A SOLUTION TECHNIQUE FOR DESIGN OF THREE- SPEED AUTOMATIC SPEED CHANGER

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ABSTRACT

Recently, there has been an increasing interest in the development of bicycles speed changers. Multi-speed changers allow cyclists to pedal at their most efficient pace at different road conditions. In this paper, a modified solution technique based on a flawed previous one is presented to investigate the kinematics, the torque distributions and the power flow through one-degree of freedom planetary gear trains. Nomographs are used as a design tool to enable the designer to simultaneously visualize these one-degree of freedom variables without first selecting a particular PGT and without the need to solve equations repeatedly.
After that, these techniques will be applied to design a three-speed automatic speed changer. Three design examples are demonstrated to illustrate this modified methodology.

KEYWORDS: Solution Technique, Automatic speed changer, Planetary Gear train, Nomography, Bicycle, Design.

أسلوبُ حَلٍ في تصميمِ مُبدّل سرعة آليّ ذو ثلاث سرع

مدرس مساعد : خلود حسن صالح
قسم المكائن والمعدات– معهد التكنولوجيا بغداد
بغداد–عراق

المـــوجز

في الوقت الراهن، هناك اهتمام متزايد في تطوّر مبدّلات سرع الدراجات الهوائيه. تُتيح المبدّلات متعددة السرعه لراكبي الدراجات ان يدوّروا الدوّاسات القدميه بافضل ما يمكن من سرعه في الظروف المختلفه للطريق.
يُقدّمْ هذا البحث اسلوب حَلّ مُنقَّح لحلٍ سابقٍ معيب ، يتحرى الحل الحالي كينماتيكا المجموعات الترسيه الكوكبيه احادية درجة الحريه وتوزيع العزوم فيها وتدفق القدره خلالها . كما يَستخدمْ المخططات النوموغرافيه ويحورها لتكون أداة تصميميه تُمكّن المصمم من ان يتصوّر كل متغيرات المجموعات الترسيه الكوكبيه احادية درجة الحريه في نفس الآن على مخطط واحد دون ان يختار مجموعه ترسيه كوكبيه معينه ابتداءً ودون الحاجه الى حل المعادلات تكراراً.

يَستعرضُ البحث، باستخدام الحل الجديد والاداة التصميميه، تصميم مبدّل سرعه آليّ ذو ثلاث سرع مع ثلاث امثله تطبيقيه توضحّ هذه الطريقه الجديده عملياً.

INTRODUCTION AND LITERATURE REVIEW

A planetary gear train (PGT) is defined as any train in which one or more gears orbit by turning about a rotating axis just as a planet turns about the sun. For the kinematic analysis of PGT's, various approaches such as the relative velocity method [1-3], energy method [**Wilkinson, 1960**], bond graph method [**Allen, 1979**] and the vector-loop method [**Gibson and Kramer, 1984**] and [**Willis, 1982**] have been proposed. [**Freudenstein and Yang, 1972**] introduced the concept of fundamental circuit to analyze PGT's. The concept was further extended by other researchers [9-10]. The analysis involves the solution of a set of linear equations for all the kinematic variables. It does not provide much insight into mechanics of a PGT.

The elementary PGT is shown in **Figure 1**, with its simplified representation. It consists of two gears, the central (1) and planet (2) gears and the planet carrier (3) or arm .The central gear may be either an external or an internal gear.

The elementary PGT is somewhat limited in practical application .More useful , however , are the PGT's referred to as the simple , compound and complex PGT's where a second central gear is used and also there is a sequence of planets ,connected in by either a shaft or by a tooth mesh , between the first and last central gears .

The simple PGT shown in **Figure 2** consists of a first central gear (F), a planet gear (P), a planet carrier or arm (A), and a last central gear (L).

[**Lévai, 1966**] identifies twelve possible variations of PGT's; they are shown in **Figure 3** using kinematical representation. These PGT's are classified to

(1) Simple and compound PGT's in which the planet gears are in mesh with both central gears, and

(2) Complex PGT's in which the planet gears are partially in mesh with each other and partially in mesh with the two central gears.

Notice that, for these PGT's and regardless of arrangement the multiple planets are always on a common carrier (multiple central gears PGT or multiple carriers PGT will be referred to as a multi-stage PGT's).

Taking the arm to be fixed, one can quickly deduce by examination that the PGT's in quadrant (1) and (2) yield negative transmission ratios between the first and last gears while PGT's in quadrant (3) and (4) yield positive transmission ratios.

[**Willis, 1870**] discusses for the first time in published literature the analytical modeling of a PGT. He suggests the use of a generic "transmission ratio" in defining the kinematic motion of a PGT. This ratio, R, is defined as the angular velocity ratio between the last and first central gears in the train relative to the arm. Mathematically, this can be written

$$R = \frac{\omega_L - \omega_A}{\omega_F - \omega_A} \qquad (1)$$

Graphically, this equation can be solved using nomography. 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 [**Corey, 2003**]. In the particular case of EGTs, a nomograph can be constructed using three vertical parallel axes [2, 17] or three or more vertical parallel axes [**Esmail, 2007**]. Corey's approach [**Corey, 2003**] is a practical one which uses nomographs to investigate both the angular velocities and the torques acting on the first, last gears and the carrier of a two input train without reaction link. However, two input planetary gear trains contain an uncertainty in the torque solution space [**Esmail and Hassan, 2010**]. Unfortunately, Corey applied his approach wrongly to two input epicyclic gear trains without reaction link. In my opinion and since there is no torque uncertainty in the torque solution space of epicyclic gear trains with a reaction link, Therefore, Corey's approach can be considered correct for such trains and in this paper it will be adopted for one-degree of freedom epicyclic gear trains. This is the main modification to Corey's approach to make it applicable to epicyclic gear trains. They origins of nomographs date back to professor Maurice d'Ocagne in late ninetieth century. In the particular case of epicyclic gear trains, nomographs are still under progress and not yet a well-established technique. This technique was first applied by Toyota to investigate their Toyota Hybrid System [17]. Toyota's solution is not a general, concise solution technique; it is only an application to their case, no details are given as to how nomographs can be drawn and where axes can be placed. Toyota uses the three parallel axes nomograph in a different manner to that used by Corey.

In this paper a modified solution technique based on a previous one is developed to solve PGT's problems with the help of nomographs.

PGT's are considered adaptable to suit automatic internal speed changers for bicycles.

A bicycle, or bike, is a pedal-driven land vehicle with two wheels arranged in line. First introduced in 19th century, it evolved quickly into its current design. With over one billion in the world today, bicycles provide the principal means of transportation in many regions and a popular form of recreation in others.

Recently, there has been an increasing interest in the development of bicycles speed changers and their shifting mechanisms [13]. Multi-speed changers allow cyclists to pedal at their most efficient pace at different road conditions. The bicycle will gradually adopt the automatic shifting fashion, by following the path of a motorcycle or an automobile. By referring to such related patents such as those on automatic bicycle transmission [8, 15 and 16], we form the following ideas about the bicycle speed changers:

The bicycle speed changers can be divided into external and internal speed changers. The external speed changers in turn are divided into front external speed changers and back external speed changers. A complete speed changer contains:

(1) The speed changer entity which is the main body of the speed changer in of transmitting the power while the bicycle is running. The chain-driven external speed changer entity consists of three parts, i.e. front chain wheel group, chain, and back chain wheel group.

(2) The shifting control device which is used in controlling the operation of each gear position of bicycle speed changer entity, and is not responsible for transmitting the driving power. It is usually located inside or nearby the speed changer entity. The chain-driven external speed changer contains the front chain-shifting mechanism and back chain-shifting mechanism. The internal speed changer contains the control mechanism inside the axle and it may be (a) revolving (b) translating or (c) revolving plus translating control mechanism.

