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About the Gear Efficiency to a Simple Planetary Train

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Abstract: Synthesis of classical planetary mechanisms is usually based on kinematic relations, considering the achieved transmission ratio input-output. The planetary mechanisms are less synthesized based on their mechanical efficiency which is developed during operation, although this criterion is part of the real dynamics of mechanisms and also the most important criterion in terms of performance of a mechanism. Even when the efficiency criterion is considered, the determination of the planetary yield is made only with approximate relationships. The most widely recognized method is one method of Russian school of mechanisms. This paper will determine the method to calculate the mechanical efficiency of a planetary mechanism. The model will resolve one important problem of the dynamics of planetary mechanisms.

Keywords: Gears, Planetary Efficiency, Planetary Gear Kinematics, Planetary Synthesis, Simple Planetary Gear, Gears Dynamic, Toothed Wheel, Dynamic Synthesis, Forces, Velocities, Powers, Efficiency

Introduction

A transmission is a mechanical device for transmitting movement from one part to another. This element of the energy chain has for function the adaptation of the torque and of the speed between the driving member and the driven member.

The transmission of motion is one of the most common functions of the elements of general mechanics, that is to say, mechanical devices intended to replace the hand of man.

According to the mechanisms, the transmission is sized according to considerations concerning:

- The position of a part of the mechanism
- The desired movement
- The force, or the couple sought
- The power

The epicyclic gear train is a mechanical transmission device. It has the peculiarity of having two degrees of mobility, like the differential, that is to say it associates three trees with different rotation speeds with a single mathematical relationship: The speeds of two of the trees to know that of the third.

These trains are often used for speed reduction because of the large reduction ratios that this configuration permits evenly compact with a single gear. Such gearboxes can be found, in particular, in automatic gearboxes, hybrid vehicle engines (Hybrid Synergy Drive from Toyota), integrated bicycle gear hubs, electric geared motors, winches and double robotic gearboxes clutch.

The epicycloidal term comes from the trajectory following an epicycloid from a point of the satellites observed with respect to the inner planetary. However, a hypocycloid is observed if the reference of the movement is the outer planetary, which is often fixed in the reducers. This corresponds exactly to what the observer observes when watching a satellite.

The configuration adopted in the automotive differential. The axis of rotation of the satellites (often in pairs) is perpendicular to that of the planets. As a result, the gears are conical.

If the satellite is stationary with respect to the satellite door, the two planets have the same rotation speed. When the planets shift in speed, the satellite rotates while transmitting power:

- Applications
- Blaise Pascal calculating machine: Pascaline
- Hub with integrated bicycle speeds
- Car differentials
- Most automatic automotive gearboxes and some mechanical gearboxes (Ford model T for example)

- The electro-mechanical transmission of the Toyota Prius and more generally of the hybrids Toyota and Lexus (HSD system)
- Wind turbine frequency multipliers
- Garden tools (plant crushers, for example)
- The helicopters

This is mainly used for epicyclic trains. They are present in automatic gearboxes and in many gearboxes coupled to electric motors. They appear in the same catalogs as the latter. Their geometry gives an output shaft coaxial to the input shaft, which facilitates its implementation. Finally, they have a great ability to reduce speed. In general, three satellites are placed on the satellite carrier. Thus, the forces in the gears are not taken up by the bearings. As a result, these reducers are very suitable for the transmission of large torques.

These devices are sometimes used as a multiplier, as on wind turbines. Here again, it is their compactness and the absence of radial force induced in the bearings of the input and output shafts which justifies their use.

From the simplest train (type I), mobility is eliminated by fixing the outer sun gear, which is also called the crown.

The input shaft, while rotating, forces the satellite to roll inside the crown. In its movement, this one drives the satellite door as if it were a crank. The planet carrier constitutes the output shaft of the device. In this configuration the output rotates in the same direction and slower than the input.

