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Energy and exergetic analysis of the transmission system

Analiza energetyczna i egzergetyczna systemu przesyłowego

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Keywords: *exergy, energy analysis, gas transmission pipelines, exergy balance.*

Abstract
The article discusses the energy and exegetic analysis of the gas transmission system using real network.

Słowa kluczowe: *egzergia, analiza energetyczna, gazociągi przesyłowe, bilans egzergii.*

Streszczenie
W artykule na przykładzie omówiono analizę energetyczną i egzegetyczną gazowego systemu przesyłowego.

1. Introduction

Knowing the hydraulic characteristics of the network is a key element for the transmission system operator to optimally manage the network. Optimal management involves determining whether a given network, with transmission contracts currently in place, has the capacity to connect a new customer without degrading the quality of service provided to other counterparties. For this purpose, the operator prepares an energy balance, in which it calculates the reserve of power in the system and based on it, makes decisions on concluding new contracts. The energy balance, in terms of the transmission system, is understood as the sum of the power supplied at sources and compressor stations and the power lost at reducing stations and pipelines. However, determining the amount of energy in the network is not a sufficient measure in terms of practical usefulness. This is due to the fact that the process of compressing gas in the compressor station is accompanied by an increase in temperature, and the higher the temperature of the medium, the more valuable is its supplied heat. So, it is required to introduce a parameter that would characterize the energy in qualitative terms.

Exergy is a quantity that determines the maximum capacity to do work in relation to the surrounding nature (Szargut, 2013). Drawing up an exergy balance will reveal any Energy losses occurring in the network, and at the same time those resulting from the irreversibility of processes (Ciesielczyk, Skoneczna-Luczków, & Kurtyka, 2013). This is done only for the non-isothermal case, because in the isothermal case, the exergy flux is equal to zero.

The difference between the energy balance and the exergy balance is significantly large for the gas compression processes and other processes taking place near ambient temperature. At temperatures lower than the ambient temperature, the sign of the exergy gain is opposite to the sign of the energy gain. The higher the energy of the refrigerant, the lower its exergy (Szargut, 2013).

2. Methodology

A high-pressure network was adopted for consideration. The analysis was performed in two cases:

- isothermal
- non-isothermal

The study was conducted with the SimNet software package for static and dynamic simulation from Fluid Systems. The program is designed to simulate networks with one or multiple pressure levels, an unlimited number of sources and non-tube elements (e.g. compressor station, reduction station). Mathematical models describing isothermal and non-isothermal steady-state flow gas flow are presented in (Osiadacz & Chaczykowski, Comparison of isothermal and non-isothermal pipeline gas flow models, 2001).

General methods for simulating transient gas flow can be divided into two groups. The first group concerns methods in which the mathematical model is transformed to differential forms and then for implicit schemes to a system of nonlinear algebraic equations. The second group of methods transforms the partial differential equation into a system of linear algebraic equations through the use of finite volume theory. For transient simulation, the generalized node method was used, the algorithm of which is presented in (Osiadacz, Numerical analysis of a method of transient simulation for gas network, 1990).

The basic system of equations for simulating isothermal transients is described by relation (1)

$$\begin{cases} \frac{A}{C^2} \cdot \frac{\partial p}{\partial t} = -\frac{\partial M}{\partial x} \\ \frac{\partial p}{\partial t} = -\frac{2f\rho w^2}{D} \end{cases} \quad (1)$$

where:
A – cross-sectional area of the pipe (m^2),
C – speed of sound (m/s),
p – pressure (Pa),
t – time (h),
M – mass flow (kg/h),
x – distance (m),
2f – coefficient of resistance (-),
ρ – density (kg/m^3),
w – speed (m/s),
D – internal diameter of the pipe (m).

