Development of a Design Tool for a Two-Degree of Freedom Gear Train with Sun-Planet-Planet-Sun Configuration

Ralph S. Jose, Gerald Jo C. Denoga

Abstract— **A two-degree of freedom gear train is a very versatile mechanism that is utilized for many different applications. There are several configurations of this gear train system with varying number of gears used to achieve the desired output motion. This paper focuses on one particular gear configuration which is the sun-planet-planet-sun arrangement. This gear train requires a designer to determine four different gear sizes that will be able to satisfy the specific gear ratio and at the same time be able to withstand the loads as required by the application. A case study on using this gear train as transmission for a particular hybrid vehicle was used in order to show the processes and results of the design tool that was developed. The result is a spreadsheet with data base search function that outputs several possible gear combinations that satisfy the required ratio and capacity. This was achieved by creating a program inside the spreadsheet tool that searches all possible combinations for the number of teeth of each of the four gears that satisfies the conditions for a particular combination to work. The output of the program is a list of gear combination that can be compared to the required capacity of each gear determining the final size of each gear that will be optimized in terms of size and mass.**

Index Terms*— *Epicyclic gear train, Power split device, Power train, Series-parallel hybrid

I. Introduction

Epicyclic gear train (EGT) is defined as gear train with one or more gear shafts revolving around another gear shaft and at the same time rotating on their own axes [1]. Fig. 1 (a) shows a regular gear train with both axes of the gears fixed to the ground. Shafts of devices, such as engine shaft, motor shaft, or output shaft, are connected to each of the two gears with one serving as the input device and the other as the output device. An inversion of this mechanism is shown in Fig. 1 (b) where the first gear is fixed to the ground the other gear is allowed to revolve around the 1st gear. This is an example of an epicyclic gear. The link connecting the axes of the two gears is the carrier. This carrier has its axis of rotation coinciding with the axis of the first gear. Given an input motion to the carrier, another member is connected in order to transmit the motion into a device with a fixed axis of rotation.

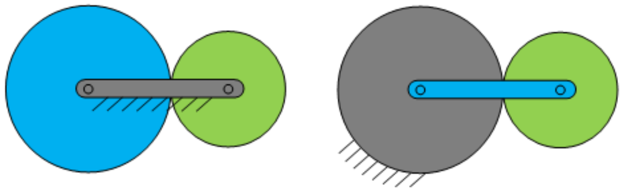
(a) (b)

Fig. 1. (a) Gear train with fixed axis of rotation (b) Gear train with one gear axis revolving around the other fixed gear axis

EGT therefore has three components that could be connected to another device or be fixed to the ground. These three members are composed of the planet carrier and two other components which could be composed of a sun gear and a ring gear, two sun gears, or two ring gears. A single EGT assembly could have zero, one, or two degrees of freedom.

Degree of freedom is the number of required independent input motion to define all the movements and positions of the rest of the mechanism [2]. Fixing two or three components to the ground makes the gear train immovable. This gives the mechanism a degree of freedom equal to zero. When one member is fixed, the system is a single degree of freedom mechanism. When all three members are allowed to rotate, the gear train is a two-degree of freedom mechanism.

A. *Types of epicyclic gear trains*

Fig. 2 shows the 12 types of epicyclic gear train. Types A, B, G, and H are classified as simple epicyclic gear trains (EGT) while types C, D, E, F, I, J, K, and L are considered complex EGTs [3]. A complex EGT has two or more pairs of planets in mesh with each other.

