R^{z}_{b,y}$ is associated with gear set 2.

4.1 First under-drive

The carrier c of gear set 1 is connected to the input, and the sun gear s is grounded, which leads the ring gear r to rotate slower than the input.

The ring gear r of gear set 1 is in turn connected to the small sun gear *2* of gear set 2, while the carrier *3* for gear set 2 is grounded, causing the output ring gear *4* of gear set 2 to be driven at a first under driven. The first under-drive power flow is shown in red on Fig. 7. Applying Eqs. (13) and (11) in turn yields

$$UD_1 = R^{s}_{c,r} \times R^{3}_{2,4} = \frac{Z_r}{Z_r - Z_s} \cdot \frac{Z_4}{Z_2}\,. \tag{14}$$

First under-drive is achieved by connecting the ring gear r of gear set 1 and the small sun gear *2* of gear set 2 by a rotating clutch C_2, and by engaging carrier *3* of gear set 2 to ground by a freewheel F_3. On the nomograph, the lines are labeled with a B or C to denote brake or clutch. Such a nomograph can also help in realizing the maximum number of feasible reductions and overdrives in a clutching sequence.

4.2 Second under-drive

Operation of gear set 1 is identical to the first under-drive; the carrier c of gear set 1 is connected to the input, the ring gear r is the output of gear set 1 and the ring gear r of gear set 1 in turn, is connected to the small sun gear 2 of gear set 2. The ring gear *7* for gear set 2 is grounded, causing the output ring gear *4* of gear set 2 to rotate in a second under driven. The speed ratio is:

$$UD_2 = R^{s}_{c,r} \times R^{7}_{2,4} = \frac{Z_r}{Z_r - Z_s} \cdot \frac{\dfrac{1}{Z_2} + \dfrac{1}{Z_7}}{\dfrac{1}{Z_4} + \dfrac{1}{Z_7}}\,. \tag{15}$$

The second under-drive is achieved by disengaging carrier *3* and engaging ring gear 7 of gear set 2 to ground by a brake B_7.

4.3 Third under-drive

The 3rd under-drive is achieved by releasing brake B_7 and grounding the large sun gear 1 of gear set 2 by a brake B_1. The speed ratio is:

$$UD_3 = R^{s}_{c,r} \times R^{1}_{2,4} = \frac{Z_r}{Z_r - Z_s} \cdot \frac{\dfrac{1}{Z_2} + \dfrac{1}{Z_1}}{\dfrac{1}{Z_4} + \dfrac{1}{Z_1}}\,. \tag{16}$$

4.4 Fourth under-drive

The upshift to fourth under-drive is accomplished by releasing brake B_7 and engaging the large sun gear 1 of gear set 2 to the ring gear r of gear set 1. Consequently, the two sun gears of gear set 2 are connected to the ring gear r of gear set 1, i.e., the Ravigneaux gear train will act as a rigid body and ring gear 4 of gear set 2 which is the transmission output, rotates at the same speed as ring gear r of gear set 1. The Ravigneaux speed ratio equals one, and the overall speed ratio is:

$$UD_4 = R^s_{c,r} \times 1 = \frac{Z_r}{Z_r - Z_s}. \tag{17}$$

The fourth under-drive is achieved by disengaging brake B_7 and engaging sun gear 1 of gear set 2 to ring gear *r* of gear set 1 by a rotating clutch C_{1r}.

4.5 Fifth under-drive

By releasing clutch C_{1r} and engaging the large sun gear 1 of gear set 2 to the carrier c of gear set 1, the transmission shifts up to the 5th under drive. First, the speed ratio of an inverted mechanism, for which the input and fixed links are interchanged, is obtained.

$$R^1_{s,4} = R^c_{s,r} \times R^1_{2,4} = \frac{Z_r}{Z_s} \cdot \frac{\frac{1}{Z_2} + \frac{1}{Z_1}}{\frac{1}{Z_4} + \frac{1}{Z_1}}. \tag{18}$$

Then, applying Eq. (12) to derive the overall speed ratio

$$UD_5 = R^s_{1,4} = \frac{1}{1 - \frac{1}{\frac{Z_r}{Z_s} \cdot \frac{\frac{1}{Z_2} + \frac{1}{Z_1}}{\frac{1}{Z_4} + \frac{1}{Z_1}}}}. \tag{19}$$

The fifth under-drive is achieved by disengaging clutch C_{1r} and engaging the large sun gear 1 of gear set 2 to the carrier c of gear set 1 by a rotating clutch C_{1c}.

