Flagship Project
TransHyDE

Compression and hydrogen transport: A technical and economic analysis

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Introduction and motivation

Demand for hydrogen transport capacity is set to increase over the coming years. Reasons include a growing use of renewables, the production of hydrogen from such energy sources and the import of hydrogen as an industrial raw material and energy carrier.

This growing demand for transport capacity will have to be met by a pipeline infrastructure which takes into account the technical differences between natural gas and hydrogen by pipeline. The compression of hydrogen in particular is a new technical challenge. Given its physical properties, such as its low molar mass, compression is technically more demanding and requires more energy.

This white paper explores various concepts for the compression of hydrogen for pipeline transportation and analyses technical and economic aspects such as capital expenditure and operating costs as well as different types of prime movers. It also provides an overview of compressor technologies (dynamic compressors, positive displacement compressors) suitable for hydrogen.

Following on from this technical and economic assessment, it offers a number of recommendations for the use of compressor technologies to meet potential transport requirements.

The findings obtained are used for simulations based on exemplary fluid mechanical calculations to derive generally valid statements on the use of compressors in hydrogen networks. In the course of these calculations, parameters such as pipe diameter and transported capacity are varied.

2

Hydrogen transport in pipelines

This section provides an overview of the physical variables that have a major impact on hydrogen transport in pipelines as well as a comparison of variables for hydrogen and natural gas with a view to categorising transport properties.

Table 2.1 shows variables that play a central role in hydrogen transport along with corresponding values for a representative natural gas composition (group H natural gas from Norway) listed here for comparison.

Table 2.1. Important physical variables of hydrogen and natural gas.

Variable	Unit	Hydrogen	Group H gas from Norway
Molar mass	g/mol	2.01	18.2
Gross calorific value	kWh/m³	3.54	11.6
Wobbe number	kWh/m³	13.4	14.7
Velocity of sound[1]	m/s	1,326	388
Gas law coefficient[1]	-	1.03	0.87
Isentropic exponent[1]		1.46	1.33

Hydrogen is the first element in the periodic table and has the lowest molar mass of all elements. At around 2 g/mol, the mass of hydrogen is around one ninth of the mass of natural gas. Among other things, the molar mass influences the maximum achievable pressure ratio where turbocompressors are employed (see section 3.1.1). The volumetric energy content of hydrogen (gross calorific value) is also significantly lower than that of natural gas – in the above case by a factor of around three.

The effects of these physical variables on important transport properties of hydrogen are explained below. Further information can also be found in the article entitled *Aktuelle Fragestellungen beim leitungsgebundenen Transport von Wasserstoff* [1].

2.1. Energy transport capacity

The maximum volumetric flow in a pipeline and hence its energy transport capacity can be determined via a calculation of the pressure drop along the pipeline. A formula commonly used is the Darcy-Weisbach equation.

[1] At 283.15 K (10 °C) and 50 bar

$$p_1^2 - p_2^2 = \lambda \cdot \frac{L}{D_5} \cdot \frac{16}{\pi^2} \cdot \rho_n \cdot \frac{T}{T_n} \cdot p_n \cdot K \cdot Q_n^2 \qquad (2.1)$$

Formula 2.1. Quadratic pressure drop equation [2].

Where:
D: Inner diameter
L: Length
K: Gas law deviation coefficient
p_1: Inlet pressure
p_2: Outlet pressure
p_n: Normal pressure
Q_n: Volumetric flow rate at standard conditions
T: Temperature
T_n: Normal pressure
λ: Friction factor
ρ_n: Normal density

The quadratic pressure drop equation can be used to drive the following simplified formula for the energy transport capacity of a pipeline [3]:

$$\dot{E} \sim C \cdot \frac{W_s}{\sqrt{K}} \qquad (2.2)$$

Formula 2.2. Energy transport capacity of a pipeline.

The energy transport capacity is determined by a constant C, the Wobbe number W_s and the gas law deviation coefficient K of a fuel gas. The constant represents the fixed properties of a pipeline: length, diameter, pressure rating, etc. The Wobbe number is a measure of the interchangeability of fuel gases. In formula 2.3 it is defined as the quotient of the gross calorific value and the square root of the relative density of a fuel gas.

$$W_s = \frac{H_{s,N}}{\sqrt{d_N}} \qquad (2.3)$$

Formula 2.3. Wobbe number.

The gas law deviation coefficient is defined as the quotient of the compression factor Z at flowing conditions and the compression factor at normal conditions Z_N.

$$K = \frac{Z}{Z_N} \qquad (2.4)$$

Formula 2.4. Gas law deviation coefficient.

The quotient of the energy transport capacity for hydrogen and natural gas expresses the degree of relative change in transport capacity.

$$\frac{\dot{E}_{H_2}}{\dot{E}_{EG}} = \frac{W_{s,H_2}}{W_{s,EG}} \cdot \sqrt{\frac{K_{EG}}{K_{H_2}}} \qquad (2.5)$$

Formula 2.5. Ratio of energy transport capacities.

Use of the aforementioned representative natural gas composition renders results in fig. 2.1.

The results show that the ratio of energy transport capacities is between 80 and 90 per cent. The greater the average pipeline pressure, the lower the ratio.

This pressure-dependent behaviour can be easily derived from formula 2.5: The quotient of the Wobbe numbers is constant. However, the ratio of the gas law deviation coefficients is largely dependent on the pressure. In addition, the gas law deviation coefficient of hydrogen is greater at high pressures, while that of natural gas is lower. Consequently, the ratio of the gas law deviation coefficients decreases as the pressure increases.

In summary, it can be said that the transport capacity of hydrogen pipelines does not decrease significantly despite the much lower gross calorific values of hydrogen compared to natural gas.

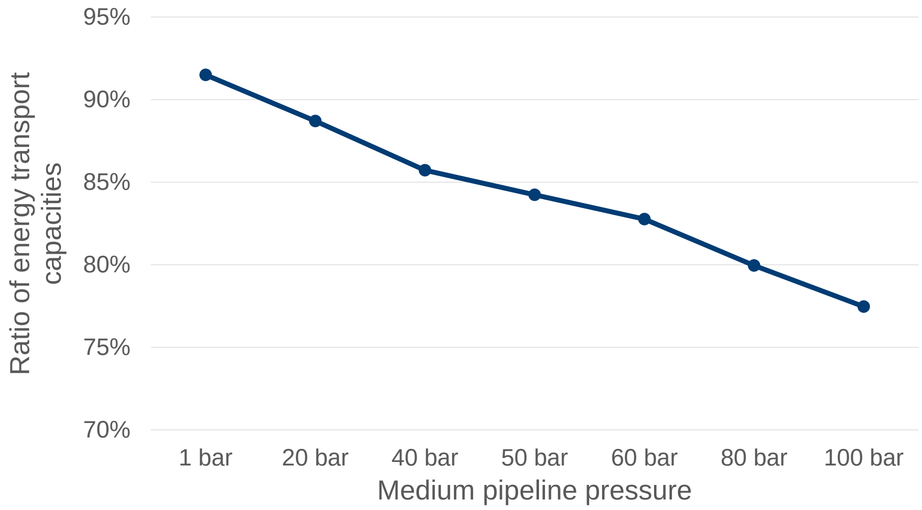


Figure 2.1. Ratio of energy transport capacities.

2.2. Volumetric flow rate at flowing conditions

The low gross calorific value has a major influence on the volumetric flow rate at flowing conditions. The definition of the volumetric flow rate at flowing conditions as a function of the volumetric flow rate at standard conditions is provided in formula 2.6.

$$Q_B = Q_N \cdot \frac{T}{T_N} \cdot \frac{P_N}{P} \cdot K \quad (2.6)$$

Formula 2.6. Volumetric flow rate at flowing conditions as a function of the volumetric flow rate at standard conditions.

The volumetric flow rate at standard conditions can in turn be calculated as the quotient of the energy transport capacity and the gross calorific value.

$$\dot{E} = Q_N \cdot H_S \iff Q_N = \frac{\dot{E}}{H_S} \quad (2.7)$$

Formula 2.7. Volumetric flow rate at standard conditions.

Inserting formula 2.7 into the definition of the volumetric flow rate at standard conditions gives the formula for the volumetric flow rate at flowing conditions as a function of capacity and gross calorific value.

$$Q_B = \frac{\dot{E}}{H_S} \cdot \frac{T}{T_N} \cdot \frac{P_N}{P} \cdot K \quad (2.8)$$

Formula 2.8. Volumetric flow rate at flowing conditions as a function of capacity and gross calorific value

The ratio of the volumetric flow rate of hydrogen and natural gas at flowing conditions is defined in formula 2.9. It is assumed that the capacity to be transported in the form of natural gas and hydrogen is the same.

$$\frac{Q_{B,H_2}}{Q_{B,EG}} = \frac{H_{s,EG}}{H_{s,H_2}} \cdot \frac{K_{H_2}}{K_{EG}} \quad (2.9)$$

Formula 2.9. Ratio volumetric flow rate at flowing conditions.

