Process knowledge based opportunistic optimization of the N₂—CO₂ expander cycle for the economic development of stranded offshore fields

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1. Introduction

When the world is looking for alternative clean energy sources, natural gas still holds the key position as the major player in the energy sector (Regulagadda et al., 2010), primarily because of its clean burning and ability to meet tough environmental regulations. The ever growing use of natural gas (NG) ranges from the chemical heating, cooling and electricity generation (Adouane et al., 2010) (Sen et al., 2010). Under the current scenario, there are strong predictions for a significant increase in the demand of NG (DOE/EIA, April). The shell is even expecting the global NG demand to increase by 60% from 2010 to 2030, reaching 25% of the global primary energy (Press Release, December 2012). NG is often found in remote and stranded locations and must be brought to the world market for trading. Pipeline transportation often has many technical and political challenges, particularly when the gas is transported between continents. Natural gas liquefaction is another safe and economic way of bringing natural gas to the potential market. To carry out liquefaction, several proven technologies are available for base-load NG plants with different capacities and complexity (Shukri, February 2004). A pure refrigerant cascade, and mixed refrigerant (MR) cascade are a broad classification of a few proven onshore technologies (Lim et al., 2013). Most of the technological developments have been made considering the operation of onshore plants. Virtually no serious improvements in offshore technology occurred until the late nineties. With the maturity and advancement of offshore technology (for oil recovery), coupled with favorable market conditions, resulted in the evolution of floating liquefied natural gas (FLNG) unit or LNG floating production storage and offloading (FPSO) unit. Since its inception, the prospects for the floating LNG industry have brightened with the announcement of the joint venture of Shell and Samsung in constructing multiple FLNGs (Press Release, December 2012). The world’s first operational FLNG unit is expected by Petronas Malaysia in 2015 designed for Kanowit gas field off Sarawak Malaysia, at the same time Prelude FLNG designed by Shell with the help of Samsung and Technip for offshore gas field in West Australia is expected to be operational in 2017 (Media releases and Shell, October 2012). Up until now, no such facility exists and FLNG remains commercially untested. Therefore, considerable work needs to be done to make FLNG commercially viable.

The aim of this paper was to fill the information gap in the literature by providing technical details for the process knowledge-
based design optimization (Khan et al., 2013) of carbon dioxide and nitrogen expander technology for (N2–CO2) NG liquefaction. As mentioned already, there are several commercial liquefaction technologies for onshore plants but only few studies have been performed on offshore plants that can satisfy the specific constraints of offshore plants, most importantly plant safety. Therefore, the little information in the open literature motivated this study of the CO2–N2 expander cycles for NG liquefaction.

Liquefaction is the heart of a NG value chain and is an energy intensive process; slight improvement in the liquefaction performance will have huge economic benefits. Liquefaction involves rejecting the heat from NG to the ambient using a refrigerant, analysis of whole plant can point out windows of opportunity for possible improvements in energy savings. This also highlights the process characteristics, fundamental design flaws and thermodynamic limitations of the process.

In this study, the CO2–N2 based expander cycle was analyzed and optimized for the compression energy requirement by identifying the key decision variables. A Honeywell UniSim Design® process plant simulator was used for rigorous process modeling, in-depth process analysis was performed and the characteristics of the main decision variables with the inherent process limitations and constraints were determined. Finally a systematic way of optimizing CO2–N2 based expander cycle was proposed.

The Honeywell UniSim design software used in this study is a commercial process plant simulator and due to its strong thermodynamic libraries the process model can be developed rigorously in less time. This simulator has macro engine and supports automation using windows COM functionality. Thus the proposed optimization methodology can be exploited using Matlab which has object oriented programming capabilities. The simulation optimization methodology using particle swarm optimization algorithm is already applied by Khan et al. and can be referred for details (Khan and Lee, 2013).

