Efficiency Improvement And Superiority of Steam Injection In Gas Turbines

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Abstract

An open cycle gas turbine with a heat exchanger and its modifications have been studied. These modifications include the combinations of the gas turbine with a steam injection system and the gas turbine with a closed cycle steam turbine. The steam is generated by a waste heat boiler. It was found that in both cases the efficiency and the net output of the gas turbine increased considerably of the order of 20 to 40 %. In order to define the superiority regions of the systems studied in various ranges of power output, an economic analysis per unit power has been done. For short duration, intermittent type of operation the steam injection was found superior. Above this mode of operation. the operational modes of electric base and continuous were covered by the gas turbine combined with the steam turbine.

1. — INTRODUCTION

Some part of the wasted energy of the open cycle gas turbine (*GT*) may be accumulated in the water to be converted into the steam through a waste heat boiler (*WHB*). This steam - injection (*SI*) can provide an increase in the net power output and the efficiency because of the additional enthalpy and mass flow of the steam.

In a recent study on the *SI*, ref. (1), a simple *GT* system was used and the cyclic calculations were based on assumptions, as used in the definition of the thermodynamic data, a constant discharge temperature from the combustion chamber, etc.

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The *SI* in the study mentioned above showed an improvement in the efficiency of the simple *GT* system. But the actual *GT* plants are usually designed from the point of view of metallurgical considerations or overall efficiency. They are not simple *GT* systems, because of having some auxiliary units.

In this study a *GT* plant near Benghazi Libya is selected as an actual model. A heat exchanger (*HE*) is used to reduce the inlet temperature of the turbine and to increase that of the combustor. In fact, the *HE* reduces the inlet and exit temperatures of the *GT* and *WHB*. In the proposed plant, the *SI*, i.e. mixing of the steam and combustion products are described between the combustor and the *HE*. The *WHB* is placed between the turbine and the chimney stack.

2. — THE MODEL GT PLANT AND THE CYCLIC ANALYSIS

The combined cycle including the actual *GT* plant, and the proposed *SI* cycle and its *T—S* diagram, are shown in Figs. 1 and 2. The volumetric flow rate of the fuel, the discharge pressure of the compressor, the inlet and exit temperatures, and the net power output of the *GT* at different operating points are measured. The fuel used is light diesel oil.

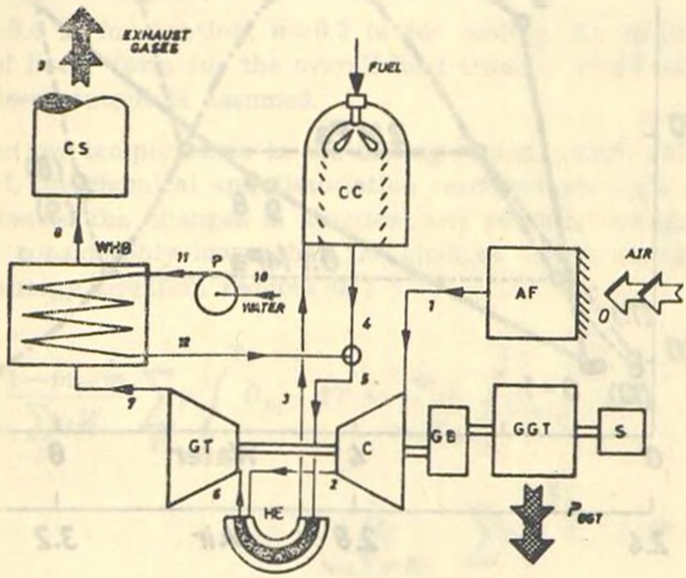

Fig. 1. — Schematic Diagram of the actual gas turbine with a heat exchanger and the proposed steam injection with a waste heat boiler.

