TECHNICAL AND ECONOMICAL ANALYSIS OF COMBINED CYCLES OF GAS TURBINE – STEAM TURBINE ON PIPELINE COMPRESSOR STATIONS

Part I – Technical analysis

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Abstract

In this part there have been shown issues of combined cycle which are on Gas Compressor Stations, reclaiming steam blocks. In analysis of this issue there have been taken few variants of steam cycle configuration and few variants of part load of gas turbine. To steam cycles there have been taken all basis parameters, and then there have been made thermodynamic calculations, which result was value of possibility to make electric energy through steam turbine. In summarizing there have been made an opinion about profitability investment and compared economic results for separate variants.

Key words: *Pipeline Compressor Stations, Combined Cycle, Thermodynamic and Economic calculations*

1. Introduction

Transport of natural gas on big scale is mainly taken with help of gas pipeline systems. Length of those gas pipelines are measure in hundreds and even in thousands kilometers. In order to forward gas on those distances will be possible, there have to deal with problem of hydraulic resistance of gas pipeline. To deal with this on the way of gas pipeline there are built Gas Compressor Stations. The exhaust gases from gas turbine have considerable amounts of heat, which can be used to production of electric energy or heat. Energy which are in gases is a spin-off and its produce does not need to incur additional costs. Installation of recover block will allow on changing heat of gases into electric energy or heat which can be sold. It only made a question of financial investments which are connected with rebuild recover block and its exploitation if there are not exceeded of income from sale of energy production. This study is analysis of this investment and attempt of finding the answer on this above question. In first part there have been shown technical analysis of rebuild recover block on gas turbine which are working on Gas Compressor Stations. In second part there have been shown economic calculation.

2. Installation of recover steam block on Gas Compressor Stations

Installation of recover steam block is aim at using energy from exhaust gases of gas turbine. Because a lot of Gas Compressor Stations are in considerably distance from bigger cities or production factories it is no way to deliver heat energy which will be probably made. So it have been accepted that steam block will make electric energy.

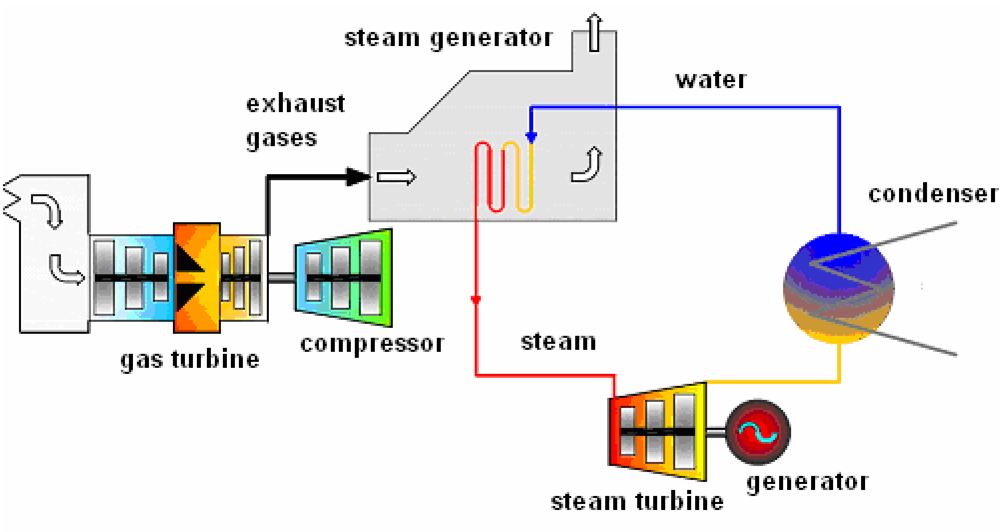

Fig..1. Visual diagram of combined cycle of gas turbine- steam turbine on Pipeline Compressor Stations

The analysis have been made on example of gas turbine ABB GT10B (Siemens SGT600), which is wide used on Gas Compressor Stations. It is double shaft construction with catalogue parameters, power output 25 MW and efficiency 34 %.

There have been analysed three types of steam system's configuration (variant A, B, C, Fig. 2÷4). All of three configurations have been projected and counted in two cases of gas turbine's overloads. The first case is in situation when gas turbine works all the time with full power. The second case consider turbine's work with part overload (from 55 % to 100 %).

