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THE EFFECT OF HYDROGEN TRANSPORTED THROUGH A GAS PIPELINE ON THE FUNCTIONING OF GAS COMPRESSION STATION WORK***

1. INTRODUCTION

The natural gas transmission in Poland is based on 14 natural gas compression stations of the Transmission System Operator (Rembelszczyzna, Hołowczyce I and II, Wronów, Kotowo, Goleniów, Jeleniów I and II, Pogórska Wola, Mirocin, Jarosław I and II, Maćkowice, Lubaczów), and a few new ones are under construction [1]. Their main function is the compression of natural gas to an appropriate pressure, thanks to which the energy carrier can be transmitted over long distances. The most important element of the compression station are compressors. They can be divided in view of the character of flow, as in Figure 1 [2–4]. Piston and centrifugal compressors are most frequently used in compression stations [5]. When deciding about the particular type of compressor, the main role is played by the character of flow of the transmitted gas. If the transport is assumed to be realized on a constant basis, at constant high transmission rates (over 5 m/s), and the compression degree exceeds 2.5, the centrifugal compressor will operate best. In the case of lower rates or lack of constant speed, and in the case of higher compression degrees, the piston variant will be more favorable. Hence, the gas compression stations in Poland on the Yamal-Europe pipeline route are equipped with gas machines compressing gas centrifugally, whereas most of compression stations belonging to the Transmission System Operator are equipped with piston compressors [3].

At present the constantly and dynamically increasing use of energy from renewable sources can be observed globally, especially in terms of electrical energy from wind and photovoltaic farms. Also, other EU countries have seen a significant development of renewable energy investments, which is in compliance with EU energy priorities [6, 7]. The energy

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technologies based on renewable energy sources have very dynamically increased in recent years. In 2005, the participation of wind farms and photovoltaic farms in the EU installed power equaled to 6% and 0.3%, respectively, to increase to 16.7% and 11% in 2016, respectively. The analysis of the structure of installed power in EU countries in 2016 reveals that wind farms (16.7%) occupy the second position after natural gas-powered utilities (20.3%). It is also Poland where a considerable increase of power (especially wind energy) has been observed. At the end of 2016 the installed power in wind farms totaled 5782 MW, placing Poland in 7th position in EU [8]. With the development of investments and the relating increase of wind energy, the participation of wind farms also systematically increases in the structure of home electrical energy production. In 2010 this participation equals only 0.8%, to increase to 7.1% in 2016 [9]. These technologies are associated with unevenness and the instability of electrical energy generation, and consequently – problems with storing energy. In light of the promotion of low-emission fuels, attention should be paid to the fact that this gas can be produced from the possible excess of energy from these sources. Energy in the form of gas can be better transported or stored. One of the possibilities is transporting it as a mixture with natural gas through the existing networks. The analyses reveal that hydrogen in certain amounts does not have any significant influence on the general level of hazard. It has a broader range of parameters affecting the flammability and explosiveness as compared with natural gas. Based on literature data, the authors [10] claim that adding small amounts of hydrogen to the existing transmission system, i.e. 20% or less, barely affects the general explosiveness hazard. A similar amount of gas escapes through the pipeline walls, and owing to the higher mobility of natural gas components, constitutes most of the emitted molecules, limiting the emission of greenhouse gases, i.e. methane. Other analyses show that the maximum amount of hydrogen which would not have any significant impact on the safety of transport and use of the mixture, totals 10% [11], 20–30% [12, 13] and even 50% or more [14], depending on the assumed criteria. The authors [12] stress that the gas installation can be constantly exposed to hydrogen (50%) for many years in distribution networks in which city gas has been used. Poland is just such a case [15].