(3) The shifting steer device is used to steer the shifting control device. It does not transmit power and is usually installed in a position far away from the speed changer entity. The shifting control device can use the shifting line or other means to connect with the shifting control device.

Most bicycle external speed changers use the parallel four-bar linkage as the main body for shifting while internal speed changers use planetary gear train (PGT) as the main body for shifting. Internal speed changers use the sun gear as the controlling element to obtain the shifting effect. The shifting effect is obtained by either turning the sun gear around the central axis or by pulling or pushing the central axis. As to the external and internal speed changers, there are two shifting types; manual and electronic. Manual shifting includes the shifting line and revolving handle. Electronic shifting uses central process unit (CPU) with speed and torque sensors to control the shifting control device.

New approaches are needed to evolve mechanical or electromechanical solutions. The prior-art inaction to evolve a widely acceptable automatic bicycle is also regrettable from environmental and socio-economic points of view, since bicycles cost much less and take much less space than automobiles, put less a load on the road, do not pollute the atmosphere like automobiles, are much less expensive to operate, and subject rider to continual salubrious exercise unavailable in any automobile.

THE MODIFIED PLANETARY GEAR TRAIN SOLUTION TECHNIQUE

1- Nomographs for PGT's

Nomograph is defined as three or more axes, or scales, arranged such that problems of three or more variables can be solved a straightedge. In the particular case of PGT's, a nomograph can be constructed using three vertical parallel axes. The ω_L axis is chosen to pass at the origin; also the ω_L and ω_F axes are chosen to be one unit [2].

Following the Technique developed by Corey {2] with the present modifications, any straight line through arbitrary points on the ω_L and ω_F axes will intersect the horizontal axis, which connect the zeros of the three axes at point s, a distance S from the origin.

From **Figure 4**, we can write

$$\frac{w_l}{w_L - w_F} = \frac{S}{1} \tag{2}$$

The location of the ω_A axis, for a particular PGT, is at a distance E from the origin. The ω_A axis may actually be to the left of the ω_L axis or between the ω_L and the ω_F axes or to the right of the ω_F axis.

In general, from **Figure 4**, we can write

$$E = \frac{W_L - W_A}{W_L - W_F}$$

E can be defined in terms of R, the general transmission ratio of the gear train.

Also, from **Figure 4**

$$\frac{W_L - W_A}{W_A - W_F} = \frac{E}{1 - E} \tag{3}$$

Re-arranging, we get

$$\frac{W_L - W_A}{W_F - W_A} = \frac{E}{E - 1} \tag{4}$$

which is exactly equation (1) when

$$R = \frac{E}{E - 1} \tag{5}$$

In general, E can be written in terms of R as

$$E = \frac{R}{R - 1} \tag{6}$$

For the purpose of analysis R can be found using the knowledge of the arrangement of the particular PGT with the arm fixed, so

$$R = \frac{W_{Lo}}{W_{Fo}} \tag{7}$$

Now this ratio R (and hence E) is a constant, regardless of the rotational speeds of the three elements of the particular PGT, and is proportional to the tooth numbers whether the arm is rotating or not, and in general we may write

$$R = \frac{\text{Product of first Gear Teeth}}{\text{Product of last Gear Teeth}} \tag{8}$$

Equation (8) is based on our arbitrary designation of first and last gear teeth (which is often referred to as "Product of Driving or Driven Gear Teeth ").

The sign of R is positive if the arbitrary designated first and last gears would rotate in the same direction with the arm stationary.

In design, E can be selected arbitrary as any value other than one or zero.

2. Modified Solution Technique for PGT's Using Nomographs

Using the principle of energy conservation, the energy balance equation for the general PGT can be written as

$$T_A W_A + T_F W_F + T_L W_L = 0 \tag{9}$$

Figure 5 shows the torques T_L, T_F and T_A represented as arrows perpendicular to the three axes at the points representing the rotational speeds. The moment of any torque vector about the zero of its rotational speed axis, represent the power flowing in that branch (T · ω).

In general, from **Figure 5**, we can write

$$\frac{W_A}{S-E} = \frac{W_F}{S-1} = \frac{W_L}{S} \tag{10}$$

from which

$$W_L = \frac{S}{S-E} \cdot W_A \tag{11}$$

And

$$W_F = \frac{S-1}{S-E} \cdot W_A \tag{12}$$

Substituting equations (11) and (12) into equation (9), and multiplying by (S – E), we get

$$T_A(S-E) + T_F(S-1) + T_L(S) = 0 \tag{13}$$

Perhaps it is more convenient to represent the torque vectors as arrows along the three axes as shown in **Figure 6**. Therefore, equation (13) is equivalent to summing moments of the torque vectors about point s.

Upward torque vector is assumed to have positive sign. Power flowing in the system is assumed to have positive sign also. Therefore, any torque vector pointing away from zero is representative of power flow in the system.

Although, the S-value is a variable quantity, which depends on the operating rotational speeds of the PGT, it allows early in the design process, to visualize how the PGT is intended to react to typical PGT rotational speeds. The S-value variation with different input speeds has the distinct advantage of providing a clear visual representation of both the torque response of the PGT and the power flow through it.

3. Nomographs for One-Degree of Freedom PGT's

A PGT has three members that can serve as either inputs or outputs; the first central gear, the last central gear and the planet(s) carrier or arm. If we lock one of the three elements of the PGT and prevent it from rotation, then we have two members that can transmit power to or from the external environment; one input and one output .In this case, since the power does not branch or meet, it follows that the input power is transmitted through every link of the PGT. The planets are considered as non-torque carrying links. It is well known that under static equilibrium, torques acting on the three elements of the one-degree of freedom PGT must be summed up to zero

$$T_A + T_L + T_F = 0 \tag{14}$$

Solving equations (9) and (14) yields

$$T_F = -R \cdot T_L \tag{15}$$

And

$$T_A = (R-1) \cdot T_L \tag{16}$$

Equations (15) and (16) express two of the external torques in terms of the third. **Figure 7** shows a nomograph for a PGT like the one shown in **Figure 3 (d)** when the first central gear is held stationary (S = 1) .

By summing the torques in the y-direction, equation (14) will be obtained.

Equations (9) and (14) together imply that the torque vectors on the two extremes of a one-degree of freedom nomograph, both must point upward or downward and that the third torque vector must point to the opposite direction.

After some examination, one finds that the torques shown in **Figure 7** can be solved for in the same way as forces on a rigid beam may be solved for.

Summing torque vectors moments about the zero of the ω_A axis yields

$$-T_L . E + T_F \cdot (1-E) = 0 \tag{17}$$

Solving for T_F and simplifying one arrives at equation (15).

Summing torque vectors moments about the zero of the ω_F axis generates the equation

$$T_A \cdot (1-E) - T_L \cdot 1 = 0 \tag{18}$$

Substituting equation (6) into equation (18), then solving for T_A and simplifying, one arrives at equation (16) which is the same as equation (9) when $\omega_F = 0$.

Summing torque vectors moments about the ω_L axis yields

$$T_F - T_A \cdot E = 0 \tag{19}$$

Similar results can be obtained when the last central gear is fixed (S = 0) or the planet(s) carrier or arm is fixed (S = E).

A nomograph for a one-degree of freedom PGT's (one of the three branches is fixed), allows designer to

(a)Define the kinematic relationships between the three branches of the PGT without first selecting a physical arrangement of gears.