Using the aforementioned natural gas composition to calculate the gross calorific value and the gas law deviation coefficient gives the ratio of volumetric flow rates at flowing conditions for hydrogen and natural gas transport shown in fig. 2.2.

The volumetric flow rates of hydrogen at flowing conditions are three to four-and-a-half times higher than for natural gas. This assumes that the capacity to be transported is the same.

2.3. Flow velocity

The increased volumetric flow rates at flowing conditions have an effect on the flow velocities of the gas in the pipeline, among other things.

The flow velocity of a gas in a pipeline is defined in formula 2.10. It is derived from the volumetric flow rate at flowing conditions Q_B and the cross-sectional area ⍰ of the pipeline.

$$v = \frac{Q_B}{A} \quad (2.10)$$

Formula 2.10. Flow velocity.

For a given pipeline, the area is constant. When the ratio of the flow velocity is calculated, the variable A is therefore cancelled out, leaving the following formula.

$$\frac{v_{H_2}}{v_{EG}} = \frac{Q_{B,H_2}}{Q_{B,EG}} \quad (2.11)$$

Formula 2.11. Ratio of flow velocities.

The ratio of the flow velocities is therefore identical to the ratio of the volumetric flow rates at flowing conditions. If a natural gas pipeline converted to hydrogen is to retain its transport capacity, the flow velocities will have to be three to four-and-a-half times as high. These high velocities can accelerate material fatigue (e.g. due to acoustic effects such as vibrations and pulsations). However, this risk can be reduced with selection of suitable materials and components.

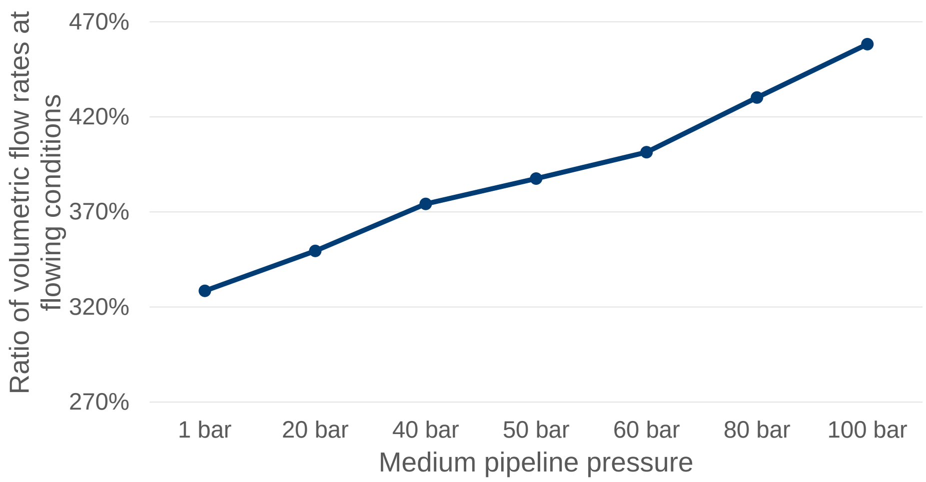


Figure 2.2. Ratio of volumetric flow rates at flowing conditions.

3

Fundamentals of compressor technology

A compressor unit consists of a compressor and a prime mover. This section explains which types of compressors are suitable for moving hydrogen in pipelines and which prime movers should be used, while also discussing the pros and cons of potential technical solutions for compressors and prime movers.

3.1. Compressors for hydrogen transport

Compressors are used to increase the pressure level of a gaseous (compressible) medium. In pipeline applications, they fulfil several functions:

- They compensate for the pressure loss that inevitably occurs during transportation through the pipeline. This pressure loss is largely determined by the properties of the pipework, the medium and the volumetric flow (see also section 2.1).
- In addition, compressors in a pipeline network are used to control gas flows by influencing both the direction and the velocity of the gas flow and the transported volumes.
- Compressors can also be used to store energy: By increasing the pressure in a pipeline, the quantity of the transported medium and hence its energy content are also increased. Such temporary increases in pressure are used, for example, to deal with short-term peak loads. This effect is often referred to as "line packing".

Compressors can basically be divided into two main groups: positive displacement and dynamic compressors (see fig. 3.1).

Positive displacement compressors work by drawing in and capturing the gas in one or several compression chambers. As the geometric volume of the chambers decreases, the gas inside is compressed. In positive displacement compressors, the maximum achievable pressure ratio does not depend on the molar mass of the gas, but only on the volume reduction.

Dynamic compressors, on the other hand, draw the gas between blades rotating on a rapidly moving impeller accelerating the gas to a high velocity. The gas is then channelled through a diffuser, where the kinetic energy is converted into static pressure. In this case, the properties of the compressed gas have a major influence on the compression process (see also section 3.1.1). The molar mass in particular has a significant impact on the maximum achievable pressure ratio.

The choice of compressor usually depends on the pressure ratios to be achieved and the maximum volumetric flows to be compressed. Dynamic compressors offer advantages for large volumetric flows and moderate pressure ratios, while positive displacement compressors tend to be the better choice for high pressure ratios and moderate volumetric flows.

In the case of hydrogen, positive displacement compressors are more suitable for initial compression and storage. For very large quantities, a geared turbocompressor can also be a suitable option. Due to the large volumetric flows, dynamic compressors are particularly suitable for moving hydrogen in pipelines. In certain cases, reciprocating compressors can be a good alternative.

The most commonly used variants in industry are centrifugal compressors and reciprocating compressors. Both compressor types have been used for more than 100 years in various applications such as pipeline transport, the chemical industry, fertiliser production and the petrochemicals sector.

Compared to natural gas, the compression of hydrogen places more demands on the material, the size of the engines

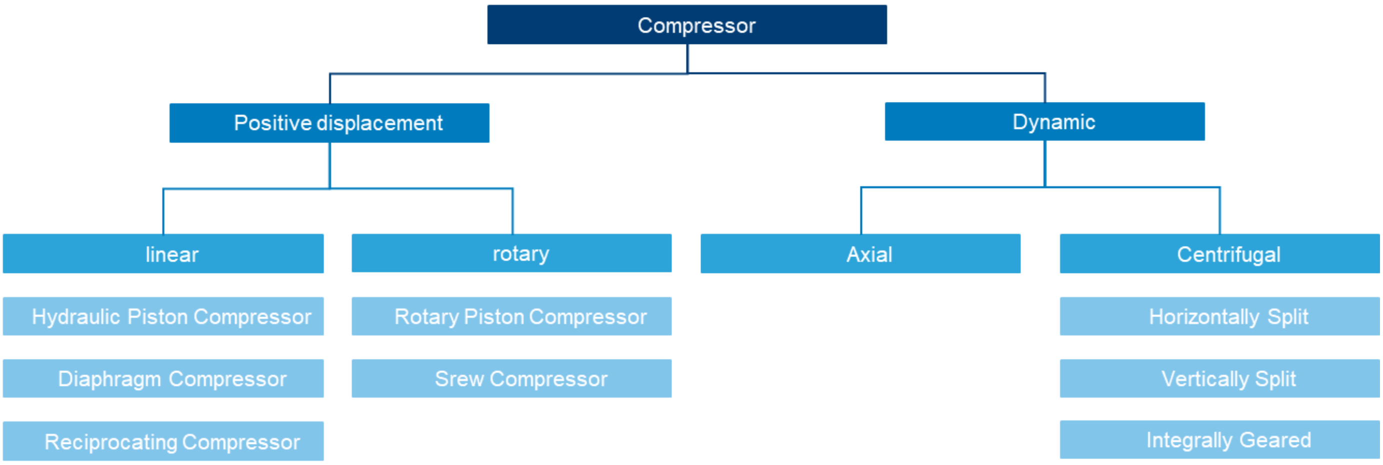


Figure 3.1. Compressor types.

and the number of compression trains, regardless of the compressor type. Given the different modes of operation of dynamic and positive displacement compressors, they each come with their own technical challenges. The advantages and disadvantages of these compressor types in terms of hydrogen compression are explained in more detail in the following sections.

3.1.1. Dynamic compressors

The most common type of dynamic compressor in pipeline applications is the turbocompressor, a radial flow compressor. In turbocompressors, the transported fluid is accelerated (by imparting kinetic energy) and then decelerated in a diffuser. Turbocompressors therefore generate pressure by converting kinetic energy. The kinetic energy in turn is proportional to the molar mass of the transported gas and the flow velocity in the compressor. Because of this correlation, the molar mass of a gas and the circumferential speed are central to the operation of turbocompressors. They will ultimately determine the maximum pressure ratio (ratio between outlet pressure and inlet pressure) that can be achieved.

