TRANSITION ACCELERATOR TECHNICAL BRIEFS
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The Transition Accelerator
L'Accélérateur de transition

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# ABOUT THE TRANSITION ACCELERATOR

The Transition Accelerator (The Accelerator) exists to support Canada's transition to a net zero future while solving societal challenges. Using our four-step methodology, The Accelerator works with innovative groups to create visions of what a socially and economically desirable net zero future will look like and build out transition pathways that will enable Canada to get there. The Accelerator's role is that of an enabler, facilitator, and force multiplier that forms coalitions to take steps down these pathways and get change moving on the ground.

Our four-step approach is to understand, codevelop, analyze and advance credible and compelling transition pathways capable of achieving societal and economic objectives, including driving the country towards net zero greenhouse gas emissions by 2050.

**1** UNDERSTAND the system that is being transformed, including its strengths and weaknesses, and the technology, business model, and social innovations that are poised to disrupt the existing system by addressing one or more of its shortcomings.

**2** CODEVELOP transformative visions and pathways in concert with key stakeholders and innovators drawn from industry, government, indigenous communities, academia, and other groups. This engagement process is informed by the insights gained in Stage 1.

**3** ANALYZE and model the candidate pathways from Stage 2 to assess costs, benefits, trade-offs, public acceptability, barriers and bottlenecks. With these insights, the process then re-engages key players to revise the vision and pathway(s), so they are more credible, compelling and capable of achieving societal objectives that include major GHG emission reductions.

**4** ADVANCE the most credible, compelling and capable transition pathways by informing innovation strategies, engaging partners and helping to launch consortia to take tangible steps along defined transition pathways.

# ABOUT THE AUTHORS

## Mohd Adnan Khan PhD

**TRANSITION ACCELERATOR**

**Adnan Khan** is an Energy Systems Analyst at the Transition Accelerator working to help design pathways towards the establishment of a sustainable energy future. Adnan has a PhD in Material Science and Engineering and is passionate about working on renewable energy systems and contributing to the development of a future hydrogen economy. He has over eight years of industrial and academic experience leading research teams across the value chain of technology development and commercialization, driving innovation, and fostering collaboration among industry, government, and academia. He has published over 35 articles in reputable scientific journals, has seven granted patents and now hopes his work will lead to the spin-out of consortia led projects, get change moving on the ground and help drive Canada towards a net-zero future.

## Cameron Young MSc, PEng

**CESAR AT UNIVERSITY OF CALGARY**

**Cameron Young** is an Energy Systems Analyst at CESAR. He joined CESAR to help create a hydrogen economy in Canada. His work will include research on different pathways for hydrogen production, transmission, and distribution to provide pragmatic information for industry and policy makers. He hopes his work will help develop projects that convert Alberta's resources into a sustainable source of hydrogen fuel. Cameron has a Chemical Engineering & Management double-major bachelor's degree from McMaster University, a Masters in Sustainable Energy Development from the University of Calgary and is registered as a Professional Engineer with APEGA. He has 10 years of process engineering and project development experience in Alberta's energy sector.

## Catherine MacKinnon MSc, PEng

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**Catherine MacKinnon** is an Energy Systems Analyst at the CESAR Initiative at the University of Calgary. She has a Master of Science degree in Sustainable Energy Development (SEDV) and a Bachelor of Science degree in Chemical Engineering from the University of Calgary. She is a Professional Engineer in good standing with The Association of Professional Engineers and Geoscientists of Alberta and has more than nine years of professional experience in the upstream energy industry in various technical, corporate, and financial roles. She recently completed her capstone project, exploring the carbon footprint and carbon management strategies of direct and indirect GHG emissions associated with operations of a remote, off-grid research station in the Yukon.

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**David B. Layzell** is an Energy Systems Architect with the Transition Accelerator, a Faculty Professor at the University of Calgary, and Director of the Canadian Energy Systems Analysis Research CESAR Initiative. Between 2008 and 2012, he was Executive Director of the Institute for Sustainable Energy, Environment and Economy (ISEEE), a cross-faculty, graduate research, and training institute at the University of Calgary. Before moving to Calgary, Dr. Layzell was a Professor of Biology at Queen's University, Kingston (cross appointments in Environmental Studies and the School of Public Policy), and Executive Director of BIOCAP Canada, a research foundation focused on biological solutions to climate change. While at Queen's, he founded a scientific instrumentation company called Qubit Systems Inc. and was elected 'Fellow of the Royal Society of Canada' (FRSC) for his research contributions.

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| | |
|---|---|
| AIH | Alberta Industrial Heartland, Edmonton, Strathcona, Fort Saskatchewan, Sturgeon, Lamont |
| Blue Hydrogen | Hydrogen produced from natural gas with carbon capture and storage |
| CESAR | Canadian Energy Systems Analysis Research |
| CCS | Carbon Capture and Storage |
| CCSU | Carbon Capture, Storage and Utilization |
| CO2 | Carbon Dioxide |
| CRF | Capital Recovery Factor |
| DTE | Drivetrain Efficiency |
| EOR | Enhanced Oil Recovery |
| EWMC | Edmonton Waste Management Centre |
| FCEB | Fuel Cell Electric Bus |
| GHG | Greenhouse Gas |
| GJ | Gigajoule ($10^9$ Joules) |
| Green Hydrogen | Hydrogen produced by water electrolysis using intermittent zero-carbon electricity generated from wind and solar facilities |
| Grey Hydrogen | Hydrogen produced from natural gas or coal |
| H2 | Hydrogen |
| HDV | Heavy-Duty Vehicle: Vehicles with a gross vehicle weight rating >= 15 metric ton or tonne |
| HFCE | Hydrogen Fuel Cell Electric |
| HFS | Hydrogen Fueling Station |
| HHV | Higher Heating Value |
| ICE | Internal Combustion Engine |
| IF | Installation Factor |
| LCOH | Levelized Cost of Hydrogen |
| LDV | Light-Duty Vehicle |
| LH2 | Liquid Hydrogen |
| MDV | Medium-Duty Vehicle |
| NG | Natural Gas |
| O&M | Operations and Maintenance |
| PJ | Petajoule ($10^{15}$ Joules) |
| SF | Scale Factor |
| SMR | Steam Methane Reforming |
| TCI | Total Capital Investment |
| TIC | Total Installed Cost |
| UC | Uninstalled Cost |

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Prairies Economic Development Canada Développement économique Canada pour les Prairies

Natural Resources Canada Ressources naturelles Canada

Canada

Alberta's Industrial Heartland

EMISSIONS REDUCTION ALBERTA

UNIVERSITY OF CALGARY

CESAR CANADIAN ENERGY SYSTEMS ANALYSIS RESEARCH

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# EXECUTIVE SUMMARY

The future of a hydrogen economy will rely on developing infrastructure for low-cost distribution and delivery of hydrogen. To this end, compression of gaseous hydrogen is a key technology which enables delivery to end user. Nonetheless the compression of hydrogen is challenging and generally considered to be one of the most expensive process units in the supply chain. There are various compressor designs that can be used and ultimately the choice of compression technology, associated costs, energy use and resulting GHG emissions will depend on where in the supply chain it is used.

The purpose of this 'technical brief' is to describe how to conduct technoeconomic analysis of hydrogen compressors, with a focus on developing a hydrogen value chain for heavy transport (buses, trucks, trains, and ships). The report is written by compiling techno-economic information from several previous studies to develop a model for students, engineers, scientists, and entrepreneurs focused on evaluating hydrogen compression technologies, power requirement and associated costs. Some key insights and highlights are as follows:

- Although compression of natural gas is widely used, the compression of hydrogen is significantly challenging due to its low molecular weight and density.
- The reported isentropic efficiency ($\eta_{isen}$) of hydrogen compressors is in the range of ~55-80%.
- Currently available reciprocating compressors which rely on mechanical pistons with several moving parts are expensive with cost varying from several hundred thousand dollars to millions of dollars depending on scale and compression ratio required. It is expected that the capital costs associated with these compressors will drop sharply with economy of scale.
- Since compressing hydrogen is an energy intensive process, the cost of operating a hydrogen compressor is dominated by energy/fuel costs rather than capital costs.
- Further research and development activities are needed to tackle issues with leakage of hydrogen, embrittlement and increase efficiency and reliability of hydrogen compressors.
- The development of new technologies such as those based on ionic liquids or metal hydrides is promising. In particular, ionic liquid compressors, which have been particularly developed by Linde, could be the key to efficient and low cost hydrogen compression.

# 1 INTRODUCTION

In the transition to net-zero emission energy systems, electricity made from very low or non-emitting carbon sources (e.g., solar, wind, hydro, nuclear, fossil fuels with carbon capture and storage) will play a major role as an energy carrier. However, electricity is not a viable energy carrier for applications such as heavy-duty or long distant transport, heavy industry (e.g., steel making) and space heating in cold climates or large buildings. These applications currently rely on energy carriers such as diesel fuel or natural gas [1,2]. To this end, hydrogen ($H_2$) could play a key role as an energy vector and become the zero/low carbon emission fuel of choice of the future for these hard-to-decarbonize sectors.

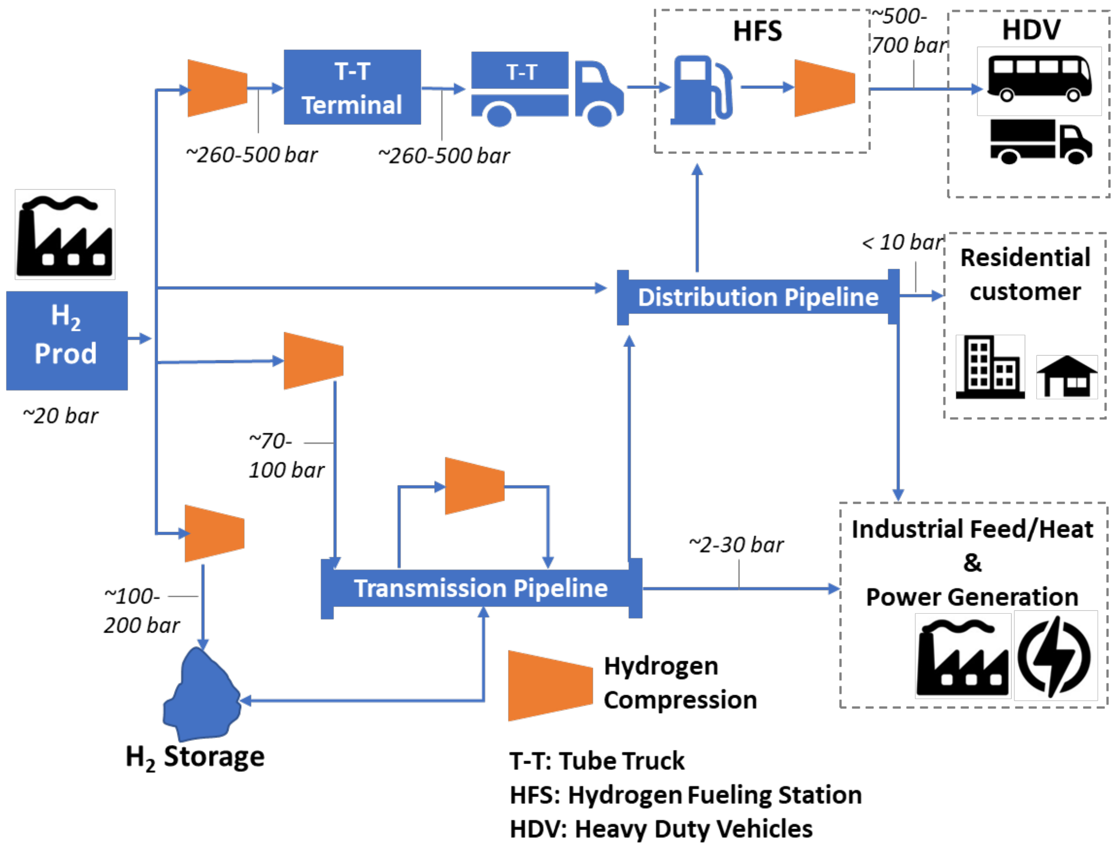


**Figure 1.1** Overview of where compression takes place in a $H_2$ supply chain.

As one of the world's lowest cost producer of low-carbon 'green' (from water electrolysis) or 'blue (from fossil fuels coupled to carbon capture and storage) $H_2$, Canada is strategically positioned to benefit from taking a leadership role in the transition to a net-zero $H_2$ economy [3]. While Canada can make $H_2$ at a cost that is lower than the wholesale price of diesel [3], the distribution and storage of $H_2$ is more challenging. The challenge arises from $H_2$'s low density of ~0.0898 kg/m$^3$ (energy density ~3 kWh$_{LHV}$/m$^3$) at standard temperature and pressure (STP) of 0 ˚C and 1 atm, respectively [4]. Therefore, a volume of ~11.1 m$^3$ (11,100 L) would be required to store 1 kg of $H_2$. In contrast, 1 kg of gasoline can be stored in a volume of ~ 0.0013 m$^3$ (1.3 L) under the same conditions [5]. Compression, liquefaction, or conversion of $H_2$ into larger molecules such as ammonia ($NH_3$) are possible options to overcome this hurdle. Each option has advantages and disadvantages, and the lowest cost option will vary according to geography, distance, scale and the required end use [6]. Compression is the ubiquitous solution in the gaseous $H_2$ supply chain whereby high pressures can help achieve acceptable energy densities. Although widely used, the compression of $H_2$ is generally considered to be one of the most expensive process units in the $H_2$ value chain [6,7]. The choice of compression technology, associated costs, energy use and the resulting GHG emissions will depend on where in the supply chain compression is used (**Figure 1.1 and Figure 1.2**). Several factors such as molar flow rates, pressure ratio, nature of gas or gas mixture (compressibility and ratio of specific heats), and purity required, dictate the choice, energy requirement and cost of compression.