(b)Yield equations completely defining the torques on the three branches of the PGT.

(c)Yield the energy balance equation for the PGT and provide a clear representation of the power flow through it.

DESIGN OF AUTOMATIC SPEED CHANGER

PGT's are considered adaptable to suit automatic internal speed changers. What follows is the design of such a gear train using the present new methodology with the help of nomographs.

The first step in the design process is to set fourth any design constraints on the design.

1. Design Constraints

In this paper, several requirements must be met for the design to be viable. These are:

1. The input and output speeds of the speed changer must rotate in the same direction.
2. The speed changer needs to possess three output speeds. They are in the rang of ±30 to ±40 % of the ideal pedaling speed (cadence) usually associated with efficient riding.
3. The speed changer needs also to possess the automatic shift function that replaces the manual or the electromechanical fashions.
4. The foot-board axle is connected with the shift mechanism's input axle.

5. Considering the effect of the output axle, the characteristic of one-way clutch is that the fast one-way clutch acts as the output. Considering the effect of the input axle, the slow one-way clutch acts as the input.

2. Design of Three-Speed Automatic Changer

In what follows one-way clutches and centrifugal clutches are reviewed

A one-way clutch (OWC) consists of an inner ring with a slanting flange, an outer ring and several rollers placed between the inner ring and the outer ring. If the one-way clutch inner ring is the driving part, the outer ring is the driven part. When the inner ring rotation speed is larger than that of the outer ring, the rollers will move to the high end of the slanting flange of the inner ring, thus causing the inner ring, outer ring and rollers of the one-way clutch to join together in one part and move as a rigid body.

By this movement, the one-way clutch transfers the driving power from the inner ring (driving part) to the outer ring (driven part). When the inner ring rotation speed is lower than that of the outer ring or has stopped, the rollers will move to the lower end of the slanting flange of the inner ring, thus causing the inner ring and outer ring to separate .The outer ring can now rotate freely in the same direction (or make circular movements. In this paper, **Figure 8(a)** shows the one-way clutch in the separated state, while **Figure 8(b)** shows the one-way clutch in the engaged state.

A centrifugal clutch consists of clutch crust, clutch blocks and clutch springs. The input axle and output axle are respectively connected with clutch blocks and clutch crust. The driving power is transferred to the driven axle by the centrifugal force caused by the clutch rotation. In this paper, **Figure 8(c)** shows the centrifugal clutch in the separated state whereas **Figure 8(d)** shows the centrifugal clutch in the engaged state.

PGT's combined with one-way clutches and centrifugal clutches, have led to the design of the automatic internal speed changer as shown in **Figure 9**.

To make the design of the system simpler, nomographs are used to quickly and simply remove ranges of R from consideration. Referring to the nomograph in figure10, the graph can be divided into three sections labeled in the figure. The sections represent the locations of the ωA axis for the cases of R listed in the figure. For example, if R=-0.5, E=⅓ from equation (6) and the ωA axis must lie between the ωL and ωF axes.

This nomograph gives the designer the ability to quickly select a broad range for R, depending on how the gear train is intended to react to typical input speeds.

Nomographs, like the one in **Figure 7** in which the first central gear is fixed, $\omega_F = 0$, allow the designers to visualize critical details of a design's response. The designer can freely select one torque and either the two remaining speeds or one rotational speed and the general transmission ratio; R. Nomographs provide some insight into the fundamental differences between these two approaches. In the first approach, one can visualize drawing a straight line between the selected speeds on the ω_F and ω_L axes and placing the ω_A axis such that it intersects this line at the selected rotational speed at the arm. In the latter approach, the designer would simply place the ω_A axis according to equation (6), draw a straight line between the two selected speeds, and read on the third axis the unspecified speed.

To search for an appropriate R, one can examine the first design constraint that the input and output speeds of the speed changer must have the same sign. **Figure 11** shows three nomographs for the system for the three ranges of R. Notice that when R is between negative infinity and zero and between zero and one, the rotational speeds ω_L and ω_A both have the same sign. Conversely when R is between one and infinity, the rotational speeds ω_L and ω_A have opposite signs.

With this range of R-values eliminated, the ranges that need to be considered during the design effort from here forward are (a) between negative infinity and zero and (b) between zero and one.

PGT's with R-values between one and infinity can have R-values between zero and one if the arbitrary designation of first and last gears is replaced by each other.

After establishing broad ranges of R-values to be considered, the next task is to attempt to apply the remaining design constraints to the governing equations of the system. Examining the second design constraint, using equation (9) and putting $\omega_F = 0$, we get, $T_A \omega_A + T_L \omega_L = 0$. It is convenient to rewrite this equation in the form

$$\frac{T_A}{T_L} = -\frac{\omega_L}{\omega_A} \tag{20}$$

Since the three output speeds of the speed changer are in the range of ±30 to ±40 % of the ideal pedaling speed as required by the second design constraint, and the input and output torques are inversely proportional to these speeds, as seen from equation (20), then the design will allow cyclist to pedal it his ideal cadence at different road conditions. The gear needed will depend on the slope of the road, the wind conditions and the cyclist own condition at any given time.

Aside from selection of a broad range for R, the nomograph also has the distinct advantage of providing a clear visual representation of both the torque response of the gear train and the power flow through the PGT.

Returning to **Figure 7**, one can visualize speeds, torques and power flow through the PGT from a single nomograph.

Since the human body provides most efficient power between speed of 80 and 100 rpm, it would be advantageous to select a gear ratio that would place the values of ωL and ωA in the range of ±30 to ±40 % of this speed. Obviously, as shown in Figure 11, a designer could select E between 0.21 to 0.29 (R = 0.23 to 0.28) or between − 0.39 to − 0.3 (R = − 0.4 to − 0.27) with a nominal cadence to force both speeds into the ideal region.

Nomographs allow engineers to define the above values and the relationships between the three branches of PGT without first selecting a physical arrangement of gears .

As can be seen from **Figure 12**, the value of ωA asymptotically approaches zero, while ωL increases linearly with decreasing R. The same results can be obtained from **Figures 12 and 13** as those obtained from **Figure 11**. **Figure 13** shows the output rotational speeds in terms of the basic transmission ratio R for nominal cadence.
,shown on the nomographs of **Figure 11**.

After selecting a gear ratio, it becomes important to decide upon one of the twelve possible arrangements of the gear trains. Returning to **Figure 3**, taking the arm to be fixed, one can quickly deduce by examination that the trains in quadrants (1) and (2) yield negative ratios between the first and last central gears while those in quadrants (3) and (4) yield positive ratios. In order to simplify the actual construction of the device, only the simple trains in Figure 3 will be considered.

3. Design of three-speed automatic speed changer of bicycle

By applying the above solution technique three types of three-speed automatic changers are created.

3.1 Plan 1

This design adopts the PGT shown in **Figure 3(d)** to achieve the front internal speed changer shown in figure 14. As required by the fourth design constraint, the foot board rotation axle is used as the input axle of the automatic speed changer.

The actuating maps of the three speeds of this plan are shown in **Figure 15 (a), (b) and (c)** with their corresponding nomographs.

At the beginning, the one-way clutch OWC-L, and one-way clutch OWC-A are engaged. The driving power is transmitted to the chain device via the last gear (ring gear), planet gear and arm of the PGT. When the speed reaches a preset level, the clutch C-L engages and the one-way clutch OWC-A separates. The driving power is then transmitted directly to the chain device. If the engine speed is increased again, the clutch C-A engages and the one-way clutch OWC-L separates. The driving power is transmitted to the chain device via the arm, planet gear and the last gear of the PGT.