2. Beginning of turbo expander device and expander technology for NG liquefaction

Until the 1960s, the Joule Thompson (JT) expansion valve was the only way to the cool gas plant streams by a pressure drop. Herrin in 1966 proposed the first turbo expander gas processing plant. Kinday et al. (2006) reported that the reversible expander cycle provides two distinct improvements over the JT valve expansion. First in the reversible expansion, a large fraction of the work required to compress the gas can be recovered and used elsewhere in the cycle, thereby increasing the cycle efficiency. Secondly, the reversible process results in a much larger cooling effect. Expander processes are simple in operation and provide several benefits compared to mixed refrigerant (MR) processes. The phase of the refrigerant is not changed, the configuration is simple, and the equipment count is low. The only disadvantage compared to MR technologies is the higher compressor energy demand (Barclay et al., 2005; Oct: 34–36). The expander cycle often uses N2 as a refrigerant which is inexpensive, non-toxic, non-flammable, and suitable for offshore applications. The expander cycle is used widely in onshore peak shaving units and is proposed for offshore liquefaction plants (FPSOs) (Mokhateb et al., 2008). The simplest type of expander process employs nitrogen as a refrigerant and a single expander, an example of which is in use at Snurrevarden in Norway (Thorsager, March 2009). This is a simple process with the shortcoming that the entire refrigerant must expand to the lowest temperature, even though most of it is needed at higher temperatures. Therefore, the temperature difference between the low temperature end inside heat exchanger is large, and the compressor work is high. Several studies related to turbo-expander cycle development have been published.

Exergy is the useful form of energy obtained from a system when it is reversibly brought into a state of thermodynamic equilibrium with the environment. Exergy gives an idea of deviation of a system from ideal state and enables determination of types and true magnitude of losses. Thus this concept is borrowed to compare the liquefaction efficiency of turbo-expander and mixed refrigerant processes. The overall exergy efficiency of turbo-expander based process taking into account the exergy loss in the aftercooler and compressor is lower than that of mixed refrigerant processes (Venkataraman and 1st ed., 2008). The large exergy loss is essentially due to a large temperature approach between the hot and cold fluid streams. A small temperature approach all along the length of the heat exchanger can never be achieved when a single-component refrigerant is used for precooling, condensation and sub-cooling of NG feed since the specific heat capacity is not same in all these cooling regions. The exergy loss is lower when a mixture of refrigerant is used that achieved slightly closer temperature approach in the heat exchanger. However, the other benefits of turbo-expander based process over mixed refrigerant process like inherent process safety, simplicity of operation and small hydrocarbon inventory make turbo-expander based process preferable for offshore platforms.

Knut et al. (Knut and Neeras) compared a N2 single expander, N2 dual expander and N2-methane (CH4) expander. The comparison results showed that a N2 single expander cycle has the highest specific energy requirement followed by a Dual-N2 expander cycle (Dubar and Process, 1997), whereas the N2–CH4 expander cycle depicts the lowest specific energy requirement. The reason for the high efficiency of N2–CH4 was attributed to the higher specific heat capacity of methane (Foglietta, 2002). Although the N2 expander cycles (single and dual) consumes higher energy and has poorer economic efficiency, they are by far the most suitable offshore liquefaction technologies because of their inherent safety in using a non-flammable refrigerant. Nitrogen expander cycles are also compact, requiring less deck area, and are less sensitive to vessel motion, rapid shut down and startup (Finn and July, 2009), and involve easier operation (Li and Ju, 2010). Inspired by the high efficiency of the N2–CH4 cycle, a new cycle based totally on a non-flammable refrigerant, N2–CO2, is being studied. CO2 is a substitute of methane as potential refrigerant with favorable environmental properties. The specific refrigeration effect of methane is higher than CO2 but the superiority of CO2 as refrigerant in meeting process safety and emission regulation makes it a suitable alternate for offshore NG liquefaction plant. With no ozone depletion and negligible global warming potential the process designed with CO2 is environmental friendly. Small inventory size of CO2 requires small footprint and provides quick start-up, ramp-up and down capacity. With all these attributes the N2–CO2 cycle is preferred over N2–CH4 for compact offshore operations (Table 1).