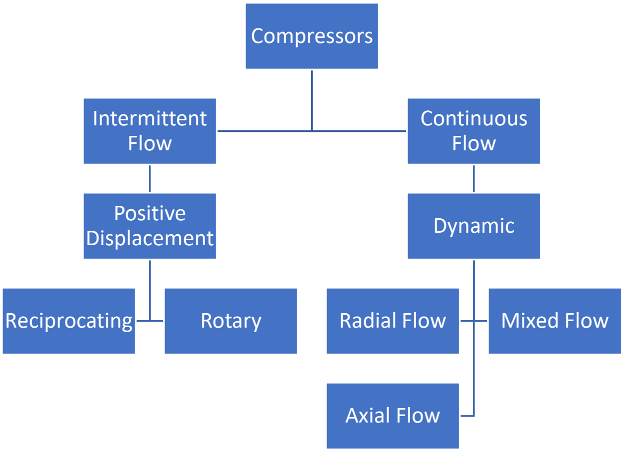

Fig. 1. Types of compressors, after [3]

Compression of gases is strictly connected with their physicochemical parameters. Hydrogen considerably differs from the remaining natural gas components, therefore tests were made on the influence of an additional component of the transmitted mixture on the compression. The compression strongly depends on the gas temperature on entry, therefore a temperature of 7°C was assumed which corresponds to the average annual observed temperature.

The entry composition of the analyzed gas was assumed after [16]:
- 96% methane,
- 1% ethane
- 3% nitrogen.

With the increase of hydrogen content in the mixture the amount of the remaining components proportionately changes, in line with equation (1).

$$x_{n\_new} = x_{n\_i}(1 - x_{H_2}) \tag{1}$$

where:

x_{n_new} – molar participation of *n*-th component at particular participation of hydrogen,
x_{H_2} – hydrogen content in the mixture,
x_{n_i} – initial participation of *n*-th component.

2. MAIN WORK PARAMETERS OF COMPRESSORS

At the selection stage, the most important parameters on the basis of which the compressor is purchased are: flow rate, gas composition, pressure and temperature on suction, discharge pressure, construction parameters (for piston compressors: number of cylinders, cooling, flow control mechanism) and number of units [17]. The main parameters characterizing compression are compression ratio, isentropic or polytropic efficiency, discharge temperature and compression labor transformed on theoretical power needed to compress a given amount of gas.

As mentioned in the Introduction, most of the compressors used for natural gas transmission are piston compressors and, for this reason, this paper is based on calculations made for these compressors. Their principle lies in increasing pressure by the reciprocating movement of the piston in the cylinder. They may work in a single- or multi-stage regime [18]. The single-stage compression lies in a single raise of pressure in the compressor from suction pressure to discharge pressure. In the case of a two-stage compression, the gas pressure is raised in the compressor twice, each time at a similar compression ratio. The compression ratio expresses a proportion of discharge pressure (outlet) and suction pressure (entry).

Gas compression is based on the polytropic process, in which heat transfer is realized through the compressor parts. It can be increased by cooling systems. Owing to the fact that the polytropy exponent characterizing the process is different for each compressor and different conditions, the adiabatic character of compression was assumed. Exemplary data coming from actual gas compression stations are given in Table 1. The temperature after compression and temperature calculated according to the procedure presented further in this paper are compared. In most cases, the difference between calculated and measured values amounts to approximately 10%.

Table 1

Comparison of temperatures after compression measured in real gas compression stations and calculated on the basis of equations describing an adiabatic process (Q – flow intensity, p_1 – suction pressure, p_2 – discharge pressure, T_1 – temperature on suction, T_2 – discharge temperature)

Compressor no.	Sample no.	Measured values						Calculated	Difference	
		Q [m^3/h]	p_1 [MPa]	p_2 [MPa]	p_2/p_1 [–]	T_1 [°C]	T_2 [°C]	T_2 [°C]		
1	1	31345	2.21	4.70	2.12	14.00	74.00	67.53	6.47	9%
1	2	80944	3.53	4.81	1.36	17.00	43.00	38.54	4.46	10%
1	3	71186	3.27	4.89	1.50	17.00	51.00	45.18	5.82	11%
2	4	64604	3.73	4.81	1.29	9.00	32.00	26.29	5.71	18%
2	5	77107	3.06	4.90	1.60	9.00	46.00	41.45	4.55	10%
2	6	49578	2.68	4.80	1.79	15.00	63.00	56.03	6.97	11%
3	7	44955	2.02	4.75	2.36	13.30	85.20	74.64	10.56	12%
3	8	56819	2.53	4.74	1.87	13.70	69.80	57.88	11.92	17%

