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INDUSTRIAL GAS TURBINES IN A DECARBONIZING ENERGY ENVIRONMENT

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Abstract

Worldwide efforts to reduce the carbon footprint of our energy use have posed a number of interesting challenges upon turbomachinery, including industrial gas turbines. Of particular importance is the requirement to reduce the carbon intensity while guaranteeing energy security.

These challenges are met by efficiency improvements to reduce carbon intensity, as well as carbon capture methods and the use of hydrogen as fuel. Stability of the energy supply, as well as electrical grid stability are paramount in these considerations. Related are concepts like dual drives, where gas turbines and electric motors are used on the same compressor train, exhaust gas recirculation to improve the effectiveness of carbon capture methods, gas turbines providing grid frequency regulation, but also the discussion about the transport of CO2, Hydrogen, Natural gas and Hydrogen Natural Gas Mixtures in pipelines, or the sequestration of CO2.

This paper will explain and evaluate some of the concepts above in the context of energy supply security and relative costs to reduce the carbon footprint for energy.

1- INTRODUCTION AND METHODS

The reduction of the carbon intensity in the energy context involves a wide range of measures. They include the improvement of the operational efficiency, fuel switch from coal to natural gas, or to hydrogen, the improvements in process efficiency, the capture and sequestration of carbon emissions, the decarbonization of fuels, and the use and storage of renewable electricity. The present discussion is by no means exhaustive, but will address a number of methods and areas where turbomachinery in general, and gas turbines in particular can contribute to the carbon reduction effort. Efficiency improvements can range for the reduction of leaks, the use of waste heat, but also improvements in the adaptation of the machine operation to the process needs. The latter topic has been discussed for many years, and are, for example, described in [1,2] While the natural gas leakage from centrifugal compressors only contributes a very minute amount to the natural gas emissions in compressor stations, available reduction methods include the improvements of Dry Gas Seal Systems, as well as leakage capture and recompression systems.

The use of renewable energy, and the increase of the use of electricity has also ignited a discussion on energy security, given the intermittent supply characteristic of many renewable sources. Here, so called dual drives can offer a number of advantages. Dual drives refer to compressors that are driven by two drivers, where one of the drivers is fossil fired, and one is an electric motor. Depending the architecture of these drives, a number of different goals can be met.

Carbon Capture from the exhaust gas of fossil fired power plants and the subsequent sequestration require a discussion of how to improve the efficiency of the capture process, as well as the organization of the transport to sequestration sites are discussed. Improving the efficiency of the capture process can include methods to increase the CO2 concentration in the exhaust gas stream, as well as discussions about improving the efficiency of the compression of the captured CO2. The transport of the CO2 to a sequestration site raises the question of the best pipeline pressure for the process. Another topic involves the question whether pre combustion or post combustion carbon removal is more advantageous. Further, the use of Hydrogen as a carbon free fuel leads to an interesting transport problem: Is it more efficient to make hydrogen at the site where energy is available, or at the place where hydrogen is used?

2- OPERATIONAL EFFICIENCY AND ENERGY SECURITY: DUAL DRIVES

While electric motor drives can reduce the carbon intensity of gas compression and boosting if the electricity is from renewable or carbon neutral sources, concerns are the supply security of electric power, as well as the cost fluctuations relative to natural gas, especially if the electric supply has to be un-interruptible. In addition, renewable electricity sources may be characterized by their variability and intermittency.

The combination of electric motor and gas turbine drivers to power a centrifugal compressor offers opportunities to take advantage of the individual strengths of each drive in a single package. In the context of carbon reduction and security of energy supply, it allows to operate the compressor with an electric motor, while having the immediate availability of a gas turbine if the electric power supply is disrupted [3,4]. Possible objectives for such drives can include managing emissions, optimizing efficiency, increasing power for specific situations (high ambient temperatures, starting, upset process conditions), the capability to optimize fuel cost by taking advantage of temporary price differences between gas and electricity (arbitrage), or the increase of availability if one fuel source is temporarily unavailable. This also reduces the reliance on electric contracts for un-interruptible supply. The large fluctuation in the availability of electricity from renewable sources can create interesting opportunities. The combination of gas turbines and electric drives can lead to trains where one of the drivers can be decoupled, and it can also lead to drives where electricity export is possible. Thus, different architectures will be advantages depending to a large degree on the project objectives.

There are a number of reasons to combine a gas turbine and an electric motor as compressor drivers in a single train:

1. The motor can serve as starter/helper driver for a large single shaft gas turbine used in LNG refrigeration trains.
2. They can also be used to allow compressors to be driven by the electric motor only if it is advantageous to use electricity (for example when a surplus green electricity is available, or to lower the carbon footprint of the station if low carbon electricity is available) or to use a natural gas fired gas turbine, either because it is economically attractive, or to protect the station from electricity outages.
3. It is also possible to configure drives to allow the electric motor to be used as a generator, using a 4 quadrant Variable Frequency Drive (VFD), thus feeding electricity in the grid when the gas turbine produces surplus power, and augmenting gas turbine power if the process or ambient temperatures require it.

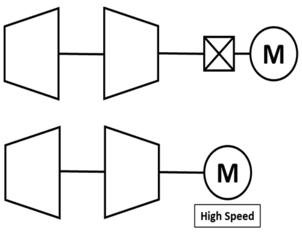
While the electric motors used as helper/starter drives (case 1) are usually significantly smaller in power than the respective gas turbines, and the same is often true for case (3), the use of dual drives in case (2) applications will usually have electric motors of about the same power range as the gas turbine.

The combination of electric and gas turbine drivers to power a centrifugal compressor offers opportunities to build on the strengths of the individual concepts. A number of technical challenges are to be addressed, not the least the torsional behavior of the train. The train configuration is determined by the different motivations and goals in the applications of dual drives. These can include the optimization of emissions, fuel consumption, available power and availability.

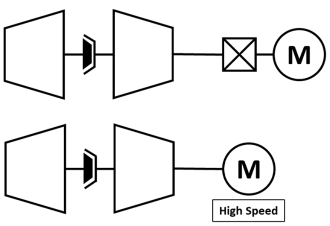
The discussion on different configurations can be made along the following lines: A key decision point is in the type of the electric motor, which can be either a constant speed motor, or a variable speed option, such as a variable speed motor (conventional with a gearbox, or with a high-speed motor), or a motor/generator (constant speed, variable speed with VFD, high-speed variable speed with VFD [3]).

Two other foundational concepts relate to the gas turbine: The gas turbine can be a single shaft machine or a two-shaft machine. Regardless of the type of gas turbine, for any concept where the electric motor operates while the gas turbine is shut down, the gas turbine must be uncoupled from the train. Otherwise, the windage losses from driving the airfoils in the turbine become unacceptably high. Also, the mechanical forces may damage the gas turbine. In other words, a clutch is always needed unless the intent is to continuously run the gas turbine. On the other hand, an electric motor (assuming it is an induction type motor) can stay engaged if it is de-energized. For the purpose of this study, we have assumed the use of two-shaft gas turbines to maintain the capability for a wide range of shaft speeds for process control. The technical challenges for a single shaft gas turbines will be similar.

From a mechanical point of view, we can distinguish drives with zero, one or two clutches (Figure 1). The number of clutches determines the possible uses.



a)



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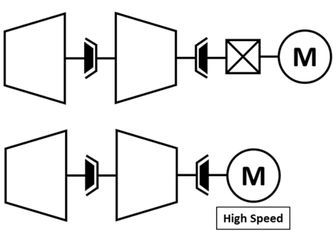

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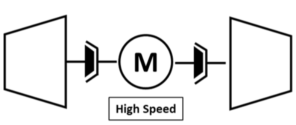


d)

Figure 1: Dual Drive arrangements: a)no clutch, b) single clutch (between gas turbine and compressor) c) 2 clutches d) 2 clutches, motor between gas turbine and compressor(requires a high speed motor)

The arrangement without clutches (Figure 1a) requires that the gas turbine always runs. Thus, the motor can be used to augment the power of the gas turbine, or, if a four quadrant VFD is used, it can act as a generator and absorb surplus gas turbine power. However, this configuration will allow only very limited arrangements for arbitrage, taking advantage of price differences between gas and electricity. It also will not allow to use only the motor without the gas turbine

Using a single clutch shown in Fig. 1b to separate the gas turbine in times when it is not used allows the switch between the use of electricity and the use of natural gas to power the compressor. The motor can be de-energized (see discussion below), but the motor, and if installed, the gearbox will create losses. The motor can potentially be used to augment the gas turbine power.

Using two clutches shown in Fig. 1c to separate the gas turbine in times when it is not used, or the motor if it is not used, allows the switch between the use of electricity and the use of natural gas to power the compressor [2]. In this configuration, there are no additional losses from the motor or the gearbox when they are not used. The discussion on the use of two clutches versus one clutch may be driven by the frequency of gas turbine use versus the frequency of motor use. If the gas turbine is the main source for powering the compressor, it is attractive to avoid the friction and windage losses from the motor.

A subset of the two clutch arrangement is an installation where the train has the motor between the gas turbine and the compressor shown in Fig. 1d. This arrangement allows the operation of the motor as a generator, without driving the compressor.

Another distinction can be the intent to operate the electric machine only as a motor, or use it as a motor and a generator. While we mostly focus on the use of the electric machine as a motor, it is possible to also use it as a generator, if either the train is controlled to run at constant speed (in this case, a single shaft machine may be attractive), or if we use a variable speed drive with a VFD that allows 4 Quadrant operation. The capability to generate electricity is usually only desirable when the electricity that is produced can be used or stored economically.

