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FLOATING SOLAR CHIMNEY POWER STATIONS WITH THERMAL STORAGE

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ABSTRACT

A floating solar chimney power station (FSCPS) has three major components:

- A circular solar collector
- A floating solar chimney in the center of the solar collector
- A set of air turbines geared to electric generators around the bottom of the solar chimney

The efficiency of the FSCPSs is roughly proportional to the height of their floating solar chimneys. The floating solar chimneys (FSCs) are made by successive balloon tubes, that are filled with He or NH_3. This permits to the FSCs to float in the air and thus to have heights a few Kms. In the present paper the effect on the FSCPS's operation of the thermal storage, due to ground or water in plastic bags or closed plastic tubes, is examined. It is shown that the FSCPSs equipped with thermal storage can operate 24 hours per day 365 days per year with a minimum guaranteed power production. This means that the FSCPSs, although renewable by nature, have a similar operation to conventional power stations. Large FSCPSs (above 20 MW) can be very cost competitive to any conventional power station due to their low construction, maintenance and operation costs.

KEY WORDS
Thermal storage floating solar chimney Power Station.

1. INTRODUCTION

The solar chimney power production method was an idea proposed at 1931 by the German writer Hanns Gunther. Solar chimney power stations were tested and supported strongly by prof. J. Schlaigh. In his book ref. [1] gives an extensive presentation for them.

The solar chimney power station is mainly a set of three components:

- A large circular solar collector of diameter D_c, supported a few meters above the ground and covered by a transparent glazing (The Greenhouse).
- A tall, air up drafting, lighter than air cylinder in the center of solar collector with internal diameter d and height H (The floating solar chimney).
- A set of air turbines geared to appropriate induction electric generators around the bottom of the solar chimney (The Turbogenerators).

An indicative diagram for a STPS is shown in fig.1.

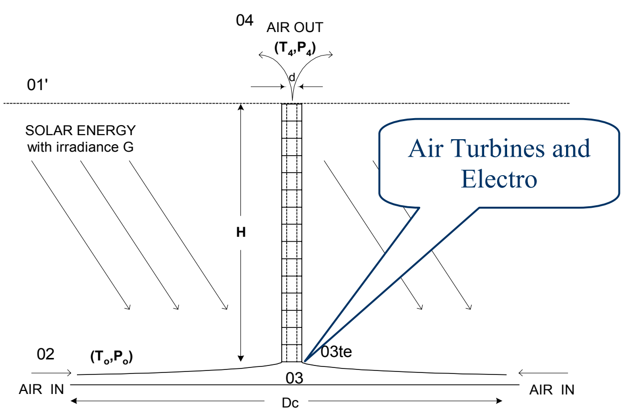

Fig. 1

Due to greenhouse effect the air is warmed in the solar collector. The warm air is moving from the periphery of the solar collector to its center towards the entrance of the solar chimney, in order to ''escape'' to upper layers of atmosphere due to buoyancy. This moving stream of warm air leaves part of its thermodynamic energy to the air turbines that are geared with appropriate induction electric generators. The shrouded axial air turbines are placed in the path of airflow, with horizontal axis in a circle of diameter 2d, around the bottom of the solar chimney. Backstrom and Cannon in ref [2,3,4,5] proposed a thermodynamic cycle analysis for the FSCPS operation. The author extended their study producing analytical results see ref [6,7,8].

2. THE FLOATING SOLAR CHIMNEY (FSC) A SHORT PRESENTATION

The Floating Solar Chimney (FSC) was invented by the author see www.floatingsolarchimney.gr, and is a lighter than air structure that can encounter all the effects arising by the FSCPS's operation and external winds see fig (2).

The FSCs are lighter than air cylindrical structures made of a set of successive parts. The latest shape, produced by the author, of such a part is made as a successive series of lifting tubes, where in between are placed supporting inflated tube rings filled with pressurized air, in order to resist to the operational sub pressure acting on its wall. A computer animation for a part of a FSC is shown in fig. (3). The heavy base at the bottom of the FSC can incline on its seat, when external winds appear, giving to the structure its deflecting ability, see fig. (4). Thus external winds are encountered.

