# Comparative Analysis Concerning the Use of the Thermal Potential of Combustion Gases in Industrial Cogeneration Systems

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*Abstract* – **Cogeneration is defined as the process of simultaneous production of heat and electricity, with the same installation (heat engine-electric generator group, turbine, etc.). Unlike classic Thermal Power Plants, cogeneration plants can be sized in correlation with the thermal energy requirement, which means that electricity is considered a "secondary" product. Throughout the article, taking into account the general theoretical aspects of the definition and operation of cogeneration systems, through the prism of specific energy indicators, the results obtained in the case of the implementation of a cogeneration system that uses the energy potential of gases are studied of combustion, for the production of the thermal agent for heating and preparation of hot water for consumption. of steam. From a functional point of view, at the level of a refinery, three superheated steam networks are needed, corresponding to three different pressure domains, which means that the results obtained in the study as the case may be, to refer to three distinct modes of operation of the cogeneration system. The main purpose of the operation of such a system for the combined production of electrical and thermal energy is to obtain as much energy as possible in the form of mechanical work by expanding steam in the turbines.**

**Cuvinte cheie:** *energie electrică, energie termică, cogenerare, producere, recuperare*

**Keywords:** *electricity, thermal energy, cogeneration, production, recovery.*

## I. INTRODUCTION

In general terms, cogeneration means the production of electricity and thermal energy, using the same fuel, in the same installations [1].

Cogeneration installations have experienced a special development, due to the energy crises, and as a result of the Kyoto Protocol, regarding the reduction of greenhouse gas emissions [2].

Cogeneration systems have high efficiences and for this reason result in cost reductions for the production of electricity and thermal energy, as well as reducing the amount of polluting emissions.

Thermal energy can be produced as hot water, steam, hot air and can be used to heat buildings and to provide hot water for consumption or in technological processes.

Combined heat and power plants can be integrated near the final consumer. Under these conditions, it results in a minimization of transport and distribution losses and an improvement of the general performance of the electricity and/or heating agent distribution network.

The implementation of the cogeneration system presents several important advantages such as: the application of the most modern energy solutions, the rational use of fuel, lower production costs, the increase in the level of comfort in the apartments, the use of electricity and the pumping of the surplus into the national energy system, becoming electricity producers, which leads to investment efficiency [3].

## II. GENERAL INDUSTRIAL COGENERATION TECHNOLOGIES

Industrial cogeneration systems are characterized by a much higher heating coefficient value. Industrial cogeneration systems can be economically efficient depending on the nature and degree of flattening of heat demand and its simultaneity with electricity demand [3], [4].

Industrial consumers mainly use steam, it being preferred to other thermal agents in heating and expansion processes in turbines and in other uses. Depending on the uses that the steam has, it is necessary for it to have certain qualities: temperature, pressure and position in relation to the phase balance curves [5].

The main industrial cogeneration systems are made with very low power backpressure groups (in piston groups), up to turbogenerator groups with superheated parameters, with condensing groups and adjustable outlets and, more recently, with condensing turbine groups with recovery the heat from the combustion gases from the technological furnaces [6].

Among the most widespread cogeneration technologies with steam turbines are: cogeneration system with counterpressure groups with high parameters (fig. 1), characterized by two pressure levels at the steam inlet to the turbines; cogeneration systems with groups with high parameters (fig. 2), which are equipped on the electric power production side with two groups of 19 MW each, simultaneously with the possibility of supplying heat to several groups of industrial consumers; cogeneration with condensing groups and adjustable outlets of 50 MW at live steam parameters of 130 at and 565°C (fig. 3) [1], [5], [6].

Lately, the development and modernization of industrial steam turbine cogeneration systems has led to the search and identification of efficient solutions for the production of both types of energy (electrical and thermal) both from the functional and economic point of view, as well as from the point of view of environmental protection.