3.2 Plan 2 and 3

These designs adopt another PGT's with positive train ratios to achieve the automatic three-speed front internal speed changers. Design 2 is shown in **Figure 16**. The actuating maps of the three speeds of this design are shown in **Figures 17 (a), (b) and (c)** with their corresponding nomographs. Nomographs are plotted for R=0.25 (E=-⅓) and 80 rpm cadence. Design 3 is shown in **Figure 18.** The actuating maps of the three speeds of this design are shown in **Figures 19 (a), (b) and (c)**. Since R=0.25 (E=-⅓) for this train which is the same as that for PGT of design 2, nomographs are the same for both of these trains and will not be repeated.

CONCLUTIONS

This study contributes to the development of a solution technique for the analysis of the kinematics, the torque distributions and the power flow through one-degree of freedom PGT's.

Nomographs are used as a design tool to enable the designer to simultaneously visualize these variables without first selecting a particular PGT and without the need to solve equations repeatedly.

After developing the solution technique and the design tool, these were applied to design a three-speed automatic speed changer. PGT's are considered adaptable to suit automatic internal speed changers. PGT's combined with one-way clutches and centrifugal clutches, have led to the design of the automatic bicycle speed changer. Three design examples were demonstrated to illustrate the methodology. The design can serve as a reference to industrial circles to develop three-speed automatic speed changers bicycles.

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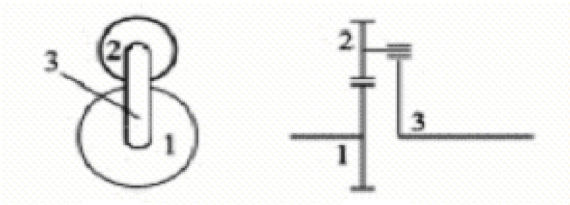


(a) (b)

Figure 1 (a) The elementary PGT and (b) its kinematical representation.

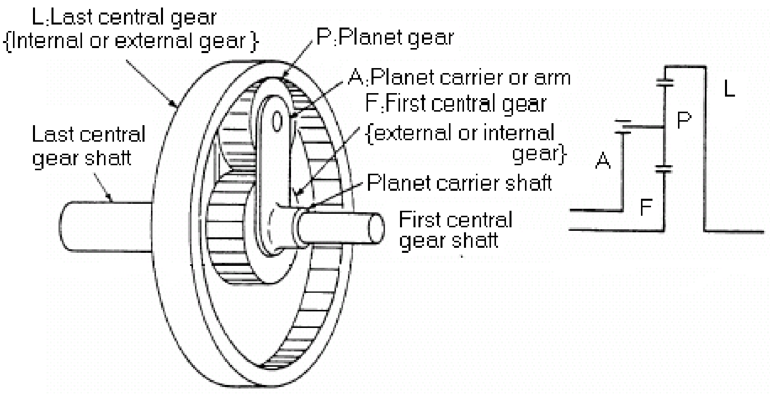


Figure 2 PGT of the lower arrangement.

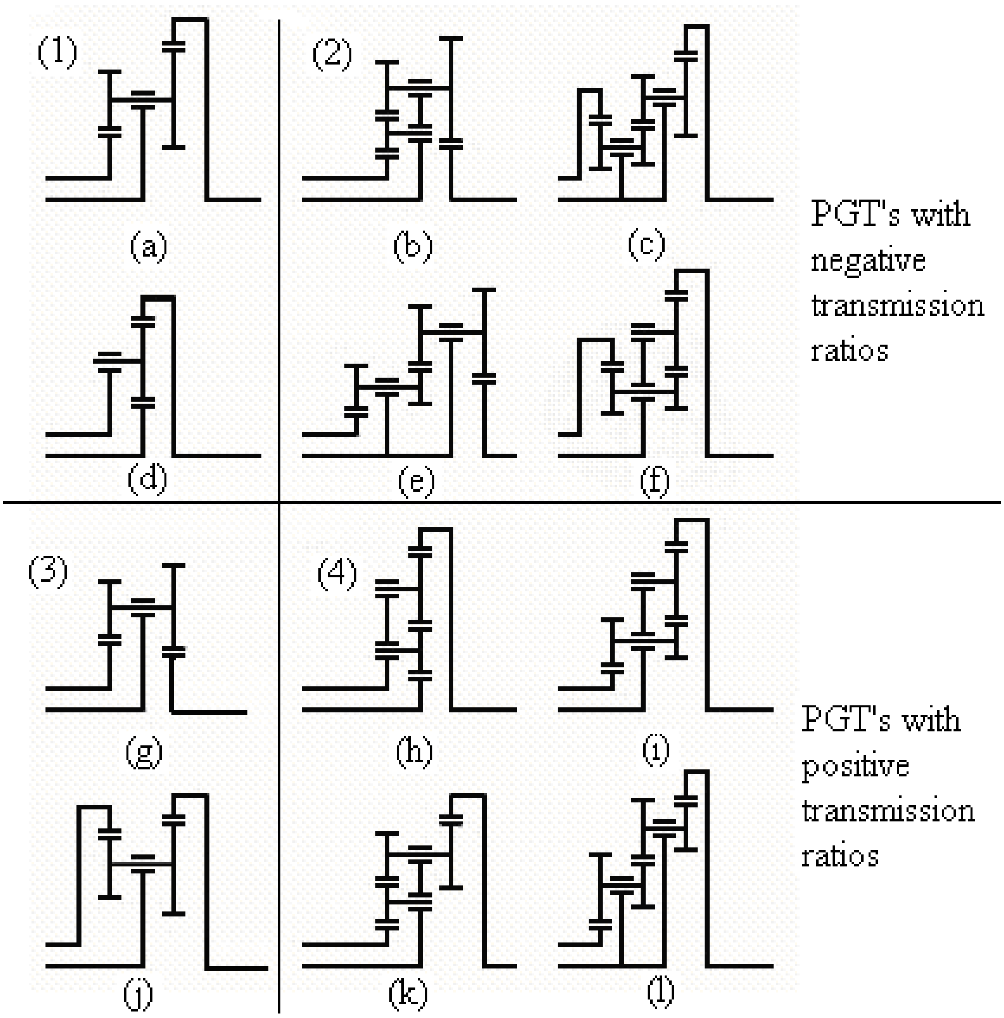


Figure 3 The simple, compound and complex PGT's as originally presented by Lévai.

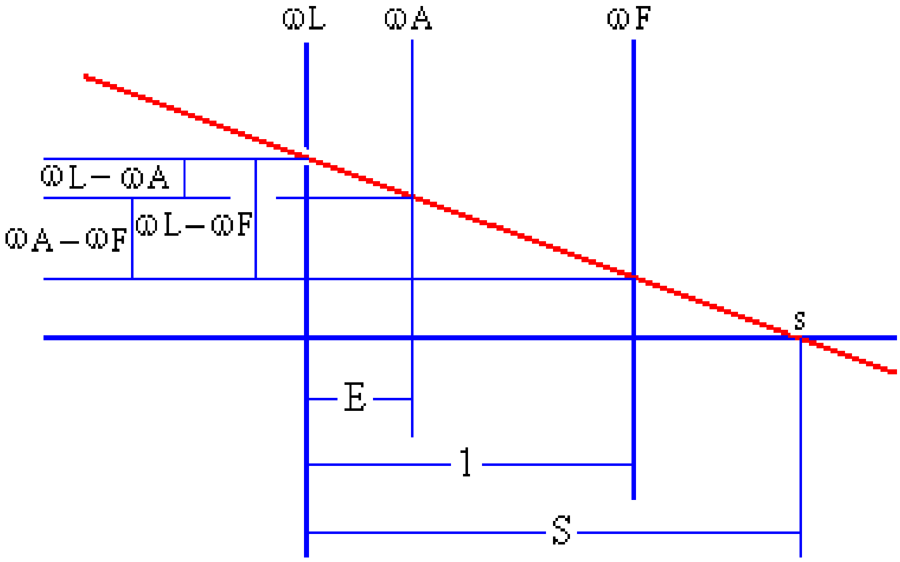


Figure 4 General layout of a nomograph for a PGT.

