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Reduced Length Eight_Velocity Automatic Transmissions

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Abstract

Although there are many epicyclic-type automatic transmissions in production, some of the related configurations are still far from attaining maximum sequential velocity ratios. This work in part will attempt to attain the *maximum possible velocity ratios* for any given epicyclic gear train.

A methodology for the design of *two-ring* Ravigneaux-type epicyclic gear transmissions for automobiles is presented. First, based on the kinematic nomographs of the corresponding basic gear ratios, clutching-sequences are enumerated. Second, a planar-graph representation is used to arrange the desired clutches for each possible clutching sequence into the epicyclic gear mechanism. Then, with the above methods, the designs of the epicylic gear mechanisms are given for demonstrating the feasibility of the proposed methodology. Next, following the general trend of increased shift stages and a wider range of velocity ratios, new six-, seven- and eight-velocity automatic transmissions are enumerated from the two-ring nine-link Ravigneaux gear mechanisms. The result of this work shows that the nine-link two-DOF Ravigneaux-type epicylic gear mechanisms could reach eight-forward speeds at most. It is a *major breakthrough* to design *eight-speed* automatic transmissions from the *nine-link* Ravigneaux gear mechanism since it has only nine links. This structural design has realized a *reduced length* automatic transmission, while having *minimal number of gears*.

The methodology can be applied to any transmission mechanism depending on its kinematic and geometric constraints.

Keywords: Automatic Transmission, Clutching-Sequence, Epicyclic Gear Train, Eight-Velocity, Feasibility Graphs, Nomographs, Ravigneaux Gear Set, Systematic Design.

الخلاصة

بالرغم من وجود العديد من آليات نقل الحركة الآلية في سوق الانتاج فأن بعض تلك الآليات ما زالت بعيده من الوصول الى القيمة القصوى من النسب السرعية المتاحة لها. أن من بعض أهداف البحث هو محاولة الوصول الى العدد الاقصى من النسب السرعية المتاحة لأي منظومة نقل سرعه. في البحث الحالي تم استعراض طريقه تصميم آليات نقل الحركة الكوكبية من نوع را فكنوز ثنائيه الترس الحلقي. في البدء وبالاعتماد على مخططات النوموغراف المناظرة للنسب الترسية الأساسية فقد تم تعداد كافة تعاشيق التروس. ثم تم استعمال تمثيل المخطط المستوي لترتيب تعاشيق التروس الممكنة لكل تتابع تعاشيق تروس متاح لآلية نقل الحركة. ثم باستعمال الطرق اعلاه تم استعراض تصاميم آليات نقل الحركة الكوكبية لأثبات أمكانية استخدامها عمليا. بعدها وبأتباع السياق العام في زياده عدد تبديلات السرعة واتساع مدى النسب السرعية فقد تم تعداد كافة تعاشيق التروس سداسيه وسباعية وثمانية السرعة للآليات الكوكبية نوع راف كنوز ثنائية الترس الحلقي. وقد بينت نتائج الدراسة الحالية أن الآليات الكوكبية نوع را فكنوز ثنائية الترس الحلقي ممكن ان تعطي ثمان سرع اماميه كحد اقصى. انه لتطور مهم أن يتم تصميم آلية نقل حركه ثمانية السرعة من مجموعة را فكنوز ذات التسعة وصلات وهو آمر لم يتم الوصول له سابقا. وفي التصميم البنائي الحالي فقد تم ادراك الطول القصير لآليه نقل الحركة بأقل عدد من التروس. أن الطريقة الحالية تصلح لأي منظومة نقل حركه بالاعتماد على المحددات الحركية والشكلية.

الكلمات المفتاحية : آليات نقل الحركة الآلية ، تعداد تعاشيق التروس ، المسلسلات الترسية الكوكبية ، منظومة التروس الكوكبية نوع راف كنوز ، تصميم منهجي.

Introduction

For a long time automatic transmissions with planetary gear trains are used in the automotive industry. An effected coupling of planetary gear entities allows automatic transmissions with a large number of gears. However, the selection of ratios is restricted, since the gearwheels are used for several gears. The individual planetary gear sets are arranged in a row like discs. More planetary gear sets also always means a greater transmission length. This must be considered in particular in the case of transmissions for front-transverse drives. To reduce the length by designing a wide transmission, as done with .

The automatic transmissions treated in the following employ fundamental planetary gear entities only. Even in a single planetary gear set, planetary transmissions already provide a large number of states of motion possible by combination.

A design often used in automatic transmissions is the Ravigneaux planetary gear set. The Ravigneaux set is a so-called reduced planetary gear. These are planetary transmissions in which the construction resources are "reduced" since parts of the individual simple planetary gear sets are grouped together.

Ravigneaux in 1940 proposed seven- and eight-link two-degree-of-freedom (DOF) epicyclic gear mechanisms (Ravigneaux, 1940a, 1940b). The seven-link two-DOF Ravigneaux gear train has been developed by nearly all automotive manufacturers as three- or four-velocity automatic transmission (Tsai ,2001). It can be found in Ford C3, Ford C5, Mercedes Benz, Toyota A40 and Nissan, to name a few three-velocity automatic transmissions. It can also be found in KM 175 and 176, Ford AOD, ZF 4 HP 14 and Borg–Warner (Wilfinger *et al.*, 1988), to name a few four-velocity automatic transmissions (Hsu *et al.,* 2009). **Figure 1** shows the ZF 4 HP 14 automatic transmission (Naunheimer *et al.,*2011), which can provide four forward speeds and one reverse speed. The transmission went into production in 1984. It consists of seven-link two-DOF Ravigneaux gear mechanism, three rotating clutches C_1, C_2 and C_3, three band clutches B_1, B_2 and B_3 and two freewheels F_1 and F_2 . With this design, there are up to four practically usable forward gears and one reverse gear available.

