TURBOEXPANDERS IN PETROCHEMICAL INDUSTRY ADVANCE TECHNOLOGY FOR GREEN HYDROGEN LIQUEFACTION

Louis Mann
R&D Manager
Atlas Copco Gas and Process
Santa Maria, CA, USA

Behrooz Ershaghi
Marketing and Technology Manager
Atlas Copco Gas and Process
Santa Maria, CA, USA

Jacob Thomas
Vice President of Process Technology and Engineering
JTurbo Engineering & Technology, LLC
Houston, TX, USA

Trevor Mayne
Lead Machinery Engineer
Qenos Altona Olefins
Melbourne, Victoria, Australia

Louis Mann is a Research and Development Manager for Atlas Copco Gas and Process Division, California, USA. Mr. Mann joined Atlas Copco in 2008, holding roles in Package Engineering, Applications, Engineering Management, and R&D. Mr. Mann received a Bachelor of Science in Mechanical Engineering from California Polytechnic State University, San Luis Obispo, and he is a registered Professional Engineer in the State of California.

Behrooz Ershaghi is Marketing and Technology Manager at Atlas Copco Gas and Process, California, USA. Mr. Ershaghi has held several positions over his 40+-year career in Gas processing, Turboexpander and Rotating equipment industries. He received his Master of Science in Petroleum Engineering and PhD in Chemical Engineering from the University of Southern California.

Jacob Thomas is Vice President of Process Technology and Engineering at JTurbo Engineering & Technology, LLC, Texas, USA. Mr. Thomas has over 40 years of experience specializing in consulting and engineering services in the natural gas industry specifically to Hydrogen Liquefaction, LNG (Liquefaction & Re-gasification), NGL and Oil & Gas processing plants.

Trevor Mayne is the Lead Machinery Engineer for the Qenos Altona Olefins refinery in Victoria, Australia. He has worked with rotating equipment in the Olefins Refinery as well as in the Plastics and Synthetic rubber plants over the last 30 years. He has held positions in reliability as well as field maintenance both at the Altona plant and in Saudi Arabia with Exxon Mobil.

ABSTRACT

Global demands for petrochemicals and green hydrogen usage are on the rise and are expected to continue for at least the next decade. Green hydrogen liquefiers are needed for the worldwide reduction of greenhouse gases and global commitments to net zero by 2050. Hydrogen is a long-term energy carrier and storage solution, and with renewable energy costs shrinking overall, green hydrogen production via electrolysis is increasingly becoming a viable solution. As a result, the demand for turbomachines to carry out cryogenic hydrogen services is growing.

Radial inflow turbines, or turboexpanders, have been used in hydrogen-rich applications in the petrochemical industry since the early 1960s. Hydrogen, or hydrogen-rich gas applications, require special turbomachines. Oil-free solutions are preferred, with high peripheral speed designs and low operating temperatures (22 K is required for liquefaction).

In this technical brief, the authors discuss turboexpanders designed for hydrogen service, featuring high enthalpy drop (head), high impeller tip speed, rotor-bearing system, material selection to avoid hydrogen embrittlement and material suitability for deep cryogenic temperatures. Turboexpander loads, varying from small friction to booster compressor or generator loads, are covered. In particular, the authors present the challenge of matching high flow coefficient booster impellers with low flow coefficient turboexpander wheels in hydrogen applications.

INTRODUCTION

Hydrogen has the potential to be one of the largest global commodities in the decades to come. Historically, hydrogen has been used in refining, chemical processing, steel industries, and ammonia production. More recently, interest in hydrogen mobility and power applications has risen due to hydrogen's potential as a carbon-free energy carrier. The growing momentum and commitments for clean fuel and energy has in turn led to enormous opportunities for the turbomachinery industry within the hydrogen supply chain.

Industry has decades of experience generating and handling hydrogen. Hydrogen is typically produced by steam methane reforming (SMR), where syngas (hydrogen and carbon monoxide) is generated by reaction of hydrocarbons with water. Hydrogen produced by SMR is considered "grey" when carbon dioxide is released to the atmosphere in the process. "Blue" hydrogen is produced in a similar fashion, only much of the waste carbon dioxide is captured to reduce its environmental impact. The future of hydrogen however is "green", where hydrogen produced by electrolysis uses zero-carbon electricity sourced from renewables.

With renewable energy costs shrinking overall, green hydrogen production is becoming a more viable solution and scalable methods of storage and transport are being addressed. Hydrogen liquefaction is one of many proven transportation methods under consideration. When converting from gas to liquid, hydrogen undergoes a 800:1 reduction in volume. Hydrogen liquefaction is a proven technology with room for improvement as it scales (Ohlig 2014). Hydrogen liquefiers were first built in the early 1960s, largely to support rocket-fuel production for the historic USA Apollo program (Krasae-in 2010). Today, global liquid hydrogen (LH2) production is near 400 tonnes per day (TPD), with over half of that production in the US. Current liquid hydrogen plants have capacities ranging from 0.3 to 40 TPD, with future plans scaling 10x those capacities.

To produce liquid at standard pressure, hydrogen must be cooled to near 20 K. Radial inflow turbines, or turboexpanders, are commonly used to produce this required cooling through near isentropic expansion of a low boiling point refrigerant (hydrogen, helium, neon, or mixtures thereof). The turboexpander performance is critical to providing enough sub-cooling to the process. Changes in the specific heat near the critical point and energy release during the ortho-para conversion in heat exchangers require careful selection of system operating temperatures. Minor performance improvements in the turboexpander can greatly impact the specific power of the plant, and thus the economic viability of the operation.

Turboexpander design for hydrogen is challenging due to high isentropic enthalpy drop across the stage, low discharge volume, and required high operating speeds. Hydrogen turboexpanders have unique aerodynamic and mechanical designs, including low flow coefficient turboexpander wheels, high peripheral speeds, heat soak and thermal management, special materials, and non-contaminating sealing systems. Practical challenges arise from hydrogen's flammability, resistance to being contained by joint seals, and its tendency to cause embrittlement in common materials. Moreover, when a turboexpander load is balanced by a high flow coefficient booster compressor for energy recovery, unique rotordynamic challenges arise.

Turboexpander technology in hydrogen-rich applications has developed greatly over the last 30 years. Turboexpanders have been increasingly used in petrochemical applications (ethylene and propane dehydrogenation plants) for cryogenic off-gas recovery. The process gas for these petrochemical applications contains up to 96 percent hydrogen. As demand for polymers increased over the years, so did turboexpander technology to support efficient operations (Agahi 2016). Developments in the petrochemical industry offer insight on how the latest turboexpander technologies can be adapted and qualified for the growing demands of green hydrogen liquefaction.

HYDROGEN EXPANSION

Hydrogen's Physical and Thermodynamic Properties

Hydrogen is the lightest element in the universe and exists as a gas (H2) at standard conditions. With a molecular weight of 2.02, hydrogen has a low density, high specific heat and high speed of sound. Table 1 compares n-hydrogen and methane properties at standard conditions.

Table 1. Hydrogen and Methane Properties at Standard Conditions (NIST REFPROP)

	Hydrogen	Methane
Molecular Weight [-]	2.02	16.0
Density [kg/m3]	0.085	0.679
Specific Heat [kJ/kg-K]	14.26	2.18
Cp/Cv [-]	1.41	1.31
Speed of Sound [m/s]	1294	441
Kin. Viscosity [CM2/S]	1.020	0.158
Joule-Thompson Coef [K/Atm]	-0.283	.4756
Max Inversion Temperature [K]	205	939
Boiling Point [K]	20.2	112

Due to its low mole weight, hydrogen has high enthalpy difference (head) for a given pressure ratio. For an ideal gas, it can be shown algebraically that enthalpy as a function of pressure ratio strongly correlates with specific heat, whereas higher specific heat results in higher enthalpy. Hydrogen's specific heat is greater than six times that of methane at standard conditions (Brun 2021).

