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The Possibility of Energy Generation within the Conventional Natural Gas Transport System

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The paper presents the possibility of electricity generation at the pressure reduction stations (PRS) in the conventional system of natural gas transmission and distribution. The main deficiency of the conventional system is its thermodynamic irreversibility, i.e. the loss of potential work that could be generated while reducing the gas pressure. Within the first part of work, a thermodynamic model was created to analyze changes in energy and exergy within the gas transport system. The second part of work deals with the application of turboexpanders (expansion turbines) which may be used to generate electricity while reducing the gas pressure. A thermodynamic model of the turboexpander system was created and the relevant thermodynamic indicators were defined. Case studies for two different PRS were performed to analyze the thermodynamic quality and the economic feasibility of turboexpander application for varying design parameters, i.e. the gas preheating rate and the expander isentropic efficiency.

Mogućnost proizvodnje električne energije u konvencionalnom sustavu transporta prirodnog plina

Izvornoznanstveni članak

Ovaj članak opisuje mogućnost proizvodnje električne energije u mjerno-redukcijskim stanicama (MRS), koje čine dio prijenosnog i distribucijskog plinskog sustava. Glavni nedostatak konvencionalnog sustava je njegova termodinamička nepovrativost, to jest gubitak rada koji bi mogao biti generiran pri redukciji tlaka plina. U prvom je dijelu ovog rada bio izrađen model za analizu promjena energije i eksergije u sustavu transporta plina. U drugom se dijelu rada razmatra upotreba turboekspandera (turbina) za proizvodnju električne energije pri redukciji tlaka plina. Izrađen je termodinamički model sustava s turboekspanderom i definirani su odgovarajući termodinamički pokazatelji.Razrađeni su primjeri za dvije različite MRS, u kojima je analizirana termodinamička kvaliteta i ekonomska opravdanost projekata turbine pri različitim projektnim parametrima, to jest stupanj predgrijavanja plina i iskoristivost turbine.

Keywords
Exergy
Expansion turbine
Natural gas
Second law analysis
Transmission and distribution

Ključne riječi
Analiza prema drugom zakonu termodinamike
Eksergija
Turboekspander (turbina)
Prirodni plin
Prijenos i distribucija

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1. Introduction

Transport of any goods requires infrastructure and energy supply. Transport of gases is more feasible and less energy-consuming if the gas is pressurized or liquefied in order to reduce its specific volume. Nowadays, the transport of natural gas is based on two main methods: Liquefied Natural Gas (LNG) and pipeline transport under high pressure. The LNG method applies to overseas gas transport as well as to inland transport in areas with no pipeline network coverage, while the conventional pipeline transport prevails in inland areas, with some offshore applications as well. Whatever be the high-distance transport method, the final supply of natural gas to consumers has to be conducted using local distribution pipelines.

Energy consumption in both transport methods may be divided into a part bound to the volume reduction and a part required to cover the distance. Theoretically, it should be possible to recover the energy consumed for volume reduction as the gas reaches its destination area. However, the current technologies applied for natural gas transport fail to utilize this latent energy.

The aim of this paper is to show the energy recovery potential within the conventional gas transport system.

This potential, related to pressure, may conveniently be described using the concept of exergy, i.e. the maximum work obtainable from a system with respect to the environmental conditions [13]. It should be stressed that a significant part of this exergy comes from nature as gas leaves the wells under high pressure. The residual part of the pipeline gas exergy results from mechanical work to be used in compressor stations.

As the gas reaches the consumption area, its pressure has to be reduced before supplying to consumers. At present, this is done in pressure regulators which destroy the work potential (exergy) bound to pressure.

Symbols/Oznake

a	- Helmholtz free energy, kJ·kmol^{-1} - Helmholtzova slobodna energija
b	- specific exergy, kJ·kmol^{-1} - specifična eksergija
CF	- cash flow, EUR - tok novca
$\dot{E}_{ch}$	- chemical energy flux, kJ·s^{-1} - tok kemijske energije
(g)	- denotes gauge pressure, Pa (g) - označuje pretlak
h	- specific enthalpy, kJ·kmol^{-1} - specifična entalpija
J_0	- investment cost, EUR - investicijski trošak
LHV	- lower heating value, kJ kmol–1 - donja ogrjevna vrijednost
$\dot{n}$	- molar flow rate, kmol s^{-1} - molarni protok
NPV	- net present value, EUR - neto sadašnja vrijednost
s	- specific entropy, kJ kmol^{-1} K^{-1} - specifična entropija
p	- pressure, Pa - tlak
q	- exchanged heat, kJ kmol^{-1} - specifična izmijenjena toplina
r	- discount rate - diskontna stopa
R	- universal gas constant, kJ kmol^{-1} K^{-1} - opća plinska konstanta
t	- operation time, years - vrijeme rada sustava, godina
T	- temperature, K - temperatura
w	- specific shaft work, kJ kmol^{-1} - specifični rad na vratilu
$\dot{W}$	- shaft power, kJ s^{-1} - snaga na vratilu
x	- mole fraction - molarni udio
α	- reduced Hemholtz energy - reducirana Helmholtzova energija
δ_b	- reduced density - reducirana gustoća
ε	- work-to-fuel ratio - omjer rad-gorivo
η	- energy efficiency - energetska iskoristivost
η_{II}	- second-law efficiency - eksergetska iskoristivost
ϑ	- temperature, ºC - temperatura
ν	- ratio of molar flow rates - omjer molarnih protoka
ρ	- molar density, kmol m^{-3} - molarna gustoća
τ	- inverse reduced temperature - recipročna reducirana temperatura

Indices / Indeksi

ch	- chemical - kemijski
ph	- physical - fizički
1 ... 9	- number of the reference section - broj referentnog presjeka
A	- air - zrak
F	- flue gas - dimni plin
G	- natural gas - prirodni plin

Utilization of the gas exergy is possible using expansion turbines, i.e. turboexpanders. The use of expansion turbines within the natural gas transport system was analyzed by Poživil [10] and Kostowski et al. [7]. Further related work concerns the application of expansion turbines for the transport of other gases (coke oven gas, [4]) or for other engineering areas (e.g. [17, 6]). Turboexpanders are also widely applied in cryogenic gas engineering [2].