Planetary gears have a number of advantages compared to transmission with fixed axis. Under similar operating conditions, the planetary transmissions serve longer and produce less noise as compared to a fixed shaft transmission (Cao *et al*., 2013; Lee, 2013; Garcia *et al*., 2007). Gearboxes are used to reduce the turbine rotational speed from generator's speed at performances of up to 110 MW or turbine rotational speeds of approx. About 60,000 rpm Parallel shaft gear units are usually used in these applications. However, some packagers and generator manufacturers use planetary gear units for gas and steam turbine systems. The majority of planetary gear unit manufacturers are located in Europe and United States.

Synthesis of classical planetary mechanisms is usually based on kinematic relations (Anderson and Loewenthal, 1986), considering mainly the achieved transmission ratio input-output. The most common model used is the differential planetary mechanism showed in Fig. 1. The formula 1 is determined by the relationship Willis (1').

For the various cinematic planetary systems presented in Fig. 3, where entry is made by the planetary carrier (H) and output is achieved by the final element (f), the initial element being usually immobilized, will be used for the kinematic calculations the relationships generalized 1 and 2.

The planetary mechanisms are less synthesized based on their mechanical efficiency developed during operation, although this criterion is part of the real dynamics of mechanisms and also the most important criterion in terms of performance of a mechanism.

Even when the efficiency criterion is considered, the determination of the planetary yield is made only with approximate relationships (Antonescu, 1979; Pelecudi *et al*., 1985; Pennestrì and Freudenstein, 1993), or having a particular character (not generalized) (del Castillo, 2002; Cho *et al*., 2006). The most widely recognized methods are the method of Russian school of mechanisms (Artobolevski, 1992) or (Martin, 1981). This paper will determine the real efficiency of the planetary trains by determining the exact method of calculation, (Petrescu and Petrescu, 2011; Petrescu, 2012).

Kinematic Synthesis

Synthesis of classical planetary mechanisms is usually based on kinematic relations, considering in especially the transmission ratio input-output achieved. The most common model used is the differential planetary mechanism as showed in Fig. 1.

For this mechanism to have one single degree of mobility, remaining in use with a drive unique and single output, it is necessary to reduce the mobility of the mechanism from two to one, which can be obtained by connecting in series or parallel of two or more planetary gear, by binding to gears with fixed axes, or the hardening of one of its mobile elements; element 1 in this case (case in which the wheel 1 is identified with the fixed element 0; Fig. 2).

The entrance to the simple planetary shown in Fig. 2 is made by the planetary carrier (H) and the output is done by the mobile cinematic element (3), the wheel (3). Kinematic ratio between the input-output (H-3) can be written as shown in relationship 1:

$$i_{H3}^{1} = \frac{1}{i_{3H}^{1}} = \frac{1}{1 - i_{31}^{H}} = \frac{1}{1 - \frac{1}{i_{13}^{H}}} \tag{1}$$

where, i_{13}^{H} is the ratio of transmission input output, corresponding to the mechanism with fixed axis (when the planetary carrier H is fixed) and is determined in function of the cinematic schematic of planetary gear used; for the model in Fig. 2 it is determined by the relation 2, depending on the numbers of teeth of the wheels 1, 2, 2 'and 3:

$$i_{13}^{H} = \frac{z_2}{z_1} \cdot \frac{z_3}{z_{2'}} \tag{2}$$

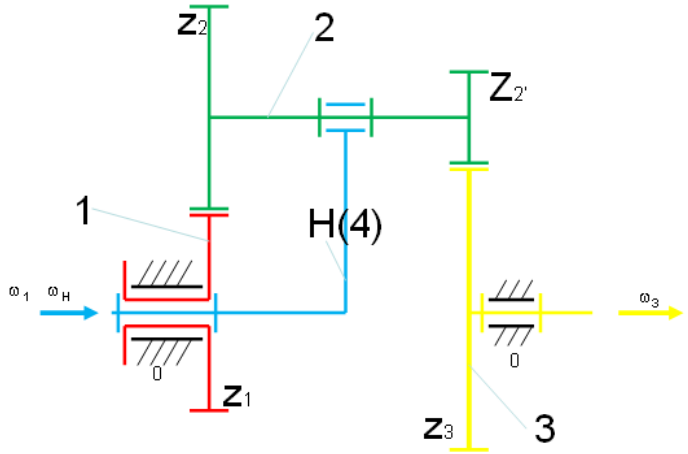

Fig. 1. Kinematic schematic of a differential planetary mechanism (M = 2)