While head is comparatively high for hydrogen, density is exceptionally low. Low density results in high volumetric flow rates and generally low energy per unit volume for turbomachinery. Conversely, low density of hydrogen also results in a high speed of sound, making Mach number limitations less of a concern.

Finally, hydrogen's inversion temperature of roughly 200 K or less (depending on the pressure) makes refrigeration via Joule-Thompson (J-T) effect unfavorable if not impossible above deep cryogenic temperatures. Refrigeration above or near the inversion temperature requires either external refrigeration or isentropic expansion, where additional energy in the form of work is removed from the gas. Turboexpanders have for decades excelled at performing this near-isentropic expansion.

Turboexpanders

In a turboexpander (see Figure 1), high-pressure fluid passes through variable inlet guide vanes (vIGV), where potential energy is converted to high tangential velocities before entering an expander wheel in the radial direction. An ideal 90-degree turboexpander is a 50-percent reaction turbine, whereby, approximately half of the enthalpy drop across the expander stage is used by the vIGVs to accelerate the fluid. The remaining expansion occurs in the turboexpander wheel, where the gas turns and exits at a lower pressure. If required by the process conditions, a turboexpander's wheel geometric design can accommodate a lower degree of reaction (<50 percent) while maintaining high efficiencies.

Figure 1. Radial Inflow Turbine, Turboexpander

Expansion through a turboexpander is a near-isentropic process in which energy extracted from a working fluid is converted to mechanical work. This mechanical work is absorbed by a variety of devices which are classified into two major categories: energy dissipating and energy recovery. Energy dissipating turboexpanders reject the produced work, typically in the form of heat. Energy recovery expanders in contrast convert the work to useful and free energy, often via a directly coupled booster compressor or generator. While turboexpander utility typically focuses on the energy extraction from the fluid (refrigeration), the free energy recovered by a loading mechanism can directly improve the specific power of a given process.

Turboexpander Optimization

Two dimensionless parameters are used when sizing turboexpanders: specific speed, Ns, and velocity ratio U_2/C_0. Specific speed is a combined factor of discharge flow coefficient (relation between fluid velocity and blade tip velocity) and head coefficient (relation of fluid energy or enthalpy drop to the wheel dynamic energy). It is a similarity parameter independent of the turboexpander size which defines the shape of the wheel. Velocity ratio is blade tip velocity to spouting velocity of total isentropic enthalpy drop, U_2/C_0. These two parameters are defined as follows:

$$N_S = \frac{\phi^{1/2}}{\psi^{3/4}}$$

$$N_S = \frac{N\sqrt{Q}}{\Delta H^{3/4}} \tag{1}$$

$$U_2/C_0 \propto \frac{D \times N}{\sqrt{\Delta H}} \tag{2}$$

Where:

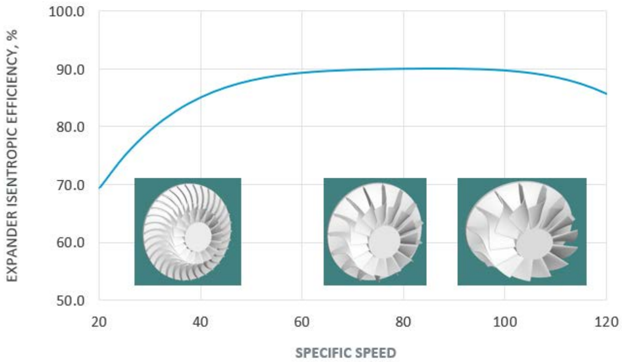
$N_S =$	Specific Speed
$U_2/C_0 =$	Velocity Ratio
$\phi =$	Discharge Flow Coefficient
$\Psi =$	Head Coefficient
$N =$	Shaft Rotation, RPM
$Q =$	Discharge Volumetric Flow, $\frac{ft^3}{s}$
$D =$	Wheel Outside Diameter, ft
$\Delta H =$	Stage Isentropic Enthalpy Drop, $\frac{ft \cdot lb}{lb_m}$

When sizing turboexpanders, volumetric flow and isentropic enthalpy are fixed by the process design, so optimization focuses on rotating speed and diameter. Turboexpander optimization considers the efficiency correlations in Figures 2 and 3, while minding physical limitations of the machinery. Optimum velocity ratio is calculated from the following equation based on maximum energy transfer relations where R is the degree of reaction or ratio of the enthalpy drop in the wheel to total stage available enthalpy drop. For 50 percent reaction turbine, the optimum U_2/C_0 is near 0.7.

$$U_2/C_{0_{opt}} = \frac{1}{2\sqrt{1-\mathrm{R}}} \tag{3}$$

Where;

$U_2/C_{0_{opt}} =$	Optimum Velocity Ratio
$R =$	Degree of Reaction



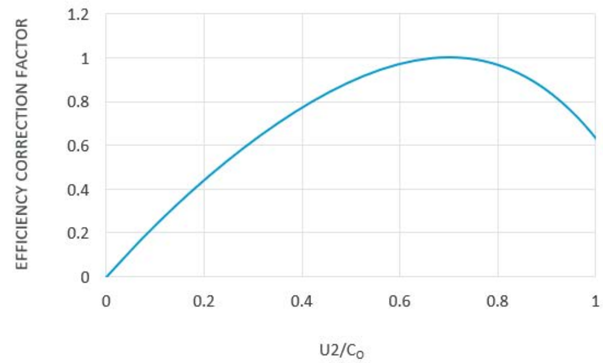


Figure 2. Optimal Specific Speed

Figure 3. Optimal U_2/C_0

When considering high-enthalpy applications, optimal velocity ratio is a first indication of stage quantities. Mechanical limitations associated with peripheral speeds limit the head that can be handled in a single stage. Hydrogen's high enthalpy drop often results in several expander stages. The only way to reduce the number of stages is to increase tip speed or consider reduced performance. Figure 4 shows an example of performance degradation of increased enthalpy for a fixed tip speed turboexpander in a hydrogen-rich application.

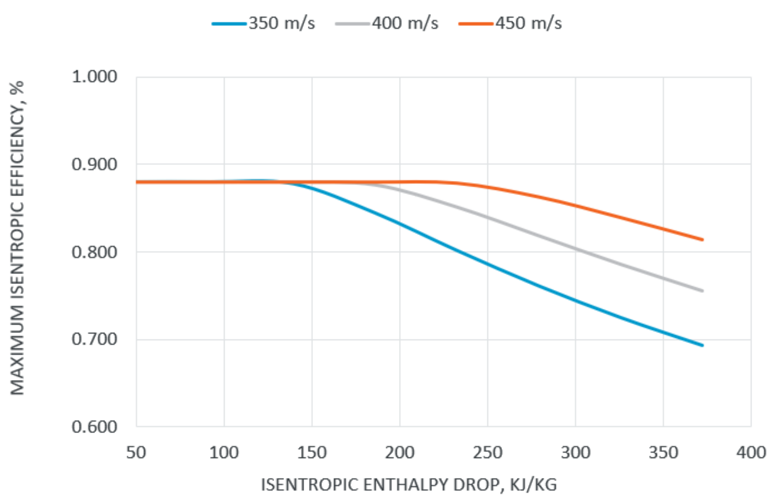


Figure 4. Tip Speed vs Enthalpy Drop (Zhao, 2020)

With enthalpy split near evenly amongst stages, shaft speed is then selected to optimize N_s, with a respective wheel diameter chosen to maximize tip speeds. When considering rotation speed alone, bearing limitations and rotordynamics are considered. Power density and shaft torque tend to be less of a limiting factor in hydrogen applications due to the low energy density per unit volume. While additional stages may add cost or complexity to the system, a multi-stage turboexpander's overall performance and produced refrigeration may be higher than individual stages due to the re-heat effect adding available enthalpy drop to the individual stage (Figure 5). Equation 4 describes multi-stage efficiency with respect to individual stage efficiencies.