2.1 Configuration of steam cycle

A) Single-Pressure System

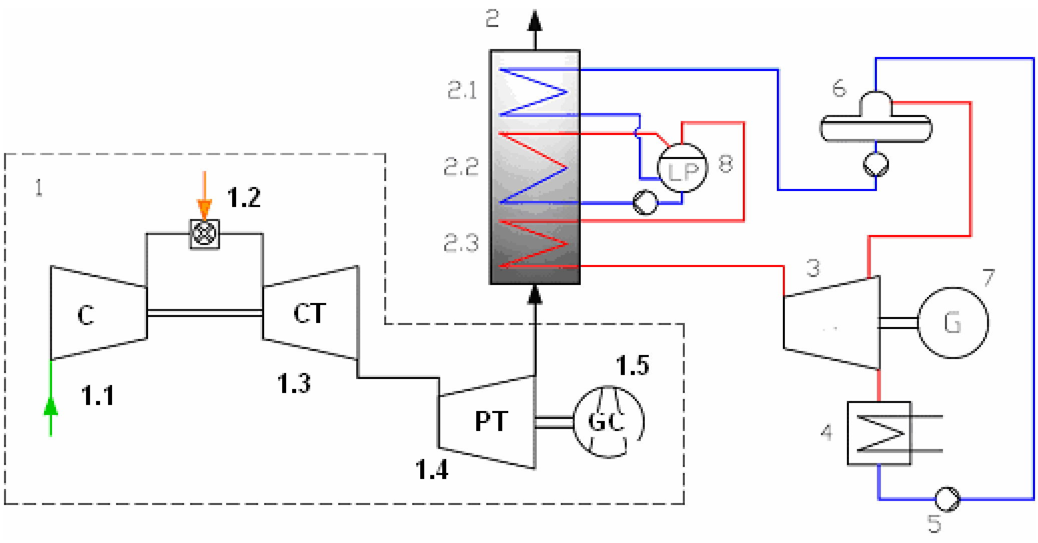

Fig. 2. Variant A of combined cycle plant
1 – Turbocompressor plant (1.1 – Compressor, 1.2 – Combustion Chamber, 1.3 –Compressor Turbine, 1.4 – Power Turbine, 1.5 – Gas Compressor), 2 – Heat Recovery Boiler, 2.1 –Economizer, 2.2 – Evaporator, 2.3 –Superheater, 3 – Steam Turbine, 4 – Condenser, 5 –Condensate Pump , 6 – Dearator, 7 – Generator, 8 – Boiler Drum

B) Single-Pressure System with a Low Pressure Evaporator as a Preheating Loop

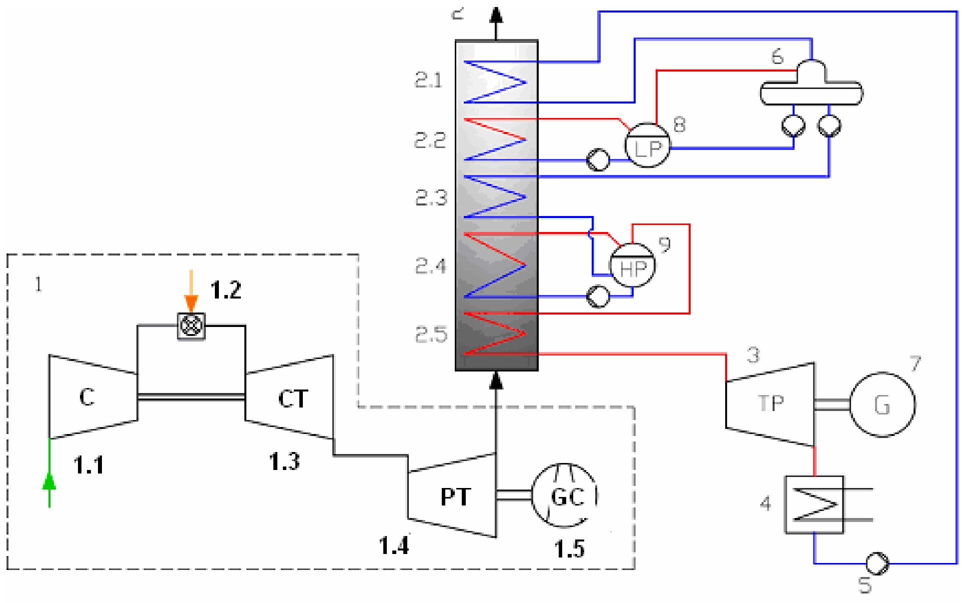

Fig. 3. Variant B of combined cycle plant : 1 – Turbocompressor plant (1.1 – Compressor, 1.2 – Combustion Chamber, 1.3 –Compressor Turbine, 1.4 – Power Turbine, 1.5 – Gas Compressor), 2 – Heat Recovery Boiler, 2.1 – Low Pressure Economizer, 2.2 –Preheating Loop (low pressure evaporator), 2.3 – High Pressure Economizer, 2.4 – High Pressure Evaporator, 2.5 –Superheater, 3 – Steam Turbine, 4 – Condenser, 5 –Condensate Pump , 6 – Dearator, 7 – Generator, 8 – Low Pressure Boiler Drum, 9- High Pressure Boiler Drum