To the best of author’s knowledge, there is no information in the open literature related to the design optimization of the N2–CO2 expander cycle. Therefore, to fill the information gap in offshore

<table>
<thead>
<tr>
<th>Refrigerant and properties</th>
<th>Methane (CH4)</th>
<th>Carbon dioxide (CO2)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ozone depletion potential</td>
<td>Yes</td>
<td>No</td>
</tr>
<tr>
<td>Global warming potential</td>
<td>56</td>
<td>1</td>
</tr>
<tr>
<td>(Time horizon 20 years)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Flammable or explosive</td>
<td>Highly flammable</td>
<td>No</td>
</tr>
<tr>
<td>Inventory size</td>
<td>Big</td>
<td>Small</td>
</tr>
<tr>
<td>Process safely</td>
<td>Low</td>
<td>High</td>
</tr>
</tbody>
</table>

Table 1: Comparison benefits of methane over carbon dioxide as potential refrigerant.
technologies, this paper analyzes and optimized the \( \text{N}_2 - \text{CO}_2 \) expander cycle.

### 3. Turbo-expander system working as a refrigerator

Carl Wilhelm first introduced the idea of an expansion device to produce low temperatures in 1857 (Siemens, 1857). Since its first introduction, expander devices have found wide applications in many fields, ranging from air separation, liquefaction of natural gas to gas purification and power generation (Bloch et al., 2001). In the later stages, the expander device was coupled with a compressor to utilize the work of expansion, which was previously wasted. The working of the turbo-expander device acting as a refrigerator is explained in Fig. 1, where the gas at high pressure is expanded isentropically to a low pressure and the work of expansion is used in gas compression. The turbo-expander system consists primarily of conventional compressors, expander, heat exchanger and intercoolers for rejecting heat to the ambient. When high pressure gas (HP Gas in, Fig. 1) is expanded to a low pressure isentropically, the work of expansion is transferred at the other end by the shaft coupled with the compressor.

Owing to the developments in expander technologies, the compressor (skid mounted on the brake end of the turbo expander) polytropic efficiencies have approached 85% (Lillard et al., 2004). To calculate the efficiency of a turbo-expander working as a refrigerator, the compressor and aftercooler were considered to be operating isothermally at a temperature, \( T_2 \), with an efficiency \( \eta_c \). The working fluid is assumed to be a perfect gas and the amount of heat, \( Q_e \), is removed and perfectly delivered to the compressor at temperature \( T_1 \) by the turbo-expander. The ideal work obtained from the expander would then be

\[
W_e = \frac{Q_e}{\eta_c} \tag{1}
\]

From the compressor work definition, the actual compressor work with efficiency \( \eta_c \) would be:

\[
W_c = \frac{Q_e}{\eta_c \eta_t} \times \frac{T_2}{T_1} \tag{2}
\]

The actual mechanical work required (\( W \)) by the compressor is equal to:

\[
W = W_c - Q_e \tag{3}
\]

From the second law, the theoretical work needed to cool the gas was calculated using Eq. (4).

\[
W_{\text{theor}} = Q_e \frac{T_2 - T_1}{T_1} \tag{4}
\]

The second law efficiency of the turbo-expander system working as a refrigerator was obtained by combining Eqs. (3) and (4)

\[
\gamma = \frac{W_{\text{theor}}}{W} = 1 - \frac{Q_e}{T_2(1 - \eta_c \eta_t)} \tag{5}
\]

The mechanical efficiency of the turbo-expander system was assumed to be 100% based on the assumption of no drag force between the expander shaft and its casing. Driven mainly by the gas processing and petrochemical industry, turbo-expanders have achieved superior mechanical integrity and thermodynamic efficiency. During early developments of turbo-expanders the oil lubricating shaft was used to deliver expansion energy but the adoption of magnetic bearing systems reduced the drag force between shaft and casing to almost zero. Thus the assumption of 100% mechanical efficiency is based on normal industrial experience. The implied result is the overall increase of expander isentropic efficiencies to 90% and compressor (break end) polytropic efficiencies to 85% (Knut, Neeraas: Dubar and Process, 1997). In this study, however, much conservative figure of 75% isentropic efficiency was used for calculations.