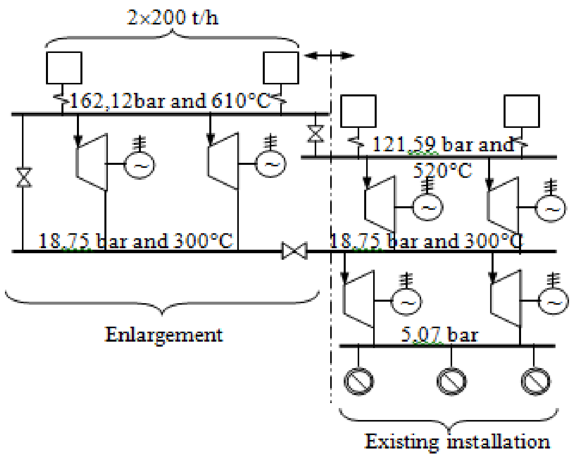


Fig. 1. Cogeneration system with backpressure groups with very high parameters [6]

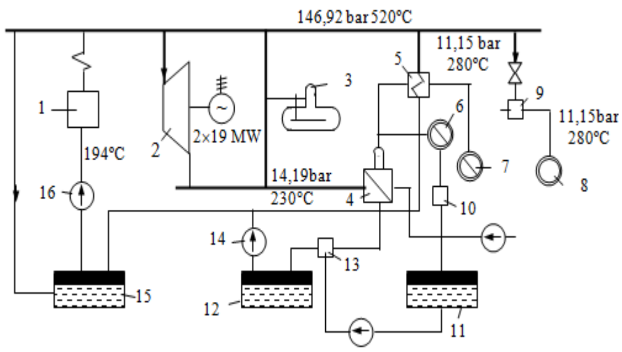


Fig. 2. Cogeneration system with high-parameter back-pressure groups:1 - forced-pass boiler; 2 - back-pressure turbine; 3 - heat accumulator; 4 - steam transformer battery; 5 - superheater; 6,7,8 - steam consumer group; 9 - reduction-cooling station; 10 - total demineralization plant; 11,12 - pressurized condensate tanks; 13 - degasser; 14 - feed pump; 15 - start-up tank; 16 - boiler water feed pump [4]

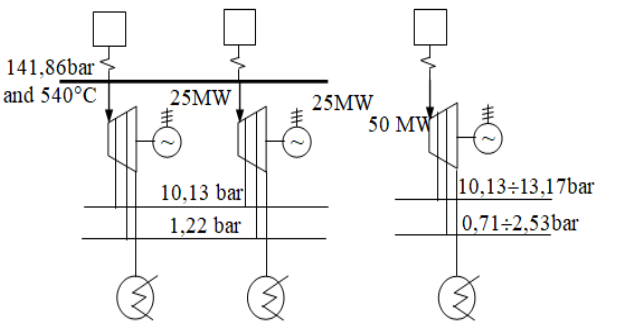


Fig. 3. Combined heat and power system with 25 MW condensing units with adjustable outlets and a 50 MW unit in expansion [3]

Such a solution can be considered as a cogeneration system with condensing turbine groups with heat recovery from the combustion gases of technological furnaces (fig. 4).

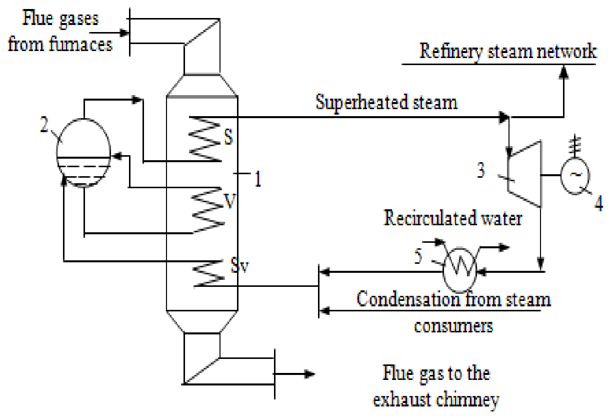


Fig. 4. Cogeneration system with heat recovery from combustion gases: 1 – recovery boiler; 2 – drum; 3 – turbine; 4 – generator; 5 – turbine condenser; E – saver; V – vaporizer; S - superheater [5]

