Journal of Engineering and Development, Vol. 12, No. 2, June (2008) **ISSN 1813-7822**

An Idea for Coupling Two Engines to Drive a Generating Head Using Epicyclic Gear Train

Lect. Ehsan S. Al-Ameen
Mechanical Engineering Department, College of Engineering
Al-Mustansiriya University, Baghdad, Iraq

Abstract

This work presents a method for designing and analyzing planetary gear trains with a discussion of potential applications in engine couplings. Furthermore, it will show to use Epicyclic gear trains as a power coupling for two-engine generators to discus the fuel consumption for (Amp/hr) 1n 2 & 4 KVA generators comparing between the use of 337 cc engine and two 163 cc engines.

Results show that the use of two 163 cc engines lunched to the generator head in the range of (1 to 12) Amp/hr may give up to 50% reduction in fuel consumption to that in single 337cc engine (at 6 Amp/hr).

الخلاصـــــــة

يبحث هذا العمل طريقة تصميم وتحليل حركة مجموعة التروس الكوكبية ومناقشة تطبيقات ربط المحركات. ويوضح العمل فكرة استخدام التروس الكوكبية كآلية ربط في مولد كهرباء ذو محركين لغرض دراسة استهلاك الوقود لكل (أمبير/ ساعة) في مولدات ذات قدرة توليد ٢ و ٤ KVA والمقارنة بين استخدام محرك حجم ٣٣٧ سم^٣ أو استخدام محركين حجم ١٦٣ سم^٣ بدلا عنه.

أظهرت النتائج بان استخدام محركين اثنين حجم ١٦٣ سم^٣ مربوطين على راس التوليد للحصول على تيار كهربائي بين ١ الى ١٢ أمبير/ ساعة يعطي تخفيضا في استهلاك الوقود وبنسبة تصل الى ٥٠% مما هو في محرك بحجم ٣٣٧ سم^٣ (عند سحب تيار ٦ امبير/ساعة).

1. Introduction

Nowadays, in Iraq, gasoline generators are taken as essential needs in every house, store, workshop, and all other places. Since, electricity is the camelish crisis for the last 10 years and will continue for the coming 10 years, so, the existence of gasoline generators every where is essential and important. Although, it is considered hard encumbrance on the employers withers due to the fuel expenses and rarity. So, we find the importance of any study to reduce these expenses.

When you need electrics about (1-6) Amp/hr you will look for a 2 KVA generator. While your needs are more than 6 Amp/hr up to 17, 18 Amp/hr or more you will go to the 5 KVA generator. Of course, every generator has its specifications and fuel consumption but, what if your needs are 6-10 Amp/hr most of the time and more than 12 Amp/hr rarely then, the 2 KVA will not be the right choice while, the 5 KVA generator will be much expensive most of the time. From here we have got the idea to use two engines from those used in 2KVA generators to operate a 5 KVA generator head. With this idea we will save the fuel comparing with the use of a 5 KVA generator for 1-12 Amp/hr but when the need is more than 13 Amp/hr the two engines will cost more. Studying the system cost, with comparison to the economic recurring is impetus.

The problem is: how to engage the two synchronously operated engines with the generator? And make them in tune and efficient. The idea is to use a planetary gear train.

2. Backgrounds

Planetary gear trains are one of the main subdivisions of the simple Epicyclic gear train family (see **Fig.(1)**). The Epicyclic gear train family has a central “sun” gear which meshes with and is surrounded by planet gears. The outer more gear, the ring gear, meshes with each of the planet gears. The planet gears are held to a cage or carrier that fixes the planets in there orbit relative to each other [1].

The three main types of simple Epicyclic gears are the planetary, the star, and the solar. These three types differ only in arrangement. The arrangement is changed by the gear that is fixed as it is defined in **Table (1)**. A planetary train has a fixed ring which allows the planets to output an input from the sun. A star train has a fixed cage; inputs in the sun gear rotate the individual planets, which allow the ring to output the torque. The solar train holds the sun gear fixed; input from the ring causes the planets and cage to rotate about the sun to produce the output. Modifications to the fixed arrangement will have large effects on gear ratios, speeds, and torque.