The necessary strength of the pressurized ring tubes, preventing the deformation of the cylinder, depends exclusively by the operational sub pressure acting on FSC's cylinder. The weight of the lifting and supporting tube rings is defining the diameter, and thus the quantity of the lifting gas, of the lifting balloon tubes in order the structure to self float in the air . The existing modern plastic and composite fabrics and fibers, tested already to air ship and inflated structures have given valuable information and know-how for an appropriate construction of the FSCs ,in order to resist to external strong winds and sub pressures. An excellent presentation on the progress in this field is given in ref [9 10 11].

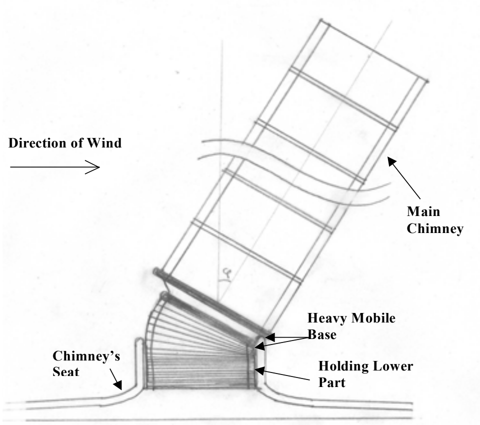

Fig. 2

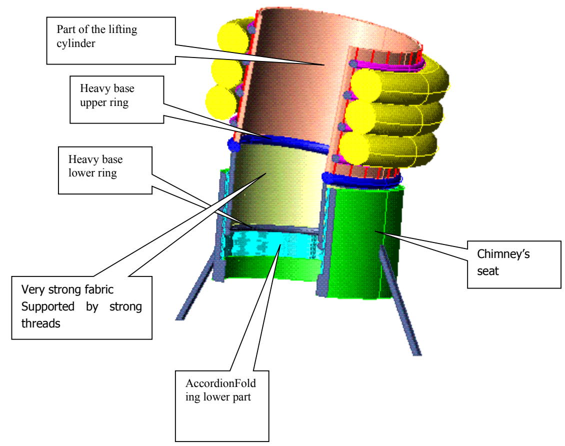

Fig. 3

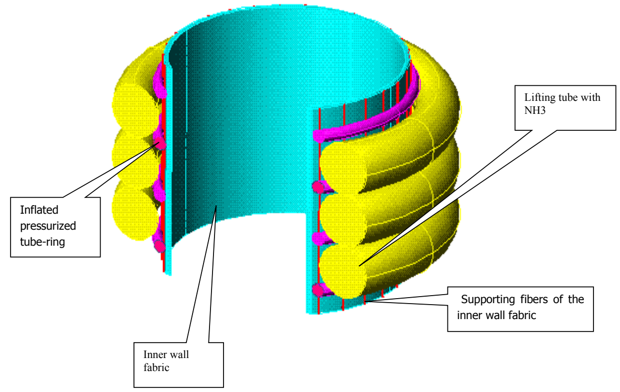

Fig. 4

3. FSCPS OPERATION

The fundamental equation for the operation of the FSCPS i.e. the electric Power Output P_{FSC} as function of the moving air mass flow $\dot{m}$ has been derived by the author in ref [6]. A presentation of the results of this analysis is given below:

The equation is the following:

$$P_{FSC} = C_p \cdot \dot{m} \cdot \left(T_{03} - C_1 - T_4 - C_2 \cdot T_4^{\,2}\right) \quad (1)$$

Where T_{03} is the entrance stagnation air temperature in the air turbines. T_{03} ,that is also the exit air temperature by the solar collector, can be defined exclusively by the solar collector thermal analysis, as it will be shown in the next paragraph.

T_4 is the appropriate root of the polynomial equation:

$$w_1 T_4^{\,4} + w_2 T_4^{\,3} + w_3 T_4^{\,2} + w_4 T_4 + w_5 = 0 \quad (2)$$

where w_1, w_2, w_3, w_4, w_5 are given by the relations:

$$w_1 = C_2^{\,2}(1-k)$$

$$w_2 = C_2\left(2 - k - n_T C_2 T_4'\right)$$
$$w_3 = C_2 C_3 (1-k) + 1 - 2n_T C_2 T_4'$$
$$w_4 = C_3 - n_T T_4'(1 + C_1 C_2)$$ and
$$w_5 = -n_T T_4' C_1$$, where:
$$C_1 = g \cdot H / C_p ,$$
$$T_4' = T_{03} \cdot (1 - C_1 / T_0) \quad C_2 = a/2 \cdot C_p (R \cdot \dot{m} / A_{ch} \cdot p_4)$$
$$C_3 = T_{o3}(n_T - 1) + C_1 \quad , A_{ch} = \pi \cdot d^2 / 4 \quad ,$$
$$p_4 = p_o (1 - C_1 / T_0)^{3.5}$$, R=287 J/Kg ^{0}C,
g=9.81 m/sec^2 and C_p=1005 J/Kg $_0$C.