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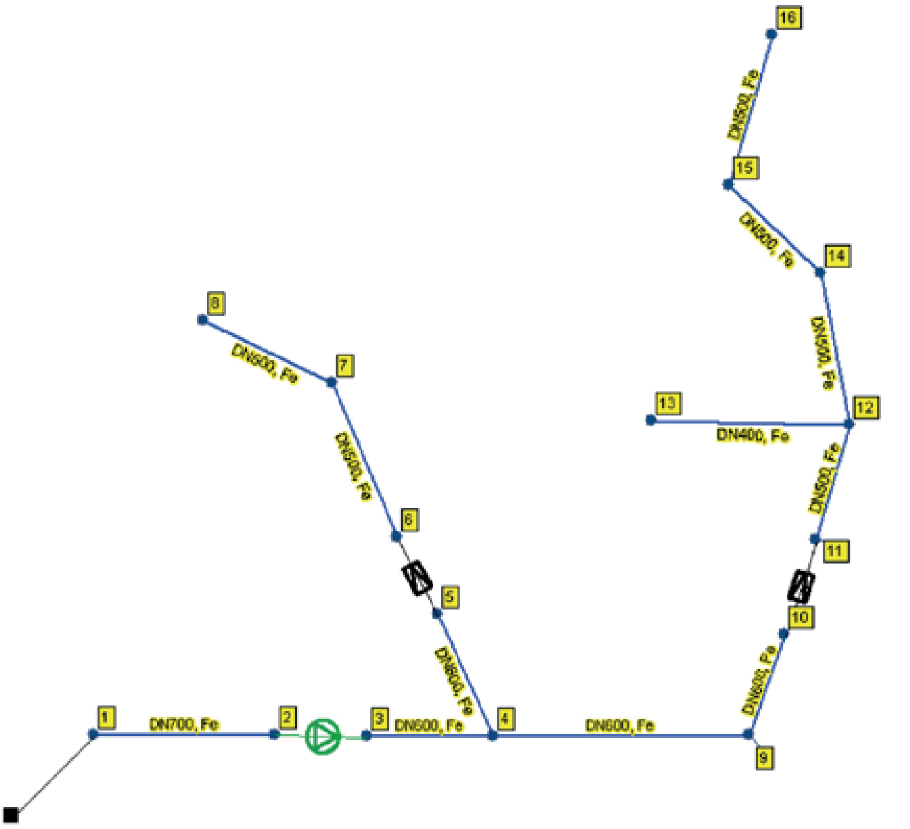

Fig. 1. Transmission network scheme – example [SimNet TSGas]

Rys. 1. Schemat sieci przesyłowej – przykład

In order to calculate the temperature distribution in the network, the energy equation was replaced by relation (2). This change greatly simplifies the transient model for non-isothermal transformation and speeds up the calculation process.

$$\theta(x) = \theta_E + \frac{\theta(x)+\theta_E}{Cx} \cdot (1 - e^{-Cx}) \tag{2}$$

where:
Φ – gas temperature (K),
Φ_E – ambient temperature (K),
C – constant containing the average value of the heat transfer coefficient between the gas and the environment ,
x – distance (m).

The transient simulation was carried out for 24 hours with a time step of 1 hour. The power and exergy values of the individual elements in the network were calculated for the states corresponding to the simulation results at the given time step with formulas (4) – (8).

3. CASE STUDY

The network consists of:

- 1 source;
- 1 compression station;
- 2 reduction stations;
- 12 pipes;
- 16 nodes.

The network scheme is presented in Fig. 1.

The energy balance is described by relation (3), while the elements included in the equation are described by relations (4) – (7).

$$P_z + P_T + P_{SR} + P_R = \Delta P \tag{3}$$

where:
P_z – power at the source (kW),
P_T – power at the compression station (kW),
P_{SR} – power loss at the reduction station (kW),
P_R – power loss in the pipelines (kW),
ΔP – power reserve (kW).