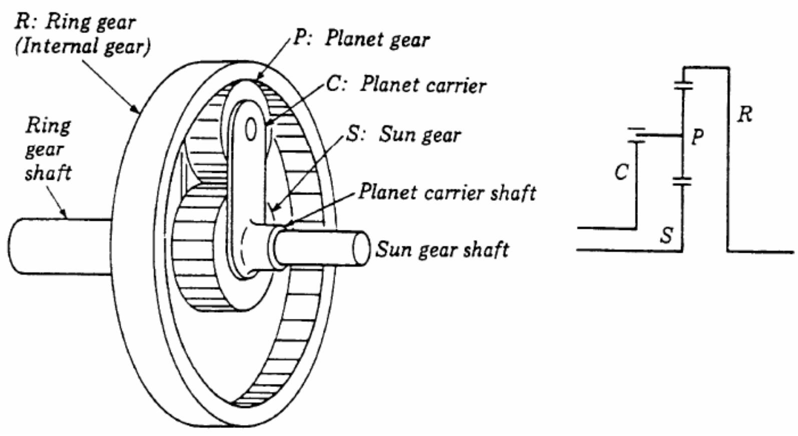

Figure (1) An Epicyclic gear train and its associated terminology

Table (1) Types of Epicyclic gear trains arrangement

Arrangement of Epicyclic train	Fixed Member	Input Member	Output Member	Over-all Gear Ratio	Range of ratios normally used
Planetary	Ring	Sun	Cage	$\left(N_r/N_s\right)+1$	3:1 – 12:1
Star	Cage	Sun	Ring	$\left(N_r/N_s\right)$	2:1 – 11:1
Solar	Sun	Ring	Cage	$\left(N_s/N_r\right)+1$	1.2:1 – 1.7:1

Epicyclic gear trains are gaining in popularity and have found use in a variety of applications over the past 300 years. James Watt patented a sun and planet gear to be used with one of his early engines in 1781. More recently, the aircraft industry has made extensive use of Epicyclic arrangements as virtually all propellers and turbine driven planes can makes use of them. Automatic transmissions use a series of complex Epicyclic gears to provide gear ratio changes. Epicyclic trains can even be combined to produce power feedback systems that allow for free rotation if an extreme feedback load is applied; such combinations have been implemented on lawn mower self propulsion units [2].

In 1857, Willis [3] was the first who published a dedication to the field now called kinematics. He discussed for the first time the analytical modeling of an Epicyclic gear train, and presented only a solution for the rotational speeds in the gear train. In 1966, Lévia [4] attempts to unify all the previously written literature on Epicyclic trains and what he calls ((Epicyclic change speed gears)), which appear to simply be multiple speed transmissions. He also identified, for the first time, the twelve possible variations on the Epicyclic train. He also stated that those twelve variations can be divided into those with and without auxiliary planets or planet pairs. After defining the Epicyclic train, Lévia turned his attention to the solution method laid out by Willis, and graphical method of Kutzbach, as it applies to trains without auxiliary planets. He discussed at length two different modifications that can be performed to apply the Kutzbach method to a train with auxiliary planets. However, he offers no treatment of the torques present in the system.

Alhimidani [5] and Dean Lent [6] presented in detail the methodology of Willis for finding the rotational speeds of each branch of the Epicyclic gear train, along with specific methods for the design of three and four gear trains. But, again there is little discussion of torques present in the system.

Joseph Shigley and John Uicer [7] give a more complete definition of the Epicyclic gear train. Most importantly, however, they present a solution technique for the torques present in the gear train. Unfortunately they do not approach the static force analysis for the general case; rather they present the solution in terms of free body diagrams for a specific arrangement of the planetary.

Hamilton Mabi & Charles Rienholtz [8], present largely the same information as Shigley and Uicer. While the treatment of the kinematics and static forces of the mechanism are nearly identical, Mabi & Reinholtz also present a brief section considering circulating power flow in controlled planetary gear systems.

In 1981, John Molnar [9] presented an excellent introduction to nomographs, as well as discussing at length their use and construction.