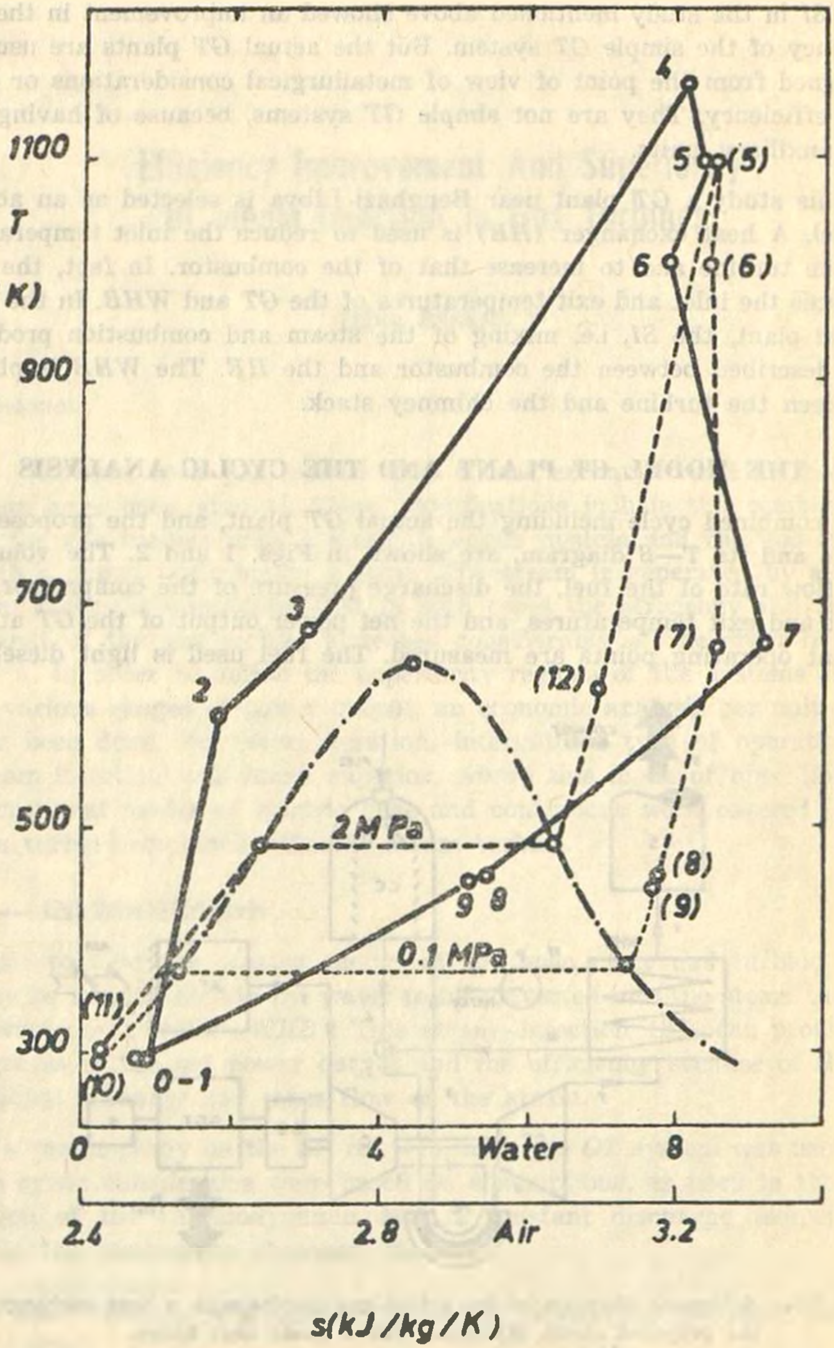

Fig. 2. — Temperature - Entropy diagram of the gas and water components of the combined cycle.

The substances considered in the system are CO_2, H_2O, N_2, O_2 in gaseous phase, H_2O and the fuel in the liquid phase. All the gases except the injected steam are treated as ideal gases. The specific heat of each substance as a function of temperature, and the pressure - loss in each unit are expressed by the empirical equations. The total enthalpy of a mixture is defined in terms of the formation and sensible enthalpies of each component at a given temperature T.

The state - equation of ideal gas, the steady state, steady flow energy equation, and the atomic mass balance of $C/H/O/N$ are used in the calculations. The irreversibilities in the compressor and turbine are defined in term of efficiencies estimated from the experimental measurements. The adiabatic flame temperature in the combustion chamber is calculated considering the components mentioned above, ref. (2).

The heat transfer rate in the toroidal HE is determined by using the overall heat transfer coefficient and the log - mean - temperature difference. Its inner and outer surface heat transfer coefficients are related to the empirical equation for tubes, ref (3).

$$Nu_d = 0.023\ Re_d^{0.8} Pr^n \tag{1}$$

where $n=0.4$ is for heating, $n=0.3$ is for cooling. An empirical relationship of linear form for the overall heat transfer coefficient against the GT power output is assumed.

Due to the low temperatures in the mixing region, points (12 - 4 - 5) in Fig. 1 & 2, the chemical and dissociation reactions are neglected. Since the heat losses, the changes in kinetical and potential energies in this region are considerably lower than the changes in the sensible enthalpies, the energy equation reduces to :

$$\frac{1-\dot{m}_{17}/\dot{m}_5}{\sum\limits_P y_i M_i} \sum_P y_i \int_{T_o}^{T_4} C_{p_i} \cdot dT + \frac{\dot{m}_{12}}{\dot{m}_5 M_{12}} \int_{T_o}^{T_{12}} \overline{C}_{p_{12}} \cdot dT =$$

$$\frac{\dot{m}_4}{\dot{m}_5 \sum\limits_{Pw} y_i M_i} \sum_{Pw} y_i \int_{T_o}^{T_5} \overline{C}_p \cdot dT \tag{2}$$

Since the solution of the equations can not be found because of the unknown temperatures, the computational procedure for the cycle analysis is based on the iterative method for the control volume around each unit. To reduce the number of iterations, closer initial guesses are obtained from the equations simplified by trating the specific heats as constants and using the heat transfer coefficient from the empirical relation, $U=f(P_{GT})$. The assumed and empirical parameters for the *GT* and *SI* are given in Table 1.

Table. 1. — Some Parameters for GT and SI

Inlet	pressure	(*)	(kPa)	101.33
	temperature	(*)	(K)	298
	relative humidity	(*)	(%)	60
Pressure loss in	air filter	(*)	(kPa)	1.2
	heat exchanger		(%)	0.5
	combustion chamber		(%)	1.0
	mixing region		(%)	0.5
	waste heat boiler		(%)	0.5
	chimney stack		(%)	0.5
Bleed flow fraction			(%)	5
Min. temperature difference at PP			(K)	27
At. 34.7 MW	air flow rate	(*)	(kg/s)	228.09
	fuel flow rate	(*)	(kg/s)	3.063
	compression rate	(*)	(–)	9.29
Efficiency of	compressor	(**)	(%)	86
	of turbine	(**)	(%)	87

(*) Measured values.
(**) Defined by iterative method.

