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Electrical Generation from Thermal Solar Energy using a Turbocharger with the Brayton Thermodynamic Cycle

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Abstract

This paper addresses the use of Solar Thermal Energy to continuously generate electrical energy with a commercially available Turbo-Charger driving an Electrical Generator. The system is based on Brayton´s thermodynamic cycle and the energy is provided by a thermal accumulator. We propose that using air as the working media, the compressor element is capable of producing sufficient boost to achieve a useful level of efficiency for the cycle at the operating speed range claimed by the factory. The design or selection of the other important components of the system, such as the electrical generator, the starting motor, the electrical drive, the rectifying and inverting of the current is discussed. A parametric analysis that proves the theoretical feasibility of this model is included.

Keywords: Brayton; Thermal; Solar; Energy; Turbo-Charger; Pressure Ratio; Electrical Generator; Simulation.

Introduction:

There have been several successful applications where Solar Thermal Energy has been used to generate Electrical Power, but they have been mostly in higher production levels than what we propose here, also they tend to be too complicated, since normally the Turbine, the Compressor and the Generator are always designed specifically for each project, resulting in an expensive and complicated unit, this becomes even worse when combinations of more than one Turbine or Compressor are used. The purpose of this investigation is to prove if a single commercially available industrial engine Turbo-Charger can implement a Brayton-cycle based engine capable of driving an Electrical Generator to produce at least one Kilowatt, providing an option for domestic scale distributed generation. The most important questions to answer are: will the compressor be able to produce a Pressure Ratio, required by a Brayton cycle engine as it is, or does it have to be modified? Will the hot compressed air drive the Turbine to the required RPMs to produce the work required by both, the Compressor (back-work) and the Generator (net-work output)? How will the required heat be conveyed to the compressed air? Will the system setup be intrinsically safe? (to operate unattended on a household´s roof).

	Nomenclature:
$\eta_{Br.}$	Brayton cycle´s overall efficiency.
$\frac{P_2}{P_1}$	Ratio between the highest and the lowest pressures in the cycle.
k	Thermodynamic capacity of the media employed
A/R	relation between cross-section area and the radius of the centroid for a Volute [20]
Trim	area of the compressor’s inducer to area of exducer (ratio), and inversely for the turbine

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The Brayton Cycle

With the *Brayton cycle*, as with all other thermodynamic cycles, it is necessary to distinguish the cycle its self from the technological applications. Brayton cycle engines have been quite varied, and will resemble modern reciprocating engines, but the most important are those with a continuous flow of a thermodynamic fluid. It is the working principle for the gas turbine.

The development of the gas turbine occurs basically in the early 20th century, and comes as a result of the settlement of the problematic main technique associated with the Brayton cycle, i.e. the compression stage. The compression of a compressible fluid is not simple: piston engines solved the problem confining the gas in a closed Chamber - cylinder-, and by reducing the volume with a piston, thus increasing the pressure; however, this requires heavy and large engines for big powers, and calls for a high mechanical inertia in order to guarantee its continuous operation. [2]

The gas turbine uses a compressor, consisting of one or more steps of rotating blades that transmit a kinetic energy to the gas by accelerating it and then, through some fixed blades or diverting passages that slow it down to convert the added energy into pressure. It was also the advance of technology, the development of new materials and getting a better understanding of fluid mechanics for men to produce the first truly effective compressors, and with that, the first gas turbines.

On these devices, after compression followed the addition of fuel in a rudimentary combustion chamber where it burned, and then expansion was set in a turbine, producing mechanical work, part of which is used to drive its compressor, and the remaining power can drive any mechanical device such as a generator. Also a Brayton Turbine can have external combustion with the hot compressed gases entering the turbine and converting its energy by expanding and making it turn.