3. Motion Analysis

The elementary planetary, or Epicyclic, gear train is shown in **Fig.(1)**. The elementary train consists of two gears, the sun (1) and planet (2) gears, and a third member, hereafter referred to as the planet carrier or arm (3). Since it is difficult to directly transmit motion to or from the planet gear, the elementary Epicyclic gear train is somewhat limited in practical application. More useful, however, is the Epicyclic train referred to as the simple and complex planetary gear trains, where a second sun gear is used. These gear trains can be realized in any of the twelve arrangements set forth in **Fig.(2)**, as originally presented by Lévia [4]. The trains in section a and c are classified as simple trains, since the planet gears mesh with both sun gears. Those in section b and d represent the complex trains, where the planet gears are partially in mesh with each other and partially in mesh with the two sun gears. Therefore, only one of the planet carriers may be used.

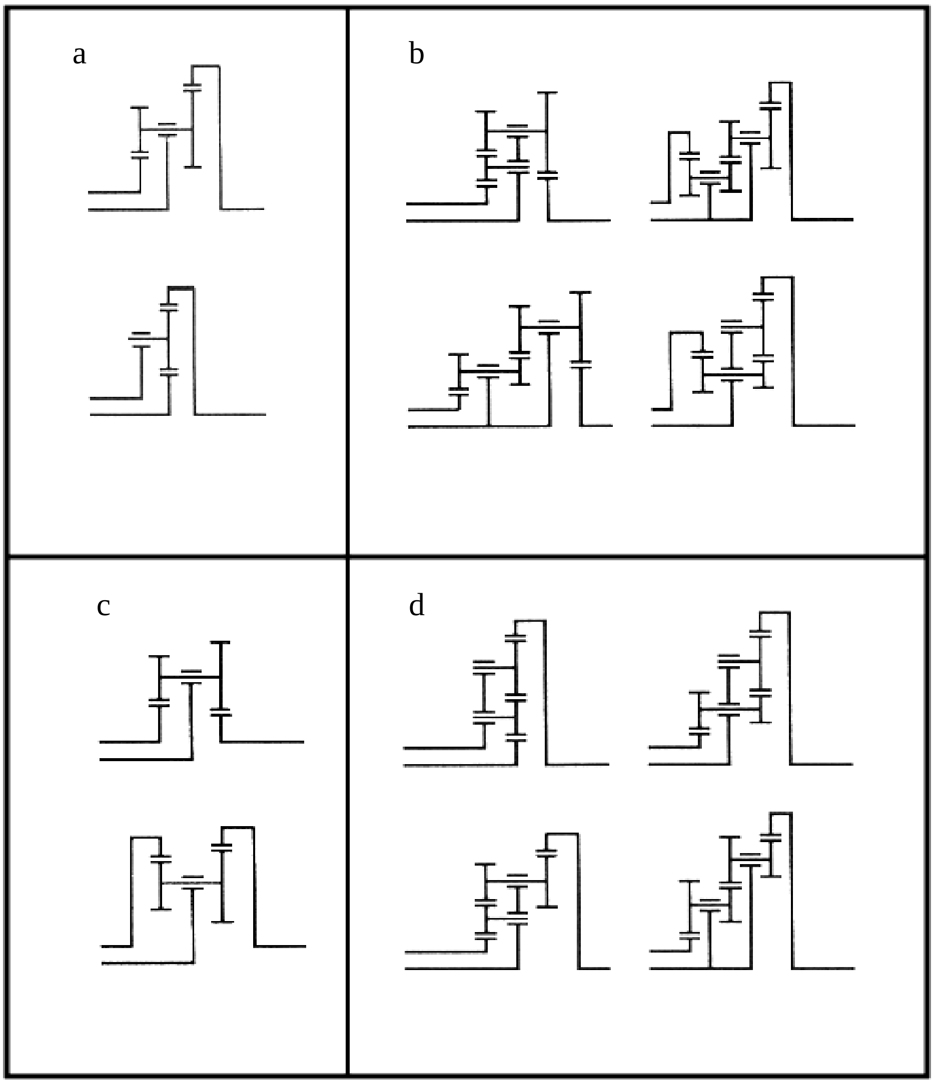

Figure (2) Epicyclic gear train arrangements

Willis [3] suggests the use of generic transmission ratio in defining the kinematics motion of a planetary gear train. This transmission ratio which is called R is defined as the speed ratio between the first and last gear in the train when the arm held stationary. For the purposes of analysis, this transmission ratio can be found using knowledge of the arrangement of the particular gear train. In design, R can be selected arbitrarily by the designer as any value other than 1. Once R is known, one can write the following

$$\frac{\omega_{FA}}{\omega_{LA}} = R \quad \text{.......} \quad (1)$$

where:

