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Power, Cool and Water Production by Innovative Cycles Fed by Solar Energy

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ABSTRACT

The paper analyses complex arrangements for engines that can be engineered using Car Engine Turbocharger Technology. Such engines can be fed by concentrated solar energy, and are capable of producing Mechanical Power, Cool Power and Water concurrently. Ideal cycle analysis demonstrate a high rate of Solar Energy utilization for both mechanical power and cool power, as well as water production. An ideal Power fraction of thermal power from the sun over 50% can be expected. The maximum overall (mechanical and cool power) utilization factor of solar thermal power entering the engine is expected to be about twice the level of previous fractions.

INTRODUCTION

Nowadays there is a drive to use solar energy, and to adopt distributed power production to reduce grid losses. The requirement is that the engine should have high Reliability, Availability and low Maintenance (RAM). Turbomachine-based engines show a higher RAM level than volumetric ones.

High conversion rates for Thermal Energy entering the Engine from the sun are related to the Maximum Cycle Temperature (T_M). The higher the T_M the higher the Heat Conversion rate. Vice versa, the higher the T_M the lower the efficiency of solar energy capture-concentration and transfer to the engine's Working Fluid (WF). The performance of simple Brayton cycle Micro Gas Turbines (MGTs) appears to be insufficient to compete against Stirling engines and concentrated solar PVs. Turbomachine-based engines derive benefits from their complexity [1-11].

In addition to power in many distributed applications, the production of Cool Power (CP) is required. This two-fold production is usually accomplished by the use of two parallel units: one producing electrical power, the other cool power. The two systems interact, exchanging electrical power (vapour compression systems) or thermal power (absorption systems, etc.). Moreover, drinkable water is sometimes required as additional production. This requires another plant interacting with previous ones. Complexity, volume occupancy and costs thus rise.

NOMENCLATURE

C	Compressor
CA	Carnot
CEG	Compressor Expander Group
CETT	Car Engine Turbocharger Technology
CP	Cool Power
d	Differential
E	Expander
ER	Ericsson
FSS	Free Standing Spool
GICE	Engine Arrangement
MGT	Micro Gas Turbine
PV	Photo Voltaic
PT	Power Turbine
Q	Heat
RAM	Reliability Availability Maintenance
S	Entropy
ST	Stirling
T	Temperature
W	Work
WF	Working Fluid

Special characters

η	Efficiency
σ	Irreversibility index
ξ	Temperature spread factor
τ	Temperature ratio
λ	Ratio
ψ	Utilization factor
χ	Ratio
Δ	Difference

Subscripts

C	Cool effect
H	High
m	Minimum
M	Maximum
T	Temperature

It should be noted that cycles can be described as a closed sequence of thermodynamic processes, and related engines can be arranged accordingly [12-16]. The most attractive ideal cycles for distributed solar energy conversions are the Stirling (ST), Ericsson (ER) and Carnot (CA) cycles, which are shown in Figure 1. All show the highest efficiency ($\eta = 1 - \frac{T_m}{T_M}$) related to T_M and to the minimum Temperature of the cycle (T_m). ST cycle applications are associated with volumetric engines, and often use high pressure Hydrogen or Helium as WF. RAM and vibration problems are the major drawbacks.