T_o is the ambient temperature, P_o is the ambient atmospheric pressure, η_T is the efficiency of the turbines and generators, k is the FSC's friction loss coefficient, α its kinetic energy correction coefficient and $\dot{m}$ the mass flow.

The optimum performance of the air turbines can be achieved see ref [8] for the maximum turbine efficiency that depends on the ratio υ/υ_{tip}. where υ is the inlet air speed (proportional to the mass flow $\dot{m}$) entering the turbines and υ_{tip} the radial speed of their blades' ends. The υ_{tip} for induction generators connected to the grid is almost constant, defined by the grid frequency. It can been proved, see ref [8], that using shrouded axial air turbines geared to induction generators we can regulate the system for optimum operation (i.e. to achieve maximum η_T), for the appropriate $\dot{m}$ where P_{FSC} it becomes also maximum.

4. CIRCULAR SOLAR COLLECTOR OPERATION

According to Shlaigh see ref [1] an approximate equation relating the exit solar collector temperature T_{03} to its input temperature T_0 valid for every circular solar collector is given by:

ta·G·A_c= $\dot{m}$ ·C_p·(T_{03} - T_{02}) + β·A_c· (T_{03}-T_{02}) (3)

β is the approximate thermal power losses coefficient of the Solar Collector, to the environment and ground per m^2 and ^{0}C for the temperature difference (T_{03}-T_0) .

An average value for β is ~5.75 W/m^2 ^{0}C.

G is the irradiance on the horizontal surface of the solar collector.

ta is the average of the product {glazing roof transmissivity for solar radiation × soil absorptivity }.

An average value for the coefficient ta is ~ 0.75 .

A_c is the Solar collector surface area.

Thus an approximation for the function T_{03} ($\dot{m}$), necessary for the FSCPS power calculation is:

T_{03} ($\dot{m}$)= [ta·G / (β + $\dot{m}$ ·C_p/A_c)] −T_{02} (4)

A more accurate function can be derived with a detailed thermal analysis of the circular collector by dividing it in circular segments of Δr width and r average radial distance as shown in fig.(5).

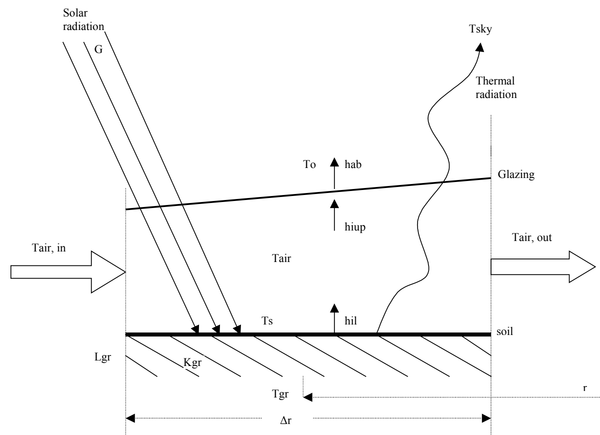

Fig 5

$T_{air\ in}$ is the moving air inlet temperature in the segment
$T_{air\ out}$ is the moving air exit temperature by the segment
T_{air} is the moving air average temperature in the segment, thus:

T_{air}= ($T_{air\ out}$ +$T_{air\ in}$)/2 (5)

T_s is the segment's average soil temperature (the soil is acting as the selective surface of the collector)

T_{gr} is the average ground temperature considered constant (~15^{o}C) after a reasonable depth L_{gr}.

Thus the conduction losses to the ground, can be estimated by the formula h_{gr}·(T_s-T_{gr}).

where h_{gr}=(K_{gr}/ L_{gr}). An estimated value for h_{gr} is ~ 0.8 W/m^2 ^{o}C.

The convection heat transfer coefficients inside the collector are h_{inl} and h_{inup}.