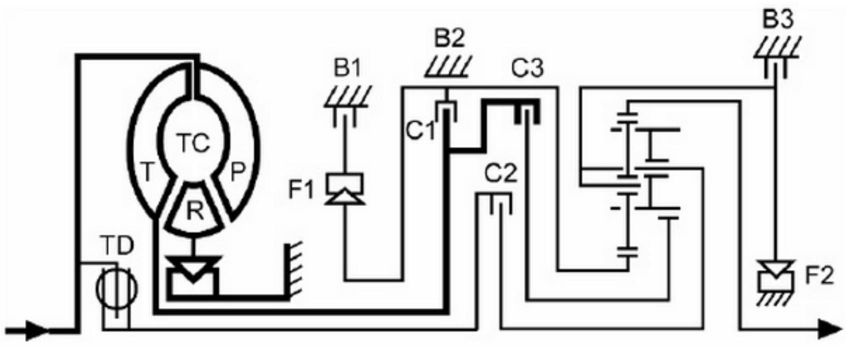


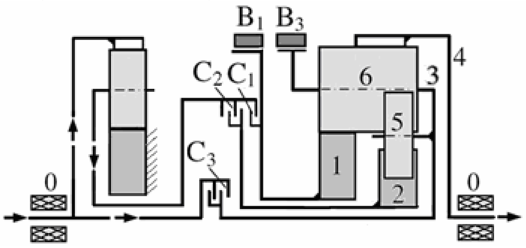
Figure 1 Gearbox diagram for ZF 4 HP 14 4-speed automatic transmission with Ravigneaux gear set (*TC*: Trilok converter, *P*: pump, *T*: turbine, *R*: reactor with freewheel, *TD*: torsion damper, *F*: freewheels, *B*: brakes, *C*: clutches) (Naunheimer *et al.,*2011).

In the clutching sequence shown in **Table 1**, X indicates that the i^{th} clutch is activated on the corresponding link for that gear. The ranges of output velocities are classified into three kinds: under drive (UD), over drive (OD) and reverse drive (RD) according to whether the velocity is between zero and the input velocity, more than the input velocity, or less than zero. A "direct drive" (DD) is equal to the input velocity.

Table 1 Clutching sequence for the transmission shown in **Fig. 1**.

Range	Activated clutches							
	C_1	C_2	C_3	B_1	B_2	B_3	F_1	F_2
First			X					X
Second			X	X			X	
Third		X	X					
Fourth		X			X			
Reverse	X					X		

A widespread gear set concept is that of Lepelletier. Lepelletier design is based on a seven-link two-DOF Ravigneaux gear train which is integrated with a simple epicyclic gear train to form ten-link three-DOF epicyclic gear mechanisms. This design is introduced to enhance the number of forward speeds (Naunheimer *et al.,*2011) and (Katou *et al.,*2004); providing six speeds. **Figure 2** shows Lepelletier automatic transmission and its clutching sequence Table (Lepelletier,1992). In 2001, ZF used this gear set design to bring the first 6-speed passenger car transmission 6 HP 26 on the market (Scherer 2003).



	C_1	C_2	C_3	B_1	B_3	Velocity ratio
UD_1		X			X	4.135
UD_2		X		X		2.363
UD_3	X	X				1.508
UD_4		X	X			1.140
OD_1	X		X			0.860
OD_2			X	X		0.675
RD	X				X	-3.127

Figure 2 Six-velocity Lepelletier transmission and its clutching sequence Table (Lepelletier 1992).

Figure 3 shows an automatic transmission concept according to Lepelletier; a nine-link two-DOF Ravigneaux gear train has been integrated with a simple epicyclic gear train to form twelve-link three-DOF epicyclic gear mechanism (Ishimaru,2007). This configuration enables the use of two additional forward gears. **Table 2** shows the corresponding eight-velocity clutching sequences for this automatic transmission.

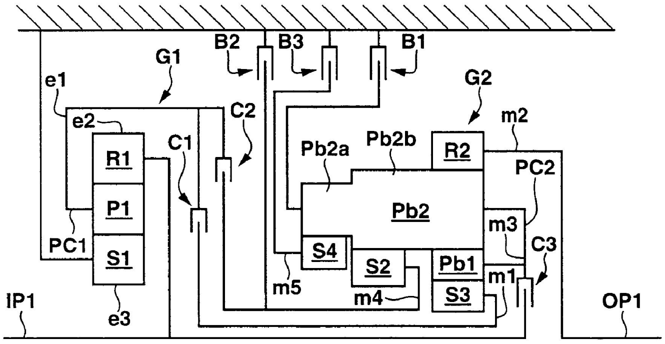


Figure 3 Eight-velocity automatic transmission concept according to Lepelletier (Patent No. US 7282007 B2) (Ishimaru 2007).

Table 2 Clutching sequences for the eight-velocity automatic transmission shown in **Figure 3** (Patent No. US 7282007 B2) (Ishimaru 2007).

	C1	C2	C3	B1	B2	B3	GEAR RATIO	6-SPEED AT
1ST	●			●			4.51	1ST
2ND	●					●	2.88	–
3RD	●				●		2.42	2ND
4TH	●	●					1.58	3RD
5TH	●		●				1.15	4TH
6TH		●	●				0.87	5TH
7TH			●		●		0.71	6TH
8TH			●			●	0.56	–
REV1		●		●			-5.53	REV
REV2		●				●	-0.49	–

Two-DOF eight-link Ravigneaux gear mechanism, shown on the right of **Figure 4**, consists of a long pinion and two or more short pinions connecting two sun gears and two ring gears. Unfortunately, two-DOF eight-link Ravigneaux gear trains have been developed as four-velocity automatic transmission (Ravigneaux,1956) while they can be used to produce automatic transmissions having more than four speeds (Hattori,1992). This work in part will attempt to attain maximum sequential velocity ratios for any given epicyclic gear train. A reduced length automatic transmission can be realized by designing transmission using Ravigneaux gear sets. This is done by adding a ring gear to the seven- or eight-link two-DOF Ravigneaux gear trains shown in **Figures 2 and 3** to obtain eight- and nine-link 2-DOF Ravigneaux gear trains, respectively. **Figure 4** shows such mechanisms that will be used to realize reduced length automatic transmissions having the maximum sequential velocity ratios. This means that the simple epicyclic gear train used in the Lepelletier design has been dispensed in the new design while still reaching the same velocity ratios as will be latter shown.