Power at the source

$$P_z = \frac{Q_n \cdot Z_{rz} \cdot p_n \cdot T_{rz} \cdot \Delta p}{Z_n \cdot p_{śr} \cdot T_n \cdot 3600} \tag{4}$$

where:
Q_n – volumetric flow under normal condition (m^3/h),
Z_{rz} – compressibility factor (-),
$p_{śr}$ – pressure (kPa),
T_{rz} – temperature (K),
Δp – pressure increase (kPa),
Z_n – compressibility factor under normal conditions, Z = 1 (-),
p_n – pressure under normal conditions, p_n = 101,325 (kPa),
T_n – temperature under normal conditions, T_n = 273,15 (K).

$$p_{śr} = \frac{2}{3} \cdot \left(p_1 + \frac{p_2^2}{p_1 + p_2}\right) (kPa) \tag{4.1}$$

where:
p_1 – gas suction pressure (kPa),
p_2 – gas discharge pressure (kPa).

Power at the compressor station

$$P_T = \frac{n \cdot p_n \cdot Q_n \cdot Z_s \cdot T_s}{\eta_m \cdot (n-1) \cdot T_n \cdot 3600} \cdot \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] (kW) \tag{5}$$

where:
n – polytropic exponent (assumed n = 1,35),
p_n – pressure under normal conditions, p_n = 101,325 (kPa),

Q_n – volumetric flow in the compressor (m^3/h),
Z_s – compressibility factor (-),
T_s – gas temperature on the suction side (K),
η_m – compressor mechanical efficiency (assumed $\eta_m = 0{,}8$),
T_n – temperaturę under normal conditions, $T_n = 273{,}15$ (K),
p_1 – gas suction pressure (kPa),
p_2 – gas discharge pressure (kPa).

Power loss at the reduction station

$$P_{SR} = \frac{Q_n \cdot Z_{rz} \cdot p_n \cdot T_{rz} \cdot \Delta p}{Z_n \cdot p_{śr} \cdot T_n \cdot 3600}\ (kW) \tag{6}$$

where:
Q_n – volumetric flow under normal conditions (m^3/h),
Z_{rz} – compressibility factor (-),
$p_{śr}$ – pressure (kPa),
T_{rz} – temperature (K),
Δp – pressure drop (kPa),
Z_n – compressibility factor under normal conditions, Z = 1 (-),
p_n – pressure under normal conditions, $p_n = 101{,}325$ (kPa),
T_n – temperature under normal conditions $T_n = 273{,}15$ (K).

Power loss in the pipelines

$$P_{SR} = \sum_{i=1}^{n} \Delta P_i\ (kW) \tag{7}$$

where:
ΔP_i – power loss on the i-th pipeline (kW),

Exergy flow

where:
B – exergy flow (kW),
P – power of the network element (compressor or reduction station) (kW),
T_{ot} – ambient temperature (K),
T – gas inlet temperature in the network element (compressor or reduction station) (K).

Exegetical balance equation

$$B = P \cdot \left(1 - \frac{T_{ot}}{T}\right)\ (kW) \tag{8}$$

where:
B_z – source exergy flow (kW),
B_T – compressor station exergy flow (kW),
B_{SR} – reduction station exergy flow (kW),
B_R – pipelines exergy flow (kW),
ΔB – exergy increase of the system (kW).

4. Results

The results from the static simulation for isothermal and for the non-isothermal scenarios are shown in Table 1 and Table 2 respectively.

The isothermal scenario

The following assumptions were made for the calculations:
- steady-state flow ($Q \neq f(t)$),
- the ambient temperature is equal to 4°C,
- heat exchange with the environment is not taken into account (T=const),
- gas pipelines are not inclined,
- gas flow is under high pressure (above 1600 kPa).

Table 1. Results of the steady-state simulation for the isothermal scenario
Tabela 1 Rezultaty symulacji statycznej dla przypadku izotermicznego

Data type		Source	Compressor station		R1 station		R2 station		Pipelines power loss	Power reserve	Units
Initial node	End node	1	2	3	5	6	10	11			-
Q_n		400 000,00			200 000,00		200 000,00				m^3/h
Z_{rz}		0,87			0,85		0,85				-
p_{rz}		6 000,00			4 947,42		5 400,62				kPa
T_{rz}		277,15			277,15		277,15				K
Q_{rz}		5 935,49			3 540,96		3 243,82				m^3/h
Δp		6 000,00			-1 763,00		-1 922,00				kPa
		P_z	P_T		P_{SR1}		P_{SR2}		P_R	ΔP	
$\Delta p \times Q_{rz}$		9 892,49	5 486,87		-1 734,09		-1 731,84		-4 817,00	**7 096,43**	kW

