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Steam Injected Gas Turbine Integrated with a Self-Production Demineralized Water Thermal Plant[1]

GIOVANNI CERRI Prof., Assoc. Member of the ASME
GIACOMO ARSUFFI Dr., Eng.

Dipartimento di Meccanica e Aeronautica
Università di Roma, ''La Sapienza'', Italy

ABSTRACT

A simple steam injected gas turbine cycle equipped with an exhaust heat recovery section is analyzed. The heat recovery section consists of a waste heat boiler which produces the steam to be injected into the combustion chamber and a self-production demineralized water plant based on a distillation process. This plant supplies the pure water needed in the mixed steam-gas cycle.

Desalination plant requirements are investigated and heat consumption for producing distilled water is given.

Overall steam-gas turbine cycle performance and feasibility of desalting plants are investigated in a firing temperature range from 1000.°C to 1400.°C for various compressor pressure and steam-to-air injection ratios. An example is reported.

NOMENCLATURE

ADt = Approach temperature difference, °C
At = Specific total desalting plant heat transfer surface, m^2
B = Gas turbine pressure ratio
BH = Brine heater
BHA = Brine heater surface, m^2
CA = Condenser surface, m^2
Cb = Brine specific heater, J/kg K
Cp = Specific heat at constant pressure, J/kg K
D = Difference
E = Envelope curve
EA = Economizer surface, m^2
FAR = Fuel air ratio
FC = Fuel cost
L = Curve of limits
LHV = Fuel low heating value, J/kg
m = Mass, kg
O = Maximum specific output work curve
P = Power, W
p = Pressure, Pa
Q = Heat, J
Qd = Specific heat consumption of the distillation plant, J/kg
S = Steam air injection ratio,
SHA = Superheater surface, m^2
SN = Distillation plant stage number
SR = Distillation plant heat transfer area parameter
T,t = Temperature, K, °C
U = Heat transfer coefficient, W/m^2K
VA = Vaporizer surface, m^2
WB = Waste heat recovery boiler
WC = Water cost
WF = Water flow, kg/s
Wo = Plant specific output work, J/kg
ε = Effectiveness
η = Efficiency
η_m = Mechanical efficiency
σ = Ratio between the steam feeding the distillation plant brine heater and the compressed air

Subscripts

a = Air, exahust
av = Available
b = Brine
bb = Brine blowdown
bc = Cold brine
bh = Hot brine
c = Compressor
cc = Combustion chamber
dw = Demineralized water
i = Inlet

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l	=	Lower, minimum
m	=	Mixture
o	=	Outlet
p	=	Politropic
pp	=	Pinch-point
s	=	Steam
ss	=	Saturated steam
sw	=	Sea water
t	=	Turbine
u	=	Upper, maximum
ut	=	Utilized
w	=	Water
1,2,3...	=	Initial and end points of transformations

INTRODUCTION

Injecting steam or water into a gas turbine combustion chamber has often been regarded as a means of increasing gas turbine output - especially during hot weather operations. Efficiency improvement has sometimes resulted when steam is produced by exhaust gas thermal energy [1,2,3,4,5,6]. Sometimes steam injection has been used to reduce gas turbine firing temperature and lengthen the life of the turbine. NOx emission reduction has also been obtained through steam injection [7,8,9].

Usually the steam injection ratio has been maintained below 5-8% due to the compressor surge margin safety of heavy-duty or aero-derived gas-turbines already on the market.

Further output improvement could have been obtained by adopting a steam-to-air injection ratio higher than 10% depending on firing temperature and pressure ratio.

Several analyses have been published during the last decade. In [10] steam injection was compared with two other kinds of plants. Water consumption and cost analyses were emphasized; lower limits of 125°C for the stack temperature and 28°C for the pinch-point temperature difference were considered. A gas turbine firing temperature of 1427°C was accounted for. The conclusions were that the water consumption was about 0.42 - 0.44 kg/MJ (1.15 - 1.60 kg/kWh) - this being less than other plants. Therefore economical convenience exists when the water cost (WC) is lower than 3.9-4.7 $\$/m^3$ (15-18 $/1000 gals) and the fuel cost (FC) is 1.9 $/GJ ($\$2/10^6$ Btu). Economically convenient water cost is also WC = 6.0-7.0 $\$/m^3$ (23-27 $/1000 gals) when FC = 2.8 $/GJ ($\$3/10^6$ Btu).

Lewis and Kinney [11] investigated steam injection for restoring maximal power when the gas turbine was operating on medium-Btu fuels during hot days. They also concluded that there was no substantial change in water consumption for the whole plant (STAG type) because reduction of cooling tower make up requirements tends to offset the water lost by steam injection.

In a previous paper [12] a thermodynamic analysis of steam injected gas turbine cycles was given. Steam injection into the gas turbine combustion chamber improves the fuel consumption (efficiency) if the steam is produced in a waste heat boiler by the exhaust gas. Improvement of efficiency can go up by 27% when the pressure ratio is 10, the firing temperature 1000°C, and the steam injection ratio S = 14%. Specific work, referred to by the compressed air taking part in the combustion, increases by about 50%.

The opinion of the authors is that the problem arising with steam injection is not connected with the water consumption but with the quality of the injected water that has to be demineralized. No suspended solids and few ppm of salts should be in the boiler feed water and hot turbine gas.

As suggested in [12], if the salinity of the raw water is not high it is possible to use special indirect boilers such as the Schmidt-Hartman. If a high salinity water source, like the sea, is available, a water treatment plant to get desalted water is required.

The aim of this paper is to investigate the performance achieved by a steam injected gas turbine equipped with a self-producing demineralized water that uses a fraction of the thermal power in the exhaust. The scheme is shown in Fig. 1.

After a brief description of some desalting thermal plants, a multistage flash distillation plant is considered. Specific heat consumption is derived as a function of stage number and heat exchanger surface extension parameter.

Steam injected gas turbine plants are studied in the firing temperature range 1000°C - 1400°C , pressure ratio range of 3 - 30 and for steam injection ratios up to 0.30. Practicability of desalting plants which supply water for injected steam is also considered.

An example of an aero-derived gas turbine plant typical for marine use is reported.

Finally, comments on costs and plant operation and complexity are also made.