Such an industrial cogeneration system, which is the subject of the energy efficiency analysis in this paper, is properly studied for three pressure ranges: low pressure steam network (p=3...6 bar); medium pressure steam network (p=12...25 bar); high pressure steam network (p=30...60 bar). [1]

The heat recovery plant basically complements a basic steam-operated plant working on the Clausius-Rankine cycle. The boiler of the basic plant produces saturated and superheated steam which in a certain proportion supplies different consumers, the unused superheated steam being diverted to a steam turbine coaxially mounted with an electric generator [7].

In the case of a variable demand for thermal energy, the system can work at maximum capacity, increasing the production of electricity. From the side outlets and through the final outlet, the $T_2$ turbine supplies all three steam networks of the refinery.

The cogeneration system with the recovery of heat from the combustion gases consists of a classic recovery boiler (1), for obtaining steam from the combustion gases, which is subsequently delivered to the networks corresponding to the respective state parameters at the refinery level [8]. Combustion gases circulate through the boiler under the action of an exhauster and are discharged into the atmosphere by means of a chimney with a high height. Part of the superheated steam is delivered to the refinery's steam network, with compatible parameters, when there is high demand from consumers, and the other part is expanded in the turbine (3). The turbine drives the electric generator (4) which supplies electricity to the refinery network. The steam released in the turbine is condensed and refuels the recovery boiler by means of a recirculation pump.

The realization of such cogeneration systems for the recovery of heat from the combustion gases, in comparison with the classical heat recovery systems, is possible only through steam generators. If during a period, the demand of steam consumers is low, the classic steam production system works with a very low economic efficiency, a possibility of efficiency being the expansion of steam not requested by consumers in turbines that drive electric generators.

In order to obtain as much energy as possible in the form of mechanical work by expanding the steam in the turbine, it is necessary that the pressure and degree of superheating of the steam be as high as possible [4].

## III. Energy Indicators Of Cogeneration Systems With Combustion Gas Heat Recovery

The technical-economic parameters associated with this system of combined production of electricity and thermal energy are imposed by the heat potential of the combustion gases, by the flow rate and temperature level and by the quantity and quality of the steam produced.

These parameters are correlated through the physical properties of combustion gases and steam with the energy requirement at the level of the industrial consumer.

The determination of the technical-economic parameters is based on the values of the constructive-functional parameters of the cogeneration plant, which depending on the pressure level of the steam produced [4].

The analysis of the operation of the cogeneration system with heat recovery from the combustion gases is based on the following energy efficiency indicators [4], [6]:

1. The heat potential that can be recovered from the combustion gases:

$$Q_{ga} = m_{ga} \cdot \left(i^{i}_{ga} - i^{e}_{ga}\right) [kJ/s] \tag{1}$$

2. Steam flow produced in the boiler:

$$m_{steam} = \frac{Q_{ga}}{i^{p}_{ab} - i^{boiler}_{water}} [kg/s] \tag{2}$$

3. Steam turbine theoretical power:

$$P_T = m_{steam} \cdot \left(i^{p}_{ab} - i^{e}_{ab}\right) [kW] \tag{3}$$

4. Total area of heat exchange surfaces in the recovery boiler:

$$A_{sch,boiler} = \frac{Q_{ga}}{k \cdot \Delta t_{med}} [m^2] \tag{4}$$

5. Power required for boiler water supply pump:

$$P_{pump,al} = \frac{m_{steam} \cdot \Delta p}{\rho_a \cdot \eta \cdot 1000} [kW] \tag{5}$$

6. Water consumption at the turbine condenser:

$$m_{water} = \frac{m_{water} \cdot \left(i^{e}_{ab} - i_{cd}\right)}{c_{water} \cdot \Delta t_{water}} [t/h] \tag{6}$$