4.6 Sixth under-drive

By releasing clutch C_{1c} and engaging the carrier 3 of gear set 2 to the carrier *c* of gear set 1, the transmission shifts up to the 6th under drive. First, the speed ratio of an inverted mechanism is obtained

$$R^3_{s,4} = R^c_{s,r} \times R^3_{2,4} = \frac{Z_r}{Z_s} \cdot \frac{Z_4}{Z_2}. \tag{20}$$

Then, applying Eq. (12) to derive the overall speed ratio

$$UD_6 = R^s_{3,4} = \frac{1}{1 - \frac{1}{\frac{Z_r}{Z_s} \cdot \frac{Z_4}{Z_2}}}. \tag{21}$$

The sixth under-drive is achieved by disengaging clutch C_{1c} and engaging the carrier 3 of gear set 2 to the carrier c of gear set 1 by a rotating clutch C_3.

At this point, the six speed ratios have been achieved by leaving clutch C_2 engaged, and cycling through the six friction elements F_3, B_7, B_1, C_{1r}, C_{1c}, and C_3.

4.7 Direct drive

The direct drive is achieved by releasing clutch C_2 and engaging clutch C_{1c}.

By simultaneously engaging clutches C_{1c} and C_3 of gear set 2 to carrier c of gear set 1, gear set 2 acts as a rigid body and ring gear 4 of gear set 2 which is the transmission output, rotates at the same speed of carrier *c* of gear set 1. The overall speed ratio is therefore quite simply equals 1.

4.8 First over-drive

Up shift to first over-drive is accomplished by releasing clutch C_{1c} and engaging clutch C_{1r}. First the speed ratio of an inverted mechanism is obtained.

$$R^3_{s,4} = R^c_{s,r} \times R^3_{1,4} = \frac{Z_r}{Z_s} \cdot \frac{Z_4}{Z_1}. \tag{22}$$

Then, applying Eq. (12) to derive the overall speed ratio

$$OD_1 = R^s_{3,4} = \frac{1}{1 - \frac{1}{\frac{Z_r}{Z_s} \cdot \frac{Z_4}{Z_1}}}. \tag{23}$$

4.9 Second over-drive

Up shift to second over-drive is accomplished by releasing clutch C_{1r} and engaging brake B_1. The engagement of brake B_1 grounds the large sun gear 1 of gear set 2, while clutch C_3 connects the carrier 3 of gear set 2 to the carrier *c* of gear set 1, which is the input shaft. Since C_3 is engaged, carrier 3 of gear set 2 turns at the input speed. This means that the total speed ratio is decided only by the speed ratio of gear set 2. The speed ratio is

$$OD_2 = R^1_{3,4} = \frac{N_{6,3} - N_{6,1}}{N_{6,4} - N_{6,1}} = \frac{\frac{1}{Z_1}}{\frac{1}{Z_4} + \frac{1}{Z_1}}. \tag{24}$$

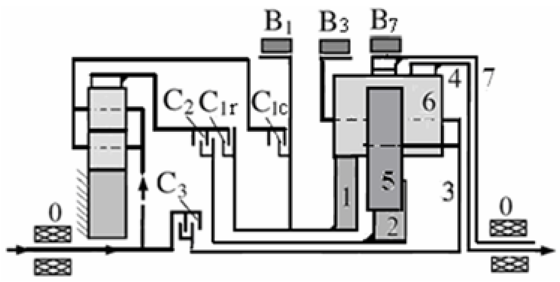


Fig. 8. The new ten-speed automatic transmission.

4.10 Third over-drive

The upshift to third over-drive is accomplished by releasing brake B_1 and engaging brake B_7. The speed ratio is

$$OD_3 = R^1_{3,7} = \frac{N_{6,3} - N_{6,1}}{N_{6,7} - N_{6,1}} = \frac{\dfrac{1}{Z_1}}{\dfrac{1}{Z_7} + \dfrac{1}{Z_1}} . \tag{25}$$

Finally, a reverse drive is added to the forward clutching sequence. By combining the forward clutching sequences with the four RDs ($C_{1r}B_3$, $C_{1r}B_7$, $C_{1c}B_3$ and $C_{1c}B_7$), four possible clutching sequences of ten-speed automatic transmissions are feasible.