The application of a gas turbine based on the Brayton cycle to airplane propulsion is due to the English engineer Frank Whittle, who in 1927 patented the idea and proposed it to the British air force. The gas turbine would only power the compressor, and propulsion would come from the high speed of gases at the output of the turbine, forming a propulsive jet which would generate a thrust force. The idea of Whittle was also raised at about the same time by the German Hans von Ohain. During World War II a frenetic race between the two sides towards the development of the first Jet engines. Since then, based on the Brayton cycle, the gas turbine would dominate as aircraft propulsion system. At the same time it continued being implemented within the generation industry.

To use air as a thermodynamic fluid, the Brayton cycle requires high temperatures to achieve reasonable performance levels.

Variations on the basic cycle, like multiple stages in compression or in expansion, or the combination with a Rankine cycle machine in what is called the “Combined cycle”.

The Brayton cycle can be open or closed. In an open-cycle, ambient air is compressed in iso-entropic form with a rotary axial or centrifugal compressor, then it enters a combustion chamber where fuel is injected burning the fuel at constant pressure, the product of this combustion then expands in a turbine, down to its output pressure which, once again is the environment pressure. Real gas turbines have open cycles, where air enters continuously.

The thermal efficiency of the Brayton cycle depends mainly on the relationship of pressures, inlet gas temperature to the turbine (TIT) and parasitic losses (especially the efficiencies of the compressor and the turbine).

$$\eta_{Br.} = 1 - \frac{1}{\left[\frac{P_2}{P_1}\right]^{\frac{(k-1)}{k}}} \qquad (1)$$

The real cycle differs from the ideal cycle due to the properties of real air (k, C_p) that are not constant over the whole range of temperatures, and as far as the internal losses, these become significant beyond 1367^0K and even useless beyond 1922^0K. [1,4].

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The closed Brayton cycle:

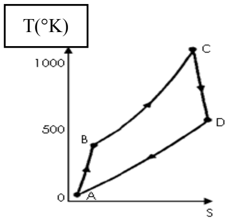

Fig. 1 Temperature as a function of Entropy in the Closed Brayton Cycle

It is assumed as an ideal cycle therefore the working fluid undergoes the four internally reversible processes.[2,3].