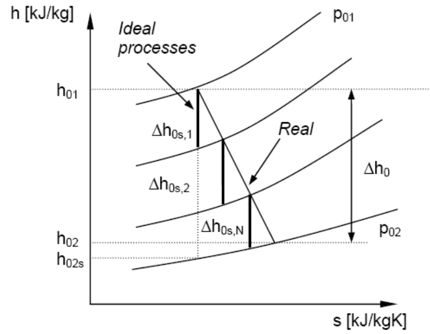


Figure 5: Multi-stage Turboexpander Performance

$$\eta_p = \frac{\Delta h_0}{\sum \Delta h_{0_{s,i}}} \tag{4}$$

Where:

$\eta_p =$	Multi-Stage Efficiency
$\Delta h_0 =$	Multi-Stage Actual Enthalpy Drop
$\Delta h_{0_{s,i}} =$	Individual Stage Isentropic Enthalpy Drop

TURBOEXPANDERS IN HYDROGEN-RICH PETROCHEMICAL APPLICATIONS

Turboexpanders have been used in the petrochemical industry since the early 1960s. In ethylene, propylene, or isobutylene production, turboexpanders are found in the cryogenic section of the plant where they expand hydrogen-rich overhead vapor (off gas) to produce deep refrigeration for product recovery. These petrochemical processes require cryogenic temperatures as low as 100 K to separate gas mixtures containing up to 96 percent hydrogen. Amongst petrochemical processing, ethylene is the world's most produced organic compound, and over the last 60 years, turboexpander technology has evolved to support improved ethylene production.

Ethylene Production

The manufacture of ethylene is primarily for the production of the plastic polymer Polyethylene. The worldwide growth in polyethylene demand required ethylene producers to not only grow their plant sizes, but to also produce the highest yield from their feedstocks. Ethylene plants typically require ethane, LPG, naphtha or a combination of them as the starting feed stock. An ethylene plant can be broken down into three basic operational units:

1- Thermal cracking and steam quenching. A low-pressure (<1 bar) high-temperature (800-900°C) process that converts the feed stock into ethylene molecules and byproducts (typically propylene, methane, and hydrogen).
2- Compression & treatment. A high-pressure process (30-40 bar) during which the products of cracking are washed and cleaned in preparation for ethylene recovery.
3- Distillation and recovery. A low-temperature cryogenic process ("chill train") down to -135-145°C, which results in the recovery of ethylene and light hydrocarbon tail gas (methane/hydrogen) that is used as fuel or sold for hydrogen recovery.

It is in the distillation/recovery "chill train" (shown in Figure 6) that the introduction of turboexpanders has played a vital role in maximizing ethylene plant yields.

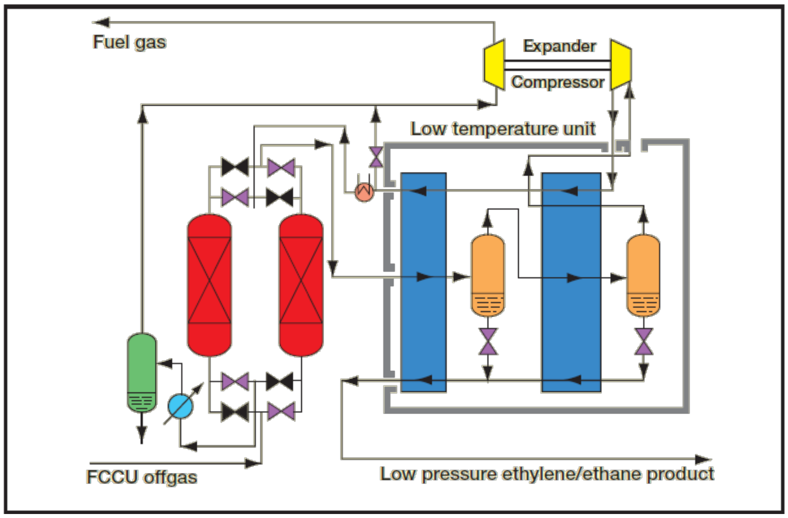


Figure 6. Ethylene Treatment, Distillation and Recovery (Tomlinson 2002)

The chill train of an ethylene plant takes advantage of byproduct propylene and the ethylene product itself as a suitable refrigerant to produce the low-temperature cryogenic process conditions required for the ethylene product recovery. At a pressure approaching atmospheric, propylene and ethylene have boiling temperatures of -50°C and -100°C respectively. At these temperatures, the original ethylene plant chill trains were configured to recover the bulk share of the ethylene production, with a considerable amount of ethylene carrying through with the tail gas. To improve on this, the final tail gas streams leaving the chill train units were further cooled using simple J-T expansion valves, letting down this tail gas stream from 30-35 bar to 5-7 bar. If the percentage of methane was high enough to offset the negative J-T effect of hydrogen, the J-T effect could push the final chill train rejection drum temperatures down towards -105-110°C. In some ethylene plants, the final chill train was also configured with an additional mechanical refrigeration system using methane as the refrigerant to be able to reach the desired cryogenic temperature levels to maximize the ethylene yield.

The introduction of turboexpanders into the ethylene process over J-T letdown or the more costly methane refrigeration process was an elegant and practical inevitability. A turboexpander is a free energy approach and as any operations manager knows, “If it’s for free, it’s for me.” The final pressure let down from the chill train unit is provided and paid for by necessary upstream compression. A turboexpander loaded by a compressor relieves some of this upstream compression duty by recovering free work in the near isentropic expansion process. Temperature reductions in the order of 40°C across the turboexpander are common for these process let-down conditions, adding to the chill train’s final cryogenic refrigeration loop and creating ethylene separation at temperatures of -135-145°C.

The big challenge for turboexpanders in ethylene plants comes from the chill train’s service factor and its absolute intolerance to any contamination that can freeze and foul out the process equipment. With the push to increase ethylene plant production, continuous operation for eight to ten years is required. In considering a turboexpander for high-purity hydrogen duty in an ethylene chill train the machine vendors must provide:

- Highest possible service factor
- Zero process contamination
- Maximum refrigeration duty with efficient means to consume produced shaft work (energy recovery)
- Robust process control (pressure) by inlet guide vanes
- Turndown capability to manage plant throughput and feedstock variations
- Minimal shaft seal leakage and compatibility with available sealing gas (hermetic design preferred)
- Ability to handle thermal cycling of startups and plant upsets
- Ability to handle liquid phase in the machine’s outlet
- Suitable for Zone 1 or 2 hydrogen gas group environment
- Safe and reliable operation of high-speed machinery

Turboexpander Configurations For Ethylene

Turboexpanders have evolved over the years to meet the increased demand of ethylene production. Energy dissipating turboexpanders were common in early low-power ethylene installations. Hydraulic brakes allowed for the high peripheral speeds required by hydrogen-rich applications. The turboexpander load was simply absorbed by a low-efficiency oil pump. The absorbed energy was not recovered; rather, it was rejected as heat by exchangers in the oil system.