7. Energy efficiency of the recovery boiler:

$$\eta = \frac{Q_{steam}}{Q_{gas}} \cdot 100 [\%] \tag{7}$$

8. Efficiency of electricity generation:

$$\eta_{el} = \frac{P_T \cdot \eta_{gen}}{Q_{ga}} \cdot 100 [\%] \tag{8}$$

were $m_{ga}$ is the mass flow of combustion gases, [kg/h], $i^{i}_{ga}$ is the enthalpy of the combustion gases at the entrance to the recovery boiler, determined according to the temperature at which the combustion gases enter the boiler, [kJ/kg], $i^{e}_{ga}$ is the enthalpy of the combustion gases at the exit from the recovery boiler, determined according to the temperature at which the combustion gases leave the boiler, [kJ/kg], $i^{p}_{ab}$ is the enthalpy of steam entering the turbine, [kJ/kg], $i^{boiler}_{water}$ is the enthalpy of the feed water of the recovered boilerr, [kJ/kg], $i^{e}_{ab}$ is the enthalpy of the steam at the turbine exit, [kJ/kg], k is the global heat transfer coefficient, [W/(m$^2$·°C)], $\Delta t_{med}$ is the average logarithmic temperature variation, [°C], Δp is the difference between the pump discharge pressure and suction pressure, [°C], $\rho_a$ is the water density, [kg/m$^3$], η is the pump's average efficiency, [%], $i_{cd}$ is the enthalpy of the condensate, [kJ/kg], $c_{water}$ is the specific heat of water at constant pressure, [kJ/(kg·°C)], $\Delta t_{water}$ is the cooling water temperature variation, [°C], $Q_{steam}$ is the heat contained in the steam produced in the recovery boiler, [kJ/s], $Q_{gas}$ is the heat contained in the combustion gases at the entrance to the recovery boiler, [kJ/s] and $\eta_{gen}$ is the efficiency of the electric generator, [%].

## IV. Data Required For Performing The Analysis

To carry out the case study, which aims to analyze the operation of industrial co-generation systems, using the thermal potential of the combustion gases, the following hypotheses will be taken into account:

- three different temperature regimes for the combustion gases at the boiler entrance will be considered heat recuperator: regime a, regime b and regime c (table 1);
- the temperature of the combustion gases at the exit from the recovery boiler is given in table.1;
- the flow of combustion gases entering the recovery boiler is determined by calculation, depending on the chemical composition of the combustion gases, related to the three thermal regimes studied;
- the pressure level of the steam produced by the recovery boiler is: 4 bar, 20 bar and 45 bar respectively;
- the chemical composition of the combustion gases is variable depending on the operating regime (table 1).

TABLE I.
INPUT PARAMETERS FOR PERFORMING THE ENERGY ANALYSIS RELATING TO THE OPERATION OF THE CONSIDERED COGENERATION SYSTEM [5], [9]

| No. crt. | Parameter | Symbol | Unit Measure | Operating mode | | |
|---|---|---|---|---|---|---|
| | | | | a | b | c |
| 1. | The temperature of the combustion gases at the entrance to the recovery boiler | $t^{i}_{ga}$ | °C | 676,30 | 483,80 | 351,70 |
| 2. | The temperature of the combustion gases at the exit from the recovery boiler | $t^{e}_{ga}$ | °C | 150 | 140 | 135 |
| 3. | The pressure of the steam produced in the recovery boiler | $P_{steam}$ | bar | 45 | 20 | 4 |