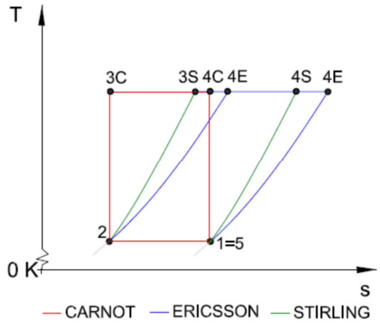

Figure 1: Highest Efficiency Ideal Cycles

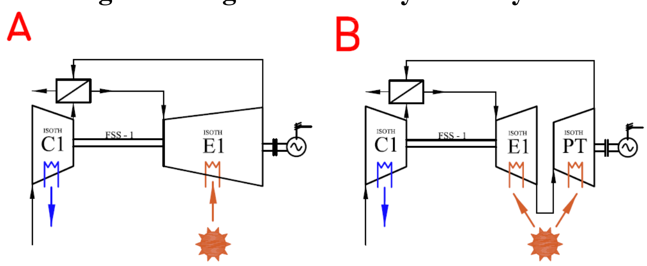

Figure 2: Ericsson Cycle Ideal CETT Based Arrangements

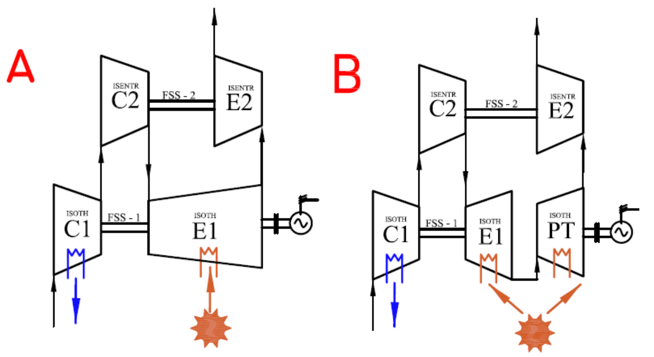

Figure 3: Carnot Cycle Ideal CETT Based Arrangements

ER and CA Cycles can be associated with turbomachinery, as shown in Figures 2 and 3. Solutions with Free Standing Spools (FSS) as Compressor Expander Groups (CEGs) can benefit from Car Engine Turbocharger Technology (CETT). They are built in millions of units, they work under severe conditions and they show very good fluid flow field performance. High availability, reliability and efficiency are expected, together with low engine costs using such a technology.

However, considering the CETT, and each CETT CEG being a FSS running at the equilibrium velocity with no power (work) production, it is possible to establish a series of GICE cycles that are a consequence of engine CEG arrangements. The paper analyses the engine's ideal cycles, and examines the possibility of concurrent Mechanical Power and Cool Power production. If the gaseous working fluid contains a liquid vapour, and if the WF temperature during engine processes decreases to values lower than the dew point temperature, some liquid will be separated. This will constitute a third product. If the working fluid is humid air, condensate water separates from the dry air and becomes a product to be added to Mechanical Power and Cool Power.

After outlining the scientific and technical background, the paper analyses some engine arrangements leading to ideal cycles whose performance constitutes the upper bounds of real machines.

SCIENTIFIC AND TECHNICAL BACKGROUND

Engines to be analysed, from a thermodynamic point of view, need to be associated with cycles. The thermodynamic analysis can be carried out by means of models and can be interpreted according to the theory given in Annex 1.

To perform the analysis, hypothesis have to be made in order to establish an upper bound for performance and useful quantities and a lower bound for consumption. Ideal cycles are taken into consideration. Consequently, the general theory described in Annex 1 is for any kind of cycle can be used. In the paper, the hypothesis that has been stated relates to the absence of irreversibility of any sort.

Therefore, the quantity σ =1 and efficiency depends only on the maximum and minimum temperature ratio $\tau = \frac{T_M}{T_m}$ and on heat source temperature spread factors (ξ^+ and ξ^-).

Maximum efficiency is achieved when heat is isothermally injected ($\xi^+ = 1$) at the T_M and heat is isothermally rejected at T_m. Cycles having such features are the ST, CA and ER cycles. These three cycles are based on a closed sequence of processes. Attempts to establish related engines have been numerous, starting from the concept that an engine can be arranged taking machine-related, heat transfer devices and practical aspects into consideration. Different ideal cycle arrangements can be associated to such engines (e.g. reciprocating Spark Ignition Engines). To select machines for an engine arrangement concept, the CETT concept can be taken as a point of reference for the application of the paper. They are built in millions of units, work under severe conditions and show very good fluid flow performance. High availability, reliability and efficiency are also expected, together with low engine costs. Moreover, the temperature at the turbine inlet for the solar application can be assumed to be between 800°C and 1,000°C according to engine exhaust temperatures at full load.

CA and ER engines show high temperature isothermal expanders. They can be arranged with freestanding groups and Power Turbine (PT) that may offer some advantages in engine turbomachine selection. CA and ER cycles and engines show that the high-temperature heat injection entropy difference is equal to that of low-temperature heat rejection.