As the size of ethylene plants has grown, energy recovery turboexpanders configured with compressors have become preferred. Turboexpanders paired with booster compressors offer a hermetically sealed system with no external shaft seals. Early designs utilized oil lubricated bearings, where the bearings and auxiliary support system (oil reservoir, pump, and so on) operate in a pressurized environment all within the process fluid. Sealing gas (required to separate the warm lubrication oil from the cryogenic process gas) could be returned to the process or routed for disposal when trace amounts of oil are not tolerated. Figure 7 shows a three-stage expander compressor with oil bearing package. Due to sealing loss and the potential for oil carryover to the process, expander compressors equipped with active magnetic bearings (AMB) ultimately became the preference for hydrogen-rich processes when introduced in the early 1990s.

AMB systems are hermetically sealed with oil-free sealing gas fully contained in the process. Sealing gas sourced from the hydrogen-rich process cools and protects the AMB components from cold, unfiltered process gas. With no permanent magnets, AMB materials are fully compatible with hydrogen service (Figure 8 shows an expander compressor with an AMB package).

Expander-compressors in hydrogen-rich applications have a unique challenge as the process tends to demand high head, low flow expanders paired with high head, high flow compressors. This imbalance in turbomachinery geometry creates challenges with both the tip speed and rotordynamics. Turboexpander designs over the last several decades have pushed these limits, increasing customer acceptance of higher tip speed machines, and optimizing compressor geometry to maximize energy recovery.

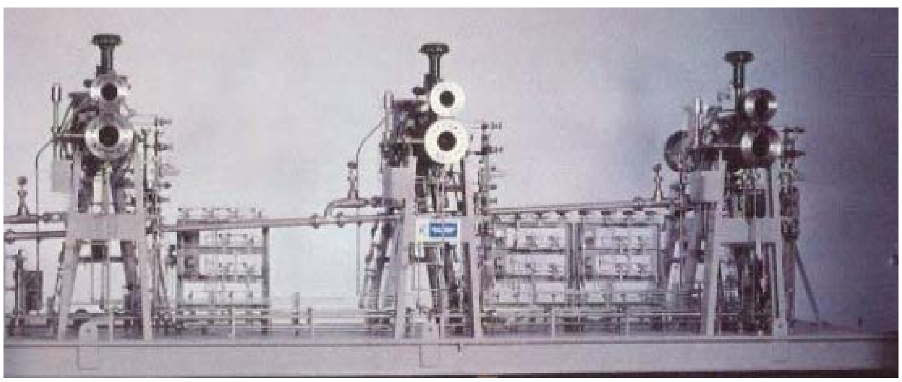


Figure 7. Multi-Stage Expander Compressor with Oil Bearing

Figure 8. Expander-Compressor with AMB

Another energy recovery option used in hydrogen-rich service is the integrally geared expander-generator. These configurations offer low seal leakage with dry gas seals (DGS), and energy recovery via generated electricity. Typical expander generators are not hermetically sealed because they rely on an oil-fed gear box to reduce shaft speeds. Expander-generator packages (Figure 9) are commonly deployed in hydrogen-rich service where recompression is not required by the process. Propane dehydrogenation (PDH), a process specific to propylene recovery, typically utilizes expander-generators in a two-stage configuration. Due to multiple conversions (speed reduction then mechanical to electric power), the energy recovered by expander-generators may be less than that for expander-compressors. A study by Zhao and Lillard (2020) offers a detailed comparison of hermetically sealed expander-compressor units to the expander-generator configuration in PDH applications.

Furthermore, innovative high-speed generators have been deployed for lighter duty applications, where gearbox cost or performance may be prohibitive for the application. In a similar arrangement to expander-compressors with AMB, permanent-magnet high-speed generators have now been arranged with two turboexpander stages rotating on a common shaft (Figure 10) (Vitt 2020). Using a permanent-magnet motor/generator connected to an inverter allows ease of variable speed startup “motoring” and transition to generation once the turboexpander takes on process gas. The inverter allows for turboexpander speed tuning in off-design conditions to optimize efficiency. To allow hermetic packaging in a hydrogen environment without hydrogen embrittlement of the permanent magnets, the central generator section is purged with a compatible externally sourced sealing gas, such as 100 percent methane.

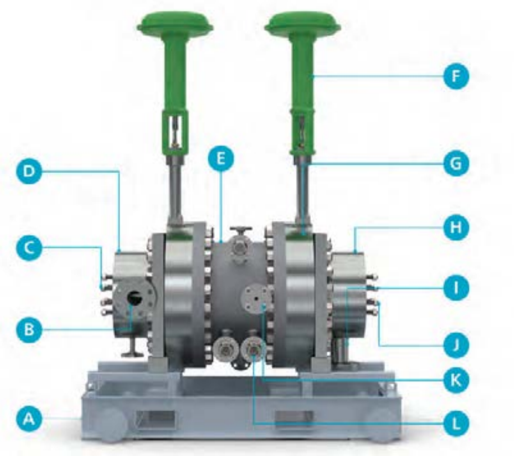


Figure 9. Expander Generator Package

Figure 10. Expander Generator (Permanent Magnet) with AMB

The introduction and evolution of turboexpanders in ethylene plants have embraced issues and faced off challenges through technology advances. The lessons learned from this evolution places the current machine configurations and vendors at an ideal point to take the next step in deeper cryogenic applications that the hydrogen industry will require.

TURBOEXPANDERS IN HYDROGEN LIQUEFACTION

Liquefaction Process

The hydrogen liquefaction process involves compression of hydrogen feed gas to the liquefier, pre-cooling normally down to 90 K, then further primary cooling down to 29 K or lower. The cooled hydrogen then expands, further reducing the temperature to about 21 K. Primary refrigeration for hydrogen liquefaction is done either in Claude cycle or closed-loop Brayton cycle. Today's liquefiers range from 10, 15, 30 to 40 TPD for local production, with plans for industrial scale up to 500 TPD in the future.

Pre-cooling is normally achieved with liquid nitrogen or cold-end, warm-end nitrogen turboexpanders. Primary cooling cycles are by means of pure hydrogen or mixed refrigerants (helium, neon, and hydrogen) based on the end user preference and the specific power required. Current Claude and Brayton hydrogen liquefaction technologies are well covered in previous work, such as in an Ohlig and Decker paper (2014), in which the authors offered potential improvements to reduce energy consumption from the usual levels of above 10 kWh per kgH2 to values closer to the benchmark of 6 kWh per kgH2. New and innovative cycles utilizing expander-compressors set out to meet or exceed this same benchmark.

Twin Expander Nitrogen – Hydrogen Liquefaction Process

JTurbo Engineering & Technology has developed a leading-edge, patent-pending twin-expander nitrogen and hydrogen cycle liquefaction technology which provides 5.2 kWh per kgH2, the lowest possible specific energy consumption when compared to the current state-of-the-art technology. The respective process scheme describing this liquefaction technology is depicted in Figure 11.