3. — CONSTRAINTS IN THE STEAM INJECTION

The technical and economic constraints, based on energy balances and the cost of the injected water, control the energy recovery from the exhaust of waste gases. One important limitation is the value of pinch point (*PP*), which is defined as the minimum temperature difference between the two fluids at any point in the *WHB*, ref. (1, 4). *PP* usually appears where the boiling starts and is given as :

$$T_e - T_{fg} = \Delta T_{PP}, \tag{3}$$

The energy equation for the beginning of the evaporation becomes :

$$\frac{\dot{m}_7}{\sum_{Pw} y_i M_i} \sum_{Pw} y_i \int_{T_8}^{T_e} \overline{C}_{p_i} \cdot dT = \frac{\dot{m}_{12}}{M_{12}} \int_{T_{11}}^{T_{12}} \overline{C}_{p_{12}}\, dT \qquad (4)$$

In comparison with the simple *GT* plant, the *HE* reduces the inlet and exit temperatures of the *WHB*. Therefore, the *PP* problem is a severe limitation on steam generation. The temperature difference on the *PP* may be taken in the range of 17 to 28 *K*, ref. (1, 4). A value of 27 *K* is used in this study. The other technical limitation is the minimum stack gas temperature. A minimum temperature of 423 *K* is assumed in the analysis of the combined cycle.

4. — RESULTS OF THE STEAM INJECTION

The results of the cycle analysis are focused on the net power of the system, the thermal efficiency of the combined cycle and the inlet temperature of the *GT*. The calculated $P_{GT}=f(r_c)$ and $\lambda=f(r_c)$, and the measured P_{GT} of the orginal plant are given in Fig. 3. It is found that the continuous power is 34.69 MW at $r_c=9.29$ and $\lambda=4.56$. The turbine inlet temperature and the thermal efficiencies of the original system are given in Fig. 4 and 5 for $x=0$. The proposed *GTSI* was studied for different water/air ratios, injected steam temperatures and various compression ratios of the *GT* for a fixed steam pressure of 2 *MPa*. In Fig. 4 the curves of $T_6=f(r_c)$ and $\eta=f(r_c)$ are presented for a constant steam temperature of $T_{12}=623$ *K*, where the water/air ratio is used as one parameter. The values of $T_6=f(T_{12})$, $\eta=f(T_{12})$, $T_6=f(P_{GT})$, $\eta=f(P_{GT})$ are given in Fig. 5 and 6, where the parameters are r_c—x and T_{12}—x, respectively. Since the stack gas temperatures in the analysis are higher than 423 *K*, the *PP* limit is selected as the controlling constraint for the maximum water/air ratio in these figures. It is clear from Fig. 4 to 6 that the injection of water around $x=0.08$ may increase the power output of the *GT* from 34.7 *MW* to 41.7 *MW*, the thermal efficiency from 26.6 % to 32.0 % (an increase of 20 %), and reduce the turbine inlet temperature from 1100 *K* to 1038 *K* (a decrease of 6 %).

5. — SUPERIORITY REGIONS OF THE STEAM INJECTION

In order to show the possibility of an increase in the thermal efficiency, upto this point the combination of a *GT* and *SI* has been studied only from a technical point of view. To prove the advantages of the *GTSI*,

