Optimization of Waste Heat Utilization in Gas Turbine Pipeline Compressor Station

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Preliminary note

Gas turbine driven pipeline compressor stations employed in existing pipelines and also in those that will be built due to increasing demand for natural gas present a significant potential for production of additional electricity and useful heat by utilizing the waste heat from the gas turbine process. The paper analyzes and compares several different configurations of heat recovery steam generator (HRSG) both in terms of thermodynamic and thermoeconomic criteria. In order to find an optimal configuration an exergo-economic HRSG optimization is conducted for finding the compromise between the cost of the exergy losses and investment costs of HRSG surfaces. Comparison and selection of configurations is based on the Pareto front through finding an optimum between maximal system efficiency and minimal cost of electricity.

Optimiranje korištenja otpadne topline u procesu plinske kompresorske stanice

Prethodno priopćenje

Plinsko turbinske kompresorske stanice u postojećim plinovodima kao i u onima koji će tek biti izgrađeni zbog povećane potražnje za prirodnim plinom predstavljaju značajan potencijal za dodatnu proizvodnju električne i korisne toplinske energije korištenjem otpadne topline iz procesa plinske turbine. U radu se analiziraju i uspoređuju nekoliko različitih konfiguracija generatora pare na otpadnu toplinu obzirom na termodinamičke i termoekonomske kriterije. Izbor optimalne konfiguracije temelji se na eksergo-ekonomskoj optimizaciji generatora pare pronalaženjem kompromisa između troškova eksergetskih gubitaka i cijene izmjenjivačkih površina. Usporedba i odabir konfiguracije temelji se na Pareto fronti traženjem optimuma između maksimalnog stupnja korisnosti procesa i minimalne proizvodne cijene električne energije.

1. Introduction

A major share in global production of electricity and heat nowadays belongs to technologies based on firing the fossil fuels i.e. coal, oil and natural gas. Thereby natural gas obtrudes since its reserves are larger than oil and its firing produces smaller amount of greenhouse gases than other fossil fuels. In such conditions natural gas demand has had a rising trend since the beginning of last decade [1]. In transport and supply of natural gas, an important role belongs to gas pipelines where compressor stations are used for maintaining the operating pressure. Generally centrifugal compressors in these stations are driven by gas turbine (GT), steam turbine (ST), electromotor (EM) or internal combustion engine (ICE) [2]. In the US and Canada ICE are represented with ca. 70 % and are followed by gas turbines whose share is ca. 25 %. The rest belongs to steam turbines and electromotors whose share is minor. In Europe the share of gas turbines is somewhat higher [3,4]. This work focuses on analysis of waste heat utilization possibilities by means of combined gas and steam turbine process. Heat and mass balance calculation followed by exergo-economic HRSG optimization is conducted by means of Matlab. System parameters at off-design regimes are calculated by means of GateCycle [5]. Comparison and selection of configurations is based on the Pareto front through finding optimum between maximal system efficiency and minimal cost of electricity. Configurations viability is examined in demand scenario in order to evaluate the viability of investment and operation with respect to assumed fuel cost, electricity sellback price and CO_2 emission certificate prices. Sensitivity analysis is performed in order to estimate system viability in case of varying fuel prices and CO_2 emission certificates.

Symbols/Oznake

A - surface area, m² / - površina

c_p - specific heat capacity, kJ/kgK / - specifični toplinski kapacitet

c - specific surface cost, \$/m² / - specifična cijena površine

C - surface cost, \$ / - cijena površine

D - functional duration of the plant / - životni vijek postrojenja

E - exergy / - eksergija

exp - surface exponent / - eksponent površine

f_c - system complexity factor / - faktor složenosti sustava

h - specific entalpy, kJ/kg / - specifična entalpija

H - entalpy, kW / - entalpija

HP - high pressure / - visoki tlak

I - exergy losses, kW / - eksergetski gubici

IP - intermediate pressure / - srednji tlak

k_I - specific cost of exergy losses / - specifični trošak eksergetskih gubitaka

K_I - cost of exergy losses / - trošak eksergetskih gubitaka

K_{HRSG} - cost of HRSG surfaces / - trošak površina kotla utilizatora

K_{total} - total annualized HRSG cost / - ukupni godišnji trošak kotla utlizatora

LP - low pressure / - niski tlak

M - mass flow, kg/s / - maseni protok

R - gas constant, kJ/kgK / - plinska konstanta

T - temperature, K / - temperatura

η - system efficiency, % / - stupanj korisnosti sustava

Indices/Indeksi

eco - economizer / - ekonomajzer

eva - evaporator / - isparivač

sh - superheater / - pregrijač

a - ambient / - okolišni

g - gas / - plin

2. Pipeline compressor stations

Since transport of natural gas via roads or sea necessitates its conversion into liquefied natural gas (LNG), compressed natural gas (CNG) or liquid petroleum gas (LPG) [1] gas pipelines present an irreplaceable supply system enabling the natural gas demand to be met both for households and various industries. Through gas pipeline networks required natural gas is supplied from the reserves owing countries to the consuming industrial and residential centers as shown in Figure 1 [6].
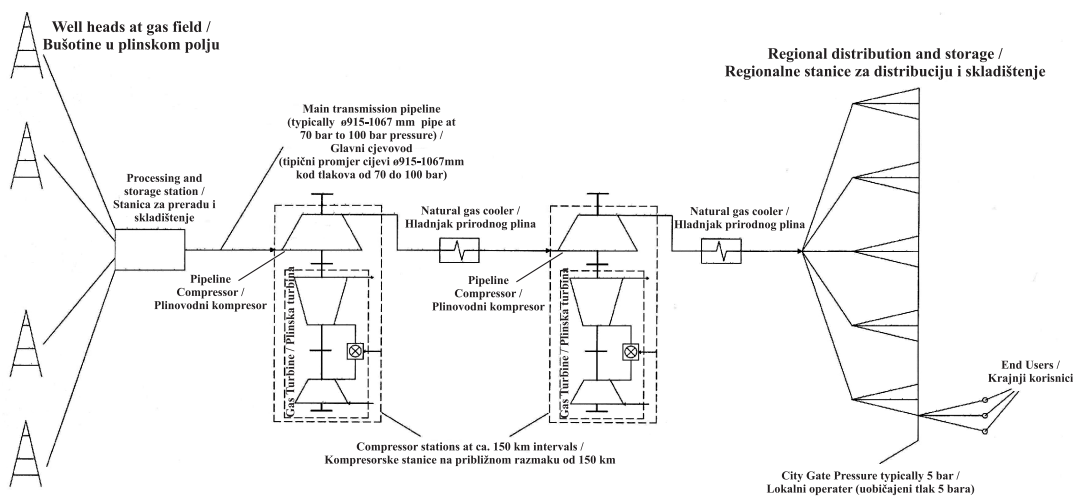