The purpose of this 'technical brief' is to describe how to conduct technoeconomic analysis of $H_2$ compressors, with a focus on developing a $H_2$ value chain for heavy transport (buses, trucks, trains, and ships). The document is intended as a beginner's guide for engineers and scientists focused on calculating compression power, associated costs and selection of appropriate compressor technology depending on where compression takes place in the supply chain. The basic operating principles and design features of various compressor technologies will be described, along with the challenges of compressing $H_2$. In addition, the report will provide a detailed step-by-step guide on power and efficiency calculations of both single and multi-stage compressors for the purpose of energy system analysis. Finally, the report will provide detailed steps on calculating capital, operating and energy costs for $H_2$ compression.

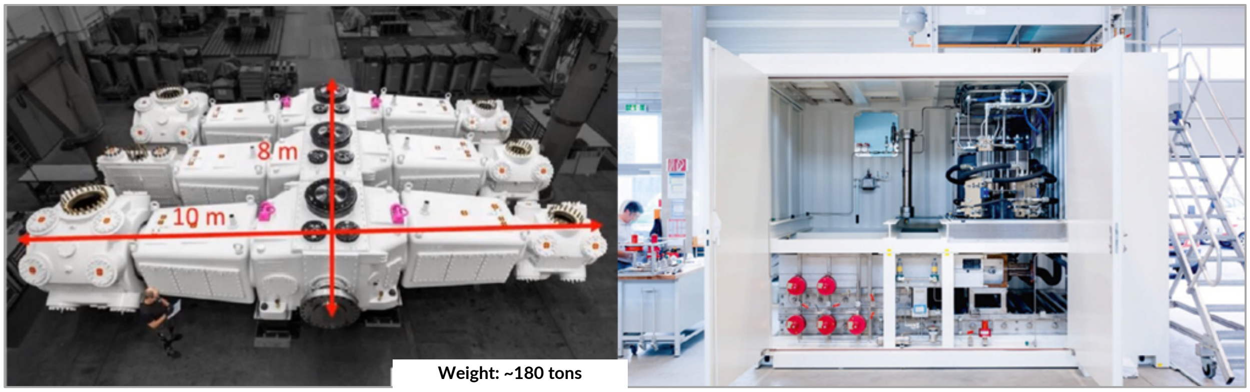


**Figure 1.2** $H_2$ compression at different scales.

LEFT: Compression at a large-scale industrial facility at a scale of ~44,000 kg $H_2$/day. RIGHT: Compression at a small $H_2$ Fueling Station (HFS) less than 40 kg $H_2$/hr.

**SOURCE:** LEFT: NEUMAN-ESSER [8]. RIGHT: LINDE [9]

# 2 TYPES OF COMPRESSORS

## 2.1 Mechanical compressors

Mechanical compressors are widely used and are designed based on the direct conversion of mechanical energy into compressed gas energy. There are several classifications, but most compressors used today for gaseous $H_2$ compression are either positive displacement compressors or dynamic compressors (**Figure 2.1**) [7,11]. In the sections below we provide a summary of various compressor technologies. For further details, readers can refer to the comprehensive review on $H_2$ compression, published by G. Sdanghi et. al. in Renewable and Sustainable Energy Reviews 102 (2019) [7].

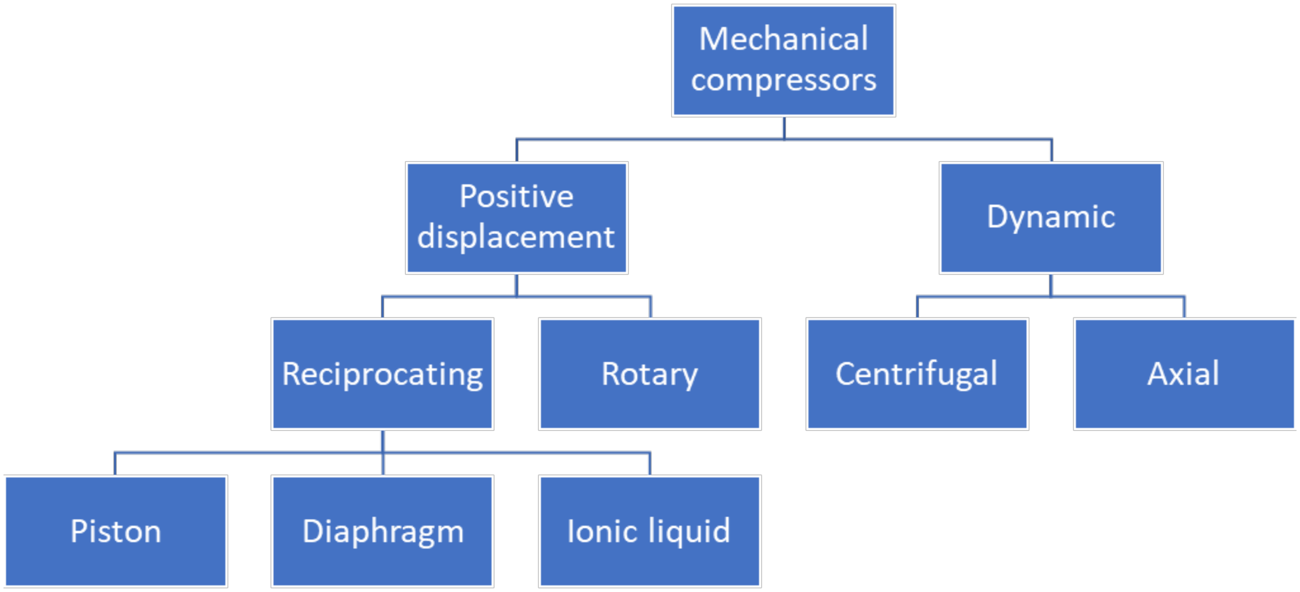


**Figure 2.1** Types of mechanical compressors broadly divided based on how they compress a gas.

**SOURCE:** ADAPTED FROM PERRY'S CHEMICAL ENGINEERS' HANDBOOK [10]

### 2.1.1 Reciprocating piston compressors

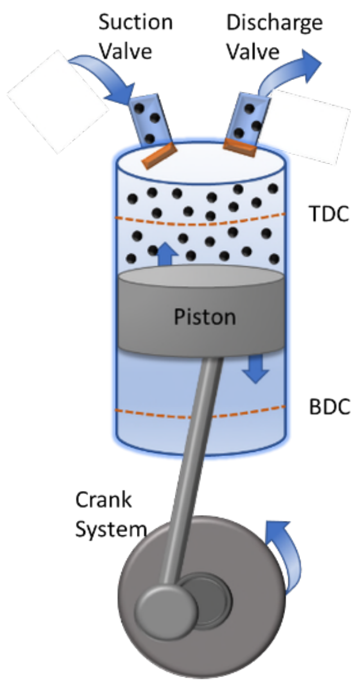


**Figure 2.2** Schematic of reciprocating piston compressor

TDC: top dead centre; BDC: bottom dead centre.

**SOURCE:** ADAPTED FROM REFERENCE [7]

Reciprocating piston compressors are ideal for low to moderate flow and high-pressure applications [7]. They are positive displacement machines that work on the concept of reciprocation i.e., via compression and displacement of gases. A single stage reciprocating compressor is designed using a piston and cylinder (**Figure 2.2**) system where the piston is driven by a crankshaft, converting rotary motion into linear motion. The cylinder uses two automatic valves – one for gas suction and the other for gas discharge and the energy needed for compression is provided by either an electrical or thermal source. They are widely used in petrochemical plants and oil refineries.

Reciprocating compressors can produce high-pressure $H_2$ particularly when a multi-stage configuration is used. Companies like Hydro-Pac Inc. have demonstrated high discharge pressures up to 850 bar, with an inlet pressure of 350 bar and flow rates up to 5084 $Sm^3/h$ [7,12]. These are typically used in refineries and chemical plants to compress industrial grade $H_2$ to high pressures. Howden Co. has recently demonstrated the world's largest reciprocating compressor with a compression power of ~ 16.6 MW [13]. Currently oil-free versions of reciprocating compressors are preferred in applications where the purity of $H_2$ is a priority. Nonetheless oil free $H_2$ reciprocating compressors are affected by embrittlement and experience frequent failure of the sealing rings due of non-uniform pressure distribution [7,14]. Furthermore, since there is no oil to act as heat sink, thermal protection of various components becomes critical in oil free compressors.

### 2.1.2 Reciprocating diaphragm compressors

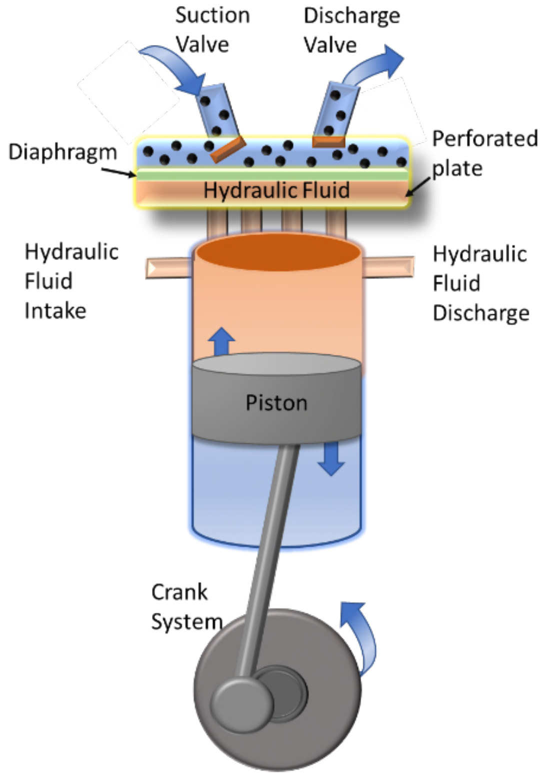


**Figure 2.3** Schematic of a metal diaphragm compressor.

**SOURCE:** ADAPTED FROM REFERENCE [7]

Diaphragm compressors also known as membrane compressors are suitable when handling high purity gases since the process gas is completely isolated from the hydraulic oil and/or piston. Similar to the reciprocating piston compressor, the piston is driven by a crankshaft but in this case the motion is then transmitted to a hydraulic fluid and then finally onto a thin metal membrane called a "diaphragm", isolating the process gas [7]. Typically, a perforated plate is used to distribute the hydraulic oil and get a uniform pressure distribution on the diaphragm plate (**Figure 2.3**). In middle and larger sized diaphragm heads, the hydraulic oil can be effectively cooled, thereby increasing the efficiency of the compressor. This makes diaphragm compressors unique as they can achieve high single stage compression ratios. For example, a compression application that could require three to five stages in traditional reciprocating piston compressors, could be done in one to two stages in diaphragm compressors.

Diaphragm compressors are a good choice for compressing gases without contamination and leakage of gas to ambient air. Diaphragm compressors are suitable for applications requiring low flow rates and are widely used in $H_2$ fueling stations (HFS) [15]. The American company, PDC Machines [16], has designed and manufactured diaphragm compressors for $H_2$ fuel cell vehicles and their compressors operate at a discharge pressure of 517 bar with flow rates ranging from 52.7 to 295.4 $Sm^3/h$ [7]. Howden, has designed diaphragm compressors that are 100% leak free, with a special 'head' of the compressor design that makes high single stage pressure ratios possible. The company claims that 1000 bar discharge pressure can be achieved with only two compressor stages, from suction pressures of 50 bar [17].