C) Two-Pressure System

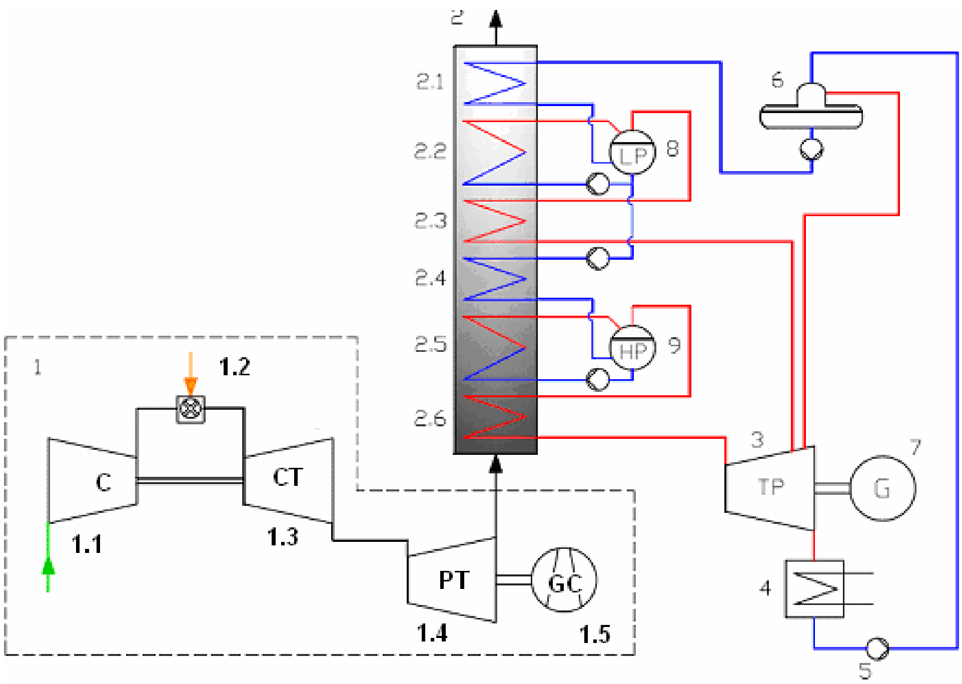

Fig. 4. Variant C of combined cycle plant: 1 – Turbocompressor plant (1.1 – Compressor, 1.2 – Combustion Chamber, 1.3 –Compressor Turbine, 1.4 – Power Turbine, 1.5 – Gas Compressor), 2 – Heat Recovery Boiler, 2.1 – Low Pressure Economizer, 2.2 – Low Pressure Evaporator, 2.5 –Low Pressure Superheater 2.4 – High Pressure Economizer, 2.5 – High Pressure Evaporator, 2.6 –High Pressure Superheater, 3 – Steam Turbine, 4 – Condenser, 5 –Condensate Pump , 6 – Dearator, 7 – Generator, 8 – Low Pressure Boiler Drum, 9- High Pressure Boiler Drum

2.2 Project assumptions and selection of system parameters

In case of analysis there have been made assumptions (joint for all three configurations) [1,2,4]:

- Range of gas turbine's power 55-100 %,
- Air intake temperature: 15˚*C*,
- Exhaust ducting pressure losses (including heat recover boiler) $\Delta p = 2\ kPa$,
- Difference between the fumes temperature after turbine and maximum live steam temperature: $\Delta T_1 = 25\ °C$ - single-pressure system ,
  $\Delta T_1 = 25\ °C$ - two-pressure system, section HP,
  $\Delta T_3 = 15\ °C$ - two-pressure system, section LP.
- pinch point temperature: $\Delta T_2 = 15\ °C$ - single-pressure system,
  $\Delta T_2 = 15\ °C$ - two-pressure system section HP,
  $\Delta T_4 = 15\ °C$ - two-pressure system, section LP.
- efficiency of economizer: $\eta_{ECO} = 0{,}9$ - single-pressure system,
  $\eta_{ECO\ LP} = 0{,}9$ - two-pressure system, section HP,
  $\eta_{ECO\ HP} = 0{,}95$ - two-pressure system, section LP.
- efficiency of steam turbine $\eta_{TP} = 0{,}9$ (steam turbine driving the generator directly),
- pressure in condenser $p = 0{,}005\ MPa$.