According to basic thermodynamics, the specific isentropic enthalpy difference (also referred to as "head") is given by the following formula:

$$h = \frac{\kappa}{\kappa - 1} \cdot \frac{R}{M} \cdot Z \cdot T \cdot (\pi^{\frac{\kappa-1}{\kappa}} - 1) \qquad (3.1)$$

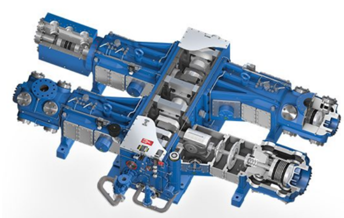
$$\text{where} \quad \kappa = \frac{c_p}{c_v}$$

Formula 3.1. Specific isentropic enthalpy difference [4].

Consequently, the specific isentropic enthalpy difference *h* depends on the isentropic exponent κ, the molar mass *M*, the inlet temperature *T*, the universal gas constant *R*, the pressure ratio π and the compression factor *Z*. In the case of hydrogen, the compression factor is about 1 and can therefore be neglected.

For a given impeller-diffuser geometry and a given speed, the compression work is relatively independent of the composition of the compressed gas. A change of gas will only change the molar mass, the isentropic coefficient (slightly), the compression factor and the resulting pressure ratio.

Table 3.1. Advantages and disadvantages of centrifugal compressors and reciprocating compressors.

Centrifugal compressor		Reciprocating compressor	
 Image source: Siemens Energy		 Image source: Ariel compression	
Advantages	**Disadvantages**	**Advantages**	**Disadvantages**
Large volumetric flow	Low pressure ratio	High pressure ratio	Small volumetric flow
Continuous compression over time	-		Discontinuous compression over time (risk of vibration)
Small footprint			Large footprint
Low leakage			Leaks possible
		Variable pressure range	

The following rule of thumb can be derived directly from the above equation:

$$\pi_{\max} - 1 \sim \mathrm{M} \tag{3.2}$$

Formula 3.2. Relationship between compression ratio and molar mass

As already shown in chapter 1, the molar mass of hydrogen (2 g/mol) is significantly lower than that of natural gas (approx. 18 g/mol). If the circumferential speed is assumed to be constant, the low molar mass means that the maximum achievable relative pressure increase for hydrogen compressors is around a factor of nine lower than for natural gas compressors. So purely hypothetically, a total of seven natural gas compressors would have to be connected in series to move hydrogen with the same pressure ratio that one of these natural gas compressors achieves with natural gas.

While a typical natural gas pipeline compressor has around two to four stages, hydrogen compressors theoretically require 14 to 28 stages, which, however, is practically impossible because of design limitations. The maximum number of stages in a centrifugal compressor would realistically be in the range of eight to ten.

The only practical way to impart more kinetic energy to the gas and thus compensate for the lower pressure build-up in each stage is to significantly increase the circumferential speed of the radial impellers. To achieve a pressure increase per stage that is comparable to that in a natural gas compressor, the impeller's circumferential speed has to be increased to approx. 600 to 800 m/s. For comparison, normal circumferential speeds during natural gas compression are in the range of 200 to 350 m/s.

The limiting factors determining the maximum achievable circumferential speeds are the velocity of sound of the transported gas and the material properties of the compressor. With natural gas, the limiting factor tends to be the velocity of sound (approx. 400 m/s). The velocity of sound of hydrogen is far higher at around 1,320 m/s. Consequently, it is not the velocity of sound that is the limiting factor for hydrogen compression, but rather the materials used. To prevent hydrogen embrittlement, the yield strength must not exceed 827 N/mm² (cf. American Petroleum Institute (API) 617).

There are centrifugal compressors on the market that deliver a pressure ratio of up to 3 with a maximum of seven impellers, which corresponds to circumferential speeds of approx. 500 to 600 m/s. It should be noted, however, that the number of these units already in use is still very limited.

To increase the circumferential speed, you can either use impellers with a larger diameter or increase the shaft speed. For design reasons, compressor manufacturers generally favour the "smaller and faster" instead of the "larger and slower" solution.

Typical hydrogen compressors used in pipeline transport can be expected to have speeds of 10,000 to 23,000 rpm. By comparison, the speeds of the compressors in today's natural gas network are between 3,000 and 10,000 rpm. Conventional prime movers are unable to achieve the high speeds required for hydrogen compression, which is why an intermediate gearbox is needed. Apart from the fact that the gearbox constitutes another element in the rotor string and therefore increases complexity, it also influences the unit's overall efficiency (approx. one to two per cent loss due to air turbulence and heat), increases its footprint and requires additional cooling.

Overall efficiencies can be expected to be lower with hydrogen than with natural gas. This is due, on the one hand, to the large number of compressor stages required to compensate for the low molar mass and, on the other hand, to the higher internal leakage rates caused by the low viscosity of hydrogen. It is probably fair to say – without any claim to accuracy – that the typical overall efficiency of a hydrogen compressor at the point of maximum efficiency (also called peak efficiency) could be around 80 per cent, while for a natural gas compressor it would be around 83 per cent.

3.1.2. Positive displacement compressors

For positive displacement compressors, the main challenge is potential hydrogen leakage. As hydrogen molecules are much smaller than natural gas molecules, seals and gaskets have to be modified. Moreover, in reciprocating compressors, the hydrogen can find its way past piston rings into the crankcase. To avoid potentially flammable mixtures, the crankcase itself or the crosshead must be flushed with an inert gas (e.g. nitrogen).

Another challenge that comes with compressing hydrogen in reciprocating compressors is the temperature at the compressor outlet. Due to the higher isentropic exponent of hydrogen, the maximum permissible outlet temperature is reached much earlier.

$$\mathrm{T}_2 = \mathrm{T}_1 \cdot \left(\frac{\mathrm{p}_1}{\mathrm{p}_2}\right)^{\frac{1-\kappa}{\kappa}} \tag{3.3}$$

Formula 3.3. Outlet temperature.

If it is assumed that the inlet temperature, inlet pressure and outlet pressure are identical, only the isentropic exponent will have an influence on the temperature increase, as shown by formula 3.3. At a pressure ratio of 5, the outlet temperature will be around 25 per cent higher than with natural gas. The higher outlet temperatures also occur with dynamic compressors, but in combination with much lower pressure ratios. Hence, the outlet temperature is mainly a challenge for positive displacement compressors and less so for dynamic compressors.

The high flow velocities and the low molar mass of hydrogen can have a negative impact on the system's pulsation behaviour. This can make it necessary to provide additional damping elements (e.g. pressure vessels).

Screw compressors are another form of positive displacement machine. They have two parallel interlocking screws that rotate relative to each other and to the housing, separated by gaps to allow the screws to move. One challenge here is that due to the pressure difference, some of the compressed gas flows back through the gaps, which reduces the compressor's efficiency. The lower the molar mass and viscosity of a gas, the lower the resistance in the gap, which results in greater backflow. Lubricants such as oil or water are often used to reduce this backflow and simultaneously cool the compressed gas. However, this means that the gas can become contaminated and must then be cleaned after compression.

3.2. Prime movers

Compressors are driven by a so-called prime mover. The choice of prime mover usually depends on the specific compressor application. Gas turbines, electric motors, steam turbines and expanders are most commonly used, with the latter found especially in the chemical and petrochemical industries, where process heat or excess pressure is available for further utilisation.

Table 3.2. Advantages and disadvantages of electric motors and gas turbines with regard to hydrogen transport.

Electric motor		Gas turbine	
 Image source: Siemens		 Image source: Baker Hughes	
Advantages	**Disadvantages**	**Advantages**	**Disadvantages**
	Dependence on appropriate power supply	Utilisation of H_2 directly as fuel gas, black start capability	
Less expensive			Expensive
Sophisticated technology			Further development necessary
High efficiencies			Low efficiencies
No direct emission of climate-forcing gases			Nitrogen oxides and noise emissions
High reliability			Risks when starting and stopping the machine
Low-maintenance			Maintenance-intensive
Broad range of services		High power density	
	Harmonics from frequency converters have adverse impact on the power grid		

Gas turbines and electric motors lend themselves to compressing hydrogen in pipeline applications.

Electric motors are an established technology that has been used in many branches of industry for over a hundred years. They require a reliable and highly available power supply and can be operated over a wide speed range with good efficiency. However, electric motors need a frequency converter to vary the speed. Frequency converters require a suitable installation location (safe area which is air-conditioned) and must be cooled. Apart from waste heat, frequency converters emit electromagnetic radiation. The biggest advantage of electric motors is their very wide range of applications combined with their good to very good efficiency. Electric motors themselves do not emit any climate-damaging gases.

It should be noted that electric motors will see a drop in efficiency if they are operated far beyond their design point. Furthermore, electric motors are not supplied with primary energy. The overall efficiency of the system must therefore be extended to include electricity generation to allow a comparison with gas turbines. Apart from power generation, transmission and the transformation process also play an important role.

Gas turbines are known in particular for their high energy density and the fact that they are independent of external energy sources. In pipeline applications, the fuel gas can usually be taken directly from the pipeline. The electrical energy required to operate the gas turbine can be provided by small generators, which means that the gas turbine can even be operated without a stable electricity grid.