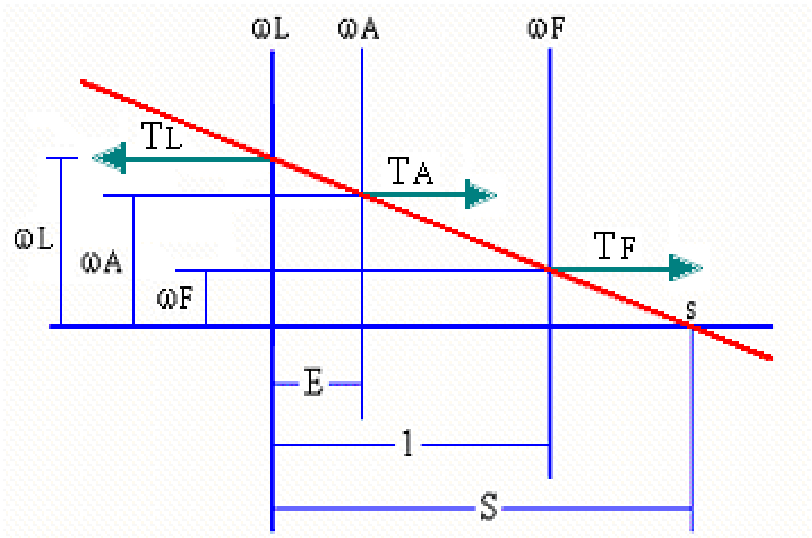


Figure 5 Nomograph with torques represented as vectors perpendicular to the rotational speeds axes.

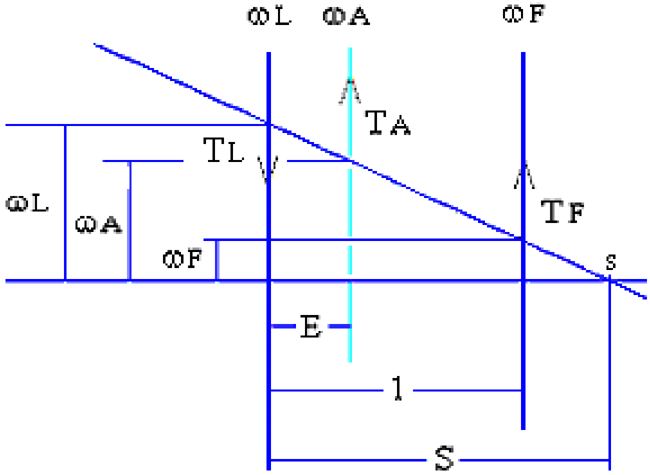


Figure 6 Nomograph with torques represented as vectors along the three axes.

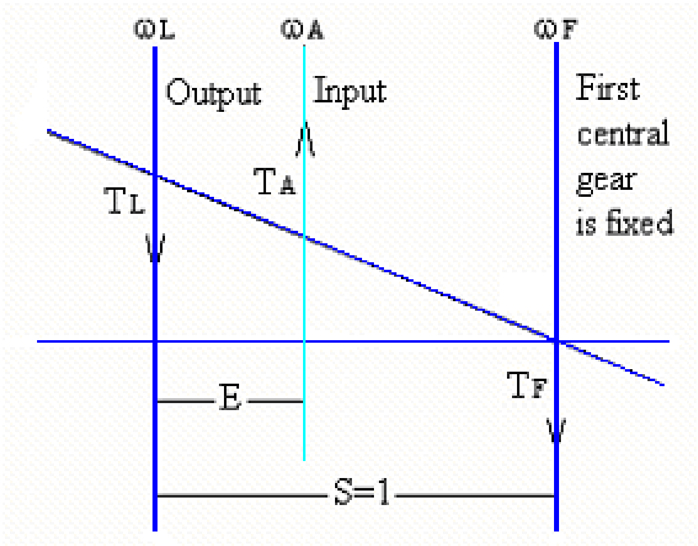


Figure 7 Nomograph for the PGT.

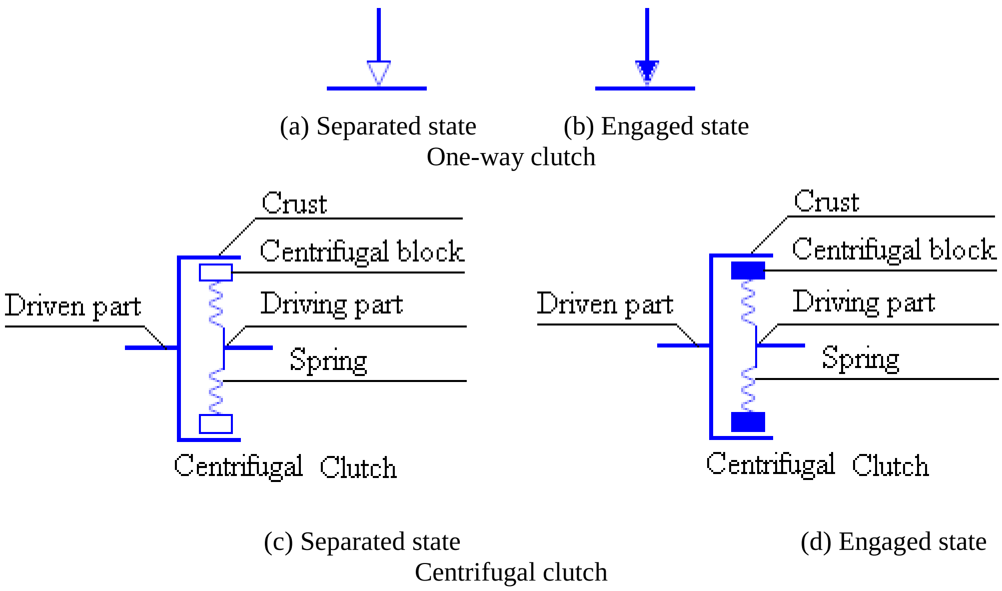


Figure 8 One-way clutch and centrifugal clutch in the engaged and separated states.

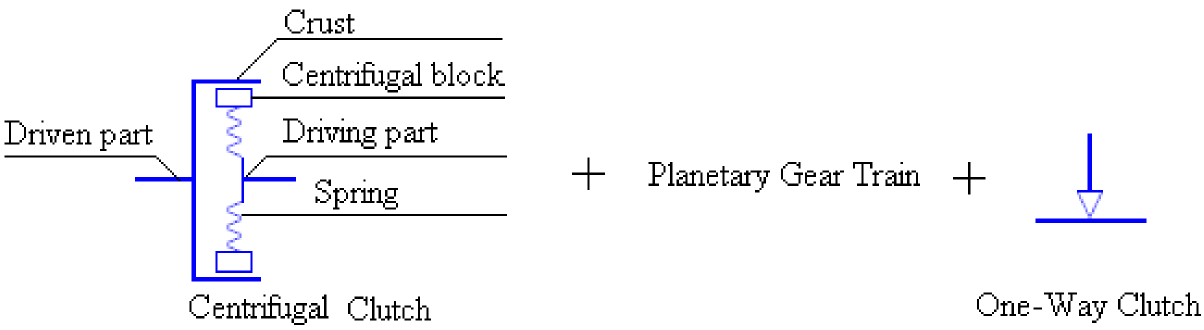


Figure 9 The automatic internal speed changer design.