The lower and upper surfaces are equal but they have different roughness. Their convection heat transfer coefficients can be calculated approximately by the following procedure.

Each segment can be considered as a special rough tube with equivalent diameter d_e equal to 2 times its height he(r). Thus its Reynold's number given by R_e=V·d_e·ρ/μ is equal to R_e= $\dot{m}$ /(μ·π·r) .

For usual values of T_{air} , μ can be received in average as equal to 1.9·10^5.

For 10^5 <Re<10^7 and roughness between 10^{-3} and 10^{-6} their Nusselt number,

according to J.P. Holman [12], is given by Nu=f·Re·Pr$^{1/3}$/8 (6)

where f is the friction factor calculated by the Moody's diagram or by the empirical relation f=1.325/[ln(ε/3.7de)+5.74/Re$^{0.9}$]^2 (7) and Pr for air is ~0.7.

Thus h_{in}=K_{air}·Nu/d_e (8) ,where the air conduction K_{air} is ~0.0275 W/m ^{0}C and Nu is calculated for the lower inner surface (l) or the upper inner surface (up) using their respective relative roughness ε/d_e.

The roughness ε for the lower surface (ground) can be considered equal to 0.05 m while for the upper (glazing) is not more than 0.001 m. Thus h_{il} > h_{iup} due to its higher roughness (that should be created in purpose facilitating the moving air's heating by the soil).

The thermal radiation losses of the soil to the sky can be estimated by the formula:

Q_{rad} =[(glazing transmissivity for thermal radiation × soil emissivity)] $\cdot\sigma\cdot(T_s^4-T^4_{sky})$ (8)

where σ is the Stephan-Boltchman constant equal to $5.67\cdot10^{-8}$ W/m^2 ^{0}K^4.

A linear approximation for the thermal radiation losses is $h_{rad}\cdot(T_s-T_{sky})$. For the usual operating temperatures inside the solar collector and average values for thermal radiation transmittance and soil emittance h_{rad} is approximately equal to ~1 W/m^2 ^{0}C.

T_{sky} is, in the worst case, equal to the ambient air temperature minus 10 ^{0}C, i.e. $T_{sky}=T_0-10^0C$.

The transparent roof conduction losses are very high thus, h_{con}~ 340÷1000 W/m^2 ^{0}C.

The convection losses above the glazing of the solar collector, are difficult to be estimated due to external variable winds. According to Duffie and Beckman [13], due to the huge and smooth surface of the external glazing its convection heat transfer coefficient should not exceed h_{ab}= 10 W/m^2 ^{0}C.

The thermal relations for each segment are:

$(T_{air\ out}-T_{air\ in})\cdot Cp\cdot \dot{m} = [h_{il}\cdot(T_s-T_{air})-h_{up}\cdot(T_{air}-T_0)]\cdot(2\pi r\Delta r)$ (9), derived by the moving air thermal equation. Where h_{up} is the overall upper thermal coefficient calculated by the formula:

$h_{up}=1/(1/h_{iup}+1/h_{con}+1/h_{ab})$ (10) .

Using (5) the equation (9) can be written as:

$h_{air}\cdot(T_{air}-T_{air\ in}) = h_{il}\cdot(T_s-T_{air})-h_{up}\cdot(T_{air}-T_0)$ (11)

where: $h_{air}=2\cdot Cp\cdot\dot{m}/(2\pi r\Delta r)$.

The soil thermal equation gives the following relation:

$h_{gr}\cdot(T_s-T_{gr})+h_{il}\cdot(T_s-T_{air})+h_{rad}\cdot(T_s-T_{sky})=ta\cdot G$ (12)

The equations (11) and (12) form a system relating T_{air} and T_s to the inlet temperature of the air to the segment $T_{air\ in}$.

Observing that the exit air temperature off the segment is the inlet air temperature to the next segment, in the direction of mass flow, we can form a set of iterative relationships for the successive segments of the solar collector. Hence for a given set of M segments of equal thickness $\Delta r=(D_c/2-d)/M$, their average radial distances are, $r(m)=D_c/2-\Delta r\cdot(m-1/2)$ (for m=1÷ M) and thus we can form a set of M iterative relationships. The exit air temperature by the solar collector T_{03} is equal to the $T_{air\ out}$ of the M_{th} segment that can be defined by the produced algorithm by the iterative relationships given the $T_{air\ in}$ of the first segment, the moving air mass $\dot{m}$ and the thermal loss of the successive segments.