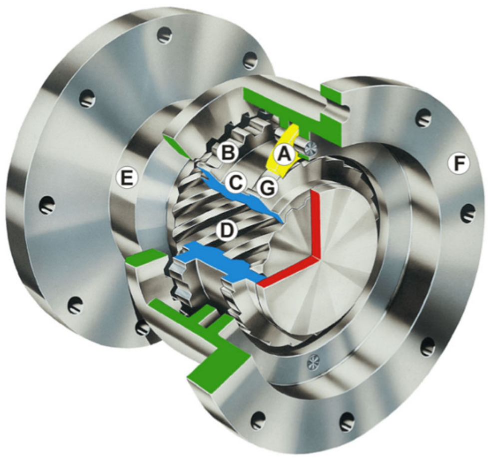
For some applications the motor will be rated for the same power as the gas turbine, while for others the motor is rated for a significantly lower power. The latter is often referred to as helper driver, and it provides additional power when the gas turbine power is insufficient, for example at high ambient temperatures, or process upsets. Also, it is often used as a starter motor in applications where the compressor has to be driven by a single shaft gas turbine. In cases where the motor and the gas turbine are used alternatively, such as for arbitrage, the motor will usually be rated similar to the gas turbine to match the compression duty. These decisions impact the design of the compressor, because the compressor aerodynamics, as well as the compressor torque capabilities, have to be sized accordingly.

In the dual driven systems discussed in this paper, the compressor is driven by either the Electric Motor (EM) or the Gas Turbine (GT). A clutch is installed on either side of the compressor and they connect the compressor to the drivers. The purpose of the clutch is to allow one driver to be stationary while the other one drives the compressor. The clutch acts like a freewheel in a bicycle and will engage whenever the driver speed matches and tends to overtake the speed of the driven machine (compressor). The clutch will disengage when one prime mover slows down relative to the compressor. The clutch engagement and disengagement is completely automatic, robust, and does not have to rely on any control system [3].

- When a driver is running, and the clutch is engaged, there is a one-to-one transmission of torque through conservatively rated gear teeth. Therefore, there is no slipping, low vibration, high overload capability and no wear to the driving parts. The clutch will remain engaged whenever it is transmitting positive torque, even when the power or speed of the compressor is varied by the operator.
- When one driver is shut down and the associated clutch disengaged there will be low losses (negligible heat rejection into the oil) and no wear to the engaging mechanism.

These requirements are met by a special type of overrunning mechanical clutch (Figure 2), with its one-way torque transmission capability using accurate gear teeth and unique engagement and driving features.

In some systems the changeovers from one driver to the other are required when the compressor is at low speed or at rest. The overrunning clutch can also engage and disengage at full speed, and this is important for fault tolerance and variable operational requirements over an extended period of time. The clutch therefore provides significant operational flexibility.



Key:

A – Pawl

B – Main driving teeth

C – Helical Sliding Component

D – Helical Spline

E – Input Component

F – Output Component

G – Ratchet Teeth

Figure 2: Clutch

The following example shows a typical layout, with a 2 shaft gas turbine, a VFD driven electric motor and a gearbox, and two clutches (Figure 3). The centrifugal gas compressor would be sized for the required pipeline flow and pressure conditions. The required power to meet the demand can be delivered by either a variable frequency controlled electric motor, that drives one shaft end through a parallel shaft gearbox, or a 2 shaft industrial gas turbine, driving the compressor from the other shaft end. Both the electric motor and the gas turbine are sized to meet the entire power demand individually. If surplus electric power from the grid is available, the compressor will be driven by the electric motor, while for all other times, the compressor will be driven by the gas turbine. Operation with compressor driven by both the electric motor and the gas turbine at the same time was not envisioned. The desired modes of operation require two clutches, allowing the separation of either the motor and the gas turbine from the drive train [3].

As can be seen from the layout view (Figure 3) the arrangement leads to a very long train. Two identical overrunning clutches are placed both on the suction and the discharge end of the gas compressor, allowing power to flow from either the gas turbine (GT) (left) or the electric motor (EMD) and gear box (right) to the compressor. Three high-speed couplings and one low-speed coupling (not visible in the layout for clarity) complete the train. Not shown here is the variable frequency drive (VFD) to provide power to the electric motor to match the speed / torque characteristics of the gas turbine. The VFD is usually placed in a safe area (for example an e-house or control building) and connected to the motor via underground cables.

When the compressor is coupled to the gas turbine, process control is performed via the power setting of the gas turbine: The gas turbine is set to produce a certain amount of power, and the compressor will run at the speed at which it absorbs that power. Compressor suction and discharge pressure are dictated by the pipeline operating point, and therefore the resulting compressor flow is dictated by the gas turbine power. When the compressor is coupled to the electric motor, process control is effected by the speed setting of the electric motor. Again, the compressor suction and discharge pressure are dictated by the pipeline operating point, and the electric motor will be operated at the necessary speed that allows for a prescribed flow. Part of the compressor control is a surge avoidance system that has to be configured to allow for the control mode of the driver for either mode of operation.

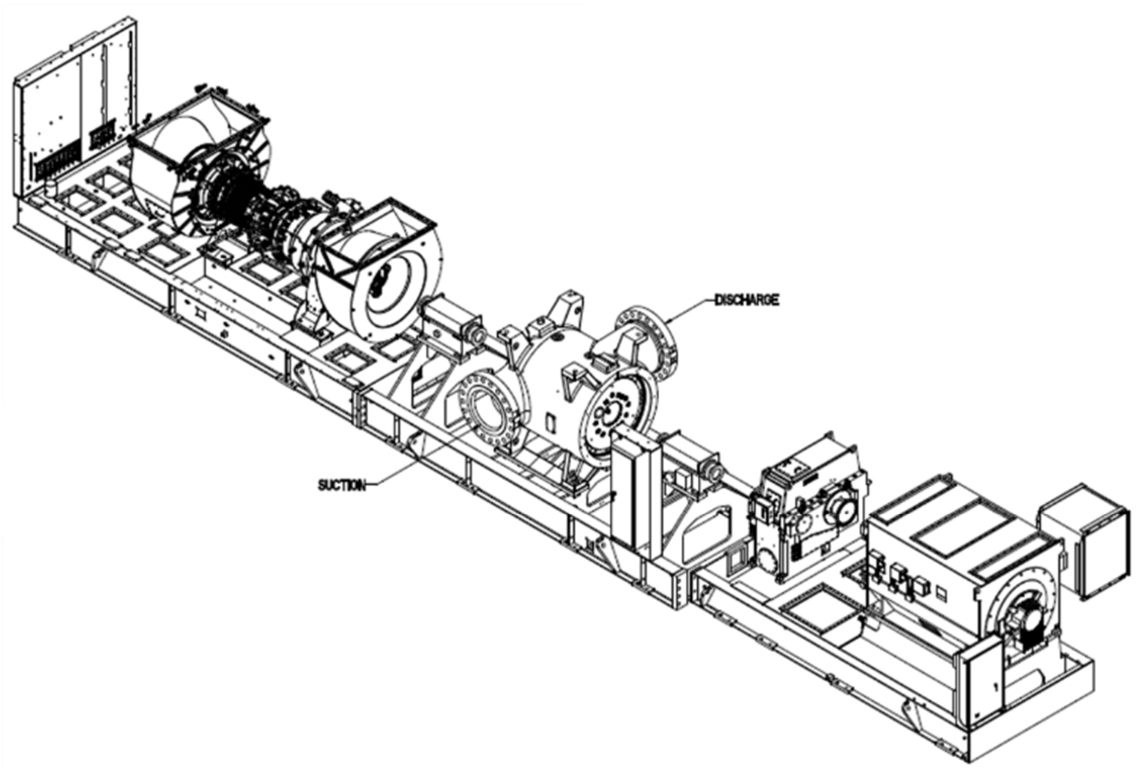


Figure 3: Example of a dual drive package for a two clutch arrangement. The two-shaft gas turbine is on the left, the variable speed electric motor on the right [4].

3-FUEL AND COMBUSTION

In the context of carbon reduction, there are two fundamentally different approaches for fossil fired engines and power plants: Capturing carbon from the exhaust of the gas turbine or providing a fuel that does not contain carbon [5].If the fuel is a fossil fuel like natural gas (NG) that will be brought to the site, one has to consider either pre- or post- combustion carbon capture.

Thus, to generate a certain amount electricity in a gas turbine plant carbon free, the choices are:

1- Feed NG to make blue hydrogen, compress the hydrogen to combustor pressure, capture the CO_2 created in the process, and compress it to the required pipeline or sequestration pressure.

2- Burn NG in a gas turbine, capture the CO_2 in the exhaust and compress the CO_2 to pipeline or sequestration pressure.

3- Bring green Hydrogen to the plant

The discussion in this section will focus on options 1 and 2.

Most hydrogen produced today in the United States is made via steam-methane reforming (SMR) , a mature production process in which high-temperature steam (700°C–1,000°C) is used to produce hydrogen from a methane source, such as NG. For the purpose of this discussion we will assume the use of an SMR process. In the steam-methane reforming process, methane reacts with steam under 3–25 bar pressure (40 to 370 psi) in the presence of a catalyst to produce hydrogen, carbon monoxide, and a relatively small amount of carbon dioxide. Steam reforming is endothermic—that is, heat must be supplied to the process for the reaction to proceed.