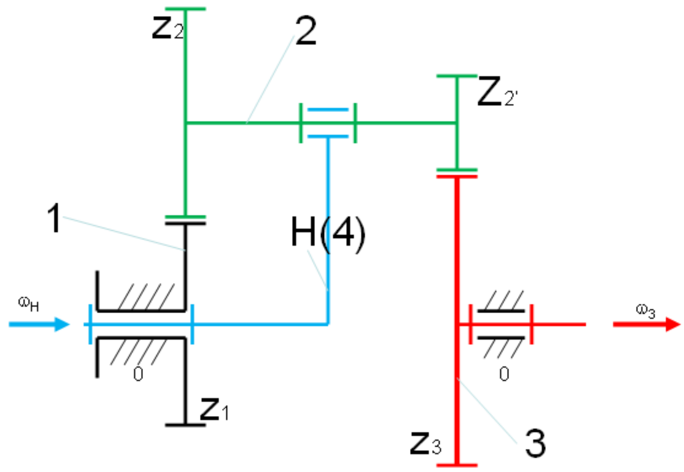

Fig. 2. Kinematic schematic of a simple planetary mechanism (M = 1)

Usually the formula 1 is determined by the relationship Willis (1'):

$$\begin{cases} i_{13}^{H} = \dfrac{\omega_1 - \omega_H}{\omega_3 - \omega_H} \equiv \dfrac{z_2}{z_1} \cdot \dfrac{z_3}{z_{2'}} \\ \dfrac{z_2}{z_1} \cdot \dfrac{z_3}{z_{2'}} = \dfrac{\dfrac{\omega_1}{\omega_H} - \dfrac{\omega_H}{\omega_H}}{\dfrac{\omega_3}{\omega_H} - \dfrac{\omega_H}{\omega_H}} \\ i_{13}^{H} = \dfrac{z_2 \cdot z_3}{z_1 \cdot z_{2'}} = \dfrac{0-1}{\dfrac{\omega_3}{\omega_H} - 1} = \dfrac{1}{1 - i_{3H}} = \dfrac{1}{1 - \dfrac{1}{i_{H3}^{1}}} \Rightarrow \\ \Rightarrow i_{H3}^{1} = \dfrac{1}{1 - \dfrac{1}{i_{13}^{H}}} \end{cases} \quad (3)$$

For the various cinematic planetary systems presented in Fig. 3, where entry is made by the planetary carrier (H) and output is achieved by the final element (f), the initial element which is usually immobilized will be used for the kinematic calculations in the relationships generalized 1 and 2; the relationship 1 takes the general form 3 and the relation 2 is written in one of the forms 4 particularized for each schematic separately, used; where i become 1 and f takes the value 3 or 4 as appropriate (Petrescu and Petrescu, 2011; Petrescu, 2012):

$$i_{Hf}^{i} = \frac{1}{i_{fH}^{i}} = \frac{1}{1 - i_{fi}^{H}} = \frac{1}{1 - \dfrac{1}{i_{if}^{H}}} \quad (4)$$

The automatic transmissions have been added slowly from airplanes to automobiles and were then generalized to various vehicles.