| No. crt. | Parameter | Symbol | Unit Measure | Operating mode | | |
|---|---|---|---|---|---|---|
| | | | | a | b | c |
| 4. | The temperature of the steam produced in the recovery boiler | $T_{steam}$ | °C | 400 | 400 | 300 |
| 5. | Water pressure at feed pump inlet [9] | $P_{water, al}$ | bar | 2 | 2 | 2 |
| 6. | Steam pressure at the turbine exit | $p_{ab}^{e}$ | bar | 0,06 | 0,06 | 0,06 |
| 7. | The temperature of the water supply to the recuperator boiler | $t_{water}^{boiler}$ | °C | 40 | 40 | 40 |
| 8. | The head of the steam at the exit of the turbine | $x_{ab}$ | - | 0,90 | 0,89 | 0,88 |
| 9. | Condensate pressure | $p_{cd}$ | bar | 0,10 | 0,10 | 0,10 |
| 10. | The average efficiency of the recovery boiler water supply pump [9] | $\eta$ | % | 70 | 70 | 70 |
| 11. | Global heat transfer coefficient in the recovery boiler [9] | k | $W/(m^2 \cdot K)$ | 45 | 45 | 45 |
| 12. | Fuel flow [9] | $B_c$ | $Nm^3/h$ | 109,87 | 129,52 | 88,125 |
| 13. | The volume of wet combustion gases when burning one $Nm^3$ of fuel [9] | $v_{ga}$ | $Nm^3/Nm^3$ | 29,59 | 20,20 | 26,54 |
| 14. | Chemical composition of combustion gases [9] | $CO_2$<br>$O_2$<br>CO<br>$N_2$ | %<br>%<br>%<br>% | 3,34<br>14,42<br>0,0004<br>82,23996 | 5,61<br>11,12<br>0,0000264<br>83,2636 | 4,19<br>13,61<br>0,00002<br>82,20998 |
| 15. | Excess air coefficient [9] | $\lambda$ | - | 4,42 | 2,07 | 3,10 |
| 16. | The temperature variation of the cooling water at the condenser | $\Delta t_{water}$ | °C | 5 | 5 | 5 |
| 17. | Average specific heat of cooling water at turbine condenser [9] | $C_{water}$ | kJ/(kg·K) | 4,184 | 4,184 | 4,184 |
| 18. | The efficiency of the electric generator | $\eta_{gen}$ | - | 0,995 | 0,995 | 0,995 |

## V. RESULTS

Determining the energy efficiency indicators that characterize the operation of the cogeneration system with heat recovery from the combustion gases of a certain fuel, requires the use of the mathematical relationships presented in Part IV of this article.

For an eloquent presentation of the results obtained, considering that it is about three different operating regimes of the analyzed cogeneration system, it is resorted to the presentation of these results both in tabular form (table 2) and in graphic form, for the main indicators of energy efficiency (fig. 5...8).