The ideal gas assumption for calculating refrigeration efficiency was made out of convenience. In more elaborated sense the assumptions underline three facts about the gas used in refrigeration equipment i) thermodynamic equilibrium of gas, ii) gas not chemically reacting, iii) internal energy, enthalpy and specific heat are the function of temperature only and not pressure. In compressor assemblies the temperature fluctuations are usually not large enough to cause any significant deviations from the thermally perfect gas model. Heat capacity is still allowed to vary though only with temperature, and molecules are not permitted to dissociate. Thus, ideal gas assumptions hold no serious problem in approximating the behavior of turbo-expander system conclusively in calculating refrigeration efficiency.

Turbo-expanders are used widely as the sources of refrigeration in industrial processes, such as the extraction of ethane and natural gas liquids (NGLs) from natural gas, the liquefaction of gases (such as oxygen, nitrogen, helium, argon and krypton) and other low-temperature processes. Depending on the operating conditions, the turbo expander reduces the load on the electric motor by 6–15% compared to the conventional vapor-compression refrigeration system that uses a throttling expansion valve rather than a turbo expander (Joost, September 2000).

### 4. Feed conditions and simulation basis

NG received at the well head contains methane and a range of heavier hydrocarbons. Depending on the heavier hydrocarbon composition, feed gas is classified as lean or rich gas. Fractionation is required after the pretreatment step to recover those heavier HC liquids commonly referred as NGL (LPG, C3/C4 and condensate C5+). NGLs are removed to maintain the product dew points and yield a source of revenue. NGL has significantly greater value as separate products in their own right. The lighter NLG fractions, such as ethane, propane, and butane can be sold as feedstock to refineries, whereas the heavier hydrocarbons can be used as gasoline blending stock. Thus with these benefits it is assumed that the heavy hydrocarbons are stripped off from feed NG and lean methane composition (91%) NG illustrated in Table 2 is used as a feed to the liquefaction plant. The techno-economic study related
to the variation on feed gas composition for refrigeration plant is performed by Getu et al. (2013, 2012) and can be referred for further information.

The Peng–Robinson equation of state was used to calculate the thermodynamic properties. Approximately 8% of the natural gas was assumed to have been converted to boil off gas (BOG) after the sub-cooled LNG was flashed to atmospheric pressure. From Table 2 it is clear that the feed gas must be cooled from 32 °C to −158.5 °C at 50 bar for flash expansion to LNG (8% BOG) storage at atmospheric pressure. To determine the cooling load, the isobars at 10 bar and 50 bar are plotted on the feed NG temperature–entropy diagram (Fig. 2). Fig. 2 shows that the liquefaction of NG at 50 bar can be classified into three stages: precooling, condensation and sub-cooling. Condensation of the feed NG at 50 bar occurs at around the critical point of the feed gas mixture and the approach temperature inside the heat exchanger at this pressure can be made small by the balanced counter-flow of the refrigerant (Chang et al., 2011). All refrigeration stages should be separately or at least pre-cooling must be done separately to save the expensive utility. Therefore, in this paper, the precooling of NG was carried out using CO2 cycle until −46 °C. The condensation and sub-cooling takes place inside the LNG exchanger using the N2 cycle.