Steam-methane reforming reaction
$CH_4 + H_2O$ (+ heat) → $CO + 3H_2$

Water-gas shift reaction
$CO + H_2O \rightarrow CO_2 + H_2$ (+ small amount of heat)

Subsequently, in what is called the "water-gas shift reaction," the carbon monoxide and steam are reacted using a catalyst to produce carbon dioxide and more hydrogen. In a final process step called "pressure-swing adsorption," carbon dioxide and other impurities are removed from the gas stream, leaving essentially pure hydrogen. Steam reforming can also be used to produce hydrogen from other fuels, such as ethanol, propane, or even gasoline. The SMR process should be capable to provide Hydrogen at up to 60 bar pressure. Given the necessary fuel gas pressure at the Gas Turbine skid edge of 400 to 500 psia (27 -35 bara), it should be possible to avoid the very energy intensive compression of Hydrogen. Similarly, natural gas is typically supplied at the necessary pressure and therefore also does not need additional compression. The CO_2 created from this process would be available, depending on the removal process at about 1 bar [5]. To produce Hydrogen with the energy content of 1 MW (LHV),ie 8.34 10^{-3} kg/s from 16.59 10^{-3} kg/s of Methane would yield 45.52 10^{-3} kg/s of CO2, not counting the external energy necessary for the reaction.

The combustion process with

$CH_4 + 2O_2 \dashrightarrow CO_2 + 2H_2O$

requires 19.98 10^{-3} kg/s of Methane, and yields 54.88 10^{-3} kg/s of CO2 for an energy content (LHV) of 1 MW. The output at the gas turbine generator terminals is now simply the energy flow divided by the thermal efficiency.

Setting aside the performance differences that may arise from different fuels (as discussed earlier), we get the same power out from the gas turbine from 8.34 kg/s of H_2 or from 19.98 kg/s of Methane. The significance of this finding is, that firing the gas turbine with SMR produced hydrogen creates 45.52 $\cdot 10^{-3}$ kg/s/MW of CO_2, while using Methane will produce 54.88 10^{-3} kg/s/MW of CO_2 for the same Gas turbine power output. However, once the energy consumption for the required reaction is accounted for, the SMR process produces about 75 $\cdot 10^{-3}$ kg/s/ MW of CO_2, which equates to 9 kg of CO_2 per 1 kg of H_2 [5].

Based on the calculations in [5] one of the key differences between pre and post combustion capture is that the SMR process produces 1.37 times the CO_2 that is produced in Methane combustion.

Carbon intensity is the rate at which CO_2 is being produced per a unit of energy. This is a normalized way to compare amount of CO_2 being generated by any fossil fuel burning energy source. Every country participating in the global effort of reducing CO_2 emissions seeks to reduce the overall carbon intensity of their energy production. Figure 7 shows the carbon intensity for a selection of industrial gas turbines running on NG, NG / hydrogen blends, gas turbines with a bottoming cycle, and gas turbines with a WHRS, all compared to the average carbon intensity of the California grid in 2020 with significant renewable energy [5]. If the fuel delivered to the engine had a blend of 20% hydrogen by volume this would reduce the carbon intensity by about 7%. At 50% hydrogen this would reduce carbon intensity by 23%. Considering configurations with a waste heat recovery system (WHRS) or combined heat and power (CHP), the carbon intensity is similar to what is being provided by the California utility grid. If CC could occur in a pre- combustion (SMR making blue hydrogen) or post-combustion method, the carbon intensity, close to zero, would be well below the California average noted in Figure 4.

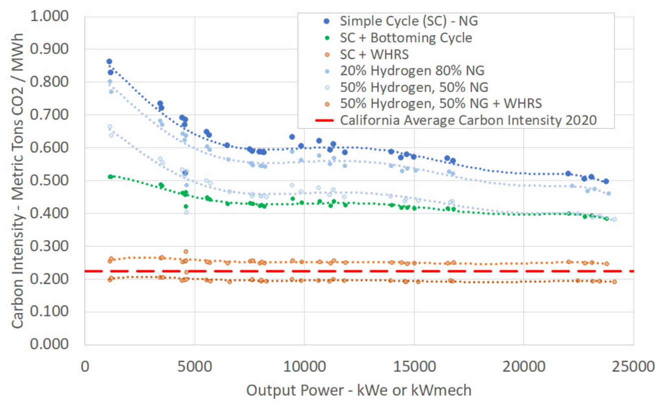


Figure 4: Full Load @ 15°C Sea Level – Carbon Intensity vs Output Power

A mitigation strategy to reduce carbon intensity is to add in hydrogen into existing NG pipelines to help reduce the CO_2 emissions at the destination where the gas is utilized. Hydrogen is a very light gas consuming much of the volume of the mixed combustible gas when combined with NG which is predominately methane CH_4. A way to look at blending the hydrogen in with the NG is to see how the resultant gas mixture changes in fuel energy and carbon content by volume as depicted in Figure 5.

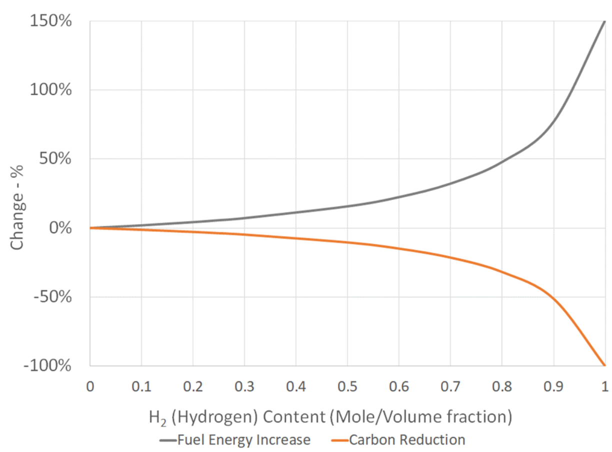


Figure 5: Change in fuel energy and carbon content versus H_2 mole/volume fraction blended with NG

With the properties of NG/hydrogen understood, we can move on to consider how this affects the performance of the gas turbine and discuss scenarios of creating hydrogen and/or capturing and processing the CO_2 to minimize carbon intensity of operating industrial gas turbine equipment.

To examine different scenarios, a performance simulation model for single shaft, 16.5MWe simple cycle gas turbine is exercised and used as the baseline specimen. The cold end drive (CED) gas turbine engine produces about 35.4% thermal efficiency, an inlet flow of 53.4 kg/s, and a P/P of 19.4:1, operating at 15°C, 60% relative humidity, full load operating on NG. The baseline configuration is the gas turbine receiving gas fuel from the extensive network of NG pipelines. In the future, hydrogen may be added in with the NG to lower CO_2 emissions and carbon intensity of power generation applications.

Figure 6 indicates how performance improves and the carbon intensity reduces as the hydrogen content increases on this gas turbine. The efficiency benefit serves to reduce the carbon intensity of the operating engine in combination with the reduced carbon in the fuel for higher amounts of hydrogen blended with the NG.

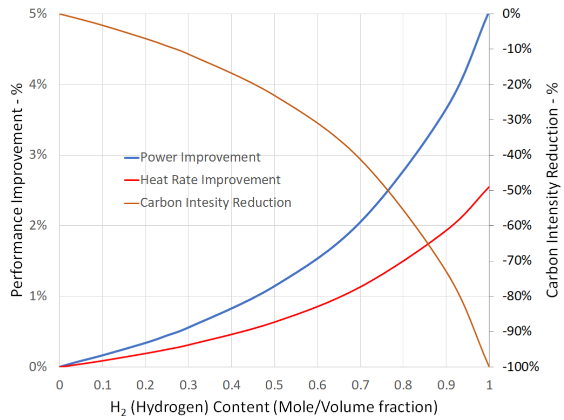


Figure 6: Performance Improvement, 16.5 MWe gas turbine. Carbon Intensity Reduction versus H_2 mole/volume fraction blended with NG

The efficiency improvement is due to the constituents in the products of combustion which alter the specific heat of the working fluid expanding across the turbine. Higher fuel H/C (Hydrogen to Carbon) ratio increases the specific heat of the vitiated gas expanding through the turbine increasing the work for a given temperature difference, improving both power and efficiency of the system. This is researched more thoroughly in a previous work [6].

SMR is the primary means to create H_2 today for industrial applications. SMR will be able to extract the H_2 from the NG fuel consisting primarily of CH_4. To make this "blue H_2", a SMR system will significantly reduce the efficiency of the gas turbine system and will produce more CO_2 per MWe than the baseline configuration. SMR is typically between 65% and 75% efficient in converting NG to H_2 [5], which is consistent with the excess CO_2 being produced in the process. SMR systems can either be sized for a single unit, but likely a larger SMR system would serve multiple gas turbines due to the economics of scale. In this case study, the gas turbine gets the benefit of running to H_2 which improves full power and efficiency, however, the inefficiency of the SMR reduces overall system efficiency substantially and generates more CO_2 emissions than any other configuration. There is a significant power parasitic to the system considering the power needed to run the SMR and the power needed to compress the CO_2 to pipeline pressure.