TABLE II.
FUNCTIONAL ENERGY INDICATORS OF THE COENERATION SYSTEM WITH HEAT RECOVERY FROM COMBUSTION GASES RESULTS

| No. crt. | The determined parameter | Symbol | Unit Measure | Operating mode | | |
|---|---|---|---|---|---|---|
| | | | | a | b | c |
| 1. | Thermal potential, recoverable from combustion gases | $Q_{ga}$ | kJ/h | 2406654,78 | 1252213,51 | 691828,87 |
| 1.1 | Combustion gas flow | $m_{ga}$ | $Nm^3/h$ | 3251,05 | 2616,30 | 2338,84 |
| 1.2 | Enthalpy of combustion gases at the entrance to the recovery boiler | $i_{ga}^{i}$ | $kJ/ Nm^3$ | 937,76 | 663,05 | 472,98 |
| 1.3 | Enthalpy of carbon dioxide from the combustion gases at the entrance to the recovery boiler | $i_{CO2}$ | $kJ/Nm^3$ | 1405,64 | 958,31 | 668,97 |
| 1.4 | Enthalpy of oxygen from the combustion gases at the entrance to the recovery boiler | $i_{O2}$ | $kJ/Nm^3$ | 967,58 | 674,98 | 481,35 |
| 1.5 | Enthalpy of nitrogen from the combustion gases at the entrance to the recovery boiler | $i_{N2}$ | $kJ/Nm^3$ | 913,53 | 641,56 | 461,55 |
| 1.6 | Enthalpy of carbon monoxide from the combustion gases at the entrance to the recovery boiler | $i_{CO}$ | $kJ/Nm^3$ | 794,23 | 511,28 | 385,21 |
| 1.7 | The enthalpy of the combustion gases at the exit from the recovery boiler | $i_{ga}^{e}$ | kJ/kg | 197,49 | 184,43 | 177,18 |
| 1.8 | The enthalpy of the carbon dioxide from the combustion gases at the exit from the recovery boiler | $i_{CO2}$ | $kJ/Nm^3$ | 261,56 | 232,01 | 223 |
| 1.9 | Enthalpy of oxygen from the combustion gases at the exit from the recovery boiler | $i_{O2}$ | $kJ/Nm^3$ | 198,95 | 183,76 | 177,11 |
| 1.10 | Enthalpy of nitrogen from the combustion gases at the exit from the recovery boiler | $i_{N2}$ | $kJ/Nm^3$ | 194,63 | 181,32 | 174,84 |
| 1.11 | Enthalpy of carbon monoxide from the combustion gases at the exit from the recovery boiler | $i_{CO}$ | $kJ/Nm^3$ | 157,73 | 147,34 | 141,99 |
| 2. | The flow of steam produced in the recovery boiler | $m_{steam}$ | kg/h | 794,14 | 406,99 | 222,79 |
| 2.1 | The enthalpy of the steam produced in the recovery boiler | $i_{ab}^{p}$ | kJ/kg | 3202 | 3246 | 3065 |
| 2.2 | Enthalpy of the water entering the recovery boiler | $i_{water}^{bolier}$ | kJ/kg | 171,5 | 169,2 | 167,7 |
| 3. | Theoretical steam turbine power | $P_T$ | kW | 193,46 | 106,85 | 48,78 |
| 3.1 | Enthalpy of steam at the exit from the turbine | $i_{ab}^{e}$ | kJ/kg | 2325 | 2300,85 | 2276,7 |

| No. crt. | The determined parameter | Symbol | Unit Measure | Operating mode | | |
|---|---|---|---|---|---|---|
| | | | | a | b | c |
| 4. | The total area of the heat exchange surfaces, at the recovery boiler level, in the case of countercurrent flow | $A^{cc}_{sch,boiler}$ | $m^2$ | 296,20 | 303,59 | 210 |
| 4.1 | Average logarithmic difference in temperature during countercurrent flow | $\Delta t_{med,cc}$ | °C | 180,56 | 91,66 | 73,21 |
| 5. | Power required for boiler water supply pump | $P_{pump,al}$ | kW | 1,37 | 0,29 | 0,02 |
| 5.1 | The difference between the water pressure entering the boiler and entering the pump | $\Delta p$ | bar | 43 | 18 | 2 |
| 5.2 | The density of the water with which the boiler is fed | $\rho_a$ | $kg/m^3$ | 992,22 | 992,22 | 992,22 |
| 6. | Water consumption at the turbine condenser | $m_{water}$ | t/h | 82,51 | 41,81 | 24,24 |
| 6.1 | Condensate enthalpy | $i_{cd}$ | kJ/kg | 151,49 | 151,49 | 151,49 |
| 6.2 | Condensate temperature | $t_{cd}$ | °C | 36,18 | 36,18 | 36,18 |
| 7. | The energy efficiency of the recovery boiler | $\eta_{CZ}$ | % | 83,41 | 76,16 | 61,73 |
| 7.1 | The heat contained in the steam produced in the recovery boiler | $Q_{steam}$ | kJ/s | 706,34 | 366,97 | 189,68 |
| 7.2 | The heat contained in the combustion gases entering the recovery boiler | $Q_{gas}$ | kJ/s | 846,86 | 481,87 | 307,28 |
| 8. | The yield of electricity production | $\eta_{el}$ | % | 28,79 | 30,56 | 25,26 |