As with all combustion processes, the choice of fuel and the combustion conditions determine the level of emissions from the gas turbine. The combustion of hydrocarbons produces carbon monoxide (CO) and carbon dioxide (CO_2), which are climate-forcing gases. These emissions do not occur where the fuel gas is hydrogen. However, combustion with air produces more nitrogen oxides (NO_x) because of the high combustion temperatures.

Electric motors and natural gas-fuelled turbines are mature technologies that have been used successfully for decades. Direct combustion of hydrogen in gas turbines, however, is still uncommon. The technical changes required have already been identified, and hydrogen-ready gas turbines are expected to be introduced to the market over the coming years.

Table 3.2 summarises the advantages and disadvantages of the different types of drive presented. In the case of gas turbines, emissions -— especially noise -— play a major role in the choice of location and actual feasibility. In contrast, the use of electric motors depends primarily on the availability of a reliable power supply, as electric motors do not have black start capability.

Most natural gas compressors are currently driven by gas turbines. In recent years, however, the use of electric motors as prime movers has increased. It is therefore only logical and sensible to transfer this trend to a hydrogen network.

The required compressor capacity as a function of the volumetric flow rate at flowing conditions and the specific isentropic enthalpy difference (see formula 3.1) can be calculated as follows.

$$P_G = h \cdot \frac{1}{\eta_s} \cdot \rho \cdot Q_B \qquad (3.4)$$

Formula 3.4. Power transferred to the gas [2].

η_s is the isentropic efficiency with due consideration of the thermodynamic and aerodynamic losses, while ρ is the density of the gas at flowing conditions and Q_B is the volumetric flow rate at flowing conditions. The latter two parameters refer to the conditions on the inlet side of the compressor.

The efficiency can be estimated at the pre-selection stage on the basis of experimental correlations or with reference to engines built previously. However, manufacturers tend to determine engine efficiency by measuring the actual compression work required on a test bench compared to the ideal compression work (i.e. without heat recovery and without losses) at the same pressure ratio.

Formula 3.5 shows that the installed power required to drive the compressor P_K (shaft output) is the product of P_G and the reciprocal of the mechanical efficiency (with bearing and sealing losses taken into account).

$$P_K = P_G \cdot \frac{1}{\eta_m} \qquad (3.5)$$

Formula 3.5. Compressor drive power.

The mechanical efficiency depends on the compressor design and is independent of the transported gas. For pipeline compressors, it is typically in the range of 95 to 98 per cent.

Assuming that the volumetric flow rate at flowing conditions is the same, there are no major differences in terms of the power required to drive a hydrogen or natural gas compressor. If, however, hydrogen and natural gas compression are compared for the same transported capacity, the shaft output required increases significantly. In this case, the volumetric flow rates at flowing conditions to be compressed and thus the shaft output to be applied are three to four and a half times greater for hydrogen than for natural gas (see also section 2.2).

Moreover, the use of repurposed natural gas pipelines in a hydrogen network may lead to an increased need for compression, as the pipe diameters are not ideal for hydrogen transmission. All these factors will increase the compressor capacity required, resulting in a greater demand for larger or additional prime movers.

4

Matching transport requirements to the compressor technology

This section explains the most important technical aspects of the selection process for a compressor unit (compressor and prime mover) using an example.

Selecting the right unit is a multi-stage process:

1. The first step is to identify transport scenarios for a future hydrogen pipeline network. The scenarios selected here are based on publications by FNB Gas on the visionary hydrogen network [5] and on findings collected at OGE as part of the company's business development activities. System-relevant parameters such as Germany's future energy demand and forecasts on the ramp-up of the hydrogen market were also taken into account.
2. In the second step, the selected transport scenarios are used to define transport requirements and operating points for compressors. To this end, the expected operating conditions of the gas pipeline system at various locations are analysed.
3. This data is then used in the third step to determine possible configurations of compressors and prime movers, with engine selection taking account of the following technical and economic criteria:
    - Investment costs, including procurement and stocking of spare parts
    - Operating costs, including energy and maintenance costs
    - Operational flexibility
    - Engine efficiency
    - System reliability and availability
    - H_2 readiness
    - Technical complexity

4.1. Input data

The starting point for these investigations was to define operating points for potential hydrogen compressors (see table 4.1). An operating point is the combination of the inlet pressure prior to compression, the desired outlet pressure after compression and the volumetric flow measured at these pressures.

The operating points described in table 4.1 have been numbered according to two criteria: the compressor station under consideration and the transport case. For example, cases 3-1 to 3-5 refer to the same compressor station but consider four different transport cases.

They are based on a fluid mechanical calculation for the visionary hydrogen network published by FNB Gas at the end of 2021. The visionary hydrogen network is a draft developed by the German gas transmission system operators for a possible hydrogen network in 2050.

The visionary hydrogen network comprises a number of compressor stations at various locations, which are shown in Figure 4.1. Some of these compressor stations were shortlisted for further analysis. In order to show as much variance as possible, only the compressor stations with widely differing operating points were selected.

Table 4.1. Operating points examined.

Operating point	Volumetric flow rate at standard conditions in 1,000 m³/h	P_{in} in bar_g	P_{out} in bar_g	Compression ratio
1-1	3,099.2	26.4	32.5	1.2
1-2	4,409.5	38.4	44.5	1.2
1-3	1,666.5	62.0	71.5	1.2
1-4	1,494.3	20.6	30.0	1.5
2-1	1,605.0	40.0	75.0	1.9
2-2	1,976.1	40.0	70.3	1.8
3-1	1,221.0	40.0	45.2	1.1
3-2	508.1	40.0	64.7	1.6
3-3	693.8	40.0	53.1	1.3
3-4	740.7	40.0	64.7	1.6
3-5	1,411.5	40.0	55.5	1.4
4-1	980.0	40.0	56.0	1.4
4-2	2,775.3	40.0	53.0	1.3
4-3	2,962.7	40.0	60.5	1.5
4-4	5,646.0	40.0	54.5	1.4
5-1	2,775.3	40.0	58.5	1.5
5-2	2,962.7	40.0	59.2	1.5
5-3	5,646.0	40.0	78.0	2.0
6-1	2,542.0	40.0	43.7	1.1
6-2	740.7	40.0	44.5	1.1
6-3	1,411.9	40.0	63.4	1.6
6-4	3,763.0	26.4	32.5	1.2

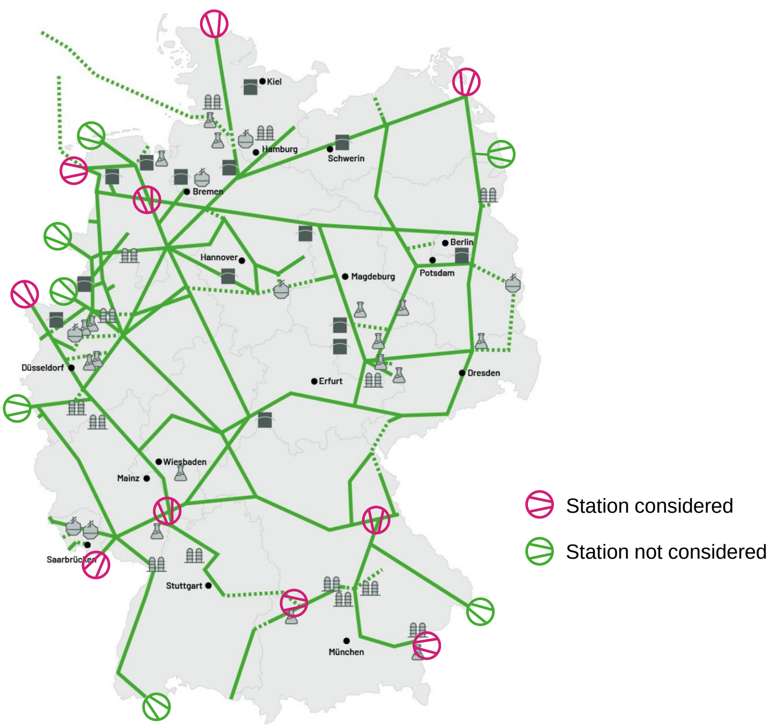


Figure 4.1. Compressor stations in the visionary hydrogen network.

4.2. Selecting the engine configuration

The operating points define the transport requirements to be met by the compressor station. Each of the compressor stations listed in table 4.1 has several operating points and therefore needs to meet different requirements in terms of engine configuration. This makes selecting the right configuration a complex optimisation process. One approach is to determine the number of compression trains for each compressor station, which involves a trade-off between CAPEX (the most cost-effective option is a single, large compressor unit) and flexibility (the most flexible option is several small compressor units), with due consideration given to the design specifications such as the achievable efficiency.

The operating points in table 4.1 were used to define possible engine configurations, which meant first calculating the required compressor capacity and specific enthalpy difference (see fig. 4.3). As the operating points vary widely in terms of flow rate and specific enthalpy difference, an initial clustering was performed based on the maximum shaft output.