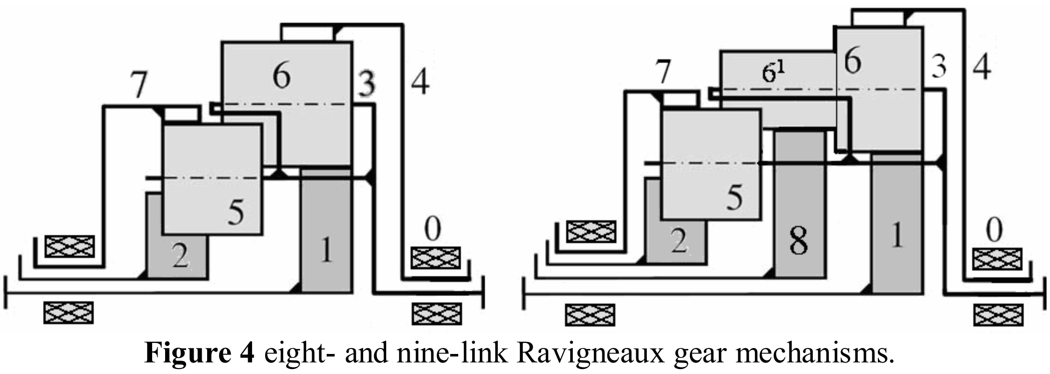


Figure 4 eight- and nine-link Ravigneaux gear mechanisms.

The literature on the design of planetary gear trains includes conceptual designs, kinematic analysis, power flow, efficiency analysis, and configuration designs. However, relatively little work has been done on the enumeration of clutching sequences and configuration design of EGMs. This paper presents a procedure to enumerate clutching sequences and to find feasible clutch layouts for nine-link 2-DOF stepped Ravigneaux planetary gear trains and attempting to attain transmissions having maximum sequential velocity ratios with reduced length.

Literature Review

The selection of an optimal clutching sequence cannot be solved analytically. (Nadel *et al.* 1991a, 1991b, and 1993) formulated the task as a constraint satisfaction problem. (Tsai *et al.* 1996 and 1998), (Hwang *et al.*, 2005) and (Hsu *et. al.*, 2009) used algorithmic techniques. (Ross *et al.* 1991) and (Esmail,2009) introduced graphic techniques. (Hattori *et. al.*, 1995) used phase geometry method.

(Nadel *et. al.*, 1991a, and 1991b) applied an artificial intelligence technique to enumerate clutching sequences for EGMs made up of two basic epicyclic gear trains. The artificial intelligence technique is a powerful tool for solving the transmission design problem. However, the technique assumes that the design variables have discrete values in prescribed domains. Furthermore, it requires a search over the entire feasible solution space. This methodology is suitable for PGTs in which two simple PGTs are combined. These shortcomings inevitably reduce the efficiency of the algorithm.

(Hattori *et. al.*, 1995) proposed twenty three phase geometric patterns for five-speed Automatic transmissions, each could provide four clutching sequences. Each feasible clutching sequence obtained could be used to construct a clutch layout. However, this approach is suitable only for PGTs consisting of two sun gears, two ring gears and one to three meshed planet gears mounted on a common arm. (Ross *et. al.*, 1991) introduced a design tool based on a lever analogy. It includes calculation of gear ratios, gear trains selection, and the construction of clutch layouts for parallel-connected PGTs. (Hwang *et al.*, 2005**)** and (Hsu *et. al.*, 2009) used similar methods to that used by (Tsai *et. al.*, 1996) to enumerate the clutching sequences of two-DOF eight-link Ravigneaux gear mechanism. Moreover, the five studies cited above are restricted to constructing clutch layouts of automatic transmissions for specific types of PGTs.

(Tsai *et. al.*, 1996) used the concept of fundamental gear entities (FGEs) proposed by (Chatterjee *et. al.*, 1995) in conjunction with their earlier kinematic study (Hsieh et al. 1996) to determine the most efficient clutching sequence associated with automatic transmission (Tsai *et. al.*, 1998). They applied combinatorial enumeration procedure to arrange the velocity ratios in a descending sequence. Then, they used an algorithm to enumerate clutching sequences for EGMs composed of two or more FGEs. A computer algorithm for the enumeration of clutching sequences is given by (Hsieh, 1996). The algorithm needs information containing the approximate gear sizes arranged in a descending order.

Most combinatorial enumeration procedures are done through the process of generating and testing. The procedure is thus divided into two parts: a generator of all possible solutions and a tester that selects only those solutions that meet the constraints. An important issue in using a generating and testing technique is the distribution of knowledge between the generator and tester. The generator produces solutions satisfying some of the constraints. The tester then selects those solutions that satisfy the rest of the constraints. While this technique is valid for solving transmission design problems, it limits the solutions to the knowledge (information) contained in the generator and tester. This inevitably reduces the efficiency of this solution technique and needs complicated computer-algorithm. Usually, putting more knowledge in the generator, results in a more efficient procedure. Nevertheless, (Tsai *et. al.*, 1996) did not develop an effective method of arranging clutch layouts for the synthesized clutching sequences. However, the elimination of invalid clutching sequences was conducted by inspection. Identifying invalid clutching sequences by inspection is not always reliable.

(Esmail, 2009) proposed a methodology, based on nomographs, for the enumeration of the associated clutching sequence table for an epicyclic gear mechanism. This method simplifies the synthesis of the clutching sequence of an epicyclic gear mechanism efficiently.