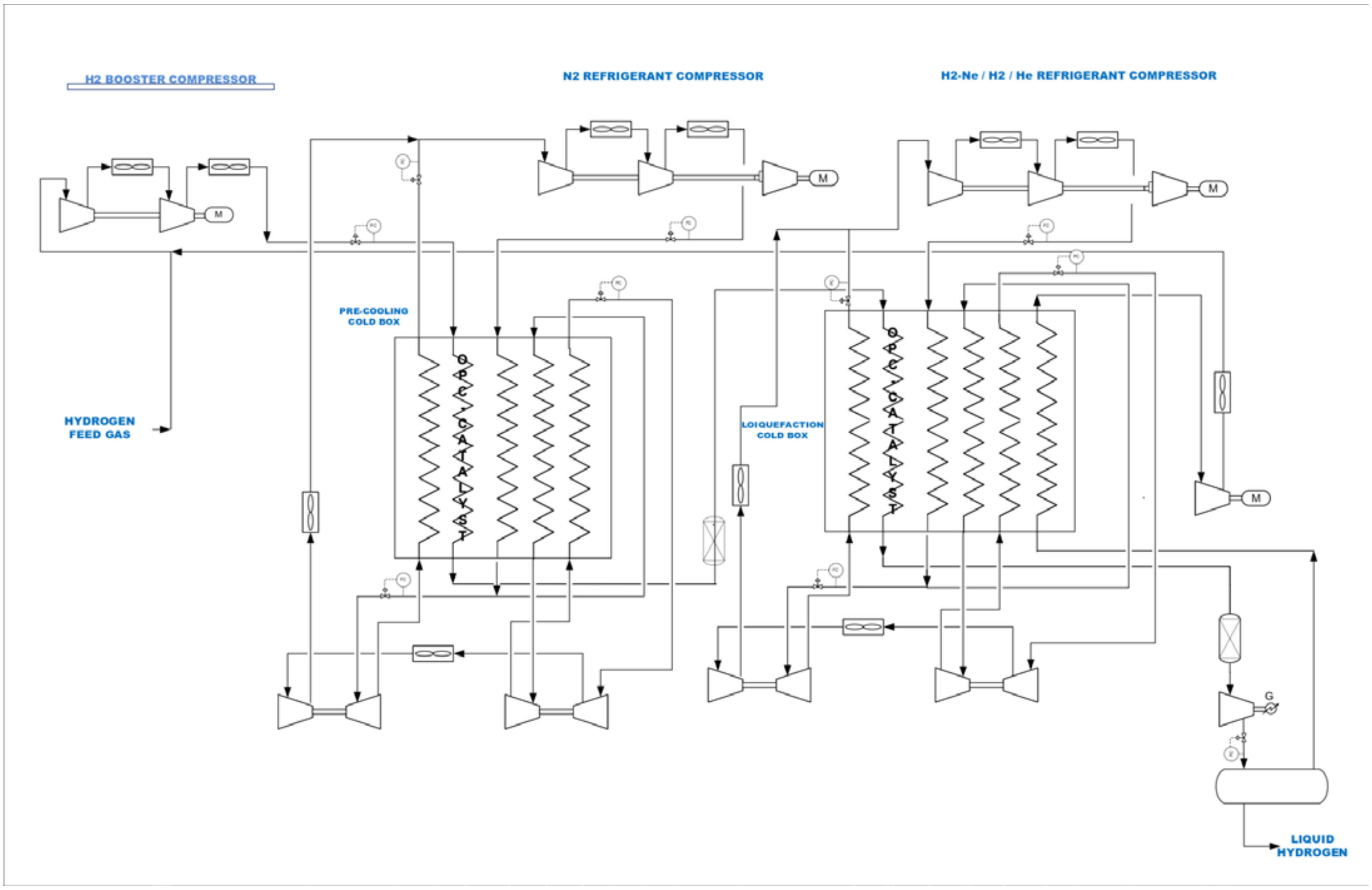


Figure 11. Twin Expander Nitrogen-Hydrogen Liquefaction Process (Closed Loop)

Gaseous nitrogen flows to a brazed aluminum heat exchanger (BAHE) cold box packed with ortho-para catalyst (OPC), for the pre-cooling hydrogen gas stream. After the first pass in the cold box, the nitrogen refrigerant stream is split, and the first portion is directed to a warm expander, which in turn drives a compressor. The second portion is routed back through and out of the cold box to a cold expander that drives a compressor.

The cold hydrogen gas stream from the precooling BAHE is sent to the hydrogen purification adsorber to remove any contaminants prior to sending it to the liquefaction and sub-cooling BAHE (vacuum-jacketed) cold box packed with OPC. From a compressor discharge stream, the hydrogen refrigerant flows to the cold box for further cooling. After the cold box, the hydrogen refrigerant stream is split, and the first portion is directed to a warm expander, which in turn drives a compressor. The second portion is routed back through and out of the cold box to a cold expander that drives a compressor.

The liquefied and sub-cooled hydrogen liquid stream from liquefaction is then routed through the adiabatic ortho-para catalytic converter bed and then directed to a liquid expander or a JT valve. This produces an expanded liquid hydrogen stream of 1-2 bar, which is sub-cooled and stored in a pressurized storage tank for liquid hydrogen export. Hydrogen flash gas stream from the liquid hydrogen storage tank is routed to the cold box and the refrigeration content of the hydrogen vapor stream is used to supplement the cold duty (compressed by a flash gas compressor), prior to recycling it back to the front-end hydrogen booster compressor.

Figures 12 and 13 show a cold and hot composite heat curve for the twin-expander nitrogen pre-cooling and hydrogen liquefaction cycle, which is closely matched to reduce overall energy consumption.