Figure 1. Simplified schematic of a natural gas pipeline system

Slika 1. Pojednostavljena shema plinovodnog sustava

Because of the pressure loss which occurs during gas transmission through pipelines due to friction and gas off-take, the gas pressure has to be boosted every 150 km in order to maintain operating pressure in the pipeline at approximately 80 bar. That is done by means of a compressor station comprising several compressor units which, depending on requirements on mass flow and operating pressure, may be connected in series or parallel. Since the supply pressure to the pipeline station can fluctuate significantly, the pressure ratio generated in the compressor has to vary accordingly. Because of economic reasons, operation at optimum efficiency is constantly required also under such variable conditions, wherefore the compressor must be capable of operating over as wide a speed range as possible. It is important to provide a compressor operation control so that on reduction of the compressor load it is possible to approach the performance map limits. Target is minimizing the energy input respectively operation at highest possible level of efficiency [7]. Except in gas pipelines, compressor stations may be employed in underground storage facilities in order to raise the gas pressure being injected into storage, or to compress the natural gas as it leaves storage to be fed into the pipeline [4]. Pipeline compressors are also used in gas collection and gas processing plants as export compressors [8]. Generally each unit consists of a centrifugal compressor driven by a gas turbine, steam turbine or in some cases electric motor. Since gas turbines are the only prime mover which releases respectable amount of heat into the atmosphere only such units are the subject of this work. At the time being heat embraced in the flue gases leaving the gas turbine of pipeline compressor stations is not utilized at all or is only partially recovered. The potential of this heat in the pipeline compressor stations which are part of gas mains of Russia and member-countries of the Commonwealth of Independent States is estimated at 1260 million GJ/year. By utilizing this heat electricity of up to 39400 GWh can be generated. However recently in some countries natural gas operators are obliged to utilize the waste heat [9]. Most compressors in pipeline compressor stations are driven by gas turbines of 6 to 25MW capacity. Typically, exhaust gas temperatures are from 450-550 °C and present valuable clean energy in the form of exhaust heat, which may be recovered [10]. Figure 2 shows a schematic of considered pipeline compressor comprising four centrifugal compressors driven by gas turbines class THM1203. Depending on required mechanical power at compressor shaft, there are two or three gas turbines in operation whilst one gas turbine is used for backup.

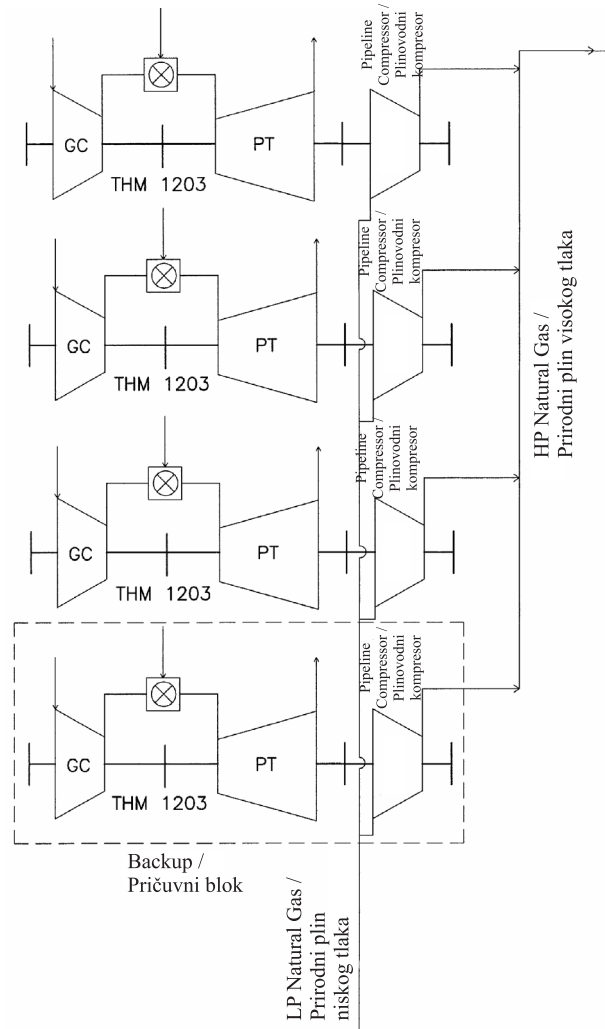From demand pattern for the year 2005 a load duration curve shown in Figure 3 is constructed.

Figure 2. Considered compressor station schematic
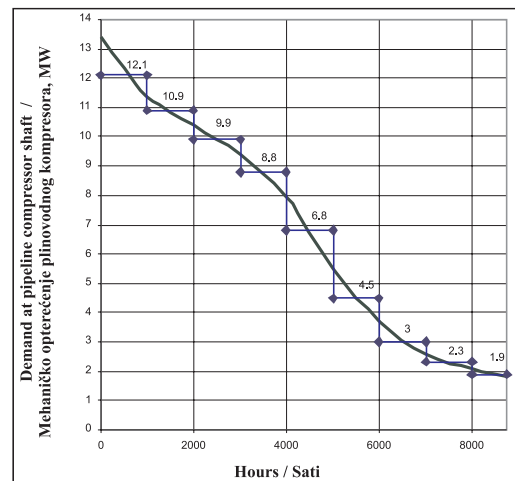**Slika 2.** Shema razmatrane kompresorske stanice

Figure 3. Load duration curve
Slika 3. Krivulja trajanja opterećenja

For simplification of calculations this annual system characteristic is divided into ten operating regimes whose power demand is obtained as an average power over respective time period. In the first five operating regimes (from 12.1MW-6.8MW) there are three gas turbines in operation and in other five regimes (6.8MW-1.9MW) there are two gas turbines in operation. Enthalpy of flue gases is calculated as a product of number of gas turbines, flue gas mass flow and flue gas specific enthalpy. Enthalpies for ten operating regimes are shown in Figure 4.
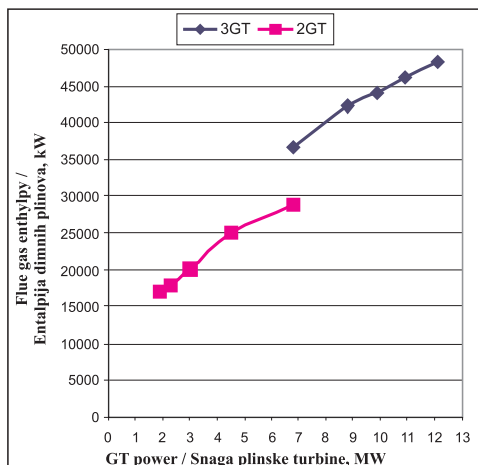

Figure 4. Flue gas enthalpy at GT outlet in compressor station operating regimes

Slika 4. Entalpija dimnih plinova na izlazu iz plinske turbine u režimima rada kompresorske stanice

Figure 4 indicates a significant decline of available flue gas enthalpy in operating regimes where flue gas enthalpy in the lowest regime reaches 31 % of flue gas enthalpy at the rated load.