Without careful examination, a clutching sequence may be mistaken for a usable clutch layout for an automatic transmission. (Hsu *et. al.*, 2009) proposed a planar-graph representation to arrange the desired clutches for each possible clutching sequence into the Ravigneaux gear mechanism. (Hwang *et. al.,* 2011) introduced coded sketches for connecting clutch elements to planetary gear trains for automotive automatic transmissions.

The literature survey reveals that although some studies are made on the clutching sequence synthesis, they are all lengthy. In this paper nomographs are used to synthesis the clutching sequences of Ravigneaux gear trains; the nomograph method is described in a series of previous papers (Esmail, 200٧, 2009, 2013, and 2016). Only the related topics needed in this paper will be reviewed wherever appeared.

In this paper, solution techniques are developed to overcome those shortcomings. This paper applies kinematic nomographs (Esmail ,2007, 2009) and feasibility graphs (Hsu *et. al.*, 2009) of Ravigneaux gear trains to achieve the goal. By virtue of these solution techniques, reduced length six-, seven- and eight-speed automatic transmissions are designed from the nine-link 2-ring Ravigneaux gear mechanism.

Enumeration of all Feasible Clutching Sequences With the Aid of Nomograph

The first stage of designing epicyclic-type automatic transmissions is to synthesize all possible clutching sequences from the epicyclic-gear mechanism. Nomographs are used for this purpose.

Nomographs and Geometry Relations for Nine - Link Ravigneaux gear trains

A nomograph is defined as three or more axes, or scales, arranged such that problems of three or more variables can be solved using a straightedge. In the particular case of EGTs, a nomograph can be constructed using three or more vertical parallel axes (Esmail, 2009, 2016).

Traditionally, the velocity ratio is used to study the velocity relations among the different links of an EGT (Tsai, 2001). Let the symbol $R^{z}_{x,y}$ 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, then

$$R^{z}_{x,y} = \frac{\omega_x - \omega_z}{\omega_y - \omega_z} \tag{1}$$

Since the gear mechanisms shown in **Figure 4** are double-planet FGEs then their nomographs can be drawn in terms of planet gear 6 (Esmail, 2009). **Figure 5** shows the nomograph of the stepped nine-link Ravigneaux gear shown on the left side of **Figure 4**.

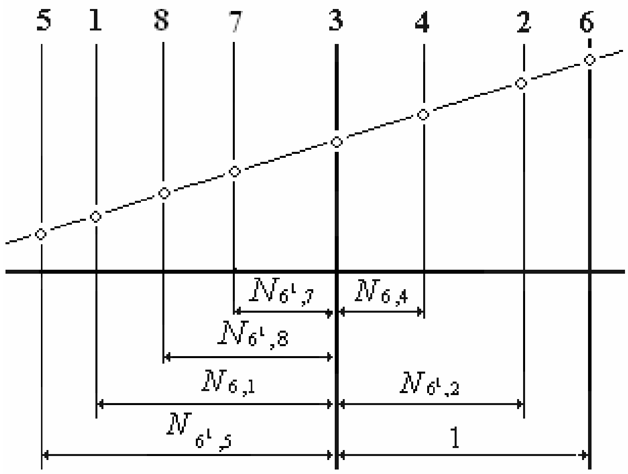


Figure 5 Nomograph for the double stepped nine-link Ravigneaux gear shown on the left side of **Fig. 4**, in terms of planet gear 6.

The term "gear ratio" is used in this paper to denote the ratio of a meshing gear pair. It is defined by the ratio of a planet gear p with respect to a sun or ring gear x

$$N_{p,x} = \mp Z_p / Z_x \tag{2}$$

Where Z_p and Z_x denote the numbers of teeth on the planet and the sun or ring gear, respectively, and the positive or negative sign depends on whether x is a ring or sun gear.

Considering the kinematics of a fundamental circuit, the fundamental circuit equation can be written as (Esmail 2009) :

$$(\omega_x - \omega_c)/(\omega_p - \omega_c) = N_{p,x} \tag{3}$$

Equation (3) can be re-written for ring, sun, and planet gears that are meshing directly to planet 6 and for the carrier to obtain $N_{p,r}$, $N_{p,s}$, $N_{p,p}$ and $N_{p,c}$, respectively. Each gear ratio is written in terms of the plant to which it is meshing, either gear 6 or 6^1 of the stepped plant.

The gear ratio for small sun gear s_s that is not meshing directly with the first planet gear p on which the nomograph is drawn and is meshing with the second planet gear p_1 can be found in terms of the gear ratio of the two planets $N_{p,p1}$ as

$$N_{p,s_s} = N_{p,p_1} \cdot N_{p_1,s_s} \tag{4}$$

The values of the gear ratios are used to place the axes of the nomograph shown in **Fig. 5**. The ω_c axis passes at the origin, and the ω_p axis is one unit apart from it.

The gear ratios for the Ravigneaux gear train are

$$N_{6,1} = -Z_6 / Z_1 \tag{5}$$

$$N_{6,4} = Z_6 / Z_4 \tag{6}$$

$$N_{6^1,8} = -Z_{6^1} / Z_8 \tag{7}$$

$$N_{6^1,5} = -Z_{6^1} / Z_5 \tag{8}$$

$$N_{5,7} = Z_5 / Z_7 \tag{9}$$

And

$$N_{5,2} = -Z_5 / Z_2 \tag{10}$$

Considering the geometry relations of the Ravigneaux gear mechanism given in equations (5) through (10), the values of the gear ratios can be deduced as following:

$$-\infty \leq N_{6,1} \leq 0 \tag{11}$$

$$-\infty \leq N_{6^1,8} \leq 0 \tag{12}$$

$$-\infty \leq N_{6^1,5} \leq 0 \tag{13}$$

$$-\infty \leq N_{5,2} \leq 0 \tag{14}$$

$$0 \leq N_{6,4} \leq 1 \tag{15}$$

and

$$0 \leq N_{5,7} \leq 1 \tag{16}$$

From Eq. (4), we can write

$$N_{6^1,7} = N_{6^1,5} \cdot N_{5,7} \tag{17}$$

$$N_{6^1,7} = -Z_{6^1} / Z_7 \tag{18}$$

Since Z_7 is greater than Z_1 and Z_6 is greater than Z_6^1 then from equations (5) and (18) it can be shown that