Arranging ideal cycles with freestanding CEGs and PT, and bearing in mind that low velocity PT implies a loss reduction in the electrical generation system, the following engines, named as GICE type, and related cycles can be established. Since the FSS does not produce any power because of the running at equilibrium velocity of compressor and expander, the entropy difference of the high temperature expander is lower than that of the compressor.

These ideal cycles GICE ER-# & GICE CA-# are characterized by PT expansion ending at a temperature lower than T_1 thus part of the Q^+ enters at temperatures lower than that of the stream entering the compressor (T_1). High-temperature injected heat Q_{HT}^+ is produced by the Sun Concentrating section, and is a valuable quantity in the same way as work (W) and cool (Q_c). Therefore, according to equation 1.14, the ratio of High Temperature to Heat conversion factor λ assumes relevance and needs to be investigated. Moreover the rate of utilization of Q_{HT}^+ for Mechanical Power and Cool Power production is an important index expressed in equation 1.18.

Two GICE engine concepts have been selected for the study. The first is based on mass flow rates that are the same at various stations. The CA-1 and ER-1 GICE cycles are different because of the devices adopted for the processes that connect the outlet of the compressor to the turbine inlet. The CA-1 uses turbomachines that change the temperatures and pressures of the WF. The ER-1 uses a heat transfer device changing only the temperatures, pressures remaining constant. Figure 4 shows the CETT CEG arrangements and the corresponding cycles. Both the CA-1 and ER-1 cycles have a PT that expands the WF from station 5 to station 6. Depending on the isothermal compressor pressure ratio, T_6 is much lower than T_1 producing the same amount of work and Maximum Cool Power (Q_{CM}).

A part of the latter quantity can be used to chill a cold enclosure for domestic and industrial purposes.

Figure 5 gives the CETT CEG based arrangements and related cycles. WF mass flow rates change along the various stations. Isothermal compressor flow rate is split into two fractions μ and $= 1 - \mu$. The former passes through the high temperature isothermal turbine and drives the isothermal compressor, the latter expands in the PT from point 6 to point 7. As for the previous cycle, Work and Cool Power are produced in the same amounts. Figure 6 shows other variable mass engine arrangements and related cycle concepts. The flow rate at the outlet of the isothermal compressor is split into three flows; the first μ has to keep isothermal CEG running, the second φ_1 expands from 6 to 7 driving an isothermal compressor and increasing the pressure of a φ_2 fraction mass flow rate which, after this compression, expands from 8 to 9, further lowering the temperature T_9 in respect of T_7. This turbine produces work and contributes to production of cool power. Mass conservation establishes that $\mu + \varphi_2 + \varphi_3 - 1 = 0$, μ being defined by the temperature ratio $\tau_1 = {}^{T_M}/_{T_1}$. All these GICE cycles receive positive heat temperature lower than T_1. Q_{LT}^+ corresponds to the maximum cool power produced (Q_{CM}). This Q_{LT}^+ lowers the average temperature of the positive heat sources that inject heat into the cycle lowering ξ^+ and reducing the efficiency. However, since the Q_{HT}^+ is connected to the Primary Energy feeding the engine, λ and ψ quantities (1.4) and (1.8) have to be investigated because they are relevant for assessing engine performance in terms of work and cool power. When the WF entering the engine is humid air and its temperature becomes lower than the dew point temperature, water separates as the temperature of working fluid at the engine's outlet is lower than that of the triple point. More water may be separated using part of the cool power available at the engine outlet.

RESULTS AND DISCUSSION

Ideal cycles of the GICE engine have been modelled and calculations performed to find the upper bounds of outcome quantities and lower bounds of consumption-related quantities.

To establish a frame of reference, the CA and ER engines have been investigated. Figure 7a shows the influence of pressure ratio on specific work and the ratio of adiabatic compressor (expander) work to specific work for the CA, q_r is recovered heat divided by specific work for the ER cycle. Curves are for a constant inlet temperature T_1 of 288 K (15°C). Figure 7b shows the influence of τ for constant isothermal pressure ratio on efficiency, specific work and isothermal compressor work divided by specific work.