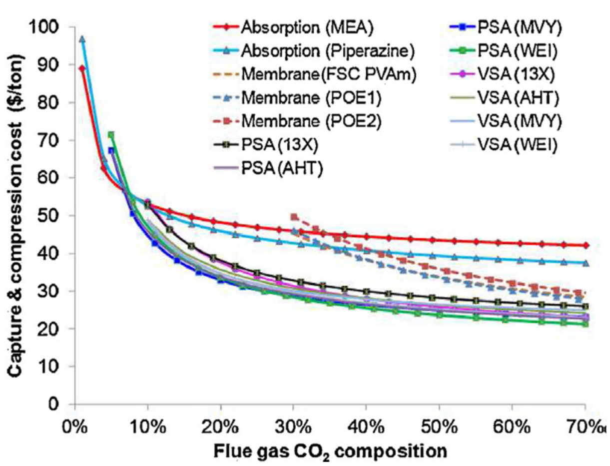


Figure 7: CO2 capture and compression costs for various materials and technologies [7].

Another option is a gas turbine system with post-combustion carbon capture (CC). The cost of carbon capture is sensitive to the CO_2 concentration in the exhaust gas (Figure 7). Therefore, this configuration uses exhaust gas recirculation (EGR) to increase the concentration of CO_2 in the exhaust to improve the economics of CC. Previous work [8,9] reviewed in detail the effect of EGR on performance and combustion behavior at full load and part load operating conditions for the same reference 16.5MW engine system. Observations and explanations of the system and component analysis results are also described in these references.

Figure 8 shows how running to different EGR levels effect the full load output power at different ambient temperatures. This additional power, especially on warm days is helpful to partially offset the power needed to run the accessory equipment to enable CC and compress the captured CO_2 to a pipeline pressure. The assumption for the power needed to run the CC and compression to pipeline pressure is roughly 10% of the rated power of the engine. To operate without the need of supplemental O_2, it is likely the EGR system will recirculate only about 50% of the exhaust flow to continue to support robust combustion and eliminate the need for an air separation plant to provide O_2.

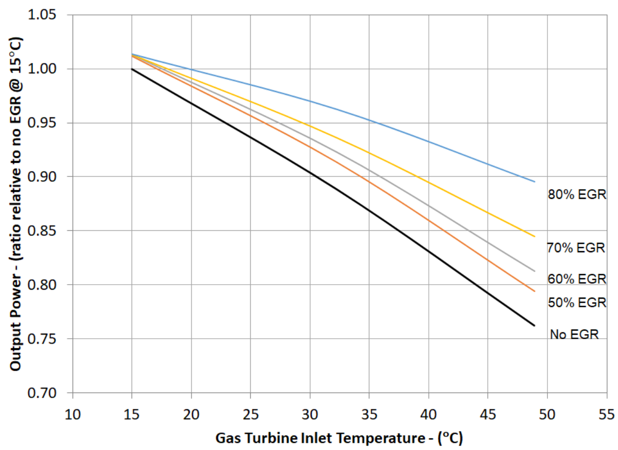


Figure 8: Full Load @ Constant EGR Levels – Relative Power vs Inlet Temperature

Figure 9 is added to show how the volumetric flow of the gas turbine changes at different EGR levels and inlet temperatures. A gas turbine is generally characterized as a constant volume machine. This plot shows how this is essentially the case where all data is within ~5% of the 15°C reference condition, but with secondary effects due to the inlet compressor characteristic and when the inlet gas constituents change. Figure 10 shows how the inlet flow to the gas turbine changes in density at different EGR levels and inlet temperatures. The 50% EGR case at hot day dips below the 0% EGR case on hot days due to being more saturated with water vapor with the wetter exhaust gas being recirculated.

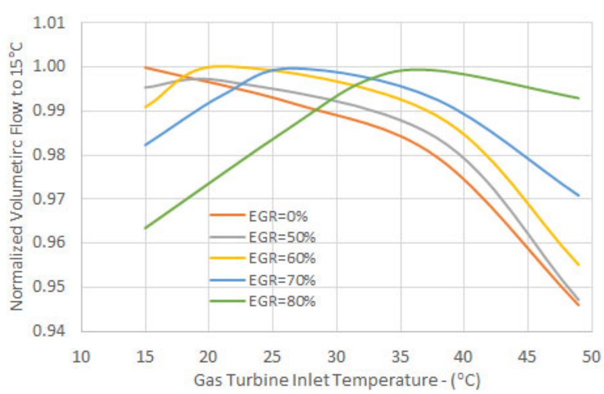


Figure 9: Full Load @ Constant EGR Levels – Relative Volumetric Flow vs Inlet Temperature

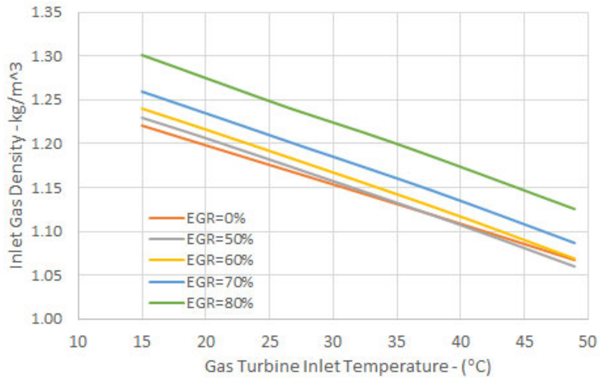


Figure 10: Full Load @ Constant EGR Levels – Gas Turbine Inlet Gas Density vs Inlet Temperature

To simplify this study, we are comparing the pre- (SMR) and post- (EGR) combustion CC configurations versus the baseline configuration at full load 15°C, standard day conditions only. A bottoming cycle is also indicated and contrasted. The general premise is that an operator of a gas turbine would continue to operate their gas turbine normally until the incentives for CC get compelling enough to invest in mitigations that reduce their carbon intensity or capture and process the CO_2 without letting the gas escape to the atmosphere. The approximate effect on how the net performance output on the gas turbine relative to the baseline configuration is shown in table 1. Both the SMR and EGR will initially provide a larger generator output power, but the parasitic loads from the balance of plant (BOP) reduces the net power below the baseline, and ultimately the power delivered to the operator's customers. The BOP includes power to run the SMR, EGR, CC systems, accountable power losses due to integration with the gas turbine, and power needed to compress the captured CO_2 to pipeline pressure. Both the SMR and EGR configurations will produce more CO_2, however 98% of this gas is being prevented from entering the atmosphere. This relative data will be used below for the techno-economic assessment.

Table 1: Performance Comparison to Pre- and Post- Combustion CC Configurations and a Bottoming Cycle Configuration relativized versus the Baseline

Configurations	Baseline	Pre-Comb - SMR	Post-Comb - EGR	Baseline+Bottoming Cycle
Gas Turbine Available Power (Full Load)	1.000	1.053	1.010	1.333
Demand Power	1.000	1.000	1.000	1.000
Thermal Efficiency of Gas Turbine	1.000	1.026	0.992	1.333
Parasitic Power rel to Full Load - %	0.0%	10.0%	10.0%	1.0%
Net Power Delivered After Parasitics	1.000	0.947	0.909	1.319
System Efficiency	1.000	0.749	0.892	1.319
Carbon Intensity prior to Capture	1.000	1.454	1.121	0.758

A power generation site needs 16.5 MWe with a CO_2 capture using EGR : Assuming that after the implementation of CO_2 tax, the gas turbines operators do not increase the price for their services i.e., electrical power, rather they find a technology for which the CO_2 tax savings justify the CAPEX and OPEX and other costs associated with it. Application of EGR increases the concentration of CO_2 in the gas turbine exhaust, which in turn helps reduce the cost of CO_2 capture. As shown in Table 1, the application of EGR enhances the gas turbine power output by 1% but makes it 0.3% less efficient. Figure 11 shows a range of CO_2 tax rate justified for implementing a post-combustion CO_2 capture technology using EGR, while using the assumptions stated above in the analysis.

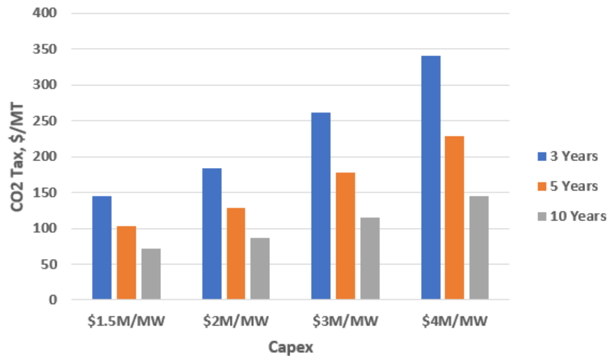


Figure 11. CO_2 tax/MT at different Capex over the discounted payback period of 3, 5 and 10 years for CCUS using EGR

At the technology CAPEX of $2M/MW, the CO_2 tax saving needed to justify the costs is between $128/MT for 5 years payback period and $87/MT for 10 years payback period. Although the CO_2 tax is on the higher side, still it is 40% less than that for Capex of $4M/MW, which is needed for CO_2 capture without using EGR. With the EGR technology advancements if the CAPEX could be reduced to $1.5M/MW, the CO_2 tax saving needed to justify the technology costs would be between $103/MT for 5 years payback period and $72/MT for 10 years payback period. If the facility has access to generate revenue from selling captured CO_2 for EOR at $35/MT, at $2M/MW CAPEX, the CO_2 tax saving needed to justify the CAPEX and OPEX deceases by 29% to $92/MT for 5 years payback period and $51/MT for 10 years payback period. And at CAPEX $1.5M/MW, the CO_2 tax saving needed to justify the costs would be between $67/MT for 5 years payback period and $36/MT for 10 years payback period. These values of CO_2 tax rates are within the range defined in the Paris Agreement. It is clear from the results in figure 11 that the expectation of 3 years payback period is unrealistic. A payback period between 5 years and 10 years is more realistic. The implementation of CO_2 capture using EGR at CAPEX $1.5M/MW could be economically viable at tax rate of $50/MT of CO_2 for payback between 5 years and 10 years [5].