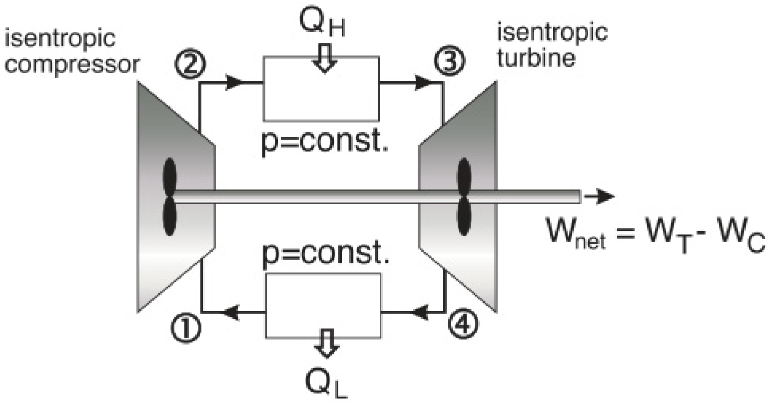

Fig. 2 Ideal Brayton Closed Thermal Engine configuration

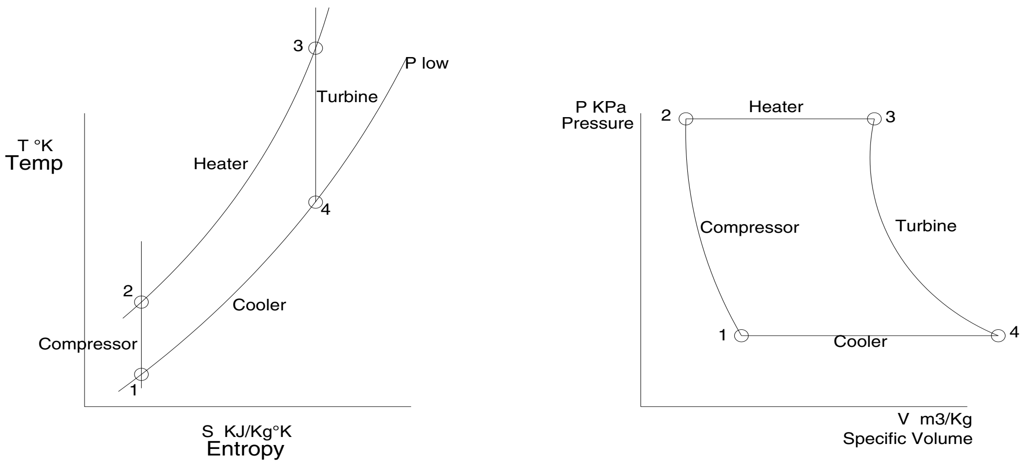

Fig. 3 Ideal Closed Brayton Cycle (T=f(S) and P=f(v))

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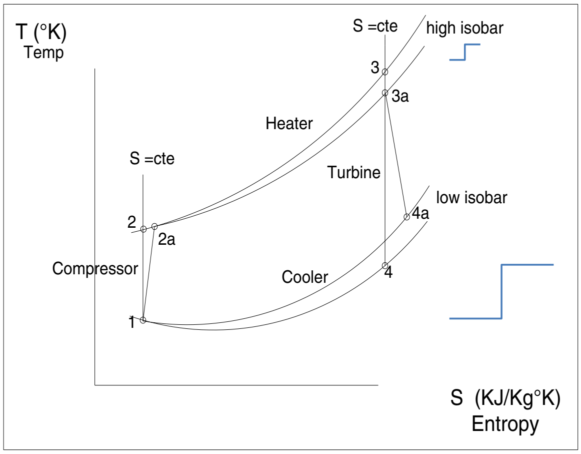

Fig. 4 Real Closed Brayton Cycle T=f(S)

In Real Cycles, compression is not completely isoentropical (1-2), but politropical (1-2a), and this is due to friction losses mostly.
Heating is not isobarical (2-3), also due to friction losses within the Heater (2a-3a).
Expansion is also politropical (3a-4a).
Cooling is polibarical (4a-1).
Every one of these four portions of the cycle may take place in a single stage, or in multiple stages, which can be accommodated to achieve efficiency improvements.

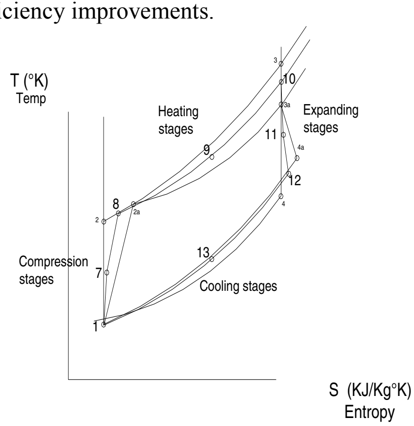

Fig. 5 Multiple stages Real Closed Brayton Cycle T=f(S)

Having a larger enclosed area on the same diagram implies larger efficiency for the same cycle, so by multiplying the stages on each sector of the cycle, an "interpolation" can be obtained as can be seen in the diagram (fig 5): Using two compression stages [14], with the same hardware efficiencies, we go from 1 to 7 and to 8 instead of from 1 to 2a, enhancing the enclosed area. Likewise heating from 8 to 9 to 10, instead of 2a to 3a. Using two turbine stages, from 10 to11 to 12, instead of 3a to 4a. And accordingly, from 12 to 13 to 1 instead of 4a to 1.