### 2.1.3 Reciprocating ionic liquid compressors

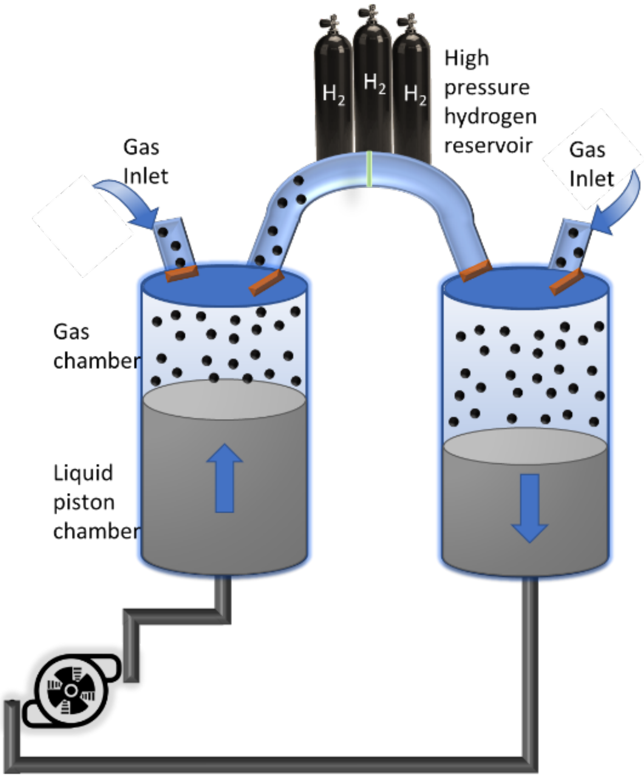


**Figure 2.4** Schematic of an ionic liquid compressor.

**SOURCE:** ADAPTED FROM REFERENCE [7]

Ionic liquid compressors are also positive displacement devices that use ionic liquids to replace the metal piston of a conventional compressor (**Figure 2.4**). A combination of various properties such as low compressibility, low vapor pressure, negative melting points, high heat capacity, high thermal conductivity, high chemical and thermal stability and low $H_2$ solubility have attracted the use of ionic liquids for $H_2$ compression [11,18]. They are known to achieve inexpensive compression because they can ensure a quasi-isothermal process [19]. The ionic liquids assist with thermal management, because of which external heat exchangers are not required, giving them a significant advantage over mechanical piston compressors. This also leads to higher efficiency values close to 70% versus ~45% for reciprocating piston or diaphragm compressors, reducing costs as well [7,11].

Additionally, these compressors do not require bearings or seals, two of the common sources of failures in piston and diaphragm compressors. Its fewer moving parts can further lead to the reduction in mechanical losses. Ionic compressors are currently available at the capacities and pressures required at HFS. Ionic liquid compressors have been particularly developed by, Linde, whose ionic liquid compressors have only eight moving parts. This reduces mechanical losses, improves overall efficiency, resulting in $H_2$ compression up to 900 bar in only five steps, at flow rates between 358.7 to 797.5 $Sm^3/h$ with an energy requirement of only 2.7 kWh/kg $H_2$ [20].

### 2.1.4 Centrifugal compressors

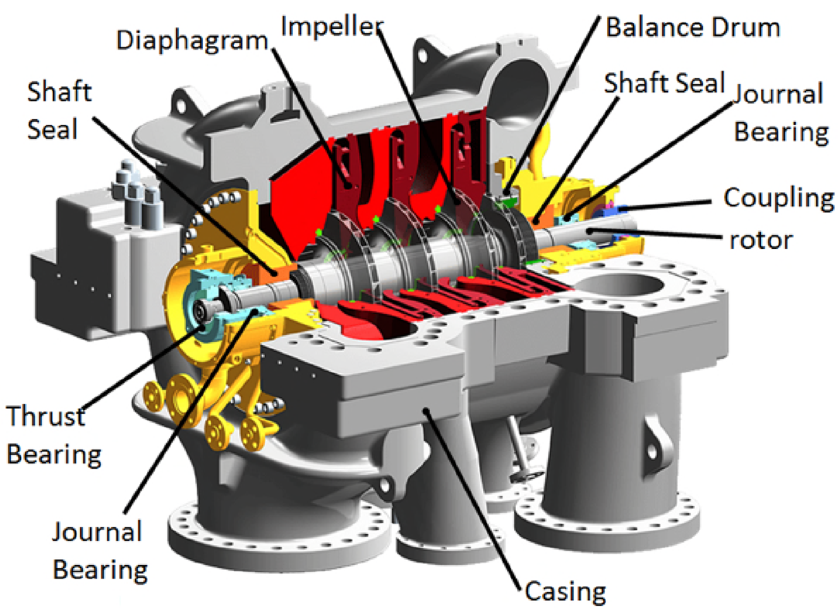


**Figure 2.5** Schematic of a centrifugal compressor.

SOURCE: THE PIPING TALK [21]

Centrifugal compressors are dynamic compressors that are most commonly used in applications that require high throughput and moderate compression ratios [22]. They compress the process gas using a rotating impeller with radial blades that imparts kinetic energy to the process gas by increasing its velocity (**Figure 2.5**). The kinetic energy is converted into pressure increase using a diffuser. Centrifugal compressors are used for pressurizing air and natural gas in petrochemical plants, refineries, gas gathering, and transmission pipelines. Unlike reciprocating compressors, the compression ratio largely depends on the molecular weight of the gas in the centrifugal compressor. Because of the low molecular weight of $H_2$, centrifugal compressors will require impeller tip-speeds around 3X higher for $H_2$ than those used for natural gas [11,23]. Therefore, when high discharge pressures are needed, the impeller speed must be increased, or additional compression stages must be added. Increasing current impeller tip speeds is very challenging due to material strength limitations and $H_2$ embrittlement issues [24]. Research and development activities over last few years has led to the evolution of titanium alloy-based impellers that can operate with 100% $H_2$ at high tip speeds to of ~700 m/s, enabling pressure ratio per stage of 1.26:1 [23]. The design and construction of centrifugal $H_2$ compressors is a multifaceted engineering task because it is affected by several interconnected aerodynamic, thermodynamic and mechanical parameters and they are currently limited to prototype demonstrations [23]. Nonetheless it can be assumed that these centrifugal compressors will be commercially available in coming years when there is a demand for them.

## 2.2 Non-mechanical compressors

Though mechanical compressors are traditionally used for $H_2$ compression, the low density of $H_2$ results in the need of large amount of energy for mechanical compression. Moreover, mechanical compressors suffer from high capital and operating costs due to the presence of many moving parts and $H_2$ embrittlement leading to reliability issues [11]. Non-mechanical $H_2$ compressors have proven to be a valid alternative due to limited moving parts, compact design and safe operation [11]. While these compressors are still in the development phase, it is worth discussing a few promising designs [11].

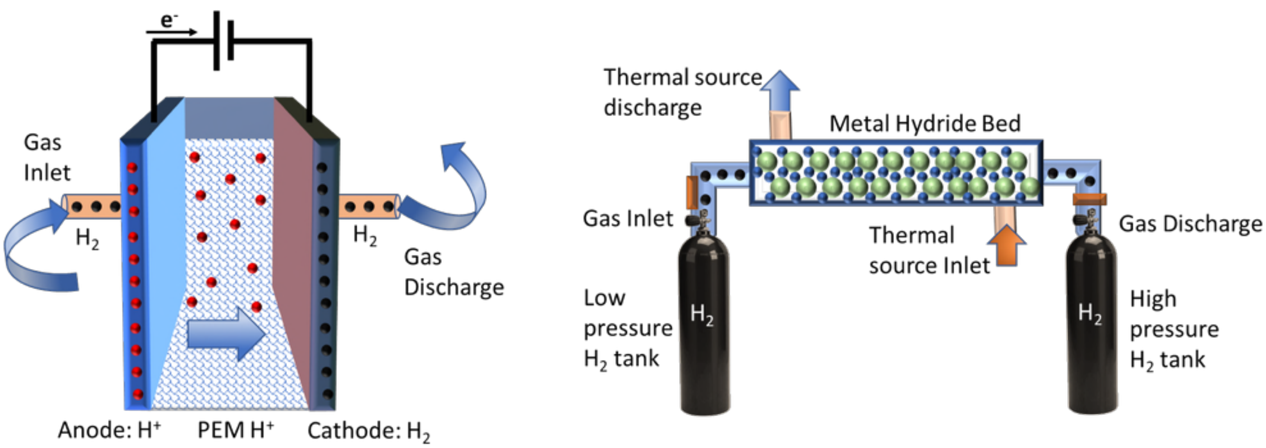


**Figure 2.6** Schematic of an electrochemical compressor (Left) and metal hydride compressor (Right).

SOURCE: ADAPTED FROM REFERENCE [7]

Electrochemical compressors enable isothermal, and consequently efficient compression of $H_2$ and can be used when required flow rates are small. Low-pressure $H_2$ flows through the gas diffusion layer to the anode electrode of an electrochemical compressor where it splits into protons and electrons (**Figure 2.6**). The protons pass through the proton exchange membrane (typically Nafion®) while the electrons flow towards the cathode via the external electrical circuit. At the cathode, the protons and electrons recombine to form $H_2$ molecules again [11]. It is important to note that unlike fuel cells, the cathode in an electrochemical compressor is blocked i.e., no gas/air can flow in. A backpressure regulator is used to attain $H_2$ at desired outlet pressure. The compression process can continue as long as the driving force provided by the potential, i.e., the electric energy supplied to the system, exceeds the internal energy of the system itself [25]. The process is selective for $H_2$, as other gases cannot pass the membrane. In reality, high-pressure electrochemical compression faces significant challenges at high pressures due to back diffusion of $H_2$ across the membrane, decreasing the system's performance [11].

Alternatively, metal hydride compressors use thermal energy for compression utilizing hydride-forming metals, alloys, or intermetallic compounds. The technology is used specifically for compressing $H_2$ gas, whereby these metal hydrides are used to absorb and desorb $H_2$ simply by means of heat and mass transfer in the reaction system (**Figure 2.6**). $H_2$ absorption is an exothermic process while desorption is endothermic and produces an increase in pressure. The selection of suitable metal hydrides is critical with several properties needed such as high $H_2$ storage capacity, fast kinetics, easy activation, and low costs [7,11]. The unique advantage of metal hydride compressors over other compressor technologies is that the system can be powered using waste industrial heat or using renewable solar energy. Recently, the technology was demonstrated for $H_2$ compression in a refueling station for fuel cell powered forklifts [26]. Nevertheless, the efficiency of metal hydride compressors is limited by the heat transfer between the heating/cooling fluid and the metal hydride alloy; efficiencies are generally below 25% at 423 K with average reported efficiencies < 10% [7,11].

# 3 COMPRESSION ENERGY AND POWER CALCULATIONS

The energy needed for compressing gases strongly depends on the required molar flow rate, the compressibility of gas and weakly on the ratio of specific heat. The first factor puts $H_2$ at a disadvantage due to its low molar energy density of ~0.066 kWh/mole versus ~0.248 kWh/mole for natural gas. Secondly, while the behavior of most gases can be approximated using the ideal gas law (PV = nRT), the behavior of $H_2$ deviates significantly from the predictions of the ideal gas model. This deviation results in expansion i.e., the $H_2$ gas occupies more space than what the ideal gas law anticipates [28]. This deviation is accounted using the compressibility factor (Z), whereby Z=1 for an ideal gas. For pressures lower than 600 bar (592 atm), Z is higher for $H_2$ versus other gases such as $CH_4$, $O_2$ and $CO_2$ (**Figure 3.1**). Indeed, by compressing $H_2$ from 1 bar to 700 bars, increases density by only 477 times from 0.0898 g/L to 42.9 g/L. This leads to higher compression power requirement for $H_2$ versus other gases due to the direct dependence on Z.

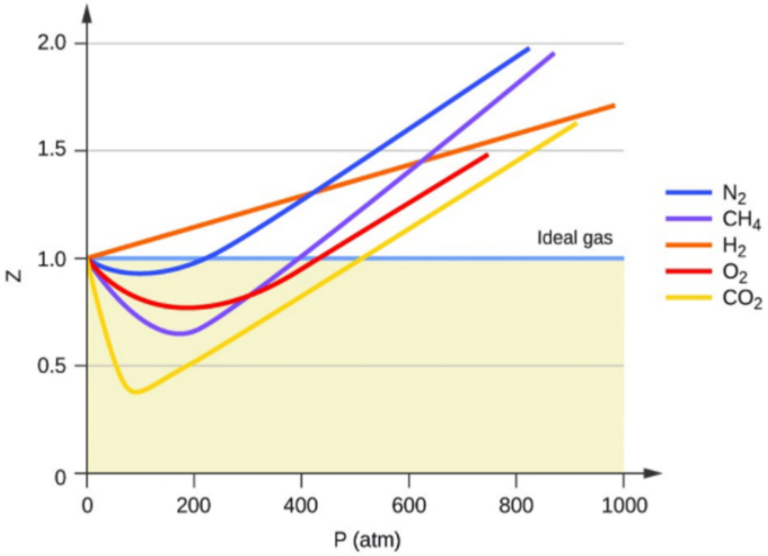


**Figure 3.1** Graph of the compressibility factor (Z) versus pressure for various gases at 273 K.

SOURCE: LUMEN LEARNING [27].