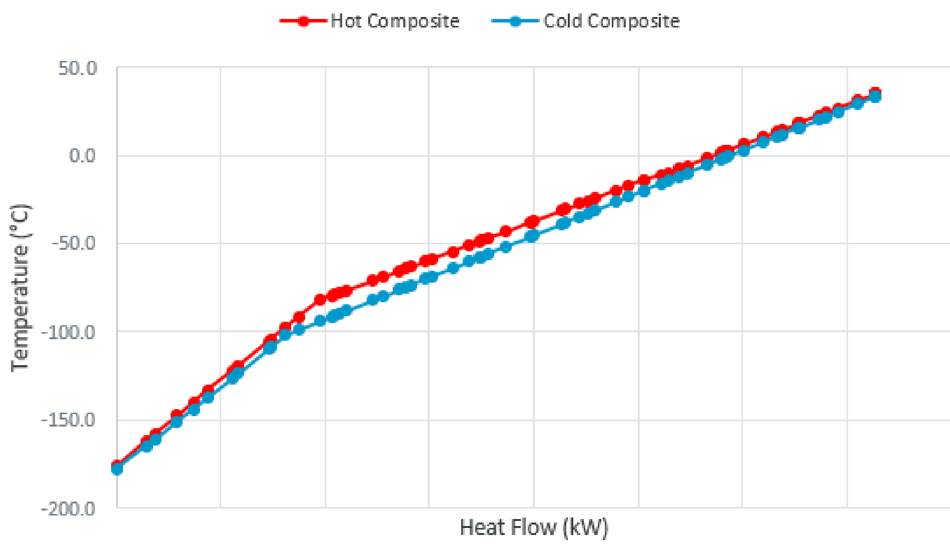


Figure 12. Twin-Expander (Nitrogen) Precooling cycle

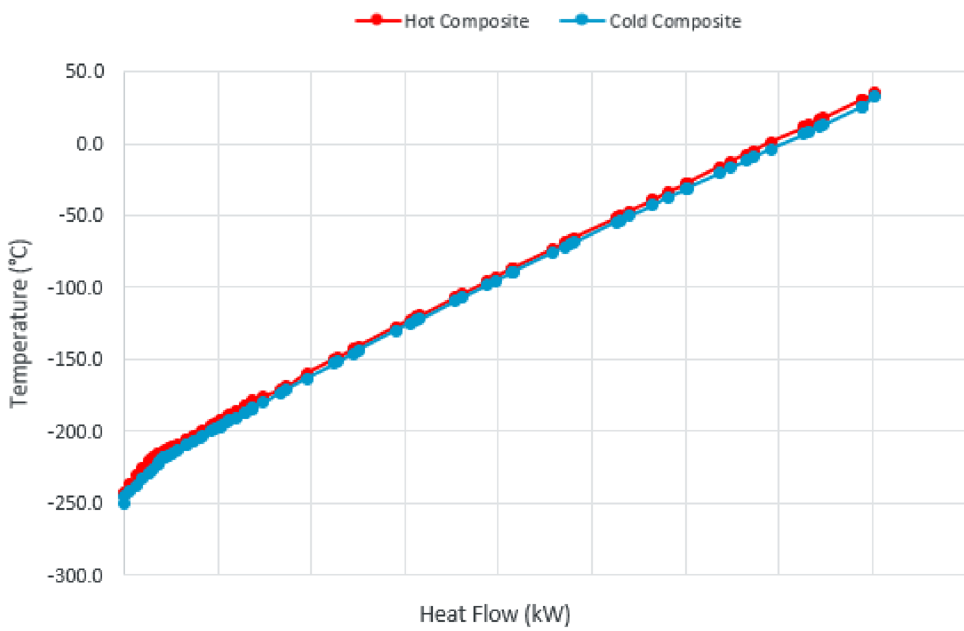


Figure 13. Twin-Expander (Hydrogen) Liquefaction cycle

Simulation Fluid Properties

The hydrogen liquefier process simulation model is mostly implemented with the fluid property estimation methods available in the Reference Fluid Thermodynamic and Transport Properties Database (REFPROP) of the National Institute of Standards and Technology (NIST), in the USA. The turboexpander and hydrogen streams that are not subject to the catalytic ortho- to para-hydrogen conversion are simulated with the fluid properties of normal-hydrogen (Leachman 2009). Normal hydrogen is a mixture of 3:1 orthohydrogen and parahydrogen. When cold or liquid hydrogen is stored for several hours, the quantum spin state of the hydrogen will convert to more parahydrogen or insufficient thermal energy. See figure 14 for parahydrogen conversion versus storage temperature.

Hydrogen liquefiers use special catalysts to reach equilibrium composition during the liquefaction process. The thermophysical properties change due to the quantum state and this will result in some evaporation of liquid hydrogen in storage tanks, which can be avoided with additional sub-cooling (Leachman 2009). The hydrogen feed stream that is cooled in the heat exchangers with continuous catalytic ortho- to para-hydrogen conversion is simulated by assuming equilibrium-hydrogen. At the final stage, the hydrogen feed stream expansion as well as the final cooling and liquefaction below 28 K are simulated with the fluid properties and the equation of state (EOS) of parahydrogen.

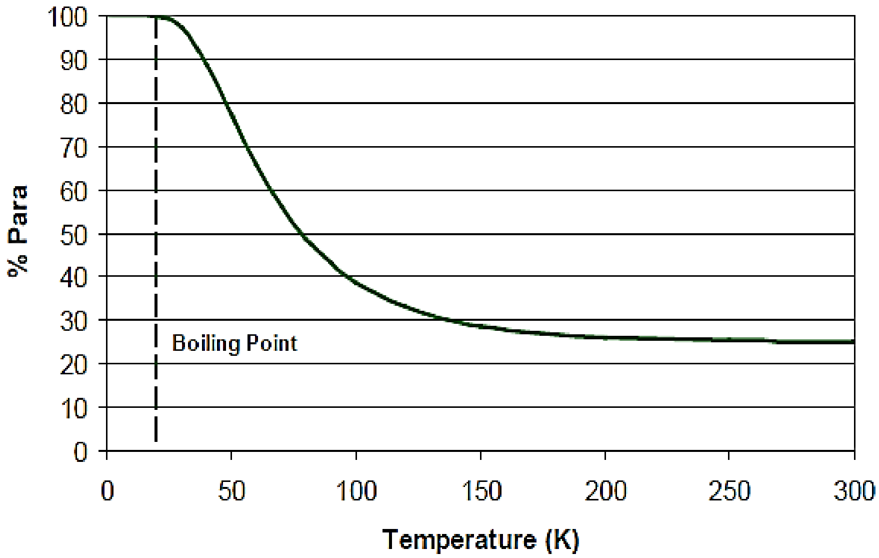


Figure 14. Para-Hydrogen Conversion Versus Storage Temperature

TURBOEXPANDER DESIGN FOR HYDROGEN LIQUEFACTION

The previously described twin-expander hydrogen cycle poses several design challenges for turbomachines. The following design considerations require evaluation when converting a traditional hydrogen-rich turboexpander to pure hydrogen liquefaction service.

Material Selection

The main factors determining material selections are suitability for hydrogen embrittlement, strength, and heat transfer. Hydrogen will diffuse into materials, and without re-diffusion out it will build pressure and cause embrittlement and cracking depending on the microstructure of the material. High-strength materials and surface hardness, or having specific distributions of grain boundary particles can result in increased susceptibility to embrittlement. This happens when sufficient stress is applied to a hydrogen-embrittled material. The stress can be caused both by the presence of residual stresses, associated fabrication operations, such as forming and welding, and applied service stresses. The severity of hydrogen embrittlement is also a function of temperature.

Some examples of suitable materials for hydrogen turbomachines are austenitic stainless steels and aluminum, depending on the effect of cryogenic temperature on their properties. Copper and other materials used in active magnetic bearings operate at relatively low temperatures and are proven to be immune from hydrogen attack. With limitations in material selection, minimizing stress from high-speed rotating components becomes an important design factor.

High Tip Speed

Most liquefier turboexpanders require high speeds to achieve higher performance due to low volume and high head. With constant mass flows in the refrigeration cycle, the high enthalpy drop on the turboexpander translates to high enthalpy rise and tip speed on the booster compressor. Aluminum alloys such as 7075-T6 have outstanding strength-to-weight ratios and enable high tip speeds for both turboexpander and booster compressors. Depending on the operating temperature, shaft fixation and optimized hub profile, tip speeds of as high as 470 m/sec are proven for high-strength aluminum impellers.

Design consideration for high tip speed include stress and deflection in the impeller disk/blade and wheel fixation to the shaft. Detailed finite element analysis (FEA) is performed to ensure stress and deflection are within acceptable limits. An example of a high tip speed compressor impeller analysis is shown in Figure 15. With proper design and feature selection, high tip speeds required for optimum aerodynamics can be achieved.