4.11 First reverse drive

A first reverse drive is achieved by locking brake B_3, and engaging clutch C_{1r}. The speed ratio is

$$RD_1 = R^s_{c,r} \cdot R^3_{1,4} = \frac{Z_r}{Z_r - Z_s}\left(-\frac{Z_4}{Z_1}\right). \tag{26}$$

4.12 Second reverse drive

A second reverse drive is achieved by locking brake B_7, and engaging clutch C_{1r}. The speed ratio is

$$RD_2 = R^s_{c,r} \cdot R^7_{1,4} = \frac{Z_r}{Z_r - Z_s}\left(\frac{-\dfrac{1}{Z_1} + \dfrac{1}{Z_7}}{\dfrac{1}{Z_4} + \dfrac{1}{Z_7}}\right). \tag{27}$$

The nomograph shown in Fig. 7 is converted back into the corresponding schematic diagram of an automatic transmission, as shown in Fig. 8, by inserting the gear trains and the embodiments of links, joints, brakes and clutches.

This novel configuration uses similar basic components as the Toyota 8-speed automatic transmission shown in Fig. 1, and has added two additional forward speeds. The main difference is that the new transmission uses the Lepelletier gear mechanism with two-ring Ravigneaux gear train. Also, this design is likely very competitive in terms of shifting elements and speed ratios for a given number of links. Only four multi-plate-clutches, three multi-disc brakes are used to operate the new ten forward speeds and one reverse speed automatic transmission. Table 2 shows the clutching sequence for the new ten-speed automatic transmission concept with double-planet simple GTE and ($C_{1r}B_7$) as a reverse drive.

Table 2. Clutching sequence table for the new ten-speed automatic transmission with ($C_{1r}B_7$) as a reverse drive.

	C_{1r}	C_{2r}	C_{3c}	C_{1c}	B_1	B_7	F_3
UD_1		X					X
UD_2		X				X	
UD_3		X			X		
UD_4	X	X					
UD_5		X		X			
UD_6		X	X				
DD			X	X			
OD_1	X		X				
OD_2			X		X		
OD_3			X			X	
RD	X					X	

Table 3. Clutching sequence table for eleven-speed automatic transmission with ($C_{1r}B_3$) as reverse drive.

	C_{1r}	C_{1c}	C_{2r}	C_{3c}	C_{7c}	B_1	B_3	B_7
UD_1			X				X	
UD_2			X					X
UD_3			X			X		
UD_4	X		X					
UD_5		X	X					
UD_6			X		X			
UD_7			X	X				
DD				X	X			
OD_1	X			X				
OD_2				X		X		
OD_3				X				X
RD	X						X	

Fig. 9 shows the new ten-speed automatic transmission that is modified from Audi 8-speed automatic transmission.

5. Feasible clutching sequences for eleven-speed automatic transmissions with twelve-link three-dof lepelletier gear mechanism

As mentioned before, Lepelletier PGTs can be analyzed as a compound of a double-planet GTE and a two-ring Ravigneaux GTE. Fig. 10 shows a clutching sequence nomograph for these two known topologies.

According to the nomograph results, the UD and OD

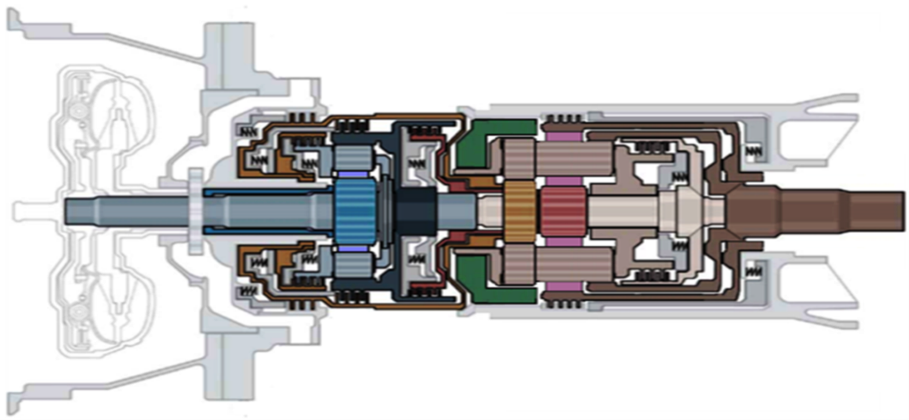


Fig. 9. New ten-speed automatic transmission adapted from Audi 8-speed automatic transmission.

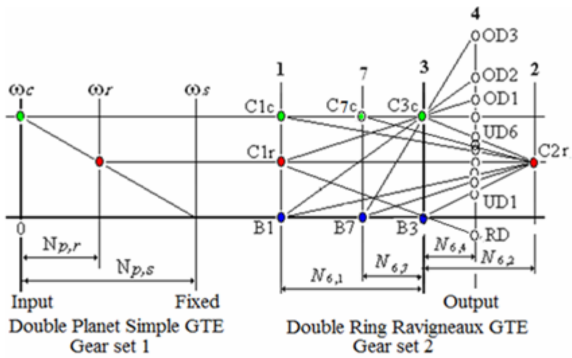


Fig. 10. Clutching sequence nomograph for the 11-speed automatic transmission.

