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Thermoeconomic assessment of a natural gas expansion system integrated with a co-generation unit

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ABSTRACT

The paper presents a thermoeconomic assessment of an expansion system applied in the natural gas transportation process. The system consists of two turboexpander stages reducing the natural gas pressure and providing mechanical energy to drive electric generators. Gas pre-heating, required to prevent hydrate formation, is performed upstream of each expansion stage using waste heat recovered from a gas engine, which contributes to the total system electricity production. The system constitutes a hybrid energy generation unit as the generated electricity derives partially from the physical exergy of pressurized natural gas, and partially from the primary energy of fuel. The presented thermoeconomic description of the system comprises definitions of system quality indicators, as well as an identification of irreversibility bound to the operation of the system's components.

1. Introduction

Natural gas (NG) is a major fossil fuel contributing to about one fourth of the world's primary energy consumption (23.8% [1]). The fuel is transported to large distances by means of pipeline systems with compressor stations consuming between 3% and 5% of the chemical energy of the transported gas, depending on the distance covered [2]. Alternatively, the gas can be transported as LNG, which is bound to an energy cost of about 2.5% of LHV for the liquefaction [3] and a boil-off loss of 2–5% in the transport and storage chain [4].

While the LNG transport dominates the overseas trading [5], the pipeline transport is fundamental for inland transmission and distribution. The pipeline networks are organized hierarchically within 3–4 pressure stages, connected by pressure reduction stations (PRSs). In most PRSs, pressure reduction is performed in a pressure regulator, i.e. an automatically operated throttling valve. This solution ensures the reliability of supply [6], but destroys the available exergy of pressurized gas which could be used to generate work in the expansion process [7]. Moreover, in order to prevent hydrate formation due to the temperature drop in the throttling process, the majority of PRS consume energy for gas pre-heating in order to maintain its temperature above the 'permissible safe temperature' [8]; detailed regulations may further specify the limit value (e.g. 5–8 °C [9]). As a result, the process of pressure reduction is energy consuming instead of being a source of energy generation.

It is technically possible to convert the available decrease in physical exergy into mechanical work by means of a piston, screw or a turbine expander (turboexpander). The generated work may be converted to electricity, some authors also suggested its direct application for hydrogen production [10] or, in particular industrial applications, to drive compressors [11]. Suitable expander technology is offered by many manufacturers worldwide (see e.g. [12,13]).

The key problem limiting the application of expanders is the significant temperature drop related to the conversion of internal energy of gas into mechanical work. It is therefore required to increase the amount of heat supplied to the gas prior to the expansion process. For industrial PRSs waste heat integration is a suitable option (see. e.g. [14]). For stand-alone PRS objects heat may be generated in gas boilers [7,15], co-generation units with IC engines [16,17], or fuel cell systems (technology under development [18]). Some authors also suggest to lead the process without pre-heating if the risk of hydrate formation is excluded [19]. Moreover, low outlet temperatures may be used for cold generation if relevant process integration is possible [20].

Despite numerous industrial applications of NG expansion plants, the thermodynamic methodology for their evaluation is not widely developed. Poživil [15] and Kostowski [7] analyzed the performance of simple turboexpander systems co-operating with a gas pre-heater supplied from a local boiler house. As pointed out by Kostowski [7], such systems are hybrid energy sources as

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Nomenclature

b	specific exergy, kJ/kg
$\dot{B}$	exergy flux, kW
$\dot{B}^*$	exergy cost of a flux, kW
e	CO_2 emission factor, kg CO_2/kW h
E	exergy flux in a productive structure, kW
F	fuel flux, kW
h	specific enthalpy, kJ/kg
$\dot{H}$	enthalpy flux, kW
I	irreversibility, kW
k^*	unit exergy cost, –
LHV	lower heating value, kJ/kg
$\dot{m}$	mass flow rate, kg/s
n	number of components of the plant, –
$\dot{n}$	molar flow rate, kmol/s
s	specific entropy, kJ/(kg K)
p	pressure, Pa
P	product flux, kW
$\dot{Q}$	heat flux, kW
T	temperature, K
$\dot{V}_n$	normalized volumetric flow rate at 0 °C and 101.325 kPa, m_n^3/h
$\dot{W}$	power, kW
η_b	exergetic efficiency, –
κ	unit exergy consumption, –