Parametric basis:

The commercially available turbo-charger selected for this experiment is the model GT5533R made by Honeywell-Garrett, meant for diesel engines with displacement between 3 and 12 Liters with power ranging from 1000 to 1600 break-Hp. The one selected has 91.2 mm diameter inductor and 133.3 mm. exductor, a trim of 47 and A/R 0.69. which has the following operating parameter ranges: [20]

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For the compressor we take

Corrected Air Flow (in lbs/min)	80	100
Speed (in rpm)	45000	62500
Pressure Ratio	2.4	2.83
η_{max} Efficiency (for the compressor).	77%	77%

To start the engine, the goal is to bring it up to 45000 rpm by means of a starting motor that would produce: the Corrected Air Flow of 80lbs/min and a Pressure Ratio of 2.4

Plotting equation (1), that relates η_{Br} (the Efficiency of the Brayton cycle) as a function of Pressure Ratio produces the following graph.

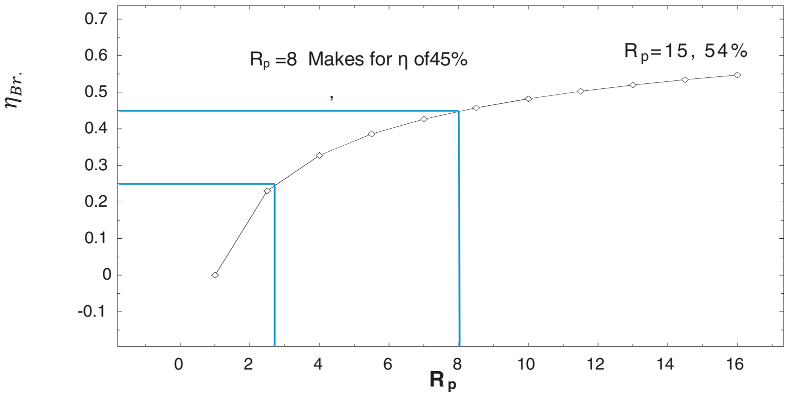

Fig.6 Brayton Cycle Overall Efficiency as function of Pressure Ratios

Since a Pressure Ratio of 1:8 brings 45%, and beyond that point the curve turns quite flat, most aircraft turbines, use three consecutive centrifugal compressor stages producing each a C.R. of 1:2. [5.8]
On the other hand, industrial turbines where weight is no object, it is normal to find total C.R.s of 14 or 15 achieving η up to 54%. [18]
In selecting the working media, the specific heat capacity of the fluid is important, on the one hand Hydrogen (H_2) has the largest by far (14.3 KJ/Kg°K), but is delicate (may be even hazardous) on its handling. Air is somewhat similar to Neon, both at the temperature ranges we will be facing have around 1.02 KJ/Kg°K. [16] it is free and calls for no concern on hermeticity.

The operational rotating speed range must be over 40,000 rpm, which works well for the efficiency of the turbine, according to its manufacturer (Garrett). [7,20].

Using air as the media, the compressor will handle 100 lbs./min, equivalent to 0.76 Kg/s.[20].

This air requires its temperature to be increased by some 300°C, assuming an overall thermal efficiency in the 25% range; an **n** account of the "back-work" the turbine must provide to compress the air.

Since the compression is expected to be only 1:2.83, then the overall thermal efficiency would only be 25%, so the heat input must be adjusted accordingly. Simulation is being made in order to validate these calculations.

As for the electrical generator, a commercially available one that operates within these ranges is being searched for, but the alternative of using a geared belt speed reducer is also under consideration.