3. Calculations

Calculations are made in case of determination of optimal live steam pressure and term other parameters of system, all the power of steam turbine and efficiency of combined cycle. For organize all calculations are divided in the following variants:

- **Variant A1** – system A, load TG – 100 %,
- **Variant B1** – system B, load TG – 100 %,
- **Variant C1** – system C, load TG – 100 %,
- **Variant A2** – system A, load TG – changing,
- **Variant B2** – system B, load TG – changing,
- **Variant C2** – system C, load TG – changing.

3.1. Parameters of gas turbine

In Table 1 has been shown the parameters of gas turbine for part load.

*Table 1. Parameters of gas turbine for part load**

Load [%]	Power [*kW*]	Electric efficiency [%]	Exhaust Gas Temperature [˚*C*]	Exhaust Gas Flow [*kg/s*]
100	23557	32,69	550	80,66
90	21196	32,19	530	77,48
80	18833	31,46	511	74,14
70	16471	30,53	491	70,76
60	14108	29,31	473	67,07
50	11745	27,77	454	63,10

* Giving parameters are only when gas turbine powers electric generator; consider inlet and exhaust ducting pressure losses(also including heat recover boiler); are for all variants; air intake temperature 15˚*C*, atmospheric pressure 1,0132 *bar*.

3.2 Power of steam turbine and efficiency of combined cycle

Because of non-typical associational system, gas turbine is powered by gas compressor, and steam turbine is powered electric generator, efficiency of all system are been expressed such as electric efficiently. There have been accepted that gas turbine and steam turbine are powered electric generator.

Power of steam turbine:

$$N_{TP} = m \cdot \Delta i \cdot \eta_{mech} \cdot \eta_{gen}$$

were:

m - steam mass flow
Δi - steam turbine specific power
$\eta_{mech.}$ - mechanical efficiency (0,99)
$\eta_{gen.}$ - generator efficiency (0,985)

Efficiency of combined cycle:

$$\eta = \eta_{TG} \cdot \left(1 + \frac{N_{TP}}{N_{TG}}\right)$$

were:

η_{TG} - efficiently of gas turbine
N_{TP} - power of steam turbine
N_{TG} - power of gas turbine

3.3 Calculation results

In above tables there have been shown calculations results of optimal parameters of combined cycle for different variants. In Table 2 there have been shown calculation results for combined cycle with load 100 % of gas turbine, and with limit of steam quality on exit from steam turbine, x>0,9.

Table 2. Optimal parameters of combined cycle *with load 100 % of gas turbine*

Variant	Optimal live steam pressure [*bar*]		Power of steam turbine [*kW*]	Efficiency [-]	Exhaust gases temperature after boiler [˚*C*]
	LP	HP			
A1	-	40	12485	50,02	157
B1	-	40	12570	50,13	143
C1	4	30	14703	53,09	96

In Table 3 there have been shown optimal parameters of combined cycle in relation to load of gas turbine in range 55-100 % of power, with limit of steam dryness' degree after steam turbine, x>0,9. results of combined cycle have been made for different variants A, B and C in relation to load of gas turbine, but fresh steam pressure have been accepted as steady pressure in load of 55 % of gas turbine independent of analyse load.

Table 3. Optimal parameters of combined cycle *for 100 % load of gas turbine with steady live steam pressure independent of gas turbine load*

Variant	Optimal live steam pressure [*bar*]		Power of steam turbine [*kW*]	Efficiency [-]	Exhaust gases temperature after boiler [˚*C*]
	LP	HP			
A2	-	22	12281	49,73	141
B2	-	22	12370	49,86	126
C2	4	16	13645	51,63	99

Fig. 5 and 6 have been shown efficiency and power output in function of gas turbine load (there have been accepted live steam pressure steady 55 % load of gas turbine independent of gas turbine load).