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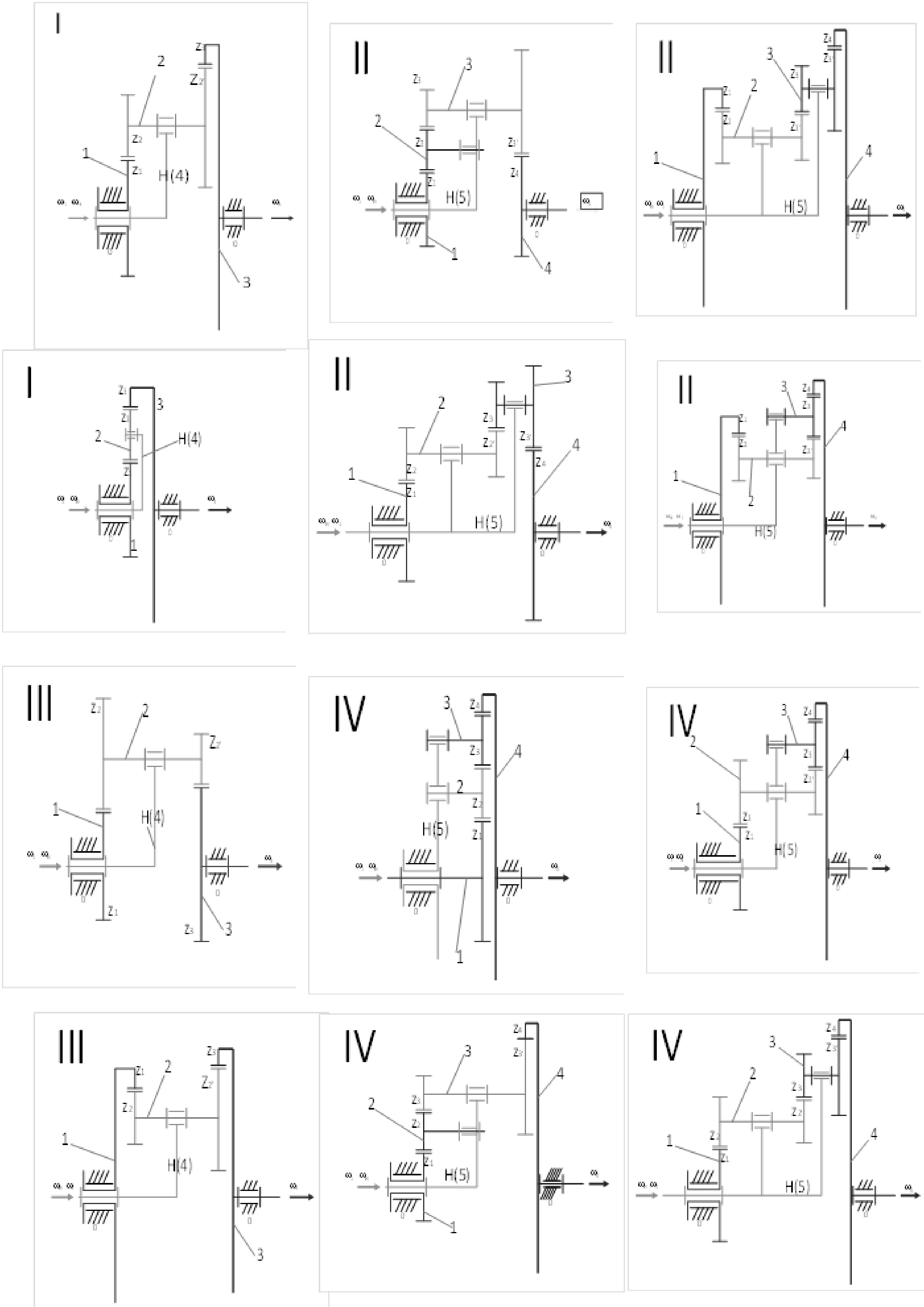

Fig. 3. Planetary systems

By using the formulas indicated in this study for calculating the dynamics of planetary mechanisms, planetary trains and planetary systems, used in aircraft and vehicles, their automatic transmissions can be achieved better than those known today.