A power generation site needs 16.5 MW with simple cycle gas turbine operating on hydrogen from SMR : The hydrogen energy vector provides a mode of pre-combustion carbon reduction. This section presents the economic analysis of operating a gas turbine on hydrogen produced from a SMR using natural gas fuel onsite. As shown in Table 1, due to the SMR-GT system thermal efficiency at 26.5%, natural gas consumption and CO_2 emissions in hydrogen production are 41% higher compared to that in the baseline. Hence, there is an added cost of operation per year from additional natural gas needed. The power penalty from the SMR operation and CO_2 capture and compression is assumed to be 10% of the total power. Based on the performance analysis, 5% higher power is expected from the gas turbine running on hydrogen than on natural gas. It helps in reducing the cost of electrical power from the grid needed to compensate the power deficit from 16.5 MW. The maintenance cost is assumed as $500k per year and CO2 storage & transport cost as $10/MT, both same as that in the EGR analysis.

The CAPEX of SMR and carbon capture system depends on the size of the system. Due to the economy of scales, larger systems are relatively less expensive than the smaller systems. A sensitivity analysis has been performed on the CAPEX values from $1.5M/MW to $4M/MW, which is the same range used in the analysis of the EGR system. The analysis results in figure 12 shows that if the CAPEX of SMR is $1.5M/MW, the CO_2 tax saving needed to justify the technology costs would be between $140MT for 5 years payback period and $108/MT for 10 years payback period, which are 35% and 49% higher than the respective EGR cases. At the technology CAPEX of $2M/MW, $3M/MW or $4M/MW, the CO_2 needed to justify the technology costs are beyond the CO_2 tax brackets in the Paris Agreement.

If the facility has access to generate revenue from selling captured CO_2 for EOR at $35/MT, at $1.5M/MW CAPEX, the CO_2 tax saving needed to justify the costs would be between $90/MT for 5 years payback period and $58/MT for 10 years payback period. These values of CO_2 tax rates are within the range defined in the Paris Agreement, but still higher by 33% and 60%, compared with those of the respective EGR cases.

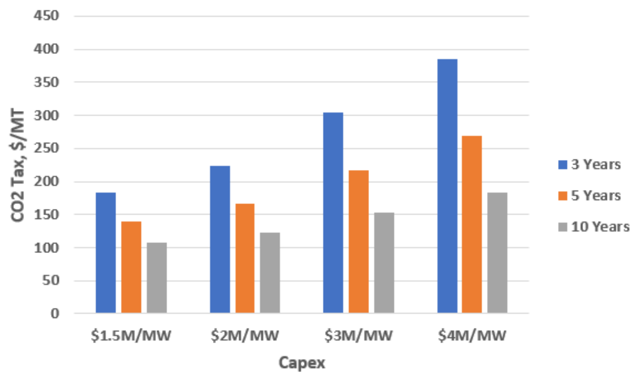


Figure 12. CO_2 tax/MT at different Capex of SMR over the discounted payback period of 3, 5 and 10 years

Currently, hydrogen produced from CO_2 free sources is very expensive. The cost of green hydrogen, produced from electrolysis of water, is about $5-$7 per kg of H2. The cost of blue hydrogen, produced from Steam Methane Reforming (SMR), is more than $3 per kg of H2. The US Department of Energy launched the first Energy Earthshot program called Hydrogen Shot on June 7, 2021. The goal of the program is to reduce the cost of clean hydrogen by 80% to $1 per 1 kilogram in 1 decade ("1 1 1") [10].

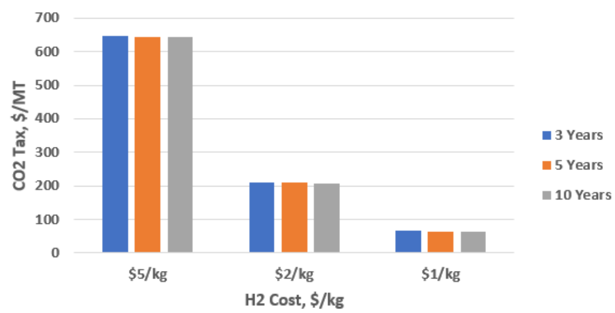


Figure 13. CO_2 tax/MT at different costs of hydrogen over the discounted payback period of 3, 5 and 10 years for the use of hydrogen a gas turbine

In this section, CO_2 tax, $/MT of CO_2, has been determined for the cost of H2 ($/kg), as shown in figure 13. These calculations assumed $1M Capex to convert the existing NG gas turbine to a H2 gas turbine. At today's cost of $5/kg of Green H2, the CO_2 tax to be avoided to justify the investments is around $645/MT. At $2/kg of H2, the CO_2 tax reduces to about $210/MT. The CO_2 tax reduces to $65/MT for the hydrogen cost of $1/kg, which is within the range of Paris Agreement.

The objective of this section was to learn more about the potential options that can be applied to existing and future industrial gas turbine systems that make them financially viable and environmentally sustainable for operators. The premise is to adapt the existing ubiquitous NG infrastructure to greatly reduce the carbon intensity in an affordable way. As carbon pricing and incentives are established more in the world marketplace, gas turbine operators may have options with pre- or post- combustion CC technologies like SMR, converting pipeline NG to H_2 while capturing and processing the CO_2 or capturing the CO_2 in the gas turbine exhaust for sequestration, with is made more affordable using EGR.

Key findings include the following:
The SMR process, converting NG fuel into H2, has been examined and from references, the inefficiency of the process produces 9 kg of CO2 per 1 kg of H2. The SMR will drop the efficiency of the gas turbine system by 25% considering the energy value of the fuel. The SMR will require 4.5 kg of water for every 1 kg of H2 produced. The techno-economic analysis suggests that using carbon capture (CC) with EGR (post-combustion) may be a better approach than using SMR to make H2. EGR will be more efficient, will produce a lesser amount of CO2 at a given power level, and will not need nearly as much water as the SMR. A bottoming cycle will be complimentary to EGR since the exhaust temperature needs to be

reduced anyway. Today's green H2 cost is too high to be affordable. Bringing the cost of H2 to the target price of $1/kg would be competitive and well within the Paris Agreement price range.

Generally, SMR will lower the efficiency of the gas turbine engine system and will produce more CO_2 emissions to dispose of due to the inefficiency of converting NG to H_2. The capital expense for a CC technology has a strong impact to determine if it is a good investment to offset carbon pricing over the long term.

The data in Figure 14 allows to determine the compression power for a given CO_2 mass flow. The fact that for a given engine output (or, for identical engine energy input) the SMR process will create more CO_2 than the Methane combustion process leads to a higher compression power consumption and a larger, and thus more expensive compressor for the option that uses an SMR to create Hydrogen, unless no additional CO_2 generation occurs from the heat energy requirements in the SMR process.

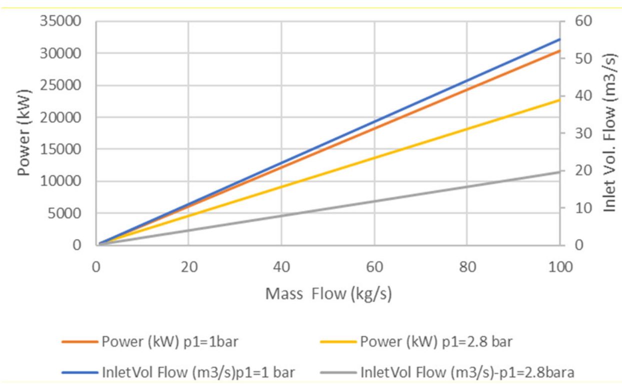


Figure 14: Compression of CO_2 (1 bar to 140 bar or 2.8 bar to 140 bar): Size of the compressor shown as inlet volumetric flow and power consumption for 83% stage efficiency,3% pressure drop per cooler. Compressor with 8+1 Stages for p1=1, 6+1 stages for p1=2.8. The last stage is dense phase compression [5].

An issue that needs to be addressed is the fact that the compression task will not be for pure CO_2. CO_2 captured from gas turbine exhaust and from the SMR process will contain some water (appr. 0.14% by mole), while CO_2 from SMR will also contain CO (appr. 1.7%), Hydrogen (appr. 1.7%) Methane (appr. 0.035%) [11]. The contaminants are small enough to avoid significant changes in the compression process, except for the dense phase compression, where the added contaminants as listed for the SMR process cause a sufficient change in the critical point to move out of the dense phase region. In either case, the water will have to be removed as part of the compression process, as water and CO_2 form acids that are corrosive, and require materials upgrades. The addition of a dehydration system will cause additional pressure losses in the compression system.

4-CARBON TRANSPORT AND SEQUESTRATION

In the previous section, methods to improve the carbon capture process were discussed. Regardless of the method, the next step is the transport of the captured CO2 to a sequestration site, and its sequestration. Usually, the sequestration sites will be at some distance from the capture site (Figure 15). For larger amounts of CO2, the transport in pipelines is advantageous. While Figure 15 indicates that most sequestration sites will be relatively close to the capture sites, there are instances where CO2 pipelines for distances exceeding 500 miles are discussed [11].

