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An evaluation of the possibilities of using turboexpanders at pressure regulator stations

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Abstract

Natural gas in Poland is transported by onshore pipelines with a maximum operating pressure of up to 8.4 MPa. The gas pressure is then reduced to 1.6 MPa or 0.4 MPa for delivery to regional/local distribution networks or to end-user installations. The pressure reduction is usually performed by a pressure regulator. Pressure reduction can also be achieved through expansion of the gas at the turboexpander, which can be harnessed to produce electricity from the recovered mechanical energy of the gas. The main objective of this study is to investigate the factors influencing the efficiency of the gas expansion process and to carry out a feasibility study involving the application of turboexpanders at selected natural gas pressure regulator stations belonging to the Polish transmission system operator Gaz System S.A.

Keywords: pressure regulator, turboexpander, waste energy recovery, city-gate station, pressure let-down station

1. Introduction

A gas pressure regulating and metering station (city gate station) consists of technical equipment and process control systems for natural gas stream pressure reduction and gas flow measurement. The process of gas pressure reduction is usually performed using a pressure regulator in the form of a throttling device, the process being accompanied by significant energy dissipation resulting in large exergy losses in respect of the gas stream.

Several authors have studied the problem of the low energy efficiency of pressure regulator stations [1–4] due to the high commercial and ecological cost involved in the natural gas pressure reduction process [5]. One well known solution for increasing the energy efficiency of the pressure reduction process is based on the application of a gas turboexpander [6] or reciprocating expansion engine [7] instead of a pressure regulator, and extracting mechanical energy from the gas while reducing its pressure, i.e. the mechanical energy of the gas is converted into shaft work, which can be used directly [8] or subsequently transmitted to a generator to produce electric energy [9].

The process of reducing the pressure of the natural gas at the pressure regulator has an undesirable side effect: a decrease in gas temperature. This is due to the Joule-Thomson effect, which can cause vapor condensation or hydrate formation [10, 11] leading to hydrate plugs in the valve seat of the pressure regulator. These phenomena have a very adverse influence on pipelines and station equipment, therefore it is common practice in high pressure regulator stations to preheat the gas before it enters the pressure regulator. Natural gas fired boilers are typically used to provide heat for the process of preheating gas in both pressure regulator and turboexpander applications. However, the effect of pressure reduction on temperature during polytropic process (expansion) is stronger than the isenthalpic process (throttling), consequently the heat demands in turboexpander applications exceed those in pressure regulator applications.

In recent years, research efforts regarding pressure regulator stations have sought to replace the traditional systems of heat production by gas boilers with cogeneration systems, including CHP units, with a gas turbine [12], internal combustion engine [13–15] and fuel cell [6, 16, 17]. Recent studies are also focusing on integrating natural gas expansion plants with renewable energy sources, including photovoltaic systems [18], heat pumps [19], geothermal heat exchangers [20], and vortex tubes [21]. These research works aim to reduce the primary energy demand for gas preheating in the process of pressure reduction.

The objective of the present study is to evaluate the feasibility of using a turboexpander on a retrofit basis in selected two real-life pressure regulator stations with significantly different demand patterns, i.e. supplying gas to a municipal area and to an industrial customer. The station delivering gas to the municipal area has seasonal fluctuations of gas flow rate caused by gas consumption for heating purposes, while the station supplying gas to the industrial customer

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enjoys relatively stable demand conditions. The flow processes present in the stations were investigated and a quantitative approach to consideration of variable demand conditions in the turboexpander selection and sizing procedure is presented and discussed.

2. Theory

Assuming that the kinetic and potential energy changes of the gas along the turboexpander are insignificant compared to the changes in enthalpy, the electric power produced in the generator with a turboexpander as a prime mover can be obtained from

$$\dot{W} = \eta_m \eta_g \dot{m} \Delta h \tag{1}$$

where $\dot{W}$ is the power available at generator terminals (W), η_m is the mechanical efficiency of the turbine, η_g is generator efficiency, $\dot{m}$ is the mass flow rate of the gas (kg/s) and Δh is the enthalpy difference between the inlet and outlet conditions (J/kg).