$$\begin{cases} i_{13}^H = -\frac{z_2}{z_1}\cdot\frac{z_3}{z_{2'}} \text{ for I of up} \\ i_{13}^H = -\frac{z_3}{z_1} \text{ for I of down} \\ i_{13}^H = \frac{z_2}{z_1}\cdot\frac{z_3}{z_{2'}} \text{ for III of up} \\ i_{13}^H = \frac{z_2}{z_1}\cdot\frac{z_3}{z_{2'}} \text{ for III of down} \\ i_{14}^H = -\frac{z_3}{z_1}\cdot\frac{z_4}{z_{3'}} \text{ for II left up} \\ i_{14}^H = -\frac{z_2}{z_1}\cdot\frac{z_3}{z_{2'}}\cdot\frac{z_4}{z_{3'}} \text{ for II right up} \\ i_{14}^H = -\frac{z_2}{z_1}\cdot\frac{z_3}{z_{2'}}\cdot\frac{z_4}{z_{3'}} \text{ for II left down} \\ i_{14}^H = -\frac{z_2}{z_1}\cdot\frac{z_4}{z_{2'}} \text{ for II right down} \\ i_{14}^H = \frac{z_4}{z_1} \text{ for IV left up} \\ i_{14}^H = \frac{z_2}{z_1}\cdot\frac{z_4}{z_{2'}} \text{ for IV right up} \\ i_{14}^H = \frac{z_3}{z_1}\cdot\frac{z_4}{z_{3'}} \text{ for IV left down} \\ i_{14}^H = \frac{z_2}{z_1}\cdot\frac{z_3}{z_{2'}}\cdot\frac{z_4}{z_{3'}} \text{ for IV right down} \end{cases} \tag{5}$$

This model resolves one important problem of the dynamics of planetary mechanisms. A complete dynamic may need to be determined and also the dynamics deformation mechanisms (Fig. 4) but this is not part of the subject matter of this paper.

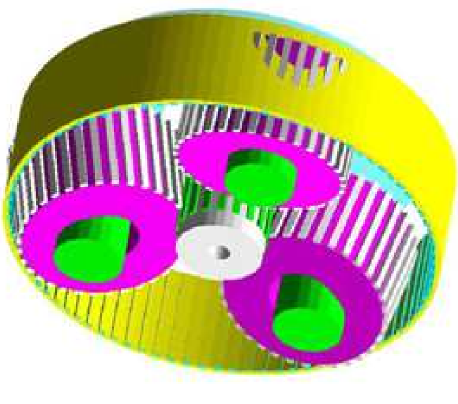

Fig. 4. The deformation of a planetary mechanism

Dynamic Synthesis Based on Performance Achieved

Dynamic synthesis of planetary trains (gears) based on performance achieved can be made with the original relationships presented in system 5 (Petrescu and Petrescu, 2011; Petrescu, 2012):