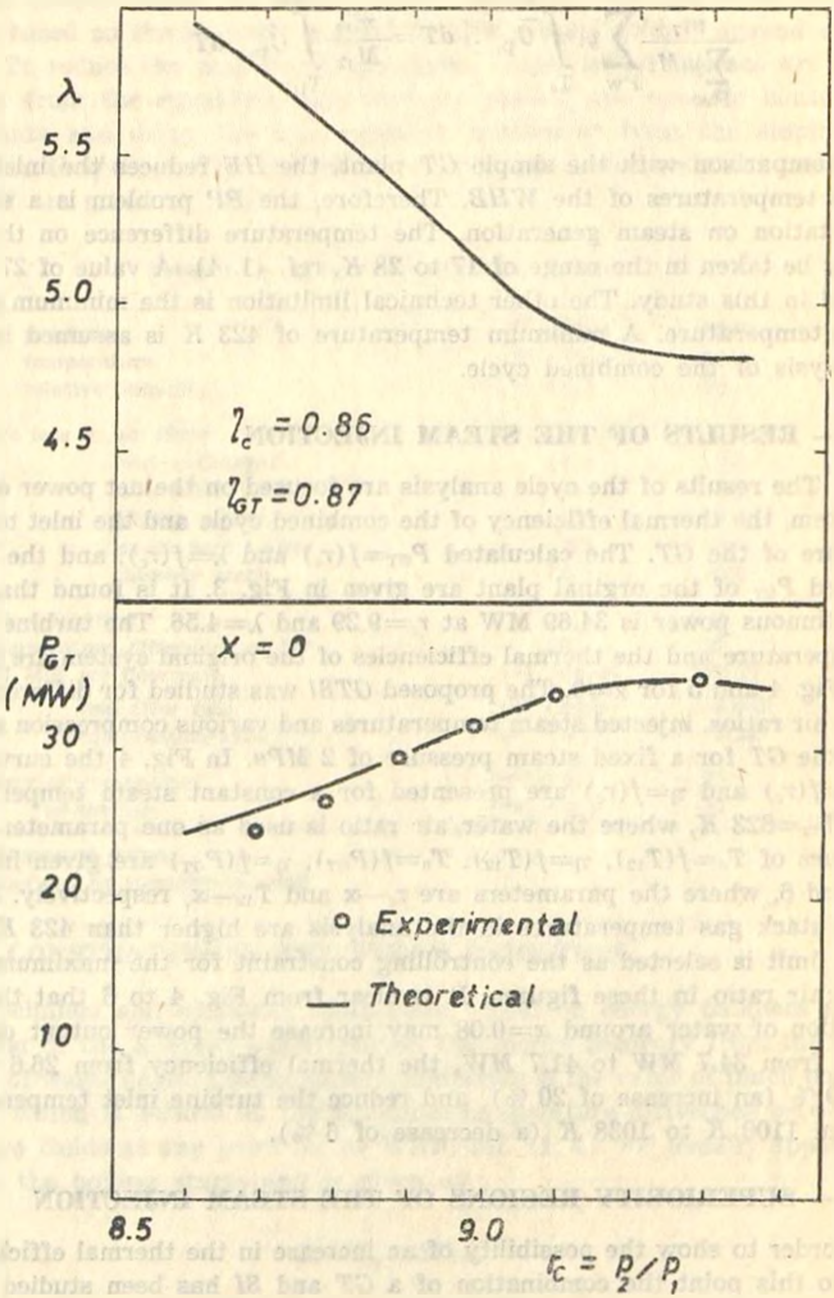

Fig. 3. — Excess air coefficient and output power of the actual gas turbine versus compression ratio.

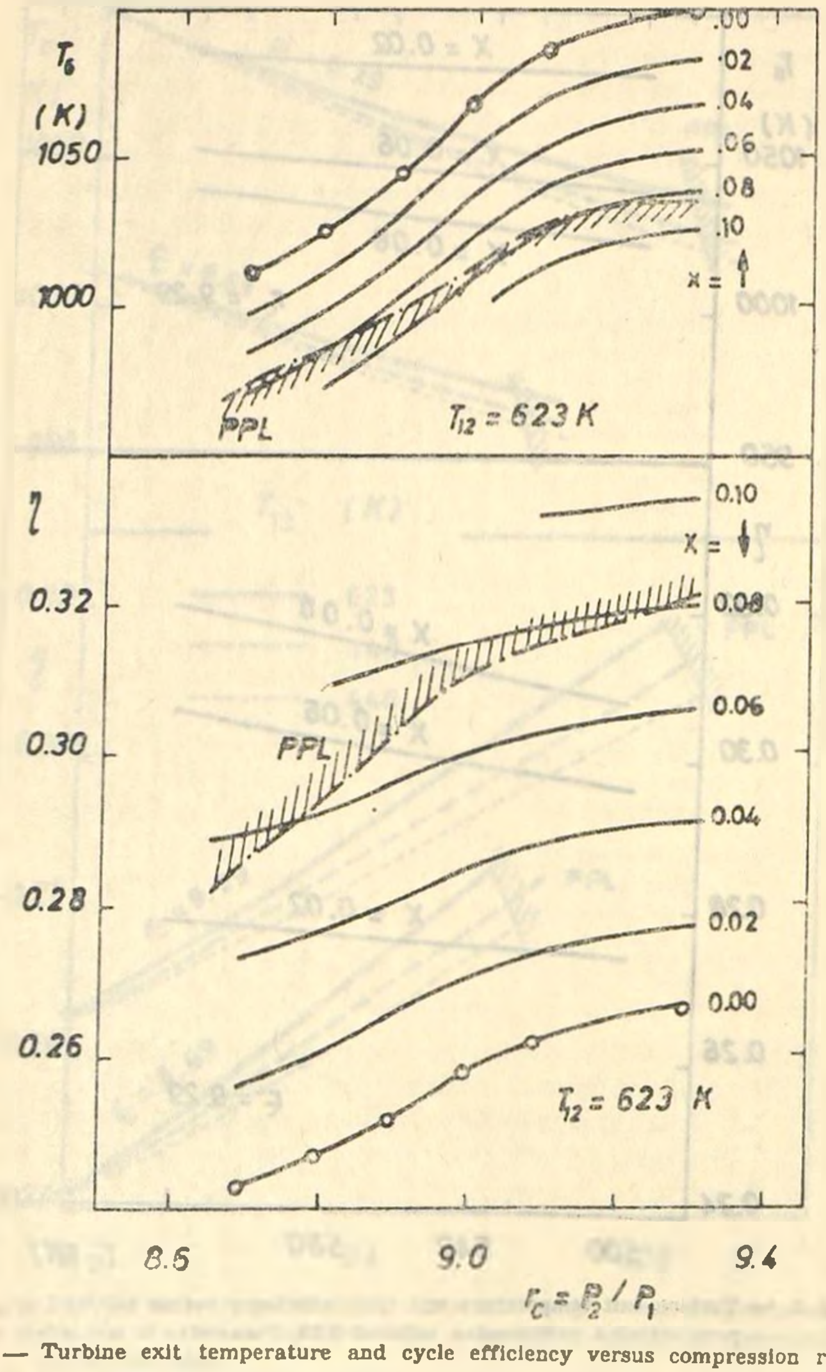

Fig. 4. — Turbine exit temperature and cycle efficiency versus compression ratio at an injected steam temperature of 623 K. Parameter is water/air ratio.