3. Waste heat utilization possibilities

Independent from GT flue gas parameters and quantity which depend on project and pipeline compressor operating regime each waste heat utilization configuration is generally set up in a way that the generated steam in HRSG is used for producing rotational energy in ST. This energy can further be transferred into different forms of energy depending on economical and technical conditions valid for the certain project in consideration.

3.1. Electricity power production

In case of electricity demand, steam turbine [11] is used for running the AC generator, which is connected in parallel with national grid. Thereby the steam cycle has to be configured in such a way as to ensure the ability of the electrical load following independently a gas turbine

operating regime. Electricity produced at generator is sold on the free market under conditions which may vary significantly depending on various parameters as shortage or excess of power in the grid. CC systems employing the HRSG for waste heat utilization from GT and ST for running the AC generator can be quite different employing one or more HRSGs and one or more steam turbines. Furthermore HRSG can be single-, double-, or triple-pressure followed by balance of plant equipment which can be with or without condensate preheaters. The approach accepted in this paper is based on application of one HRSG and one ST as such concept offers advantages in terms of design, capital costs and operation in comparison with concept with several HRSGs and ST [4].

3.2. Electricity power and heat (cooling) production

Depending on demand at location production of electricity via ST can be coupled with production of useful heat or cooling energy. In the case of heat load cogeneration cycles are employed whereby exhaust steam from the ST is supplied to consumer where it is fully utilized. That is accomplished by employing a backpressure steam turbine (or alternatively extraction condensing steam turbine) offering overall thermal efficiencies of up to 90 % [12]. Furthermore, in case of a cooling demand a trigeneration system is applied. It commonly comprises a cogeneration system equipped with an absorption chiller producing the cooling energy from the LP steam which is used for process or space cooling.

3.3. Natural gas compressing by means of additional compressor

In this case the steam turbine is directly coupled to the pipeline compressor. Thereof it enables the pipeline compressor station to pump higher gas quantity than it has been designed for. Such configuration is used in the case of unavailability of electric grid followed by increased demand of natural gas. The disadvantage of such configuration in comparison with gas turbine driven stations is a longer startup time of steam turbine and slower respond to load oscillations. Some steam turbine manufacturers have developed for this purpose a special system of load following via injection of variable quantities of feed water into superheated steam. Through injection of feed water the fresh steam temperature decreases resulting in a drop of inlet enthalpy of steam. Thus available enthalpy drop decreases resulting in lower ST power. Should the power be increased the procedure goes in the opposite direction. Since portion of the injected water into the superheated steam can in such processes be higher than in conventional steam cycle or combined gas and steam cycle, particular attention has to be paid to allowable water content in the last ST stages.

3.4. Natural gas compressing by means of electromotor

In this configuration produced electricity by AC generator is used for running the electromotor which drives the pipeline compressor. Such configuration is used in the case of unavailability of electrical grid. Further argument for this configuration might be the fact that considered electromotor has preferable characteristic of part load following than a steam turbine system in the previous case. The disadvantage of this system is increased cost due to the fact that there it comprises four components: steam turbine, generator, electromotor and pipeline compressor.

3.5. Electricity production via organic Rankine cycle (ORC)

LDC of compressor station shown in Figure 3 shows that during certain periods its part load is quite low i.e. 12 % and less. However flue gas temperatures at the lowest considered operating regime is 380 °C. Compressor stations in which the part load is quite low and appears over a longer time period exhaust gas temperatures can be below 300 °C. If that is followed by the fact that the gas turbine efficiency at part load decreases significantly exhaust gas temperatures below 250 °C can be met. In such cases it makes sense to consider installation of ORC (organic Rankine cycle) which uses Ammonia (or isopentane or silicon oil) as a working medium instead of water. Ammonia has lower evaporating temperature than water, making this cycle favorable in the case of lower temperatures in a topping cycle. Practical applications have shown that usage of ORC cycle in waste heat utilization from pipeline compressor stations is viable if the exhaust gas temperatures are between 200-250 °C. Should the temperatures be higher the preference belongs to a Rankine cycle [8].

3.6. Hydrogen production

Steam produced in HRSG can be used in different hydrogen production processes like methanol or natural gas steam reforming processes, carbon monoxide processes or Kvaerner carbon black & hydrogen processes [13]. Thereby production of hydrogen can be coupled with electricity production into electric power generation and hydrogen production combination plant as it was patented recently [14].

From the technologies mentioned production of electricity is chosen. Firstly, economic revenue of such a system can be assessed quite easier than in the case of other technologies. Furthermore, the idea is to find an optimization method which takes into account the electricity sellback price.

Best available and most acceptable technology for electricity production nowadays with respect to environmental aspects is combined gas and steam turbine process or commonly combined cycle (CC) [15,16].

Compared with separate gas or steam plants their main advantages are:

- thermodynamic efficiency,
- flexibility,
- lower cooling water equipment cost,
- acceptable influence on environment,
- high reliability and relative short delivery time,
- possibility of application of broad fuel spectrum.

However there are also at least two disadvantages which can mainly be emphasized into:

- lower thermodynamic parameters of a steam process
- inability of a fast startup

For production of electricity from waste heat recovery system it is possible to apply one of the following operating strategies:

- Constant ST power in all regimes,
- Constant ST power in regimes 1-5, ST out of operation in regimes 6-10,
- Maintaining constant flue gas temperature.

A drawback of the first strategy is limited steam production at operating regimes with two gas turbines. A second restriction is quite high firing temperatures reaching 900 °C necessitating employment of water cooled enclosure making the system more expensive. A third restriction is high fresh steam temperature which cannot be held within the limits despite employment of three HPSH and two injection coolers.

In case of a second strategy installation of a flue gas bypass upstream the HRSG is considered. Its function would be to vent flue gases into the atmosphere as soon as their enthalpy falls beneath 35000 kW. Knowing that compressor station demand is quite stochastic, such prediction would place high requirements on control systems. So as to avoid jeopardizing steam production in the steam drum and overheating of tubes consideration may be given to installing a steam bypass upstream of the steam turbine instead. In this case, steam would be produced in all operating regimes; however in case of flue gas enthalpies below 35000 kW steam would be vented into the atmosphere or eventually utilized for district heating or space heating system.

So as to overcome drawbacks of the first two strategies, a system shall be designed and operated in a way that the flue gas temperature of 496 °C is kept constant in all operating regimes by means of supplementary firing.

4. HRSG optimization model

In recent history there was not much attention paid to HRSG and its efficiency since it was understood as another kind of steam boiler [17]. Since at the present time CC power plants enable the growing energy demand with the least fuel consumption to be met, it is of great interest to get greater performances and efficiencies from them. From this reason world manufacturers involved in this sector are working on improving the GT inlet temperatures, which directly increases the CC efficiency. Further efficiency improvement can be achieved by optimizing the HRSG operative parameters so as to maximize the work obtained in the steam cycle. With pre-selected number of pressure levels the operating parameters to be optimized are the pressures and temperature differences between the flue gases and water (steam) known as the pinch point.