The average height of the m_{th} segment is given by: $he(m)=H_{in}+(H_{out}-H_{in})\cdot(m-1/2)/M$ (13)

where H_{in} and H_{out} are the input and output heights of the solar collector.

As an example we have calculated the exit air temperatures T_{03} for a circular solar collector of: D_c=2550 m ,H_{in}= 3 m and H_{out}= 10 m, by the two procedures for various values of $\dot{m}$.

If we assume that this Solar Collector is part of a FSCPS (of rated Power output ~100 MW) with a set of air turbines and electric generators with efficiency η_t=0.8 and a Floating Solar Chimney with H=3000m, d=50m, k=1 and a=1.058 and G=667 W/m^2 , T0=20 ^{0}C and p_0=101300 Pa, we can produce its operation curves for the two calculated functions T_{03} ($\dot{m}$) derived by the approximate (1) and the more accurate procedure (2) for M=100.

The curves P_{FSC} ($\dot{m}$) are shown in fig (6). As it is obvious the curves are reasonably close.

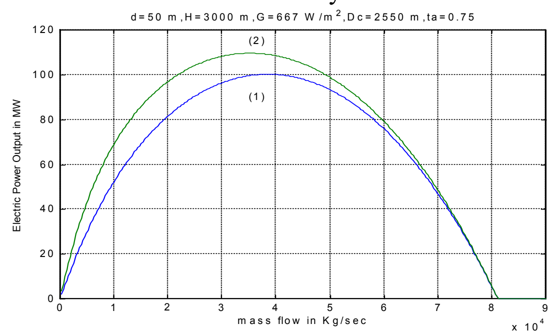

Fig 6

5. THERMAL STORAGE

The latest paper for thermal analysis of FSCPSs with thermal storage, was presented by Bernades e.t.c in ref [14]. Their analysis is rigorous, rather complicated and interconnected with the thermodynamic analysis of FSC's. My approach, that will follow, I believe that is simpler, independent of FSC's thermodynamics, and can be used easily for the solution of any problem related to thermal storage operation of FSCPSs.

The equations of the previous paragraph, for the segments of the solar collector, they did not take into account the ground thermal storage or any artificial water thermal storage. In order to simplify the thermal storage analysis, the solar collector's thermal storage can be considered as a thin layer of negligible thickness on its surface with an appropriate surface storage ability C (in J/m^2 ^{0}C). C will count the ground's soil thermal storage or and any artificial water thermal storage applied to the ground of it. For example a layer of one cm water thermal storage will have a C=41868 J/m^2 ^{0}C .

The thermal equations of each peripheral segment, of width Δr and average radial distance r, become time dependent in order to take into consideration the thermal storage. Thus the equations become as follows:

$h_{air}\cdot(T_{air}-T_{air\ in}) = h_{il}\cdot(T_s-T_{air})-h_{up}\cdot(T_{air}-T_0)$ (14)

$C\cdot dT_s/dt = ta\cdot G-[h_{gr}\cdot(T_s-T_{gr})+h_{il}\cdot(T_s-T_{air})+h_{rad}\cdot(T_s-T_{sky})]$ (15)

T_{air} ,$T_{air\ in}$ and T_s ,as G ,T_0 and T_{sky} are functions of time (t). The thermal constants h_{il},h_{up},h_{gr},h_{rad}, ta and C are assumed constants for each segment of area 2πrΔr. The mass flow $\dot{m}$ can be constant or time dependent and thus

$h_{air}=2\cdot Cp\cdot \dot{m}/(2\pi r\Delta r)$ the same. The relations (14),(15), in their diurnal variation, can be written as functions of a time integer n (where n=1:N+1) related to time t in hours, through the equality : t=24·(n-1)/N (division of the day in time intervals of dt=24/N hours=86400/N sec):

$h_{air}(n)\cdot(T_{air}(n) - T_{air\ in}(n)) = h_{il}\cdot(T_s(n)-T_{air}(n)) - h_{up}\cdot(T_{air}(n)-T_0(n))$ (16)

$C[T_s(n+1)-T_s(n)]/(86400/N) = 1/2[F(n+1)+F(n)]$ (17)

Where :