$$\begin{cases} \begin{cases} i_{13}^H = \frac{\omega_1-\omega_H}{\omega_3-\omega_H} \Rightarrow \omega_3-\omega_H = \frac{\omega_1-\omega_H}{i_{13}^H} \Rightarrow \\ \omega_3 = \frac{\omega_H\cdot\left(i_{13}^H-1\right)+\omega_1}{i_{13}^H} \text{ and with } \omega_1 = 0 \Rightarrow \\ i_{13}^H = \frac{z_2\cdot z_3}{z_1\cdot z_{2'}} \qquad \Rightarrow \omega_3 = \frac{\omega_H\cdot\left(i_{13}^H-1\right)}{i_{13}^H} \end{cases} \\ P_i + P_o = 0 \\ \begin{cases} M_1\cdot\omega_1^*\cdot\left(\eta_{13}^H\right)^x + M_3\cdot\omega_3^* = 0 \Rightarrow \\ \eta_{13}^x = \frac{-M_3\cdot\omega_3^*}{M_1\cdot\omega_1^*} \Rightarrow \eta_{13}^x = \frac{-M_3}{M_1}\cdot\frac{\omega_3-\omega_H}{\omega_1-\omega_H} \Rightarrow \\ \Rightarrow \eta_{13}^x = \frac{-M_3}{M_1}\cdot\frac{\omega_3-\omega_H}{-\omega_H} \Rightarrow \\ \eta_{13}^x = \frac{-M_3}{M_1}\cdot\frac{1}{i_{13}^H} \Rightarrow M_1 = \frac{-M_3}{\eta_{13}^x\cdot i_{13}^H} \end{cases} \\ M_1+M_3+M_H = 0 \Rightarrow M_H = -\left(M_1+M_3\right) \\ = -\left(\frac{-M_3}{\eta_{13}^x\cdot i_{13}^H}+M_3\right) = -M_3\cdot\frac{\eta_{13}^x\cdot i_{13}^H-1}{\eta_{13}^x\cdot i_{13}^H} \\ \begin{cases} M_H\cdot\omega_H\cdot\eta_{H3}^1 + M_3\cdot\omega_3 = 0 \Rightarrow \eta_{H3}^1 = \frac{-M_3\cdot\omega_3}{M_H\cdot\omega_H} \Rightarrow \\ \eta_{H3} = \frac{-M_3\cdot\omega_3\cdot\eta_{13}^x\cdot i_{13}^H}{-M_3\cdot\left(\eta_{13}^x\cdot i_{13}^H-1\right)\cdot\omega_H} \Rightarrow \\ \Rightarrow \eta_{H3} = \frac{\left(\omega_H-\frac{\omega_H}{i_{13}^H}\right)\cdot\eta_{13}^x\cdot i_{13}^H}{\left(\eta_{13}^x\cdot i_{13}^H-1\right)\cdot\omega_H} \Rightarrow \\ \eta_{H3} = \frac{\left(i_{13}^H-1\right)\cdot\eta_{13}^x}{\left(\eta_{13}^x\cdot i_{13}^H-1\right)} \Rightarrow \eta_{H3} = \eta_{13}^x\cdot\frac{\left(i_{13}^H-1\right)}{\left(\eta_{13}^x\cdot i_{13}^H-1\right)} \\ \begin{cases} x = 1 \text{ for } i_{13}^H \le 1 \;;\; i_{13}^H - 1 \le 0 \\ x = -1 \text{ for } i_{13}^H > 1 \;\; i_{13}^H - 1 > 0 \end{cases} \end{cases} \end{cases} \tag{6}$$

For a normally planetary system (Fig. 2) the mechanical efficiency can be determined based on system 5, which shows that the power between input-output is conserve. With Willis one determines w3. It uses then the conservation of power between input-output for a simple planetary gear apparently fixed axis (1-input, 3-output), (Petrescu and Petrescu, 2011; Petrescu, 2012). Continue with relative preservation of moments on an axis

(Pelecudi *et al*., 1985; Petrescu and Petrescu, 2011; Petrescu, 2012) and finally write again simple planetary gear power conservation (Pelecudi *et al*., 1985).

Useful Used Relations

For the calculation of return on fixed axis mechanism consider these three relationships, 6-8 (Petrescu and Petrescu, 2014a-c):

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

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

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

Discussion

The epicyclic gear train is a mechanical transmission device. It has the peculiarity of having two degrees of mobility, like the differential, that is to say it associates three trees with different rotation speeds with a single mathematical relationship: The speeds of two of the trees to know that of the third.

These trains are often used for speed reduction because of the large reduction ratios that this configuration permits evenly compact with a single gear. Such gearboxes can be found, in particular, in automatic gearboxes, hybrid vehicle engines (Hybrid Synergy Drive from Toyota), integrated bicycle gear hubs, electric geared motors, winches and double robotic gearboxes clutch. The epicycloidal term comes from the trajectory following an epicycloid from a point of the satellites observed with respect to the inner planetary. However, a hypocycloid is observed if the reference of the movement is the outer planetary, which is often fixed in the reducers. This corresponds exactly to what the observer observes when watching a satellite.

The configuration adopted in the automotive differential. The axis of rotation of the satellites (often in pairs) is perpendicular to that of the planets. As a result, the gears are conical.

If the satellite is stationary with respect to the satellite door, the two planets have the same rotation speed. When the planets shift in speed, the satellite rotates while transmitting power.