There are different methods which can be used for HRSG optimization. In selection of a proper method, the following criteria have been observed:

- Practical and reliable for computations with many iterative runs,
- Considering economizer, evaporator and superheater sections of HRSG in terms of heat transfer and in terms of cost,
- Considering economic aspects of the plant.

A general heat exchanger optimization method known as the "pinch point method" is based on optimizing the min. temperature difference between the two streams without taking into account any economic aspects [18,19].

Another approach is based on the gas side pressure drop. This parameter affects the performance of the gas turbine and the size of the heat recovery unit. The method used for such optimization [20] necessitates detail designing or knowing the detailed HRSG design tube size, arrangement, length and type of finning.

There are number of HRSG optimization methods based on maximization of exergy transfer respectively on second law analysis of HRSG. In [21] a general equation for entropy generation number is introduced.

A method for design and operation of HRSG with minimum irreversibility is described in [22]. It suggests a thermodynamic optimum based on estimation of non-dimensional saturation temperature with a proper evaluation of the number of evaporator transfer units. A disadvantage is not taking into account the superheater. Furthermore it does not take into account either surface cost or economic aspects of the plant (e.g. electricity sellback price).

The approach based on exergetic HRSG optimization striving to maximize the exergy transfer from flue gases to water (steam) circuit is explained in [23]. A drawback of the method is assumption that the total heat transfer area of HRSG is constant. Thereby it does not consider separately surface costs of economizer, evaporator and superheater or their influence on overall cost and efficiency.

A general method for the optimum design of HRSG is explained in [24]. Thereby optimization is organized at two levels: first one obtains the main HRSG operating parameters, while the second involves the detailed design of the component concerning the geometric variables of the heat transfer sections. The second level of the optimization is split in two different steps. The first step aims to minimize the pressure drop for a given heat flow. The second step leads to a reduction of the overall dimensions, maintaining the imposed performance of the HRSG in terms of heat flow and pressure drop. The method seems to be complex for computations with many iterative runs.

Broader combined cycle optimization interconnecting a conceptual design, performance, cost and financial analysis is given in [25]. The methodology takes into account several input variables; gas turbine type, fuel type (cost), efficiency, capital cost, return on investment, electricity sellback price followed by emission limits. A limitation of the method does not take into account detailed HRSG optimization. Furthermore it is applicable for projects where complete CC plant including gas turbine is to be proposed.

One recent heat exchanger exergy analysis conducted in [26] presents complete non-dimensional analytical model of calculation of the exergy destruction for evaporator and condenser. It provides a user's diagram from which exergy destruction and effectiveness for the given work-points of the evaporator or the condenser can be directly read.

The methodology used in this work is based on minimizing the objective function comprising both the thermodynamic and thermoeconomic component [27,28]. Thereby the thermodynamic component tends to minimize the exergy losses and thermoeconomic component tends to minimize the total HRSG cost by introducing the reduction to a common monetary base of the costs of exergy losses and of HRSG surface costs. A final solution expresses compromise between the two criteria.

Generally, exergy is defined as a maximal work that can be done from internal energy of a substance with regard to the state of the environment. By considering HRSG as an open system, its exergy flow balance can be expressed per (1).

$$E_{\text{gas,in}} + \sum_{\text{in}} E_{\text{w,in}} = E_{\text{gas,out}} + \sum_{\text{out}} E_{\text{w, out}} + E_Q + I \;. \qquad (1)$$

The term $\sum\limits_{\text{in}} E_{\text{w,in}}$ is negligible since the inlet temperature of water is similar to environmental. The term $E_{\text{gas,out}}$ is negligible since the exhaust gas leaves the HRSG and is no longer used. Furthermore the exergy losses E_{Q} related to the heat exchange with the environment are considered negligible. Approach of neglecting the terms $\sum\limits_{\text{in}} E_{\text{w,in}}$, $E_{\text{gas,out}}$ and E_{Q} is overtaken from [27,28]; however it does not seem to be very realistic since selected feed water inlet temperature of 60 °C is quite above environmental temperature. The temperature of the exhaust gas is also expected to be quite higher than the environmental temperature. Therefore, an alternative approach considering these terms as part of the objective function is also examined. By rearranging equation (1) the exergy loss can be expressed in dimensionless terms:

$$I^* = \frac{I}{M_{\text{g}} c_{\text{pa}} T_{\text{a}}} = \frac{E_{\text{gas,in}} - \sum\limits_{\text{out}} E_{\text{w, out}}}{M_{\text{g}} c_{\text{pa}} T_{\text{a}}} . \tag{2}$$

Wherein E means exergy, M_{g} flue gas mass flow, c_{pa} specific heat of flue gases at environmental temperature T_{a}. Exergy of inlet gas respectively outlet steam is calculated per (3).

$$E = H_2 - H_1 - T_1 (S_2 - S_1) . \tag{3}$$

Wherein H_2 and S_2 mean enthalpy and entropy of inlet gas respectively outlet steam and H_1, T_1 and S_1 mean the enthalpy, temperature and entropy at the state of environment. Enthalpy and entropy of each medium are calculated as a product of its mass flow and specific enthalpy respectively specific entropy according (4).

$$H = \dot{m} \cdot h ; S = \dot{m} \cdot s . \tag{4}$$

Thereof calculation of enthalpy and entropy difference reduced to calculation of specific enthalpy respectively specific entropy difference. Thereby specific enthalpy and specific entropy of outlet steam respectively inlet feed water is calculated as a function of respective pressure and temperature [29]. Specific entropy difference of the flue gases is calculated per (5) assuming flue gases as an ideal gas.

$$s_2 - s_1 = c_{\text{p}} \ln \frac{T_2}{T_1} - R \ln \frac{p_2}{p_1} . \tag{5}$$

Specific enthalpy of flue gases is calculated for the respective temperature from the h-t polynomial (6) which is constructed from exhaust gas composition and specific enthalpies of constituent gases.

$$h = 1.05 \cdot 10^{-13} t^5 - 3.54 \cdot 10^{-10} t^4 + 4.03 \cdot 10^{-7} t^3$$
$$-6.09 \cdot 10^{-5} t^2 + 1,05 t - 1.3217 \cdot 10^{-2} \tag{6}$$

Cost of exergy losses is expressed per (7).

$$K_{\text{I}} = k_{\text{I}} M_{\text{g}} c_{\text{pa}} T_{\text{a}} H I^* . \tag{7}$$

In equation (7) k_{I} represents the specific cost of exergy losses, which are considered equal to an average value of the selling price of the electrical energy, and H is the functioning duration of the plant. HRSG surface cost can be expressed as the sum of the costs of various sections; economizers, evaporators and superheaters.

$$K_{\text{HRSG}} = \left(\sum\limits_{\text{eco}} C_{\text{eco}} + \sum\limits_{\text{eva}} C_{\text{eva}} + \sum\limits_{\text{sh}} C_{\text{sh}} \right) \cdot f_{\text{c}} . \tag{8}$$