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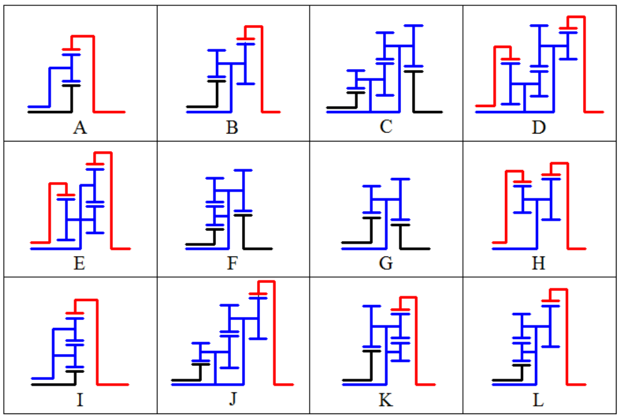

Fig. 2. 12 possible configurations of epicyclic gear trains by Levai [4]

Type-A has a sun-planet-ring configuration and is the most common type of EGT. It has the simplest structure having all gear components aligned or meshing in a single plane. From the four simple EGTs, type-G is the one that does not have a ring gear. This is the configuration that will be focused in this study. It has a sun-planet-planet-sun configuration.

B. Hybrid Vehicle Application

One application that has fully utilized advantages of the epicyclic gear train is the hybrid vehicle technology. A hybrid vehicle is defined as vehicle having a combination of two or more power sources [5]. Car manufacturers have already produced different kinds and types of configurations of hybrid cars. One of the most common hybrid vehicles in the market is the Toyota Prius. It uses the sun-planet-ring configuration. The configuration of the Toyota Prius is shown in Fig. 3. The sun gear is driven by the MG1 while the petrol engine is connected to the planet carrier. The output of the PSD is directly coupled to the MG2 which drives the wheels.

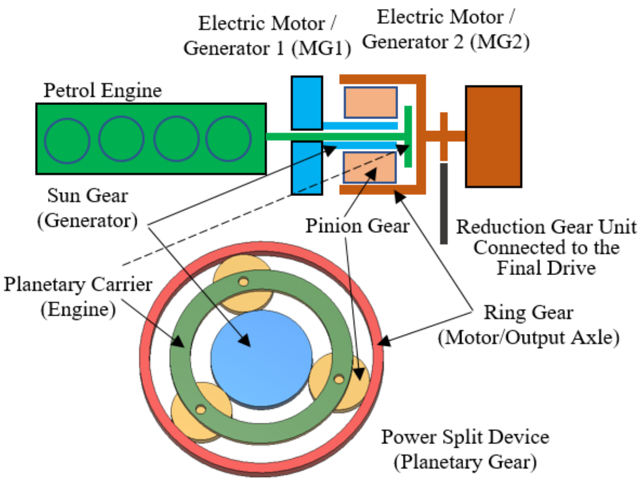

Fig. 3. Toyota Prius planetary gear unit [6]

C. Case study

This study however will use a hybrid configuration for a bus as a case study for development of the gear selection methodology. The EGT configuration that was used in the case study is the sun-planet-planet-sun configuration. Fig. 4 shows the schematic diagram of a series-parallel hybrid configuration.

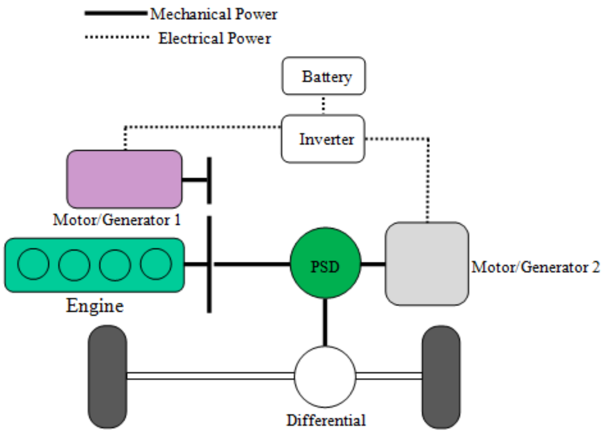

Fig. 4. Series-parallel hybrid configuration

The hybrid set-up is composed of an internal combustion engine that is coupled to a motor/generator 1, motor/generator 2, and an output shaft going to the vehicle's differential that transfers the power to the wheels. The coupling of the Engine and the motor-generator 1 is combined with the motor-generator 2 (MG2) using the power split device (PSD). [7]

D. The Power Split Device

The power power-split device (PSD) is considered as the heart of the series-parallel hybrid vehicle. The PSD is essentially an epicyclic gear train that physically connects the engine, the electric motors, and the output shaft of the hybrid vehicle.