$F(n)=[\ ta\cdot G(n)\ -[\ h_{gr}\cdot(T_s(n)-T_{gr})+\ h_{il}\cdot(T_s(n)-T_{air}(n))+h_{rad}\cdot(T_s(n)-T_{sky}(n))]$ (18)

We can use the relationships (16),(17) and (18) as an iterative algorithm in order to calculate the diurnal variations of $T_s(n)$ and $T_{air}(n)$, for n=1:N+1 with initial conditions: $T_{air\ in}(n)$ a given function and $T_s(1)$ a given initial soil temperature. If we would like to eliminate the transient effects, due to arbitrary choice of $T_s(1)$, we can run the algorithm a few times replacing $T_s(1)$ with the $T_s(N+1)$ of the previous run.

Assuming that the algorithm is applied to the first segment (m=1) the function $T_{ain\ in}(n)$ is equal to$T_0(n)$.

For the next segment $T_s(1)=T_s(N+1)$ of the previous segment and $T_{air\ in}(n)=T_{air\ out}(n)$ of the previous segment that is equal to $[2*T_{air}(n)-T_{air\ in}(n)]$ of the previous segment e.t.c. up to the M_{th} segment.

Thus finally $T_{air\ out}(n)$ of the last segment M, is calculated. This is the function $T_{03}(n)$ i.e. the diurnal variation of the inlet air temperature to the air turbines. Hence using the equations (1) and (2) the diurnal variation $P_{FSC}(n)$ can be calculated for the assumed mass flow $\dot{m}$ (n) .

6. THERMAL STORAGE EFFECTS ON THE DIURNAL OPERATION OF THE FSCPSs

In order to apply the previous theory to a given FSCPS the functions G(n) and $T_0(n)$ should be given. These functions are depending on the place of FSCPS's installation where the weather data are varying day by day. As an average day we can consider the day for which the daily irradiation on horizontal surface is equal to the annual irradiation W in KWh/m^2 divided by 365 (days) . For simplification we can consider that the daily irradiation is a sinusoidal function of the number of day D (D=1: 365) given by:

$w(D)=W/365+\Delta w\cdot\cos[(D-212)\cdot 2\cdot\pi/366]$ (19) in (KWh/m^2), where Δw is the maximum variation of the daily irradiation from its average value W/365, the day 212 is, as an example, the first day of august considered as the day with the maximum irradiation for the place of installation of the FSCPS.

The diurnal variation of the solar irradiance G(D,n) it can be considered as a sinusoidal function for the daylight hours dlt given by:

$G(D,n)=G_M(D)\cdot\cos((t-12)\cdot\pi/dlt)$ (20) for t=24·(n-1)/N, while for non daylight hours G(D,n)=0. Where: $G_M(D)=\pi\cdot w(D)/(2\cdot dlt)$ and 12 (i.e. the noon time) is assumed as the time of maximum irradiance.

The diurnal variation of the ambient temperature $T_0(D,n)$ it can be assumed as sinusoidal given by:

$T_0(D,n)=T_{00}(D)+\Delta T_0(D)\cdot\cos[(t-15)\cdot\pi/12]$ (21) , where $T_{00}(D)$ is the average temperature in day D

$\Delta T_0(D)$ is the maximum temperature variation in day D, and 15 is assumed as the hour for maximum day temperature.

Assuming that the previous FSCPS (as the example in paragraph 4) is installed in a place with W=2000 KW/m^2, for a typical average day its average day irradiation should be w=2000/365. Let us assume furthermore that $T_{00}=20\ ^0C$, $\Delta T_0=4\ ^0C$ and dlt=13 hours. Using the previous algorithm for N=144 and M=100, we have produced the curves shown in the fig(7), for thermal storages 2.5 cm, 12.5 cm and 22.5 cm of equivalent water, assuming that the ground soil is equivalent to 2.5 cm of water storage. The calculated average Power $P_{FSC,av}$ is approximately the same for the three cases (~31.25 MW) and almost equal to the Maximum Power calculated for the same $\dot{m}$, for the average G and without any thermal storage.