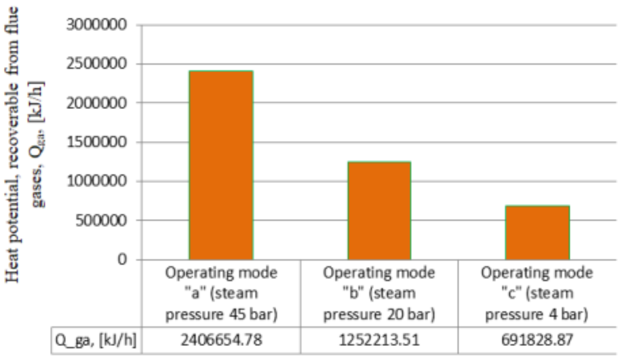


Fig. 5. The theoretical power of the steam turbine when operating in the three pressure regimes

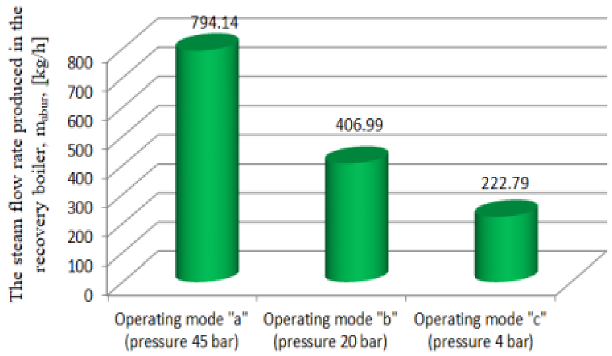


Fig. 6. The variation of the steam flow produced in the recovery boiler depending on the pressure regime of the steam produced

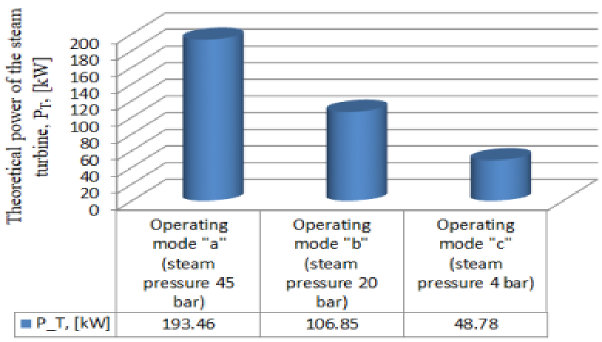


Fig. 7. The theoretical power of the steam turbine when operating in the three pressure regimes

From the analysis of the diagram in figure 6, it can be seen that with the increase in the pressure of the produced steam from 4 bar to 45, the steam mass flow rate increases approximately 3.56 times (from 222.79 kg/h to 794.14 kg/h).

The increase in the theoretical power at the turbine shaft is directly proportional to the increase in the pressure of the superheated steam released in the turbines, and from the diagram in figure 7, an increase in the electric power from 48.78 kW to 193.46 kW can be observed, i.e. approximately 4 times.

At the same time, the increase in the pressure of the steam produced in the boiler from 4 to 45 bar leads, in addition to the increase in the theoretical power of the steam turbine, to an increase in the yield of electricity production from 25.26% to 30.56% (fig. 8). It is observed that the highest electrical efficiency is obtained in the case of expansion of medium-pressure steam in the steam turbine, even if the electrical power produced is lower than in the variant of expansion of high-pressure steam.

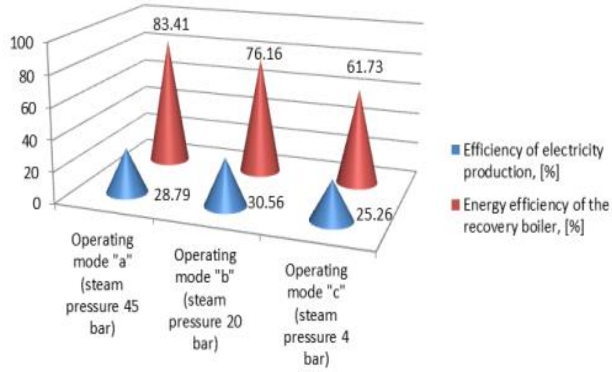


Fig. 8. The variation of the yield of electricity production, $\eta_{el}$ and, of the energy yield of the recovery boiler depending on the pressure regime of the produced steam

Also, the same increase in the pressure of the steam produced in the recovery boiler, leads to the increase of the recovery boiler's load by approximately 35%, from 61.73% to 83.41% (fig.8).