E. Sun-Planet-Planet-Sun Configuration

The configuration of the epicyclic gear train as PSD for the case study is illustrated in Fig. 5. It is composed of two sun gears (s1 and s2) and two planet gears (p1 and p2). The arrangement is described as sun-planet-planet-sun.

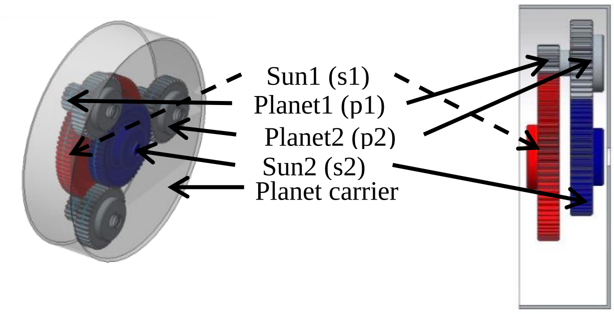

Fig. 5. PSD-Phantom gear configuration (sun-planet-planet-sun)

Two quantities are needed to be solved to define the dynamics of the PSD. These are rotational speeds and torques of the components.

The sun-planet-planet-sun configuration presents some advantages and disadvantages in the design and manufacture process. The sun-planet-ring configuration is one of the easiest to design as it only requires a simple gear train arrangement. The challenge however occurs in the manufacturing process. The gear train requires a ring gear of which ordinary fly cutter or gear hub cannot produce. It requires special type of machine and cutting tool. This implies an increase in the manufacturing cost. Such cost could easily be compensated by the increase of volume of

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production. This on the other hand requires that the design process will decrease the customization of the vehicle.

The sun-planet-planet-sun configuration requires no ring gear. This means that all gears to be used are external gears which are commonly manufactured and is easy to be customized. However, the downside of this configuration is that it is more difficult to design as there will be more gear configurations that are possible for a given size range as compared to that of the sun-planet-ring configuration. This design challenge is answered in this study. A spreadsheet tool is developed such that it will be able to give a list of possible configurations from a database of standard gear sizes given the design specification for each gear.

The case study presented in this paper maximizes the power that the electric motors while sustaining the operating speed of the engine at its optimum condition. This will enable the vehicle to run at its maximum power while maintaining an efficient operation of the engine. The resulting methodology requires a properly designed gear configuration with a gear ratio that satisfies the required operations parameters.

II. Statement of the Problem

Designing the epicyclic gear train is a challenging task when given the limitations in the size for a specific gear ratio. Given a particular setup ensuring maximum efficiency of the engine and that the electric motor will be able to boost the needed power of the vehicle while maintaining the most efficient operating condition of the engine makes the hybrid bus an interest for study. Applying the EGT mechanism as a power split device for a hybrid vehicle requires several design steps.

Given that it is a two-degree of freedom gear train, the mechanism requires two independent inputs to have a combined dependent output. This setup is desired for a series-parallel hybrid system. The problem now with using an epicyclic gear train is that the gear ratios of the EGT are fixed. This gear ratio determines the distribution of power contribution of each of the components. This means that for a power contribution from the ICE-MG1 and the power contribution from the MG2, it will add up as the output power to the wheels. Selecting any gear ratio will not ensure that the desired output will be achieved. A particular gear ratio may result to maximized component contributions while not achieving the maximum vehicle output.

The challenge therefore is to determine the gear ratio of the EGT that will combine the powers of the ICE, MG1, and MG2 that will match the desired output of the vehicle.