Figure 16 shows a Mollier diagram for CO2. While CO2 transport in dense phase requires the least amount of energy, lower pressures may be considered, even though the power consumption for the transport will increase. Lower pressure pipelines are considered for 2 reasons: (1) The transport is for a relatively short distance or (2) Existing pipelines with a lower pressure rating (for example natural gas pipelines with a typical pressure rating of about 105 bara) are to be used, to avoid the requirement for new pipeline construction [12].

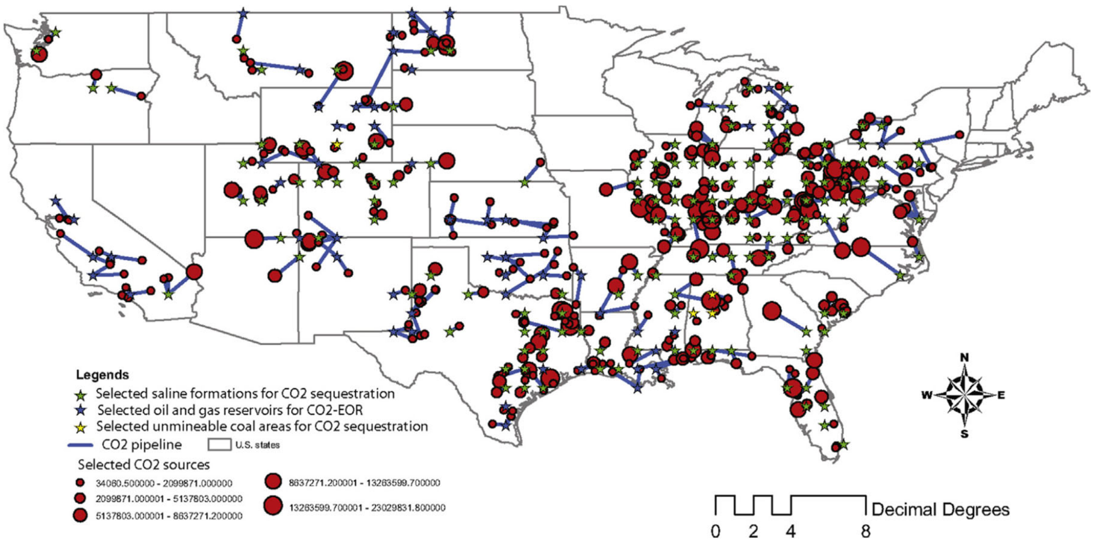


Figure 15: Nationwide CCUS supply chain network to reduce 50% CO2emissions from the stationary sources. The network includes 444 sources (red circles) for CO2capture,76 oil and gas reservoirs (blue stars) for CO2utilization, and 151 saline formations (green stars) and 6 unmineable coalbeds (yellow stars) for CO2sequestration. Blue lines represent the pipelines connecting sources to the injection sites. [7].

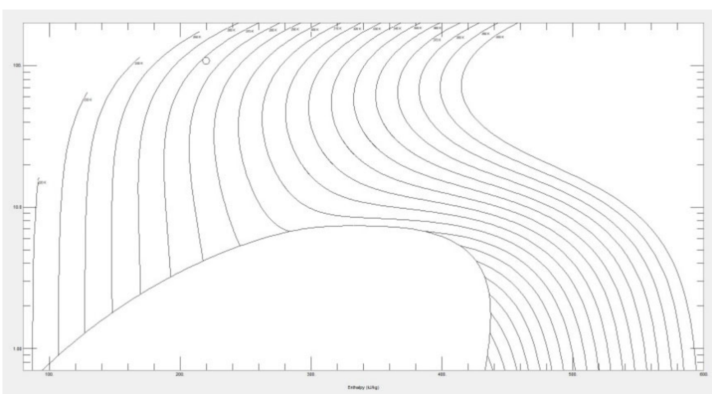


Figure 16: Pressure -enthalpy diagram for CO2. If the Pipeline pressure drops below the critical pressure, liquid dropout can occur at common ambient temperatures.

Thus, depending on the distance the CO2 has to be transported, different pipeline options exist:

- Either, the pipeline pressure p_p for CO_2 has to be in the dense phase region to reduce the energy cost of transportation,
- or the pressure can be lower, for example to allow the use of existing pipelines, or to lower the cost of compression from capture pressure to pipeline pressure.

Current CO_2 pipelines use an operating pressure of 140 bara (2000 psi) which gives sufficient margin to the critical pressure of CO_2 [11,12]. Generally, to take advantage of dense phase transport, pipeline pressures can be between about 110 to 150 bar. In dense phase, CO_2 has a density near that of liquids, but is compressible like a gas, and has a low viscosity. This also allows to limit flange ratings to ASME/ANSI 900#. Dropping the pressure below the critical pressure creates the risk of two phase (ie. liquid and gas) operation of the pipeline, which has to be avoided. If, on the other hand, existing natural gas pipelines are to be used, the pressure rating of such pipelines in the range of 100 to 105 bar precludes the operation in dense phase. Further, in order to avoid the formation of liquid CO2 in the pipeline, especially at low soil or ambient temperatures will further limit the acceptable operating pressure to about 35 to 50 bar.

In either case, the CO2 will be available after capture at pressures that are slightly above ambient pressure. Thus, the first question is a discussion on power requirements to bring the CO2 to the required pipeline pressure, especially if dense phase transport is required. The power demand is expressed as work, with work (or enthalpy rise) defined as power consumption per unit of mass flow. Given the high pressure ratio required, studies show that an economical way to increase from the pressure from 1 bar to 140 bar requires multiple compression steps with intercooling in between (Figure 17). In the example, after 5 stages of compression (each followed by a gas cooler), the gas is brought to dense phase by cooling. The last part of the compression, which occurs in dense phase, can either use a compressor stage or a pump stage.

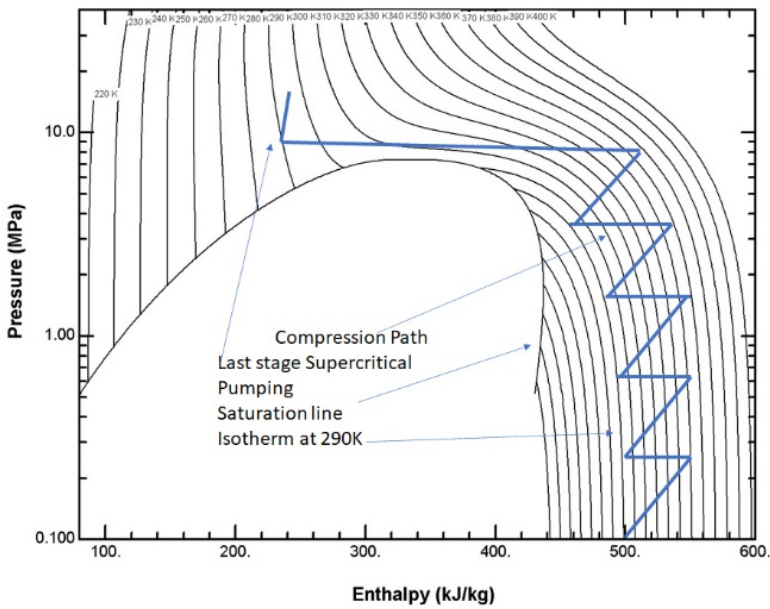


Figure 17: Compression of CO_2, Pressure versus Enthalpy [9]

The compression to lower pressures would require somewhat less power (Figure 18), with compression to only 35 bar requiring about 73% of the power necessary to reach 140 bar.

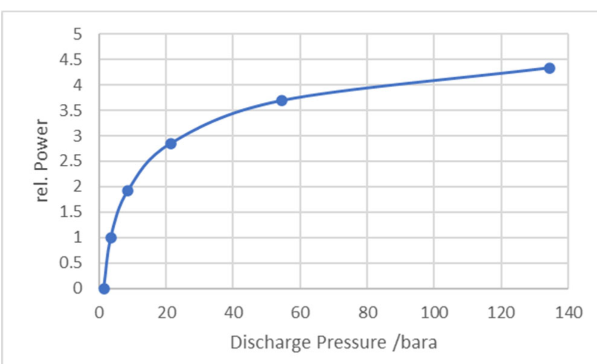


Figure 18: Compression Power for Carbon capture compression (Suction pressure 1.3 bar).

Transporting CO_2 in dense phase conditions has a number of advantages, since it combines a high density with a viscosity that is in the same range as a gas, rather than that of a liquid. Supercritical CO_2 is compressible, so increased pressure leads to increased density, and like a gas, it fills any available volume. Unlike gas in the subcritical range, there is no phase change when it is cooled at constant pressure. Existing CO_2 pipelines operate at pressures between the critical point and up to 150 bar to maintain single phase operations in the dense phase region [11]. The upper pressure limit is based on the mean allowable operating pressure and maximum allowable pressure as determined by the physical pipeline design. Pressure will vary along the length of the pipeline due to viscous, or pipe friction losses, change in elevation, and thermodynamic effects associated with changes in temperature. Pressure changes due to environmental thermal input and elevation profile can be significant for CO_2 pipelines. In addition to thermal flux from the environment, isenthalpic effects associated with a sudden pressure drop, such as across a throttling valve, can cause localized temperature changes. Care must be taken to open valves slowly in order to minimize local thermal gradients which can result in phase changes and transients within the piping system. If a phase change were to occur at a valve, significant damage could result from the high gas flow velocities associated with the liquid to gas phase transition [13]. Due to these features, both pumps and compressors can be used as boosters for pipelines. Compressors would run relatively slow, but significantly faster than a pump. The required pressure ratio is low. Booster stations will be placed about 150 to 250 km apart, thus requiring a pressure ratio of about 1.5 to 1.7.