The scope of this paper is limited to exergy recovery within the conventional (pipeline) transport of natural gas. The exergy potential available within the LNG transport chain was analyzed by Kaneko et al, 2004 [5] as well as by Szargut and Szczygiel, 2008 [14]).

2. Thermodynamic analysis of the gas transmission and distribution system

2.1. Technical description

A conventional pipeline transport system has a hierarchical structure, based on 3 or 4 pressure stages, interconnected via Pressure Reduction Stations (PRS). In most countries, the high-pressure stage is referred to as the transmission sector, while lower stages constitute the distribution sector. In Europe, this division is regulated by the Directive No. 2003/55/EC [18], which lays down a requirement to split the transmission and the distribution sector between different companies in order to clarify the transport costs in both sectors and to support market liberalization. However, national regulations still differ in defining the limits between both sectors and between the particular pressure stages.

The function of the high-pressure stage is to transport gas to long distances ranging between 10^2 and 10^4 km under pressure between 1.6 and 10 MPa. Due to the long distances, energy has to be supplied to compressor stations in order to compensate for friction losses in pipelines. The compression cost is a function of the transport distance, the gas flow rate and of the applied pressure level and pipe diameter. The energy consumption in compressor stations corresponds to 3–5 % chemical energy of the transported gas [16] and constitutes between 25 and 50 % of the total transmission cost [11].

The transmission sector supplies gas to local distribution systems and to the large-scale industrial consumers. Within a distribution system, following the gas flow to subsequent lower pressure stages, its pressure is reduced in pressure regulators in order to set the required downstream pressure levels. The total length of a distribution system, measured as a path of gas from the first-stage PRS to the final consumer, ranges between 10^0 and 10^2 km. Due to the proximity of the high-pressure system, no further energy supply is required for gas distribution.

2.2. Analysis of a model system

A model system, illustrating the flow of energy and exergy in the gas transmission and distribution system, was chosen, based on parameters typical for the gas infrastructure in Poland, with three pressure levels applied. The system is presented in Figure 1a.

It was assumed that the transport path starts in a high pressure pipeline at a compressor station inlet, with gas pressure p_1 = 3.0 MPa (g) and temperature ϑ_1 = 10 °C. Gas is compressed to p_2 = 4.5 MPa (g) with the isentropic efficiency of η_c = 0.82. At the compressor outlet, gas is cooled to ϑ_3 = 30 °C in order to increase the downstream pipeline flow capacity. Due to friction losses, gas pressure decreases to p_4 = 2.5 MPa (g) along the transmission pipeline, at the same time gas temperature equalizes with soil temperature of $\vartheta_{soil} = \vartheta_4$ = 10 °C.

The transmission system splits in point 4 to supply gas to several local distribution zones via first-stage reduction stations; one process path was chosen for further analysis. Before reducing the pressure to p_6 = 0.40 MPa (g), gas has to be pre-heated in order to compensate for the Joule-Thomson effect and thus avoid low temperatures at the regulator outlet, which could lead to hydrate forming. Gas pre-heating was assumed to the level of ϑ_5 = 20 °C. Within the medium pressure network friction losses decrease the gas pressure to p_7 = 0.25 MPa (g). At this point the system splits again to supply several directions. The second-stage reduction stations set the pressure at a level of p_8 = 2.5 kPa (g), which is suitable for direct use in domestic gas appliances. Within the low-pressure network the gas pressure decreases eventually to p_9 = 1.8 kPa (g). Equalization to the soil temperature was

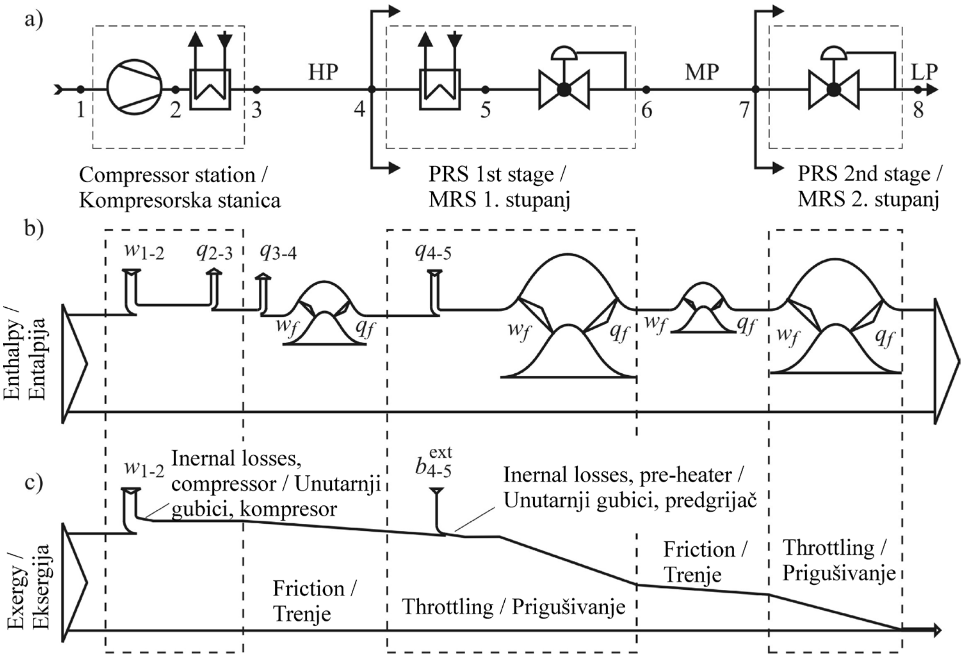

Figure 1. Energy and exergy flow diagram for the gas transmission and distribution system. Abbreviations: HP, MP, LP – high/medium/ low pressure. The low-pressure pipeline ends at point 9 not shown in the diagram.

Slika 1. Dijagram tokova energije i eksergije u prijenosnom i distribucijskom plinskom sustavu. Oznake: HP, MP, LP – visoko-/ srednje-/ niskotlačna plinska mreža. Niskotlačni cjevovod završava u točki 9 (izvan dijagrama).

assumed in points 7 and 9. No preheating is necessary in the second-stage PRS as the pressure drop is less than 0.3 MPa.