TYPICAL DESALTING PLANT AS REFERENCE POINT

Water desalination treatment (total dissolved solids < 10 ppm) is based on many processes, the most important being:

- distillation by thermal energy
- ion exchange
- freezing
- electrodialysis
- reverse osmosis
- solvent extraction.

The two processes of interest for the present application are: i) distillation by thermal energy; ii) ion exchange.

In the second method cation and anion ion exchange resins are used to remove salts. The exhausted resins are regenerated with acids, ammonia, or lime depending upon the specifics of the process. This kind of plant is well-developped for other industrial uses. Foreign matter can foul ion exchange resins so pretreatment of some saline waters may be needed. Production costs are about proportional to total dissolved solids removed and depend on the load factor. Research carried out on a steam injected gas turbine plant whose power is 100 MW and firing temperature 1000°C showed that if the hours of operation are over a thousand the water cost is lower than 2.8% of the fuel cost. The ratio between the unit mass of fuel and water costs was at least 200; in this case the water source was a river.

Water treatment plants based on distillation processes are more suitable for the present application because they can use the heat in the exhaust gas stream directly. This is particularly important when the primary water source is the sea.

In these kinds of plants a saline solution is vaporized and pure water is obtained by condensation (a final salt content less than 10 ppm can be obtained by these processes).

Water from the sea (or from a river) is pumped into the plant, deareated, treated with chemicals and combined with a stream of recycled brine (if a recycle loop exists). The saline solution is usually heated progressively up to a maximal temperature with chemical treatment dependent on this temperature (acid treatment allows higher temperatures (> 120°C) than polyphosphate or similar). The peak temperature is reached by means of an external heat source; in the present situation the heat is from the gas-turbine exhaust gas. The heat available in the brine and condensed pure water is partially saved in some recovery stages [13,14,15,16].

Due to the pure water separation the brine salinity rises. Then one fraction is drained out of the plant while sea water is added.

A fraction of fresh sea water is usually used as cooling water and then discharged without taking part in the distillation process.

Fundamentally two kinds of distillation processes are used:

a - multiple-effect evaporation (MEE)
b - multistage flash distillation (MSF).

In the MEE plant, schemed in Fig. 2, once the brine has been heated up to the peak temperature it enters into the first evaporator (first effect) where the steam distillates due to external heat. Distilled steam goes into the first brine preheater where it partially condenses and then goes into the second evaporator where it condenses totally causing the distillation of other steam from the brine. The second evaporator pressure is lower than that in the first one, so then the temperature is also lower. The process is repeated for each stage. The final brine preheater condenses all the distilled vapor from the last stage and then a certain amount of brine is discharged as surplus. The brine exiting from the last stage at the minimum pressure and temperature is blown off. A fraction of the heat available in the blowdown brine can be recovered by a freshwater-brine heat exchanger. This can assume a certain importance for a small stage number.

The process can operate at higher blowdown concentrations than the MSF.

Capital investment is higher than for the MSF; it may be quite competitive for very high pure water production (exceeding 1600 m^3/h corresponding to about 500 l/s).

A multistage flash distillation (MSF) plant having three sections is schematically represented in Fig. 3. The 1st section is where the brine is heated up to the maximal temperature by an external source. The 2nd (central) section is for recovery and the 3rd one is for cooling. One stage consists of a chamber where the vapor distillates at the bottom and condenses at the top in a brine preheater. A drop separator divides the upper part from the lower part of the chamber. In the cooling section the distilled steam is condensed by the feedwater which is partially discharged out from the plant. The brine from the last stage can be recycled to lower the heat consumption.

Both these above processes can be schemed from the energetic point of view as a black box where heat from the outside enters together with the feedwater that is at its inlet temperature T_{bci}. The brine blowdown, having the temperature T_{bho}, goes out of the plant as well as the cooling water at its temperature T_{cwo} and the demineralized water at its temperature T_{dwo} (see Fig. 4).

A multistage flash distillation plant is considered here; however, a multiple-effect evaporation plant can be treated in the same way.

The question arising in the present study regards the feasibility of a distillation plant able to give the required amount of pure water by using a fraction of exhaust heat.

To answer this question it is necessary to determine for a certain number of stages, SN, what the maximum and the minimum amount of heat consumption (Qd) for the unit mass of demineralized water is. Moreover, for such a specific heat consumption range a parameter has to give the heat transfer surface extension. This parameter, SR, varies from zero to one with the minimum and maximum surface. The distilled water final temperature is also in relation to the number of the stages and SR. The following relationships can be found:

- specific heat consumption

$$Qd = Qd(SN,SR) \tag{1}$$

- specific total heat transfer area

$$At = At(SN,SR) \tag{2}$$

- distillate temperature

$$T_{dwo} = T(SN,SR) \tag{3}$$

The above are shown in Fig. 5. How these relationships are found is reported in [17].

Data assumed for plots in Fig. 5, are:

T_{bci} = 15°C
T_{bhu} = 120.°C
DT_{db} = 1.75°C
Cb = 4.2 kJ/kg°C
U = 1.63 kW/m^2 °C
ADT_1 = 1.5°C

Fig. 5 plots do not account for the possible heat recovery from the blowdown brine that lowers Qd_1 and can be really important for SN = 1-4. Instead of using the blowdown brine recovery system the brine recycling could be adopted.

Fig. 5 also gives the corresponding condenser effectiveness for $SR = 0$, and for $SR = 1$. The latter value is related to the minimum approach which corresponds to the maximum heat transfer area.

The maximum heat consumption for SN stages is assumed equal to the minimum value for $SN - 1$ stages [17].

Specific heat consumption for desalting water decreases by increasing the stage number as well as the heat transfer surface, therefore increasing initial capital investment.

CYCLE ANALYSIS

Performance of steam injected gas turbines is analyzed with reference to the Fig. 1 plant scheme showing a traditional gas turbine plant (compressor, combustion chamber, and turbine) equipped with a waste heat recovery section that produces the injected steam by using sea water as the primary water source.

The thermodynamic analysis has been carried out taking the mixed gas steam cycle, shown in Fig. 6, and the heat recovery section into consideration.