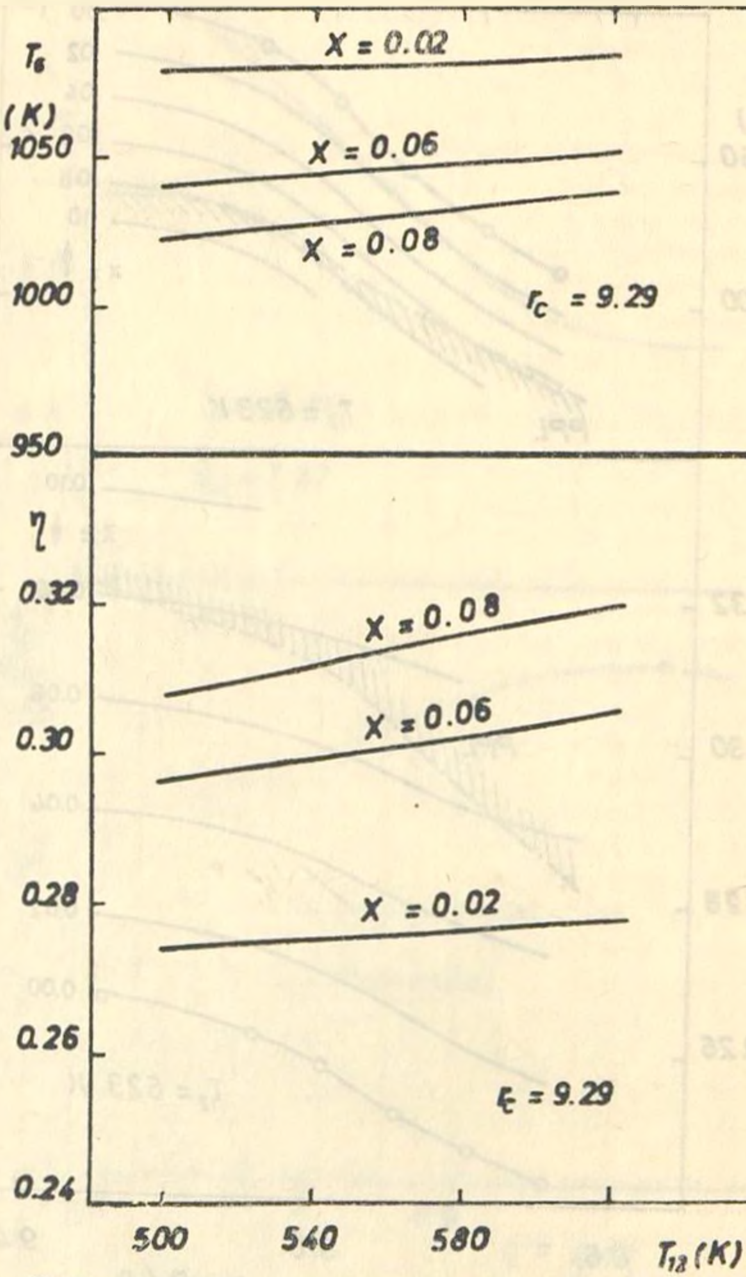

Fig. 5. — Turbine exit temperature and cycle efficiency versus injected steam temperature at a compression ratio of 9.29. Parameter is water/air ratio.