Synthesis of classical planetary mechanisms is usually based on kinematic relations, considering the achieved transmission ratio input-output. The planetary mechanisms are less synthesized based on their mechanical efficiency which is developed during operation, although this criterion is part of the real dynamics of mechanisms and also the most important criterion in terms of performance of a mechanism. Even when the efficiency criterion is considered, the determination of the planetary yield is made only with approximate relationships. The most widely recognized method is one method of Russian school of mechanisms. This paper will determine the method to calculate the mechanical efficiency of a planetary mechanism. The model will resolve one important problem of the dynamics of planetary mechanisms. Major producers provide power transmission solutions utilizing gear drive and brake products for industrial and mobile equipment all over the world. Today we need to understand the necessity of keeping businesses running with a minimum of inventory and maintaining an excellent track record for on-time delivery and providing expedited services.

Planetary gear drives provide the reliability and performance needed in the demanding mobile and industrial applications. Combine modern planetary drives with electric or hydraulic motors for the most dependable and efficient solution in swing, drive and lift applications. Boom rotation, slewing drives, winch drives, demolition equipment and conveyor drives are just some of the applications of modern planetary gear drives and multi disc brakes. Planetary mechanisms have become indispensable in the modern industry. Used as automatic gearboxes are becoming increasingly prevalent and even indispensable. The automatic transmissions have been added slowly from airplanes to automobiles and were then generalized to various vehicles. The major problem that arises is their dynamic design judicious and particularly to operate with very high efficiency. Paper presented address all these goals simultaneously. Dynamic synthesis of planetary trains (gears) based on achieved performance can be made with the relationships presented above. The used calculation program written in Excel can be found on appendix (Fig. 9).

Examples of Calculation

Next, some examples of calculation are presented in the four Fig. 5-8. It analyzed the mechanism model shown in Fig. 2. Input-output gear ratio achieved, iH3, is: 11 (Fig. 5), -10 (Fig. 6), 5 (Fig. 7), -0.2 (Fig. 8). On the left side of a Fig. using standard pressure angle of 20 degrees [deg] and the right of every Fig. uses a pressure angle decreased to 10 degrees [deg].

ηH3	0.25622	ηH3	0.37708
i13H	1.1	i13H	1.1
x	-1	x	-1
iH3	11	iH3	11
z1	42	z1	42
z2	42	z2	42
z2'	40	z2'	40
z3	44	z3	44
alfa012 [deg]	20	alfa012 [deg]	10
alfa023 [deg]	20	alfa023 [deg]	10
beta12 [deg]	15	beta12 [deg]	15
beta23 [deg]	15	beta23 [deg]	15

Fig. 5. i13H=1.1

ηH3	0.18391	ηH3	0.32408
i13H	0.90909	i13H	0.90909
x	1	x	1
iH3	-10	iH3	-10
z1	42	z1	42
z2	42	z2	42
z2'	44	z2'	44
z3	40	z3	40
alfa012 [deg]	20	alfa012 [deg]	10
alfa023 [deg]	20	alfa023 [deg]	10
beta12 [deg]	15	beta12 [deg]	15
beta23 [deg]	15	beta23 [deg]	15

Fig. 6. i13H=0.9(09)

ηH3	0.45429	ηH3	0.57698
i13H	1.25	i13H	1.25
x	-1	x	-1
iH3	5	iH3	5
z1	32	z1	32
z2	40	z2	40
z2'	36	z2'	36
z3	36	z3	36
alfa012 [deg]	20	alfa012 [deg]	10
alfa023 [deg]	20	alfa023 [deg]	10
beta12 [deg]	15	beta12 [deg]	15
beta23 [deg]	15	beta23 [deg]	15

Fig. 7. i13H=1.25

ηH3	0.68961	ηH3	0.84792
i13H	1.6666	i13H	1.16666
x	1	x	1
iH3	-0.2	iH3	-0.2
z1	48	z1	48
z2	32	z2	32
z2'	64	z2'	64
z3	16	z3	16
alfa012 [deg]	20	alfa012 [deg]	10
alfa023 [deg]	20	alfa023 [deg]	10
beta12 [deg]	15	beta12 [deg]	15
beta23 [deg]	15	beta23 [deg]	15

Fig. 8. i13H=1.(6)