The next step then was to choose the operating points to be used for selecting the engine configuration. Operating points 1-1 to 1-4 and 4-4 were disregarded. 1-1 to 1-4 represent a main line compressor station and have comparatively low requirements for the engine configuration. The maximum volumetric flow for point 4-4 is so high that the compressor station would require 170 MW of capacity. This is such an extreme and rare case that the decision was made to exclude this point from the selection process as an outlier.

The remaining operating points have outputs of up to 80 MW and specific enthalpy differences of up to 900 kJ/kg. For comparison, the installed capacity of existing natural gas compressor stations is generally between 20 and 200 MW.

4.2.1. Selecting the prime movers

The operating points in table 4.1 were used to select appropriate compressor configurations. The process of selecting compressors and prime movers to match the operating points is shown in fig. 4.4.

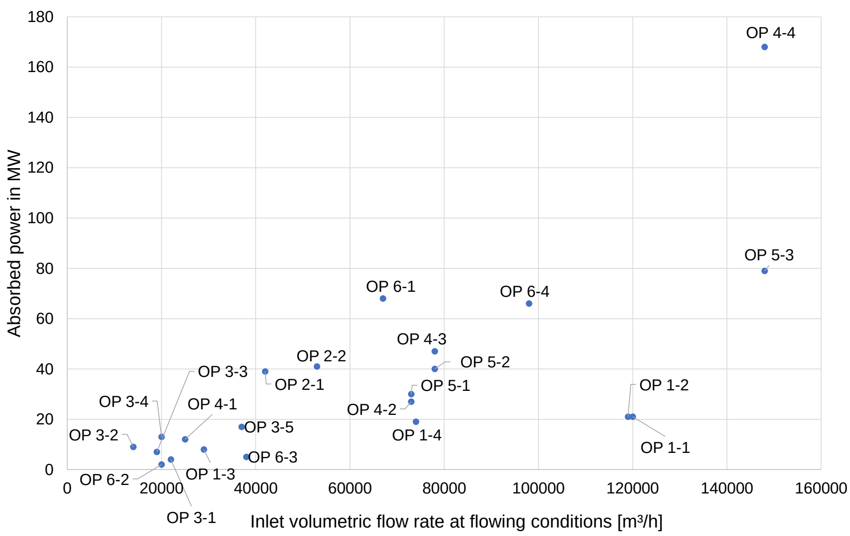


Figure 4.2. Absorbed power.

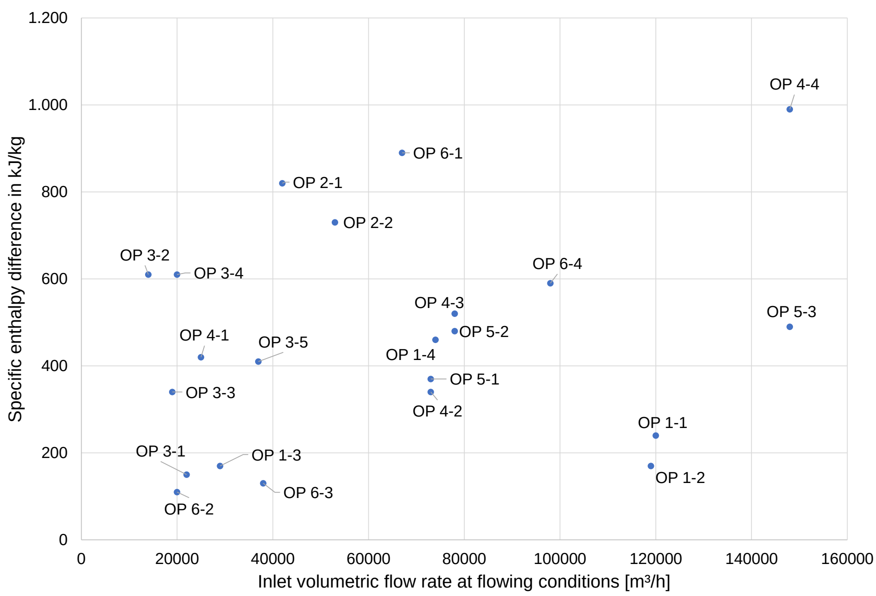


Figure 4.3. Specific enthalpy difference.

In order to find a suitable configuration for the prime movers, the first step is to compare the required compressor power per station with the maximum shaft output of the prime movers. Three specific options were considered:

1. Gas turbine with a maximum output of around 14 MW under site conditions.
2. Gas turbine with a maximum output of around 29 MW under site conditions.
3. Electric motor with a maximum output of around 40 MW. The following ratings were considered in detail:
   - 14 MW, medium voltage, asynchronous, water-cooled
   - 29 MW, medium voltage, synchronous, water-cooled
   - 40 MW, medium voltage, synchronous, water-cooled

Upon completion of step 1, the number of prime movers required at the station and hence the number of compression trains is determined. Prime mover selection is deemed to have been successful if it delivers the necessary shaft output and the resulting number of compression trains is not too high (e.g. less than seven trains).

4.2.2. Selecting the compressors

In this process step, the required specific enthalpy difference of an entire station is compared with the individual values of different compressor types and a decision is taken as to whether the compressor units should be operated in parallel (to provide high flow rate for the same specific enthalpy difference) or in series (to provide high specific enthalpy difference for the same flow rate). Compressor selection is considered to have been completed successfully if the overall transportation requirements can be met and the resulting number of compressor stages (impeller + diffuser) of each compressor unit is not too high (e.g. less than eight stages).

Table 4.2. Compressor configuration with gas turbines and electric motors.

Operating point	Gas turbine	Electric motor
2-1	3	
2-2	4	
3-1	1	
3-2	1	
3-3	1	
3-4	2	
2-1	3	
3-5	2	
4-1	1	
4-2	3	
4-3	4	
5-1	3	
5-2	4	
5-3	4	3
6-1	4	2
6-2	1	
6-3	1	
6-4	4	3

When it comes to determining the configuration of the compressor units, there are a number of options for compressors and prime movers. In table 4.2 shows an example of such a configuration using gas turbines and electric motors rated 14 MW

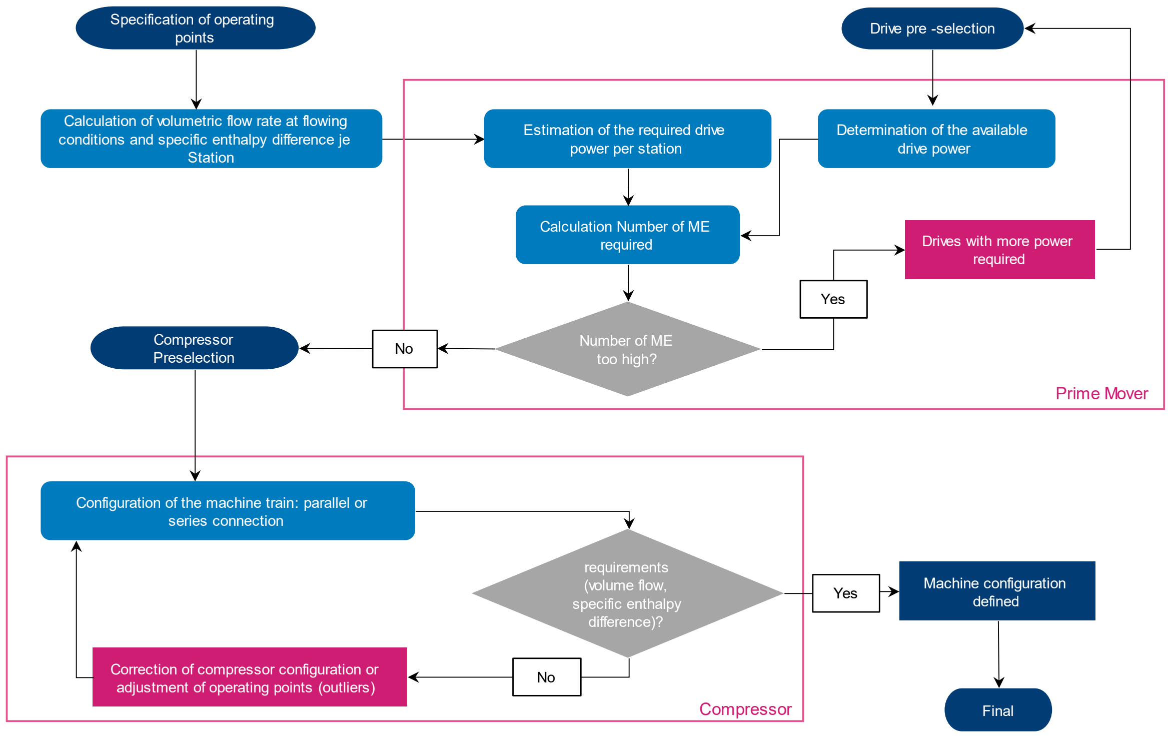


Figure 4.4. Selection process for the compressor unit configuration.

each.

The configuration in table 4.2 can cover all operating points, provided that the compressor units are suitably connected in series or parallel as required for the station in question. In order to maximise flexibility, the compressor units should be arranged back-to-back.