Air and steam can be considered separately for the mixed gas-steam cycle thermodynamic analysis. The air is compressed up to pressure 2a then is mixed with steam. The air (combustion gas) is then heated up to the firing temperature, and then expands from its upper partial pressure to the lower partial pressure p4a. The environment produces the steam-air (exhaust gas) separation.

The steam is produced at a pressure just higher than p2 (ideally $p2s = p2$) and is injected into the combustion chamber. Then it goes with the air at its partial pressure, into the combustion, expansion, and heat recovery processes. Finally it goes out into the environment where it separates in a dew temperature range that has a lower boundary value approximately close to the feed water temperature T_{dwo}.

Specific work refers to the compressed air unit mass taking part in the combustion, and cycle efficiency can be evaluated according to [12,17] as functions of five quantities:

- the specific work

$$Wo = W(T_3, B, S, SN, SR) \tag{4}$$

- and the efficiency

$$\eta = \eta(T_3, B, S, SN, SR) \tag{5}$$

It is worth pointing out that the above two functions can be calculated with the condition that the efficiency is maximum for each pair (B,S). Once T_3 has been assumed this condition leads to the steam temperature calculation by taking the heat transfer process in the heat recovery section into account.

According to Figs. 7 a & c, the heat recovery section can be arranged in two ways.

In the first scheme (Fig. 7.a) there are the waste heat boiler (WB), that produces the injected steam, and the desalting plant brine heater where the brine is heated directly by the exhaust gas.

In the second scheme (Fig. 7.c) the waste heat boiler produces the steam to be injected as well as the steam condensing in the desalination plant brine heater. The exhaust gas in the heat recovery section must supply the heat needed for the injected steam.

$$Q_s = Q(S, B, T_{dwo}, T_{2s}) \tag{6}$$

and for desalting the right amount of pure water

$$Q_{dw} = Q(S, SN, SR) \tag{7}$$

where the heat required by the desalting plant depends on its specific heat consumption and steam-to-air injection ratio

$$Q_{dw} = Q_d(SN, SR)S \tag{8}$$

The heat recovery section introduces further conditions:

1 - steam and exhaust temperature difference must be higher than a lower limit

$$T_{4m} - T_{2s} \geq DT_{sml} \tag{9}$$

2 - the pinch-point temperature difference has to be higher than a lower limit

$$DT_{pp} \geq DT_{ppl} \tag{10}$$

3 - Another condition regards the steam quality; due to combustion stability [4] it has to be at least dry steam. Due to technological reasons [even if condition (9) is satisfied] maximum steam temperature must not exceed an upper limit:

$$T_{ss2} \leq T_{2s} \leq T_{2su} \tag{11}$$

To avoid low temperature sulphur corrosions stack exhaust temperature can't become lower than a limit connected with the fuel quality.

$$T_5 \geq T_{5l} \tag{12}$$

Typical exhaust gas and water-steam temperature distribution profiles for both heat recovery section schemes are represented in Figs. 7 b & d.

The calculation scheme follows this procedure:

1 - the gas turbine firing temperature, T_3, is assumed as known besides the GT pressure ratio, B, and the injection ratio, S;

2 - the exhaust temperature is then calculated in addition to the turbine work and the maximum value available heat in the exhaust that is:

$$Q_{avm} = (1 + FAR + S)C_{pm}(T_{4m} - T_{5l}) \tag{13}$$

3 - since the feedwater temperature T_{dwo} is not known "a priori" it is assumed as a tentative one. Then the heat required for producing the injected steam is calculated assuming T_{2s} in the range established by conditions (9) and (11). The positive difference of heat

$$DQ = Q_{avm} - Q_s \tag{14}$$

available in the exhaust is compared with the heat needed to desalt the sea water. If this difference is higher than the maximum value of heat for distillating the sea water with an one stage desalting plant:

$$DQ \geq SQ_d(1,0) \tag{15}$$

the desalination plant is assumed to have one stage and the minimum heat transfer surface.

If inequality (15) is not satisfied the minimum number of stages is calculated as well as heat transfer surface parameter that satisfies

$$Qd(SN,SR) = DQ/S \tag{16}$$

Of course conditions (9) , (10) and (11) are verified. Iterations are needed for the variable T_{dwo} to evaluate Qd, the difference DQ, and to get the maximum efficiency for the pair (B,S).

Once the heat recovery section quantities, i.e. injected steam temperature T_{2S}, desalting plant stage number SN, desalting plant heat transfer surface parameter SR, feedwater temperature T_{dwo}, and all the temperature differences (pinch-point, etc.), have been found, the feasibility of the heat recovery section is possible and then the following calculation step is performed;

4 - the compressor requirements are calculated as well as combustion process - then the specific work and efficiency are found. Next a new set of variables (T_3, B, S) can be assumed and calculation is repeated.

A typical plot of the heat balance in the heat recovery section is given in Fig. 8 where for the gas turbine firing remperature $T_3 = 1000°C$ and pressure ratio B=7, the heat required for the injected steam, Q_s, and for the desalting plant, Q_{dw}, and the overall utilized heat

$$Q_{ut} = Q_s + Q_{dw} \tag{17}$$

divided by the available exhaust heat are given versus the steam-to-air injection ratio. On the desalting heat curve (Q_{dw}) the number of flash stages, SN, are given. The injected steam temperature (T_{2s}) is reported on the Q_s curve.

Only a fraction of the available exhaust heat is utilized when steam-to-air injection ratios are lower than $S \simeq 6\%$ (see Fig. 8). In this case 1 or 2 flashing stages are needed and the steam is produced at the highest temperature possible. For steam-to-air injection ratios higher than 6% the stage number of the desalting plant increases while the corresponding heat is reduced, and the steam temperature remains at the maximum. This solution is of course the most convenient from an energy saving point of view (maximum efficiency).

For steam-to-air injection ratios over than about 13%, steam temperature decreases until the saturation temperature is reached.

The pinch-point temperature difference is always higher than 80°C for the scheme in Figs. 7 a & c. The steam injected gas turbine maximum efficiency is reached when the steam air ratio corresponds to all available exhaust heat used as well as to the highest possible steam temperature ($S \simeq 13\%$, in Fig. 8).

DISCUSSION OF RESULTS

The main outcome of the present analysis is calculation of overall cycle efficiency, specific work output (referring to the unit mass of compressed air taking part in the combustion) and the feasibility of a sea water desalting plant using heat from exhaust gas.