$$N_{6^1,7} > N_{6,1} \tag{19}$$

Similarly $N_{6^1,8} > N_{6,1}$ and $N_{6^1,7} > N_{6^1,8} > N_{6,1}$. Moreover, from Eq. (4), we can write

$$N_{6^1,2} = N_{6^1,5} \cdot N_{5,2} \tag{20}$$

$$N_{6^1,2} = Z_{6^1} / Z_2 \tag{21}$$

$$N_{6^1,2} = Z_{6^1} / Z_2 \tag{22}$$

Since Z_4 is greater than Z_2 and Z_6 is greater than Z_6^1 then from equations (6) and (22) it can be shown that these gear ratios work oppositely; while one increase the gear ratio, the other decrease it, but, without loss of generality, we can select the most probable case where ;

$$N_{6^1,2} > N_{6,4} \tag{23}$$

Enumeration of all Feasible Clutching Sequences

Kinematic relationships among the links of this FGE can easily be visualized by observation from the nomograph shown in **Figure 5**. Any straight line through the input operating velocities of the EGT will intersect other axes at the operating velocities of the links representing to those axes. From the nomograph, $R^{z}_{x,y}$ can be written as (Esmail, 2009)

$$R_{x,y}^{z} = \frac{N_{p,x} - N_{p,z}}{N_{p,y} - N_{p,z}} \tag{24}$$

Because there are six coaxial links and link four or eight can be assigned as the output link, this gear train can provide twenty overall velocity ratios for each assignment (Esmail, 2016). **Figure 6** shows the clutching sequence nomograph for the nine-link Ravigneaux gear mechanism shown in **Figure 4**.

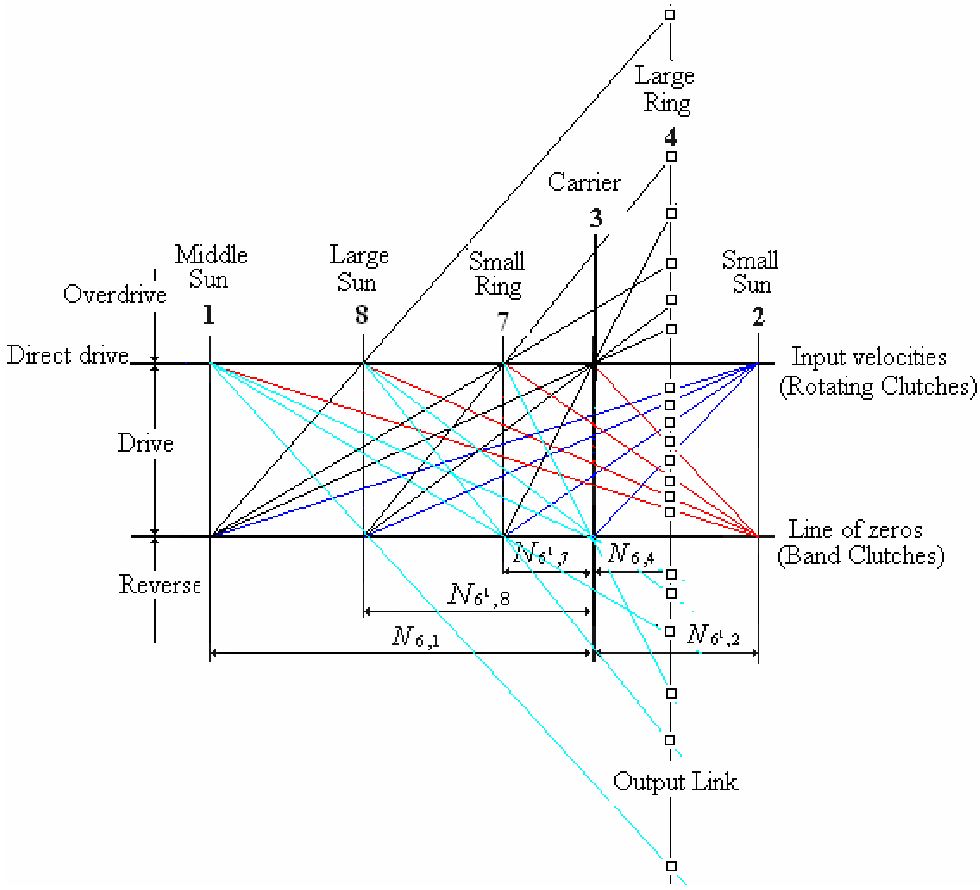
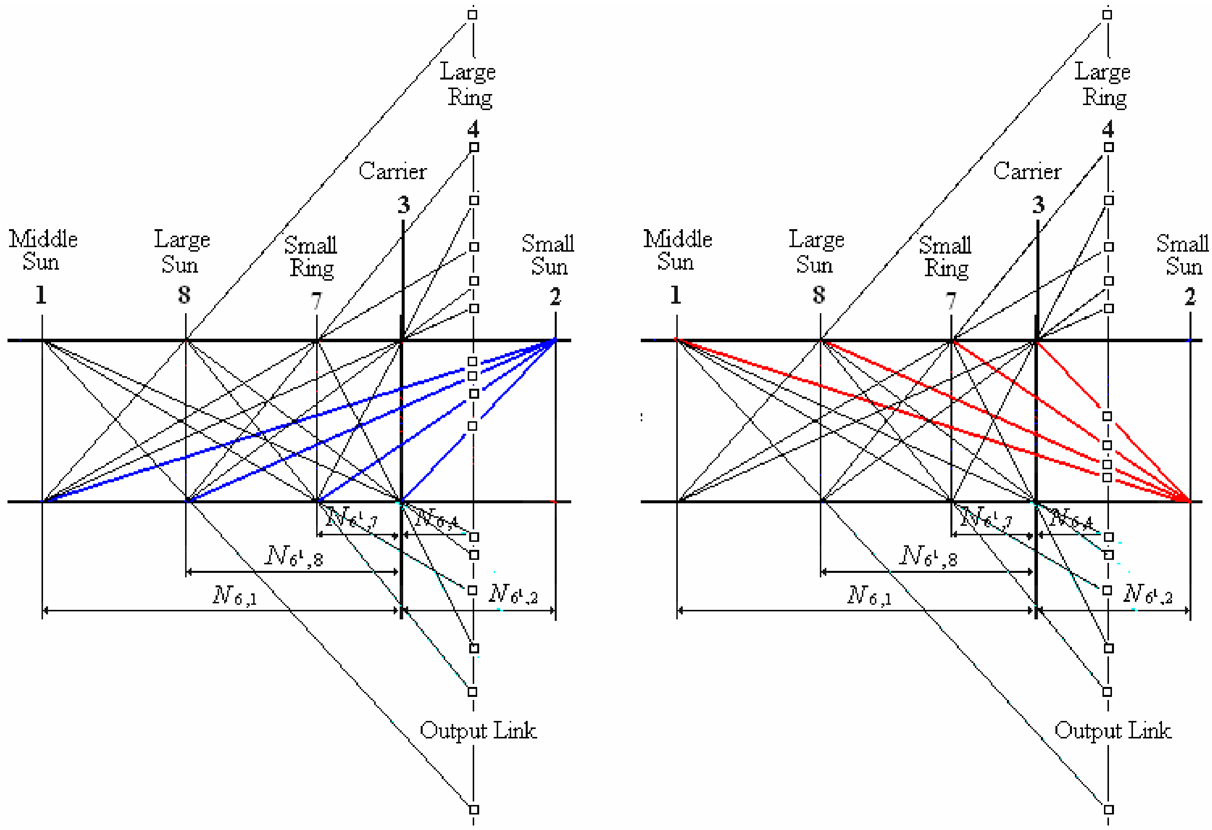