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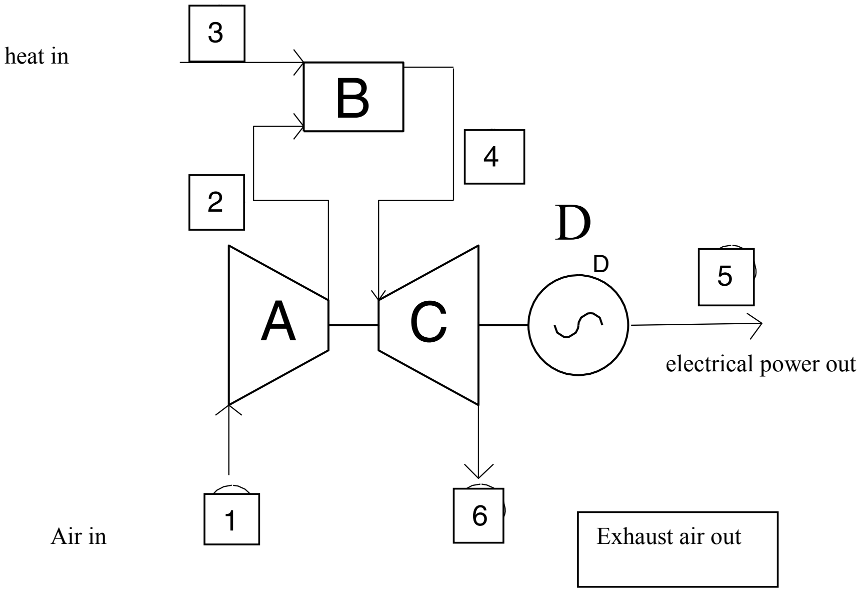
This constitutes the initial setup: It consist of a single spool industrial diesel turbocharger (Garrett GT5533R) and a heater placed between the compressor and the turbine (fig. 6). This Heater, for the time being is fed by electrical resistors, (a total resistor load of 5000Watt), this way a precise amount of energy can be supplied in order to achieve the proof of concept. It is anticipated, that a motor/generator (45 KRPM and some 3-4 KW), will start it and drive it to a speed (above 20000 rpm), then the electrical heat energy must be applied to achieve at least a self-sustained continuous motion and then, the motor shall be switched to work as a generator. At first as mentioned above an open circuit is used, so it will take atmospheric air, compress it, heat it, then pass the heated air mass through the turbine, expanded and expel it back to the atmosphere.

Parametric Analysis:

Assuming the initial setup in order to prove the concept, the turbocharger (GT5533R) is connected on it's discharge (exducer) to a heater where electrical resistors are used to provide in a practical and measurable way the required energy to make it work. The final objective will be to supply it with Concentrated Thermal Heat from Solar origin. This setup is shown in fig (6)



A compressor	1 fresh atmospheric air
B heater	2 compressed air
C turbine	3 heat input
D electrical generator	4 hot compressed air
	5 electrical output
	6 exhaust air output

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Item	name	parameter	Value	unit	relation	comments
1	**INTAKE AIR**	Temp 1	298	°K	T1=To	Ambient
		Press1	100,000	Pa	P1=Po	
		F1 flow	0.76	Kg/s	= 100lbs/min	Nominal flow of Turbo-Charger at 62500 rpm
		H1	298.18	KJ/Kg		
		S1	1.69528	KJ/Kg°K		
2	**COMPRESSED AIR**	T2	405	°K		$T_2 = T_1.R_c{}^{(k-1)/k}$ formula(5)
		P2	283000	Pa	P2=P1*CR	
		F2	0.76	Kg/s	F2=F1	
		H2	405.86	KJ/Kg		H1+1.005*(T2-T1)
		S2	1.70	KJ/Kg°K	S2=S1	assumed isoentropic as first approach
3	**INPUT HEAT**	HEAT 3	5000	J/s (W)		added heat
		T3	983	°K		
		P3	283000	Pa		same as P2 disregarding losses
		H3	6984.80	KJ/Kg		H2+HEAT3/F2
		S3	6.78	KJ/Kg°K		S2+HEAT3/T3
4	**HOT-COMPRESSED AIR**	T4	934	°K		T3*ηH/100
		P4	283000	Pa		same as P3 disregarding losses
		F4	0.76	Kg/s		same as F1
		H4	6984.80	KJ/Kg		same as H3
		S4	6.78	KJ/Kg°K		same as S3
5	**EXHAUST AIR**	T5	692	°K		$T_2 = T_1.R_c{}^{(k-1)/k}$ formula(5)
		P5	102807	Pa		the turbine´s expansion ratio
		F5	0.76	Kg/s		same as F1
		H5	-242.71	KJ/Kg		1.005*(T5-T4)
		S5	6.78	KJ/Kg°K		same as S4