The system was investigated using the energy and the exergy analysis. As the flux of gas in the system is not constant, molar-based specific values are used.

2.3. Energy analysis

Assuming steady-state gas flow with negligible changes in the kinetic and the potential energy, the energy conservation equation for an arbitrary section *i–j* of the entire gas transport path 1–9 may be written:

$$q_{i-j} = h_j - h_i + w_{i-j}, \tag{1}$$

where q_{i-j} is the heat transferred to the system along the section, w_{i-j} is the shaft work (positive if extracted and negative if supplied) and h is the gas enthalpy. Note that Eq. (1) does not contain work w_f done against friction forces, as it is entirely converted to friction heat q_f which is assumed to return to the system. The chemical energy of natural gas does not change along the process, hence it is not presented in the balance. However, it should be stressed that its value is of an order of magnitude higher than changes in enthalpy.

Gas enthalpy $h\ (p,\ T,\ \bar{x})$ is a function of its pressure p, temperature T and composition $\bar{x} = \{x_1,\ x_2,...,\ x_n\}$. The composition was chosen according to the NIST2 reference natural gas (Table 1) which is comparable to Russian gas from the Yamal region, covering most of gas demand in Poland. Details on calculating the natural gas enthalpy are given in Appendix A.

Table 1. Composition of the NIST2 reference natural gas

Tablica 1. Sastav referentnog prirodnog plina NIST2

Component / Sastojak	Mole fraction / Molarni udio
methane / metan	0.906720
ethane / etan	0.045279
propane / propan	0.008280
butane / butan	0.001563
isobutane / izobutan	0.001037
pentane / pentan	0.000443
isopentane / izopentan	0.000321
hexane / heksan	0.000393
nitrogen / dušik	0.031284
carbon dioxide / ugljični dioksid	0.004676

Results of the energy analysis are shown in Figure 1b with numerical values in Table 2. In addition to external energy transfer by work and heat, also the friction work w_f converted to friction heat q_f was shown schematically in the energy flow diagram. It should be stressed that the energy balance fails to entirely explain the thermodynamics of the gas transport system, as it assigns the same value to both work and heat. A more informative system evaluation is possible based on the exergy analysis.

2.4. Exergy analysis

As may be concluded from Figure 1b, energy does not represent the ability to produce work. The enthalpy of gas remains approximately constant throughout the process. However, as the gas achieves point 9, it could not be transported any further since it has a very low pressure, and thus it has lost the ability to perform work against the friction forces. A function defining the work potential of a thermodynamic medium with respect to the environment is the physical exergy, defined as:

$$b_{ph} = h - h_0 + T_0(s - s_0), \tag{2}$$

where 0 denotes the state of the analyzed medium under ambient conditions.

Exergy may be supplied or extracted to a system by means of work or heat. Exergy of work equals the work itself. On the contrary, exergy of heat exchanged with a heat source (or sink) depends on its temperature T:

$$b^{in/out} = q\frac{T - T_0}{T}. \tag{3}$$

Heat exchanged with the environment has no exergetic value.

Exergy does not satisfy the conservation law. Therefore, a quasi-balance of exergy has to be closed using the internal exergy loss δb:

$$\sum b^{in} = \Delta b_{system} + \sum b^{out} + \delta b. \tag{4}$$

Table 2. Energy and exergy balance for elementary processes of the gas transport system

Tablica 2. Energetska i eksergetska bilanca elementarnih procesa transporta plina

Process/ Proces	q_{i-j}	w_{i-j}	b_i	b^{ext}_{i-j}	δb
	kJ / kmol				
1–2	0	−1102	8047	+1102	182
2–3	−558	0	8968	0	39
3–4	−422	0	8929	0	1365
4–5	+396	0	7564	+89	89
5–6	0	0	7564	0	3747
6–7	+27	0	3817	0	845
7–8	0	0	2973	0	2904
8–9	+49	0	69	0	17

Transformations of exergy for the analyzed gas transport path are shown in Figure 1c with numerical values in Table 2. As can be seen in the diagram, the physical exergy of gas is almost entirely destructed along the path. The exergy utilization for performing work against friction forces in pipelines is thermodynamically justified if the system is properly designed. However, most of the available work potential is destroyed in a pressure reduction station, which could be replaced by expansion turbines.

3. Thermodynamic analysis of the expansion turbine design parameters

3.1. Technical aspects of expansion turbines

A possible replacement of a pressure regulator station with an expansion turbine should take into account the primary functions of a pressure reduction station in the gas transmission and distribution system, i.e.:

- matching the flow through the regulator to the current demand at the consumers',
- reducing the gas pressure,
- stabilizing the outlet pressure.

It should be stressed that only the second function may be carried out using an expansion turbine. Therefore, a turbine has to be installed in parallel to existing regulating lines of a pressure reduction station. Furthermore, a pressure regulator should be installed downstream of the turbine in order to stabilize the outlet pressure. The reduction line with the turbine has to be equipped with a standard overpressure protection system, e.g. with two slam-shut valves.

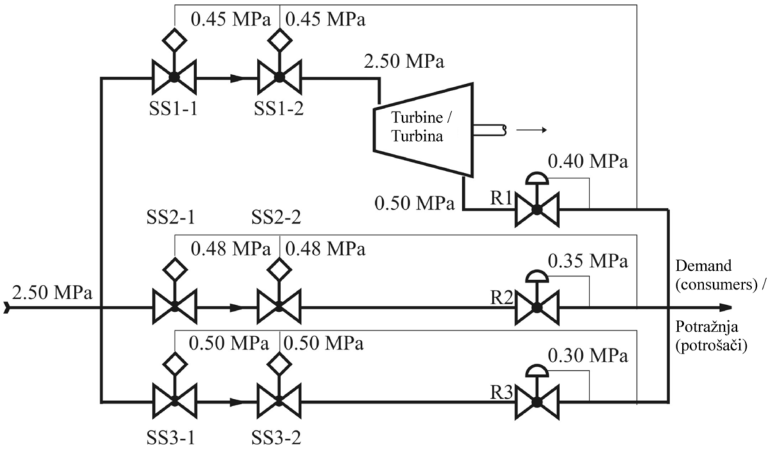

Figure 2. A pressure regulation system with an expansion turbine. R – pressure regulators, SS – slam shut valves