Figure 6 Clutching sequence nomograph for the nine-link Ravigneaux gear train shown in **Figure 4**.

In arranging a clutching sequence, it is highly desirable to achieve a single-shift transition (Esmail, 2009). A direct drive is obtained by simultaneously clutching two coaxial links of an EGT to the input power source. A reverse drive can be obtained by applying one or two of the clutches designed for the forward drives to the reverse drive.

The velocity ratios are classified into three groups; under-drive, over-drive and reverse drive. If there are more than two sets of velocity ratios in any group, they are further classified into all possible sets or subgroups based on the constraint that only one clutch can be shifted in each set. Possible clutching sequences are generated by combining the UD, DD, OD, and RD subgroups together. The velocity ratios in any set are arranged in a descending order, and then the corresponding clutching sequence is generated.

Returning to **Fig. 6** and to achieve single-shift transitions, the UD velocities are classified into two sets as shown in **Figure 7**. The OD and RD subsets are used with both of the UD sets.



(a) First UD set (b) Second UD set

Figure 7 Clutching sequence nomograph for the two UDs of the nine-link Ravigneaux gear train with all the OD and RD subsets.

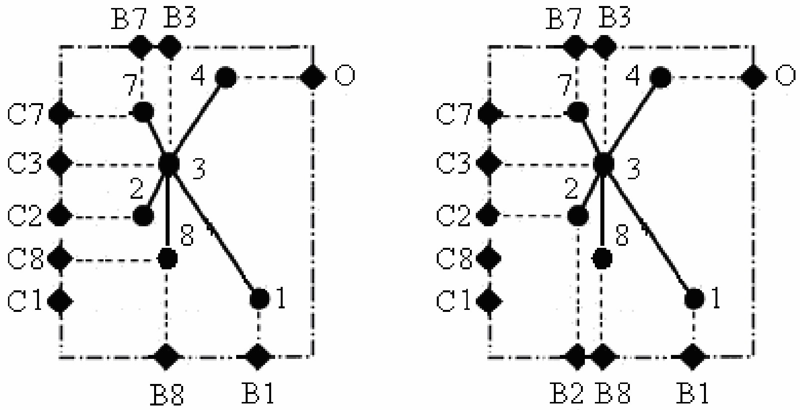
2.3. Feasibility Graphs for an Epicyclic Gear Train to Form an Automatic Transmission

A graphical representation is proposed by (Hsu *et. al.*, 2009) to easily and quickly detect the possibility of the arrangement of all the rotating clutches and band clutches into the gear mechanism to forming an automatic transmission.

A graph that can be drawn on a plane such that no two of its edges intersect is called planar; meaning it can form an automatic transmission. A graph that cannot be drawn on a plane without crossover between its edges is called non-planar, meaning there is no possibility to form an automatic transmission. Here we shall call such graphs as the feasibility graphs of a gear train to form an automatic transmission.

For the first set, the UD clutching sequences shown in **Figure 7 (a)**, are controlled by C_2, B_1, B_3, B_7 and B_8, while those for the second set shown in **Figure 8 (b)**, are controlled by B_2, C_1, C_3, C_7 and C_8.

The feasibility graphs of the nine-link Ravigneaux gear train to form automatic transmissions are shown in **Fig. 8 (a) and (b)**.



(a) First UD set (b) Second UD set

Figure 8 Feasibility graphs for the nine-link Ravigneaux gear train to form six-, seven-, or eight-velocity automatic transmissions.

For the first UD set, the clutching sequences, shown in **Figure 8 (a)**, are controlled by C_2, C_3, C_7, C_8, B_1, B_3, B_7 and B_8. As a result, all of the UD and OD clutching sequences that can be obtained from the first set are feasible to form automatic transmissions. The RDs are only obtained from one of the rotating clutches C_7 or C_8 and one of the band clutches B_3 or B_7.