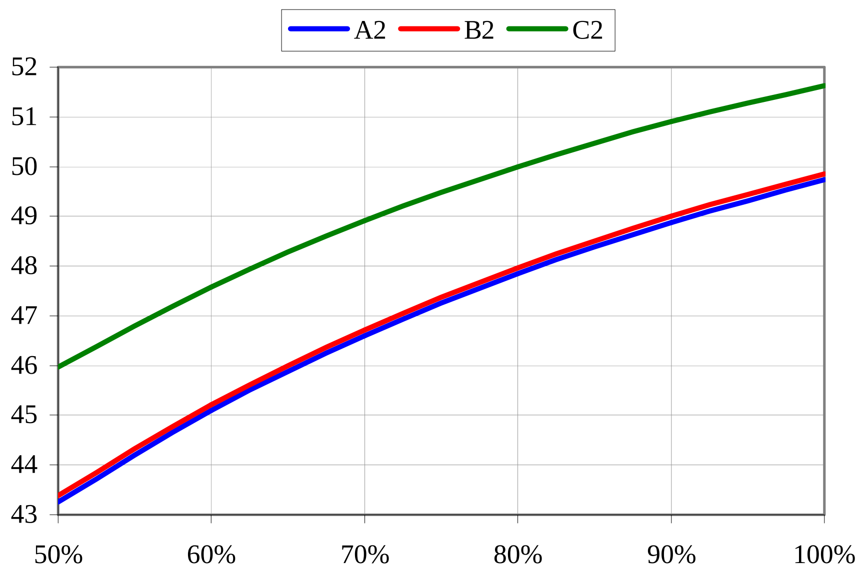

Fig. 5. Efficiency of combined cycle as a function of gas turbine load

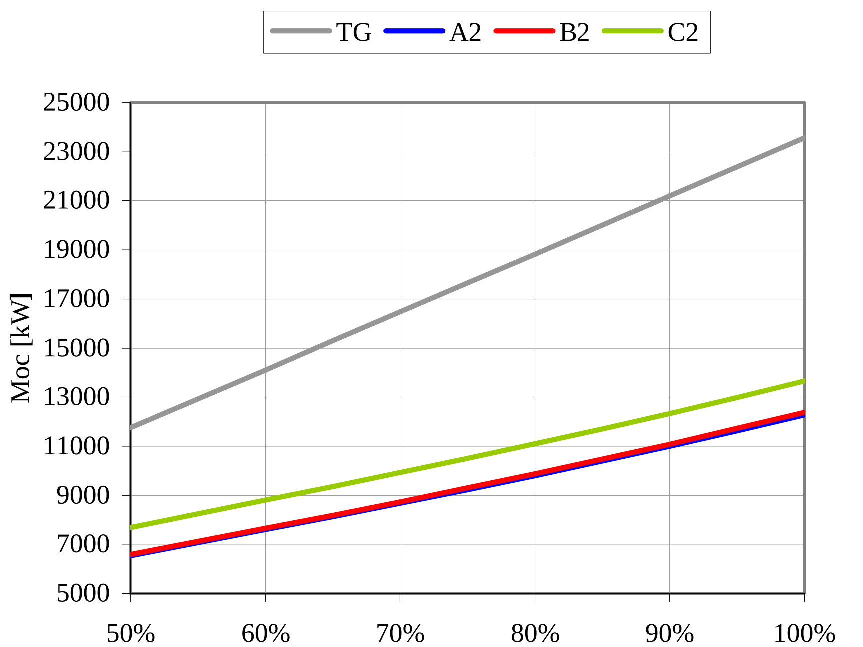

Fig. 6. Power output of gas turbine and steam turbine as a function of gas turbine load

4. Conclusion

From technical analysis of calculated variants there are following results. For variant with steady load of gas turbine, optimal live steam pressure is also maximum pressure. Futher increasing pressure is impossible because of reducing of steam quality (x) after steam turbine which should not be lower than 0,9 [1,4]. For system which are used to work with gas turbine working with changing loads, maximum live steam pressure conditions temperature of exhaust gases from gas turbine which is in minimum load. For variant A and B with load of 100 % power of gas turbine reducing steam pressure does not have a bigger influence on efficiency of combined cycle in comparison with calculated system without limits, while for two-pressure system it has considerably influence, see Tab. 2 and 3.

Exhaust gases temperature after boiler for variants A and B show that specific enthalpy of gases was not used exactly. Using variant C allows to reduce a gas temperature below 100˚*C*, which of course gives obviously increase of efficiency in comparison with single-pressure systems.

Comparison calculation results for variant A and C shows that using additional section of boiler allows a little increase in using specific enthalpy of gases. The differences are not so big. It can be said that, in this range of power, there are omitted and also it should be remembered that recover boiler is equipped in two additional section which considerably increase investment cost.

Choosing optimal associational system needs economic analysis of optimal solutions which are achieved from thermodynamic calculations.

Additional issue without economic analysis which have influence on analysis combined cycle is check if slope of gas turbine's power is acceptable about 230 kW, which is created by additional installation of recover boiler.

References

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