Gas can be compressed when the compressors perform their work. The ratio of work put in isentropic process to real work in the same conditions is defined as the isentropic efficiency of a compressor η_{is}. As already mentioned, in practice compressors are purposefully cooled to minimize the work input. In such cases the process in the compressor can be considered as isothermal and the isentropic efficiency can be substituted with isothermal efficiency, referred to the irreversible isothermal process. For comparison's sake, both the polytropic process (heat transfer is possible), and isentropic process (no heat transfer) can both be referred to [19]. The difference lies in the fact that the polytropic process is based on the same discharge temperature, both for the real process and when the discharge temperature in the isentropic process is different (lower) [20], as visualized in Table 1. For a polytropic process we can determine the efficiency on the same basis as in the case of isentropic and isothermal processes.

The adiabatic exponent is constant for a given adiabatic process. For the sake of calculating the discharge temperature, equation (2), based on adiabatic process equations, was used:

$$T_2 = T_1\left(r_p \frac{\kappa-1}{\kappa}\right) \tag{2}$$

where:

T_1 – gas temperature on suction [K],

r_p – compression ratio, $r_p = \dfrac{p_2}{p_1}$,

p_1, p_2 – suction (1) and discharge pressure (2) [Pa],

κ – adiabatic exponent.

The final most important parameter of each compressor is its power. Using equation (3) we can define the BHP (brake horsepower) needed for a single compression cycle, expressed in horsepower [4, 20]. The parasitic efficiency E expresses mechanical losses and pressure losses in valves and pulsation dampers (lower efficiencies are usually connected with low compression ratios).

$$BHP = 0.653 \cdot Z_{ave} \left[\frac{(Q_n)(T_1)}{E \cdot \eta} \right] \left[\frac{\kappa}{\kappa - 1} \right] \left[\left(\frac{p_2}{p_1} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right] \tag{3}$$

where:

- BHP – brake horsepower per stage,
- Z_{ave} – average supercompressibility factor,
- Q_n – gas flow intensity in normal conditions [m_n^3/s],
- T_1 – temperature of gas on suction [K],
- p_1, p_2 – suction pressure (1) and discharge pressure (2) [Pa],
- E – parasitic efficiency (for high-speed reciprocating units assumed as 0.72–0.82; for low-speed reciprocating units assumed as 0.72–0.85),
- η – compression efficiency (for piston compressors 1.0),
- κ – adiabatic exponent.

3. EFFECT OF HYDROGEN ADDITION ON THE MAIN COMPRESSION PARAMETERS

Knowing that the parameters of hydrogen considerably differ from the parameters of the remaining natural gas components, the authors performed calculations of basic parameters characterizing gases in reference to the share of hydrogen in the whole mixture. An adiabatic character of the transformation was assumed. The adiabatic exponent closely depends on the heat capacities referred to isochoric process c_v and isobaric process c_p, as in equation (4). These parameters assume various values for each substance depending on temperature. The analysis of the list in [4] attention was paid to the fact that their change can be described as a linear function of temperature. It should be stressed that hydrogen has considerably higher thermal capacities (at 10°C: 14.23 $kJ \cdot kg^{-1} \cdot K^{-1}$) in reference to the mass as compared to other gases analyzed (c_p methane in 10°C: 2.20 $kJ \cdot kg^{-1} \cdot K^{-1}$).

$$\kappa = \frac{C_p}{C_v} \tag{4}$$

where:

- C_p – heat capacity in isobaric process [kJ/(kg·K)],
- C_v – heat capacity in isochoric process [kJ/(kg·K)].

Making use of equations (1), (2) and (4), the plots illustrating the dependence of compression ratio on discharge temperature for definite hydrogen contents in the mixture were drawn.

The major parameter influencing the changes of temperature in the compression, as in equation (2), is the adiabatic exponent. As the exponent assumes higher values for a mixture with higher hydrogen content, then the higher the hydrogen content in the whole mixture is, the higher will be the discharge temperature values. Figures 2 and 3 show the relation between the compression ratio and discharge temperature after compression for mixtures of natural gas and hydrogen 0 to 15% with a 5% step, at gas temperature on suction of 7°C. Much higher compression ratios can be obtained for particular temperatures on suction for natural gas without a hydrogen admixture. Analogously, a high compression ratio will result in a considerably higher difference between suction and compression for higher hydrogen contents. It is worth noting that with the increase in the amount of added hydrogen, the difference of temperatures between successive steps decreases.