Figure 8 (a) reveals that a clutch cannot be attached to link 1, thus, the last three reverse clutching sequences (C_1B_3, C_1B_7, and C_1B_8) shown in **Figure 7 (a)** are inadmissible. As a result, only thirteen clutching sequences shown in **Figure 9** are feasible at this design stage.

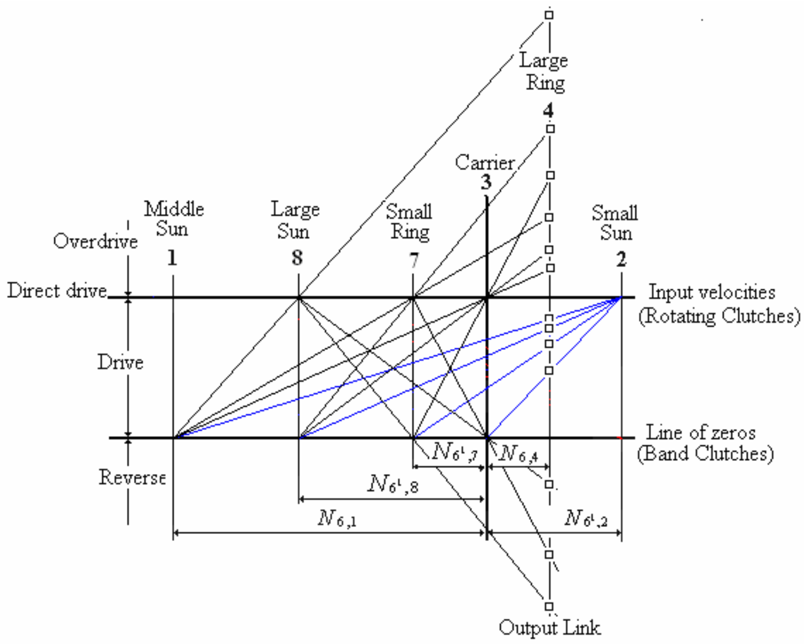


Figure 9 Clutching sequence nomograph for the nine-link Ravigneaux gear train based on the first UD feasibility graph.

Based on the first UD set of feasibility graph, the general clutching sequence layout for the nine-link Ravigneaux gear train is shown in **Figure 10**. All of the feasible clutching sequence layouts can be extracted from this general layout.

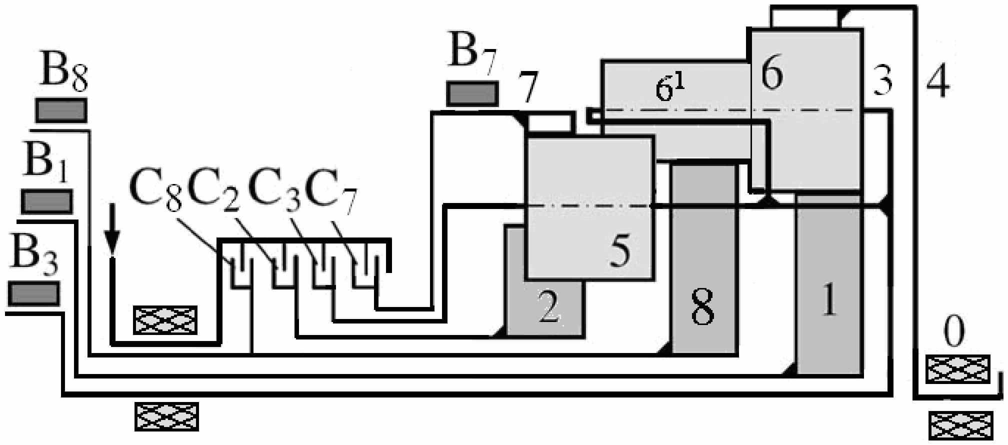


Figure 10 Clutching sequence layout for the nine-link Ravigneaux gear train based on the first UD feasibility graph.

In order to achieve single-shift transitions, the ODs can be further classified into seven sets; four sets consist of three-over-drives, two sets consist of two-over-drives and one set of one-over-drive as shown in **Table 3**.

Table 3 The seven over-drive sets and their associated direct drives for the nine-link Ravigneaux gear train.

	1st set	2nd set	3rd set	4th set	5th set	6th set	7th set
DD	$C_2 C_3$	$C_2 C_3$	$C_2 C_3$	$C_2 C_3$	$C_2 C_7$	$C_2 C_7$	$C_2 C_8$
OD_1	$C_3 B_1$	$C_3 B_1$	$C_3 B_1$	$C_3 B_1$	$C_7 B_1$	$C_7 B_1$	$C_8 B_1$
OD_2	$C_3 B_8$	$C_3 B_8$	$C_7 B_1$	$C_7 B_1$	$C_7 B_8$	$C_8 B_1$	-----
OD_3	$C_3 B_7$	$C_7 B_8$	$C_7 B_8$	$C_8 B_1$	-----	-----	-----

By combining the first set of UD clutching sequences with the four sets of three-velocity OD clutching sequences and three RDs (C_7B_3, C_8B_3, and C_8B_7) twelve possible clutching sequences of eight-velocity automatic transmissions are feasible at this design stage.

Moreover, the first set of UD clutching sequences can also be combined with two sets of two-velocity OD clutching sequences and three sets of RDs, and six possible clutching sequences of seven-velocity automatic transmissions are feasible at this design stage. It can also be combined with the 7th over-drive set to obtain three possible clutching sequences of six-velocity automatic transmissions.

For the second set, since B_2 is essential for the UD clutching sequences, then only band clutches can be attached to links 1 and 8 as shown in **Figure 8 (b)**. Therefore, the clutching sequences are controlled by C2, C3, C7, B1, B2, B3, B7 and B8. The clutch C2 has no role in the clutching sequence. By excluding C_2 from the clutching sequences nomograph, only two UDs, five ODs and one reverse will remain as shown in **Figure 11.** They are controlled by C_3, C_7, B_1, B_2, B_3, B_7 and B_8.