A	B	C	D
ηH3	=B34^B35*(1-B33)/(1-B34^B35*B33)		
i13H	=B33		
x	=B35		
z1	20		
z2	30		
z2'	20		
z3	20		
alfa012 [deg]	20		
alfa023 [deg]	20		
beta12 [deg]	15		
beta23 [deg]	15		
sign wheel teeth z1	1	1=ext	-1=int
sign wheel teeth z2'	1	1=ext	-1=int
sign gear 12	1	1=ext	-1=int
sign gear 23	1	1=ext	-1=int
alfa012 [rad]	=B9*PI()/180		
alfa023 [rad]	=B10*PI()/180		
beta12 [rad]	=B11*PI()/180		
beta23 [rad]	=B12*PI()/180		
cos(beta12)	=COS(B20)		
tan(alfa012)	=TAN(B18)		
tan(beta12)	=TAN(B20)		
ε12	=(1+B24^2)/2/PI()*(SQRT(((B5+2*B22)*B23)^2+4*B22^3*(B5+B22))+B15*SQRT(((B6+B15*2*B22)*B23)^2+B15*4*B22^3*(B6+B15*B22))-(B5+B15*B6)*B23)		
η12	=B5^2*B22^2/(B5^2*(B23^2+B22^2)+2/3*PI()^2*B22^4*(B25-1)*(2*B25-1)+B13*2*PI()*B23*B5*B22^2*(B25-1))		
cos(beta23)	=COS(B21)		
tan(alfa023)	=TAN(B19)		
tan(beta23)	=TAN(B21)		
ε23	=(1+B29^2)/2/PI()*(SQRT(((B7+2*B27)*B28)^2+4*B27^3*(B7+B27))+B16*SQRT(((B8+B16*2*B27)*B28)^2+B16*4*B27^3*(B8+B16*B27))-(B7+B16*B8)*B28)		
η23	=B7^2*B27^2/(B7^2*(B28^2+B27^2)+2/3*PI()^2*B27^4*(B30-1)*(2*B30-1)+B14*2*PI()*B28*B7*B27^2*(B30-1))		

Fig. 9. Calculation program written in Excel

It was considered for all cases a tilt teeth angle of 15 degrees [deg].

Conclusion

The input shaft, while rotating, forces the satellite to roll inside the crown. In its movement, this one drives the satellite door as if it were a crank. The planet carrier constitutes the output shaft of the device. In this configuration the output rotates in the same direction and slower than the input.

Planetary gears have a number of advantages compared to transmission with fixed axis. Under similar operating conditions, the planetary transmissions serve longer and produce less noise as compared to a fixed shaft transmission (Cao *et al*., 2013; Lee, 2013; Garcia *et al*., 2007). Gearboxes are used to reduce the turbine rotational speed from generator's speed at performances of up to 110 MW or turbine rotational speeds of approx. 60,000 rpm. Parallel shaft gear units are usually used in these applications. However, some packagers and generator manufacturers use planetary gear units for gas and steam turbine systems. The majority of planetary gear unit manufacturers are located in Europe and United States.

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For the various cinematic planetary systems presented in Fig. 3, where entry is made by the planetary carrier (H) and output is achieved by the final element (f), the initial element being usually immobilized, will be used for the kinematic calculations the relationships generalized 1 and 2.

The planetary mechanisms are less synthesized based on their mechanical efficiency developed during operation, although this criterion is part of the real dynamics of mechanisms and also the most important criterion in terms of performance of a mechanism.

The planetary system efficiency given by the presented formula has the great advantage to be easily determined and can be used for any type of simple planetary gear.

The great advantage of this formula is its generality. A second great advantage of this formula is its precision. By this we have now high precision formula which can be used in any case.

Analyzing the previous calculation examples, it can be noticed that the yield (of a simple planetary gear) increases when the sun gear transmission ratio input-output decreases and when $alfa_0$ angle is decreased.

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Ethics

This article is original. Author declares that are not ethical issues that may arise after the publication of this manuscript.

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Appendix

Calculation program written in Excel (See Fig. 9).