Therefore, due to these factors, compression is the most energy intensive component of the $H_2$ supply chain. If we consider $H_2$ gas initially generated at 20 bar such as that from steam methane reforming (SMR) or autothermal reforming (ATR) units, the lowest possible energy to compress $H_2$ isothermally in a single stage from 20 bar to 350 bar at 20 °C is 1.08 kWh/kg $H_2$ and only 1.48 kWh/kg $H_2$ to compress from 20 to 700 bar. In practice, greater compression energies are required reach these high pressures due to compressor inefficiencies and leaks. The United States Department of Energy (DOE) Technology Validation Project data for compression from on-site $H_2$ production ranges from 1.7 to 6.4 kWh/kg $H_2$, depending on inlet, outlet pressures and compressor efficiencies [29].

Typically, calculations for compressor power are performed for an ideal process. The results are then adapted to the practical scenario employing thermodynamic efficiency factors. Both positive displacement and dynamic compressors are controlled by some basic principles derived from the laws of thermodynamics. There are three ideal processes that can be used to describe the compression process: 1) Isothermal process ($PV^1=C_1$), 2) Isentropic process ($PV^k=C_2$) and 3) or a polytropic process ($PV^n=C_3$) [30]. While either of these processes is acceptable as a basis for evaluating compression power requirements, isentropic process is most common and will be discussed in next section.

## 3.1 Power calculation for single stage compressors

The thermodynamic power calculation for single stage compressors is generally idealized using an isentropic process that is both adiabatic and reversible. *"The compression is said to be isentropic when it is carried out by an ideal compressor, without friction, without internal leakage and while being perfectly insulated"* [31] *(No net transfer of heat or matter)* [32]. This process does not occur as adiabatic and reversible would mean that the initial and final entropies are the same. To account for this non-ideality, an isentropic efficiency factor ($\eta_{isen}$) is used which is defined as ratio of minimum isentropic work to actual work [31]. In other words, the $\eta_{isen}$ accounts for the deviation from isentropic case where all the shaft work is used for compression and actual case where some of the shaft work goes to increasing internal energy or temperature of system. $\eta_{isen}$ is generally quantified and mentioned by the manufacturer or can be quantified if the suction, discharge pressures and temperatures are known using the equation:

$$\eta_{isen} = \frac{T_{suc}}{T_{disc} - T_{suc}}\left[\left(\frac{P_{disc}}{P_{suc}}\right)^{\left(\frac{k-1}{k}\right)} - 1\right]$$

Ideal equation with compressibility,

$$PV = Z\frac{m}{M}RT$$

$$P_{suc}V_{suc} = Z_{suc}\frac{m}{M}RT_{suc}$$

Isentropic Equation

$$PV^k = C$$

$$V = C^{\frac{1}{k}} P^{-\frac{1}{k}}$$

The power for an isentropic (reversible and adiabatic) single stage process is calculated by Compressor power = VdP.

$$P_{Single\ stage(ideal)} = \int_{P_{suc}}^{P_{disc}} V dP$$

$$P_{Single\ stage(ideal)} = \int_{P_{suc}}^{P_{disc}} C^{\frac{1}{k}} P^{-\frac{1}{k}} dP$$

$$P_{Single\ stage(ideal)} = C^{\frac{1}{k}} \frac{P_{disc}^{-\frac{1}{k}+1} - P_{suc}^{-\frac{1}{k}+1}}{-\frac{1}{k}+1}$$

$$P_{Single\ stage(ideal)} = P_{suc}^{1/k} V_{suc} \frac{k}{k-1} \left( P_{disc}^{\frac{k-1}{k}} - P_{suc}^{\frac{k-1}{k}} \right)$$

$$P_{Single\ stage(ideal)} = P_{suc}^{\frac{1}{k}} Z \frac{m}{M} RT_{suc} * \frac{1}{P_{suc}} \frac{k}{k-1} \left( P_{disc}^{\frac{k-1}{k}} - P_{suc}^{\frac{k-1}{k}} \right)$$

$$P_{Single\ stage(ideal)} = P_{suc}^{\frac{1}{k}-1} Z \frac{m}{M} RT_{suc} \frac{k}{k-1} \left( P_{disc}^{\frac{k-1}{k}} - P_{suc}^{\frac{k-1}{k}} \right)$$

$$P_{Single\ stage(ideal)} = \frac{1}{P_{suc}^{1-\frac{1}{k}}} Z \frac{m}{M} RT_{suc} \frac{k}{k-1} \left( P_{disc}^{\frac{k-1}{k}} - P_{suc}^{\frac{k-1}{k}} \right)$$

$$P_{Single\ stage(ideal)} = \frac{1}{P_{suc}^{\frac{k-1}{k}}} Z \frac{m}{M} RT_{suc} \frac{k}{k-1} \left( P_{disc}^{\frac{k-1}{k}} - P_{suc}^{\frac{k-1}{k}} \right)$$

$$P_{Single\ stage(ideal)} = Z \frac{m}{M} RT_{suc} \frac{k}{k-1} \left( \left( \frac{P_{disc}}{P_{suc}} \right)^{\frac{k-1}{k}} - 1 \right)$$

$$\boldsymbol{P_{single\ stage} = \left(\frac{k}{k-1}\right) \left(\frac{Z}{\eta_{isen}}\right) T_{suc}\ (q_M) R \left[ \left(\frac{P_{disc}}{P_{suc}}\right)^{\left(\frac{k-1}{k}\right)} - 1 \right]}$$

Where,

- $P_{single\ stage}$: Power (W)
- $k$: Ratio of specific heat under constant pressure $(C_p)$ to specific heat under constant volume $(C_v)$: $k = \frac{C_p}{C_v} = 1.41\ (Hydrogen)$
- $Z$: Average compressibility factor [dimensionless]: Typically, Z = 1.0 - 1.4 for $H_2$ in the pressure and temperature ranges examined in this report. Z can be determined using the CoolProp excel plugin or other applications such as NIST REFPROP.
- $T_{suc}$: Suction/inlet temperature (K); $T_{disc}$: Discharge/outlet temperature (K)
- $P_{suc}$: Suction or inlet pressure (bar); $P_{disc}$: Discharge or outlet pressure (bar)
- R: Universal gas constant, R = 8.314 J/mol.K
- $q_M$: Molar flow rate (mole/s)

## 3.2 Power calculation for multistage compressors

When overall high compression ratios (> 2) are needed, compression is usually carried out in multiple stages with intercooling and same per-stage compression ratio. This leads to significant power savings versus single stage compression and the advantage is illustrated in **Figure 3.2** [31]. Thermodynamic power calculation for multistage compressors with intercooling can be done using the isentropic model based on several assumptions:

i. The work at each stage is equal.
ii. Pressure ratio per stage is equal.
iii. Temperature of gas in intercoolers is cooled to original suction temperature at first stage.
iv. There is no pressure drop or heat losses that occur in intercoolers between stages.

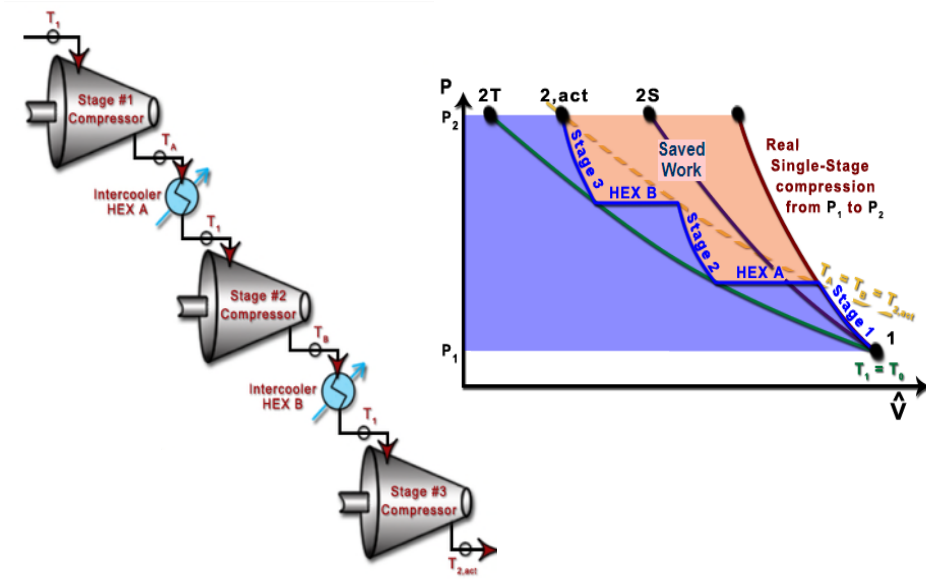


**Figure 3.2** Multistage compressors with intercooling and corresponding P-V diagram in blue line.

SOURCE: ADAPTED FROM LEARN THERMO [33].

Therefore, considering N = number of compressor stages and $x$ = compression ratio for single stage,

$$\left(\frac{P_{disc}}{P_{suc}}\right) = (x)^N$$

$$N = \frac{log\left(\frac{P_{disc}}{P_{suc}}\right)}{log(x)}$$

where $x$ = 2.1-4.0 as reported in literature. (Source: References [11,15,18,19,22])

The power for an isentropic (reversible and adiabatic) multistage process is calculated by,

$$P_1 = P_2 = P_3 = P_N = \left(\frac{k}{k-1}\right)\left(\frac{Z}{\eta_{isen}}\right)T_{suc}\,(q_M)R\left[(x)^{\left(\frac{k-1}{k}\right)} - 1\right]$$

$$P_{multi\ stage} = P_1 + P_2 + P_3 + \ldots + P_N$$

$$\boldsymbol{P_{multi\ stage} = N\left(\frac{k}{k-1}\right)\left(\frac{Z}{\eta_{isen}}\right)T_{suc}(q_M)R\left[\left(\frac{P_{disc}}{P_{suc}}\right)^{\left(\frac{k-1}{Nk}\right)} - 1\right]}$$

The rate compressor power can be calculated using equation:

$$Compressor\ motor\ rating\ (W) = \frac{P_{multistage}}{Motor\ efficiency\ (\%)}$$

## 3.3 Isentropic efficiency

The above equations can be used when there is no heat or pressure loss in the intercoolers and the $\eta_{isen}$ number used in calculating compression power should specifically be taken for $H_2$ compressors. There is limited literature available concerning $\eta_{isen}$ for $H_2$ compressors, but depending on the compressor type, size (scale) and design, $\eta_{isen}$ varies in the range of ~55-80% for most compressor designs [34]. It is well known that larger compressors are more efficient than smaller ones [35,36] and Amos [36] states that large compressors have an $\eta_{isen}$ of ~65-70%, while small compressors have an $\eta_{isen}$ of ~40-50%, though there is no quantification for "big" and "small". The $H_2A$ model developed by National Renewable Energy Laboratory (NREL) uses an $\eta_{isen}$ of 88% for large scale reciprocating compressors up to max capacity of 16 MW and an $\eta_{isen}$ of 65% for smaller scale diaphragm/reciprocating compressors with flow rates up to 500 kg/h [37]. A more detailed look at the Tables 2-18 and 2-22 in $H_2A$ Analysis Results report [37] suggests that $\eta_{isen}$ can vary between 75-88% for large reciprocating compressors and between 45-70% for smaller reciprocating or diaphragm compressors [37]. The HDSAM model follows the $H_2A$ model by assuming $\eta_{isen}$ of 88% for large scale central compressors and $\eta_{isen}$ of 75% for smaller scale refueling station compressors [38]. The DOE validation data for small scale (1,000 kg/day) reciprocating compressors reports an $\eta_{isen}$ of ~56% [29].

# 4 CHOICE OF COMPRESSOR, PRIME MOVER, AND COST CALCULATIONS

Only after careful evaluation of various parameters can the proper compressor type and number of stages be determined. Some important parameters include [39]:

- Volume and mass flow of the gas.
- Inlet or suction pressure
- Outlet or Discharge pressure
- Inlet or Suction temperature
- Specific gravity of the gas to be compressed.