At an elevated pressure above the critical pressure, CO_2 can be transported in a dense phase state (Figure 16) using electric driven pumps or compressors that maintain pressure and temperature above its critical point. This strategy of pumping or compressing the CO_2 in dense phase versus compressing CO_2 in gas phase compressors is often dictated by the volume flow rate of the pipeline and the distance from the source to the sequestration site . For smaller volumes or shorter distances, compression in the gaseous phase is preferable because the overall power consumption will be less since pressurization above 140 bar (2200 psi) is not required. However, for greater volumes of CO_2, or longer distances, it is worth the cost of initially compressing the CO_2 to the dense phase state since it can then be transported at overall lower power costs. Figure 19 illustrates this breakover point, where the overall power consumption between supercritical state and gas phase (y-axis) will be less for the supercritical pumps, at three diameters of 20", 24" and 30" lines and for flow rates between 100-700 mmscfd (x-axis). For example, for a 24" CO_2 pipeline (middle orange line), the power spent in compressing up to the supercritical state and then pumping CO_2 is less than the gaseous compression power for flow rates of 300 mmscfd or more.

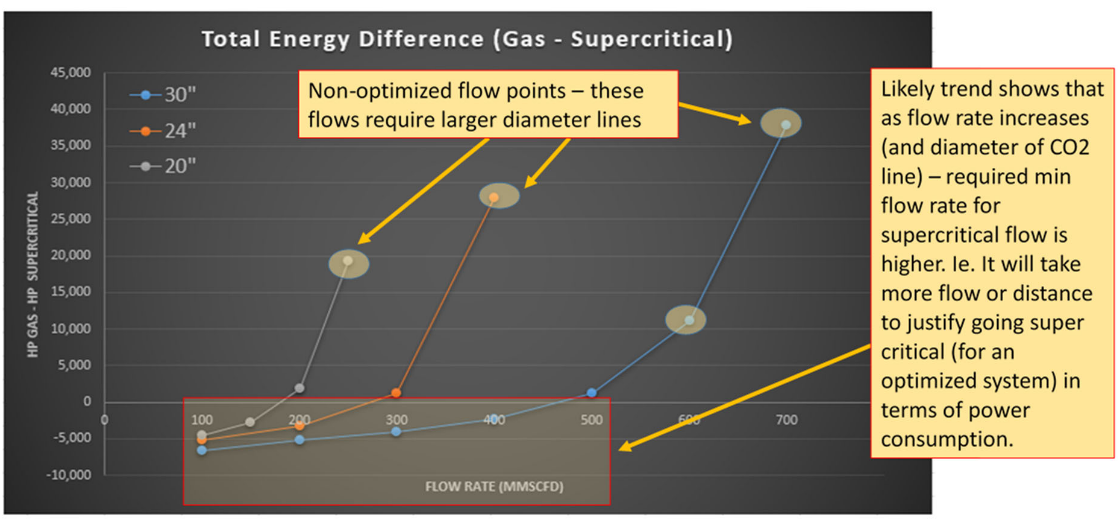


Figure 19: CO_2 Transport as a gas or in supercritical/dense phase state.

It should also be noted that the pipeline flows should be optimized to diameter such that a larger diameter choice will incur less pressure drop and keep the power consumption lower. Once the power consumption difference starts to increase exponentially (Figure 19), a pipeline designer would choose to increase the pipeline diameter to incur a lower power demand, in either the gas phase or supercritical state. Finally, temperature effects should also be taken into account since the dense phase CO_2 transport will be more sensitive to temperature changes, especially for the pipeline route ground temperatures. A CO_2 pipeline optimization study should consider these temperature effects to assure that the pumps can maintain pipeline pressures, given the expected fluid temperatures and mixture composition.

Another consideration is that transporting CO_2 in dense phase will not allow the re-use of existing natural gas pipelines, which are typically rated for approximately 1500psi (105 bar) MAOP. While 1500psi (105 bar) is still above the critical pressure, operability concerns (for example the large change in volume for relatively small changes in temperature and the risk of forming liquids when temperatures drop, may prevent operation in that range. The option then becomes to reduce the operating pressure even further, ie to 400 to 600 psi (28 to 41 bar) MAOP, because at higher pressures, changes in ambient temperature affect gas temperatures, which may lead to the formation of liquid CO_2 (Figure 16).

While CO2 transport in dense phase requires the least amount of energy, lower pressures may be considered, even though the power consumption for the transport will increase. Lower pressure pipelines are considered for 2 reasons: (1) The transport is for a relatively short distance or (2) Existing pipelines with a lower pressure rating (for example natural gas pipelines with a typical pressure rating of about 105 bara) are to be used, to avoid the requirement for new pipeline construction [12].

Since the CO2 to be transported will not be pure, the impact of impurities must be considered. For a given pressure drop, and a given pipe diameter, and compared to the transport of pure CO2, typical oxyfuel would reduce the pipeline capacity by 25%, while CO2 with 5% hydrogen content leads to a reduction of 11.5%, CO2 with 5% Nitrogen or 5% CO leads to a 6% reduction. Other impurities show little impact on the flow capacity [11].

5- POWER GENERATION: BLUE HYDROGEN OR CARBON CAPTURE?

In the context of carbon reduction for natural gas fired power plants, there are two fundamentally different approaches: Capturing carbon from the exhaust of the gas turbine or providing a fuel that does not contain carbon [5,12]. To generate a certain amount electricity in a gas turbine plant carbon-free, the choices are [5]:

1. Feed natural gas to make blue hydrogen, compress the hydrogen to combustor pressure, capture the CO2 created in the process, and compress is to a pipeline or sequestration pressure
2. Burn natural gas in a gas turbine, capture the CO2 in the exhaust and compress the CO2 to pipeline or sequestration pressure.

This raises the question: If the CO2 stream, the methane stream and the hydrogen stream must be transported, what is the relative energy consumption? With the following assumptions for the gas streams to and from a 1GW power plant with 50% thermal efficiency we can compare the options.

Lubomirsky et al. [14] performed pipeline simulations comparing a realistically sized hydrogen pipeline, an actually operating CO_2 pipeline, and a realistically sized natural gas pipeline. The natural gas and the hydrogen pipeline were assumed for a operating pressure of 100 bar, while 152 bar operating pressure were used for the CO_2 pipeline to stay in the supercritical region.

If a CO_2 stream, a natural gas stream and a hydrogen stream have to be transported, what is the relative energy consumption? For the evaluation, the natural gas and the hydrogen pipelines were assumed for an operating pressure of 100 bar, while 155 bar operating pressure was used for the CO_2 pipeline to stay in the supercritical region. As stated earlier, the transport of CO_2 is most economic if it can be done in its supercritical or dense phase state, both due to the high density, but also due to the low amount of work needed to achieve the necessary pressure rise at a booster station. On the negative side of this approach is the requirement for initial compression at the header station, and the fact that the pipeline has to be designed for a high pressure, typically in the range of 140 to 155 bar (2000 to 2250 psi) operating pressure. The distance between booster stations is dictated by the requirement to stay above approximately 90 bar (1300psi), to avoid entering the two-phase region [5].
For shorter distances, especially when the injection pressure for sequestration is relatively low, transport of CO_2 in the gas phase, at pressures below 105 bar is possible, and would allow the use of existing natural gas pipelines. However, the required compression power is significantly higher. In many

instances, especially if within a region multiple sources of CO_2 need to be accommodated, the creation of a CO_2 hub could be considered. From this hub, CO_2 from the entire region can be compressed to an injection pressure.

A 1 GW gas turbine combined cycle power plant will produce about (350 t/h) 97kg/s, or 162 MMSCFD of CO2 [13]. It will consume approximately 40 kg/s of natural gas (180 MMSCFD). On the other hand, a 1 GW hydrogen fired power plant uses about 16.7 kg/s (60,000 kg/h) or 600 MMSCFD of hydrogen. In the discussion of hydrogen and CO_2 applications, the question is whether it is more appropriate to transport hydrogen to a power station, or to run the power station with natural gas, and transport the captured CO2 to a sequestration site.

To cover the discussion, a CO_2 pipeline, a natural gas pipeline, and a hydrogen pipeline have been modelled using commercially available pipeline simulation software, and described in [14]. In all cases, station spacing(125 miles) and overall length(500 miles) were kept the same. The pipe diameter was adapted to achieve realistic flow velocities. In the case of the CO_2 pipeline, the flow velocity was calculated from an existing pipeline. The flow velocity in the pipelines will be significantly different, the hydrogen pipeline with about 12m/s (40 ft/s), the CO_2 pipeline at about 2 m/s (6.5 ft/s), and the natural gas pipeline at 7m/s (23.5ft/s). Only the CO_2 pipeline is operated in the dense phase region at about 2200 psi. Based on the results of the pipeline simulation, the flow is then, based on keeping the flow velocity the same (which would lead to different pipeline diameters, adjusted to the flow that would be the result of the fuel consumption (with hydrogen or natural gas), and the CO_2 production (if natural gas is used as fuel) for a 1 GW gas turbine power plant.