Table 4.3. Price indications for prime mover and compressor configurations.

Compressor unit configuration	Price indication (April 2023)	
Large gas turbine + compressor	~ € 25 million/unit	~ € 0.9 million/MW
Small gas turbine + compressor	~ € 15 million/unit	~ € 1.0 million/MW
Large electric motor + compressor	~ € 15 million/unit	~ € 0.5 million/MW
Small electric motor + compressor	~ € 10 million/unit	~ € 0.8 million/MW

The back-to-back arrangement allows a compressor unit to be operated both in series and in parallel (see Figure 4.5). The unit is divided into a low-pressure and a high-pressure section. When used in series, the gas is first compressed in the low-pressure section and then in the high-pressure section over several stages. When the unit is operated in parallel, the gas flow is split into a low-pressure flow and a high-pressure flow and compressed separately.

In addition to the technical design, the compressor configurations also have to undergo a commercial assessment. Table 4.3 provides a rough price overview for various configurations.

In this list, the variants with electric motors have lower costs than those with gas turbines. However, the electric motors incur additional costs for the grid connection.

4.3. Results

Compared to natural gas, the compression of hydrogen requires more compressor units with higher ratings. This is because hydrogen has a lower volumetric energy content and a lower molar mass.

The choice of prime mover is driven by the system and should take into account both the technical complexity and the availability of a grid connection. From a mechanical perspective, both gas turbines and electric motors are suitable drives for dynamic compressors. Electric motors have the advantage of not generating any direct CO_2 emissions while also requiring less maintenance than gas turbines. On the other hand, gas turbines offer black-start capability. Of course, combinations of gas and electric drives are also conceivable.

A certain number of compression trains are required for each transport case. Depending on the specific enthalpy difference and the volumetric flow to be transported, the compressors should be connected either in parallel or in series. This is where a back-to-back arrangement increases flexibility, as it allows all operating points reviewed to be covered with just one type of compressor.

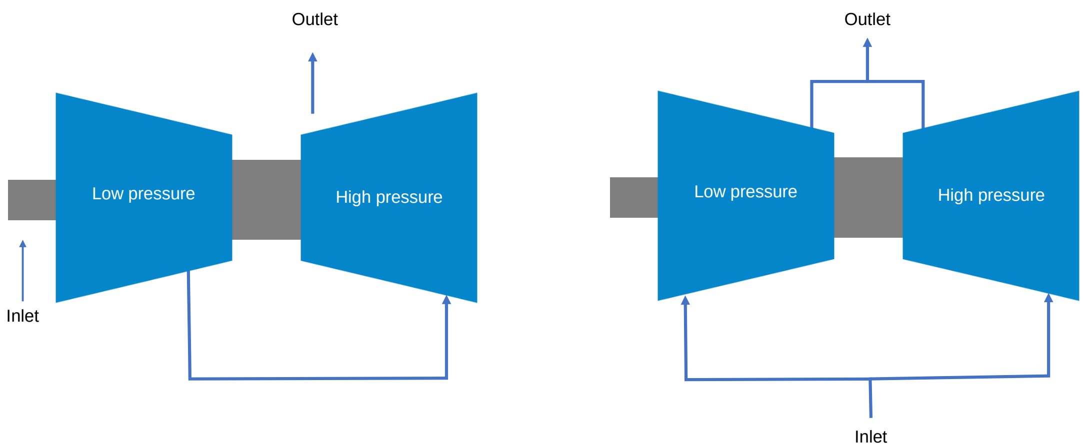


Figure 4.5. Back-to-back arrangement.

5

Fluid mechanical calculations for pipeline configurations

This section presents the results of several fluid mechanical calculations for various pipeline configurations along with the conclusions drawn for the transportation of hydrogen. To this end, a series of basic calculations are first made for a basic pipeline. This basic pipeline is then modified to include additional elements in order to show what options are available for increasing the transmission capacity of a pipeline system (e.g. loop lines, compressors) and what advantages and disadvantages they entail. The calculations were carried out using the SIMONE simulation programme. Cost estimates are based on the indicative rates from the core network.

5.1. Fundamentals

One of the main aims of fluid mechanical calculations is to determine the pressure drop in pipelines (see section 2.1). The pressure drop is used to determine the energy transport capacity of a pipeline or pipeline network.

Figure 5.1 is a schematic representation of the pressure drop during hydrogen transportation in a pipeline. The hydrogen enters a given pipeline at a pressure of 80 bar_g. Due to the pressure drop during transport, the pressure at the end of the pipeline is only 30 bar_g. The pressure drop is non-linear, increasing more and more towards the end of the pipeline section. In this example, the minimum pressure at the end of the line (30 bar_g) corresponds to the calculated pressure. This means that the maximum energy transport capacity of the pipeline has been reached.

One way to increase the transport capacity of the system is to parallelise the existing pipeline. The pressure drop across the two pipelines is then lower, which increases the transport capacity (see fig. 5.2).

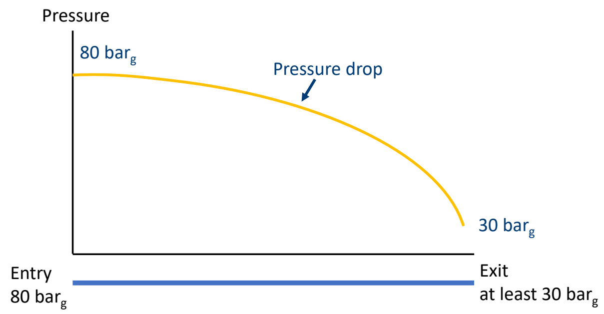


Figure 5.1. Pressure drop across a pipeline.

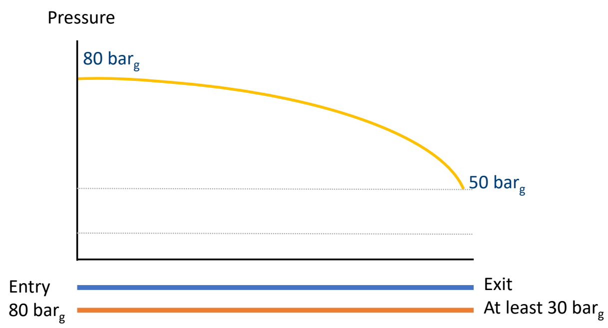


Figure 5.2. Pressure drop with parallel pipelines.

The capacity of the pipeline system can also be increased through the installation of compressors to boost the pressure along the pipeline.

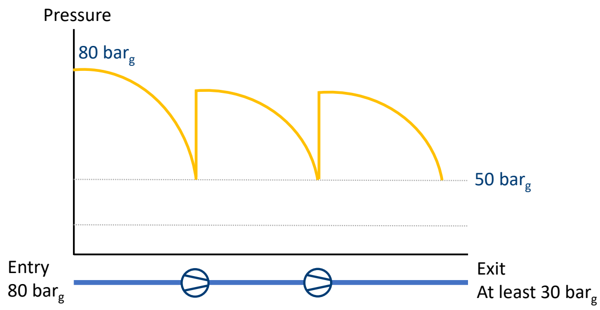


Figure 5.3. Pressure drop with compression.

5.2. Basic calculations

For the basic calculations, the pressure drop over a 1,000 km pipeline is determined for different pipe diameters and different transmission capacities.

Fixed parameters:

- The pipeline under consideration is 1,000 kilometres long, which is roughly the length of Germany from north to south.
- The pressure at the entry point is 80 bar_g.
- The pressure at the end of the pipeline should be at least 30 bar_g.

Variable parameters:

- Pipeline diameters: 800 mm, 1,000 mm, 1,200 mm and 1,400 mm.
- Transported capacity: 5 GW, 10 GW and 20 GW in the form of pure hydrogen.

The pressure drop was calculated for each combination of pipe diameter and transported capacity. The pressure drop is considered acceptable if the pressure at the end of the pipeline is at least 30 bar_g. If it is lower, the pipeline is unable to complete the specified transportation task. In this case, system expansion is required, which can be achieved either with construction of a series of loop lines in parallel to the existing pipeline or with the installation of compressors at suitable locations.

Figure 5.4 shows a pressure drop calculation across a pipeline with a diameter of 1,000 mm. In this example, the planned transmission capacity is 10 GW. However, this is not achieved as the maximum capacity is around 8 GW. If it were higher, the pressure at the end of the pipeline would fall below the desired minimum pressure.

Raising the capacity to 10 GW therefore requires the system to be expanded. Figure 5.5 shows the network used to determine how many additional kilometres of loop lines are required, with black arrow symbols indicating that a pipeline section has actually been used. Each section has a length of 100 km, resulting in a total length of 500 km of additional pipelines.

Alternatively, the capacity of the pipeline system can be increased through the installation of compressors. In the example of 10 GW to be transported via a pipeline with a diameter of 1,000 mm, a single compressor station after 400 kilometres would be sufficient to achieve the minimum pressure at the end of the pipeline. This can be recognised from the colour of the compressor symbol; (see Figure 5.6) the active compressor station is shown in black. In this case, the required compressor capacity is around 38 MW.