Given that this gear ratio is already determined, a new challenge is presented in selecting the gears to use in order of get the required gear ratio. In the usual sun-planet-ring setup, the designer only needs to select 3 gear sizes. One of the gears is dependent on the size of the other two gears.

A. *Face width*

For a sun-planet-ring configuration, once the face width of a gear is selected, all the other gears will have a same face width. Given an effective gear ratio, the three gears could easily be tabulated and compared to the required strength of each gear. The face width could then be adjusted in order to make sure that all gear will be able to withstand the given loads.

The case is different for a sun-planet-planet-sun configuration. Since the configuration is a compound gear set-up, the 1st sun-planet interface could be different to the other sun-planet interface.

B. *Gear combinations*

Aside from having 2 different face widths, there could be more than one gear set that will satisfy the required gear ratio. Tabulation of all the gear combinations possible makes the process longer. The challenge now is to device a way to make the process faster.

III. Methodology

A. *The Gear Ratio k*

Changing the gear ratio of the PSD affects the distribution of power from the E+G and the MG2. Different values for r_{p2} and r_{s2} can result in the same output of the torque distribution on the components of the PSD. The question now is if the different gear ratios of the component result to the same output, what then is constant in the relationships of the gear sizes? In order to find this out, equations for the speed and torque relationships shall be explored. The following are the definition of variables used:

T = Torque
r = radius
ω = angular velocity
pc = planet carrier

Equation (1) and (2) show the relationships of torques of Phantom components.

$$T_{s1} = -T_{s2}\frac{r_{p2}r_{s1}}{r_{s2}r_{p1}} \tag{1}$$

$$T_{pc} = -T_{s1}\frac{r_{pc}(r_{p1}-r_{p2})}{r_{p2}r_{s1}} \tag{2}$$

For (2),

$$r_{pc} = r_{s1} + r_{p1} = r_{p2} + r_{s2} \tag{3}$$

therefore,

$$T_{pc} = -T_{s2}\frac{(r_{s1}+r_{p1})(r_{p1}-r_{p2})}{r_{s2}r_{p1}} \tag{4}$$

$$T_{pc} = -T_{s2}\frac{r_{s1}r_{p1}-r_{s1}r_{p2}+r_{p1}{}^2-r_{p1}r_{p2}}{r_{s2}r_{p1}} \tag{5}$$

$$T_{pc} = -T_{s2}\left(\frac{(r_{s1}+r_{p1}-r_{p2})r_{p1}}{r_{s2}r_{p1}} - \frac{r_{s1}r_{p2}}{r_{s2}r_{p1}}\right) \tag{6}$$

$$r_{s1} + r_{p1} - r_{p2} = r_{s2} \tag{7}$$

$$T_{pc} = -T_{s2}\left(1 - \frac{r_{s1}r_{p2}}{r_{s2}r_{p1}}\right) \tag{8}$$

Equation (3) and (4) both have a common factor of $\frac{r_{s1}r_{p2}}{r_{s2}r_{p1}}$.

A variable now is defined as

$$k = \frac{r_{s1}r_{p2}}{r_{s2}r_{p1}} \tag{9}$$

The relationship for the torques can now be summarized as

$$T_{pc} = -T_{s2}(1-k) \tag{10}$$

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$$T_{s1} = -T_{s2} k \tag{11}$$

$$T_{PC} = -T_{s1} \frac{(1-k)}{k} \tag{12}$$

The velocity relationship from (1) can be written as

$$\omega_{s2} - \frac{r_{s1} r_{p2}}{r_{s2} r_{p1}} \omega_{s1} + \frac{-(r_{p1} - r_{p2}) r_{pc}}{r_{s2} r_{p1}} \omega_c = 0 \tag{13}$$

Substituting k will give

$$\omega_{s2} - k\omega_{s1} - (1-k)\omega_c = 0 \tag{14}$$

Table I shows that different gear combination can give the same ratio k. Noting this, there will be infinitely many possible gear combinations that can satisfy the required drive specifications. These different combinations however have certain physical implication on the implementation or production of the PSD. Each combination has its own overall mass, size, inertia, and load capacity.