The non-isothermal scenario

The following assumptions were made for the calculations:
- steady-state flow ($Q \neq f(t)$),
- heat exchange between the pipe walls and the surroundings is taken into account ($T \neq$ const),
- gas pipelines are not inclined,
- gas flow is under high pressure (above 1600 kPa),
- the ambient temperature is equal to 0°C,
- the source outlet temperature is equal to 10°C ,
- the compressor station outlet temperature is equal to 60°C ,
- the reduction station outlet temperature is equal to 7°C .

Table 2. Results of the steady-state simulation for the non-isothermal scenario
Tabela 2. Rezultaty symulacji statycznej dla przypadku nieizotermicznego

Data type		Source	Compressor station		R1 station		R2 station		Pipelines power loss	Power reserve / Exergy increase	Units
Initial node	End node	1	2	3	5	6	10	11			-
Q_n		400 000,00			200 000,00		200 000,00				m^3/h
Z_{rz}		0,86			0,84		0,84				-
p_{rz}		6 000,00			4 985,29		5 439,48				kPa
T_{rz}		283,15			271,95		288,15				K
Q_{rz}		6 021,98			3 407,66		3 309,17				m^3/h
Δp		6 000,00			-1 776,00		-1 935,00				kPa
		P_z	P_T		P_{SR1}		P_{SR2}		P_R	ΔP	
$\Delta p \times Q_{rz}$		10 036,63	5 328,76		-1 681,11		-1 778,68		-4 894,00	7 011,60	kW
		B_z	B_T		B_{SR1}		B_{SR2}		B_R	ΔB	
Exergy flow		354,46	39,30		-7,42		-92,59		-264,03	33,43	kW

The results from the transient simulation for the isothermal scenario are shown in Table 3. as for the non-isothermal scenario the results are presented in Fig. 2 and 3. Gas demand changes at the nodes are shown in Fig. 4.

The isothermal scenario

The following assumptions were made for the calculations:
- transient flow ($Q = f(t)$),
- the ambient temperature is equal to 4°C,
- heat exchange with the environment is not taken into account (T=const),
- gas pipelines are not inclined,
- gas flow is under high pressure (above 1600 kPa).

Table 3. Results of the transient state simulation for the isothermal scenario

Tabela 3 Rezultaty symulacji dynamicznej dla przypadku izotermicznego

	Source	Compressor station	R1 station	R2 station	Pipelines power loss	Power reserve / Exergy increase	Units
Time	P_z	P_T	P_{SR1}	P_{SR2}	P_R	ΔP	
00:00	7527,42	3814,00	-903,78	-1062,64	-1481,90	7893,11	kW
01:00	7475,33	3730,00	-873,48	-1056,52	-483,47	8791,86	kW
02:00	7448,58	3829,00	-1003,39	-1056,27	-498,12	8719,80	kW
03:00	7517,29	3924,00	-1142,72	-1052,09	-522,27	8724,21	kW
04:00	7636,22	4027,00	-1282,84	-1048,10	-557,05	8775,23	kW
05:00	7788,90	4140,00	-1423,78	-1044,53	-603,11	8857,48	kW
06:00	7967,05	4261,00	-1498,32	-1040,76	-661,12	9027,85	kW
07:00	8165,10	4390,00	-1706,67	-1037,93	-732,05	9078,45	kW
08:00	8366,28	4468,00	-1725,30	-1037,37	-786,57	9285,05	kW
09:00	8550,21	4564,00	-1724,72	-1144,57	-869,99	9374,93	kW
10:00	8738,49	4694,00	-1719,61	-1265,66	-990,08	9457,14	kW
11:00	8948,38	4850,00	-1713,50	-1423,15	-1139,19	9522,55	kW
12:00	9186,63	5027,00	-1706,56	-1678,72	-1345,21	9483,13	kW
13:00	9451,75	5214,00	-1700,64	-1744,55	-1628,77	9591,79	kW
14:00	9744,11	5449,00	-1817,74	-1826,61	-1974,34	9574,42	kW
15:00	19530,91	5894,00	-1968,96	-1893,10	-3269,31	18293,53	kW
16:00	14184,64	6604,00	-2137,20	-1962,80	-3782,07	12906,57	kW
17:00	13763,41	6892,00	-2050,42	-2034,43	-4228,07	12342,50	kW
18:00	13677,19	6975,00	-1914,53	-1984,93	-4373,09	12379,64	kW
19:00	13675,86	6967,00	-1776,45	-1896,46	-4266,67	12703,29	kW
20:00	13678,07	6906,00	-1640,76	-1793,07	-4020,47	13129,76	kW
21:00	13653,18	6807,00	-1504,82	-1681,80	-3700,04	13573,52	kW
22:00	10598,47	6637,00	-1364,75	-1609,64	-3085,77	11175,31	kW
23:00	9111,98	6260,00	-1209,57	-1543,82	-2506,52	10112,08	kW
00:00	7873,61	5755,00	-1053,30	-1378,55	-1862,78	9333,98	kW