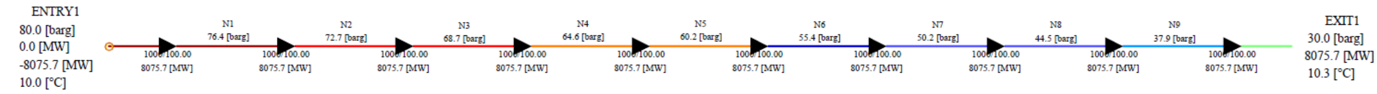


Figure 5.4. Pressure drop over a 1,000 km pipeline in SIMONE.

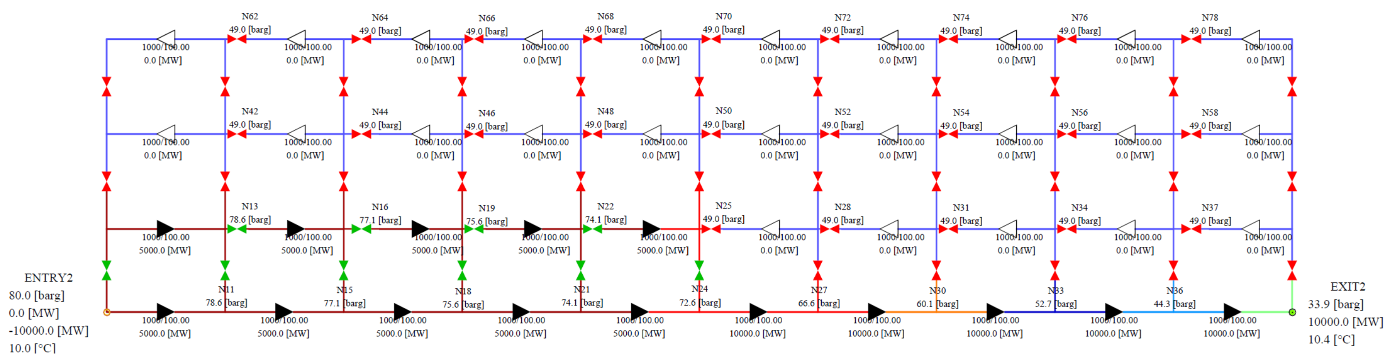


Figure 5.5. Loop lines in the simulated SIMONE pipeline network.

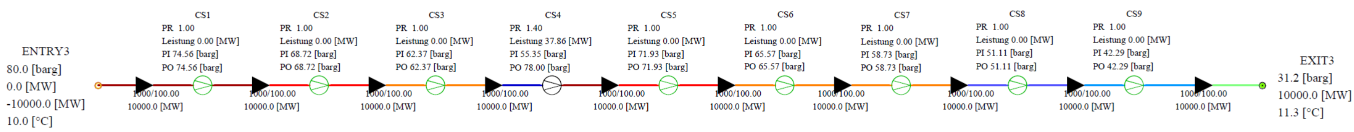


Figure 5.6. Use of compressors in the simulated SIMONE pipeline network.

The results of the simulation calculations for all combinations are summarised in Table 5.1.

The result shows which diameter/capacity combinations in the table meet the specified transportation goal (text in black in Table 5.1). For example, with the combination DN1000 and 5 GW (scenario no. 10), the existing pipeline capacity is sufficient and no expansion is necessary. Scenarios 11 and 12 were therefore not considered any further.

For the scenarios that meet the transportation goal, the specific transportation costs in ct/kWh were determined using the indicative costs of the core network. For this purpose, a mixed cost approach including CAPEX and OPEX was used, for which two variants for compressor running hours per year were assumed: 8,760 hours (all hours of a year) vs. 2,190 hours (a quarter of the hours of a year). The different running hours were used to calculate the minimum and maximum specific costs. In Figures 5.7 to 5.9 the minimum costs (at 8,760 hours) are shown in blue and the maximum costs (at 2,190 hours) in orange.

Table 5.1. Results of simulation calculation.

Nr.	Szenariobeschreibung	Ergebnis
1	5 GW, 800 mm, no expansion	Pipeline capacity is not sufficient, loop line or compressor required
2	5 GW, 800 mm, loop line	300 km of loop line required
3	5 GW, 800 mm, compressor	1 compressor station required
4	10 GW, 800 mm, no expansion	Pipeline capacity is not sufficient, loop line or compressor required
5	10 GW, 800 mm, loop line	1,400 km of loop line required
6	10 GW, 800 mm, compressor	8 compressor stations required
7	20 GW, 800 mm, no expansion	Pipeline capacity is not sufficient, loop line or compressor required
8	20 GW, 800 mm, loop line	3,600 km of loop line required
9	20 GW, 800 mm, compressor	19 compressor stations required
10	5 GW, 1000 mm, no expansion	Pipeline capacity is sufficient
11	5 GW, 1000 mm, loop line	No additional pipeline capacity required
12	5 GW, 1000 mm, compressor	No additional compressor capacity required
13	10 GW, 1000 mm, no expansion	Pipeline capacity is not sufficient, loop line or compressor required
14	10 GW, 1000 mm, loop line	500 km of loop line required
15	10 GW, 1000 mm, compressor	1 compressor station required
16	20 GW, 1000 mm, no expansion	Pipeline capacity is not sufficient, loop line or compressor required
17	20 GW, 1000 mm, loop line	1,700 km of loop line required
18	20 GW, 1000 mm, compressor	9 compressor stations required
19	5 GW, 1200 mm, no expansion	Pipeline capacity is sufficient
20	5 GW, 1200 mm, loop line	No additional pipeline capacity required
21	5 GW, 1200 mm, compressor	No additional compressor capacity required
22	10 GW, 1200 mm, no expansion	Pipeline capacity is sufficient
23	10 GW, 1200 mm, loop line	No additional pipeline capacity required
24	10 GW, 1200 mm, compressors	No additional compressor capacity required
25	20 GW, 1200 mm, no expansion	Pipeline capacity is not sufficient, loop line or compressor required
26	20 GW, 1200 mm, loop line	800 km of loop line required
27	20 GW, 1200 mm, compressor	3 compressor stations required
28	5 GW, 1400 mm, no expansion	Pipeline capacity is sufficient
29	5 GW, 1400 mm, loop line	No additional pipeline capacity required
30	5 GW, 1400 mm, compressor	No additional compressor capacity required
31	10 GW, 1400 mm, no expansion	Pipeline capacity is sufficient
32	10 GW, 1400 mm, loop line	No additional pipeline capacity required
33	10 GW, 1400 mm, compressor	No additional compressor capacity required
34	20 GW, 1400 mm, no expansion	Pipeline capacity is not sufficient, loop line or compressor required
35	20 GW, 1400 mm, loop line	100 km of loop line required
36	20 GW, 1400 mm, compressor	1 compressor station required

The specific transportation costs are between 0.2 and 5.9 ct/kWh, depending on the transportation task, with the higher value resulting from a very unfavourable and therefore unrealistic combination of transported capacity and pipeline system.

For 5 GW of transportation capacity, there are no major differences in costs, as Figure 5.7 illustrates. If the pipeline has a diameter of 800 mm, it is slightly more cost-effective to install compressors to boost the capacity instead of using loop lines. This is due, among other things, to the fact that in order to simplify system planning the minimum length of loop is 100 km, while compressors are deployed more precisely to meet requirements. This results in a slight advantage for the compressor costs.

From a diameter of 1,000 mm, the pipe is sufficient to transport the required capacity. Although the costs are higher for diameters of 1,200 mm and 1,400 mm, these cases are not relevant as the smaller diameter is sufficient for transportation.

A pipeline with a diameter of 800 mm is not sufficient to transport 10 GW, so further expansion will be required. In this case, it is more cost-effective to plan loop lines rather than compressors. At a diameter of 1,000 mm, there is no major cost difference between the two expansion options. From a diameter of 1,200 mm, the existing pipeline is sufficient to meet the transportation goal. Therefore, the 1,400 mm diameter case is again not relevant.

The transportation of 20 GW shows greater differences in costs compared to the lower capacities previously considered. The most expensive case is the combination of a pipeline diameter of 800 mm with additional compressors. However, this case is so unfavourable that it must be regarded as unrealistic because, according to the simulation calculation, compressors would be required every 50 km.

Even for a diameter of 1,000 mm, it is significantly more expensive to add compressors than it is to use loop lines. From a diameter of 1,200 mm, there are no longer any major differences.

Moreover, there is a general trend for 20 GW of transportation capacity: The larger the diameter of the initial pipeline, the lower the specific transportation costs.