A multistage flash distillation plant is considered even if, conceptually, a multiple-effect evaporation plant can be utilized.

Major assumptions which underlie the analysis are:

- compressor inlet temperature and sea water temperature $t_1 = t_{sw} = 15°C$.
- compressor inlet pressure and stack back pressure $p_1 = p_6 = 100$ kPa
- pressure loss in the combustion chamber $Dp_2/p_2 = 3\%$
- back pressure at the gas turbine exit $p_4 = 105$ kPa when there is the exhaust heat recovery section; $p_4 = 101{,}5$ kPa without heat recovery section
- polytropic efficiency of:
  - compressor $\eta_{pc} = 0.89$
  - turbine $\eta_{tc} = 0.90$
- mechanical efficiency of compressor and turbine $\eta_{m,c,t} = 0.98$
- combustion chamber efficiency $\eta_{cc} = 0.96$
- heat exchanger external losses 4%
- fuel low heating value LHV = 42 MJ/kg
- steam upper temperature limit $t_{2su} = 538°C$
- boiler steam-gas temperature difference lower limit $DT_{sal} = 50°C$
- pinch-point temperature difference lower limit $DT_{ppl} = 30°C$
- minimum stack exhaust temperature $t_{5l} = 160°C$
- auxiliary power requirements as well as steam loop losses have not been accounted for because they give negligible effects
- gas turbine firing temperature has been investigated in the range $t_3 = 1000 - 1400°C$
- gas turbine pressure ratio, B, and steam-to-air injection ratio variable in proper ranges according to present application
- desalination plant performance shown in Fig. 5 is considered.

The analysis has been carried out according to what is stated in [12,17].

Results given in Fig. 9 show typical efficiency and specific work curves versus gas turbine pressure ratio - the parameter being steam-to-air injection ratio, S. Figures are for a gas turbine firing temperature equal to 1000°C. For each value of B there is a particular $S\eta$ that makes for maximum efficiency.

Curve E is the envelope line $[E = E(B,S\eta)]$ of curves $\eta = \eta(B)S = \text{const}$ and is the locus of maximum efficiency.

Specific work curves are given in Fig. 9.b where the curve L represents a boundary that is related to low steam temperature, (saturation point) at low pressure ratio. For high pressure ratios (low steam-to-air injection ratios), usually the limit is the pinch-point. This condition is not interesting for the applications. In Fig. 9.b the locus of maximum efficiency, curve E, (which corresponds to the Fig. 9.a envelope curve) is also given together with the maximum specific work curve O. Minimum stage number of the desalting plant is also given in the Fig.

Fig. 10 gives some plant parameters: exhaust, steam, stack exhaust, steam saturation temperatures as well as efficiency and specific work versus steam-to-air injection ratio. Gas turbine firing temperature is $t_3 = 1000°C$ and pressure ratio $B = 7$. Specific work increases with S. Efficiency rises too when S is lower than S_η while steam temperature rises slightly or remains constant and stack exhaust temperature is reduced. Peak efficiency is achieved when $T_5 = T_{5}1$ and steam temperature is maximum.

In Fig. 11 some cycle temperatures are given versus gas turbine pressure ratio.

Water consumption versus pressure ratio is given in Fig. 12, the parameter is steam-to-air injection ratio. Near the maximum efficiency ($B = 12$ and $S = 10\%$ for $t_3 = 1000°C$) water consumption is about 0.3 kg/MJ; that means a 100 MW plant would consume about 111 m^3/h of pure water.

Fig. 8, already discussed, gives an idea of the heat recovered for steam injection and the heat used in the desalination plant besides the desalting stage number.

Steam injected gas turbine performance has been investigated in the firing temperature range 1000 - 1400°C. The envelope curves which represent maximum efficiency $[\eta = \eta(B, S\eta) T_3 = const.]$ are given in Fig. 13 where iso-$S\eta$ curves are also shown.

Overall plant efficiencies between 40% and 50% can be obtained for pressure ratios between 10 and 20, and steam-to-air injection ratios from 8% to 14%.

In the same above firing temperature range, conventional combined cycle of performances [18,19,20] when afterburning is not considered, are compared with injected gas turbine plant performances in Table I.

Data reported in Table I are close to maximum efficiency conditions and are for plants designed "ad hoc". Conventional combined gas-steam plants present better performances (n and Wo) than steam injected gas turbines and their compressors are less expensive. However in steam injected plants only one turbine exists - there are no steam turbine, condenser, and cooling water system.

Figs. 14 a & b give efficiency and specific work versus steam-to-air injection ratio (parameters are firing temperature and pressure ratio). Curves show that for low pressure ratios it is not convenient to raise the firing temperature and the highest steam injection ratio would be about 22% - 25% (small gas turbine plants), while for medium-high pressure ratios the higher the firing temperature the higher the efficiency and the maximum efficiency steam-to-air injection ratio is.

AN EXAMPLE

As an example a gas-turbine plant is considered with main characteristics being:

- firing temperature 1250°C
- compressor pressure ratio B = 18
- compressed air flow ma = 65 kg/s

Without steam injection, mechanical power is P=20.0 MW. Exhaust temperature is $t_a = 532.°C$, and overall plant efficiency $\eta = 34.8\%$.

If steam injection is used, plant performance is given in Table II. The assumption is that the same compressor is used while the turbine would be designed "ad hoc", as well as the combustion chamber secondary flow.

A maximum of three distillation stages is assumed.

The most convenient steam injected gas-turbine plant would be n. 4 with a steam-to-air injection ratio $S = 12\%$; since the enthalpy drop in this turbine is about 7% greater than one without steam, it would have the same stage number as a dry gas turbine, while the blade height would be greater.

Adopting a direct steam production boiler that produces the steam for the injection and the dry steam to feed the desalting plant, the heat transfer boiler surfaces would be:

- economizer EA = 2900. m^2
- vaporized VA = 7700. m^2
- superheater SHA = 1400. m^2

The desalting plant is constituted of a brine heater fed by 3,2 kg/s of dry steam with pressure reduced to 350, kPa. The brine heater surface is:

BHA = 150. m^2

Three flash stages must be adopted without brine recycling. There has to be a brine-to-distilled water ratio of about 7.5. The brine temperature rise is 26°C in each stage, and each condenser is made up of a heat transfer surface

CA = 1200. m^2

CONCLUSIONS

The present analysis has pointed out that a steam injected gas turbine can be equipped with a desalting plant that produces the required pure water from the sea by using the exhaust heat.