The non-isothermal scenario

The following assumptions were made for the calculations:

- transient flow (Q = f(t)),
- heat exchange between the pipe walls and the surroundings is taken into account (T ≠ const),
- gas pipelines are not inclined,
- gas flow is under high pressure (above 1600 kPa),
- the ambient temperature is equal to 0°C,
- the source outlet temperature is equal to 10°C ,
- the compressor station outlet temperature is equal to 60°C,
- the reduction station outlet temperature is equal to 7°C.

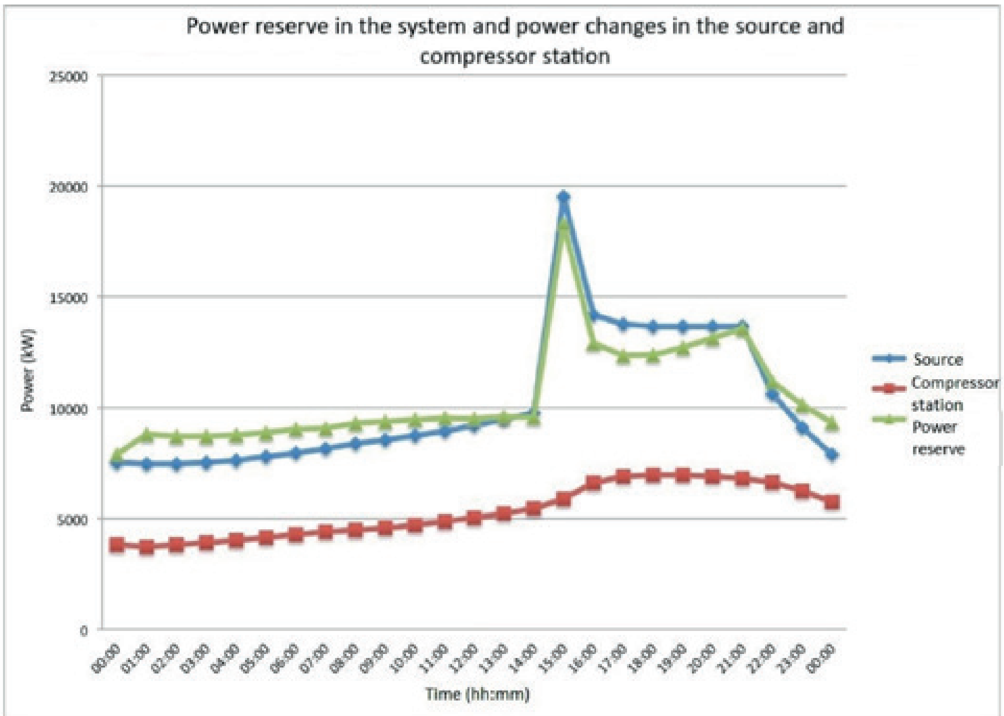

Fig. 2. Results of the transient state simulation for the non-isothermal scenario.