Slika 2. Sustav za redukciju tlaka s turboekspanderom. R – regulator tlaka, SS – sigurnosno-zaporni ventil

Figure 2 shows sample settings for the first-stage reduction station with three regulation lines. Under normal operating conditions, the gas flows through the turbine which reduces its pressure to 0.50 MPa while the regulator R1 provides further reduction and stabilization of the pressure to 0.40 MPa. If the gas flow through the line 1 is insufficient to cover the demand, the pressure in the downstream system decreases, and the regulator R2 opens. Each line is supplied with an overpressure protection system, based on two slam-shut valves, closing the gas flow in case of excess outlet pressure. Line 3 opens if the lines 1 and 2 are closed due to failure or if the line 2 is closed and the flow through line 1 is insufficient.

The slam shut valves in each regulation line control the outlet pressure and close if the set values are exceeded. The turbine is connected to a generator converting the mechanical energy to electricity. In the solutions available in the market (e.g. RMG, General Electric, Gascontrol), the generator is integrated with the turbine in order to maintain the rotating shaft inside the housing as the machine operates in an explosive atmosphere. Both synchronous and asynchronous AC generators are applied with expansion turbines, the former providing less disturbance to the electricity grid. Micro expansion turbines with DC generators are also produced to cover the local electricity demand at the pressure reduction stations.

3.2. Analysis of the model system

The thermoeconomic analysis of an expansion turbine was performed for a model system as shown in Figure 3. The system consists of a turbine with a heat exchanger (HE) serving as a pre-heater. The required heat is produced in a boiler, fired with natural gas taken from point 5 situated downstream of the turbine. The overpressure protection system and the parallel standard regulation lines are not analyzed assuming they have no impact on the system under normal operating conditions.

Two control volumes defining the system are shown in Figure 3. The control volume A is used to compare the generated work with the fuel input to the system, while the volume B may be used to define the overall system efficiency.

The system has to be supplied with two pressure regulators. The main regulator (R1, Figure 3) provides final adjustment to the outlet pressure; to simplify the

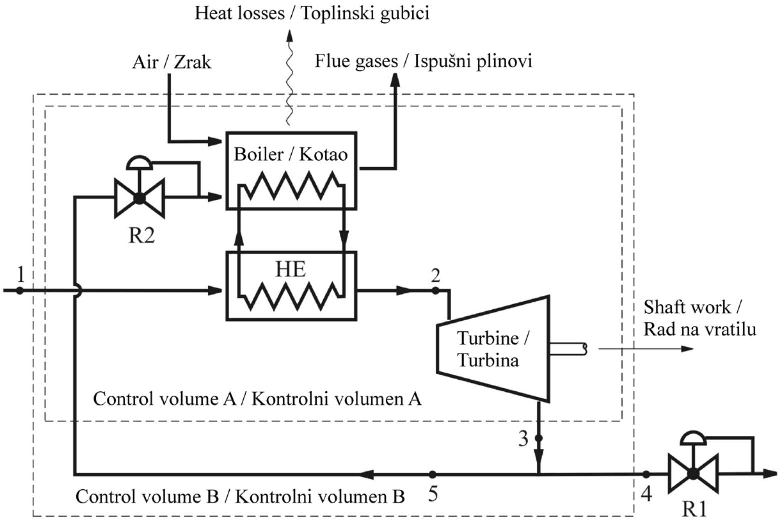

Figure 3. Model of the expansion turbine system. Symbols: HE – heat exchanger, R1, R2 – pressure regulator,

Slika 3. Model sustava s turbinom. Oznake: HE – izmjenjivač topline, R1, R2 – regulator tlaka

analysis this regulator is situated outside both control volumes. The boiler regulator (R2) reduces pressure to a level suitable for combustion; this regulator is situated within the control volume A.

Following simplifying assumptions are done to the model:

- the inlet pressure and temperature are constant,
- heat losses from the system are assigned to the heating system, i.e. the turbine is adiabatic,
- the heat exchanger efficiency is $\eta_{\mathrm{HE}} = 0.9$ regardless of the preheating temperature T_2.

Furthermore it is assumed the thermodynamic quality of the system is scale independent. Hence, for any required outlet pressure p_3, the independent design parameters are the pre-heating temperature T_2 and the turbine isentropic efficiency η_{T}. The system response to the varying parameters may be characterized by the following thermodynamic and economic indicators:

- the specific work output,
- the work-to-fuel ratio, relating the produced work to the supplied chemical (fuel) energy,
- the overall system efficiency, describing the level of energy losses in the system related to chemical energy of the entire gas flux,
- the second-law efficiency, describing how much of the available work potential is recovered in the process,
- the net present value ratio (NPVR) showing the financial result of the system installation at given price levels, with respect to the installation cost.

Relevant expressions are derived in the following subsections. Numerical results for two case studies are presented in section 4.

3.3. Energy analysis

Since natural gas is a fuel, its enthalpy comprises both the physical and the chemical part, the latter by an order of magnitude higher. A notation based on enthalpy changes rather than on absolute values allows one to account for the chemical enthalpy only for the combusted flux of gas. The lower heating value (LHV) was chosen as a measure of chemical enthalpy. Accordingly, the reference state for physical enthalpy must be chosen at 298.15 K and 101 325 Pa if LHV is present in an equation.