Therein is the cost of each section calculated from (9).

$$C_{\text{eco}} = c_{\text{eco}} A_{\text{eco}}^{\text{exp}} ; C_{\text{eva}} = c_{\text{eva}} A_{\text{eva}}^{\text{exp}} ; C_{\text{sh}} = c_{\text{sh}} A_{\text{sh}}^{\text{exp}} . \tag{9}$$

In equation (9) *exp* means a surface exponent used for assigning proper weighting factor to the thermoeconomic component of the objective function. Total annualized cost of the HRSG to be minimized is expressed per (10).

$$K_{\text{total}} = K_{\text{I}} + K_{\text{HRSG}} , \tag{10}$$

$$........ = k_{\text{I}} M_{\text{g}} c_{\text{pa}} T_{\text{a}} H I^* + \frac{1}{D} \left(\sum\limits_{\text{eco}} C_{\text{eco}} + \sum\limits_{\text{eva}} C_{\text{eva}} + \sum\limits_{\text{sh}} C_{\text{sh}} \right) \cdot f_{\text{c}} .$$

Where D means the economic life of the plant. Input data to be used for calculation are given in Table 1 [27,28].

Table 1. Input data for exergo-economic HRSG optimization
Tablica 1. Ulazni podaci za eksergo-ekonomsku optimizaciju generatora pare

Variable / Varijabla	Value / Vrijednost	Unit / Jedinica	Reference / Literatura
D	20	years / godine	[27]
H	7884	hours / sati	[8]
k_{I}	0.03	\$/kWh	[30]
c_{hpsh}	382.5	\$/m²	[31]
c_{hpeva}	66	\$/m²	[31]
$c_{\text{hpeco}}, c_{\text{ipeco}}, c_{\text{lpeco}}$	39	\$/m²	[31]
$c_{\text{ipsh}}, c_{\text{lpsh}}$	75	\$/m²	[31]
$c_{\text{ipeva}}, c_{\text{lpeva}}$	69	\$/m²	[31]
f_{c} (1P system)	1	-	-
f_{c} (2P system)	1.1	-	-
f_{c} (3P system)	1.2	-	-

The objective function given in (10) is solved by means of Nelder-Mead method [13]. The method is also known as downhill simplex method or amoeba method and is a numerical method for minimizing an objective function in a many-dimensional space. It uses the concept of a simplex, which is a polytope of N+1 vertices in N

dimensions. Examples of simplexes include a line segment on a line, a triangle on a plane, a tetrahedron in three-dimensional space and so on. The method approximately finds a locally optimal solution to a problem with N variables when the objective function varies smoothly. For proper convergence and determination of independent variables by means of Nelder-Mead method it is desirable to approximately define a local minimum. Actually that means that order of magnitude of parameters being optimized; pressure and pinch point have to be compared with relevant literature so as to verify results. A better possibility is to verify the results with another program routine. In this work optimization is conducted by means of the Nelder-Mead optimization method using Matlab whereupon the optimization results are verified by means of GateCycle respectively its subprogram CycleLink.

5. System configurations

Optimal CC configuration in terms of the lowest total annualized HRSG cost is determined. So as to simplify the optimization procedure number of pressure levels and arrangement of heat exchangers in HRSG are predefined. Thereof seven configurations are proposed with main features listed in Table 2 followed by flow schemes given in Figure 5 to Figure 11.

Table 2. Configuration features
Tablica 2. Karakteristike konfiguracija

Configuration / Konfiguracija	Main features / Glavne značajke
1P	Single pressure / Jednotlačna konfiguracija
2P1	HPECO arranged after LPECO / Visokotlačni ekonomajzer smješten poslije niskotlačnog ekonomajzera
2P2	HPECO arranged after HPEVA / Visokotlačni ekonomajzer smješten poslije visokotlačnog isparivača
2P3	HPECO1 arranged after HPEVA/ Prvi visokotlačni ekonomajzer smješten poslije visokotlačnog isparivača HPECO2 arranged after LPECO / Drugi visokotlačni ekonomajzer smješten poslije niskotlačnog ekonomajzera
3P1	One HPECO arranged after HPEVA / Jedan visokotlačni ekonomajzer smješten poslije visokotlačnog isparivača
3P2	HPECO1 arranged after HPEVA / Prvi visokotlačni ekonomajzer smješten poslije visokotlačnog isparivača HPECO2 arranged after LPECO / Drugi visokotlačni ekonomajzer smješten poslije niskotlačnog ekonomajzera
3P3	Triple pressure, three HP economizers / Trotlačna konfiguracija, 3 visokotlačna ekonomajzera

All configurations are characterized by the following common features:

- condensate degassing with extracted LP steam from the steam turbine
- two high pressure superheaters (HPSH) and one intermediate injection cooler
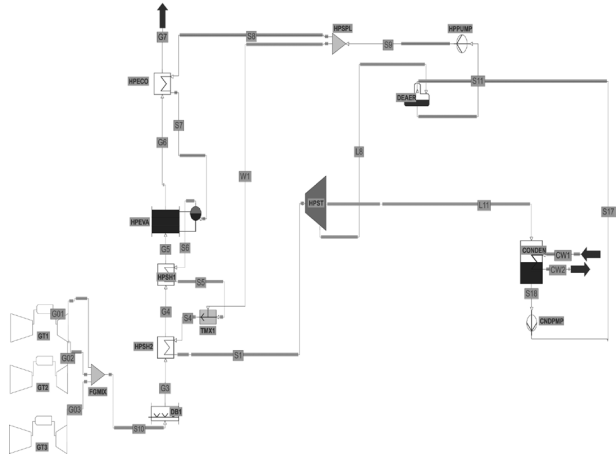- natural circulation drum

Figure 5. Flow scheme 1P configuration
Slika 5. Shema 1P konfiguracije

Basic configuration 1P shown in Figure 5 is equipped with a single pressure HRSG and uses a common way of degassing the feed water by means of extracted low quality steam from the steam turbine. Selected ST operating mode is fix pressure. ST is equipped with one unregulated extraction for condensate deaeration. Since considered fuel is natural gas containing none or very small amount of sulphur selected feed water temperature at HRSG inlet is 60 °C. Saturation pressure at this temperature appearing in deaerator is 0.2 bar at which pressure is the steam is extracted from the ST via unregulated extraction. Extracted steam leaves the ST and enters the condenser at condensing pressure of 0.1 bar. Since the obtained fresh steam pressure in all configurations is not above 100 bar natural circulation [32] drum is assumed. As to design of evaporators and economizers, finned tubes are used in order to improve the heat transfer from the flue gases to water respectively steam through adding the additional layer called finning (or extended surfaces) which typically reduces the HRSG size [8, 17]. Finning materials need to have high conductivity. Fin shape and pitch have to be selected properly to prevent excessive pressure drops in HRSG, respectively too high gas side backpressure to the gas turbine [17]. Superheater tubes are to be designed without any finning and are called bare tubes, because of the possibility of oxidation of the finning [8]. In terms of flow direction, all heat exchangers are designed as counter flow heat exchangers. Since gas turbines operate over a quite wide load range, it is the function of supplementary firing to maintain constant