ω_{FA} *: is, the rotational speed of the first sun gear in the train and*

ω_{LA} *: is, the rotational speed of the last sun gear, both relative to the arm.*

Mathematically, this can be written:

$$\frac{\omega_F - \omega_A}{\omega_L - \omega_A} = R \quad \text{.......} \quad (2)$$

It is convenient to represent this in terms of the rotational speed of the arm, ω_A as:

$$\omega_A = \frac{R\omega_L - \omega_F}{R - 1} \quad \text{.......} \quad (3)$$

Equation (3) makes it mathematically apparent why 1 is an invalid selection for the transmission ratio, R. physically; a transmission ratio of 1 represents the indeterminate case where the arm can rotate at any speed regardless of the speeds of the first and last gears of the train.

4. Torque Analysis

Planetary gear trains are given a clear treatment with regard to the pure kinematics of the system, but little literature exists that includes the torques present in the system. Unfortunately, very few publications present a simple, concise design and analysis technique that considers both the motion and forces present in a gear train in the general case.

With the motion of the planetary gear train fully defined, it becomes important to understand the torque requirements of the system. This has traditionally been achieved by a static force analysis of a specific gear train. While this is a valid technique, it requires the selection of a specific planetary arrangement, and it is a long computational process, introducing many opportunities for error on the part of the designer. It is simpler to use the principal of energy conservation. The energy balance equation for the general planetary gear train can be written as:

$$\tau_A\omega_A + \tau_F\omega_F + \tau_L\omega_L = 0 \quad \text{.......} \quad (4)$$

where: *τ 's: represent the torques applied to each branch of the gear train. While this at first does not appear to yield a great amount of information, the careful selection of two specific cases for ω_F and ω_L will quickly yield equations completely defining the torques on the three brunches of the gear train.*

The first case to examine is the instance when the gear train is moving as a solid axle, that is $\omega_F = \omega_L = \omega_A = \omega$. In this case, equation 4 can be rewritten as:

$$\tau_A\omega + \tau_F\omega + \tau_L\omega = 0 \quad \text{.......} \quad (5)$$

Collecting terms and assuming ω is non-zero, it can be quickly deduced that:

$$\boldsymbol{\tau_A + \tau_F + \tau_L = 0} \quad \textbf{(6)}$$

This is the first of two equations that will define the torque requirements of the planetary gear train.

The second case of interest is that of zero speed at the arm. Using equation 3 and substituting zero for ω_A, it becomes readily apparent that the numerator on the right side must be zero. Hence,

$$\boldsymbol{\omega_F = R\omega_L} \quad \textbf{(7)}$$

Making this substitution and substituting zero for ω_A in equation 4.

$$\boldsymbol{\tau_F(R\omega_L) + \tau_L\omega_L = 0} \quad \textbf{(8)}$$

Again, collecting terms and rearranging into a more convenient form, the second torque governing equation is found to be:

$$\boldsymbol{\tau_L = -R\tau_F} \quad \textbf{(9)}$$

Using this equation along with equation 6, one can fully characterize the torque requirements at any two branches of the gear train, given the torque at the remaining branch. With this analysis complete, the system has been reduced to the solution of equation 3, 6, and 9, involving a total of seven variables. While this seems to imply that the designer has free choice of any four variables, the torque equations are independent of rotational speed. This means that the designer must select one torque, along with either three speeds or two speeds and a gear ratio. With so many variables being selected by the engineer, it becomes important to be able to visualize the response of the design to changes in any of these variables. A convenient graphical aid to the design is the nomograph proposed by Corey [10].