In order to determine all energy fluxes in the system it is first necessary to formulate balance equations for the particular devices. Solving the energy balance for the turbine (T) and the heat exchanger (HE) allows one to determine the specific work output w and the required heat production in the boiler q_B. The specific fuel consumption in the system may be determined from the boiler balance (B) under the assumption that the flue gas temperature is known. The balance equations related to 1 kilomole gas at point 1 are:

$$\text{T:}\ \ w = h_2^{\mathrm{G}} - h_3^{\mathrm{G}}, \tag{5}$$

where δ_{q} is

$$\text{HE:}\ \ q_{\mathrm{B}} = h_2^{\mathrm{G}} - h_1^{\mathrm{G}} + \delta q \quad \text{or} \quad q_{\mathrm{B}} = \left(h_2^{\mathrm{G}} - h_1^{\mathrm{G}}\right)\cdot\frac{1}{\eta_{\mathrm{HE}}}, \tag{6}$$

$$\text{B:}\ \ \nu_5\left(h_5^{\mathrm{G}} + \mathrm{LHV}\right) + \nu_5 \nu^{\mathrm{A}} h^{\mathrm{A}} = q_{\mathrm{B}} + \nu_5 \nu^{\mathrm{F}} h^{\mathrm{F}}. \tag{7}$$

The upper indices G, A, F denote natural gas, air and flue gases, respectively. The molar fluxes of air and of flue gases expressed per unit of combusted gas are denoted with ν^A and ν^F. The specific fuel consumption ν_5 is defined as the fraction of gas extracted as a fuel for the boiler:

$$\nu_5 = \frac{\dot{n}_5}{\dot{n}_1}. \tag{8}$$

System indicators resulting from the first-law analysis are the specific work, resulting directly from Eq. (5) and the work-to-fuel ratio defined at the control volume A as:

$$\varepsilon = \frac{w}{\nu_5\,\mathrm{LHV}}. \tag{9}$$

Note the parameter ε should not be referred to as 'efficiency' as its value may be greater than unity. This is because the expansion turbine is a work recovery and not a standard energy generation device.

It is also possible to define the overall system efficiency η_s describing the operation of an expansion turbine with respect to the entire natural gas transport system. This definition compares the useful energy fluxes at extended system boundaries (control volume B):

$$\eta_\mathrm{s} = \frac{\dot{E}^\mathrm{G}_{\mathrm{ch},4} + \dot{W}}{\dot{E}^\mathrm{G}_{\mathrm{ch},1}} = 1 - \nu_5 + \frac{w}{\nu_5\,\mathrm{LHV}}. \tag{10}$$

3.4. Exergy analysis

The thermodynamic quality of a system may be evaluated using a second-law efficiency. For systems other than a heat engine, this efficiency may be formulated as [1]:

$$\eta_\mathrm{II} = \frac{\text{Exergy recovered}}{\text{Exergy supplied}}. \tag{11}$$

For the analyzed expansion turbine system, the exergy fluxes crossing the system boundary may comprise the following components:

- the chemical exergy, bound to the content of combustible components as well as to the composition other than that of the environment,
- the physical exergy, resulting from pressure or temperature different from ambient conditions,
- the exergy of work and heat exchanged with the surroundings.

The chemical exergy of natural gas was calculated after Szargut's results for high-methane gas [13]:

$$b^\mathrm{G}_\mathrm{ch} = 1.04 \cdot \mathrm{LHV}^\mathrm{G}. \tag{12}$$

The calculation of the chemical exergy of flue gases takes into account only the change of the components' concentration compared to the environment:

$$b^\mathrm{F}_\mathrm{ch} = T_0 R_\mathrm{u} \sum_{i=1}^{N} x_i \ln \frac{x_i}{x_{i,0}}, \tag{13}$$

where $x_{i;0}$ denotes the concentration of given species in the environment.

The specific physical exergy of a gas flux was calculated from Eq. (2). Heat losses to the environment have a zero exergy content. Hence the significant exergy fluxes crossing the control volume A comprise the following terms:

$$\begin{aligned} b^\mathrm{G}_{\mathrm{ch},1} &= \nu_5 \left(b^\mathrm{G}_{\mathrm{ph},5} + b^\mathrm{G}_{\mathrm{ch},5} \right) = \\ &= w + b^\mathrm{G}_{\mathrm{ph},3} + \nu_5 \nu^\mathrm{F} \left(b^\mathrm{F}_\mathrm{ph} + b^\mathrm{F}_\mathrm{ch} \right) + \delta b, \end{aligned} \tag{14}$$

where δb is the specific internal exergy loss in the system. Note the significant chemical exergy of the main gas flux is not included in Eq. (14) as it is not destroyed in the system.

The second-law efficiency results from Eq. 14. It should be noted that the recovered exergy may or may not contain the physical exergy of the outlet stream, depending on whether or not it is intended for further work generation:

$$\eta_\mathrm{II} = \frac{\varphi\, b^\mathrm{G}_{\mathrm{ph},3} + w}{b^\mathrm{G}_{\mathrm{ph},1} + \nu_5 \left(b^\mathrm{G}_{\mathrm{ch},5} + b^\mathrm{F}_{\mathrm{ch},5} \right)}, \tag{15}$$

where $\varphi = 0$ for last-stage pressure reduction and $\varphi = 1$ elsewhere.

3.5. Economic analysis

The objective of the economic analysis is to investigate the impact of the design parameters on the feasibility of turboexpander installation. The differential NVP method was applied, analyzing the changes of the cash flow elements due to expander installation. The Net Present Value was defined as:

$$\mathrm{NPV} = -J_0 + \sum_{t=1}^{t=15} \frac{\Delta \mathrm{CF}}{(1+r)^t}, \tag{16}$$

where r is the discount rate, J_0 is the investment cost and ΔCF is the change in annual cash flow due to turboexpander installation. Assuming own capital investment, for any year t the cash flow is:

$$\Delta \mathrm{CF} = \mathrm{ES} - \Delta \mathrm{OC} - \Delta \mathrm{CIT}, \tag{17}$$

where ES are the electricity sales, Δ OC is the change of operating cost and Δ CIT is the change of the company income tax, accounting for the change of costs and sales as well as for the depreciation expense.

The ratio of the NPV and the investment cost NPV / J_0 is the Net Present Value Ratio (NPVR), describing the relative feasibility of an investment. The time necessary to achieve NPV > 0 is the Discounted Pay-Back Time (DPB). The two latter indicators are used for the case study analysis.

4. Case study analysis

Two sample pressure reduction stations were chosen to perform detailed analysis of the design parameters. Input data for both stations are presented in Table 3.

Both cases correspond to the system structure shown in Figure 3.