Given the isentropic efficiency of the turboexpander η_0, the enthalpy difference is

$$\Delta h = \eta_0 \left(h_{in} - h_{0,out} \right) \tag{2}$$

where h_{in} is the enthalpy of the gas at the turboexpander inlet calculated from the property relation $h_{in} = h\left(p_{in}, T_{in}\right)$, and $h_{0,out}$ is the enthalpy of the gas at the turboexpander outlet if the process were isentropic, i.e.

$$h_{0,out} = h\left(p_{out}, s_{out}\right) \tag{3}$$

Since for the isentropic process $s_{out} = s_{in}$, while the entropy of the gas at the turboexpander inlet can be calculated from the pressure and temperature conditions with the appropriate property relation $s_{in} = s\left(p_{in}, T_{in}\right)$, the enthalpy $h_{0,out}$ is obtained from Eq. (3). The enthalpy of the gas at the turboexpander outlet is

$$h_{out} = h_{in} - \Delta h \tag{4}$$

and the temperature at the turboexpander outlet can be calculated iteratively from the property relation $T_{out} = T\left(p_{out}, h_{out}\right)$.

The isentropic efficiency of the turboexpander at part-load conditions can be obtained through dimensional analysis. The maximum efficiency curve as a function of turboexpander specific speed can be approximated by

$$\eta = \eta_d \left[2\left(\frac{\dot{V}}{\dot{V}_d}\right) - \left(\frac{\dot{V}}{\dot{V}_d}\right)^2 \right] \tag{5}$$

where η_d is turboexpander efficiency at the design (optimum) conditions, while $\dot{V}$ and $\dot{V}_d$ are volumetric gas flow rates at part-load and design conditions, respectively.

The thermodynamic property relations were calculated in this study using GERG 2004 Equation of State as implemented in the NIST REFPROP database [22].

3. Results

3.1. Case study I: local distribution network

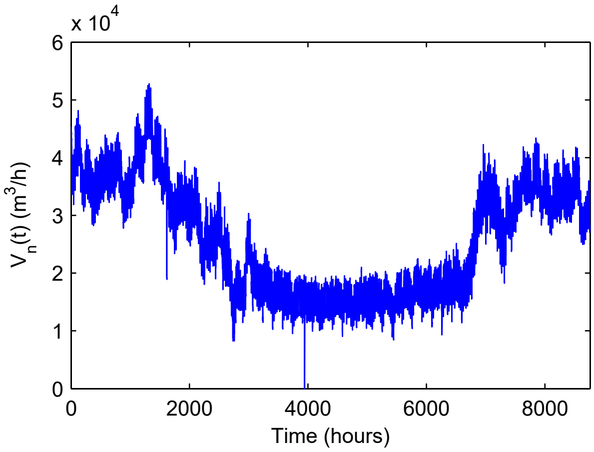

Figure 1: Hourly demand curve in a municipal area (flow at standard conditions of 101.325 kPa and 273.15 K)

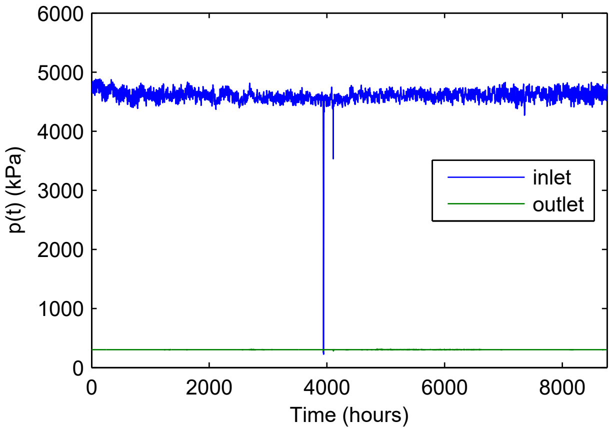

Figure 2: Inlet and outlet pressure profiles (gauge pressure) in Case study I

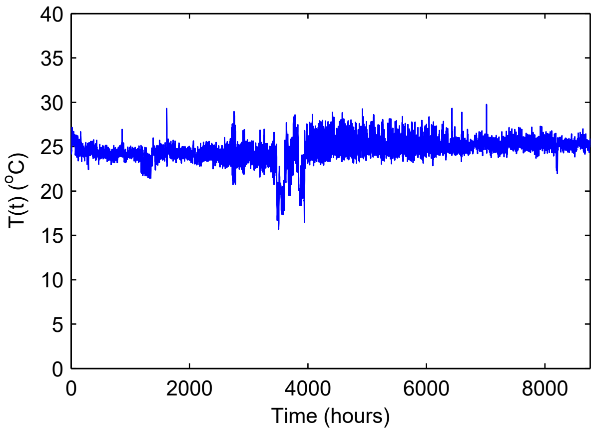

Figure 3: Inlet temperature profile in Case study I

Fig. 1 shows the hourly demand curve (volumetric flow rate at standard conditions) for a representative period of one year of operation of the pressure regulator station delivering gas from an upstream high pressure gas transmission pipeline to a local gas distribution network in a municipal area. The corresponding values of pressure at the inlet and outlet of the station are shown in Fig. 2, while the values of the temperature at the inlet of the station are shown in Fig. 3.