5. Application

The idea of this study is to use the Epicyclic gear train to engage a couple of engines to drive one generator. The specifications of the engines and the generator are derived from ASTRA gasoline generators manual for engine models 2500 and 5000 as shown in **Table (2)** below.

Table (2) Technical Specifications for ASTRA generators (models 2500 and 5000)

Generator model			2500	5000
Engine model			160	340
Displacement in cm^3			163	337
Rated Horsepower HP	3000 rpm		3.5	7.2
	3600 rpm		4	8
AC Output KVA	Rated	50 Hz	2	4
		60 Hz	2.2	4.5
	Maximum	50 Hz	2.2	4.5
		60 Hz	2.5	5

The goal is to save the gasoline required to operate the 5000 model by the use of two (160) model engines instead of the (340) model. Since it is not assumed to get the maximum power most of the time, and only one engine is enough to operate the generator for half power and the second engine may be lunched if more power is demanded. In his case study on tandem bicycling, Corey [10], found that the gear train arrangement in section c of **Fig.(2)** is the better to be considered for simplicity. In this work the same system is considered with some modifications as shown in **Fig.(3)**.

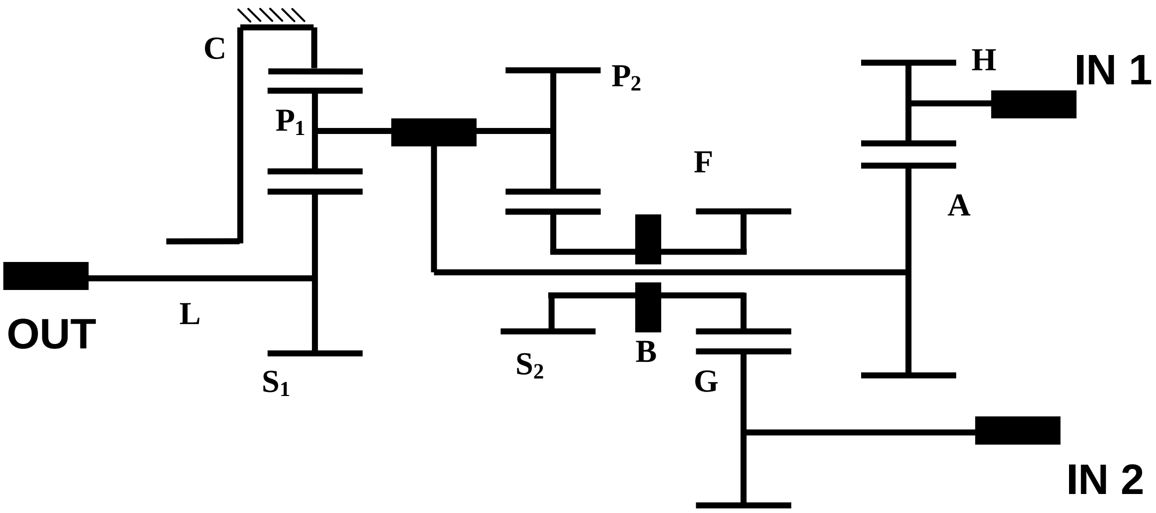

Figure (3) Epicyclic Gear train arrangement for two input and one output

5-1 System Arrangement and Operation

The gearing system arrangement consists of a ring gear (C) fixed to the casing , two fixed together planet gears (P_1) & (P_2) , two sun gears (S_1) fixed to Output shaft & (S_2), planet carrier fixed to gear (A), Input shaft(1) fixed to gear (H)which meshes with gear (A) , Input shaft (2) fixed to gear (G) which meshes with gear (F) and a mechanical unidirectional (selective) coupling (B)between gears (S_2 and F) . When less than two KVA is demanded the first engine on input shaft (1) lunches operating at (3000-3600) rpm delivering (3.5-4) (HP) and a maximum torque of 87.7 N.M at (H). Keeping the ring (C) fixed the output shaft (L) will turn in the same direction of (H) running the generator and delivering electricity. The gears (F) & (G) are kept stationary by means of the selective coupling (B). If more electrical power is wanted, the second engine may lunch and the input shaft (2) delivers torque to the system through the gears (G), (F) and the coupling (B). The two engines will run in parallel leading a double amount of power to generate up to 4 KVA.