## VI. Conclusion

The cogeneration systems with the recovery of heat from combustion gases requires steam generators with special construction that allow operation at variable regimes of flow rates, pressures, thermal energy and electricity.

When, in a certain period, the steam demand of thermal consumers is low, the operation of the steam production system would be performed with a reduced economic efficiency.

For the energy efficiency of the operation of such a system, functional measures are required that concern the expansion of steam not requested by thermal consumers on the industrial platform, in turbines that drive electric generators.

Obtaining the greatest possible amount of energy in the form of mechanical work by expanding the steam in the turbine is possible in the conditions where the pressure and degree of superheating of the steam are as high as possible.

In case of high pressure steam production ($p_{steam}$=45 bar), the efficiency of the recovery boiler is approximatively 83%, obtaining a flow of 0,22 kg/s, at a temperature of 400°C and a pressure of 45 bar, which implies a thermal potential of approx 706 kJ/s. The flow of combustion gases required in this case is 0.903 kg/s with a temperature of approximately 676°C, which represents a thermal potential of approximately 846 kJ/s.

The efficiency of electricity production under these conditions is approximately 29%.

In the case of medium pressure steam production ($p_{steam}$=20 bar), the efficiency of the recovery boiler is approximately 76%, obtaining a flow rate of 0.11 kg/s, at a temperature of 400°C and a pressure of 20 bar, which implies a thermal potential of approximately 367 kJ/s. The flow of combustion gases required in this case is 0.727 kg/s with a temperature of approximately 483°C, which represents a thermal potential of approximately 482 kJ/s.

The efficiency of electricity production under these conditions is approximately 30%.

In the case of low pressure steam production ($p_{abur}$=4 bar), the efficiency of the recovery boiler is approximately 62%, obtaining a flow rate of 0.06 kg/s, at a temperature of 300°C and a pressure of 4 bar, which implies a thermal potential of approximately 190 kJ/s. The flow of combustion gases required in this case is 0.650 kg/s with a temperature of approximately 351°C, which represents a thermal potential of approximately 307 kJ/s.

The efficency of electricity production under these conditions is approximately 25%.

Analyzing further the results obtained in the three operating regimes of the turbine cogeneration system with steam produced by recovering the thermal potential from the combustion gases, the following can be observed:

1. the cooling water flow required at the turbine condenser decreases with the decrease in the pressure of the generated steam, from 82.51 t/h in the case of steam produced at 45 bar, to 24.24 t/h in the case of steam produced at 4 bar , due to the variation of the steam flow rate;

2. the power required for the water supply pump of the boiler, although it increases approximately 68 times at the pressure of 45 bar compared to the pressure of 4 bar, the absolute level of consumption is quite small, representing approximately 0.71% of the electrical power produced by turbines - in the case of operating mode a, 0.27% of the electrical power produced by the turbines - in the case of operating mode b, respectively, 0.04% of the electrical power produced by the turbines - in the case of operating mode c operation;

3. The main disadvantage of increasing the pressure of the produced steam is represented by the increase in investment. This increase is due, on the one hand, to the need for a larger heat exchange surface (a 44.57% increase in the surface of the coils at a pressure of 20 bar compared to 4 bar) and, on the other hand, to the increase in thickness tube walls.

When the cogeneration system with the recovery of heat from the combustion gases does not operate in cogeneration mode, the production of low pressure steam is efficient, and if the installation operates in cogeneration mode, the production of steam of the highest pressure is very advantageous.

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