TABLE I
EXAMPLE OF DIFFERENT GEAR COMBINATIONS HAVING THE SAME GEAR RATIO K

	s1	p1	p2	s2	k
N	36	66	34	68	0.2727
	45	55	25	75	0.2727
	48	80	40	88	0.2727

B. Case Study

The case study for this paper uses the sun-planet-planet-sun configuration. The transmission mechanism is labelled the Phantom dive. The epicyclic gear train of Phantom configuration is a type of gear train called reverted compound train [8]. Reverted compound train is a gear train with concentric input and output shaft. In order to transmit the motion back to the same axis, an intermediate compound gear is to be used. Naming the gears as s1, p1, p2, and s2 as shown in Fig. 6 the following relationship shall be followed.

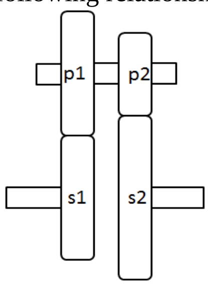

Fig. 6. Reverted compound train

Converting Eq. 3 in terms of the diameter

$$d_{s1} + d_{p1} = d_{p2} + d_{s2} \tag{15}$$

given that all gears will have the same module,

$$m{*}N_{s1} + m{*}N_{p1} = m{*}N_{p2} + m{*}N_{s2} \tag{16}$$

therefore,

$$N_{s1} + N_{p1} = N_{p2} + N_{s2} \tag{17}$$

This condition is true for the equations derived for the computation of the radii of the gears in the PSD. Hence, it shall now be emphasized that the gear components for the PSD shall have the same module or diametral pitch.

The gear sizes are determined by module and number of teeth. The value of the number of teeth is in whole number format. The radii solved in the optimization step are not in whole numbers with respect to the size of the base gear, sun1. Hence, the number of teeth for sun1 can be set as base for the gear ratios.

Table II shows the number of teeth of the other gears solved. The values obtained in the power optimization gives fractions for the number of teeth, thus, it shall be rounded off to the nearest integer. This rounding off will affect the optimized value of the gear ratio of k. Hence, gear ratios may not give the exact value of k. Acceptable deviation in percent of k shall be defined.

TABLE II
SAMPLE GEAR RATIO SOLVED FOR THE PSD COMPONENTS WITH SUN1 AS BASE VALUE 1.

PSD - C1	R
Sun1	1.000
Planet1	1.857
Planet2	1.000
Sun2	1.857
Carrier	2.857
k	0.290

Possible gear combinations are tabulated and will be tested for the dynamic properties to be used for the PSD. Still, the goal is to find the smallest gears possible to achieve the desired dynamic behavior and at the same time within the capacity that can handle to loads transmitted through the components.

To select the appropriate gear sizes, a catalogue for gears will be used as reference for gear selection. Gears in the catalogue are tabulated by module. Standard modules are 1, 1.5, 2, 2.5, 3, 4, 6, 8, and 10. Gear catalogue includes pitch diameter, outside diameter, face width, the gear torque capacity, and other essential information. As an example, module-3 has 63 standard gear sizes available. Tabulation is shown in

Table III.

TABLE III
SAMPLE GEAR RATIO SOLVED FOR THE PSD COMPONENTS WITH SUN1 AS BASE VALUE 1.