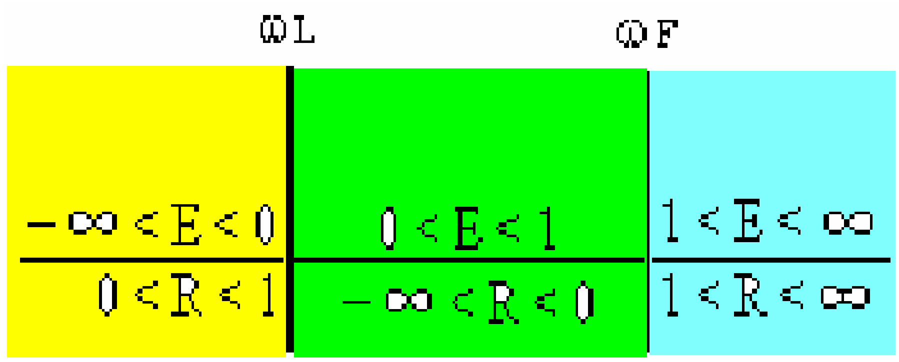


Figure 10 Nomograph with ranges of basic transmission ratio R labeled.

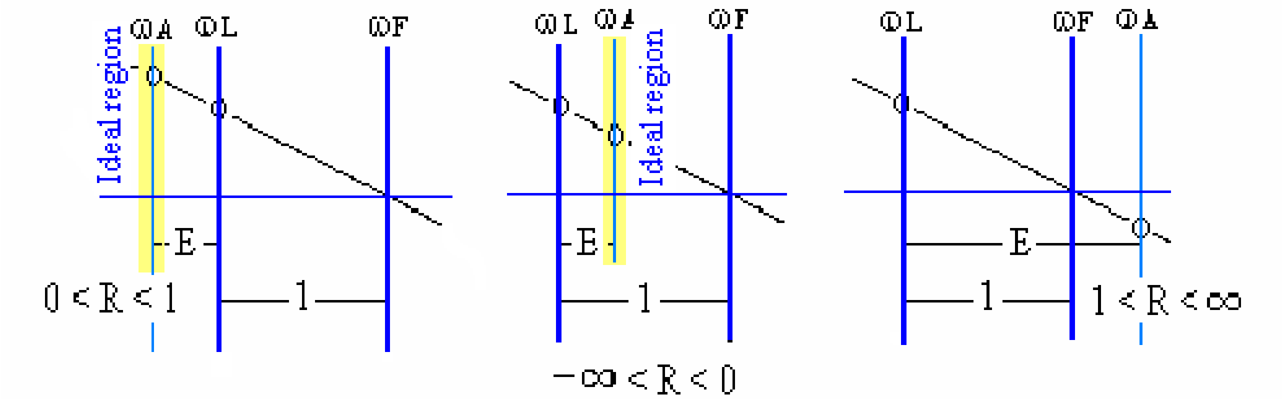


Figure 11 Nomographs for the system for the three ranges of R with ideal regions.

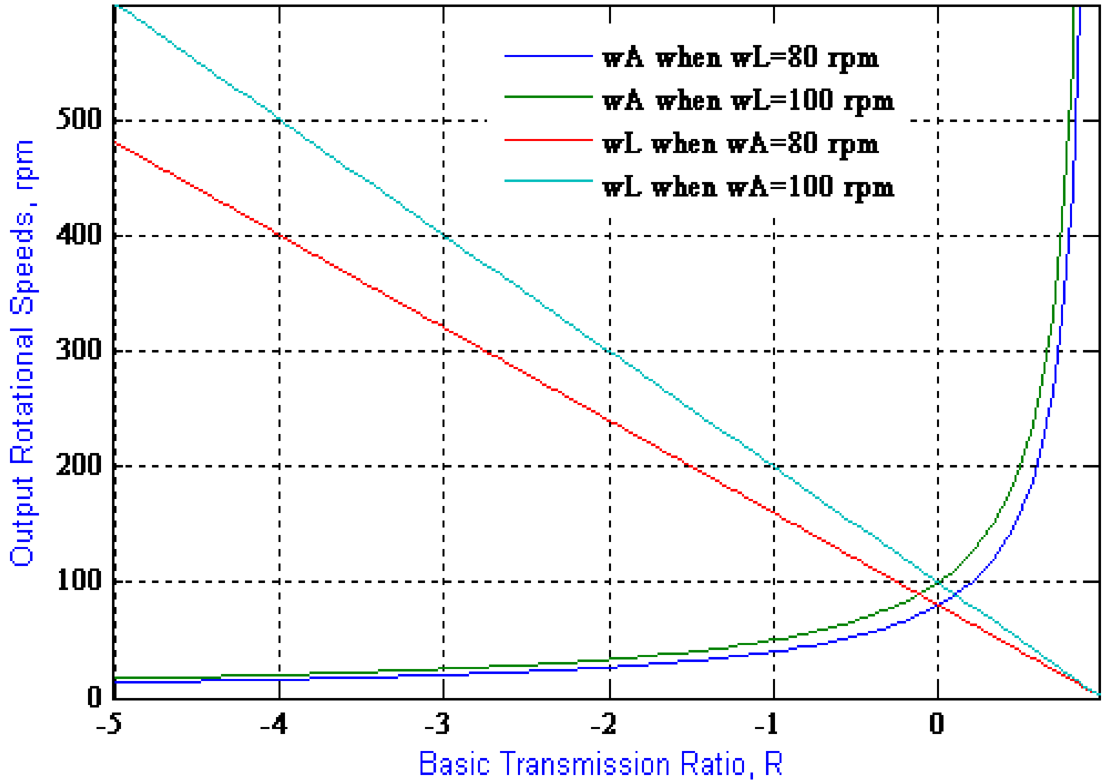


Figure 12 Output rotational speeds ωA and ωL as a function of train ratio R, for two pedaling speeds 80 and 100 rpm