5. N2–CO2 single expander process description

The N2–CO2 expander process is a simple refrigeration cycle, where two separate loops of refrigerants render the task of NG liquefaction. Fig. 3 shows a schematic diagram of the N2–CO2 expander process flow sheet. CO2 is used to precool the feed NG and nitrogen to approximately −46 °C. N2 is used to condense and subcool natural gas. CO2 is compressed from 6.8 bar to 72 bar in three compression stages along with intercooling at 30 °C. The compression of nitrogen occurs in three compression stages and the pressure is increased from 6.3 bar to 100 bar. The intercoolers are used to maintain the temperature of nitrogen at 30 °C. During compression, care is taken to maintain a compression ratio at 1:3, which is to prevent the excessive power consumption that causes an increase in process irreversibility (Khan et al., 2012). Fig. 3 shows that pretreated natural gas feed composed mainly of methane (91%) enters the exchanger LNG-100 along with the N2 at ambient temperature of 30 °C. The CO2, which is expanded from 72 bar to 6.8 bar in a Joule–Thompson valve, provides a sufficiently low temperature to precool the feed NG and N2 to −46 °C. The feed NG and N2 then enters exchanger LNG-101. N2 is expanded from 99 bar to 6.4 bar, creating a cold stream of N2, exchanging heat with warm N2 and NG, and provides sufficient cooling for NG to preclean from −46 °C to −149 °C. NG is still in a high pressured state, after leaving LNG-101 exchanger. NG is then flashed to atmospheric pressure and its temperature decreases to −158.5 °C, which is sufficient to liquify 92% of NG at atmospheric pressure and 8% NG leaves as BOG. Owing to the non-flammable nature of the working refrigerant, this process is suitable for offshore purposes and enhancing their safety measures. Different flow rates of N2 and CO2 required producing various quantities of LNG is illustrated in Table 3. Note that Table 3 shows the general trend of refrigerant requirement at fixed pressure of N2 and CO2 levels and does not represent the optimum flow rates.

5.1. Decision variables and efficiency improvement analysis in the N2–CO2 expander process

Fig. 3 outlines the N2–CO2 expander cycle used for the liquefaction of natural gas. The degree of freedom analysis led to the selection of four main decision variables that helps minimize the energy requirement during the liquefaction of NG. The CO2 degree of superheating, operating pressure levels of N2 (condensation and suction pressure) and N2 flow rate are the four main decision variables. The increase in the CO2 degree of superheating decreases the plant compression energy requirement because the demand of the overall CO2 flow rate decreases to provide the same cooling duty, as shown in Fig. 4. The increase in the CO2 degree of superheat has a benefit in the compression energy requirement but the approach temperature in LNG exchanger-101 decreases, which bring the system to an infeasible domain. The effect of the CO2 degree of superheat will be discussed further in detail in Section 7.6.

The condensation and sub-cooling of the feed NG occurs in LNG exchanger-101, where nitrogen working as a refrigerant provides cooling by isenthalpic expansion from a high pressured state. Expanding nitrogen gas from a high pressure provides refrigeration that is used to liquify NG at the expense of the compression energy requirement. When the N2 discharge pressure from the compressor assembly is fixed to a higher value (e.g. 100 bar), the overall operating pressure difference between the compressor cooler assemblies can be varied by varying the N2 suction pressure, as shown in Fig. 5.

The compression power decreases monotonously with increasing N2 suction pressure. This is because the overall operating pressure difference between compressor ends decreases. When the suction pressure is 25 bar, restoring this pressure to a 100 bar