Table 3. Input data for the turboexpander case studies

Tablica 3. Podaci za studije slučaja turboekspandera

Parameter / Parametar	Case 1/ Slučaj 1	Case 2/ Slučaj 2
Inlet gauge pressure / Ulazni nadtlak plina, kPa	2400	250
Outlet gauge pressure, kPa / Izlazni nadtlak plina	400	2.6
Inlet temperature / Ulazna temperatura, °C	10	10
Minimum outlet temperature / Minimalna izlazna temperatura, °C	0	0
Existing gas pre-heater/ Postojanje predgrijača plina?	yes/ da	no/ ne
Generator efficiency/ Iskoristivost generatora	0.95	0.95

Calculations were performed for atmospheric air with the temperature 15 °C, relative humidity 60 % and CO_2 content 380 ppm. Flue gas composition was calculated assuming 15 percent excess air. Temperature of flue gases was assumed 120 °C.

Installation costs were estimated based on prices given for two reference devices:

- turboexpander: Gascontrol, J_{ref} = €32 100, N_{ref} = 15 kW electric power,
- boiler installation: De Dietrich, J_{ref} = €60 222, N_{ref} = 280 kW thermal power.

The prices were scaled according to the required power:

$$J = J_{ref}\left(\frac{N}{N_{ref}}\right)^{a}, \tag{18}$$

where a was assumed 0.60 for turboexpanders and 0.73 for boilers [12]. Boiler installation cost was determined only for Case 2; in Case 1 it was assumed that the existing boiler system may be used with increased heating medium temperature.

The following economic data were assumed:

- 100 % own capital investment,
- discount rate r =1.93 %,
- annual depreciation rate 18 %,
- gas price € 0.23/Nm^3 (0 °C / 101 325 Pa),
- electricity sale price € 50/MWh.

Analyses were performed for the pre-heating temperature ϑ_2 varying from 20 °C to 110 °C and for the turbine isentropic efficiency η_T between 0.1 and 1.0. Results of the thermodynamic analysis for Cases 1 and 2 are shown in Figures 5–7. As can be seen in the figures, the specific work output, the work-to-fuel ratio and the second-law efficiency increase with the isentropic turbine efficiency. However, lower values of isentropic efficiency allow one to reduce the preheating temperature

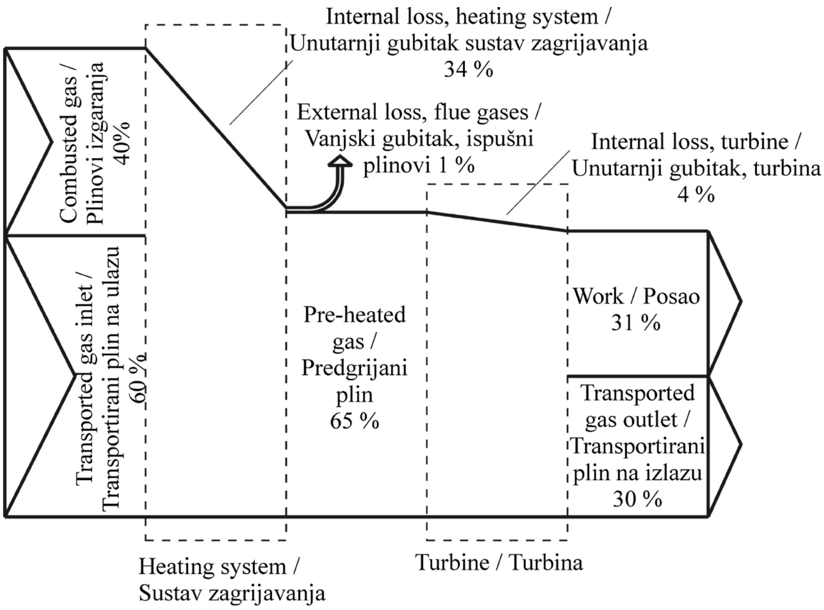

Figure 4. Transformations of exergy in the expansion turbine for the Case 1. Assumed parametres: gas-pre-heating to 110 °C, turbine efficiency 0.90

Slika 4. Promjene eksergije u ekspanzijskoj turbini u slučaju 1. Pretpostavljeni parametri: predgrijavanje plina na 110 °C, stupanj djelovanja turbine 0.90

and thus to obtain more favourable values of the work-to-fuel ratio.

The obtained specific work is higher in Case 1, which is bound primarily to a higher inlet-to-outlet pressure ratio. Maximum work may be obtained for the highest available turbine efficiency. As can be concluded from Figure 5, further increasing of the pre-heating temperature allows one to increase the specific work beyond the range obtained in this study.

used for further expansion while reducing to the low-pressure level. Therefore the second-law efficiency is not zero even in the case of a standard pressure regulator (depicted in the graph as a turbine with 0 efficiency). In Case 2, the outlet gas exergy is not intended for further work generation and therefore not accounted for in the definition. However, the exergy content of the outlet gas stream is greater than zero, hence the second-law efficiency would not achieve unity even for isentropic

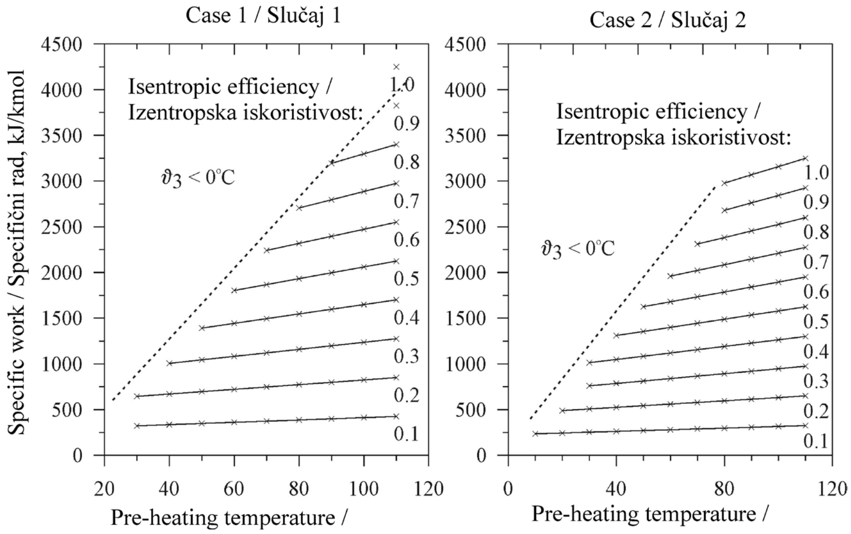

Figure 5. The specific work production for the Cases 1 and 2.