Calculations and thermodynamic analysis of GICE engine ideal cycles have been performed to widely investigate state variables and performance quantities. In this paper however performance are discussed for any of the engine ideal cycles. Such quantities are: λ, that gives the fraction of the high-temperature heat that enters the cycle and is converted into work; ψ gives the ratio between work and maximum cool power. Efficiency, Work and Cool power are also given. Figure 8 shows λ, ψ, η, W and T_C. They are practically the same for the engines in fig.4 and fig.5 and do not depend on the CA-# or ER-# arrangement. It is interesting that λ is very close to the ideal CA cycle efficiency and is always higher than the cycle efficiency. The overall $Q^+{}_{HT}$ utilization factor ψ is always higher than 100%, this is due to the fact that CP and W are produced by the same process. GICE 3 (Fig.6) engine ideal cycle performance are given in figure 9, for a maximum temperature of 900°C. The figure on the left is for an isothermal pressure ratio of 3, the abscissa φ_1 being the percentage of the flow extracted from the upper cycle that is expanded to drive the compressor of the split φ_2 flow fraction that produces work (PT) and part of the cool power. It can be observed that a lower cool temperature that that of the previous cycle is reached. The λ quantity is over 50% but always lower than that of the previous cycle. When about half (φ_1) of the split flow is used to compress the second half (φ_2), the maximum factor of $Q^+{}_{HT}$ reaches its maximum being always 125%. ψ and λ quantities versus β_{12} for T_M=900°C and φ_1=20% are given on the right in figure 10. Low pressure ratios improve λ, ψ, η while specific work decreases and cold temperature decreases too.

The use of humid air leads to separated water as the GICE1 engine brings all the mass flow rate entering the compressor to a temperature lower that the dew point and triple point temperatures.

Under these conditions the water separates and can be collected. The blue curve of figure 11 gives the water that can be collected by GICE 1 engines. The red curve is the water that is produced by GICE2 and GICE 3 engines. Water quantity changes with T_M and β_{12}.