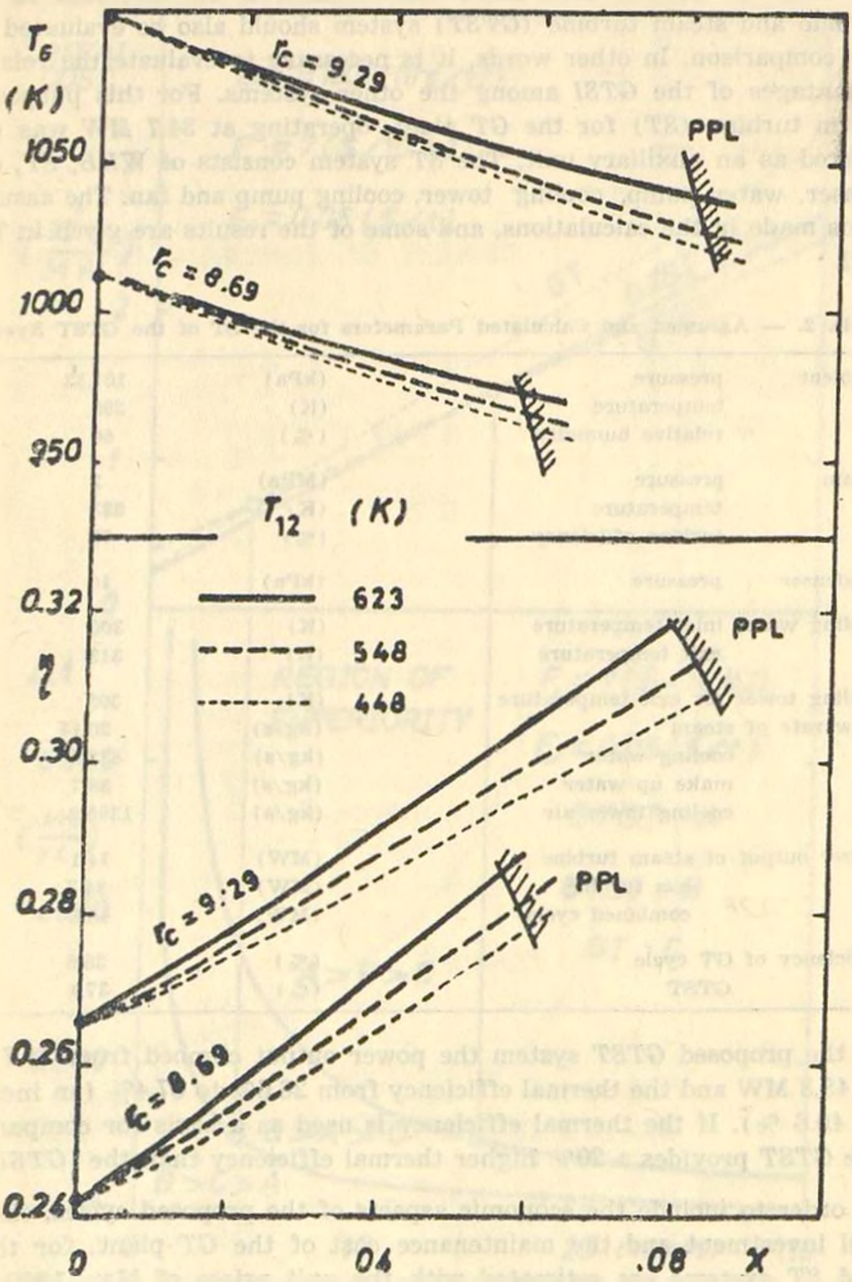

Fig. 6. — Turbine exit temperature and cycle efficiency versus output power of the turbine at an injected steam temperature of 623 K. Parameter is water/air ratio.

not only the *GT* but some other combinations of the *GT*, such as gas turbine and steam turbine (*GTST*) system should also be evaluated for the comparison. In other words, it is necessary to evaluate the relative advantages of the *GTSI* among the other systems. For this purpose a steam turbine (*ST*) for the *GT* plant, operating at 34.7 *MW* was considered as an auxiliary unit. The *ST* system consists of *WHB, ST,* condenser, water pump, cooling tower, cooling pump and fan. The assumptions made in the calculations, and some of the results are given in Table 2.

Table. 2. — Assumed and Calculated Parameters for the ST of the GTST Systems.

Ambient	pressure	(kPa)	101.33
	temperature	(K)	298
	relative humidity	(%)	60
Steam	pressure	(MPa)	2
	temperature	(K)	623
	turbine efficiency	(%)	75
Condenser	pressure	(kPa)	10
Cooling water	inlet temperature	(K)	300
	exit temperature	(K)	313
Cooling tower air exit temperature		(K)	303
Flow rate of	steam	(kg/s)	20.14
	cooling water	(kg/s)	832.5
	make up water	(kg/s)	38.7
	cooling tower air	(kg/s)	1395.2
Power output of	steam turbine	(MW)	14.1
	gas turbine	(MW)	34.7
	combined cycle	(MW)	48.8
Efficiency of	GT cycle	(%)	26.6
	GTST	(%)	37.4

In the proposed *GTST* system the power output climbed from 34.7 *MW* to 48,8 MW and the thermal efficiency from 26.6% to 37.4% (an increase of 40.6 %). If the thermal efficiency is used as a basis for comparison, the *GTST* provides a 20% higher thermal efficiency than the *GTSI*.

In order to include the economic aspects of the proposed cycles, the initial investment and the maintenance cost of the *GT* plant, for the *SI* and *ST* systems are estimated with the unit prices of Nov. 1980. The prices of the fuel F_f and water F_w, the annual operating hours Δt, the