Catalog No.	No. of teeth	Bore (mm)	Pitch dia. (mm)	Outside dia. (mm)	Face width (mm)	Allowable torque (N-m)	Weight (kg-f)
	z	A_{H7}	C	D	E	Bending strength	
SS3-17	17	15	51	57	30	121.5	0.65
SS3-18	18	15	54	60	30	132.6	0.67
SS3-19	19	15	57	63	30	143.8	0.73
SS3-20	20	15	60	66	30	155.1	0.8
SS3-21	21	15	63	69	30	166.6	1
SS3-22	22	15	66	72	30	178.2	1.1
SS3-23	23	15	69	75	30	189.9	1.1
SS3-24	24	15	72	78	30	201.6	1.2
SS3-25	25	20	75	81	30	213.5	1.3
SS3-26	26	20	78	84	30	225.5	1.5
SS3-27	27	20	81	87	30	237.4	1.6
SS3-28	28	20	84	90	30	249.5	1.7
SS3-29	29	20	87	93	30	261.6	1.7

The gear size to be selected will be based on the number of teeth. Three gear sizes will be selected. The first gear is assigned to the sun1. A table can now be generated for possible combinations for sun2 and planet2. Given these

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three gears, the corresponding size for planet1 can be calculated using

$$p1 = s2 + p2 - s1 \tag{18}$$

The ratio k can then be calculated using

$$k = \frac{s1\ p2}{s2\ p1} \tag{19}$$

The ratio k will be the basis for gear selection. Acceptable values for k will be a designer's prerogative on which the deviation from the desired dynamic characteristics of the train is proportional to the deviation of the selected k from the optimum k. Hence, all gear combinations that will fall in the range of k plus or minus deviation (k ± k*dev) will be tabulated in the list of candidates of possible combinations.

It can also be noticed here that there will be values of p1 on which there is no corresponding gear available on the list. If the computed value of p1 is not on the list, then the combination for it will be discarded.

Table IV shows possible gear combinations of s2, p2, and p1 for a chosen s1. Changing the number of teeth of sun1 will result to a new set of combinations.

The process will be done for all sizes for sun1 with a total of N number of iterations. This will give N^3 combinations that would mean a total of 63^3 = 250047 combinations. Tabulating all of this will be a very demanding task. Hence, a program using Visual Basic function in MS Excel will be created to do the task.

The program will output a list of all the possible gear combinations satisfying the criteria that the gear train will have a k close if not equal to the optimum k and that all gears to be used are available in the catalogue list.

TABLE IV
TABULATION OF AVAILABLE GEAR SIZES FOR A MODULE AND THE RESULTING RATIO K.

	15		16		17		18		19		20	
	k	p1	k	p1	K	p1	K	p1	k	p1	K	p1
15	1.000	15	1.000	16	1.000	17	1.000	18	1.000	19	1.000	20
16	0.879	16	0.882	17	0.885	18	0.888	19	0.891	20	0.893	21
17	0.779	17	0.784	18	0.789	19	0.794	20	0.798	21	0.802	22
18	0.694	18	0.702	19	0.708	20	0.714	21	0.720	22	0.725	23
19	0.623	19	0.632	20	0.639	21	0.646	22	0.652	23	0.658	24
20	0.563	20	0.571	21	0.580	22	0.587	23	0.594	24	0.600	25
21	0.510	21	0.519	22	0.528	23	0.536	24	0.543	25	0.549	26
22	0.465	22	0.474	23	0.483	24	0.491	25	0.498	26	0.505	27
23	0.425	23	0.435	24	0.443	25	0.452	26	0.459	27	0.466	28
24	0.391	24	0.400	25	0.409	26	0.417	27	0.424	28	0.431	29
25	0.360	25	0.369	26	0.378	27	0.386	28	0.393	29	0.400	30
26	0.333	26	0.342	27	0.350	28	0.358	29	0.365	30	0.372	0
27	0.309	27	0.317	28	0.326	29	0.333	30	0.341	0	0.347	32
28	0.287	28	0.296	29	0.304	30	0.311	0	0.318	32	0.325	0
29	0.268	29	0.276	30	0.284	0	0.291	32	0.298	0	0.304	34
30	0.250	30	0.258	0	0.266	32	0.273	0	0.279	34	0.286	35

C. *Load Sharing on planets*

The load on an EGT is distributed among the planets. In effect, it increases the capacity of the sun gear. Ideally, increasing the number of planets will increase the division of load and will reduce the load on the planet gears. This will only be true if manufacturing tolerances and errors are not accounted for.