A chart of the inlet volume flow versus discharge pressure (**Figure 4.1**; Adapted from Reference [9]) reveals that centrifugal compressors are appropriate for high flow applications while reciprocating compressors are better suited to low flow rates.

Furthermore, it is important to note that compressors are driven by different types of engines such as reciprocating engines, gas turbines or electric motors which are also known as “prime movers”. Reciprocating engines are like internal combustion engines where gas is ignited in a chamber to move a piston in a reciprocating movement. In contrast gas turbines rely on hot exhaust gas to run a power turbine in a rotational movement which in turn drives the centrifugal compressor. Recently pipeline companies are designing inlet pipeline compressors using modern electric motors. These electric motors are more reliable and efficient than either reciprocating engines or gas turbines with a faster ramp up. An added advantage of using electric motors is that they do not emit toxic emit $NO_x$ and $CO_2$ at the point of use. However, the availability and reliability of grid electricity is the biggest concern when using electric motors. The selection of the compressor dictates the choice of the prime mover as well. In the natural gas industry, reciprocating compressors are generally driven by natural gas-powered reciprocating engines and centrifugal compressors are driven by natural gas turbines.

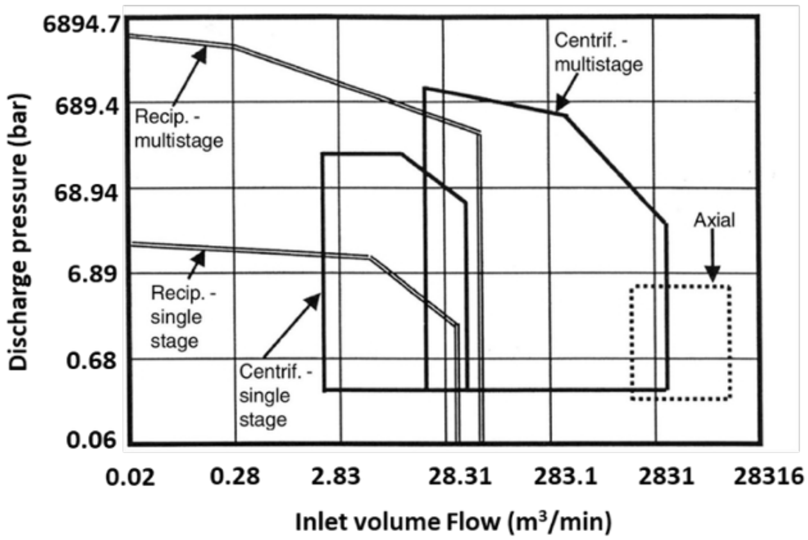


**Figure 4.1** Compressor selection based on discharge pressure (bar) and flow rate ($m^3$/min).

SOURCE: ADAPTED FROM PETROWIKI [9]

## 4.1 Assumptions used

For the analysis presented in this report we assume compressors are driven by electric motors with a motor efficiency of 95%. Furthermore, we assume the use of large centrifugal $H_2$ compressors for pipelines application following the HDSAM model developed by Argonne National Laboratory [30,32]. We use an isentropic efficiency of 80% and fix the maximum compressor capacity at 16,000 kW. Following the HDSAM model we also assume the use of smaller diaphragm compressors for HFS applications, as they are more reliable. An isentropic efficiency of 60% and maximum capacity 1,000 kW was assumed for these smaller diaphragm compressors. For higher power levels multiple compressors were assumed.

## 4.2 Determining the capital cost of a hydrogen compressor

If production costs are not considered, then compression of $H_2$ dominates the delivery cost of $H_2$ in the supply chain [6]. The purchase price of a compressor can vary from several thousand dollars to millions of dollars depending on scale and compression ratio required. Therefore, generally compression must be done at a large scale to remunerate this cost. Academic literature and industry both use empirical cost correlations or rules of thumb to determine how much a compressor will cost based on its size. The cost of compressor is calculated using these correlations and is based on the required compressor motor power (determined in **Section 0**). In this section we describe the calculation of capital costs associated with compressors using the correlations provided in the HDSAM model [32].

## 1) Total Installed Costs (TIC)

The cost is defined here as the total installed cost (TIC), which is the cost of the compressor itself (uninstalled cost (UC)) and the labor/parts required to install it (installation factor (IF)). Correlations and factors are provided in more detail in **Table 4.1**.

**TIC = UC * IF**

Some notable resources for more information include:

- $H_2A$ / HDSAM
- Perry's Chemical Engineering Handbook

**Table 4.1** Detailed cost assumptions for total installed cost of hydrogen compressors.

| Compressor Type | Value / Conversion factor | Notes |
|---|---|---|
| **High flow rate – Moderate compression ratio** | Pipeline compressor:<br>UC [2019 C$] = 3,083.3* kW^SF, where Scale Factor (SF) = 0.8335; IF = 2.0 | Source: HDSAM<br>kW = kW motor power<br>Converted to 2019 C$ by escalating cost using CEPCI (2007 = 525.4, 2013=567.30, 2019 = 619.2) followed by the C$/US$ exchange rate.<br>0.75 US$/C$ (2019 average) |
| **Small flow rate – High compression ratio** | Terminal storage or Refueling station main:<br>• For 350 bars refueling<br>UC [2019 C$] = 63,684.6 * kW^SF, where SF = 0.4603; IF = 1.3<br>• For 700 bars refueling<br>UC [2019 C$] = 62,909.9 * kW^SF, where SF = 0.6038; IF = 1.3<br>Terminal loading or refueling station booster compressors:<br>UC [2019 C$] = 8,731.88 * kW; IF = 1.3 | Source: HDSAM<br>kW = kW motor power<br>Converted to 2019 C$ by escalating cost using CEPCI (2007 = 525.4, 2013=567.30, 2019 = 619.2) followed by the C$/US$ exchange rate.<br>0.75 US$/C$ (2019 average) |

## 2) Indirect costs

The simplest way to determine indirect costs is by calculating it as a percentage of the TIC. The advantage of this approach is that larger and more complex projects, which have a higher TIC, have higher associated costs. The indirect costs used in this whitepaper are based on established literature (SOURCE: HDSAM) and are detailed below:

- **Site preparation = 5% of TIC**; This cost helps cover purchase of the land, site preparation costs including building infrastructure and installation of electrical, water, HVAC, and sewer systems. Furthermore, this would also cover construction of internal roads, walkways, and parking lots [40].

- **Engineering & Design = 10% of TIC**; This helps cover salaries and overhead expenditures for the engineering and project management personnel on the project [40].

- **Project Contingency = 10% of TIC**; Unforeseen events, such as project risks or uncertainties are factored in. This cost also helps cover delays caused by storms and strikes, as well as minor design modifications and unanticipated price rises [40].

- **Permitting = 3% of TIC**; The price of obtaining the appropriate approvals to design and install the control equipment are covered by these indirect costs. This is a site-specific expense, meaning that the costs sustained by one facility may not be easily transferred to another.

- **Owner's Costs = 12% of TIC;** For significant investments, an owner's cost component is used to account for additional owner's engineering, prospective construction debt origination, closure costs, and due diligence studies. The 12% estimate is based on construction experience and is only applied for large scale compressors [41].

$$Indirect\ costs = 40\%\ for\ large\ pipeline\ compressors; 28\%\ for\ small\ HFS\ compressors$$

## 3) Total Capital Investment (TCI)

Once the TIC of a compressor and indirect costs are known, the total capital investment (TCI) can be determined. TCI is the capital expenditure (Capex) at the beginning of a project and can occur over several years depending on how long it takes to design & procure equipment, deliver it to a project site, and construct the project.

$$TCI\ (Capex) = TIC + Indirect\ costs$$

## 4) Annualized TCI

The annualized TCI converts the TCI, which usually occurs at the beginning of the project lifecycle, into an annual expenditure so it can be compared equitably with other annual expenditures such as electricity costs and non-energy OPEX (Discussed in next sections).

$$Annualized\ capex \left[\frac{\$}{yr}\right] = TCI\ (\$) * Capital\ recovery\ factor\ (CRF)$$

$$CRF = \frac{i\,(1+i)^n}{(1+i)^n - 1};\ (i - Discount\ rate\ (\%);\ \ n - Compressor\ lifetime)$$

- Finally, the TCI can be normalized to the $H_2$ throughout using the equation:

$$Capex_{comp}\left[\frac{\$}{kgH_2}\right] = \frac{\left(Annualized\ TCI\ \left[\frac{\$}{yr}\right]\right)}{\left(Availability\ [\%] \times Design\ Capacity\left[\frac{kgH_2}{day}\right] \times 365\left[\frac{days}{yr}\right]\right)}$$

In the above equation, availability is the fraction of the year the asset (a compressor in this case) can operate. When multiplied with the compressor's design capacity, it determines how much $H_2$ can be compressed (throughput) in a year. This white paper assumes the availability is related to compression at a large-centralized $H_2$ production facility that only needs to be taken offline for maintenance for few weeks of the year or any unplanned outage i.e., ~ 10%, therefore availability = 90%. If $H_2$ compression was used for a production process that only runs a small fraction of the year, such as an electrolyzer using electricity from a wind farm, then setting the availability to the capacity factor of the wind farm (30-40%) may be more appropriate.

## 4.3 Determining the operating cost of a hydrogen compressor

The followed factors need to be considered to determine the operating costs associated with a $H_2$ compressor.

### 1) Energy/Electricity costs

In compression, the energy used is the electricity consumed to power the compressor motor. Energy costs are a form of variable operating expenditure (OPEX), broken out because energy use (and associated cost) is of particular interest in the $H_2$ supply chain. The formula for energy cost per year for compression, can be expressed as:

$$Electricity\ cost\ \left(\frac{\$}{yr}\right) = \ Compressor\ rating\ (kW) * Operating\ hours\ \left(\frac{hr}{yr}\right) * Electricity\ price\ \left(\frac{\$}{kWh}\right)$$

**Note:** Electricity price = Average industrial electrical price in Alberta for 2019 (0.11 C$/kWh)

$$Energy_{comp}\left[\frac{\$}{kgH_2}\right] = \frac{\left(Electricity\ cost\ \left[\frac{\$}{yr}\right]\right)}{\left(Availability\ [\%] \times Design\ Capacity\left[\frac{kgH_2}{day}\right] \times 365\left[\frac{days}{yr}\right]\right)}$$

## 2) Non-energy OPEX:

Non-energy OPEX costs found in literature (SOURCE: HDSAM) for $H_2$ compression include labor costs and fixed O&M costs.

$$Non - energy\ OPEX \left(\frac{\$}{yr}\right) = Total\ labor \left(\frac{\$}{yr}\right) + Fixed\ O\&M \left(\frac{\$}{yr}\right)$$

### i. Total labor cost:

$$Total\ labor \left(\frac{\$}{yr}\right) = Direct\ labor \left(\frac{\$}{yr}\right) + Indirect\ labor \left(\frac{\$}{yr}\right)$$

**Direct labor cost:**

$$Direct\ labor \left(\frac{\$}{yr}\right) = Annual\ hours \left(\frac{hours}{yr}\right) * Labor\ rate \left(\frac{\$}{hour}\right)$$

$$Annual\ labor\ hours \left(\frac{hr}{yr}\right) = 288 * \left(\frac{x}{100{,}000}\right)^{0.25}$$

where x = compressor flow rate (kg $H_2$/day) and Labor rate = 49.66 2019 $CAD/hour. (SOURCE: HDSAM)

**Overhead indirect labor cost:**

$$Indirect\ labor \left(\frac{\$}{yr}\right) = Direct\ labor \left(\frac{\$}{yr}\right) * Indirect\ labor\ factor\ (\%)$$

$Indirect\ labor\ factor = 50\%$; used to consider the cost of overhead (i.e., head office, personnel)

### ii. Fixed O&M costs:

All non-labor fixed O&M costs ($/yr) are calculated as a fraction of the TCI or TIC (Section 4.1) to reflect that the larger and more complex, and therefore more expensive, projects have higher upkeep costs throughout the project life.