Slika 5. Specifični rad u slučajevima 1 i 2

The work-to-fuel ratio (Figure 6) increases with decreasing pre-heating temperature. This ratio is limited by the condition of minimum turbine outlet temperature, however it does achieve values greater than unity for the Case 2.

The second-law efficiency (Figure 7) was calculated based on different definitions for Cases 1 and 2. In the Case 1, the outlet medium-pressure gas flux can be

expansion with no outlet temperature limitation. The overall system efficiency as defined in Eq. 10 is not shown graphically as its values vary between 99.4 and 99.9 %, which shows that the analyzed transformations of physical energy and exergy are very small compared to the chemical energy content of natural gas.

In addition to the analysis of design parameters, a detailed calculation of exergy fluxes in the system was

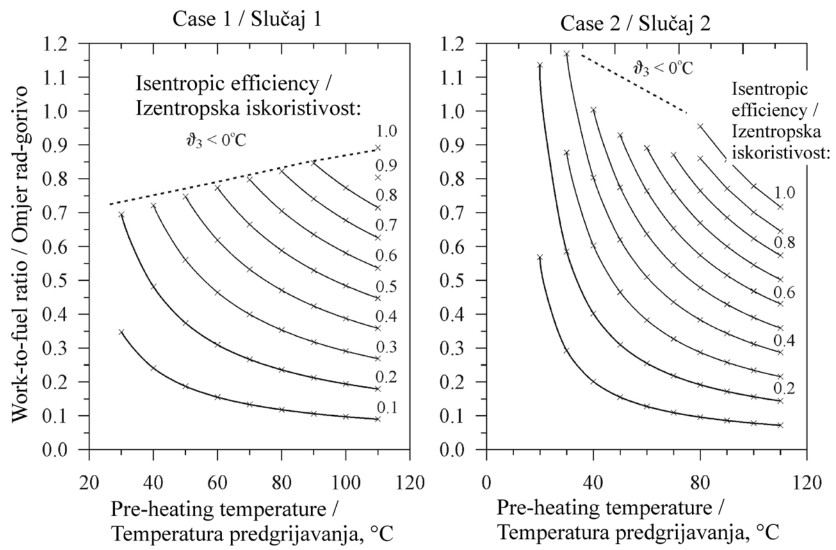

Figure 6. The work-to-fuel ratio for the Cases 1 and 2

Slika 6. Omjer rad-gorivo u slučajevima 1 i 2 performed for selected parameters (η_T = 0.9 and ϑ_2 = 90 °C) in the Case 1. Results are shown in Figure 4. It is worth stressing that the main area of exergy losses in a system with an expansion turbine is the pre-heating system. The exergy losses result from irreversible heat transfer from hot flue gases to relatively cold natural gas. This indicates that an alternative, low-temperature heat source would improve the second-law efficiency of the system.

the installation of a turboexpander is characterized by a long pay-back time of minimum 7 years.

Results of the presented case studies may be used for a preliminary evaluation of the energy generation potential within the entire gas transport system. In general, this potential depends on disposable pressure drops as well as on available baseload flow rates at the particular PRS's, and it is thus not possible to determine without detailed information on gas flow characteristics in the system.

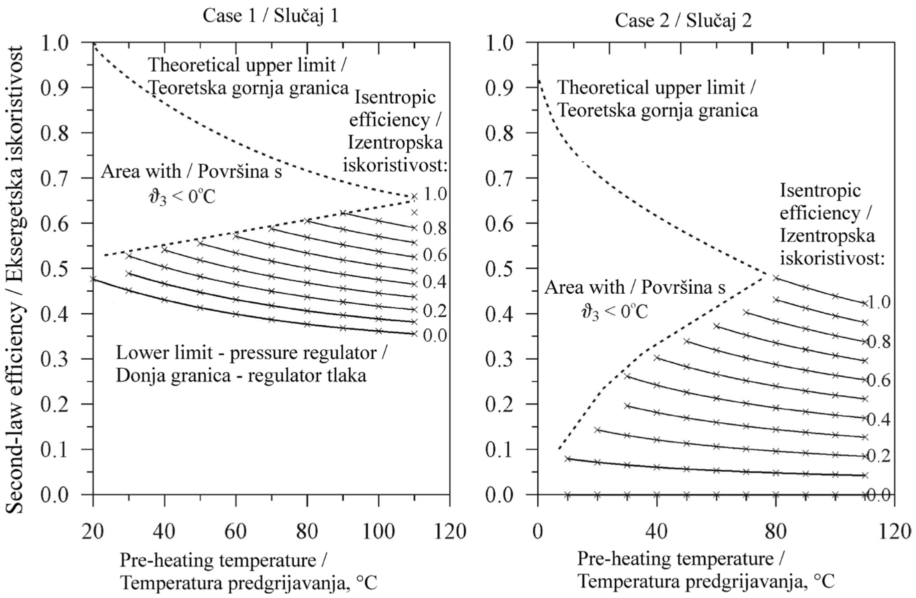

Figure 7. The second-law efficiency for the Cases 1 and 2.