In a study by Bodas and Kahraman, the difference on the load sharing characteristics of three, four, five, and six planet systems are explored [9]. Their experimental setup was done such that the sizes of the gear components are the same for all setups. Their findings show that a three-planet system has perfect load sharing regardless of manufacturing errors. For systems with more than three planets, there is a poor sharing of loads. This poor distribution increases with the degree of manufacturing errors such as carrier pinhole position error, planet tooth thickness error, and run-out errors. The effect is that the expected reduction on planet loads is disregarded and the system will have the loads equivalent to three-planet system.

In the dissertation of Ligata [10], he investigated the effects of different factors that can affect the load sharing on the planets of a PGT. He was able to establish a database relating manufacturing errors and planet loads. His finding agrees with that of Bodas and Karahman [9] that a three-planet PGT has perfect load sharing. Presence of manufacturing errors fails to reduce the planet load.

IV. RESULTS AND DISCUSSION

A. *Visual Basic Program in MS Excel*

A self-developed search program using Visual Basic is created to generate a complete list of combinations gears such that it will fall under the range of acceptable deviation from the solved k. It also considers that the resulting gear sizes are all available in the gear catalogue. Fig. 7 illustrates the interface of the developed search tool. The clear button clears the list of all gear sizes. This operation is done before generating new set of gear sizes when value of k or the percent allowable deviation is changed.

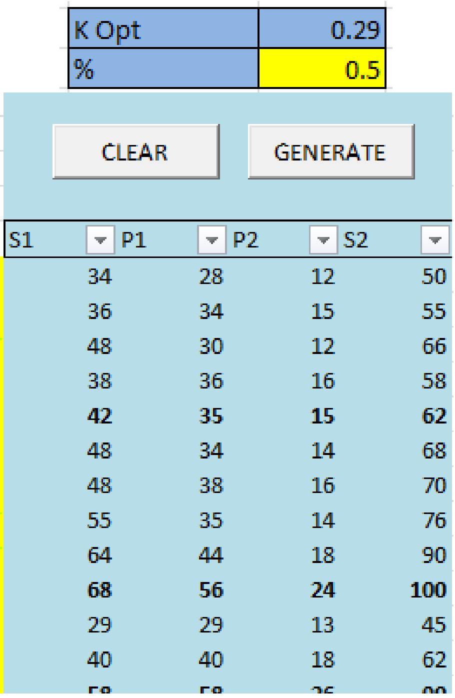

Fig. 7. MS Excel Visual Basic program output. list of combination of gear sizes having a gear ratio k.

After having a list of all possible gear combinations, the *vlookup* function will be utilized to find pertinent values needed for deciding which gear combination will be used. The first criteria to be satisfied is that the gear components of the device should be able to handle the torques as computed from the previous process. There are three gear

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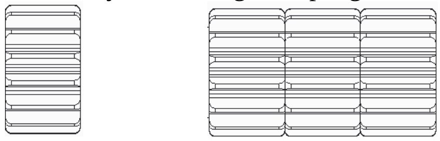
torque values to be tested. Since planets 1 and 2 are connected in a single shaft, both will have the same torque. In order to increase the capacity of the gear components, say for sun2 which will experience the largest torque, it will be assembled with two gears layered with each other. This in effect will increase the face width. This increase in the face width, for example having two layers and in effect doubling the area of the gear teeth, decreases the stress on the teeth, thus doubles the capacity. Fig. 8 shows the increase in the face width of by assembling multiple gears in layers.