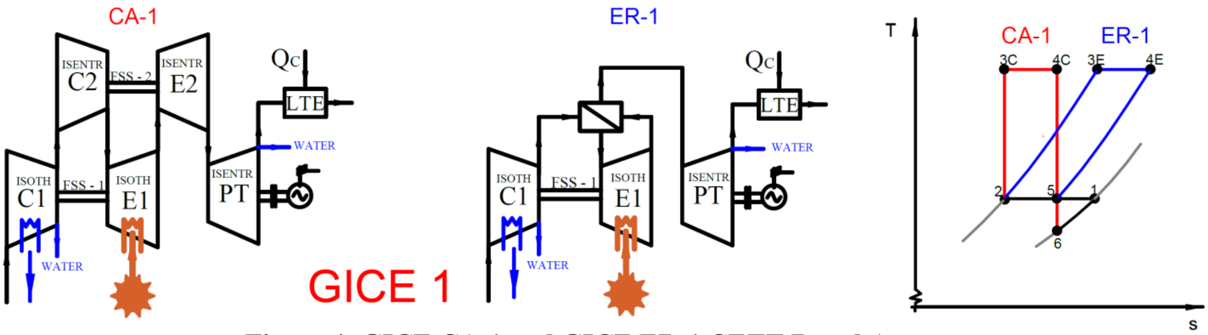

Figure 4: GICE CA-1 and GICE ER-1 CETT Based Arrangements

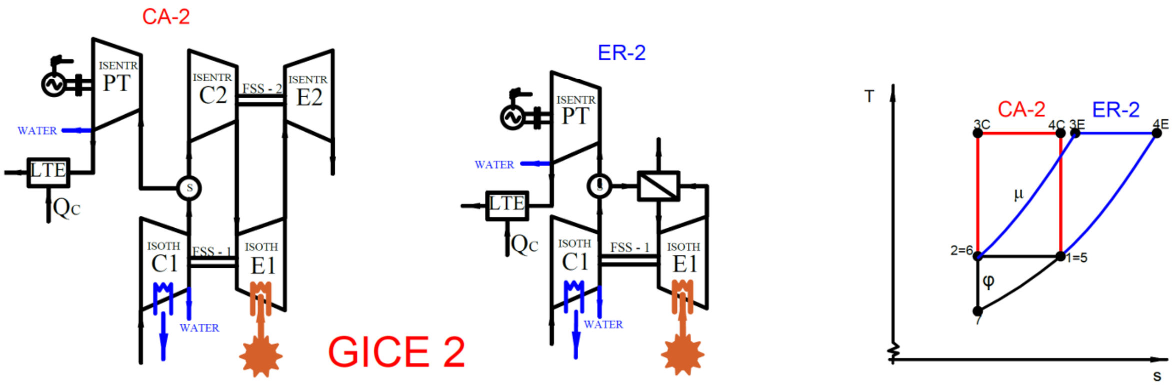

Figure 5: GICE CA-2 and GICE ER-2 CETT Based Arrangements

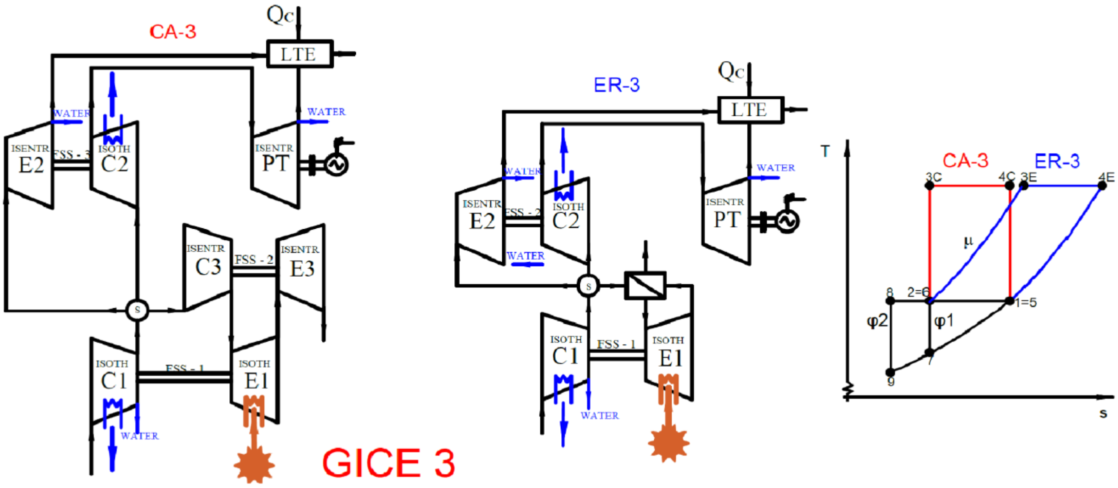

Figure 6: GICE CA-3 and GICE ER-3 CETT Based Arrangements

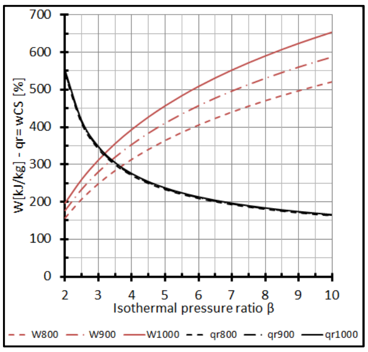

Figure 7-a: Carnot and Ericsson Cycle Performance

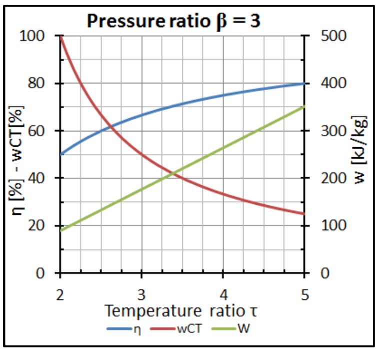

Figure 7-b: Carnot and Ericsson Cycle Performance

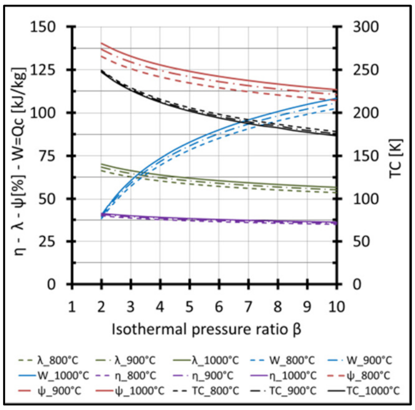

Figure 8: GICE 1 & GICE 2 Cycle Performance

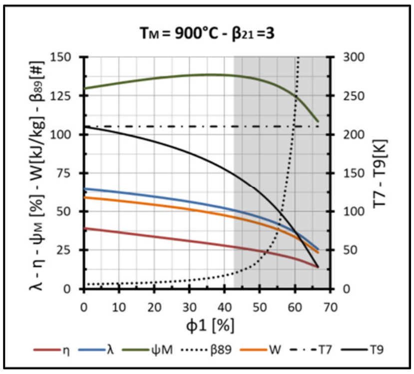

Figure 9: GICE 3 - Cycle Performance

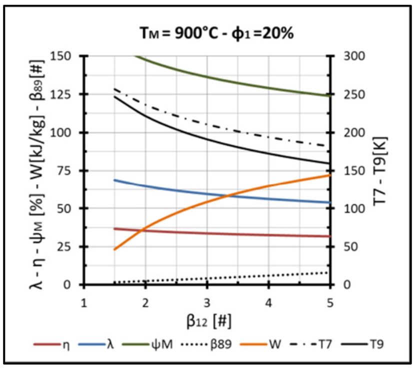

Figure 10: GICE 3 - Cycle Performance

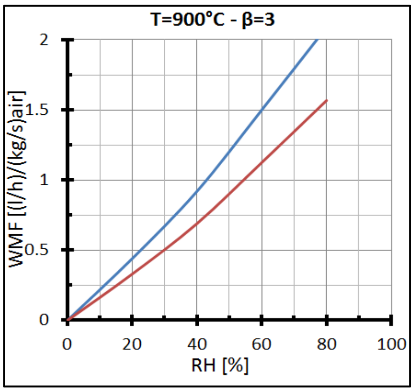

Figure 11: Water production versus relative humidity

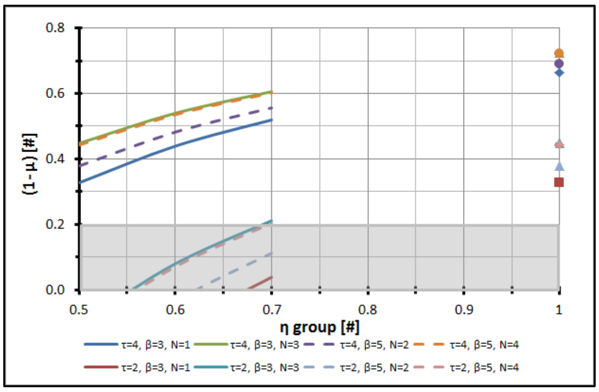

Figure 12: Real machines (1-μ) fraction mass flow Vs η_{group}

CONCLUSIONS

The paper has shown that using CETT CEGs it is possible to arrange various GICE engines that, according to ideal cycle analysis, are capable of high rates of utilization of the high temperature heat entering the engine. Such engines produce mechanical power, cool power and water concurrently when the working fluid is humid air.