Rys.2. Rezultaty symulacji dynamicznej dla przypadku nieizotermicznego.

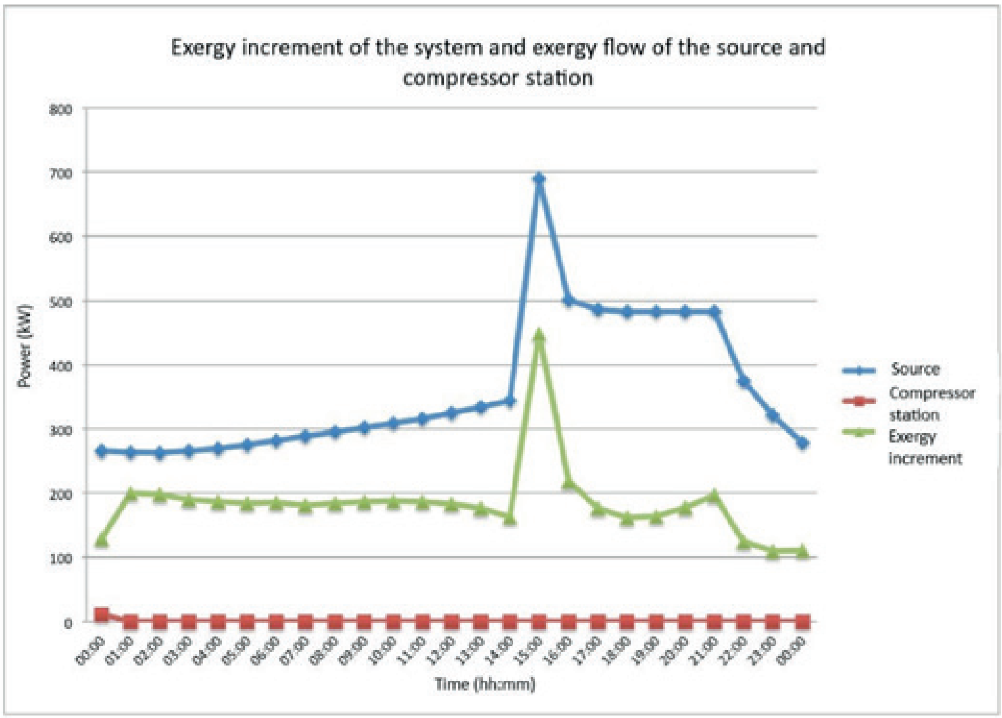

Fig. 3. Results of the transient state simulation for the non-isothermal scenario