Matrices and vectors

$\mathbf{F_S}$	vector of external resources ($n \times 1$), kW
$\langle\mathbf{KP}\rangle$	matrix of unit exergy consumptions ($n \times n$)
$\mathbf{P}$	vector of products of the components ($n \times 1$), kW
$\mathbf{P_S}$	vector of final products of the plant ($n \times 1$), kW
$\lvert\mathbf{P}\rangle$	product operator ($n \times n$)
$\mathbf{U_D}$	identity matrix ($n \times n$)
$\boldsymbol{\kappa}_\mathbf{e}$	vector of unit exergy consumptions of external resources ($n \times 1$)

Acronyms

GP	gas preheater
HE	heat exchanger
HT	high temperature
LNG	liquefied natural gas
LT	low temperature
NG	natural gas
ppm	parts per million (molar)
PR	performance ratio
PRS	pressure reduction station
TE	turboexpander

the produced electric energy derives partially from the exergy of the pressurized, expanded gas, and partially from the primary energy of fuel. Therefore, the thermodynamic quality of the system should be described with the exergy rather than energy efficiency. It has also been shown that exergy losses in the system are primarily bound to the pre-heating process due to high irreversibility of heat transfer occurring at high temperature differences. Expansion plants integrated with a co-generation unit have been analyzed by Mirandola et al. [16]. Although the authors present interesting case studies, the related methodology is not explained in detail.

The aim of the present paper is to apply thermoeconomic analysis [21] for a turboexpander system co-operating with a co-generation unit. The design data of the Arlesheim expansion plant in Switzerland [22] have been used for developing the model. Within this paper it has been shown that this type of hybrid energy generation systems is characterized by good thermodynamic feasibility as well as very low emission indicators; therefore, its application deserves attention and promotion. Furthermore, exergy and thermoeconomic analysis has been used to point out how the performance of the system can be improved.

2. Process description

A scheme of a two-stage turboexpander installation with a CHP unit is shown in Fig. 1. High pressure (HP) natural gas destined for expansion is first heated in the Low Temperature Gas Pre-heater 1 (LTGP1) and in the High Temperature Gas Pre-heater 1 (HTGP1) and then expanded in the first expansion stage (Turboexpander TE1). Subsequently, the middle pressure (MP) gas is heated in the pre-heaters LTGP2 and HTGP2 and then expanded in the second stage TE2. The expanders generate shaft power ($\dot{W}_{37}$ and $\dot{W}_{38}$), which is then transformed to electricity ($\dot{W}_{el,40}$) through a common gearbox and an electric generator.

The heat required for pre-heating the NG is generated in a CHP module comprising a gas engine and an electric generator. The heat is transported to the HP and MP gas fluxes by means of two water circuits (low- and high-temperature, LT and HT). The shaft power generated by the gas engine is converted to electricity ($\dot{W}_{el,39}$) in the generator. The entire expansion system has thus two independent electric outputs.

Nominal system data are collected in Table 1.

3. Thermo-economic assessment methodology

The thermo-economic assessment has been preceded by solving mass and energy balances for the system and by calculating the values of exergy of all relevant fluxes. Modelling has been performed under some simplifying assumptions which do not introduce major changes to the thermodynamic evaluation of the system:

- NG is approximated by methane (LHV = 49.91 MJ/kg).
- All heat exchangers and expanders are adiabatic with the environment; heat exchangers are isobaric.
- All fluxes in the HT and LT circuits have the nominal feeding/return temperature; no mixing irreversibilities occur at flow junctions.

3.1. Mass and energy balance

A system of equations based on steady-state mass and energy balances has been used to determine fluxes and the values of enthalpy in all system points. For split and junctions in the HT and LT water circuits, the relevant fluxes have to meet the *mass conservation* equation:

$$\sum \dot{m}_{in} = \sum \dot{m}_{out}. \qquad (1)$$

The flux of fuel supplied to the CHP module has been calculated based on the electric output and the electric efficiency of the CHP module, and on the LHV of the fuel:

$$\dot{n}_9 = \frac{\dot{W}_{el,39}}{\mathrm{LHV}_9 \cdot \eta_{el,\mathrm{CHP}}}. \qquad (2)$$

The flux and the composition of exhaust gases have been obtained by means of stoichiometric equations, relating the quantity of reagents and products. Calculations were performed for