- Operating, maintenance and repairs = 4% of TIC
- Insurance = 1% of TCI
- Property tax = 1% of TCI
- Licensing and permitting = 0.1% of TCI

$$Non - energy\ OPEX_{comp} \left[\frac{\$}{kgH_2}\right] = \frac{\left(Non - energy\ OPEX \left[\frac{\$}{yr}\right]\right)}{\left(Availability\ [\%] \times Design\ Capacity \left[\frac{kgH_2}{day}\right] \times 365 \left[\frac{days}{yr}\right]\right)}$$

## 4.4 Determining the lifecycle cost of hydrogen compression

A useful indicator of the relative economic viability of $H_2$ is the Levelized Cost of $H_2$ (LCOH). The simple definition of LCOH for compression is as follows and the detailed assumptions are in **Table 4.2**

$$LCOH_{comp}\left[\frac{\$}{kgH_2}\right] = Capex_{comp}\left[\frac{\$}{kgH_2}\right] + Non-energy\ OPEX_{comp}\left[\frac{\$}{kgH_2}\right] + Energy_{comp}\left[\frac{\$}{kgH_2}\right]$$

This technical brief report focuses on the LCOH of compression only. However, it is possible to determine the LCOH for multiple units in a supply chain to determine the overall cost of $H_2$ as an energy carrier in different applications. For example, in a supply chain where $H_2$ is generated as a zero-emission fuel for heavy duty trucks, delivered via pipeline to a HFS (**Figure 1.1**), the LCOH could be determined as:

$$LCOH_{Total} = LCOH_{Prod} + LCOH_{comp} + LCOH_{Pipeline} + LCOH_{HFS}$$

This is a simplified example that does not necessarily consider all essential steps for a supply chain.

**Table 4.2** Detailed economic assumptions for calculating the LCOH for compression.

| Factor | Value / Conversion factor | Notes |
|---|---|---|
| **Exchange rate** | 0.75 US$/C$ | Source: 2019 average |
| **Inflation Rate** | e.g., CAPEX from 2007 to 2019 = 619.2 / 525.4 = 1.179 | Source: CEPCI – Plant Cost Index for CAPEX/Equipment (US$)<br>2007 = 525.4. 2013 = 567.30.<br>2019 = 619.2. |
| **Discount Rate** | 8% | Discount rate = weighted average cost of capital (WACC) (Assumed) |
| **Project Lifetime** | 15 years | Source: HDSAM |
| **Electricity cost** | 0.11 C$/kWh$_e$ | Rate Alberta Industrial Electricity in Alberta; Source: NRCAN |
| **Availability** | 90% | Assumed |

# 5 ANALYSIS AND RESULTS

## 5.1 Analysis of a large hydrogen compressor for use in pipelines

In this example we will demonstrate the energy and cost calculations for a large centrifugal compressor driven by an electric motor to be used for $H_2$ pipelines. The capacity needed is 50,000 kg $H_2$/day, inlet temperature of 305.15 K, an inlet (suction) pressure of 20 bar and required outlet (discharge) pressure of 70 bar. A compression ratio per stage (x) of 2.1, isentropic efficiency ($\eta_{isen}$) of 80% and electric motor efficiency of 95% is considered as the model. The steps involved are listed below with the results presented in **Table 5.1**.

**Table 5.1** Power and cost calculation of a large centrifugal $H_2$ compressor for pipeline use.

| Steps | Calculation | Notes |
|---|---|---|
| **N** | $= \frac{[\log(70/20)]}{\log(2.1)} = 2$ | $N = \frac{\log\left(\frac{P_{disc}}{P_{suc}}\right)}{\log(x)}$; Round N up to the nearest whole number, i.e., 1.7 → 2. |
| $\mathbf{T_{disc}}$ | $= 305.15\left(1 + \frac{\left(\frac{70}{20}\right)^{\left(\frac{1.4-1}{2*1.4}\right)} - 1}{0.8}\right) = 379.9\ \mathrm{K}$ | $T_{disc} = T_{suc}\left[1 + \frac{\left(\frac{P_{disc}}{P_{suc}}\right)^{\left(\frac{k-1}{Nk}\right)} - 1}{\eta_{isen}}\right]$ |
| $\mathbf{P_{avg}(bar)}$ and $\mathbf{T_{avg}(K)}$ | $P_{avg} = \frac{2}{3}\left(\frac{70^3 - 20^3}{70^2 - 20^2}\right) = 49.62\ \mathrm{bar}$ <br> $T_{avg} = \frac{305.15 + 379.9}{2} = 342.5\ \mathrm{K}$ | $P_{avg} = \frac{2}{3}\left(\frac{P_{disc}^3 - P_{suc}^3}{P_{disc}^2 - P_{suc}^2}\right)$ [17] <br> $T_{avg} = \frac{T_{suc}+T_{disc}}{2}$ |
| **Z** | At calculated $T_{avg}$ and $P_{avg}$; $Z = 1.024$ | Using CoolProp excel plugin |
| $\mathbf{q_M}$ | $= \frac{\left(\frac{50{,}000}{0.002}\right)}{24*60*60} = 289.35\,\frac{\mathrm{moles}}{\mathrm{sec}}$ | Molar flow rate from Mass flow rate |
| Actual Compressor power (kW) | $= 2\left(\frac{1.4}{1.4-1}\right)\left(\frac{1.024}{0.8}\right)305.15\ (289.35)\ 8.314\left[\left(\frac{70}{20}\right)^{\left(\frac{1.4-1}{2*1.4}\right)} - 1\right]$ <br> $= 1{,}289{,}410\ \mathrm{W} = 1{,}289.41\ \mathrm{kW}$ | Power = <br> $N\left(\frac{k}{k-1}\right)\left(\frac{Z}{\eta_{poly}}\right)T_{suc}\ (q_M)R\left[\left(\frac{P_{disc}}{P_{suc}}\right)^{\left(\frac{k-1}{Nk}\right)} - 1\right]$ |
| Rated Compressor Power (kW) | $= \frac{1{,}289.41\ \mathrm{kW}}{0.95}$ <br> $= \mathbf{1{,}357.28\ kW}$ | Rated Compressor Power (kW) <br> $= \frac{\text{Actual Compressor Power (kW)}}{\text{Motor Efficiency (\%)}}$ |

| Steps | Calculation | Notes |
|---|---|---|
| Energy Intensity ($kWh/kg\ H_2$) | $= (1357.28\ \text{kW} * 24\frac{\text{hrs}}{\text{day}})/\ (50{,}000\ \text{kg/day})$ <br> $= \mathbf{0.65\ kWh/kg\ H_2}$ | |
| UC (2019 C$) | = 3083.35 * 1,357.28 ^ 0.8335 <br> $= \mathbf{1,259,222.1\ C\$}$ | UC = 3083.3 * [kW]^SF, where SF = 0.8335 |
| TIC (2019 C$) | = $1,259,222.11 * 2 <br> $= \mathbf{2,518,444.5\ C\$}$ | TIC = UC * IF; where IF = 2. |
| TCI (2019 C$) | = $2,518,444.58+ (0.4* $2,518,444.58) <br> $= \mathbf{3,525,822.4\ C\$}$ | TCI = TIC + Indirect Costs; where Indirect costs = 40% TIC |
| Annualized TCI (2019 C$/yr) | $\text{CRF} = \frac{0.08(1+0.08)^{15}}{(1+0.08)^{15}-1} = 0.1168$ <br> Annualized TCI = $3,525,822.41 * 0.1168 <br> $= \mathbf{411,920.23\ C\$/yr}$ | Annualized TCI $\left[\frac{\$}{\text{year}}\right]$ = TCI ($) * Capital recovery factor (CRF) <br> $\text{CRF} = \frac{i\,(1+i)^n}{(1+i)^n-1}$ (i – Discount rate (%); n- Plant lifetime) |
| Electrical energy cost (2019 C$/yr) | $= 1{,}357.28\ \text{kW} * 24\frac{\text{hrs}}{\text{day}} * 365\frac{\text{days}}{\text{year}} * 0.90 * \frac{0.11\$}{\text{kWh}}$ <br> $= \mathbf{1,177,085.98\ C\$/yr}$ | Electrical energy cost ($/yr) = Power (kW) * Operating hours (hours/yr) * Electricity price ($/kWh) |
| Direct labor cost (2019 C$/yr) | = (288*(50,000/100,000) ^0.25) *49.66 <br> $= \mathbf{12,026.67\ C\$/yr}$ | Direct labor cost ($/yr) = Annual hours(hours/yr) * Labor cost ($/hour) <br> Annual hours(hours/yr) = 288 * (x/100000)^0.25 |
| Indirect labor cost (2019 C$/yr) | = 12,026.67 * 50% <br> $= \mathbf{6,013.33\ C\$/yr}$ | Indirect labor cost $\left(\frac{\$}{\text{yr}}\right)$ = Direct labor $\left(\frac{\$}{\text{yr}}\right)$ * Indirect labor factor (%) <br> Indirect Labor factor = 50% |
| **Fixed O&M (2019 C$/yr)** | = (0.04 * $ 2,518,444.58) + (0.021 * $3,525,822.41) <br> $= \mathbf{174,780.05\ C\$/yr}$ | ✓ O&M & repairs = 4% of TIC <br> ✓ Insurance = 1 % of TCI <br> ✓ Property tax = 1 % of TCI <br> License & permits = 0.1% of TCI |
| **Non – Energy OPEX (2019 C$/yr)** | = $12,026.67 + $ 6,013.33 + $174,780.05 <br> $= \mathbf{192,820.06\ C\$/yr}$ | O&M $\left(\frac{\$}{\text{yr}}\right)$ = Total labor $\left(\frac{\$}{\text{yr}}\right)$ + Additional O&M $\left(\frac{\$}{\text{yr}}\right)$ |

| Steps | Calculation | Notes |
|---|---|---|
| $Capex_{comp}$ (2019 C$/kg $H_2$) | $= \frac{\$411{,}920.23/yr}{(0.90 * 50000 * 365)}$ $= \mathbf{0.025\ C\$/kg\ H_2}$ | $Capex_{comp}\left[\frac{\$}{kgH_2}\right] = \frac{\left(\text{Annualized TCI}\left[\frac{\$}{yr}\right]\right)}{\left(\text{Availability}[\%]\times\text{DesignCapacity}\left[\frac{kgH_2}{day}\right]\times 365\left[\frac{days}{year}\right]\right)}$ |
| Non-energy $OPEX_{comp}$ (2019 C$/kg $H_2$) | $= \frac{\$192{,}820.06/yr}{(0.90 * 50000 * 365)}$ $= \mathbf{0.011\ C\$/kg\ H_2}$ | $\text{Non}-\text{energy Opex}_{comp}\left[\frac{\$}{kgH_2}\right] = \frac{\left(\text{Non}-\text{energy Opex}_{comp}\left[\frac{\$}{yr}\right]\right)}{\left(\text{Availability}[\%]\times\text{DesignCapacity}\left[\frac{kgH_2}{day}\right]\times 365\left[\frac{days}{yr}\right]\right)}$ |
| $Energy_{comp}$ (2019 C$/kg $H_2$) | $= \frac{\$1{,}177{,}085.98/yr}{(0.90 * 50000 * 365)}$ $= \mathbf{0.071\ C\$/kg\ H_2}$ | $Energy_{comp}\left[\frac{\$}{kgH_2}\right] = \frac{\left(\text{Electrical Energy costs}\left[\frac{\$}{yr}\right]\right)}{\left(\text{Availability}[\%]\times\text{DesignCapacity}\left[\frac{kgH_2}{day}\right]\times 365\left[\frac{days}{yr}\right]\right)}$ |
| $LCOH_{comp}$ (2019 C$/kg $H_2$) | $= 0.025 + 0.011 + 0.071 = \mathbf{0.108\ C\$/kg\ H_2}$ | $LCOH_{comp}\left[\frac{\$}{kgH_2}\right] = Capex_{comp}\left[\frac{\$}{kgH_2}\right] + \text{Non}-\text{energy OPEX}_{comp}\left[\frac{\$}{kgH_2}\right] + Energy_{comp}\left[\frac{\$}{kgH_2}\right]$ |

## 5.1.1 Effect of design capacity

The TCI (MM 2019 C$) as a function of compressor size or capacity is shown in **Figure 5.1(a).** As mentioned earlier the uninstalled capital costs were calculated using correlations built by the $H_2A$ and HDSAM models. The earlier versions of the HDSAM models were based on cost data of two and three stage reciprocating compressors which was assembled from data supplied by Air Liquide, Neuman & Esser, Burckhardt Compression, Ariel Compressors, and Dresser-Rand [37]. However the latest versions of the HDSAM uses cost projections based on an existing centrifugal $H_2$ compressor design and protype developed by Concepts NREC for the US Department of Energy (DOE) [23]. The cost correlation does not predict a significant cost reduction with compressor capacity with a scaling factor of ~0.83 for TCI. A breakdown of $LCOH_{comp}$, CAPEX, electricity and non-energy operating costs is shown in **Figure 5.1(b)** which shows the dominant contribution from energy/electricity costs. More importantly, the results indicate that at capacities > 100 $t_{H2}$/day, there is no advantage of scaling up compressors. This is because the electricity/energy costs become the dominant contributor to the $LCOH_{comp}$, which is independent of scale/capacity.