Even these simple basic calculations show that it is crucial from a transportation cost perspective to choose the right pipe dimensions for each transportation task. If the initial diameter is

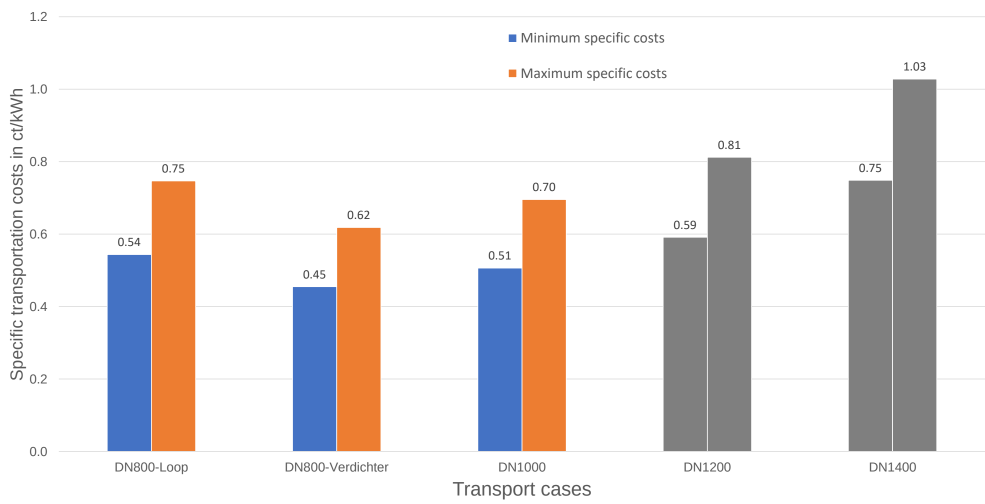


Figure 5.7. Specific costs for the transportation of 5 GW.

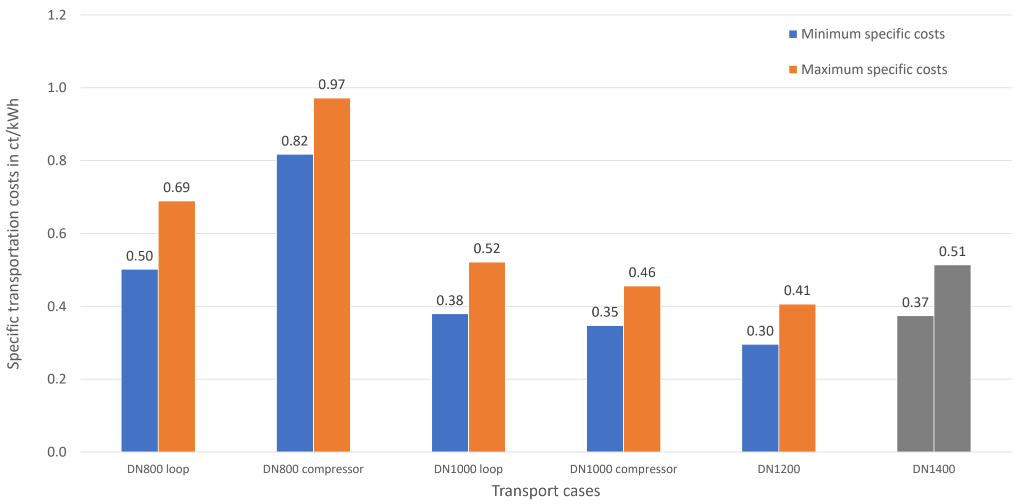


Figure 5.8. Specific costs for the transportation of 10 GW.

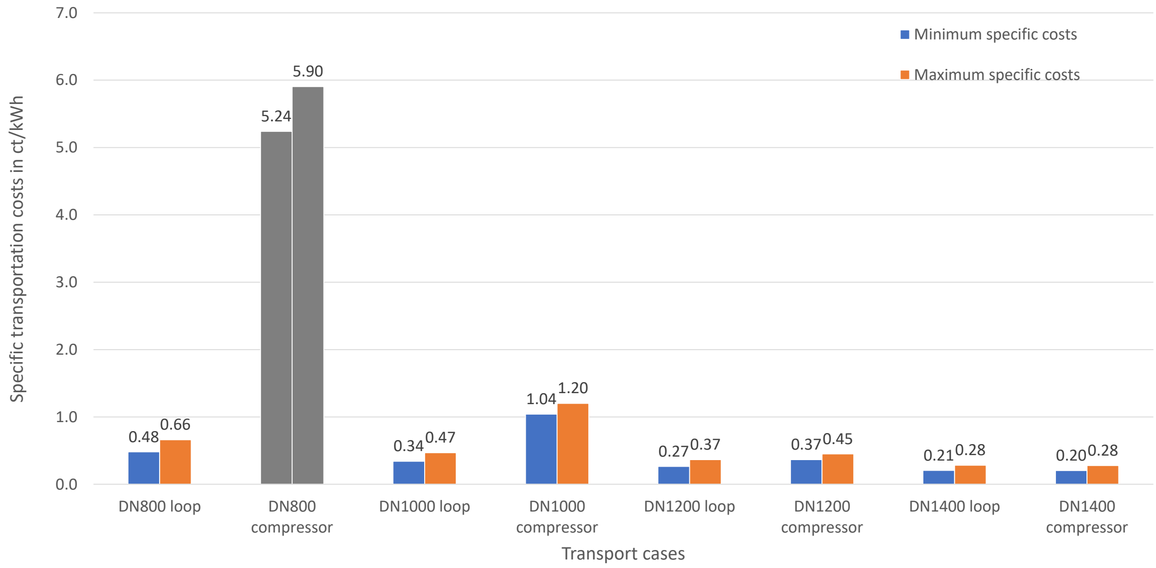


Figure 5.9. Specific costs for the transportation of 20 GW.

far too small, transportation will become much more expensive due to the additional expansion work required. If, on the other hand, the pipeline is oversized in the initial pipeline network, the costs will be higher than necessary from the outset.

The calculations performed allow the following basic conclusions to be drawn:

- If large quantities of hydrogen are to be transported (e.g. 10 GW and 20 GW), it is most cost-effective to use large-diameter pipelines.
- If a large increase in transport capacity is to be expected, loop lines are generally more cost-effective than additional compressors.
- If the existing transportation system is only slightly undersized, it is worth adding compressors.

6

Summary

This white paper presents concepts for hydrogen compression in a growing transportation system and provides an assessment from a technical and economic perspective.

Chapter 2 uses formulas to show that the transportation capacity of a natural gas pipeline converted to hydrogen is around 80 per cent of its previous capacity. This is mainly due to the similar Wobbe numbers of hydrogen and natural gas. When used for hydrogen, the volumetric flow rate at flowing conditions and thus the flow velocity are significantly higher (by a factor of 4.5) for the same transported capacity. This is due to the fact that the volumetric energy content (gross calorific value) of hydrogen is significantly lower than that of natural gas.

Chapter 3 provides some basic information about compressors and prime movers. Dynamic compressors combined with electric motors or gas turbines are suitable for compressing hydrogen. Electric motors have very high efficiencies but must have a reliable power supply. Gas turbines can use hydrogen for operation but emit noise and nitrogen oxides.

The compression of hydrogen with turbocompressors requires seven times more compression stages than with natural gas to generate the same high pressure difference, unless the compressors rotate significantly faster.

For the same volumetric flow rate at flowing conditions, there are no major differences in terms of shaft output between hydrogen and natural gas. However, if the compression of hydrogen and natural gas is compared at the same transported capacity, the shaft output needed to drive the compressor increases significantly. Due to the increased volumetric flow rate at flowing conditions, the shaft output to be applied is three to four and a half times greater when compressing hydrogen.

Chapter 4 explains how compressor configurations can be selected to meet specific transportation requirements, describing possible compressor configurations for some operating points derived from the visionary hydrogen network. A flexible back-to-back compressor arrangement makes it possible to cover all operating points with compressors of a single performance class.

Chapter 5 presents the results of simple fluid mechanical calculations, exploring the options available to increase the transportation capacity of a pipeline network. They show that the transportation situation determines whether loop lines or additional compressors are more cost-effective.

Bibliography

[1] F. Graf, V. Isik, M. Heneka, W. Köppel, T. Kolb, A.-C. Fleer, and R. W. Verbücheln, "Aktuelle Fragestellungen beim leitungsgebundenen Transport von Wasserstoff," *Chem. Ing. Tech.*, vol. 96, no. 1-2, pp. 74–85, Jan. 2024.

[2] J. Mischner, H.-G. Fasold, and K. Kadner, *Systemplanung in der Gasversorgung, Gaswirtschaftliche Grundlagen, Kapitel: Transportkapazität von Gasleitungen und Leitungssystemen*. Essen, Germany: Oldenbourg Industrieverlag GmbH, Oct. 2011.

[3] J. Mischner and H.-G. Fasold, "Gasbeschaffenheiten in Deutschland: Was zum Wobbe-Index noch gesagt werden sollte," *gwf Gas+Energie*, 2019.

[4] K. H. Ludtke, *Process centrifugal compressors*. Berlin, Germany: Springer, Nov. 2010.

[5] "FNB Gas — fnb-gas.de," https://fnb-gas.de/wasserstoffnetz/h2-netz-2050/, [Accessed 09-04-2025].