Main remarks are the following:

- desalting plants based on distillation methods may be employed integrated into steam injected gas turbine power plants;
- a low number of stages (3 - 5, max 10) are enough for maximum efficiency;
- waste heat boiler pinch-point difference is not really a limit;
- by adopting a proper steam-to-air injection ratio, efficiency can be increased up to 40-50% and specific power output can increase more than 50-80% depending on the firing temperature and gas turbine pressure ratio.

The following points have to be considered by power plant designers when they compare traditional steam gas turbine combined plants and simple steam injected gas turbine plants:

- both plants necessitate a waste heat recovery boiler. The steam injected gas turbine boiler pressure is lower than the combined plant one owing to the use of appropriate pressure ratios for steam injected gas turbines. These pressures should be lower than those of high pressure gas turbines (high firing temperature aero-derived gas turbines). Moreover, by lowering the steam injected gas turbine pressure ratio, high steam-to-air injection ratios can be adopted, therefore increasing the specific work;
- the steam injected plant does not require a condenser and the condensing water loop (possibly equipped with cooling towers);
- the steam turbine does not exist and then the electrical equipment is simplified. In fact,only one electrical generator is needed or alternatively the double coupling at both electric generator shaft ends is reduced to one coupling;
- steam injected gas turbine plants should have a water treatment section that is a more sophisticated plant than the make up system in the traditional combined steam and gas turbine plants;
- steam injection leads to improved pollution characteristics due to NOx emission abatement.

All the above statements make for a low equipment cost for steam injected gas turbine plants. In fact, comparing a steam injected gas turbine plant and a conventional combined gas-steam plant having the same power, the latter should have an initial cost 25-50% higher than the former due to simplified cycle equipment.

For plants having main parameters according to Tab. I, the steam injected gas turbines have fuel costs 5-10% higher than conventional combined gas-steam plants with both plants using the same fuel.

However, high quality fuel is required - especially for the highest firing temperatures. Steam injected gas turbine plants can also use coal derived gas fuel by integrating the plant with a gasification system.

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TABLE I - Comparison between steam injected gas turbine and conventional combined gas-steam plants

	STEAM INJECTED GAS TURBINES			CONVENTIONAL COMBINED GAS-STEAM PLANTS		
Firing temperature °C	1000	1200	1400	1000	1200	1400
Efficiency %	42	46	51	44	50	55
Pressure ratio	15	17	23	6-10	10	14
Steam-to-air ratio %	8	11	14	16-12	16	20
Specific work [kJ/kg]	300	450	630	420-360	550	720

TABLE II - Steam injected gas turbine main parameters

PLANT No.	S %	AFR	η %	P MW	t_4 °C	t_{2s} °C	WF kg/s	SN No.	SR -
1	0	48.3	34.8	20.0	532	-	-	-	-
2	10	41.0	41.6	28.2	546	496	6.5	2	.90
3	11	40.4	42.2	29.0	548	498	7.2	3	.55
4	12	39.2	42.1	29.8	549	427	7.8	3	.80
5	13	37.5	41.4	30.7	550	314	8.5	3	.80

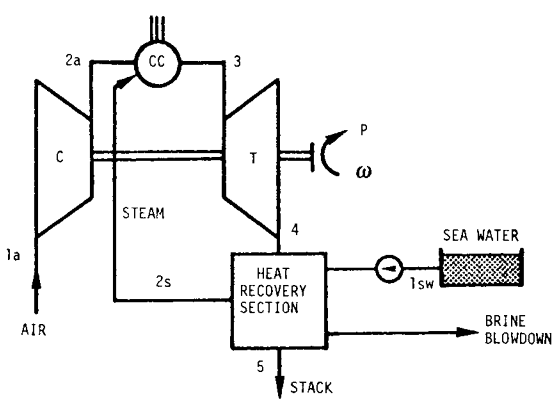


Fig. 1 - Scheme of the gas turbine with steam injection plant equipped with a heat recovery section fed by sea water.

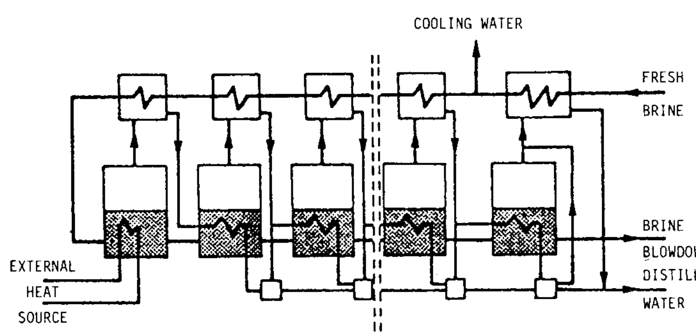


Fig. 2 - Multiple effect evaporation plant scheme

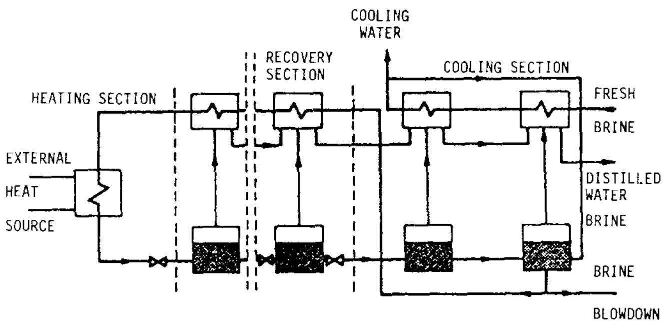


Fig. 3 - Multistage flash distillation plant scheme

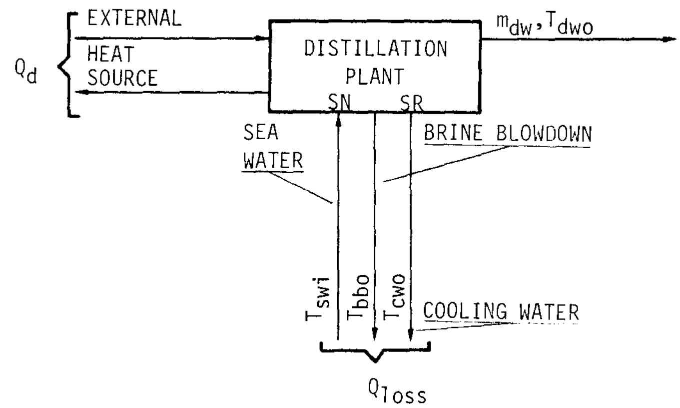


Fig. 4 - Reference distillation plant scheme seen as a black box for energy and mass conservation point of view