The CA-# that uses compression to achieve T_M has the drawback of the very high pressure and temperature at the compressor outlet. Making the compressor wheel and case of the same material as the turbine, the temperature problem is reduced. High pressure is a really serious problem that needs to be investigated further. While the ER-# cycles show a higher degree of feasibility, future works have to report on complete thermodynamic analysis and investigate the reality of turbomachinery and its selection according to engine power.

Looking at figure 12, the reality of the machines influences the μ fraction mass flow because of group efficiency variability ($\eta_c\eta_e\eta_m$). The overall pressure ratio β being set 3 and 5 and varying the phases of compression and expansion N, for τ equal 2 and 4, the cycle performance can be established according to the group efficiency. Not convenient area is filled in grey.

ANNEX 1 - THERMODYNAMIC CYCLE THEORY

Thermodynamic cycle arrangements are made of closed sequences of processes. A thermodynamic cycle to accomplish the task of continuous production of Work (W) has to receive a certain amount of positive Heat

$$Q^+ = \oint T^+ ds^+ = \Delta S^+ . \bar{T}^+ \qquad (1.1)$$

T^+ being the actual cycle temperature of elementary heat and ds^+ the related entropy change, and has to reject a certain amount of negative heat

$$Q^- = \oint T^- ds^- = \Delta S^- . \bar{T}^- \qquad (1.2)$$

T^- and ds^- are the analogous of T^+ and ds^+, ΔS^+ and $\bar{T}^+$ (ΔS^- & $\bar{T}^-$) being the entropy change and the average temperature of positive (negative) heat injection (rejection).