(a) (b)

Fig. 8. Increasing face width through multiple layer arrangement. (a) Single-layer gear. (b) Triple-layer gear

This case allows smaller gear diameter to handle higher loads. Making two gears in tandem requires that its mating gears should also be doubled. This will result into an increase in the overall mass of the PSD. Having this noted, the mass will be tabulated together with the gear capacity. The overall diameter will also be listed as reference on how large the construction of the PSD will be.

B. *Number of planets*

Designing an epicyclic gear train is not a straightforward approach. Arriving at a desired gear ratio will not guarantee that the train will be physically feasible to construct. For the PGT, the number of teeth of the planet is dependent on the number of teeth of the sun gear and the ring gear. The number of teeth of the planet is equal to half of the number of teeth of the ring gear less the number of teeth of the sun gear.

After selecting the number of teeth of the gears, the gear train can now be physically assembled for a single planet. That means the position of the sun and the ring gear is now fixed for the given position of the planet. This will leave the positioning of additional planet gear to be possible only on certain locations. If the planet gears are to be equally spaced, the sum of the number of teeth of the sun gear and the ring gear should be divisible to the number of teeth desired [11].

The same rule is true with the sun-planet-planet-sun epicyclic gear train but requires match markings. These match markings are necessary before the assembly of the PSD. Assembling the gear train without these marks may not insure proper mesh.

Fig. 9 shows a sample result of the search program combined with the *vlookup* function to show all the possible combinations that will satisfy the gear ratio and the availability of the gear size. The first four columns give the number of teeth of the components. The next four columns check if the combination can be assembled with 3, 4, 5, or 6 planets respectively. It is followed by the required size of the housing for the gear train. The next sets of columns are the load capacity of the gears plus their corresponding weight. The succeeding columns are for the increase in the number of layers to increase the capacity.

S1	P1	P2	S2					R_h	MT_s	MxT_p1	MxT_p2	Mxt_s2
34	28	12	50	ok	ok		ok	270	323	249.5	68.71	525.1
36	34	15	55					312	347.8	323	99.73	589.5
48	30	12	66	ok			ok	324	499.5	273.8	68.71	610.1
38	36	16	58	ok	ok		ok	330	372.8	347.8	110.5	628.2
42	**35**	**15**	**62**		ok			**336**	423.3	335.4	99.73	680.1
48	34	14	68		ok			348	499.5	323	89.18	631.8
48	38	16	70					372	499.5	372.8	110.5	653.5
55	35	14	76					375	589.5	335.4	89.18	718.9
64	44	18	90					456	588.4	448.6	132.6	872.3
68	**56**	**24**	**100**	ok	ok		ok	540	631.8	602.3	201.6	982.6
29	29	13	45					261	261.6	261.6	78.83	461.3
40	40	18	62	ok			ok	360	397.9	397.9	132.6	680.1
58	**58**	**26**	**90**		ok			522	628.2	628.2	225.5	872.3
15	25	13	27	ok			ok	195	99.73	213.5	78.83	237.4
14	26	14	26		ok	ok		198	89.18	225.5	89.18	225.5
13	27	15	25					201	78.83	237.4	99.73	213.5
21	27	13	35		ok			225	166.6	237.4	78.83	335.4
18	29	15	32			ok		228	132.6	261.6	99.73	298.3

Fig. 9. Spreadsheet output of self-developed search program

V. CONCLUSION

This study shows that for any selected effective gear ratio for an epicyclic gear train with a sun-planet-planet-sun configuration, there will be several possible gear combinations what will give the desired ratio. However, the process of obtaining a list of all the gear combinations goes beyond a two-dimensional row vs column listing. Also filtering those gear combinations that satisfies the required strength for all the gears selected requires more than just a look-up function. Hence, a compact tool was developed where a user inputs a database from standard gear sizes and runs a program resulting to a list of gear combinations that is physically possible to assemble and at the same time capable of withstanding the designed load and kinematic behavior of the mechanism.

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