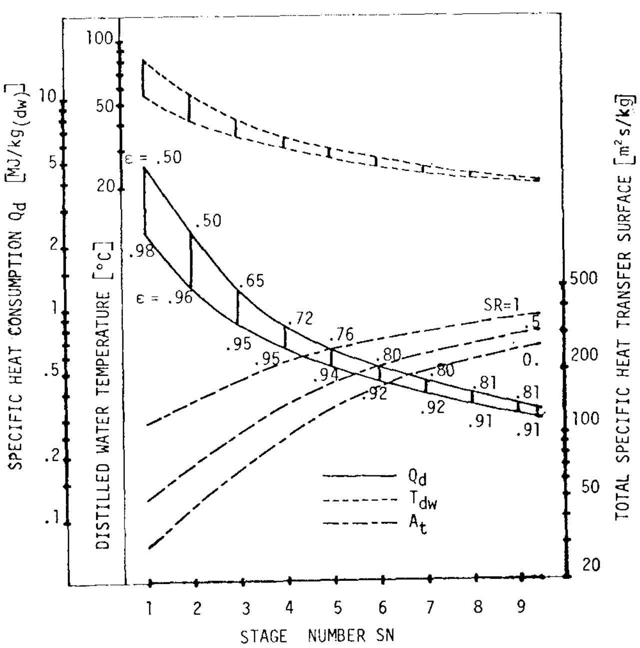


Fig. 5 - MSF plant specific heat consumption, outlet water temperature, and total heat transfer area versus stage number and SR parameter

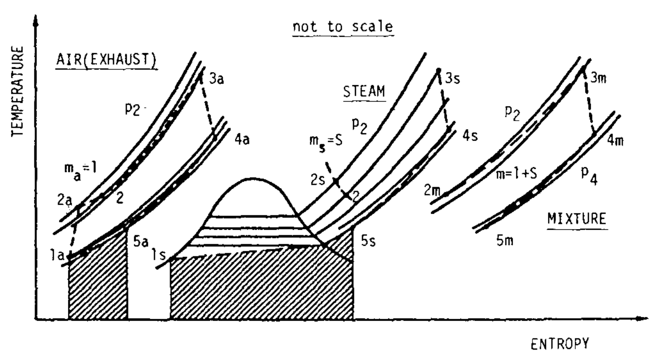
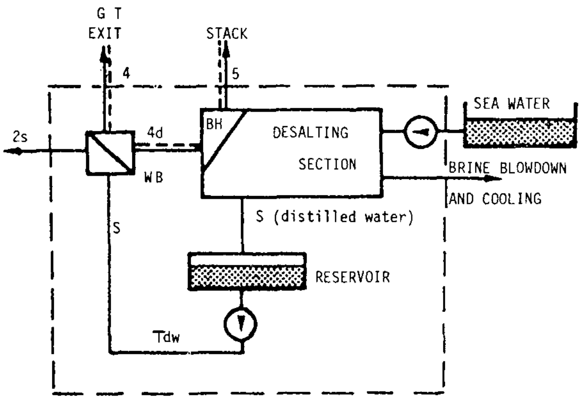
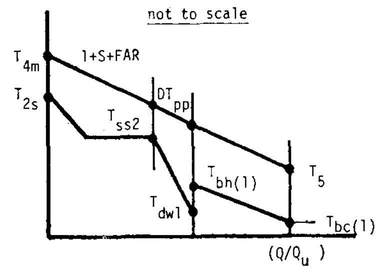


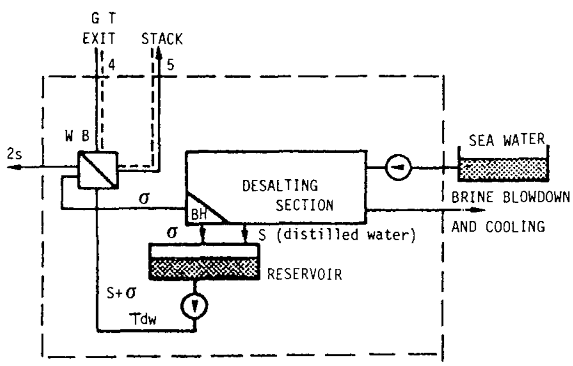
Fig. 6 - Temperature-entropy diagram of steam injected gas turbine cycle



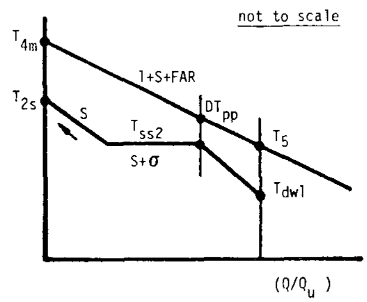
a) Brine heater fed by exhaust gas.



b) Temperature profile for scheme (a).



c) Brine heater fed by dry steam from the boiler.



d) Temperature profile for scheme (c).

Fig. 7 - Two possible schemes for the heat recovery section (HRS) and typical temperature profiles.