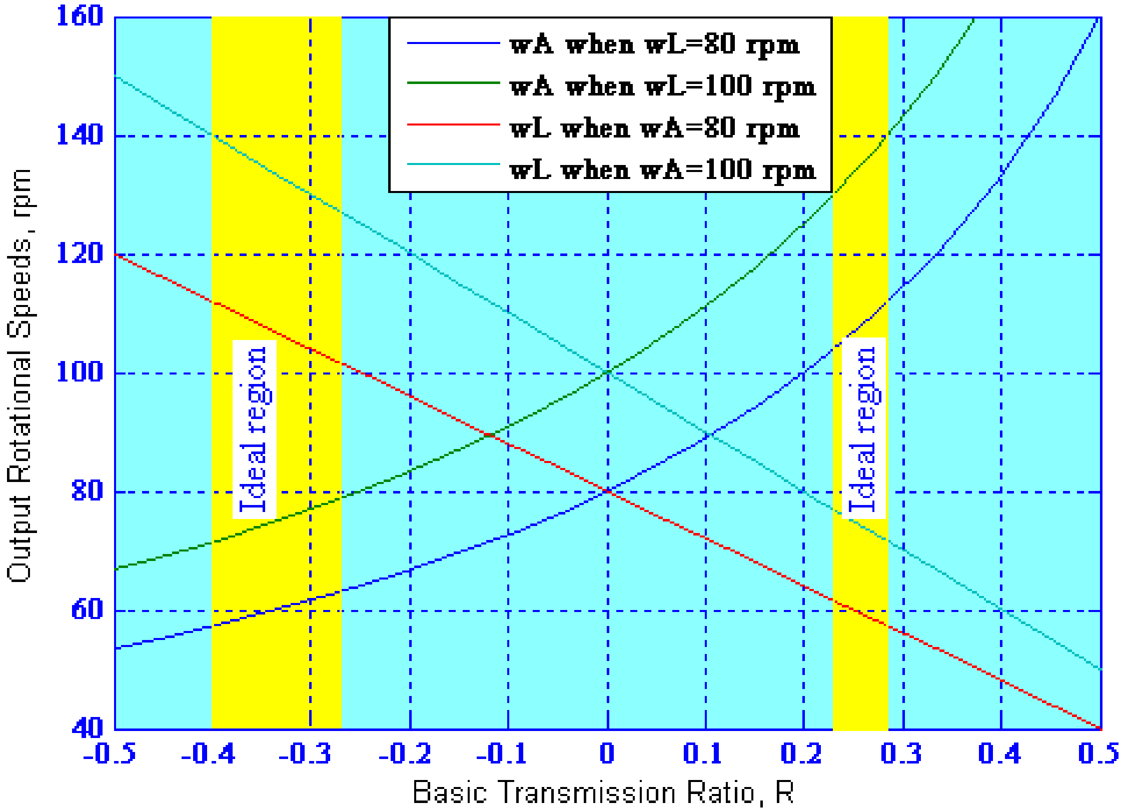


Figure 13 Ideal regions for the basic transmission ratio R

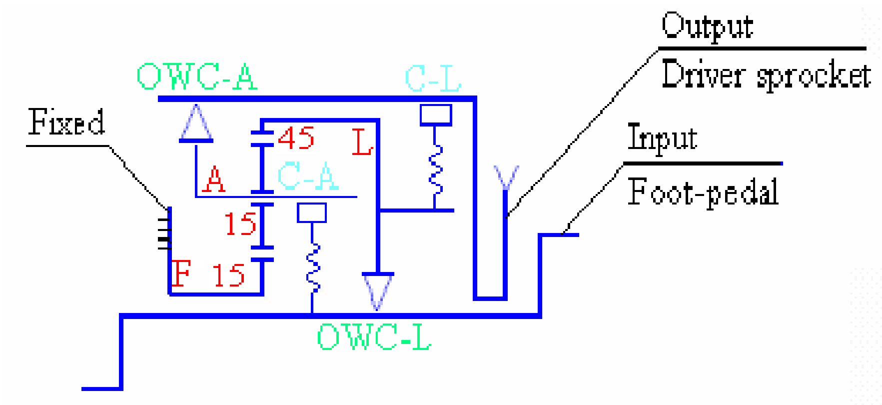
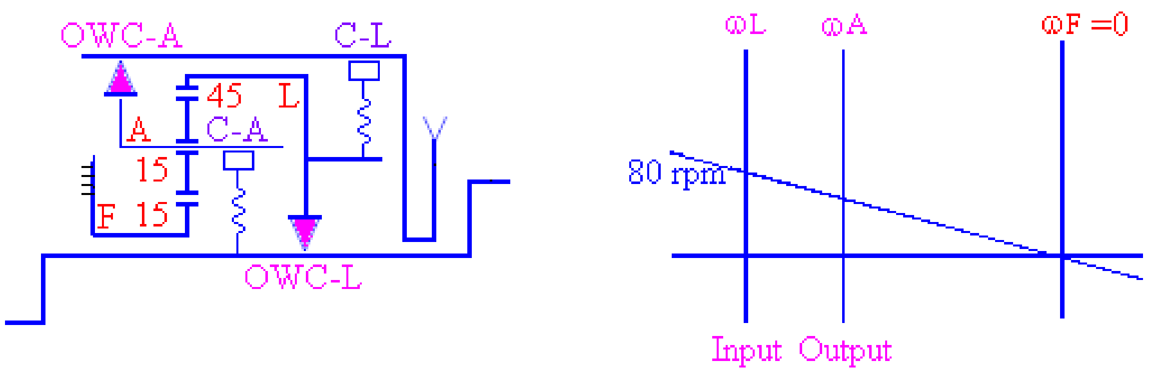
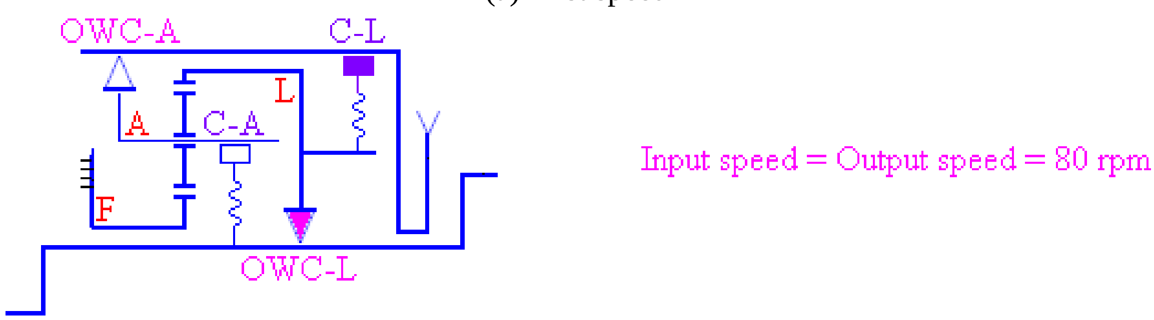


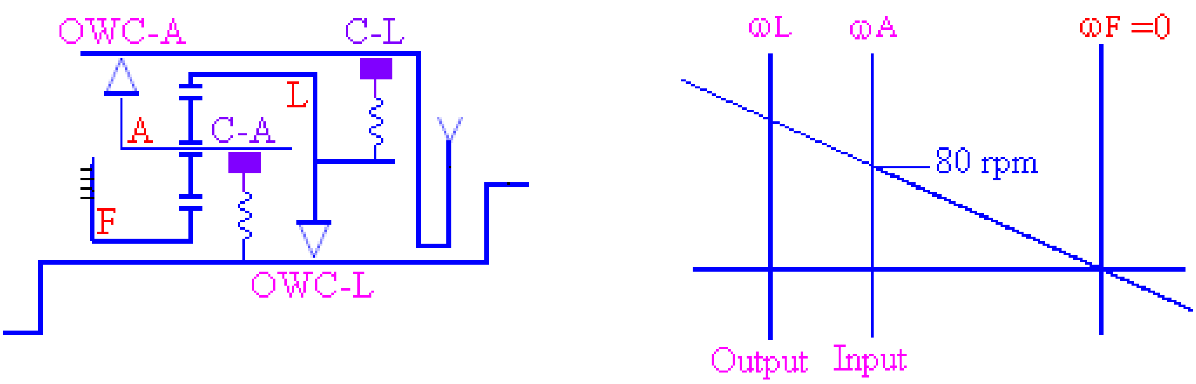
Figure 14 Design 1 for the bicycle three-speed automatic changer.



(a) First speed



(b) Second speed



(c) Third speed

Figure 15 The three speeds of the speed changer of plan 1.