5-2 Calculations of Gear ratio and Torque

Table (3) gives the number of teeth of the gears. Each engine delivers a power of 3.5(HP) 4.82 Kw at 3000 rpm up to 4(HP) 5.508 Kw at 3600 rpm.

Table (3) Number of teeth of the gears

	Gear	Number of teeth
1	P_1	10
2	P_2	12
3	S_1	20
4	S_2	18
5	C	40
6	F	18
7	G	22
8	A	30
9	H	10

All gears are of module (5). And the number of teeth is determined according to the reasonable available size

Using the tabular method to get the speed ratio we get

Function	Carrier A	C	P_1 & P_2	S_1 & L	S_2
Give whole system + a revolution	+ a	+ a	+ a	+ a	+ a
Fix A and give C +b revolution	0	+ b	$+b\, T_C / T_{P1}$	$- b\, T_C / T_{S1}$	$- b\, (T_C\, T_{P2}) /(T_{P1}\, T_{S2})$
Add	a	a + b	$a +(b\, T_C / T_{P1})$	$a-(bT_C / T_{S1})$	$a- b\, (T_C\, T_{P2}) /(T_{P1}\, T_{S2})$

Take the input speed 1 to be 3600 rpm

i.e. $N_H = 3600$ *C.W*

$N_H\ T_H = -\ N_A\ T_A$

$N_A = a = -\ 3600\ (10/\ 30) = -\ 1200$ *rpm*

Since C is fixed (the case) *i.e.* $N_C = 0$ *then*

$a + b = 0$

$b = -\ a = 1200$

$N_L = -\ 1200 - (1200)\ (40/20) = -\ 1200 - 2400 = -3600$ *rpm*

The gear ratio R is 1:3

The input torque at H is 87.7 N.M

And at A it is 263 N.M

From eq.9 the output torque is (-87.7) N.M which gives (2.2) KVA.

Later when the second engine works an additional torque will be delivered to the system through (G- F- B- S_2) and the total output torque will be (-175.4) N.M giving up to (4) KVA.

The experimental fuel consumption for (ASTRA) model (5000) is shown in **Fig.(4)** which clears that the minimum fuel consumption is at (0 to 2 Amp/hr) about 600 cc reaching the maximum value 2800 cc at (20 Amp/hr).

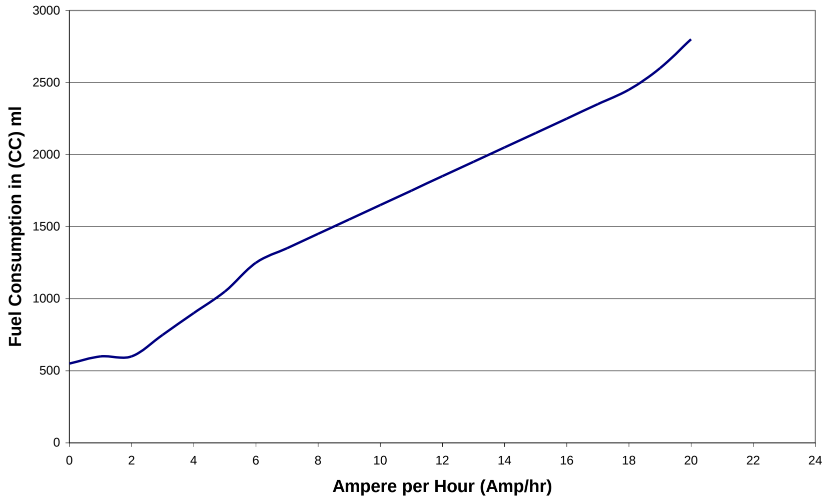

Figure (4) Fuel consumption for ASTRA 5000 generator

Figure (5) gives the experimental fuel consumption for model (2500) starting at 300 cc reaching maximum value 1450 cc at (10 Amp/hr).