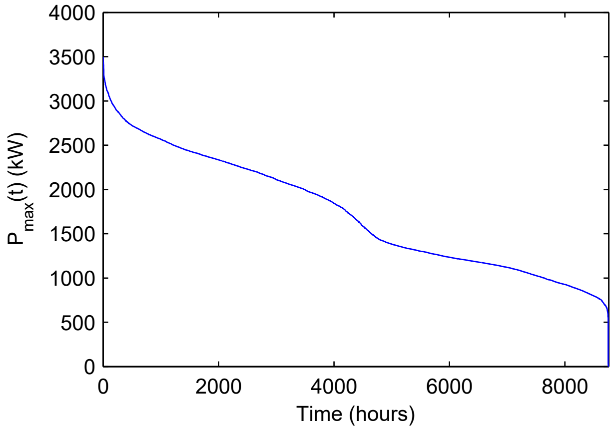

Figure 4: Duration curve of power output (theoretical limit value) in Case study I

Fig. 4 shows the estimated theoretical limit value of power output from the process, assuming isentropic expansion of the gas between the inlet state (p-T conditions) and the outlet state (pressure setpoint) in the form of a load duration curve corresponding to the load curve presented in Fig. 1. Based on cumulative load duration curve, the discreet optimization problem of turboexpander sizing was formulated with the net income from the operation of the turboxpander as an objective function, and the number of turboexpanders, nominal power output and annual working hours of each turboexpander as decision variables. The solution of the problem led to the selection of one two-stage turboexpander with nominal power output of 1,800 kW (Table 1).

3.2. Case study II: industrial customer

Fig. 5. shows the hourly demand curve for the period of 1 year at the pressure regulator station delivering gas from a high pressure gas pipeline to the small scale end-user installation. The pressure profiles at the inlet and outlet of the station ae shown in Figure 6, and the respective temperature profile at the inlet of the station is shown in Fig. 7. The analogously estimated theoretical limit value of power output from the process is shown in Fig. 8. The solution of the sizing problem results in the selection of one single-stage turboexpander with nominal power output of 160 kW (Table 1).

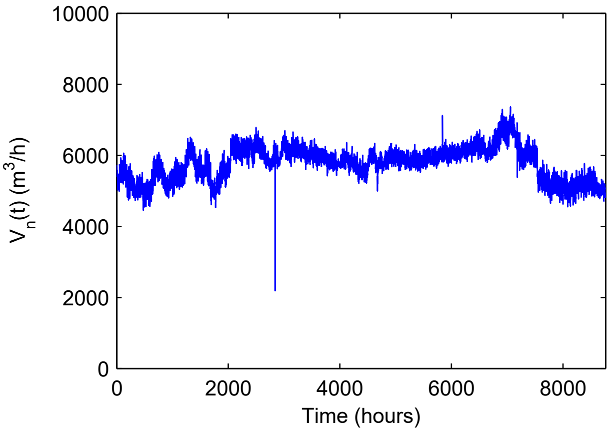

Figure 5: Hourly demand curve of industrial customer (flow at standard conditions of 101.325 kPa and 273.15 K)