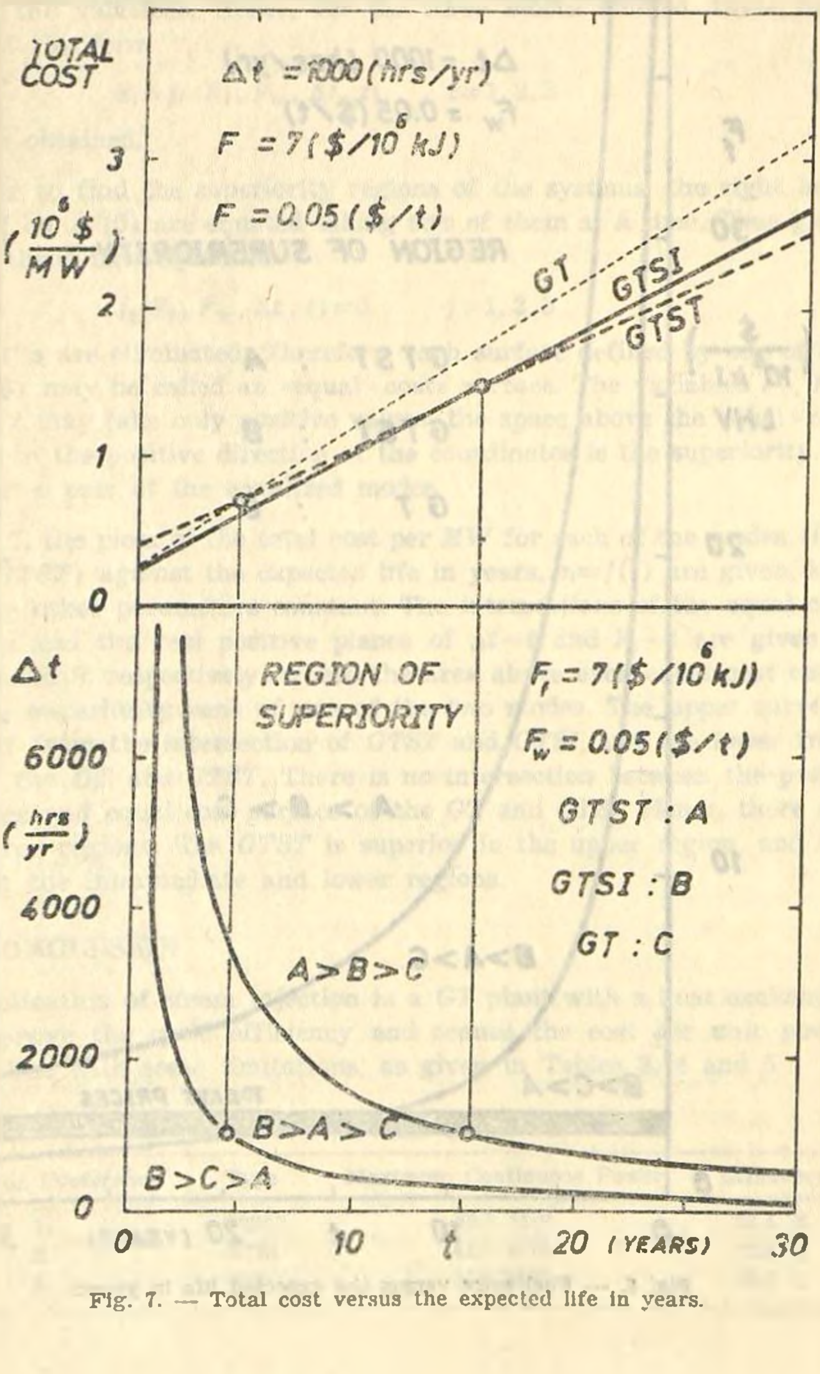

Fig. 7. — Total cost versus the expected life in years.

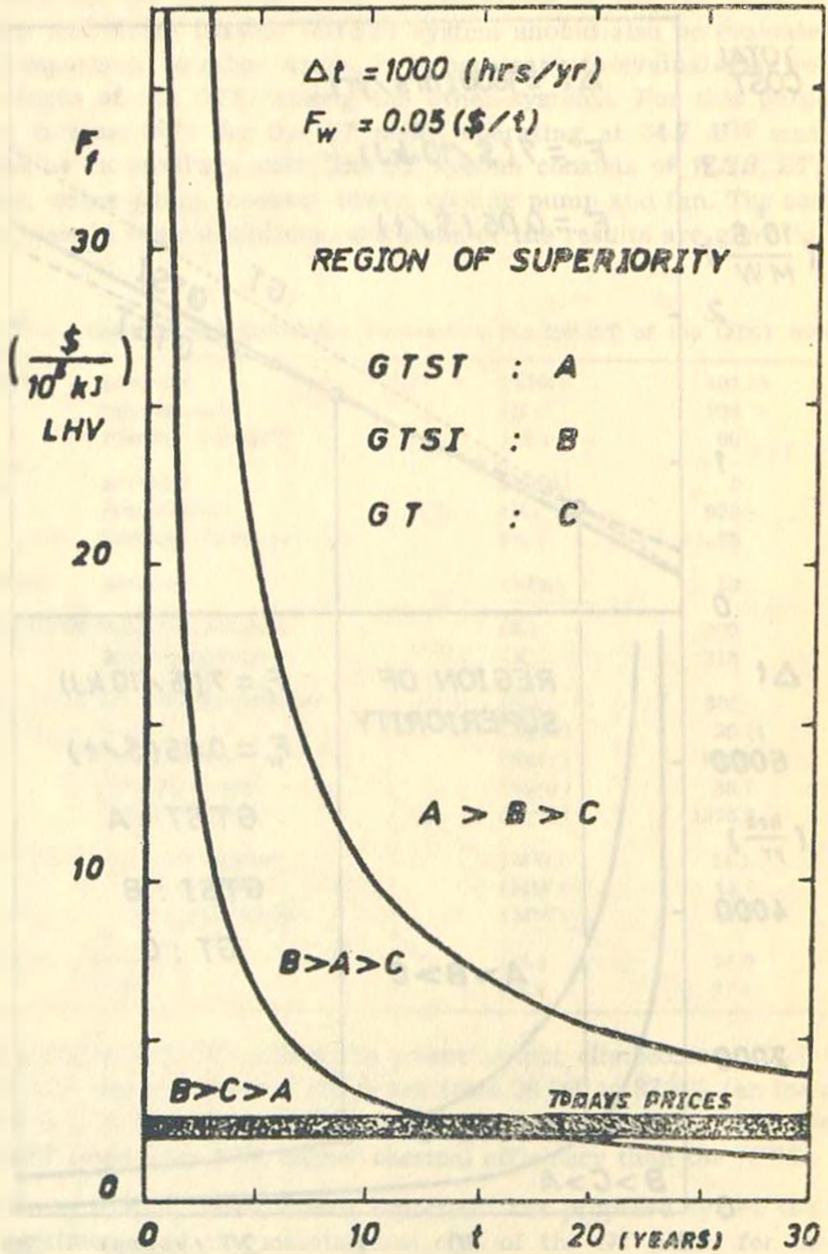

Fig. 8. — Fuel price versus the expected life in years.

expected life in years t, and the total cost per unit power z, are selected as the variables. Hence, for the three modes studied, three functions of the form

$$z_i = f_i\,(F_f\,,\,F_w\,,\,\Delta t\,,\,t) \qquad i = 1, 2, 3 \tag{5}$$

may be obtained.