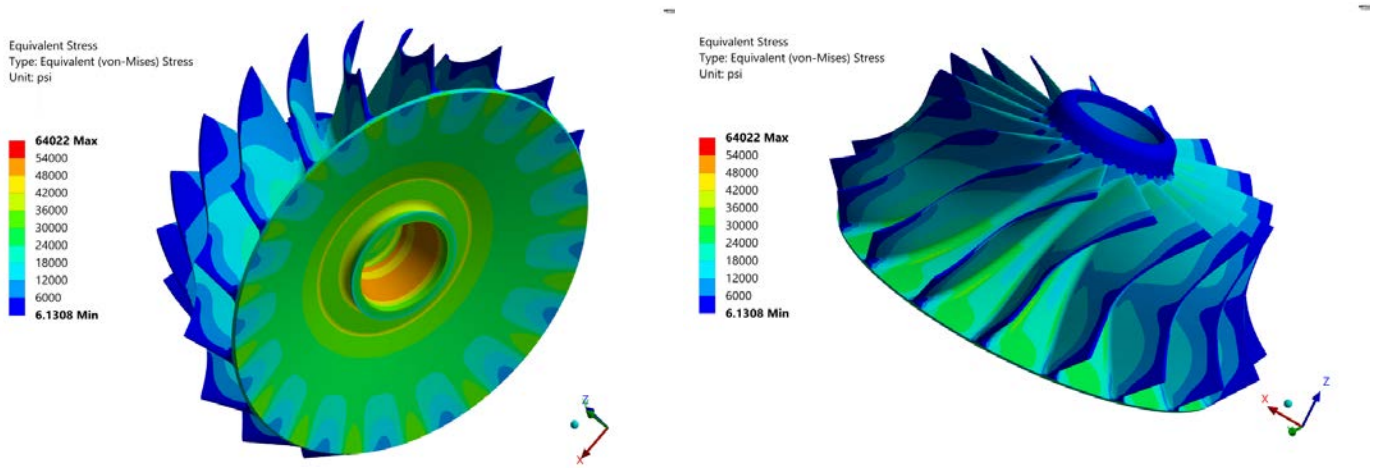


Figure 15. Stress Analysis on 470 m/sec Tip Speed Impeller

Aerodynamics

For radial inflow turboexpanders, optimum specific speed (N_s) is in the range of 70-90 and optimum velocity ratio (U_2/C_0) is in the range of 0.67-0.7. High enthalpy drop, low volume and limitation on the operating speed on hydrogen expanders often shift the design towards the low N_s (Figure 16) and U_2/C_0 (Figure 17) range. U_2/C_0 values of 0.6-0.65 are common for hydrogen, resulting in fluid spouting velocity over 1.5 times that of blade velocity. Higher rotational speeds may improve turboexpander efficiencies, but they are limited by bearing selection. Backswept inlet blade angles and full or partial shroud impellers improve the performance of low N_s, low degree of reaction turboexpander designs. The aerodynamic design must optimize blade geometries for minimizing the aerodynamic losses while maintaining lower stress levels and avoiding hub and blade resonances.

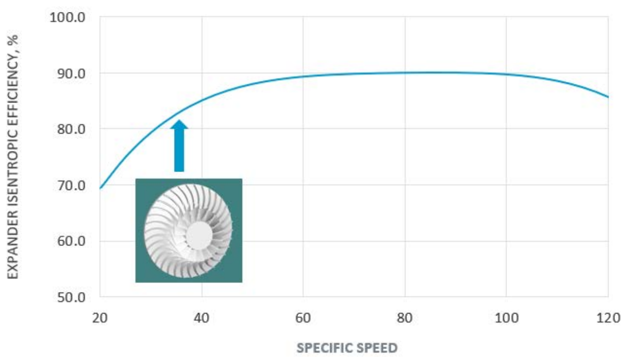


Figure 16. Low Specific Speed Selection

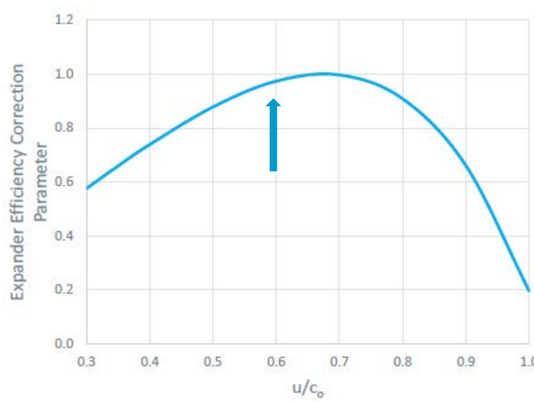


Figure 17. Low U_2/C_0 Selection

The high temperature difference across the heat exchangers in a hydrogen refrigeration cycle result in extreme differences in density between expander and compressor streams. This results in a low flow coefficient expander, paired with high flow coefficient compressor operating at the same speed (the booster compressor maintains turboexpander speed by absorbing the turboexpander power). The hydrogen booster compressor pressure ratio is very low due to the light gas. For a given compressor impeller tip speed, hydrogen compression pressure rise is approximately 1/8 of natural gas. Because the compressor boost is less important than turboexpander refrigeration, the compressor impeller design may focus more on mechanical limits rather than aerodynamic optimization. The high specific speed compressor impeller must be custom engineered for mechanical integrity, with minimal overhung mass to improve the rotordynamics for maximum speed. Furthermore, higher head coefficient compressor designs enable smaller diameter impellers and lower tip speeds.

Rotor-Bearing System

Depending on the turboexpander load, cryogenic turboexpanders typically use active-magnetic, oil-lubricated or gas bearings. Turboexpanders with magnetic or gas bearings do not require an additional oil supply system, and they offer no means in which oil can enter the process. Gas-bearing turboexpanders can be either dynamic or static gas-bearing turbines. Static gas-bearings require an external supply of process gas to the bearing, while dynamic gas bearings circulate available gas in the bearing housing. For smaller liquefiers, gas-bearing turboexpanders are a proven solution that can reach high peripheral speeds, which supports high expander efficiencies. Due to the lower-load capacity of gas bearings, larger liquefiers may require AMB technology for oil-free expander compressors.

Liquefier expander-compressors have a very low pressure ratio across the compressor, making automatic thrust load control per conventional design less effective. Thrust load compensation on the expander side is not recommended since it will add to the leakage and thermal loss. AMB technology (figure 18) offers improved thrust capability when compared to gas bearings, and it can typically handle the process loads without the need for additional thrust compensation.

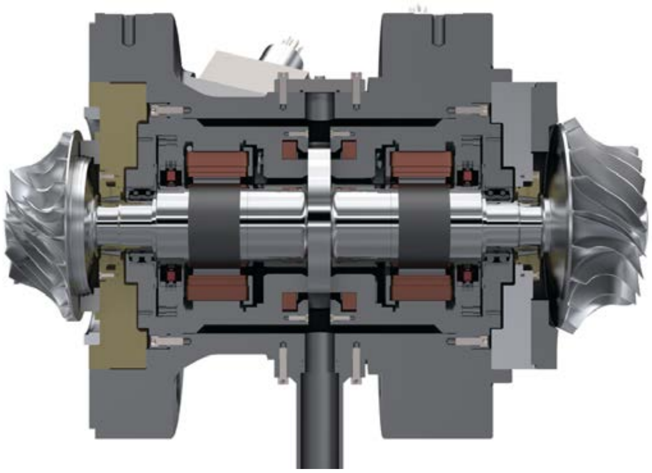


Figure 18. Magnetic Bearing Rotor-Bearing System

The rotor design aims to maximize the speed with the required double overhung impeller weights and maximize thrust load capacity to cover different process conditions. Figures 19 to 22 show rotordynamic analysis of a high-speed hydrogen turboexpander with active magnetic bearings. Aluminum expander and compressor wheels are mounted on the steel shaft by tension tie bolts. The magnetic bearing stiffness is in the range of 100-500 thousand lb./in (10-60 MM N/mm).

The large mass and stiffness of the housing and support structure have negligible effect on the rotor natural frequency and are not considered in the analysis. Undamped critical speed analysis shows the minimum synchronous forward bending critical speed at >120 percent of the trip speed.