Slika 7. Eksergetska iskoristivost u slučajevima 1 i 2

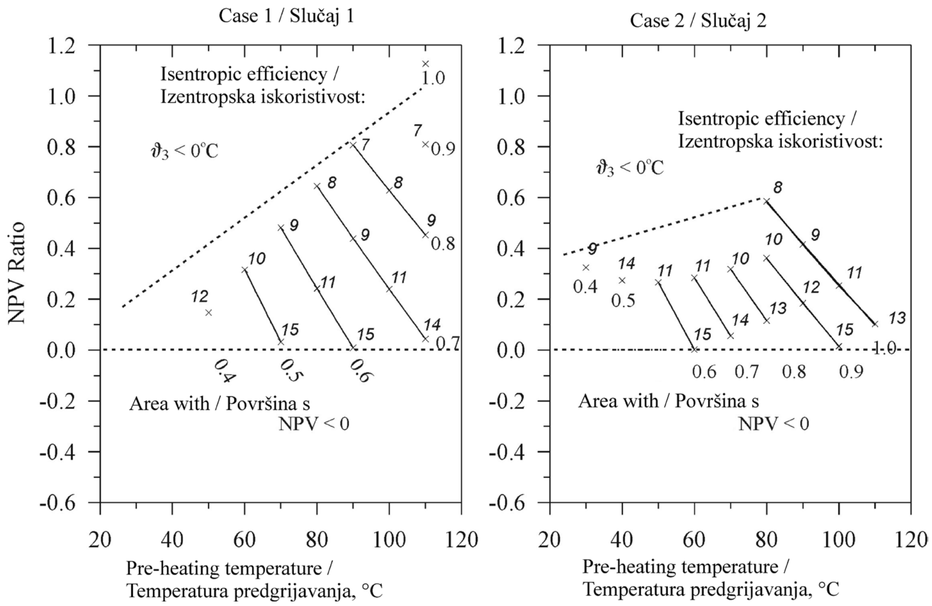

Figure 8. The Net Present Value Ratio for the Cases 1 and 2. Numbers in italic denote the discounted pay-back time (years).

Slika 8. NPVR – omjer neto sadašnje vrijednosti NPV i investicijskog troška u slučajevima 1 i 2. Brojevi u kurzivu predstavljaju diskontirani period povrata kapitala u godinama.

Results of the economic analysis are shown in Figure 8. Only positive NVP results are shown in the graphs. Maximum NPV ratio is obtained for maximum isentropic efficiency with minimum pre-heating temperature corresponding to that efficiency. The results indicate that for the current ratio between the price of gas and electricity

Assuming, as a rule of thumb, that 50 percent of natural gas transported can pass an installation corresponding to that of Case 1 (specific work 3500 kJ/kmol), one can estimate the generation potential at ca. 22 GWh per 10^9 normal cubic metres of gas.

5. Final conclusions

Expansion turbines provide utilization of the physical exergy of the pressurized natural gas. However, due to technical limitations bound with the minimum gas temperature at the turbine outlet it is necessary to preheat gas before expanding in a turbine. This fact does not allow to classify expansion turbines as clean or renewable energy sources, since a part of the input exergy is obtained by combustion of fossil fuels.

However, as shown in the sample exergy flow diagram, it is possible to indicate which part of the input exergy derives from the inlet gas exergy, which is otherwise destroyed in conventional regulating devices. Accordingly, a corresponding part of the produced energy should be classified as clean or waste energy.

The calculated thermodynamic indicators obtained from the analysis show the high thermodynamic quality of the proposed turboexpander installation. The work-to-fuel ratio may reach values higher than unity. Although this ratio should not be referred to as efficiency, it may be compared with the efficiency of the most current combined cycle power plants, as this efficiency, reaching about 60 %, is also a work-to-fuel ratio.

Results of the economic analysis show that positive economic results of turboexpander installation are possible to obtain if the design parameters are chosen appropriately. However, a higher feasibility might be expected in the case of rapid growth of electricity prices that may occur in the near future.

Last but not least, the environmental effect should be underlined, as the installation of turboexpander allows one to replace electricity production in a conventional power plant with an efficiency much less than the work-to-fuel ratio of the expander.

6. Appendix A Thermodynamic properties of natural gas

Calculation of thermodynamic properties of natural gas was done based on a real-gas equation of state (EOS) [8] explicit for the Helmholtz free energy $\alpha(\rho, T, \bar{x})$, from which any other thermodynamic parameter may be derived.

For a pure substance, the Helmholtz free energy a consists of an ideal gas term a^0 and a residual term for real gas behaviour a^r:

$$a(\rho,T) = a^0(\rho,T) + a^{\mathrm{r}}(\rho,T). \tag{19}$$

Eq. (19) may also be written in a dimensionless form with the parametres $\alpha = a/(RT)$, $\delta = \rho/\rho_c$ and $\tau = T_c/T$:

$$\alpha(\delta,\tau) = \alpha^0(\delta,\tau) + \alpha^{\mathrm{r}}(\delta,\tau). \tag{20}$$

For a mixture with a given molar composition, the reduced Helmholtz energy consists of three terms: a weighted sum of ideal gas terms increased by the entropy of mixing, a weighted sum of residual terms and of a so-called departure function accounting for non-ideal mixing:

$$\alpha(\delta,\tau,\bar{x}) = \sum_{i=1}^{N} x_i\left[\alpha_i^0(\rho,T) + \ln x_i\right] + \\ + \sum_{i=1}^{N} x_i\alpha_i^r(\delta,\tau) + \Delta\alpha(\delta,\tau,\bar{x}). \tag{21}$$

The departure function $\Delta\alpha$ (δ, τ, $\bar{x}$) of a mixture is calculated using specific mixing functions listed in [8]. Moreover, the reduced density δ and the inverse reduced temperature τ are related to pseudocritical parametres of the given mixture which also need to be calculated using particular mixing rules.

Pressure, enthalpy and entropy may be derived from the reduced Helmholtz free energy:

$$p(\rho,T,\bar{x}) = \rho RT\left(1+\delta\alpha_{\delta}^{\mathrm{r}}\right), \tag{22}$$

$$h(\rho,T,\bar{x}) = RT\left[1+\tau\left(\alpha_{\tau}^{0}+\alpha_{\tau}^{\mathrm{r}}\right)+\delta\alpha_{\delta}^{\mathrm{r}}\right], \tag{23}$$

$$s(\rho,T,\bar{x}) = R\left[\tau\left(\alpha_{\tau}^{0}+\alpha_{\tau}^{\mathrm{r}}\right)-\alpha^{0}-\alpha^{\mathrm{r}}\right]. \tag{24}$$

Lower indices in Eqs. (22)–(24) denote partial derivatives.

Calculations were performed using REFPROP NIST Standard Reference Database v. 8.0, with ideal gas properties, real gas EOS and mixing functions according to the GERG model, which is given in details in the GERG Technical Monograph [8].

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