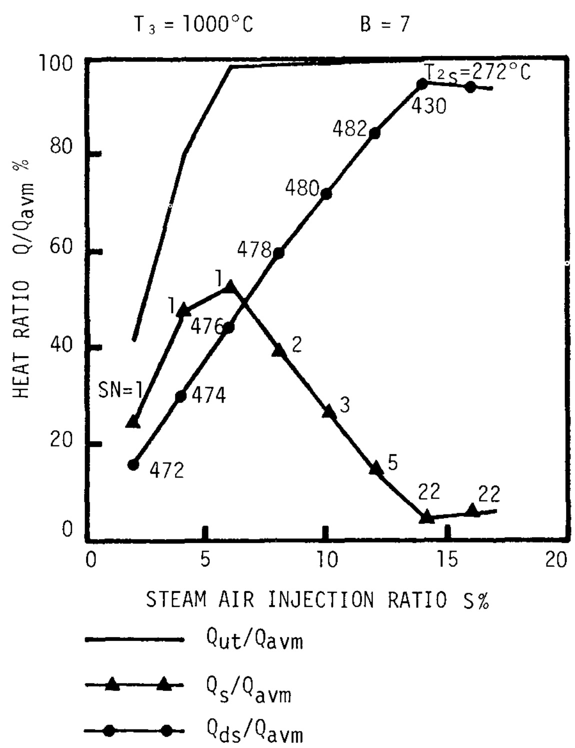
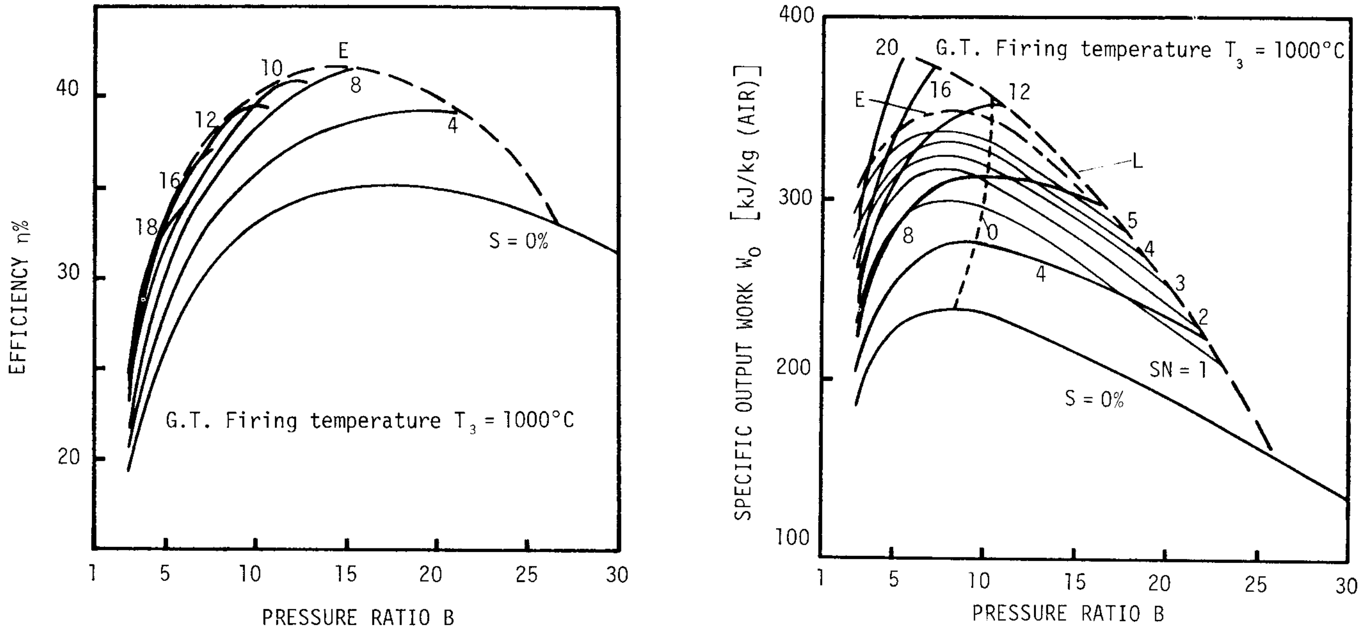


Fig. 8 - Heat balance in the heat recovery section versus steam air injection ratio. Desalting stage number and steam temperature are indicated.



(a) (b)

Fig. 9 - Efficiency (a) and specific work (b) versus pressure ratio. Parameter is steam air injection ratio. E = locus of max efficiency; L = locus of limits; 0 = locus of maximum specific work.

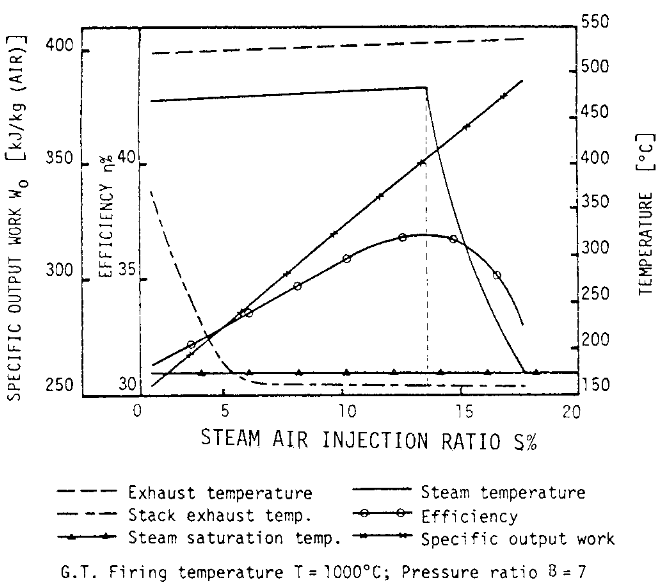


Fig. 10 - Some plant parameters versus steam air injection ratio.

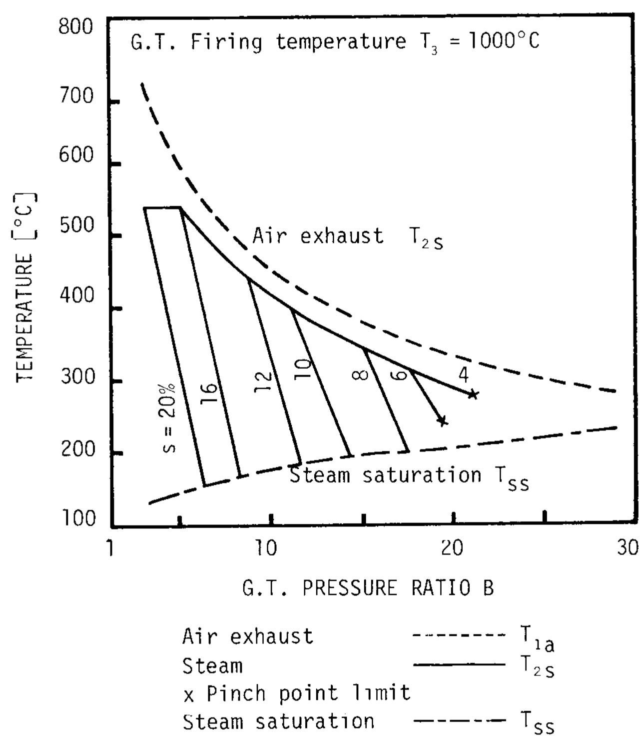


Fig. 11 - Exhaust and steam temperature curves versus G.T. pressure ratio.