Using a simulation to the system of two (163 cc) engines (which is used in model 2500) engaged through the gear train delivering (in tune) power to the generator of model (5000) to get a maximum power at (17 Amp/hr) taking into consideration the mechanical efficiency; the fuel consumption versus (Amp/hr) trend shown in **Fig.(6)**. The gain in fuel of the two engine system against the single engine is clear at (1-13 Amp/hr) starting with 300 cc getting the peak (at 6 Amp/hr) 475 cc falling to 200 cc and keep constant during (9-11 Amp/hr) then falling down and reaches the fair zone at (12.5 Amp/hr). After that the two engine arrangement will be non practical this can be clear in **Figs.(7 and 8)**.

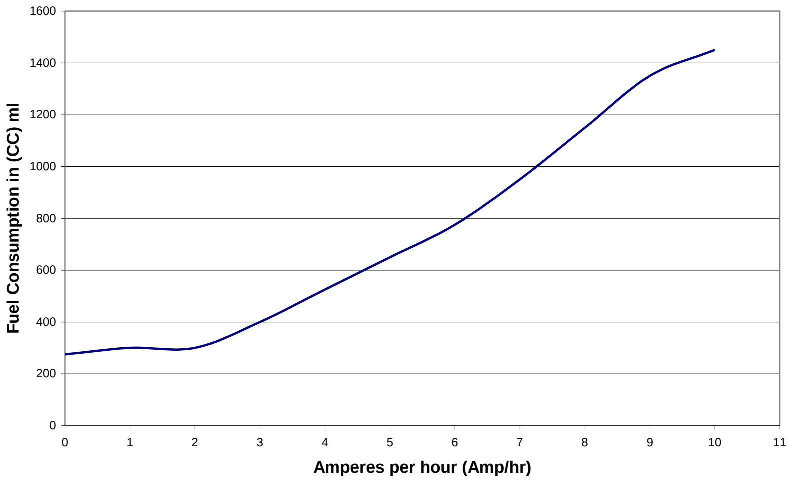

Figure (5) Fuel consumption for ASTRA 2500 generator

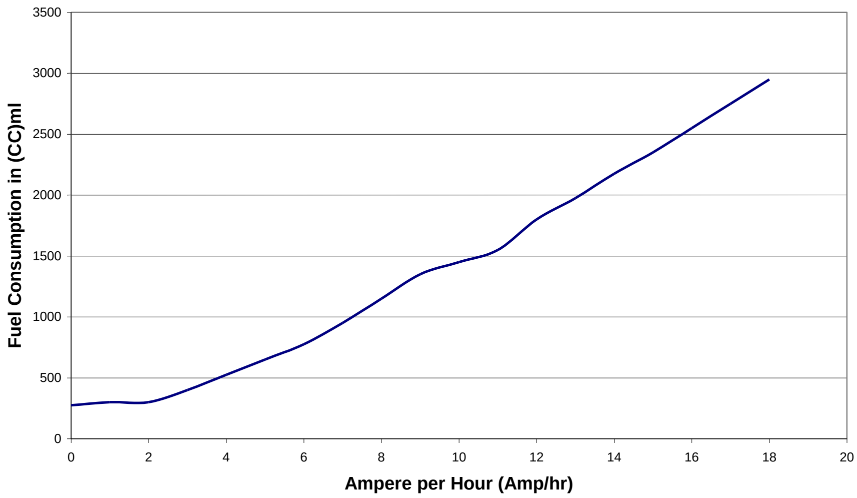

Figure (6) Fuel consumption for 2x(163) engines coupled by the gear train

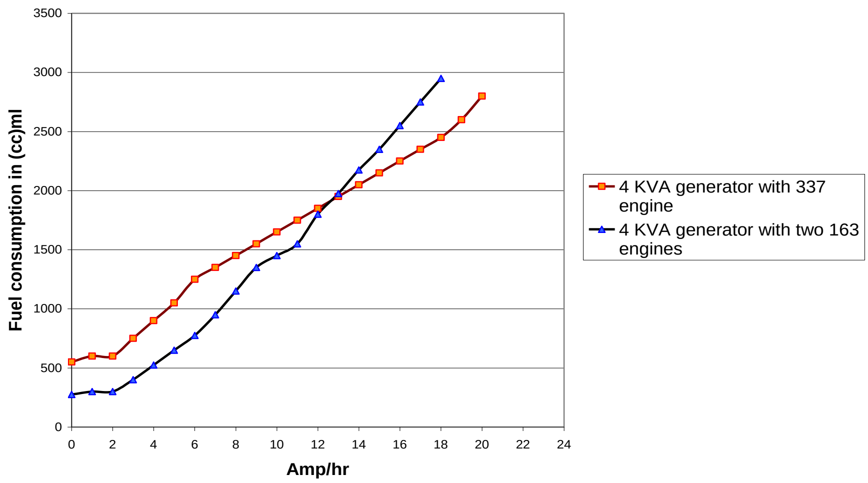

Figure (7) Fuel consumption versus Amp/hr for both cases

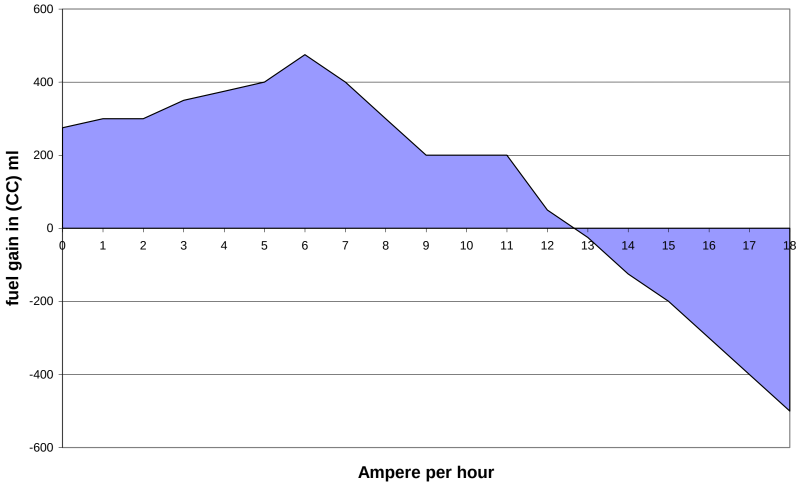

Figure (8) The gain in fuel when using 2(163) engines

6. Conclusions

From **Figs.(7 and 8)** above, it can be concluded that using single engine for the first (9 amperes) saves more than (300 CC) fuel per hour and starting the second engine when more than (9 amperes) demanded is more economic until reaching (12 amperes per hour) the point where the single (337) engine type will be more economic. So, the user can make a decision when buying a generator according to this study and find the suitable type for his daily needs.