Rys.3. Rezultaty symulacji dynamicznej dla przypadku nieizotermicznego

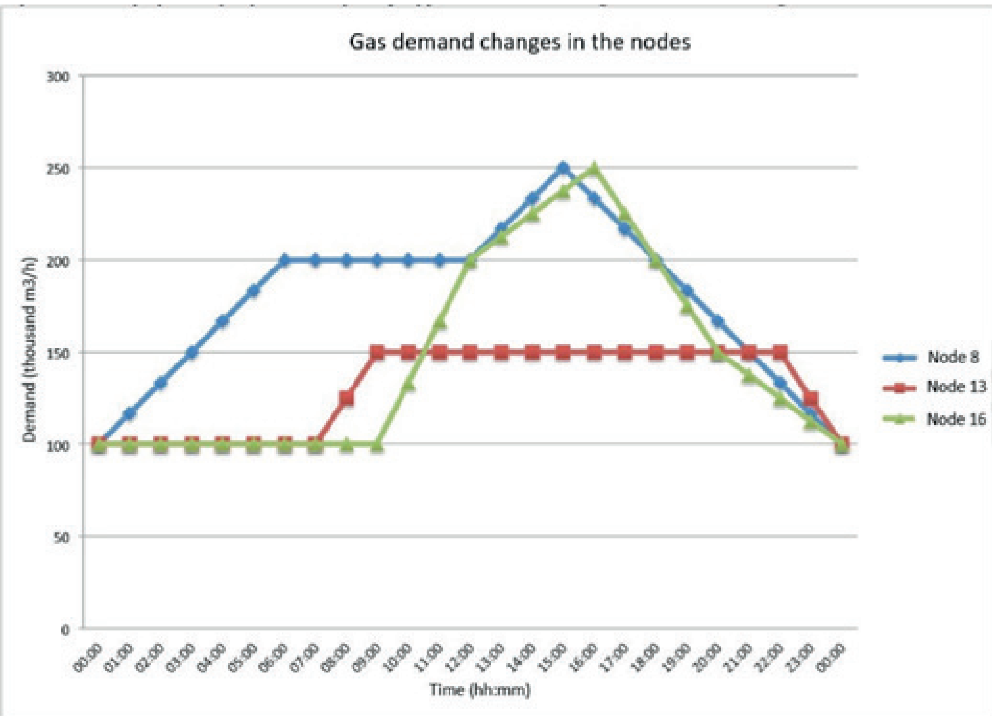

Fig. 4. Results of the transient state simulation for isothermal and non-isothermal scenario

Rys.4. rezultaty symulacji dynamicznej dla przypadku izotermicznego i nieizotermicznego

Results

Table 4 and 5 summarize the calculation results from both simulation cases.

For the given network and its physical parameters, it came out in case of the steady state simulation that the larger power reserve is in the isothermal scenario (46,15%) than in the non-isothermal scenario (45,63%). On the other hand, the transient state simulation for the 18:00 data shows that the greater power reserve is in the non-isothermal case (59,94%) than in the isothermal case (57,79%). The differences are due to the adopted gas temperature, which for the isothermal scenario each grid element is constant, while in the non-isothermal scenario each grid element varies

Table 4 Steady-state simulation summary

Tabela 4 Zestawienie wyników symulacji statycznej

Data type		Source	Compressor station		R1 station		R2 station		Pipelines power loss	Power reserve / Exergy increment	Units
Initial node	End node	1	2	3	5	6	10	11			-
		P_z	P_T		P_{SR1}		P_{SR2}		P_R	ΔP	
Isothermal		9 892,49	5 486,87		-1 734,09		-1 731,84		-4 817,00	**7 096,43**	kW
Non-isothermal		10 036,63	5 328,76		-1 681,11		-1 778,68		-4 894,00	**7 011,60**	kW
		B_z	B_T		B_{SR1}		B_{SR2}		B_R	ΔB	
Exergy flow		354,46	39,30		-7,42		-92,59		-264,03	**33,43**	kW

Table 5. Transient-state simulation summary

Tabela 5. Zestawienie wyników symulacji dynamicznej

Data type	Source	Compressor station	R1 station	R2 station	Pipelines power loss	Power reserve / Exergy increment	Units
Initial node / End node	1	2 / 3	5 / 6	10 / 11			-
	P_Z	P_T	P_{SR1}	P_{SR2}	P_R	ΔP	
Isothermal	13 296,72	7 141,00	-1 859,49	-1 981,69	-4 786,17	**11 810,37**	kW
Non-isothermal	13 677,19	6 975,00	-1 914,53	-1 984,93	-4 373,09	**12 379,64**	kW
	B_Z	B_T	B_{SR1}	B_{SR2}	B_R	ΔB	
Exergy flow	483,04	0,00	-81,85	-0,73	-239,01	**161,45**	kW

from each other. Ambient temperature boundary conditions for both scenarios should be taken into account, which have high impact on non-tube elements.

Conclusions

It can be concluded from the analyzed case study that the isothermal scenario is a reliable approach for managing and developing the transmission network. The implementation of the isothermal flow model is much simpler than the non-isothermal model, which is a huge advantage. However, there is no substitute for the non-isothermal model, as it is important to monitor the network for hydrate formation. Calculation from the isothermal model can be treated as a preliminary outline of the processes taking place in the network, while calculation from the non-isothermal model should be used to effectively manage and develop the transmission network. The non-isothermal model allows to **qualitative examination of** the power of the elements in the system also it helps to find points in the network that cause significant drops in quality which can provide a space to improve the thermal performance in such points.

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