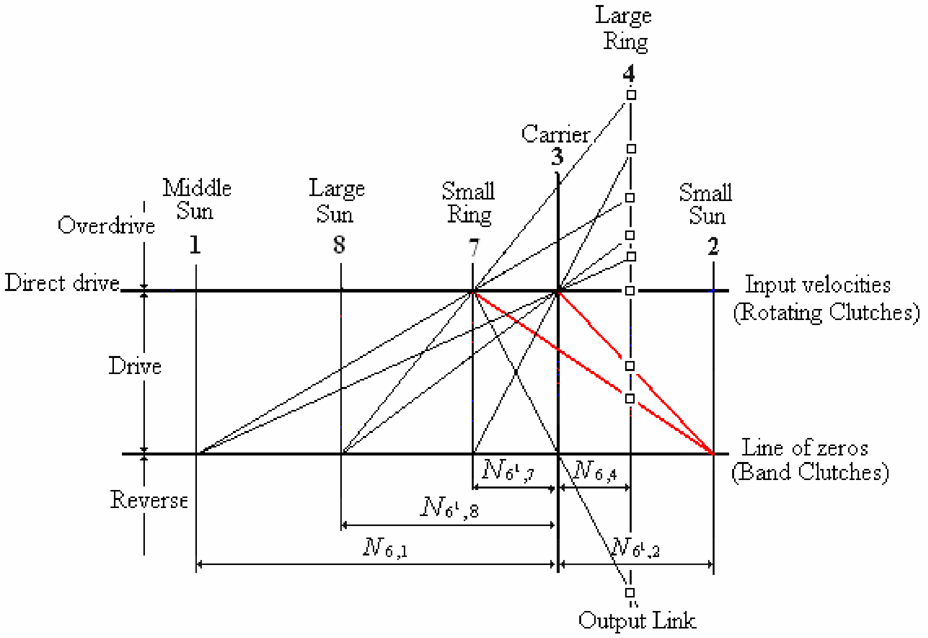


Figure 11 Clutching sequence nomograph for the nine-link Ravigneaux gear train based on the second UD feasibility graph.

Based on the second UD set of feasibility graph, the general clutching sequence layout for the nine-link Ravigneaux gear train is shown in **Figure 12**. All of the feasible clutching sequence layouts can be extracted from this general layout.

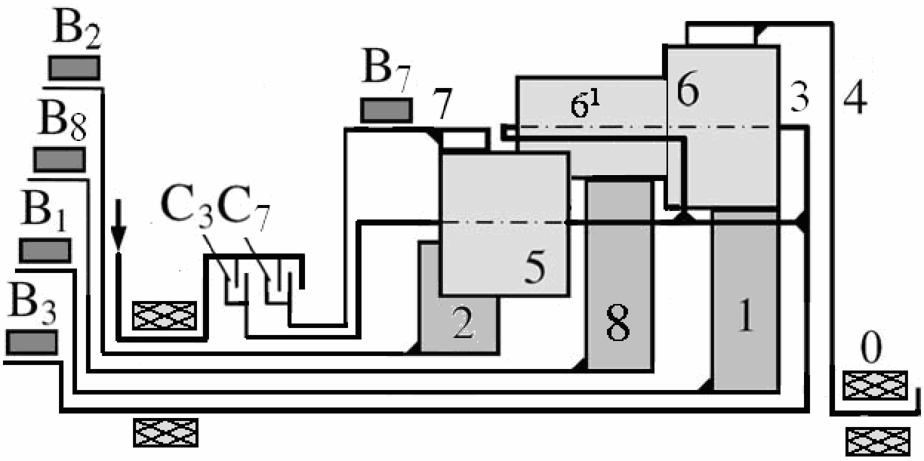


Figure 13 Clutching sequence layout for the nine-link Ravigneaux gear train based on the second UD feasibility graph.

The RD is obtained from the rotating clutches C_7 and the band clutches B_3. To achieve single-shift transitions, the ODs can be further classified into five sets; three sets consist of three-over-drives, one set consists of two-over-drives and one set having one-over-drive as shown in **Table 4**.

Table 4 The five over-drive sets for the nine-link Ravigneaux gear train.

	1st set	2nd set	3rd set	4th set	5th set
DD	C_3 C_7	C_3 C_7	C_3 C_7	C_3 C_7	C_3 C_7
OD_1	C_3 B_1	C_3 B_1	C_3 B_1	C_7 B_1	C_7 B_8
OD_2	C_3 B_8	C_3 B_8	C_7 B_1	C_7 B_8	-----
OD_3	C_3 B_7	C_7 B_8	C_7 B_8	-----	-----

By combining the second set of UD clutching sequences with the three sets of three-velocity OD clutching sequences and the one RD, three possible clutching sequences for six-velocity automatic transmissions are feasible at this design stage. One clutching sequence is possible too for four- or five-velocity automatic transmissions.

Conclusions

A methodology for the design of *two-ring* Ravigneaux-type epicyclic gear transmissions for automobiles has been presented. Following the general trend of increased shift stages and a wider range of velocity ratios, new six-, seven- and eight-velocity automatic transmissions have been enumerated from the two-ring nine-link Ravigneaux gear mechanisms. The result of this work shows that the nine-link two-DOF Ravigneaux-type epicylic gear mechanisms could reach eight-forward speeds at most. The completeness of the results cannot be confirmed since no publications exist for eight-velocity Ravigneaux gear mechanisms. But it is a major breakthrough to design a completely satisfactory eight-speed automatic transmission from the two-ring Ravigneaux gear train since it has only nine links. This structural design has realized a *reduced length* automatic transmission, while having *minimal number of gears*. Feasibility graphs have been used as a design tool, allowing the designer to quickly selecting the most viable clutching sequence. The proposed methodology can be used for the systematic design of any epicyclic-type automatic transmission.

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