pressure of fresh steam independently on enthalpy of flue gases from the gas turbines.
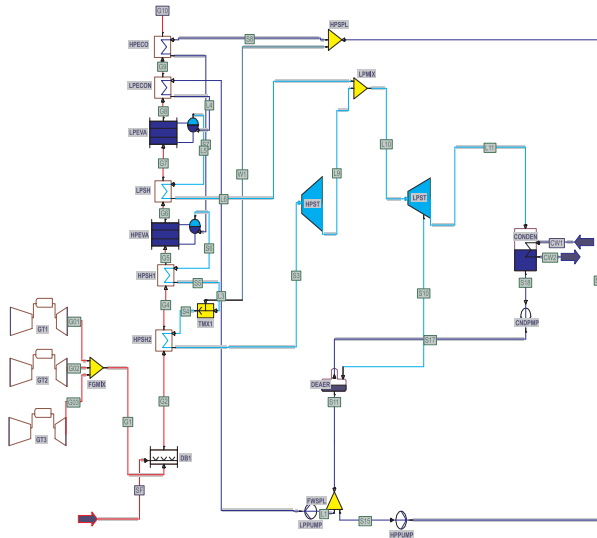

Figure 6. Flow scheme 2P1 configuration
Slika 6. Shema 2P1 konfiguracije

2P1 configuration is shown in Figure 6. In the direction of flue gas path, it comprises two HPSH, HPEVA followed by LP loop consisting of LPSH, LPEVA and LPECO. HPECO is arranged after the LP loop at the end of the flue gas path. Low pressure pinch point (LPPP), high pressure pinch point (HPPP) and fresh steam pressures are optimized. The ST is equipped with one admission for LP steam and one uncontrolled extraction for deaerator steam. Analogically as in the case of 1P configuration extraction pressure is 0.2 bar, which is also the operating pressure of the deaerator.
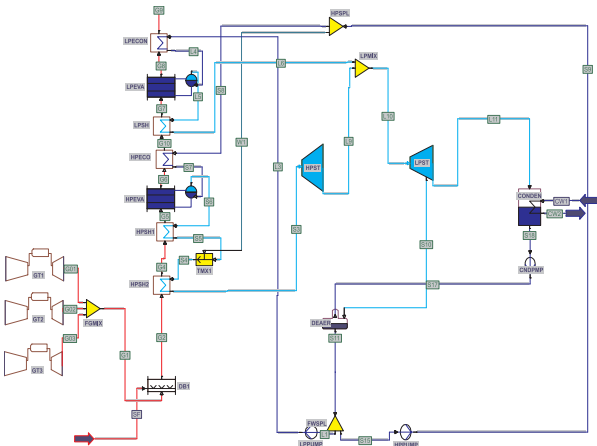
Figure 7. Flow scheme 2P2 configuration
Slika 7. Shema 2P2 konfiguracije

2P2 configuration shown in Figure 7 is similar to configuration 2P1 with HPECO arranged after the HPEVA in the flue gas path so as to provide lower approach point

at HPEVA inlet. Thereof preheating of feed water to a saturation temperature is done via an economizer unlike configuration 2P1 where a great portion of heat required for preheating the feed water is transferred by means of an evaporator.

Configuration 2P3 shown in Figure 8 is similar to 2P2 configuration with the addition of a second HP economizer installed after the LPECO in the flue gas path. The intention is further to increase heat recovery efficiency through closing the water-steam temperature profile to the temperature profile of the flue gases.
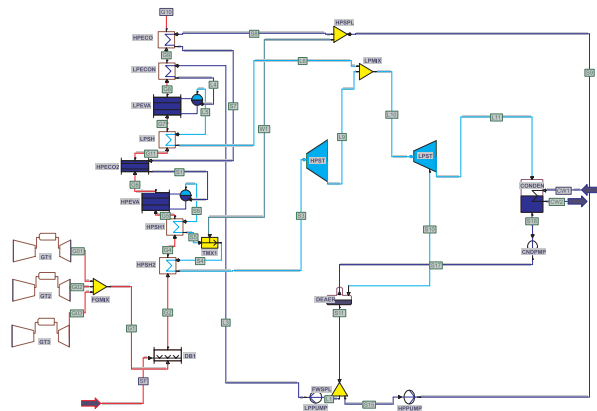

Figure 8. Flow scheme 2P3 configuration
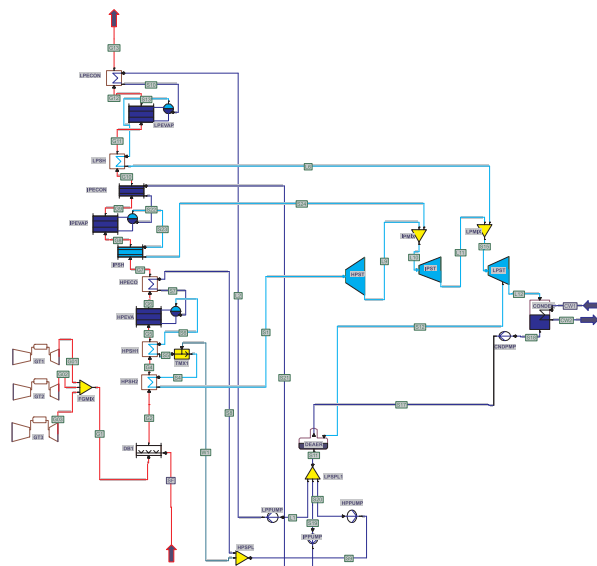**Slika 8.** Shema 2P3 konfiguracije

Figure 9. Flow scheme 3P1 configuration
Slika 9. Shema 3P1 konfiguracije

3P1 configuration is shown in Figure 9. Additional to 2P1 configuration here is an IP loop installed between the LP and HP loop comprising IP economizer, IP evaporator and IP superheater. HP/IP/LP fresh stem pressures and pinch points are optimized. The ST is of single casing design and is equipped with two admissions; one for IP

steam comprising one stop valve and two control valves and one for LP steam comprising one stop valve and two control valves. Furthermore the ST is equipped with one uncontrolled extraction for deaerator steam. Analogically as in the case of 2P1 configuration extraction pressure is 0.2bar which is also the operating pressure of the deaerator.