Table 2

<table>
<thead>
<tr>
<th>Property</th>
<th>Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Property calculation</td>
<td>Peng–Robinson, Lee–Kesler (for Enthalpy)</td>
</tr>
<tr>
<td>NG temperature</td>
<td>32 °C</td>
</tr>
<tr>
<td>NG pressure</td>
<td>50 bar</td>
</tr>
<tr>
<td>NG mass flow rate</td>
<td>1.0 kg/h (the unit flow rate was used to calculate the specific compression power out of convenience)</td>
</tr>
<tr>
<td>NG composition (mole fraction)</td>
<td></td>
</tr>
<tr>
<td>Nitrogen</td>
<td>0.0022</td>
</tr>
<tr>
<td>Methane</td>
<td>0.9133</td>
</tr>
<tr>
<td>Ethane</td>
<td>0.0536</td>
</tr>
<tr>
<td>Propane</td>
<td>0.0214</td>
</tr>
<tr>
<td>i-Butane</td>
<td>0.0046</td>
</tr>
<tr>
<td>n-Butane</td>
<td>0.0047</td>
</tr>
<tr>
<td>i-Pentane</td>
<td>0.0001</td>
</tr>
<tr>
<td>n-Pentane</td>
<td>0.0001</td>
</tr>
</tbody>
</table>
discharge pressure obviously requires less power compared to restoration from less than 25 bar. The suction pressure also alters the heat transfer driving force and prevents the occurrence of a pinch zone in the LNG exchanger at the expense of the compression energy requirement. An optimal N₂ suction pressure that corresponds to minimum compression power with maximum heat transfer driving force in the heat exchanger exists and will be determined in the next Section 7.4.

The increase in N₂ flow rate increases the compression energy demand monotonically (Fig. 6). With a low N₂ flow rate, the refrigeration demand cannot be met in the LNG exchanger and a pinch zone occurs at the cold end of the heat exchanger. To avoid the pinch zone, the minimum flow rate of N₂ is sought, which provides a sufficient heat transfer driving force to carry out the liquefaction of NG with minimum compression energy. The effect of the N₂ flow rate will be discussed in detail in Section 7.5.

6. Optimization objective and process constraints

After analyzing the trends in the decision variables, a knowledge based methodology was proposed to determine the decision variables that provide the minimum compression energy requirements for NG liquefaction. Minimization of the total compression shaft work defined by Eq. (6) (Nitrogen and CO₂ compressor demand) is selected as the optimization objective in the present study.

$$\min W_{\text{total}} = \sum \left( W_{\text{N₂ compressor}} + W_{\text{CO₂ compressor}} - W_{\text{N₂ Expander}} \right)$$  \hspace{1cm} (6)

where $W_{\text{N₂ compressor}}$ and $W_{\text{CO₂ compressor}}$ are the compression energy demand by separate N₂ and CO₂ compressors, respectively. The energy received from the expander is assumed to have been used elsewhere in the plant for partial recompression of the refrigerant. Therefore, the total compression energy demand is reduced by the energy supplied from the expander, as expressed in Eq. (6).

To yield meaningful results and a practical solution from the optimization problem, the unique constraints posed by the N₂–CO₂ process must be satisfied. Some constraints are posed by the hardware like the highest and lowest operating pressure in the process plant and others are related to the physical nature of the process. Finding the optimal solution that satisfies all the constraints requires good engineering judgments coupled with robust optimization schemes. Relying merely on an optimization algorithm with poor process understanding is unlikely to yield better results, therefore process understanding is important. To develop process understanding the process constraints are described individually in the following section.

6.1. Compressor model: constraints and efficiencies

The compressor isentropic efficiency in the simulation and the mechanical efficiency were assumed to be 75% and 100%, respectively, based on normal industrial experience (Wang et al., 2012). The compression task was assumed to have been performed in stages. The compression work on every stage was fixed equal to the specific enthalpy change in the refrigerant multiplied by its mass flow rate (Wang et al., 2011). The work consumption at each stage of the CO₂ and N₂ cycle was calculated using Eqs. (7) and (8)

$$W_{\text{N₂}}, i = f_{\text{N₂}} \Delta H_{\text{N₂}, i} \quad i = 1, 2, 3$$  \hspace{1cm} (7)

$$W_{\text{CO₂}}, i = f_{\text{CO₂}} \Delta H_{\text{CO₂}, i} \quad i = 1, 2, 3$$  \hspace{1cm} (8)

where $W_{\text{N₂}, i}$ and $W_{\text{CO₂}, i}$ are the required compression work at stage $i$. Apparently, there are three stages in both N₂ and CO₂ compression cycles. $f_{\text{N₂}}$ and $f_{\text{CO₂}}$ are the total mass flow rate of nitrogen and carbon dioxide flowing through the individual compression stage. $\Delta H_{\text{N₂}, i}$ and $\Delta H_{\text{CO₂}, i}$ are the changes in the specific enthalpies of N₂ and CO₂ refrigerant at the corresponding compression stage. The
specific enthalpies involved above are basically a function of the inlet and outlet temperatures and pressure.