Figure 19. Rotor 3D Model

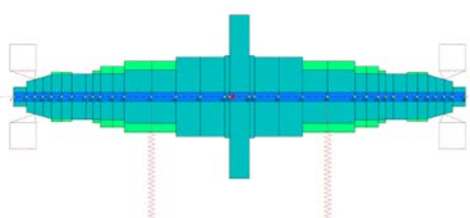


Figure 20. Rotor Mass Model

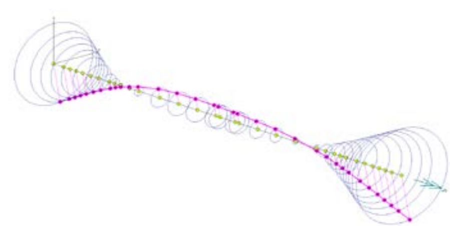


Figure 21. First Bending Mode

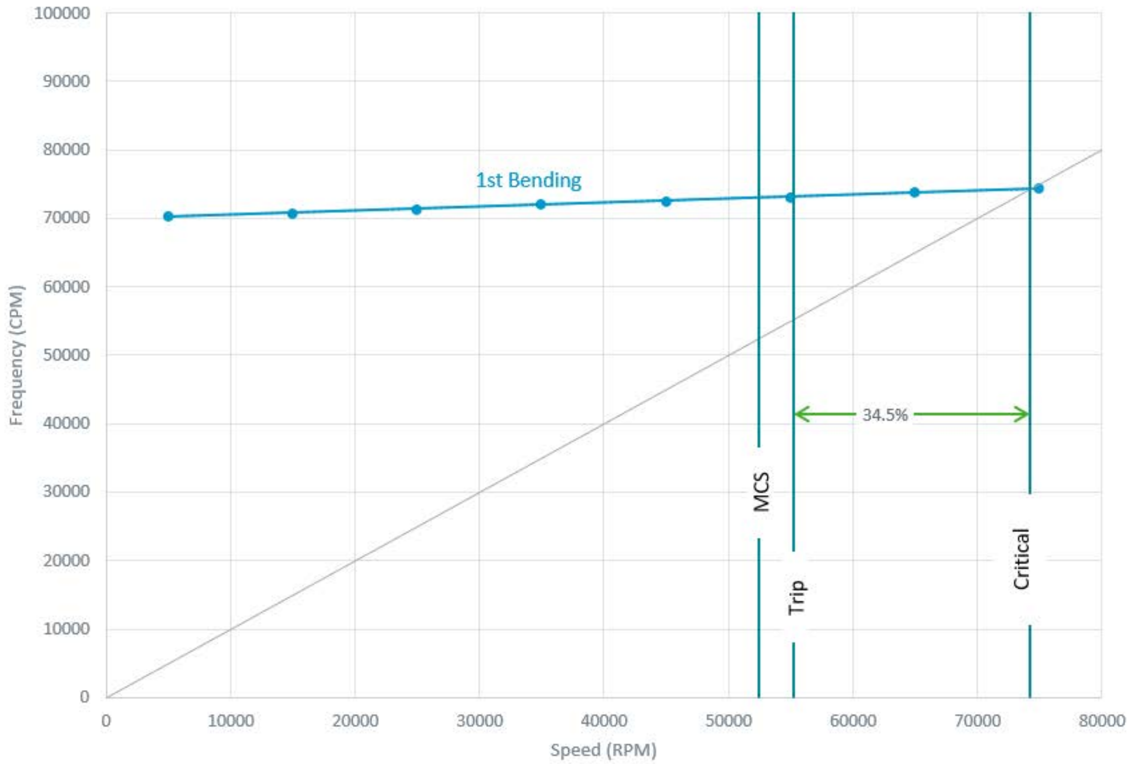


Figure 22. Campbell Diagram

Wheel and Shaft Sealing

Conventional turboexpander shaft sealing for hydrogen rich turboexpanders are oil-free labyrinth (Figure 23), carbon rings or dry gas seals (Figure 24). Conventional oil-free seals require warm seal gas towards the process, which will have a significant performance loss when mixing with deep cryogenic hydrogen gas. Innovative shaft sealing for liquefiers is required to avoid this mixing. The parasitic heat loss from conventional shaft sealing on AMB can be as high as 4-6 percent without additional mitigation. Other losses to consider include turboexpander wheel seals, in which flow bypassing the turboexpander wheel undergoes isenthalpic expansion, decreasing overall stage efficiency. Capable thrust management allows for fewer wheel seals and therefore reduced parasitic losses.

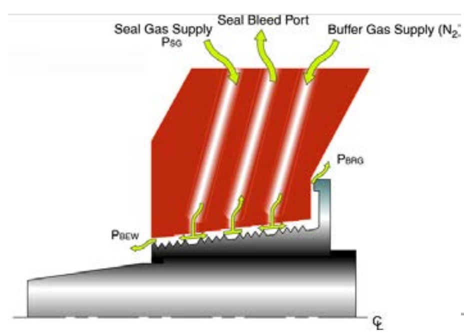


Figure 23. Labyrinth Seal

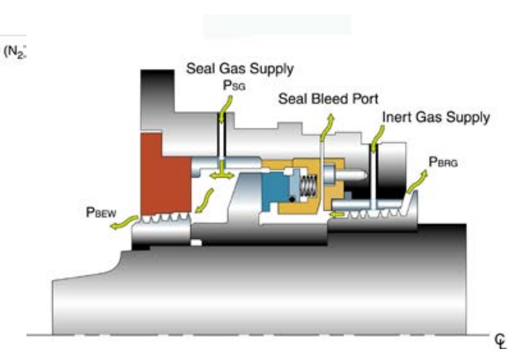


Figure 24. Dry Gas Seal

Thermal Management

Cold box interface, bolt stress going through start up and shut down resulting in joint leakages requires extensive analysis. The net turboexpander refrigeration depends both on aerodynamic performance and parasitic losses through the system, such as seal gas leakage, shroud or tip clearance leakage, and thermal soak into the system. The heat conduction of the rotor is minimal and is not significant compared to the casing interface with bearing carrier. A well designed thermal barrier between the joint will minimize the parasitic heat loss.

Turboexpander casing flanges are designed in one plane for the interface with the cold box. Interstage piping is also inside the cold box and/or vacuum jacketed. Integration of the turboexpander inside the liquefier cold box requires collaboration with cold box suppliers.

CONCLUSIONS

Turboexpanders have been used extensively in hydrogen-rich applications in the petrochemical industry for the last half century. Developments in the petrochemical industry can be adapted to the demands of green hydrogen liquefaction in order to support its growth. Turboexpander design for hydrogen liquefaction requires challenging aerodynamics, high tip speeds, innovative sealing systems, minimal clearance losses, suitable material selections and robust thermal management. Numerous hydrogen-rich turboexpanders designed and operating in the field have validated the design process. With careful evaluation, these machines can be adapted and qualified for hydrogen liquefaction at all scales.

NOMENCLATURE

AMB – Active Magnetic Bearings
BAHE - Brazed Aluminum Heat Exchanger
DGS – Dry Gas Seals
EOS – Equation of State
LH2 – Liquid Hydrogen
OPC - ortho-para catalyst
PDH – Propane Dehydrogenation
SMR – Steam Methane Reforming
TPD – Tons per Day
vIGV – Variable Inlet Guide Vanes

REFERENCES

Agahi, R., Ershaghi, B., Al Halaki, S., and Mayne, T., 2016, "Turboexpander Technology Contribution in Development of Ethylene Plant Process", Turbomachinery Laboratories, Texas A&M Engineering Experiment Station

Brun, K. and Simmons, S.B., 2021, "Compression Options for the Hydrogen Economy", *Gas Compression Magazine*, May 2021

Krasae-in, S., Stang, J. H., and Neksa, P., 2010, "Development of Large-Scale Hydrogen Liquefaction Processes From 1898 to 2009", *International Journal of Hydrogen Energy*, Volume 35, Issue 10, May 2010, pp. 4524-4533

Leachman, J. W., Jacobsen, R. T., Penoncello, S. G., and Lemmon, E. W., 2009, "Fundamental Equations of State for Parahydrogen, Normal Hydrogen and Orthohydrogen", *Journal of Physical and Chemical Reference Data 38* (2009) pp. 721-748

Ohlig, K. and Decker, L., 2014, "The Latest Developments and Outlook of Hydrogen Liquefaction Technology", *AIP Conference Proceedings 1573* (2014), pp. 1311-1317

Tomlinson, T. R., 2002, "Improved operating margins with Ethylene recovery", *Petroleum Technology Quarterly*, Spring 2002, pp. 125-131

Vitt, J. and Mayne, T., 2021, "Expander boosts Ethylene Recovery", *Turbomachinery International*, January/February 2021, pp. 26-27

Zhao, S., Zhao, J., and Lillard, J., 2020, "Selecting turboexpanders for PDH", *Petroleum Technology Quarterly*, Q4 2020, pp. 109-113