7. References

1. Lyn wander, Peter, ***"Gear Drive Systems: Design and Application"***, Marcel Dekker, Inc., 1983

2. Dudley, Darle, ***"Gear Handbook: the Design, Manufacture, and Application of Gears"***, McGraw-Hill Book Company, Inc., 1962.

3. Willis, Robert, ***"Principles of Mechanism"***, Second Edition. London: Longmans, Green and Co., 1870.

4. Lévai, Zoltan, ***"Theory of Epicyclic Gears and Epicyclic Change speed Gears"***, Budapest: Technical University of Building, Civil, and Transport Engineering, 1966.

5. Fahim Alhimidani, ***"Developing a Simplified Model for Epicyclic Gear Train"***, The 1st Scientific Meeting for Engineering Materials Research Unit, Engineering College, Mosul University, Iraq, 2002.

6. Lent, Dean, ***"Analysis and Design of Mechanisms"***, Second Edition, Englewood Cliffs, NJ: Prentice Hall, 1970.

7. Shigley, Joseph Edward, and John Joseph Uicker, Jr., ***"Theory of Machines and Mechanisms"***, Second Edition, New York: McGraw-Hill Book Company, 1980.

8. Mabie, Hamilton H., and Charles F. Reinholtz, ***"Mechanism and Dynamics of Machinery"***, Fourth Edition. New York: John Wiley and Sons, 1987.

9. Molnar, john, ***"Nomographs"***, Ann Arbor: Ann Arbor Science Publishers, Inc, 1981.

10. Corey, Christopher A., ***"Epicyclic Gear Train Solution Techniques with Application to Tandem Bicycling"***, Thesis Submitted to the Virginia Polytechnic Institute, 2003.