In order to find the superiority regions of the systems, the right hand sides of eq. s (5) are equated taking two of them at a time. Thus generating three new equations

$$f_j(F_f\,,\,F_w\,,\,\Delta t\,,\,t) = 0 \qquad j = 1, 2, 3 \tag{6}$$

where z' s are eliminated. Therefore, each surface defined by one of the eq. s (6) may be called an «equal - cost» surface. The variables F_f, F_w, Δt and t may take only positive values. the space above the equal - cost surface in the positive direction of the coordinates is the superiority region for a pair of the equalized modes.

In Fig. 7, the plots of the total cost per MW for each of the modes (GT, $GTSI$, $GTST$) against the expected life in years, $z_i = f(t)$ are given, keeping the other parameters constant. The intersections of the equal cost surfaces and the real positive planes of $\Delta t - t$ and $F_f - t$ are given in Figs. 7 and 8, respectively; where the area above each equal cost curve shows a superiority zone of one of the two modes. The upper curve is obtained from the intersection of *GTST* and *GTSI*, and the lower from that of the *GT* and *GTST*. There is no intersection between the positive planes and equal cost surface of the *GT* and *GTSI*. Hence, there are only three regions. The *GTST* is superior in the upper region, and the *GTSI* in the intermediate and lower regions.

6. — CONCLUSION

The application of steam injection in a *GT* plant with a heat exchanger can improve the cycle efficiency and reduce the cost per unit power output, but with some limitations, as given in Tables 3, 4 and 5.

Table. 3. — Thermal Comparison

Order of Preference	Type	Maximum Continuous Power	Efficiency
1.	GTST	48.8 MW	37.4 %
2.	GTSI	41.7 MW	32.0 %
3.	GT	34.7 MW	26.6 %

Table 4. — Economic Comparison

Order of Preference	Continuous ($<$8700 hrs/yr)	Electric base ($<$4000 hrs/yr)	Peak ($<$1000 hrs/yr)
1.	GTST	GTST	GTSI
2.	GTSI	GTSI	GT*
3.	GT	GT	GTST*

* This column is for the fuel price up to 2.35 \$/10⁶ kJ LHV. Above this price the order is: 1. GTSI, 2. GTST, 3. GT.

Table. 5. — Fuel Price Effect on the Peak Operation (expected life 15 yrs)

Order of Preference	Fuel Price, F_f (\$/10^6 kJ LHV) $F_f < 2.35$	$2.35 < F_f < 7.17$	$7.17 < F_f$
1.	GTSI	GTSI	GTST
2.	GT	GTST	GTSI
3.	GTST	GT	GT

If the *GTSI* is compared with the *GT* the following conclusions may be drawn: The thermal efficiency and power increase are more in the *GTSI* than in the *GT*. An increase of 20 % is obtained at a water/air ratio of 0.08 with a steam of 2 *MPa* and 623 *K*. The inlet and exit temperatures of the *GT* and the *WHB* are lower in the *GTSI* than in the *GT*. The complexity, maintenance and investment are higher in *GTSI*.

If the *GTST* is compared with the *GTSI* and the *GT* the following conclusions may be drawn: The thermal efficiency and power increase are more in the *GTST* than in the *GTSI*. An increase of 40.6 % is obtained at a water air ratio of 0.089 with a steam of 2 *MPa* and 623 *K*. The inlet and exit temperatures of the gas turbine and *WHB* are higher than the *GTSI,* but same as the *GT*. The complexity, maintenance and investment are higher in the *GTST* than in the *GTSI*.

Economic considerations indicate that a *GTST* plant may be preferred if the operational mode is continuous or electric base which are above 500 (hrs/yr) to 1000 (hrs/yr).

If the operation is limited up to 1000 (hrs/yr) the *GTSI* may be used for short duration, intermittent type of operation. In this, so called peak operation, the *GTSI* superiority continues up to a fuel price of 7.17 (\$/10^6 *kJ LHV*).

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NOMENCLATURE

C = Specific heat
F = Price
$\dot{m}$ = mass flow rate
M = molar mass
Nu = Nusselt no.
P = power
Pr = Prandtl no.
r = pressure ratio
Re = Reynolds no.
t = expected life in years
Δt = annual operating hours
T = temperature
U = overall heat transfer coefficient
y = mole fraction
z = total cost per MW
x = water/air ratio (mass)
λ = excess air coefficient
= 1/(equivalence ratio)
η = thermal efficiency

Superscript :

n = values corresponding to heating or cooling
$\overline{(\ )}$ = molar value

Subscript. :

c = compression
e = exhaust gas
f = fuel
fg = value corresponding to evaporation
GT = gas turbine
i, j = no. of component or equation
o = reference value
p = constant pressure
PP = pinch point
w = water
1, 2, ... = values at characteristic point at Fig. 1 & 2.

Abbreviations :

GT = gas turbine
GTSI = gas turbine+steam injection
GTST = gas turbine+steam turbine
HE = heat exchanger
LHV = lower heating value
P = combustion products
Pw = combustion products & steam mixture
SI = steam injection
ST = steam turbine