Cooling water was employed between the stages. Therefore, the outlet temperature from each compression stage was equal to 30°C. The first stage compression in the CO2 cycle was exceptional, where aftercooler inlet and outlet temperatures were 7.98°C and 13.9°C, respectively, and the cooling provided by the CO2 released elsewhere in the plant was obviated for flow sheet simplification. Once the temperature and pressure values were specified and calculated, the actual required power was then calculated based on the compressor efficiency definition. The following definitions of adiabatic efficiency (isentropic efficiency) and polytropic efficiencies were used for the calculations based on the Schultz method (Schultz, 1962).

\[
\text{Adiabatic efficiency} = \frac{\text{Power Required}_{\text{isentropic}}}{\text{Power Required}_{\text{actual}}} \times 100\% \quad (9)
\]

\[
\text{Polytropic Efficiency} = \left( \frac{P_{\text{out}}}{P_{\text{in}}} \right)^{\frac{n}{k}} - 1 \times \left( \frac{n}{n-1} \times \frac{k-1}{k} \right) \times \text{Adiabatic Efficiency} \quad (10)
\]

\[
n = \frac{\log(P_{\text{out}})}{\log\left(\frac{P_{\text{out}}}{P_{\text{in}}}\right)} \quad (11)
\]

\[
k = \frac{\log(P_{\text{out}})}{\log\left(\frac{P_{\text{out}}}{P_{\text{in}}}\right)} \quad (12)
\]

\[
\text{Power Required}_{\text{actual}} = \text{Heat Flow}_{\text{outlet}} - \text{Heat Flow}_{\text{inlet}} \quad (13)
\]

### 6.2. Compression ratio constraint

The compression ratio is defined as the ratio of the outlet pressure to the inlet pressure, which can vary in the range of 1.5–3.5 (Khan et al., 2012). In the current study, the intermediate compression stage pressure (in both N2 and CO2 cycle) was fixed evenly based on the overall compression ratio and is not an optimizing variable. The overall compression ratio is expressed as Eq. (14)

\[
r_j = \left( \frac{P_{\text{out}}}{P_{\text{in}}} \right)^{1/3} \quad (14)
\]

where \(P_{\text{out}}^j\) and \(P_{\text{in}}^j\) are the inlet discharge and suction pressure from the compression assembly and \(r_j\) is the compression ratio, where \(j \in \{\text{CO2}, \text{N2}\}\).
6.3. Heat transfer feasibility constraints

For a feasible counter current heat transfer operation, the outlet temperature of the hot stream must be higher than the inlet temperature of the cold stream and the same is true for the exchanger hot end. The cryogenic heat exchanger involves a large heat transfer area that assists in bringing a heat transfer driving force to a minimum. Consequently, an approach temperature as low as 2 °C was possible. The minimum temperature difference in this study was assumed to be 3 °C (Khan and Lee, 2013) (Hasan et al., 2009). The heat transfer feasibility constraints could be modeled using equation Eqs. (15) and (16) for the cold and hot end of the exchanger, respectively.

\[
T_{\text{Ex, out}}^\text{hot} - T_{\text{Ex, in}}^\text{cold} \geq 3 \tag{15}
\]

\[
T_{\text{Ex, in}}^\text{hot} - T_{\text{Ex, out}}^\text{cold} \geq 3 \tag{16}
\]

where \(T_{\text{Ex, out}}^\text{hot}\) and \(T_{\text{Ex, in}}^\text{cold}\) are the outlet temperature of the hot stream and inlet temperature of the cold stream from exchanger, respectively, and the corresponding definition for Eq. (16).