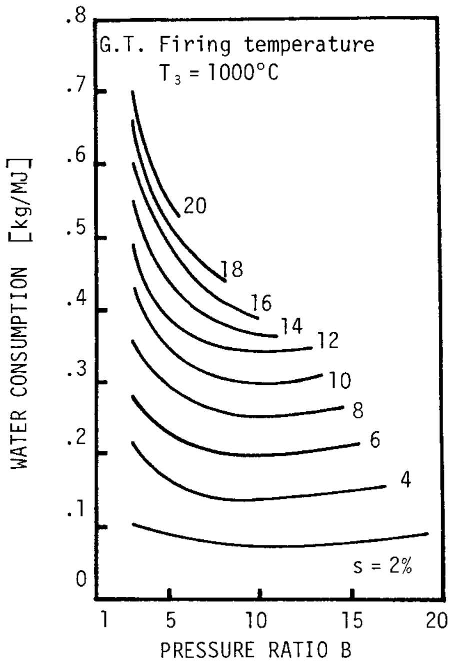


Fig. 12 - Water consumption versus pressure ratio

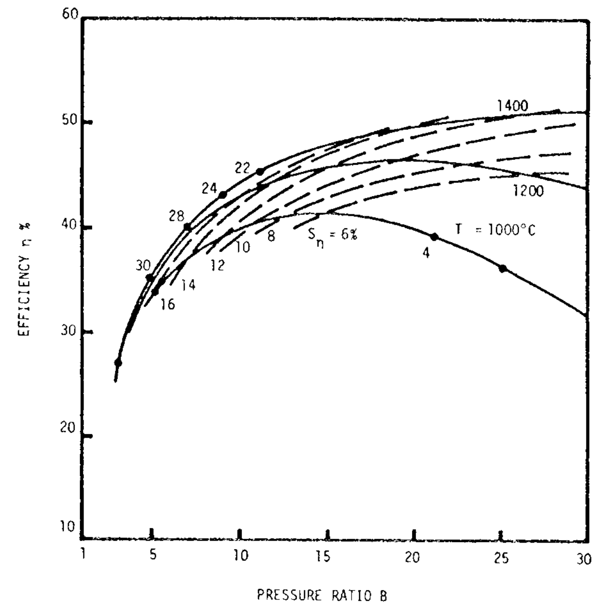


Fig. 13 - Maximum efficiency for T = const. and iso-S_η curves versus pressure ratio.

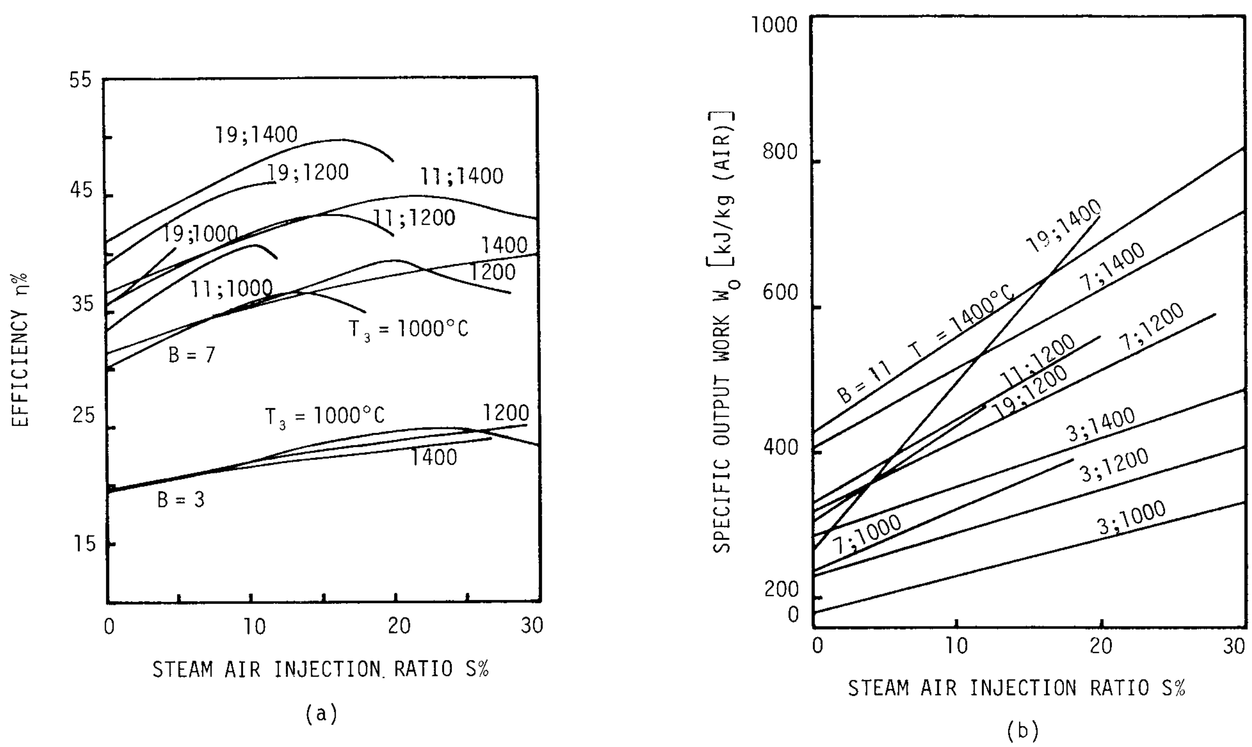


Fig. 14 - Efficiency (a) and specific work (b) versus steam air injection ratio.
Parameter: firing temperature, and pressure ratio.