The simulations include:

1-A dense phase CO_2 pipeline with 32'' nominal diameter, 3 compressor stations equally spaced, transporting about 766 kg/s or 1300 MMSCFD, for a distance of 500 miles (800 km). The operating pressure is 2200 psig, and the inlet pressure for the compressor station is kept at above 1320 psig to avoid two-phase flow at all prevailing ambient temperatures. The calculations shown are for 37.8 ^oC (100 ^oF) gas temperature at the head station discharge.

2- A hydrogen pipeline for the same distance (500 miles,800km), 1440 psig operating pressure, transporting 1000 MMSCFD, also with 3 compressor stations equally spaced.

3- A 500miles (800km) natural gas pipeline, for 700 MMSCFD, 1440 psig operating pressure, gas with SG=0.58.

It should be noted that the power consumption of the header station is very large relative to the pipeline stations both for the CO_2 and hydrogen cases. This power consumption is very sensitive to the suction pressure, which could be as low as atmospheric pressure for CO_2, or as low as 5 bar for hydrogen (Figures 6). The results are summarized in Table 2.

Table 2: Summarized results for the pipeline simulations

Gas	Flow (MMSCFD)	Station PR	Head (ft)	Power (hp)
Hydrogen	1000	1.21	79,928	10666
Natural gas	700	1.45	17,000	13300
CO2	1300	1.65	2,191	8298

The head and power requirements outlined in Table 3 would require a 7 to 8 stage centrifugal compressor, with 585 mm (23 in) diameter impellers for hydrogen, while the CO_2 application can be covered with a single stage compressor with a 380mm (15 in) impeller. The natural gas compressor would require a single impeller or two impellers, typical diameter would be 530 to 560mm (21 to 22 in).

In Table 3, the data is shown by scaling the pipeline to the flow required for a 1 GW power plant, to be able to compare power demands. The pipelines themselves would probably be too small to be economic. This was the reason why the pipeline flows in Table 2 were used in [14].

The data in Table 3 shows the significant difference in consumed power in these cases. It becomes clear that for a power plant of a given size, transporting hydrogen to the plant requires more energy than the transport of natural gas to the plant, and the transport of CO_2 from the plant. Table 3 indicates that even if the power for the Natural gas and the CO_2 pipeline are combined, they are still significantly lower than the power for the hydrogen transport. Bringing natural gas to a power plant and transporting the generated CO_2 to a sequestration site is more energy efficient than transporting hydrogen, generated elsewhere over larger distances.

Table 3: Summarized results for the pipeline simulations, corrected to 1 GW plant flow

Gas	Flow (MMSCFD)	Station PR	head	power
Hydrogen	600	1.21	79,928	6400
Natural gas	180	1.45	17,000	3420
CO2	162	1.65	2,191	1034

This raises the question: If the CO_2 stream, the Natural Gas stream and the Hydrogen stream have to be transported what is the relative energy consumption. In Figure 20, the power requirements for identical transportation distances are compared to answer that question.

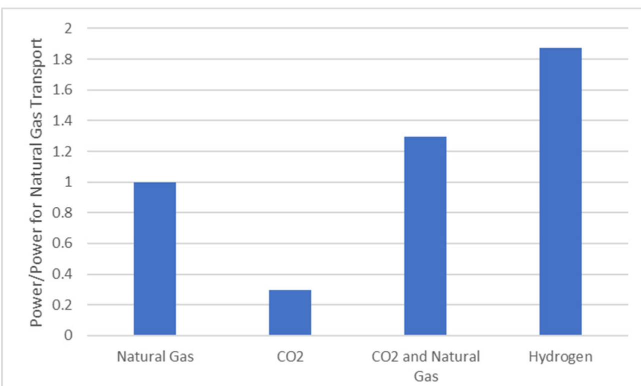


Figure 20: Relative power requirement for transportation of CO_2 and Hydrogen, compared to transportation of natural gas [14].

The key finding (Figure 20) is, that hydrogen transport requires the most energy to transport. CO_2 transport in the supercritical state requires only a small fraction of the energy needed for hydrogen or natural gas. This means that bringing natural gas a power plant and transporting the generated CO_2 to a sequestration site is more energy efficient than transporting hydrogen, generated elsewhere over larger distances. The calculations do not account for the compression requirements at the head station for the hydrogen and natural gas pipeline, nor the compression requirements to bring CO_2 from capture pressure to pipeline pressure. These compression requirements can vary over a large range.

The power required for transportation of the different gases involved is important for the discussion on the placement of blue or green hydrogen generation relative to the point of usage. It is evident that the long distance transport of pure hydrogen is rather unattractive, while the energy consumption for CO_2 (either from combustion of Methane or from the SMR) is very small. This suggests that hydrogen production should be close to the power plant.

For green hydrogen, the transportation method would be determined from the hydrogen usage at the receiving point, and from the purpose of hydrogen generation. Hydrogen can be used as fuel for processes that require a high energy density fuel, or require combustion, such as steel making, but it also can be used for storing large amounts of energy to balance renewable energy production and demand. The latter, which can be realized either in pure hydrogen storage and transport, or by mixing hydrogen in a natural gas pipeline, is thus an attractive method to balance the intermitted nature of renewable energy supply.

5. STORAGE

Storage of gases in geological formations is well established for natural gas storage, and under discussion for CO_2 sequestration as well as hydrogen storage [15,16]. Concerns include the question whether formations that are gas tight for long duration storage.

Depleted natural gas or oil fields, often close to consumption centers are frequently used for storage. Conversion of a field from production to storage duty takes advantage of existing wells, gathering systems, and pipeline connections. Depleted oil and gas reservoirs are currently the most commonly used underground storage sites for natural gas due to their wide availability. Natural aquifers have also been converted to gas storage reservoirs. An aquifer is suitable for gas storage if the water bearing sedimentary rock formation is overlaid with an impermeable cap rock. While the geology of aquifers is similar to depleted production fields, their use in gas storage usually requires more base (cushion) gas and greater monitoring of withdrawal and injection performance.
Salt caverns provide very high withdrawal and injection rates relative to their working gas capacity. Cavern construction is more costly than depleted field conversions when measured on the basis of cost per working gas capacity. Their ability to perform several withdrawal and injection cycles each year reduces the per-unit cost of each thousand cubic feet of gas injected and withdrawn, but this advantage is only relevant for natural gas or hydrogen storage.

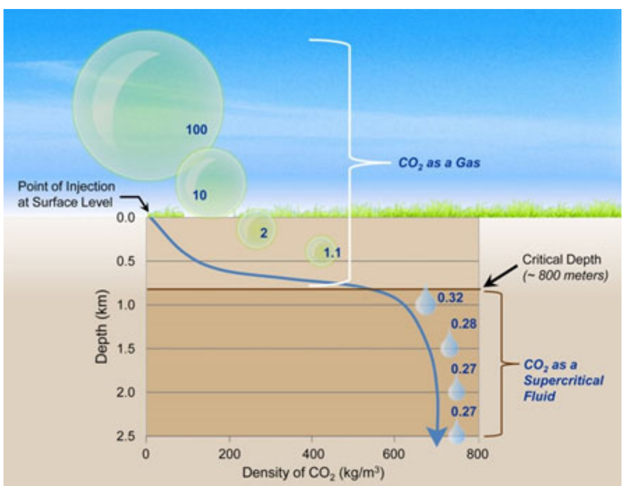


Figure 21: CO_2 density as a function of storage depth. CO_2 reaches supercritical pressure and volume at about 800 m depth.[19]

There also have been efforts to use abandoned mines to store natural gas, with at least one such facility having been in use in the United States in the past. Further, the potential for commercial use of hard-rock cavern storage is currently undergoing testing. None are commercially operational as natural gas storage sites at the present time.

The storage pressures would be in the range of 100 to 200 bar, or higher, thus requiring significant amounts of compression. Compression to inject the CO_2 into the storage facility brings the CO_2 from delivery pressure (20 to 150 bar) to injection pressure. Depending on the type of storage facility, the required injection pressure could be 200 bar (3000 psi) or less in abandoned gas fields, below about 250 bar (3750 psi) in aquifers. Injection for enhanced oil recovery (EOR) could be 200 to 400 bar or higher (Figure 21) The injection flow rates are to some extent determined by facility limitations (for example, erosion limits).

CONCLUSIONS

The paper describes a number of concepts to reduce the carbon footprint in the energy industry, using gas turbines while retaining energy supply security. Relative cost factors have been considered.
Dual drive concepts can contribute to energy security, and to a reduced energy footprint in situations where electricity is available, but may only be available intermittently.

Exhaust gas recirculation, and its positive impact on the viability of carbon sequestration has been described. This is very important in the discussion on blue hydrogen versus carbon capture when operating gas turbines. Generally, Steam Methane Reforming (SMR) to create blue hydrogen will lower the efficiency of the gas turbine engine system and will produce more CO_2 emissions to dispose of due to the inefficiency of converting NG to H_2. The capital expense for a carbon capture technology has a strong impact to determine if it is a good investment to offset carbon pricing over the long term.

Whether the source of CO2 is from making blue hydrogen, or from exhaust carbon capture, the transport is energy intensive. Transport in the dense phase state versus gas phase transport will likely depend on the transportation distances, and the desire to re-use existing natural gas pipelines. General transport considerations will also favor the long distance transport of Natural gas and CO2 over the long distance transport of hydrogen, unless the use of pipelines can be advantageous for energy storage purposes.

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