Configuration 3P2 shown in Figure 10 is similar to 3P1 configuration with the addition of the second HP economizer installed after LPECO in the flue gas path. The intention is to further increase heat recovery efficiency through closing the water-steam temperature profile to the temperature profile of the flue gases.
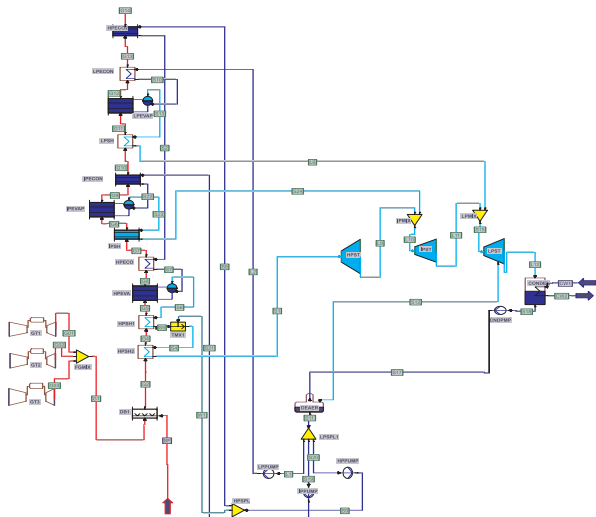

Figure 10. Flow scheme 3P2 configuration
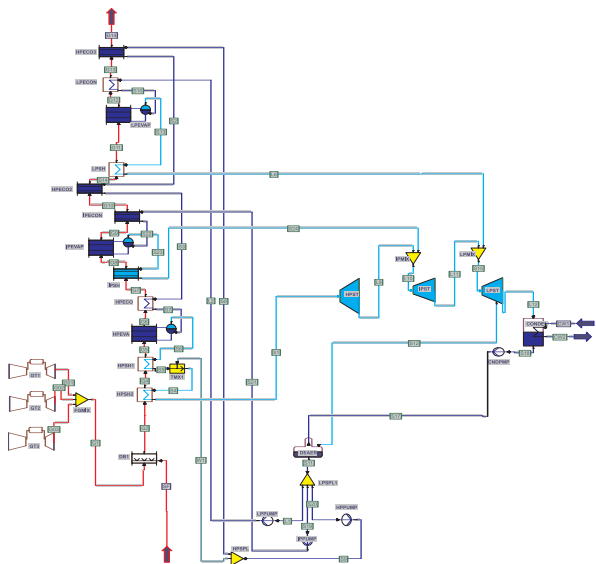**Slika 10.** Shema 3P2 konfiguracije

Figure 11. Flow scheme 3P3 configuration
Slika 11. Shema 3P3 konfiguracije

Configuration 3P3 shown in Figure 11 is similar to 3P2 configuration with the addition of the third HP

economizer installed between IPECO and LPSH in the flue gas path. The intention is to further increase heat recovery efficiency through closing the water-steam temperature profile to the temperature profile of the flue gases.

6. Results and discussion

Upon solving the optimization algorithm with assumed surface costs values and assigning value of 1 to the surface exponent (*exp*) objective function for 1P configuration shown in Figure 12 is obtained.
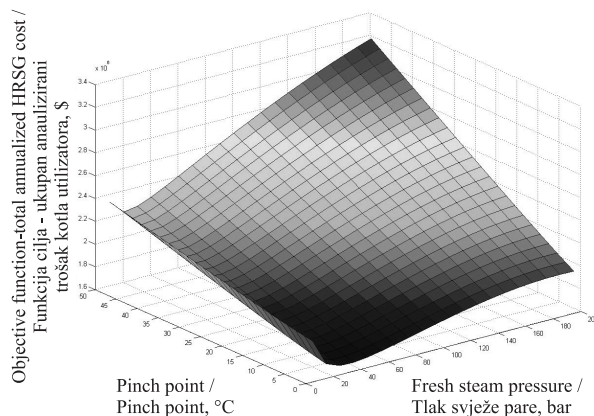

Figure 12. Objective function for initial assumptions
Slika 12. Funkcija cilja za početne pretpostavke
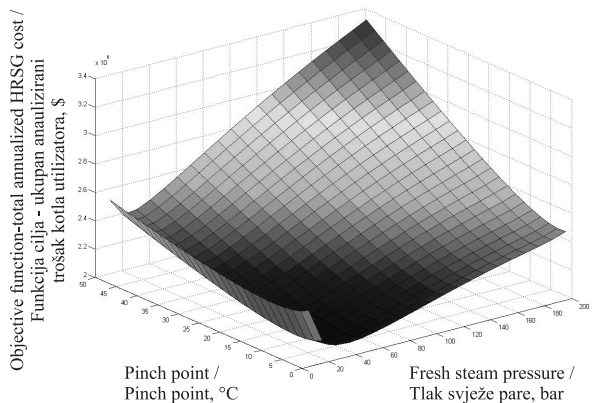

Figure 13. Objective function upon introducing surface exponent
Slika 13. Funkcija cilja sa uvedenim eksponentom površine

Apparently objective function reaches its minimum at fresh steam pressure value around 30 bar and pinch point of 0 °C. However it is not possible to design a steam generator with zero pinch point that would necessitate infinite large surfaces. Furthermore, according to the second law of thermodynamics for heat transfer between two media, the existence of finite temperature difference is essential. Pinch point should also not reach too low values due to practical limitations. Too large surface areas for the respective HRSG capacity cause an increase

of the tube package size increasing thereof the size and cost of the passive HRSG part. An increase of HRSG overall dimensions lead also to an increase in storage and transportation cost. Since this impact cannot be mathematically described and calculated it makes sense to limit the lower value of the pinch point. Therefore, a surface exponent with a value of 1.18 is introduced, which provides pinch point of 5 °C. This value of pinch point is actually minimal feasible pinch point for unfired HRSG obtained from [31]. With this value of a surface exponent, the objective function is given the shape as shown in Figure 13.

Apparently, the objective function for modified assumptions reaches its minimum at fresh steam pressure value of 30.6 bar and pinch point of 4.98 °C. Obtained surface exponent value of 1.18 is further used for optimization of other configurations. Optimization progress proves that the thermodynamic component has an order of magnitude of 10^6 whilst the thermoeconomic component has an order of magnitude of 10^5. Thereof it may be concluded that the cost of exergy losses has greater influence on optimization results than the cost of HRSG surfaces. Comparison of minimal values of objective function and its components for all configurations is given in Figure 14.
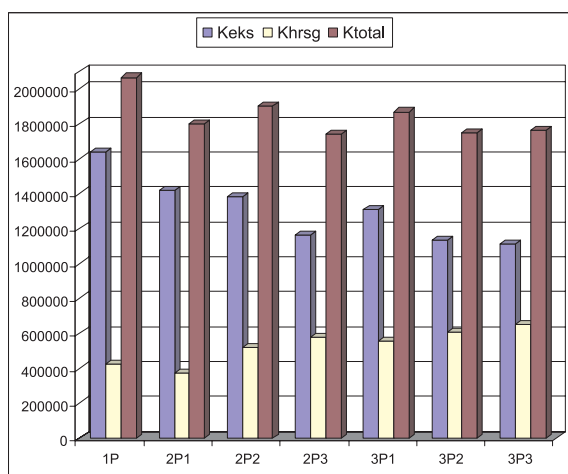
A falling trend of the thermodynamic component and rising trend of the thermoeconomic component is notable. It is also apparent that the objective function reaches minimal value in case of configuration 2P3.