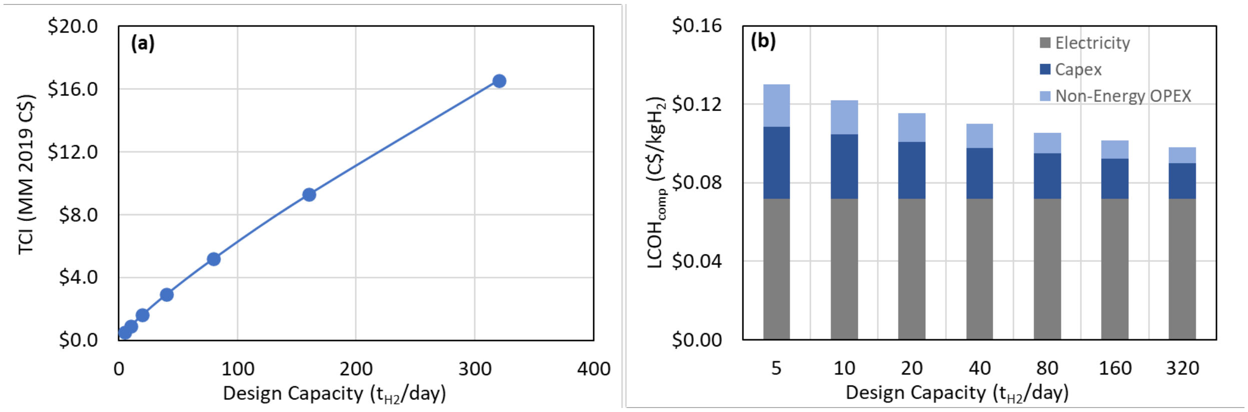


**Figure 5.1** Impact of design capacity on (a) TCI (MM 2019 C$) and (b) $LCOH_{comp}$ (2019 C$/kg $H_2$) for large scale centrifugal $H_2$ compressors.

## 5.1.2 Effect of suction/inlet pressure and isentropic efficiency

We have also analyzed the effect of two key parameters i.e., suction/inlet pressure and isentropic efficiency on the energy requirement and cost of compression as show in **Figure 5.2**. As expected, the suction pressure has a significant effect on the energy/power requirement and thereby both capital and electricity costs decrease with increase in suction pressure as shown in **Figure 5.2(c).** This leads to significant decrease in $LCOH_{comp}$ indicating that the pressure drop in pipelines should be minimized to lower the cost of compression. The isentropic efficiency of the compressor also has a similar but more subtle effect on the energy requirement and cost of compression as shown in **Figure 5.2(d-f)**. The combined effect of these two parameters is depicted in the contour plots of **Figure 5.3** showing the dominant effect of suction pressure on both energy intensity and $LCOH_{comp}$.

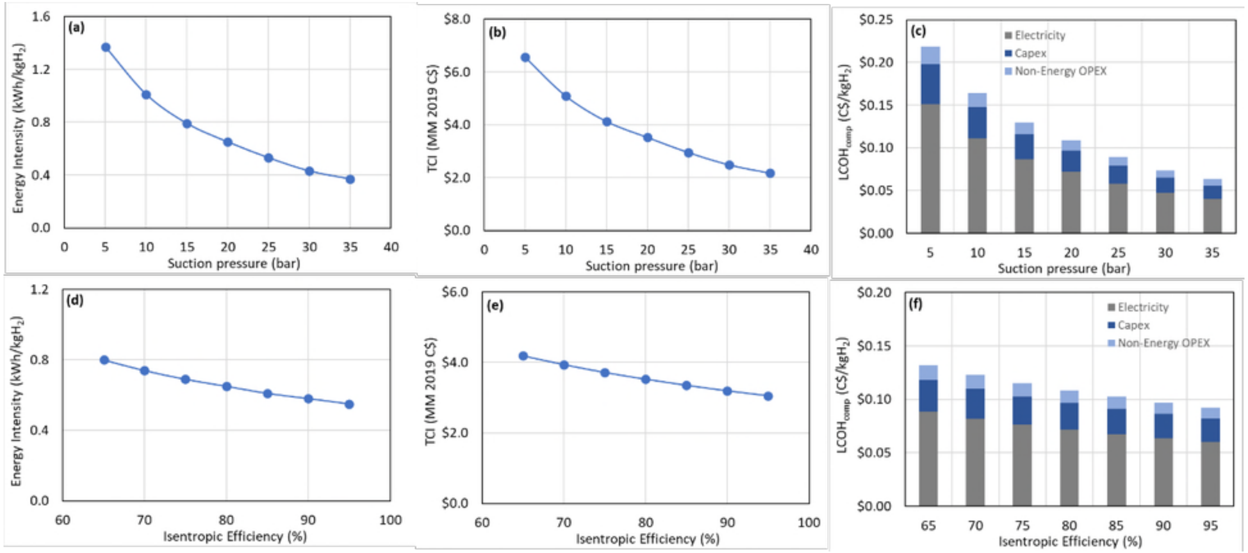


**Figure 5.2** Impact of suction pressure (a-c) and isentropic efficiency (d-f) on performance of large scale centrifugal $H_2$ compressors.

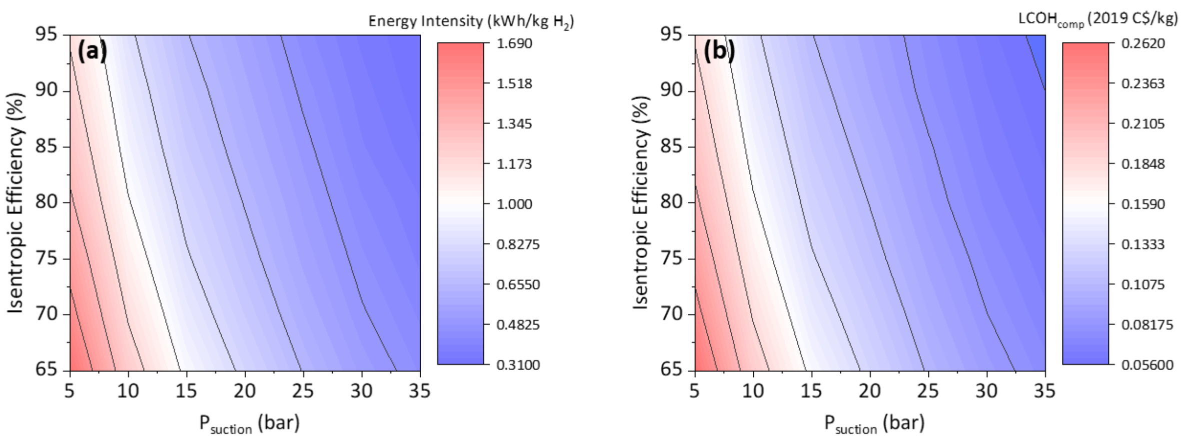


**Figure 5.3** Combined effect of suction pressure and isentropic efficiency on (a) energy intensity (kWh/kg $H_2$) and (b) $LCOH_{comp}$ (2019 C$/kg $H_2$) for large scale centrifugal $H_2$ compressors

## 5.2 Analysis of a small hydrogen compressor for use at HFS

In this example we calculate the energy and cost of a small $H_2$ diaphragm compressor to be used at HFS. The capacity needed is 2,000 kg $H_2$/day, inlet temperature of 305.15 K, an inlet (suction) pressure of 20 bar and required outlet (discharge) pressure of 500 bar. A diaphragm compressor with compression ratio per stage ($x$) of ~3.1, isentropic efficiency ($\eta_{isen}$) of ~60% and motor efficiency ~95% is considered as the model. The steps involved are listed below with the results presented in **Table 5.2**.

**Table 5.2** Power and cost calculation of a small diaphragm hydrogen compressor for HFS's.

| Steps | Calculation | Notes |
|---|---|---|
| **N** | $= \frac{[\log(500/20)]}{\log(3.1)} = 3$ | $N = \frac{\log\left(\frac{P_{disc}}{P_{suc}}\right)}{\log(x)}$; Round N up to the nearest whole number, ie. 1.7 → 2 |
| $\mathbf{T_{disc}}$ | $= 305.15\left(1 + \frac{\left(\frac{500}{20}\right)^{\left(\frac{1.4-1}{3*1.4}\right)} - 1}{0.6}\right) = 487.6$ | $T_{disc} = T_{suc}\left[1 + \frac{\left(\frac{P_{disc}}{P_{suc}}\right)^{\left(\frac{k-1}{Nk}\right)} - 1}{\eta_{isen}}\right]$ |
| $\mathbf{P_{avg}(Pa)}$ and $\mathbf{T_{avg}(K)}$ | $P_{avg} = \frac{500 + 20}{2} = 260\text{ bar}$<br>$T_{avg} = \frac{305.15 + 487.6}{2} = 396.4\text{ K}$ | $P_{avg} = \frac{2}{3}\left(\frac{P_{disc}^3 - P_{suc}^3}{P_{disc}^2 - P_{suc}^2}\right)$ [17]<br>$T_{avg} = \frac{T_{suc}+T_{disc}}{2}$ |
| **Z** | At calculated $T_{avg}$ and $P_{avg}$; $Z = 1.126$ | Using CoolProp excel plugin |
| $\mathbf{q_M}$ | $= \frac{\left(\frac{2{,}000}{0.002}\right)}{24*60*60} = 11.57\frac{\text{moles}}{\text{sec}}$ | Molar flow rate from Mass flow rate |
| Actual Compressor power (kW) | $= 3\left(\frac{1.4}{1.4-1}\right)\left(\frac{1.12}{0.6}\right)305.15\ (11.57)\ 8.314\left[\left(\frac{500}{20}\right)^{\left(\frac{1.4-1}{3*1.4}\right)} - 1\right]$<br>$= 207{,}702\text{ W} = 207.7\text{ kW}$ | Power = $N\left(\frac{k}{k-1}\right)\left(\frac{Z}{\eta_{poly}}\right)T_{suc}\ (q_M)R\left[\left(\frac{P_{disc}}{P_{suc}}\right)^{\left(\frac{k-1}{Nk}\right)} - 1\right]$ |
| Rated Compressor Power (kW) | $= \frac{207.7\text{ kW}}{0.95}$<br>$= \mathbf{218.63\ kW}$ | Rated Compressor Power (kW) $= \frac{\text{Actual Compressor Power (kW)}}{\text{Motor Efficiency (\%)}}$ |
| Energy Intensity (kWh/kg $H_2$) | $= (218.63\text{ kW} * 24\frac{\text{hrs}}{\text{day}}) / (2{,}000\text{ kg/day})$<br>$= \mathbf{2.62\frac{kWh}{kg\ H2}}$ | |

| Steps | Calculation | Notes |
|---|---|---|
| UC<br>(2019 C$) | $= 63{,}684.6 * 218.63 \wedge 0.4603$<br>$= \mathbf{760,338.49\ C\$}$ | <u>For 350 bars refueling</u><br>UC [$C 2019] = 63,684.6 * kW^SF, where SF = 0.4603; IF = 1.3 |
| TIC<br>(2019 C$) | = $760,338.49* 1.3<br>$= \mathbf{988,440.04\ C\$}$ | TIC = UC * IF; where IF = 1.3 |
| TCI<br>(2019 C$) | = $988,440.04 + (0.28* $988,440.04)<br>= **1,265,204.31 C$** | TCI = TIC + Indirect Costs; where Indirect costs = 28% TIC |
| Annualized TCI<br>(2019 C$/yr) | $CRF = \frac{0.08(1+0.08)^{15}}{(1+0.08)^{15}-1} = 0.1168$<br>Annualized TCI = $1,265,204.31 $*$ 0.1168<br>$= \mathbf{147,813.24\ C\$/yr}$ | Annualized TCI $\left[\frac{\$}{year}\right]$ = TCI ($) $*$<br>Capital recovery factor (CRF)<br>$CRF = \frac{i\,(1+i)^n}{(1+i)^n-1}$ (i: Discount rate (%); n-Plant lifetime) |
| Electrical energy cost<br>(2019 C$/yr) | $= 218.63\ kW * 24\frac{hrs}{day} * 365\frac{days}{year} * 0.90 * \frac{0.11\$}{kWh}$<br>$= \mathbf{189,608.62\ C\$/yr}$ | Electrical energy cost ($/yr) = Power (kW) $*$ Operating hours (hours/yr) $*$ Electricity price ($/kWh) |
| Direct labor cost<br>(2019 C$/yr) | = (288*(2,000/100,000) ^0.25) *49.66<br>$= \mathbf{5,378.49\ C\$/yr}$ | Direct labor cost $\left(\frac{\$}{yr}\right)$ =<br>Annual hours(hours/yr) $*$ Labor cost ($/hour)<br>Annual hours(hours/yr) = 288 $*$ (x/100000)^0.25 |
| Indirect labor cost<br>(2019 C$/yr) | = $5,378.49 $*$ 50%<br>$= \mathbf{2,689.24\ C\$/yr}$ | Indirect labor cost ($/yr) =<br>Direct labor($/yr) $*$<br>Indirect labor factor (%)<br>Indirect Labor factor = 50% |
| **Fixed O&M<br>(2019 C$/yr)** | = (0.04 $*$ $988,440.04)<br>+ (0.021 $*$ $1,265,204.31)<br>$= \mathbf{62,717.90\ C\$/yr}$ | ✓ O&M & repairs = 4% of TIC<br>✓ Insurance = 1 % of TCI<br>✓ Property tax = 1 % of TCI<br>✓ License & permits = 0.1% of TCI |
| **Non − energy OPEX<br>(2019 C$/yr)** | = $5,378.49 + $2,689.24 + $62,717.9<br>$= \mathbf{70,785.72\ C\$/yr}$ | $O\&M\left(\frac{\$}{yr}\right) = \text{Total labor}\left(\frac{\$}{yr}\right) + \text{Additional O\&M}\left(\frac{\$}{yr}\right)$ |
| $Capex_{comp}$<br>**(2019 C$/kg H2)** | $= \frac{\$147{,}813.24\ /yr}{(0.90 * 2000 * 365)}$<br>$= \mathbf{0.225\ C\$/kg\ H2}$ | $Capex_{comp}\left[\frac{\$}{kgH_2}\right] = \frac{\left(\text{Annualized TCI}\left[\frac{\$}{year}\right]\right)}{\left(\text{Availability}[\%]\times\text{DesignCapacity}\left[\frac{kgH_2}{day}\right]\times 365\left[\frac{days}{year}\right]\right)}$ |