Table 1, Parametric Analysis of the Flows

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item	name	parameter	Value	unit	relation	comments
A	**COMPRESSOR**	CR	2.83		P2=P1*CR	Compression Ratio
		FA	0.76	Kg/s		
		Wc	81.57	J/s (W)		Work Compressor
		ΔPCf	193,000	Pa	P2-P1	boost
		RPM	62,000	rpm		RPM
		ΔH	107.68	KJ/Kg		increase in enthalpy
		ΔS	0	KJ/Kg°K		iso entropic
		ΔTA	107.14	°K		$T_2 = T_1.{R_c}^{(k-1)/k}$ formula(5)
		ηC	85	%		compressor efficiency
B	**HEATER**	heat req.	1331.88	J/s (W)		net Heat required
		Heat	5000	J/s (W)		Heat Input in Joules/sec (W)
		FH	0.76	Kg/s		same as F1
		ΔTh	392.93	°K	T4-T2	
		ΔHh	3960	KJ/Kg	H4-H2	
		ΔSh	3.05	KJ/Kg°K	S4-S2	
		ΔPCf	-24172.5	Pa	´-3% (P3)	Pressure Loss by friction (-3%)
		ηH	95	%		
C	**TURBINE**	Er	2.85			Expansion Ratio turbine
		Ft	0.76	Kg/s		same as F1
		Wc	-78.31	J/s (W)		Work Compressor (out)
		Wgen	-1052.63	J/s (W)		Work on Generator
		ΔHt	-242.71	KJ/Kg		
		ΔSt	0	KJ/Kg°K		iso entropic
		ΔPt	-523030.702	Pa		expansion
		RPM	62,000	rpm		RPM high spool
		ΔTt	-241.51	°K		cooling in turbine
		ηt	78	%		turbine´s efficiency
D	**GENERATOR**	EPout	1	KW	Electric Power output =	Design Principle
		Vout	28	Vdc		Voltage output (dc)
		Iout	35.71	Adc		Current output (dc)
		Wgen	1052.63	J/s (W)		Work on Generator
		RPMh	62,000	rpm		RPM
		ηE	95	%	Epout/Wgen	Overall Efficiency generator

Table 2. Operating Parameter Analysis for the System Components

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Modeling and simulation:

Based on the calculations shown above, simulation for the existing mechanical elements is being made as follows [7,11]: The compressor´s rotor has been faithfully modeled using Inventor®, as well as the compressor body in order to make simulations looking forward to modify their design pursuing higher boost, within its operating parameters such as speed and flow-rate.

Fig 8 Modeled compressor rotor.

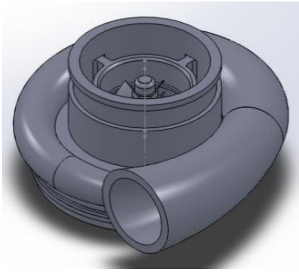

Fig 9 Modeled volute

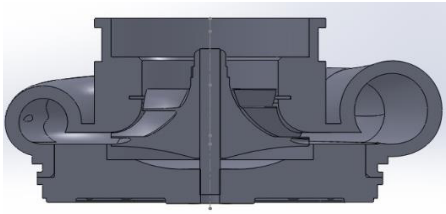

Fig 10 Modeled assembly

Conclusions:

According to the analysis made, the use of a commercially available Turbo/Charger to drive an electrical generator to produce continuously approximately one Kilowatt, and the application of Thermal Stored Solar Energy is proven feasible.
Air can be selected as the working media, it has high enough thermal properties, it is free and requires no handling consideration.
Simulations and D.O.E. are being prepared to identify the optimal operating conditions.

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