From Table 3 it is apparent that the configuration 3P3 offers the highest CC efficiency, steam cycle efficiency and heat recovery efficiency. Configuration 2P3 provides the lowest total annualized HRSG cost. Calculation results prove that configuration with the lowest total annualized HRSG cost does not obtain the highest thermodynamic efficiency (with respect to optimized pressure and pinch point values).

From Figure 15 it is apparent that each configuration provides further increase of net CC efficiency with the exception of 3P1 compared with 2P3 configuration. Furthermore heat recovery efficiency shown in Table 3 does not follow the values of CC efficiency due to the fact that it mostly depends on the flue gas temperature at HRSG exit.
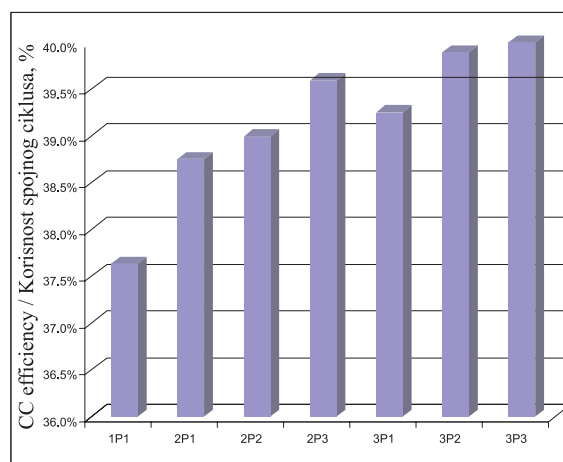

Figure 14. Value of optimal objective function and its components
Slika 14. Vrijednost optimalne funkcije cilja i njene komponente

Figure 15. CC net efficiency
Slika 15. Korisnost spojnog ciklusa

Table 3. Comparison of optimization results for all configurations
Tablica 3. Usporedba rezultata optimizacije za sve konfiguracije

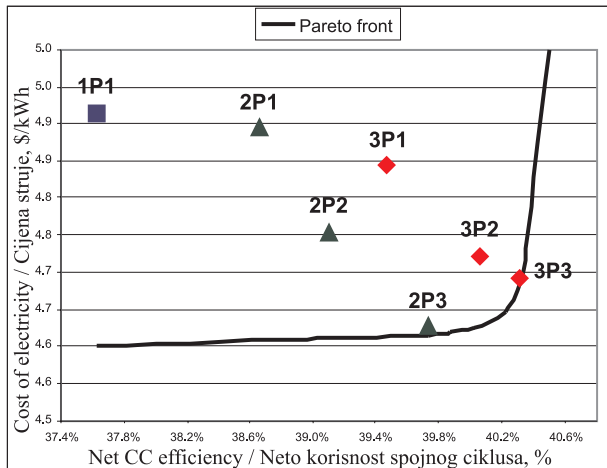| Configuration / Konfiguracija	1P	2P1	2P2	2P3	3P1	3P2	3P3
Steam turbine power / Snaga parne turbine, kW	10989	11820	12077	12514	12253	12726	12844
Net CC efficiency / Neto korisnost spojnog ciklusa, %	37.6	38.8	39.0	39.6	39.3	39.9	**40.0**
Steam cycle efficiency / Korisnost parnog ciklusa, %	19.2	20.7	21.1	21.9	21.4	22.3	22.5
Heat recovery efficiency / Korisnost procesa za iskorištavanje otpadne topline, %	71.4	79.6	76.6	80.4	79.7	82.9	**83.3**
HRSG surface cost / Cijena izmjenjivačkih površina kotla utilizatora, $	1955398	2427076	2366647	2352113	2362172	2463025	2571792
$K_{total, min}$, $	2066746	1802987	1899096	**1740566**	1869441	1746602	1763105

Figure 16. Cost of electricity as a function of net CC efficiency

Slika 16. Cijena električne energije u ovisnosti o stupnju korisnosti spojnog ciklusa

Figure 16 shows dependency of cost of electricity on net CC efficiency for all configurations. So as to compare configurations based on criteria of maximal efficiency and the lowest cost of electricity, a Pareto front is used. It is constructed by approximation of a polynomial which passes through the points with maximal efficiency and minimal cost of electricity. Furthermore the polynomial is extended in both *x* and *y* directions so as to illustrate vicinity of other configurations to the Pareto front [13, 33]. Apparently 2P3 and 3P3 configurations are the closest to the Pareto front, whereby the first is characterized by lower cost of electricity and the latter by higher system efficiency.

7. Conclusion

In this work an exergo-economic HRSG optimization based on minimization of the objective function comprising the HRSG cost of exergy losses and HRSG surface cost is conducted. By means of this method considered configurations are optimized for finding HRSG operative parameters providing the lowest total annualized HRSG cost. Employed optimization method [27,28] and obtained results manifest the meaning of the thermoeconomic optimization whereby more expensive HRSG surfaces cause less heat transfer from flue gases to water (steam) respectively an increase of pinch point and decrease of fresh steam pressure followed by smaller surface size. However initial optimization results show insensibility of the objective function on the cost of surfaces by resulting in zero pinch point. Such result demonstrates low surface costs compared to the cost of exergy losses for given parameters of available flue gases and assumed factors of the objective function. Incorporation of exergy of inlet feed water and outlet

gas (which are neglected by author of the selected optimization method) result in max. pinch point and high values of pressure close to critical one. Such result of pressure value is, however, not feasible for design of the industrial steam turbine. Furthermore, a tendency of reaching an upper limit of pinch point shows inadequacy of this approach. So as to obtain positive but realistic pinch point values a surface exponent is introduced into the equation for calculating the HRSG surface cost. The idea is to find the value of the surface exponent obtaining minimal feasible pinch point provided by the heat recovery steam generator manufacturer. Referent minimal feasible pinch point is selected for unfired case. The proper value of the surface exponent is determined on the basic single pressure configuration. Upon defining optimization criteria this value is further used for optimization of other configurations. Calculation results show that the configuration 3P3 offers the highest CC efficiency whilst the configuration 2P3 offers the lowest total annualized HRSG cost. Comparison of configurations is based on a diagram presenting dependency of electricity cost on net system efficiency. The pareto front with respect to both criteria; maximal system efficiency and minimal cost of electricity is constructed. Thereof 2P3 and 3P3 configurations are the closest to the Pareto front whereby the first is characterized by the lowest cost of electricity and the latter by the highest system efficiency. Economic results show that all configurations are viable whereby the preference in terms of viability belongs to configuration 2P3. Based on results presented in this work guidelines for efficient waste heat utilization from pipeline compressor stations might be emphasized into:

• Focus should be on compressor stations providing considerable enthalpy of available flue gases,

• Annual demand and operating mode should be such that gas turbines in operation do not run below 50 % load,

• Preference should be given to sites where a significant demand for process heating, district heating or district cooling exists.

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