| Steps | Calculation | Notes |
|---|---|---|
| Non-energy $OPEX_{comp}$ (2019 C$/kg $H_2$) | $= \frac{\$70{,}785.72/yr}{(0.90 * 2000 * 365)}$ $= \mathbf{0.108\ C\$/kg\ H2}$ | $Opex_{comp}\left[\frac{\$}{kgH_2}\right] = \frac{\left(O\&M\left[\frac{\$}{year}\right]\right)}{\left(Availability[\%]\times DesignCapacity\left[\frac{kgH_2}{day}\right]\times 365\left[\frac{days}{year}\right]\right)}$ |
| $Energy_{comp}$ (2019 C$/kg $H_2$) | $= \frac{\$189{,}608.62/yr}{(0.90 * 2000 * 365)}$ $= \mathbf{0.289\ C\$/kg\ H2}$ | $Energy_{comp}\left[\frac{\$}{kgH_2}\right] = \frac{\left(Energy\left[\frac{\$}{year}\right]\right)}{\left(Availability[\%]\times DesignCapacity\left[\frac{kgH_2}{day}\right]\times 365\left[\frac{days}{year}\right]\right)}$ |
| $LCOH_{comp}$ (2019 C$/kg $H_2$) | $= 0.225 + 0.108 + 0.289$ $= \mathbf{0.621\ C\$/kg\ H2}$ | $LCOH_{comp}\left[\frac{\$}{kgH_2}\right] = Capex_{comp}\left[\frac{\$}{kgH_2}\right] +$ $Non-energy\ OPEX_{comp}\left[\frac{\$}{kgH_2}\right] +$ $Energy_{comp}\left[\frac{\$}{kgH_2}\right]$ |

### 5.2.1 Effect of design capacity

As discussed earlier, small-scale compressors for use at HFS need to compress $H_2$ to high discharge pressures of ~450-850 bars. While both reciprocating and diaphragm compressors can be used for this purpose, diaphragm compressors are more common due to higher reliability and purity of $H_2$ at discharge. The TCI (MM 2019 C$) of these small-scale compressors as function of design capacity is shown in **Figure 5.4(a).** In contrast to large scale centrifugal compressors, there is a significant advantage of compressor size on TCI with a scaling factor of ~0.46. A breakdown of $LCOH_{comp}$, CAPEX, electricity and non-energy operating costs is shown in **Figure 5.4(b)** which shows the dominant contribution from energy/electricity costs at capacities > 4000 kg $H_2$/day. This leads to a drastic drop in in $LCOH_{comp}$ up to capacity of ~4000 kg $H_2$/day due to the effect of scaling factor on capital and non-energy operating costs. At higher capacities, the scaling factor effect is minimized due to the dominant contribution of electricity/energy costs, and we observe a more gradual decrease in $LCOH_{comp}$.

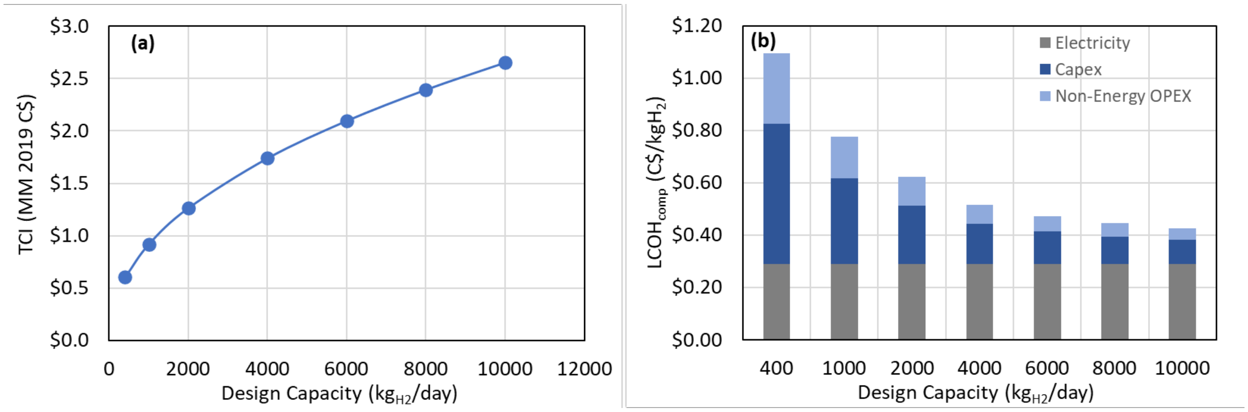


**Figure 5.4** Impact of design capacity on (a) TCI (MM 2019 C$) and (b) $LCOH_{comp}$ (2019 C$/kg $H_2$) for small scale diaphragm compressors.

### 5.2.2 Effect of suction/inlet pressure and isentropic efficiency

**Figure 5.5** highlights the impact of suction pressure and isentropic efficiency on the energy requirement and cost of compression. Like the trend seen with large scale reciprocating compressors, the suction pressure has a significant impact on the $LCOH_{comp}$ and electricity costs. But unlike what we have observed till now, when suction pressure is > 80 bar (Compression ratio <6.25), electricity costs for small scale compressors decrease to a point where capital costs contribution becomes dominant as seen in **Figure 5.5(c)**. Also, like large scale compressors, the isentropic efficiency of small-scale compressors has a smaller effect on the energy requirement and cost of compression versus suction pressure as shown in **Figure 5.5(d-f)** and contour plots of **Figure 5.6**.

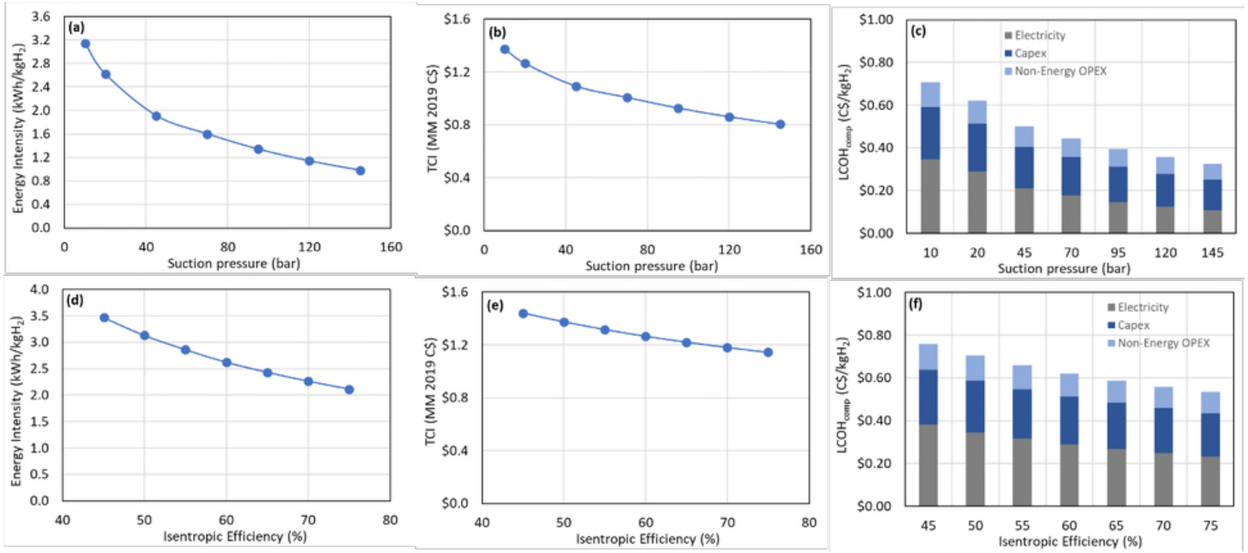


**Figure 5.5** Impact of suction pressure (a-c) and isentropic efficiency (d-f) on performance of small-scale diaphragm $H_2$ compressors.

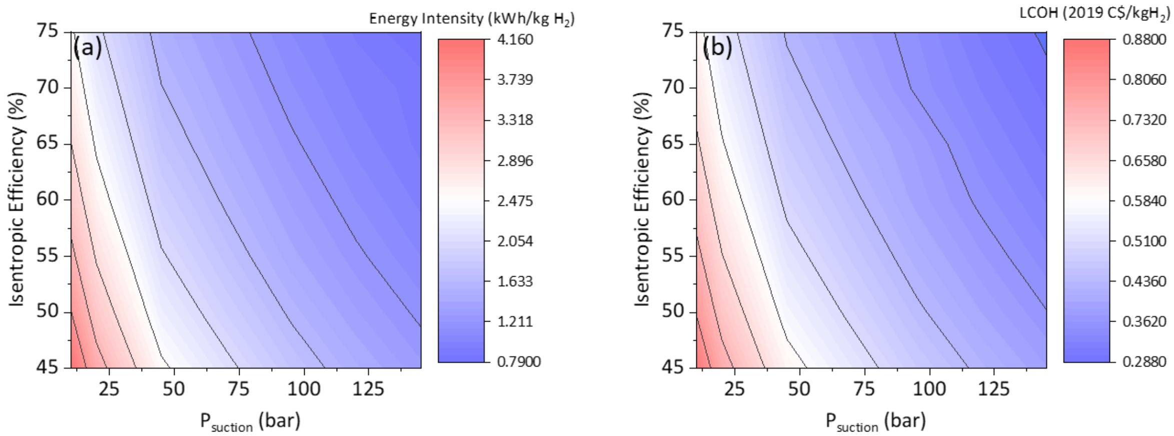


**Figure 5.6** Combined effect of suction pressure and isentropic efficiency on (a) energy intensity (kWh/kg $H_2$) and (b) $LCOH_{comp}$ (2019 C$/kg $H_2$) for small scale diaphragm $H_2$ compressors.

# 6 SUMMARY AND OUTLOOK

Low carbon $H_2$ is projected to play a key role as an energy carrier in future energy systems and become the fuel of choice in hard-to-decarbonize sectors such as heavy transport, heating, and steel production. At present, almost all the $H_2$ consumed in the world is close to the production site. The development of low cost and efficient technologies for storage and transportation of $H_2$ will determine its role in a net-zero future. To this end, compression is the key technology which enables delivery of $H_2$ from production site to end user.

Although compression of natural gas is widely used, the compression of $H_2$ is significantly challenging due to its low molecular weight and density. Currently available compressors which rely on mechanical pistons are expensive and reported efficiencies are low when compressing $H_2$ to high pressures (> 200 bar). Moreover, they suffer from frequent mechanical failure which increases operating expenses.

Further research and development activities are needed to design high efficiency compressors that can deliver $H_2$ at high pressures without compromising on the purity and reliability. The development of new technologies such as those based on ionic liquids or metal hydrides is promising. In particular, ionic liquid compressors which have been particularly developed by, Linde, could be the key to efficient and low cost $H_2$ compression. These compressors do not require bearings or seals, two of the common sources of failures in piston and diaphragm compressors.

Finally, as we move towards the implementation of net-zero energy systems, the capital costs associated with compressors is forecasted to drop sharply with economy of scale. These are exciting times and present both challenges and opportunities for different stake holders involved in the $H_2$ economy.

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