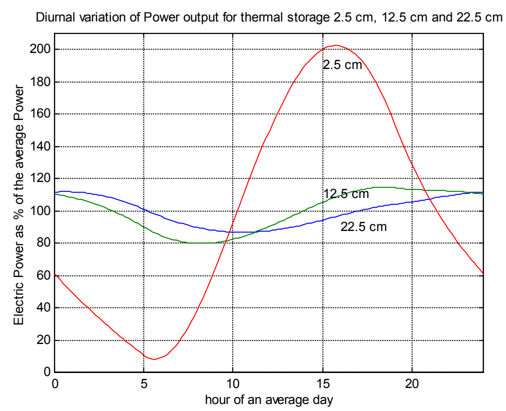

fig 7

On the results we should make the following remarks:

- The $\dot{m}$ was assumed constant (near the value for maximum power under the average constant G) With an appropriate control of Turbines and Generators we can vary slightly $\dot{m}$ during the diurnal cycle in order to increase the overall produced electric power.
- The initial inlet air temperature in the entrance of the solar collector (Tin(n)) actually should be greater than the ambient temperature To(n), due to ground thermal storage effects near the edge of the solar collector i.e. Tin(n)=T0(n)+ΔTin. Thus the daily produced electric energy will be bigger. An average reasonable estimation for ΔTin is ~2 0C. Taking this into consideration PFSC,av is increasing to ~33.46 MW.

- Considering that the annual efficiency is approximately equal to average efficiency, it can be proved that the thermal storage is lowering the FSCPS's efficiency. This is because with thermal storage the FSCPS is working with average $G=w/24$, while without thermal storage the average G should be approximately double. This for example means that the efficiency of the FSCPS with thermal storage is equal to $\eta thst=PFSC,max(G)/(G\cdot Ac)$. This efficiency calculated for this FSCPS is ~2.9 % ,while without thermal storage is equal to $\sim PFSC,max(2G)/(2G\cdot Ac)$. This figure for the same FSCPS is ~ 3.2%. Thus a full thermal storage can decrease the annual Power output up to 10% and should be avoided or used partially for certain time periods only when a 24 hours/day operation with a minimum guaranteed Power Output is necessary .

7. SEGMENTAL VARIATION OF THERMAL STORAGE

Let us now examine the effect of a radially non-uniform solar thermal storage. For example let us assume that in the inner part of $r<D_c/2$ the thermal storage is 72.5 cm (of equivalent water), while in the rest area we have 2.5 cm (of equivalent water) i.e. just the soil thermal storage. This is equivalent to a uniform thermal storage of 20cm of water.

The results are shown in the curves of fig (8). By these curves it is evident that the uniform thermal storage has beneficial characteristics for the same water storage capacity. This means that has a bigger minimum Power output for the same average Power.

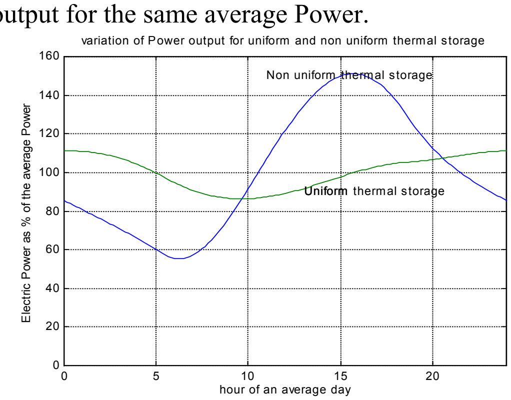

fig. 8

Thus when it is necessary to leave free land in the collector's ground, for agriculture or any other appropriate use, it is better the water thermal storage to be arranged with radial uniformity and peripheral non-uniformity.

For example in case of 36 air turbines and generator units, we can put thermal storage in 36 radial sectors of 2.5 degrees each. The rest 36 radial sectors of 7.5 degrees each can be preserved free for any appropriate use.

8. ANNUAL OPERATION OF FSCPSs OF GUARANTEED POWER OUTPUT

For the FSCPS under consideration let us examine his behavior during the annual cycle. In order to do so the following assumptions are considered:

- $G_{av}=w/24$, $G_{av,\,summer}=1.5\cdot G_{av}$, $G_{av,winter}=0.5\cdot G_{av}$
- $T_{00,av}=20\ ^0C$,$T_{00,summer}=30\ ^0C$, $T_{00,winter}=10\ ^0C$, $\Delta T_0=4\ ^0C$, $\Delta T_{in}=2\ ^0C$
- dlt_{av}= 12 hours , $dlt_{av\ summer}$= 14 hours, $dlt_{av\ winter}$= 10 hours
- Thermal storage homogeneous 20 cm of equivalent water

The produced results are shown in fig.(9).