The compression finally brings about higher transmission pressures, and this is closely connected with a change in the volume of the transmitted gas. The basic adiabatic equation is expressed with equation (5). This relation is constant for the whole process, therefore the ratio of gas volume after and before compression was calculated with equation (6), being a process of (5). The results are presented in the form of a plot in Figure 4. As compared to previously calculated data, the differences between different hydrogen contents are not so large. For a compression ratio of 2.5, the volume ratio may be higher by 8% for natural gas without hydrogen than in the case of a 30% H_2 admixture.

$$pV^{\kappa} \tag{5}$$

where:

p – gas pressure during adiabatic process [Pa],
V – gas volume during adiabatic process [m^3].

$$\frac{V_1}{V_2} = \sqrt[\kappa]{\frac{p_2}{p_1}} \tag{6}$$

where:

p_1, p_2 – gas pressure before (1) and after (2) adiabatic compression [Pa],
V_1, V_2 – gas volume before (1) and after (2) adiabatic compression [m^3].

Calculations presented above and those performed with the use of equation (3) clearly show to that the process in which a mixture with a higher hydrogen content is compressed, is more demanding. The results suggest that from the compressors' point of view it is easier to use natural gas without any hydrogen admixtures. After re-calculating the obtained results with equation (3) on SI units, the data were presented in the form of a plot in Figure 5. An average flow of 88,000 m_n^3/h was obtained from one compressor in a real natural gas compression station, and a suction pressure of 1 MPa. The parasitic efficiency of the compressor was assessed at 78%. The higher the required compression ratio is, the bigger is the difference of required compression power for gas of assumed flow intensity between natural gas without and with hydrogen. For a compression ratio of 2.5, the difference between natural gas and its mixture with 30% H_2 totals almost 200 kW. This value considerably increases with the growth of hydrogen participation and the compression ratio; however the increase of hydrogen content in the mixture is accompanied by a drop in compressor power.

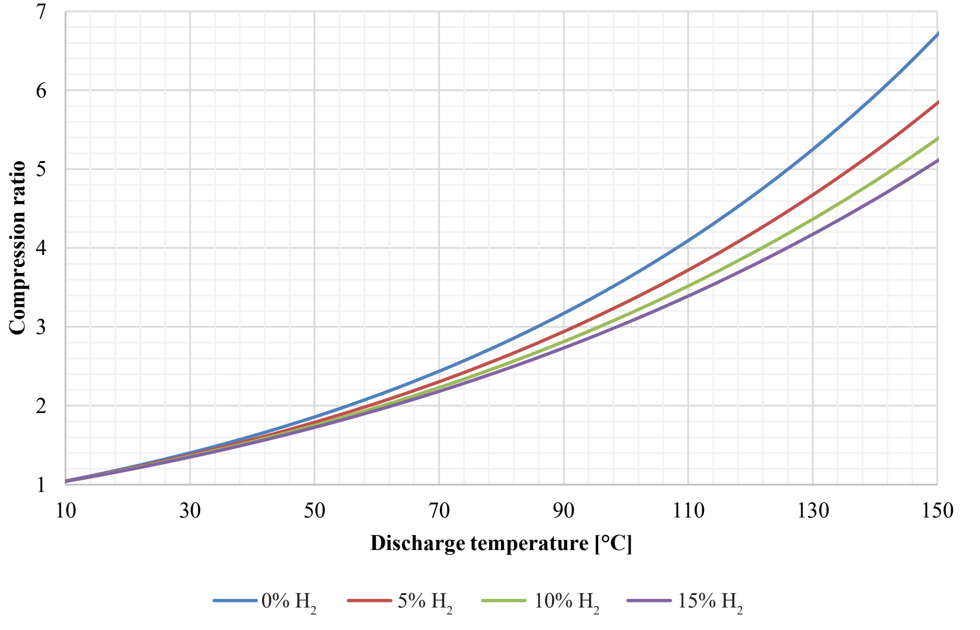

Fig. 2. Compression ratio vs. adiabatic discharge temperature for natural gas with various hydrogen admixtures