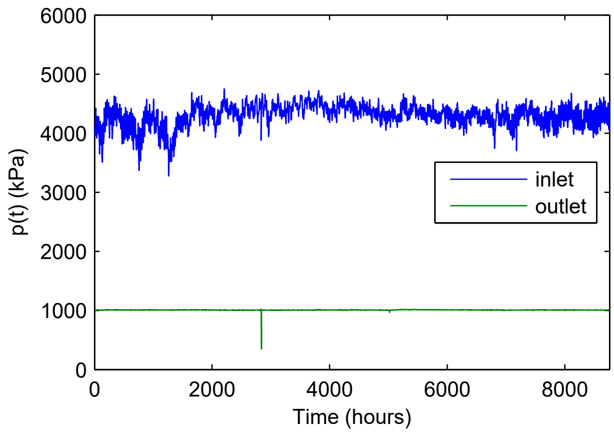

Figure 6: Inlet and outlet pressure profiles (gauge pressure) in Case study II

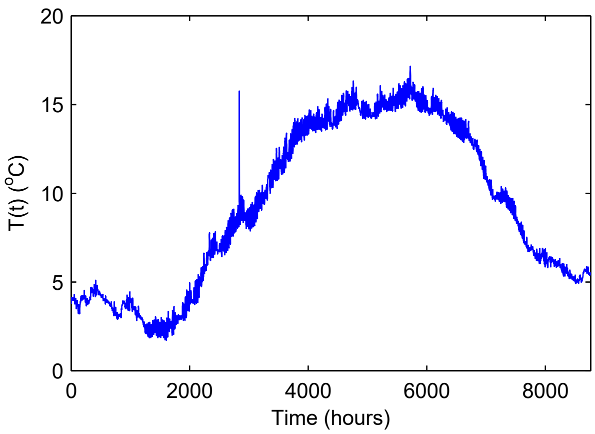

Figure 7: Inlet temperature profile in case study II

3.3. Equipment sizing calculations

Given the turboexpander efficiency at design conditions, the efficiency at part-load operational conditions was calculated from Eq. 5. In the case of the local distribution network, the configuration with a two-stage turboexpander was adopted, with inlet and intermediate pressures of 3.3 MPa and 1.2 MPa, respectively. A series pressure regulation system before gas expansion was assumed in order to limit the enthalpy decrease on a single stage of the turboexpander,

Table 1: Results of the turboexpander sizing calculations

Case study	I (distribution network)	II (industrial customer)
Turboexpander manufacturer	CRYOSTAR	RMG
Type	TG200	MTG160
Number of stages	2	1
Nominal power output (kW)	1,800	160
Efficiency at design conditions (%)	92	80
Nominal gas flow rate (m^3/h)	40,000	9,000
Maximum gas flow rate (m^3/h)	44,000	10,000
Minimum gas flow rate (m^3/h)	10,000	3,000
Annual working hours	8,725	8,760
Turboexpander capital expenditures (PLN)	6,420,700	1,060,500

Table 2: Characteristics of a preheating system with a gas boiler

Case study	I (distribution network)	II (industrial customer)
Annual heat demand (MWh)	16,043	1,921
Heating system supply temperature (°C)	90	110
Boiler power output (kW)	3,800	300
Fuel price (PLN/kWh)	0.1203	0.1137
Annual fuel cost (PLN)	2,031,000	229,000
Preheating system capital expenditures (PLN)	800,000	78,500

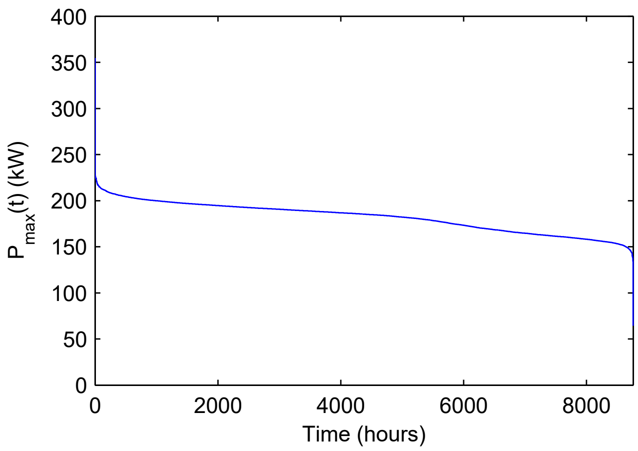

Figure 8: Duration curve of power output in Case study II

which should not exceed ~150 kJ/kg due to turboexpander design constraints. Two alternative solutions regarding the configuration of the preheating system were considered in this study, namely a) the configuration with gas boilers, and b) the configuration with a CHP unit with an internal combustion engine. The results of the preliminary equipment sizing calculations for the above scenarios are presented in Table 2 and Table 3.

The annual heat demand in Table 2 is the integral with respect to time of the instantaneous heat rate over a period of 1 year. Boiler power outputs result from the maximum instantaneous heat rate occurring at the stations. Annual fuel costs were determined on the basis of annual heat demand, assuming a fuel price of 0.1203 PLN/kWh for group E high methane natural gas in the local distribution network and 0.1137 PLN/kWh for group L sub-group Lw nitrified natural gas for the industrial customer. For the calculation of heat demand, constant efficiency was assumed for the gas preheater and gas boiler. The figures were 85% and 95%, respectively.

The peak demand for heat in Table 3 is the maximum instantaneous heat rate occurring at the stations, which served as the design parameter for the selection of the CHP engines. The sizing of the engines was based on heat demand, since heat production for the gas preheating is the main product of the cogeneration process. Electric power of the engines results from the heat-to-power ratio declared by the engine manufacturers.

The required heating system supply temperature results from the assumptions that the minimum temperature difference between the heating medium and the gas is 10°C, while the inlet gas temperature of the preheater should ensure the minimum temperature of 5°C at the turboexpander outlet (operational constraint). Fuel consumption and annual fuel costs were determined from engine manufacturer data, based on the assumption that the engines operate at nominal power output throughout the year.