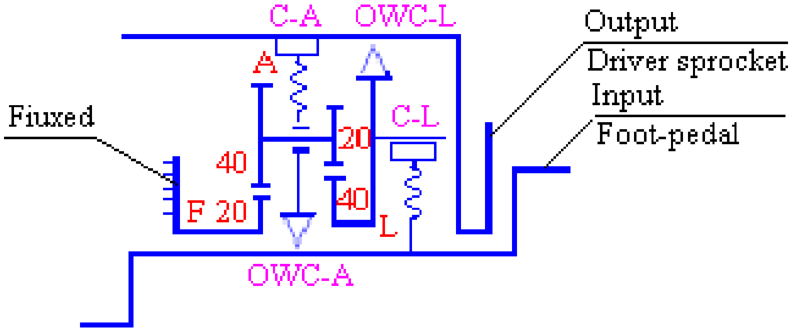
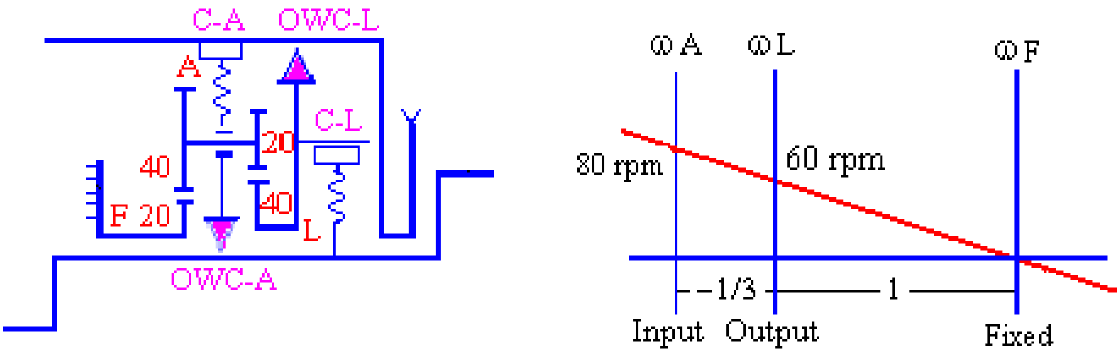
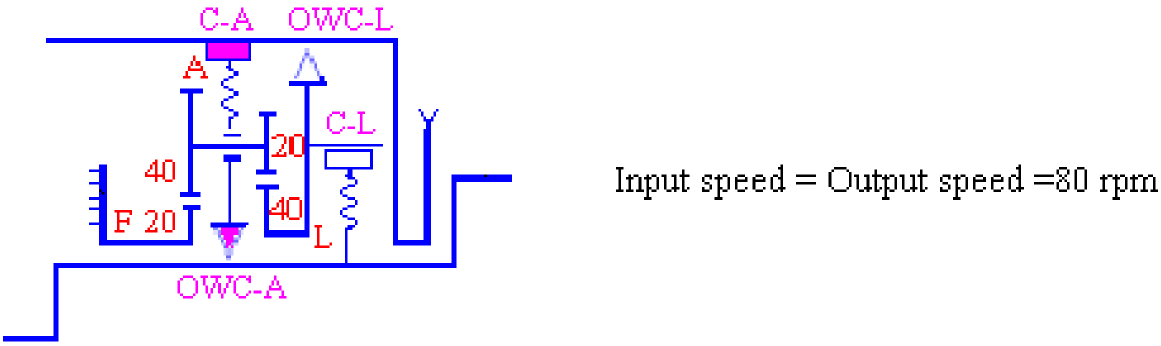


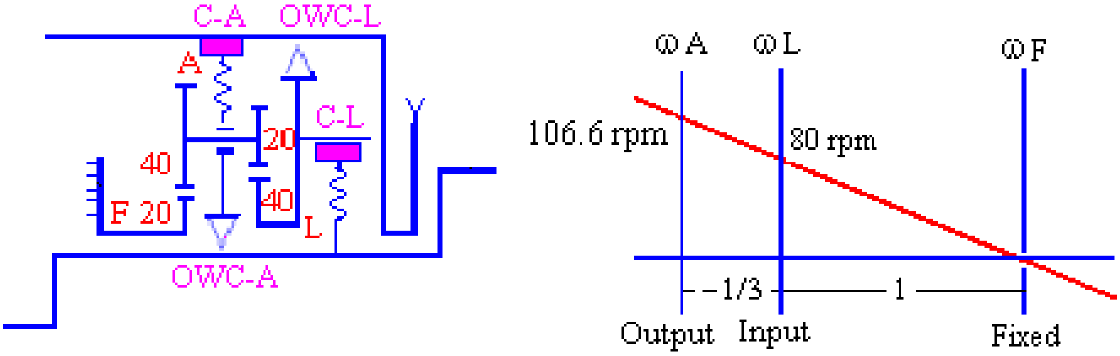
Figure 16 Design 2 for the bicycle three-speed automatic changer.



(a) First speed



(b) Second speed



(c) Third speed

Figure 17 The three speeds of the speed changer of plan 2.

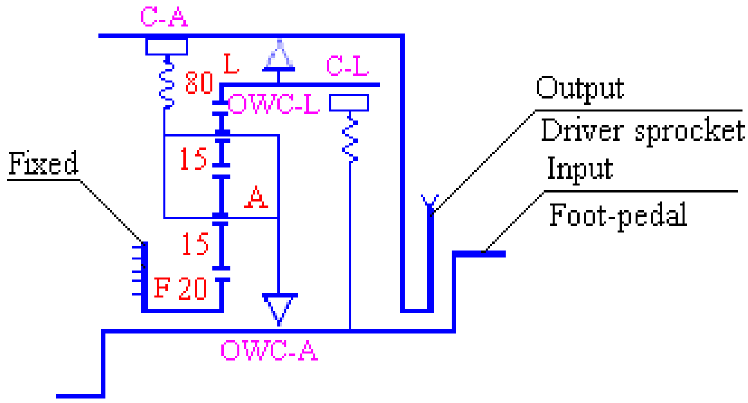
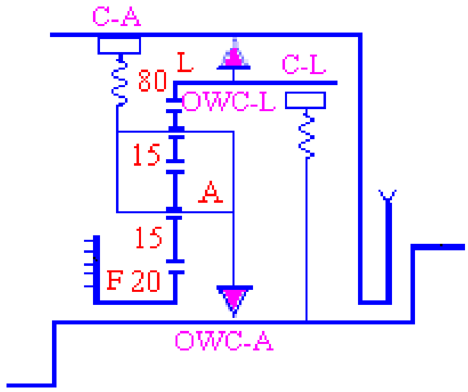
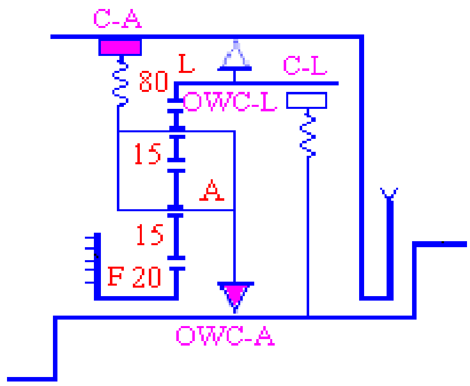


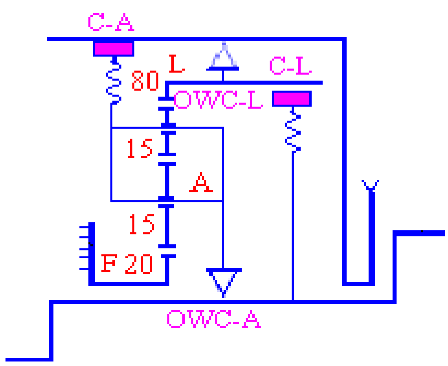
Figure 18 Design 3 for the bicycle three-speed automatic changer.



(a) First speed



(b) Second speed



(c) Third speed

Figure 19 The three speeds of the speed changer of plan 3.