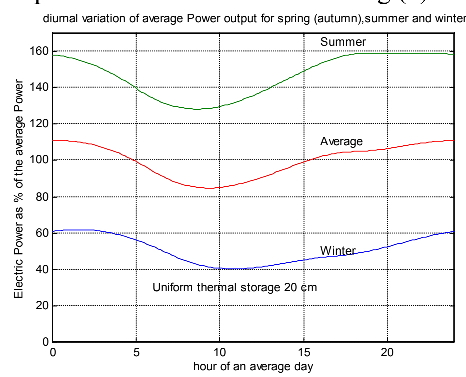

fig. 9

Thus the minimum guaranteed Power of the FSCPS is 40 % of the average Electric Power Output. In summer the FSCPS can produce up to 160 % of the average Power. It is evident that if we decrease appropriately the water thermal storage in all seasons except for winter, keeping always the minimum guaranteed Power the FSCPS can produce more energy annually,.

In fig(10) the Power output is shown for winter thermal storage 20 cm (of water),

for summer 3.5 cm and for the rest seasons 5 cm. In this case in summer the Power could be as high as 265 % of the average. This means that for the FSCPS under examination with a rated Power ~100 MW, and average ~33.5 MW the minimum guaranteed Power is ~14 MW (winter night) while at summer noon can be as high as ~90 MW.

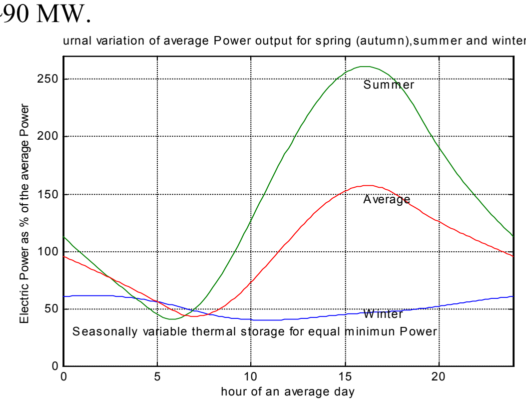

fig. 10

Due to the variable potentialities of the FSCPSs to produce Electric Energy during the year I think that is necessary to have an objective Rating for the FSCPSs. This can be (for example) the maximum Power output of the FSCPS under the average irradiance on horizontal surface in its place of installation,
i.e. $P_{Rate}=P_{max}(G_{av})$. This means that approximately the annually produced electric energy should be
~ $3000 \cdot P_{Rate}$,considering that the insolation in a favorite place of installation should exceed the 3000 hours. This energy can be produced with FSCPS under any reasonable demand supported with the appropriate, for every season, thermal storage.
The FSCPSs, assuming that are installed in places where the annual solar irradiation exceeds 2000 KWh/m^2, can have construction costs in average in the range of 300÷500 €/rated KW.
Their operation and maintenance costs are very small in comparison to any other Electric Power generating system. As a rough estimation their annual maintenance and operation costs should be less than 2% of their construction cost. In order to compare FSCPSs with other conventional power stations, we should take into consideration that all the fossil fuel power stations in average can operate easily above 6000 hour per year supplying their rated Power. Due to thermal storage FSCPSs can also work continuously all year producing above a minimum (~40%) of their average Power. The construction costs, for the same annual Energy production, of the conventional, fuel consuming Power Stations, is higher than the construction cost of their respective FSCPS. The maintenance and operation costs, due to their fuel consumption, are far more expensive in comparison to the FSCPSs respective costs. This means that the KWh produced by FSCPSs will have a smaller production cost.

9. CONCLUSION

An innovative and cost effective method of producing electricity by solar energy was presented. This method is based on floating solar chimneys cooperating with large circular solar collectors and ducted air turbines geared with electric generators. The collector thermal storage analysis was presented based on reasonable heat transfer coefficient assumptions. It was shown that the uniform spread of thermal storage is preferable of concentrating it to the inner area of the solar collector. It was also shown that using thermal storage for FSCPSs can make them Electric Power generating systems that can operate 24 hours per day, 365 days per year, producing always above a minimum of their annual average Power. That is why I strongly believe that the FSCPS will become eventually the major Electromechanical Electric Energy Production System in benefit of the environment and our planet.

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