Cycle theory states that the entropy integrated along the cycle

$$\oint ds = \Delta S^+ + \Delta S_i - \Delta S^- = 0 \qquad (1.3)$$

ΔS_i being the production of entropy connected with irreversibility, then negative divided by positive entropy changes as the heat is rejected and injected.

$$\sigma = \frac{\Delta S^-}{\Delta S^+} \geq 1 \qquad (1.4)$$

σ being equal to 1 for ideal cycles.

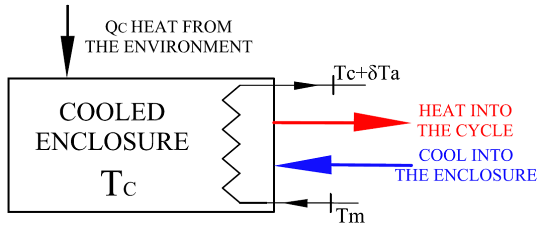

Figure 1.1: Low Temperature Enclosure Schema

Since the heat injected in to a cycle is

$$Q^+ = \Delta S^+ \cdot \bar{T}^+ \quad (1.5)$$

and the heat rejected into the environment is

$$Q^- = \Delta S^- \cdot \bar{T}^- \quad (1.6)$$

work produced by the cycle is

$$W = Q^+ - Q^- = \Delta S^+ \cdot \bar{T}^+ \cdot [1 - \sigma \cdot (\frac{\bar{T}^-}{\bar{T}^+})] \quad (1.7)$$

and the term inside the square brackets represents efficiency

$$\eta = 1 - \frac{\sigma \cdot \bar{T}^-}{\bar{T}^+} \quad (1.8)$$

for a constant entropy production factor, the greater the ratio $\frac{\bar{T}^+}{\bar{T}^-}$ the greater the efficiency. The temperature ratio τ is established

$$\tau = \frac{T_M}{T_m} \quad (1.9)$$

as technological-physical-chemical index, for the dispersion of temperatures of the heat sources: for heat injection

$$\xi^+ = \frac{\bar{T}^+}{T_M} \leq 1 \quad (1.10)$$

and for heat rejection

$$\xi^- = \frac{\bar{T}^-}{T_m} \geq 1 \quad (1.11)$$

the cycle efficiency becomes

$$\eta = 1 - \frac{\sigma}{\tau} \frac{\xi^-}{\xi^+} \quad (1.12)$$

This means that to have a high efficiency the external heat transfer from outside and into the environment must be concentrated as close as possible to the Maximum cycle Temperature (T_M) for positive heat sources, and as close as possible to the minimum cycle Temperature (T_m).

The positive heat entering the cycle can be split into two parts

$$Q^+ = \oint dq^+ = Q_{HT}^+ + Q_{LT}^+ \quad (1.13)$$

Q_{HT}^+ being the heat that enters the cycle when $T^+ > T_1$ and Q_{LT}^+ the heat that enters the cycle when $T^+ \leq T_1$; T_1 being the ambient reference temperature. Q_{LT}^+ spontaneously flows inside the cycle while Q_{HT}^+ has to be produced by a device (combustor, CSP, etc.). It can easily be seen that the ratio

$$\lambda = \frac{W}{Q_{HT}^+} \quad (1.14)$$

assumes relevance while efficiency is established as

$$\eta = \frac{W}{Q_{HT}^+ + Q_{LT}^+} \quad (1.15)$$

Q_{LT}^+ being equal to the maximum Cool Heat (Q_{CM}) that spontaneously flows from the cycle into the environment.

If between the ambient and the cycle there is an enclosure at temperature T_C between $T_m \leq T_C \leq T_1$

$$Q_C = \chi Q_{LT}^+ \quad (1.16)$$

$$\chi = \frac{T_C - \delta T_a - T_m}{T_1 - T_m} \quad (1.17)$$

Since the engine will run because of Q_{HT}^+, it's utilization

$$\psi = \frac{W + \chi Q_{LT}^+}{Q_{HT}^+} \quad (1.18)$$

assumes maximum values when $\chi = 1$

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