3.4. Economic analysis

A summary of the economic indicators for the two analyzed cases is presented in Table 4.

The revenue from sales of electricity produced by the turboexpander was calculated by multiplying the amount of electricity produced in a power generator driven by the turboexpander by the sell price of electricity, which was set at 200 PLN/MWh. The sell price of electricity produced in CHP accounts for the premium of 29.84 PLN/MWh, set by the Polish energy regulator as financial support for CHP electricity. In the case of the CHP unit, the total revenue from electricity sales covers the electricity produced in the generators powered by both the turboexpander and CHP unit.

Table 3: Characteristics of a preheating system with a CHP unit

Case study	I (distribution network)	II (industrial customer)
Peak heat demand (kW)	3,833	302
Heating system supply temperature (°C)	90	110
Electric power output (kW)	2×2,022 = 4,044	400
Heat output (kW)	2×2,265 = 4,530	300
Annual fuel cost (PLN)	10,024,000	942,000
Preheating system capital expenditures (PLN)	8,610,000	1,640,000

Table 4: Results of cost-effectiveness analysis

Case study	I (distribution network)		II (industrial customer)	
Preheating system type	Boiler	CHP unit	Boiler	CHP unit
Revenue from electricity sales - turboexpander (PLN)		1,977,400		242,700
Revenue from electricity sales - CHP unit (PLN)		8,142,000		805,000
Total annual revenue from electricity sales (PLN)	1,977,400	10,119,400	242,700	1,047,700
Total capital expenditures (PLN)	7,220,700	15,030,700	1,139,000	2,700,500
Annualized capital expenditures (PLN)	1,247,900	2,597,600	196,800	466,700
Operation and maintenance costs (PLN per annum)	128,000	1,627,000	21,000	144,000
Cash flow from investing activities (PLN per annum)	- 1,429,500	- 4,129,200	- 204,100	- 505,000
Annual EBITDA (PLN)	-181,600	-1,531,600	-7,300	-38,300

The annualized capital expenditures were determined from the following formula

$$CAPEX_a = CAPEX \frac{r}{1-(1+r)^{-n}} \quad (6)$$

where r is the discount rate (%) and n is the project lifespan in years. For the purpose of the economic effectiveness evaluation, a constant discount rate of 5% with a seven-year amortization period was assumed.

The costs of project documentation, adaptation of existing station facilities and the electricity connection are subject to local conditions, and there is no literature data allowing reliable estimation. Based on market research, project design and construction work was assumed to amount to 12% of the turboexpander and preheating system capital expenditures. Turboexpander service and maintenance costs were set at 2% of the capital expenditures [23], while the service and maintenance cost of a CHP unit, confirmed by the manufacturer, was 10 EUR/MWh (EUR/PLN = 4.1).

Total capital expenditures in Table 4 are the sum of turboexpander capital expenditures and the capital expenditures of the respective preheating system.

4. Discussion

Table 1 shows that considering the minimum gas flow rate required for the turboexpander operation, the availability of the pressure regulator station is 99.6% for case study I and 100% for case study II, i.e. the turboexpander could run 8,725 h annually in the station supplying the gas distribution network, and 8,760 h in the station delivering gas to the industrial customer. However, part-load operating conditions are as much as 92% and 100% of the operation time for the first and second case study, respectively.

Although the project enjoyed good thermodynamic and environmental performance, Table 4 shows that the project is likely to be a commercial failure owing to the current relationship between electricity and natural gas prices in Poland. Both case studies show negative cash flow from the investment activities. Furthermore, all scenarios lead to negative annual EBITDA, which indicates that the investment has fundamental problems with both profitability and cash flow. Consequently, the project cannot be profitable without subsidies in the foreseeable economic and industry climate.

The income obtained as a consequence of implementation of the project could be increased by considering the gas distribution system operator as a prosumer, thanks to savings on the electric power annual invoice. However, current energy market regulations in Poland do not allow pipeline operators to become energy producers.

5. Conclusions

Currently, the commitment to improving energy efficiency and protecting the environment is present in all business decisions regarding natural gas midstream infrastructure planning. Waste energy recovery in pressure regulator stations contributes to power generation and, as such, promotes independence from external energy supplies, which follows the precepts of European Union law on the promotion of cogeneration. Conversion of hitherto wasted energy resources into power reduces the use of fossil fuels and lowers greenhouse gas emissions. Low CO_2 emission costs and falling prices for coal are contributing to a decrease in electricity prices, which in turn makes the investment considered here unprofitable. Current price relations between electricity produced from coal and from other energy sources can be varied through a decision-making process, which may improve the cost effectiveness of the discussed solution.

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