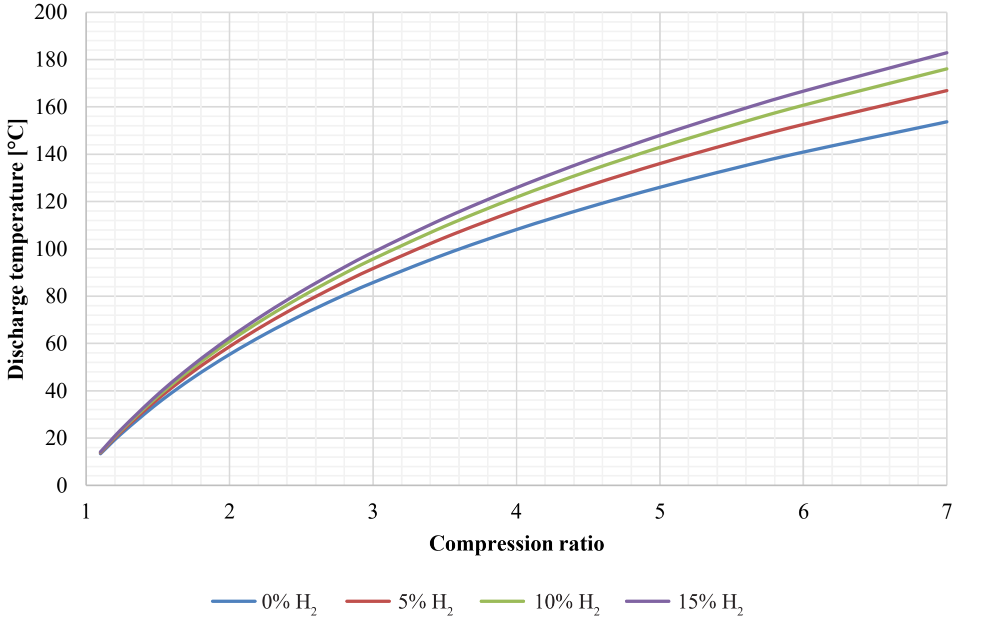

Fig. 3. Temperature after adiabatic compression vs. compression ratio for natural gas with different hydrogen admixtures

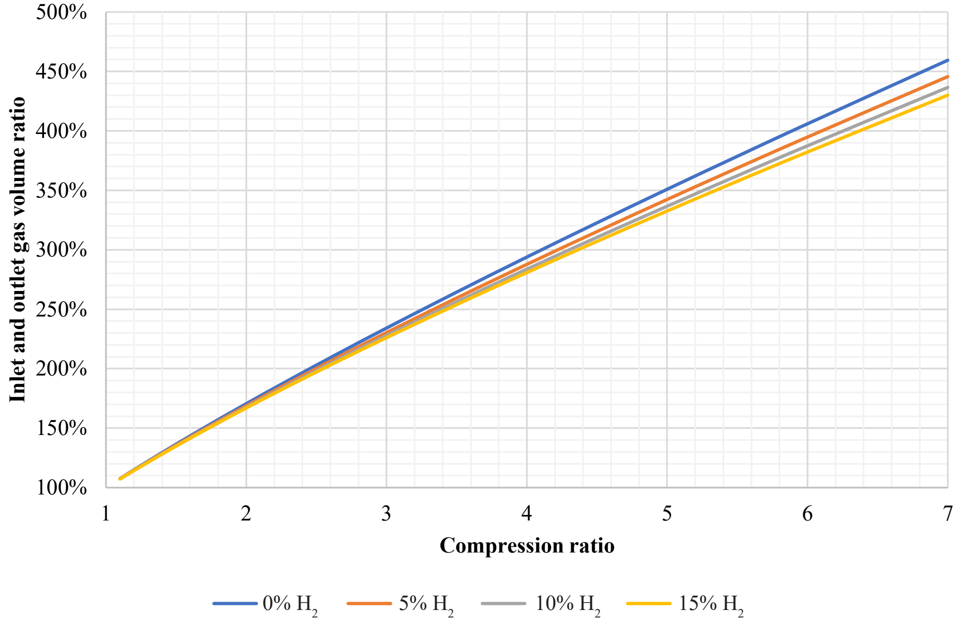

Fig. 4. Gas volume before and after adiabatic compression vs. compression ratio for natural gas with different hydrogen admixtures