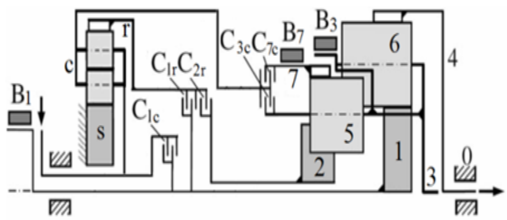


Fig. 11. Eleven-speed automatic transmission concept.

clutching sequences are given in Table 3. By combining this set of clutching sequences with the five RDs ($C_{7c}B_3$, $C_{1c}B_3$, $C_{1c}B_7$, $C_{1r}B_3$, and $C_{1r}B_7$) five possible clutching sequences of eleven-speed automatic transmissions are feasible.

As the associated clutching sequence is synthesized, all the rotating clutches and brakes are then arranged between the simple and Ravigneaux gear mechanisms to obtain an 11-speed Lepelletier automatic transmission. For this clutching sequence, the clutches include five rotating clutches C_{1r}, C_{1c}, C_{2r}, C_{3c} and C_{7c} and three brakes B_1, B_3 and B_7, which are arranged in Fig. 11 to form 11-speed transmission. In this transmission, five RDs ($C_{1r}B_3$, $C_{1r}B_7$, $C_{1c}B_3$, $C_{1c}B_7$ and $C_{7c}B_3$) are possible.

To reduce the number of rotating clutches to four, one of the rotating clutches (C_{1c} or C_{7c}), which produces only one velocity ratio, is considered each time, and thus two 10-speed Lepelletier automatic transmissions are obtained. For the first clutching sequence UD_1, UD_2, UD_3, UD_4, UD_5, UD_6, DD, OD_1, OD_2 and OD_3 shown in Table 3, the highest under-drive (UD_7) is eliminated and the desired shifting elements include four rotating clutches (C_{1r}, C_{3c}, C_{2r} and C_{7c}) and three brakes (B_1, B_3, and B_7). In this transmission, three RDs ($C_{1r}B_3$, $C_{1r}B_7$ and $C_{7c}B_3$) are possible.

For the second clutching sequence UD_1, UD_2, UD_3, UD_4, UD_5, UD_7(will become UD_6) , DD, OD_1, OD_2 and OD_3 shown in Table 3, the second highest under-drive (UD_6) is eliminated and the desired shifting elements include four rotating clutches (C_{1r}, C_{1c}, C_{2r} and C_{3c}) and three brakes (B_1, B_3, and B_7). In this transmission, four RDs ($C_{1r}B_3$, $C_{1r}B_7$, $C_{1c}B_3$ and $C_{1c}B_7$) are possible.

The methodologies developed in this paper contribute to the early phase of the design process. The next phase of the design process should address the following issues:

- Find the optimal gear ratios of three-DOF twelve-link Lepelletier gear mechanism to achieve a set of speed ratios.
- Determine the dimensions of the gears of the ten-speed Lepelletier-type automatic transmission especially for that modified from the Audi 8-speed automatic transmission.
- Perform the design optimization by simultaneously considering the speed ratios, geometric constraints and design constraints.

6. Conclusions

This work is concerned with the development of a systematic methodology for the enumeration of feasible clutching sequences associated with Lepelletier-type gear mechanism. Eight configurations of ten-speed automatic transmissions have been synthesized from the twelve-link Lepelletier gear mechanism. A novel eleven-speed automatic transmission has been also synthesized from the twelve-link Lepelletier gear mechanism.

It appears to be a major breakthrough in the design of eleven-speed automatic transmission, because the Lepelletier gear mechanism had only twelve links.

Nomographs are found practical in detecting possible configurations and clutching sequences without knowing the exact dimensions of the gears. Design and analysis techniques presented here, although developed for a specific system arrangement, are generally applicable to all types of compound epicyclic gear mechanisms where maximum sequential clutching sequence is intended.

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Essam Lauibi Esmail received the B.S. and M.S. from the Department of Mechanical Engineering, Baghdad University, Baghdad, Iraq, in 1980 and 1985, respectively. His Ph.D. in Mechanical Engineering is from the University of Technology in 2009. He is currently an assistant professor in the Department of Mechanical Engineering, University of Qadisiyah. His research interests include mechanisms, machine theory, design methodology, automotive engineering, hybrid transmissions, robot manipulators, fuzzy logic control, and genetic algorithm optimization.