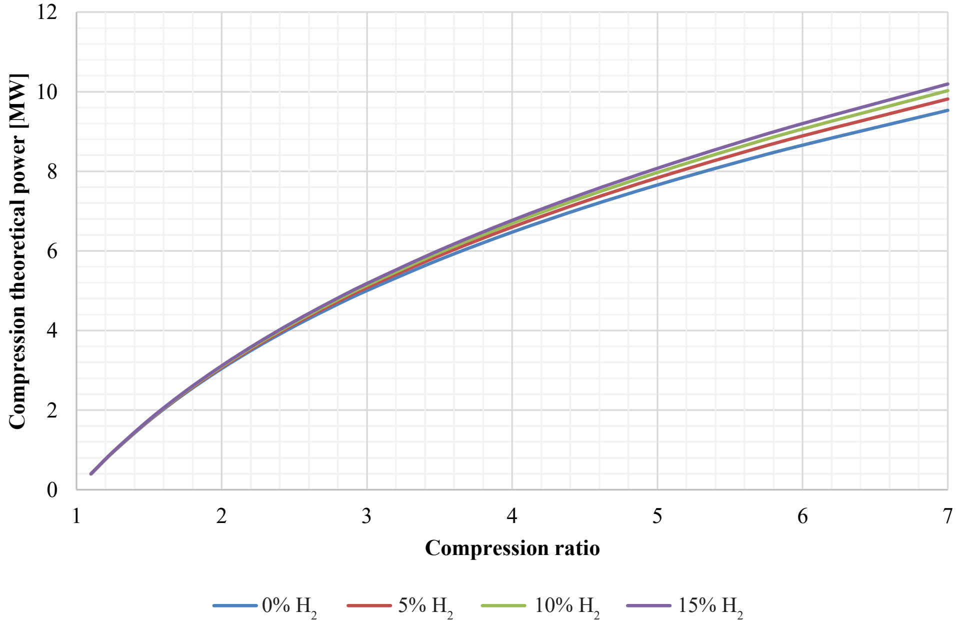

Fig. 5. Theoretical adiabatic compression power vs, compression ratio for natural gas with different hydrogen admixtures

4. CONCLUSIONS

The difference between the physicochemical parameters of hydrogen and other natural gas components significantly affects the character of transformations associated with gas compression. No wonder that considerable attention has been paid to hydrogen, as evidenced by the dynamic development of fuel cells, the introduction of cars powered with them and new hydrogen refueling stations. The possibility of adding hydrogen to the gas transmission was studied, although further measurements in real conditions should be carried out to confirm our theoretical considerations.

A few important conclusions should be highlighted after analyzing the characteristics of gas compression in a function of the quantity of added natural gas. Firstly, the comparison of the obtained results for natural gas without hydrogen with the results obtained for natural gas with the increasing participation of H_2 clearly reveal that the addition of this gas has a negative impact on the work parameters of piston compressors. This is mainly connected with the characteristically high heat capacity values for hydrogen, which result in high, as compared to natural gas, differences of temperatures between suction and compression and the required compressor power. What is important, the increase of energy demand decreases with the increase of hydrogen participation in the mixture.

On the other hand, it is hard to ignore the aspect of transmission of the gas mixture through the pipelines. Owing to the lower resistance, and consequently lower pressure losses, much higher pressure has been observed for the mixture than for the natural gas itself over the same distance [16]. The influence of temperatures after compression and the relation between the compressor power on the compression ratio mean that this issue is of the utmost importance. Although in the comparison of natural gas with and without hydrogen admixture, 'clean' natural gas turns out to be more advantageous, the situation may change when we consider real conditions in the transmission network and pressure losses in the pipeline.

It should be stressed that the effect of hydrogen permeability on the construction materials of the compressor installation, much higher than in the case of natural gas, was not discussed in this paper. Another problem is the increasing corrosiveness of the system when the gaseous mixture is used over a long time. Hence, when making analyses of real compression systems, attention should be paid to all aspects connected with the physicochemical characteristic of hydrogen. A full set of analyses of the long-term impact of the mixture on the designed system is recommended.

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