6.4. Energy balance constraints

The energy balance equation for the LNG-101 exchanger based on the assumption of a 0.1 bar pressure drop in every pass and is generalized as follows:

\[
m' (h_{\text{in}} - h_{\text{out}}) + Q_{\text{in}} + Q_{\text{out}} = 0 \tag{17}
\]

where \(m'\), \(h_{\text{in}}, h_{\text{out}}, Q_{\text{in}},\) and \(Q_{\text{out}}\) denote the fluid flow rate in the exchanger, specific enthalpy at the inlet and outlet, and heat gained from the surrounding layers and external environment, respectively. For simplicity, the heat losses of the exchanger were ignored in the calculations. The enthalpies of the warm streams (NG and warm refrigerant) were transferred to the cold refrigerant stream. For a feasible energy balance, the following constraints must hold:

\[
f_{\text{N}_2} \Delta H_{\text{N}_2, \text{warm stream}} + f_{\text{NG}} \Delta H_{\text{NG}} - f_{\text{N}_2} \Delta H_{\text{N}_2, \text{cold stream}} = 0 \tag{18}
\]

where \(f_{\text{N}_2}\) and \(f_{\text{NG}}\) are the flow rate of \(\text{N}_2\), feed NG, \(\Delta H_{\text{N}_2, \text{warm stream}}, \) \(\Delta H_{\text{NG}}, \) and \(\Delta H_{\text{N}_2, \text{cold stream}}\) are the changes in the enthalpies of warm \(\text{N}_2\), feed NG and cold \(\text{N}_2\) across the exchanger ends, respectively.

7. Opportunistic optimization scheme

Section 5.1 describes the tradeoff between the decision variables and objective value. Proceeding with that knowledge, the search for the values of the decision variables that gives the minimum compression energy requirement in \(\text{N}_2-\text{CO}_2\) liquefaction cycle was pursued in two steps: i) identifying the operating pressures (suction and discharge) of the \(\text{N}_2\) cycle; and ii) identifying the \(\text{CO}_2\) degree of superheating and the \(\text{N}_2\) flow rate.

7.1. \(\text{N}_2\) cycle suction pressure and feasible range

The discharge pressure from \(\text{N}_2\) compressor was assumed to be 100 bar, which is a general trend in suction pressures that provides a feasible operating range, as illustrated in Fig. 7. At a suction pressure below 6.2 bar, there was no feasible solution and an increase in \(\text{N}_2\) flow rate had almost no effect on the approach temperature. As the \(\text{N}_2\) suction pressure was increased from 6.2 bar, a feasible solution (MITA = 3 °C) was obtained at a \(\text{N}_2\) flow rate of approximately 5.2 kg/h. The continued increase in \(\text{N}_2\) suction pressure brought no further improvements except for an increase in \(\text{N}_2\) flow rate, which eventually increased the compression energy demand (see Fig. 6). Fig. 7 indicates the minimum and maximum operating pressure levels, beyond which there is no feasible solution. Once the feasible operating range of suction pressures is obtained, efforts can be directed to determine its optimal value.

7.2. \(\text{N}_2\) cycle discharge pressure and feasible range

A similar approach was applied to determine the feasible range and highest possible discharge pressure for the \(\text{N}_2\) compressor. Fig. 8 shows that when the discharge pressure is 101 bar, the approach temperature remains below zero and there is no change with increasing \(\text{N}_2\) flow rate. Below this pressure, there are changes in the system feasibility. Therefore, this pressure dictates the highest possible operating pressure that provides feasible results. As the pressure decreases from 101 bar, the demand for \(\text{N}_2\) flow rate increases. This is because when a \(\text{N}_2\) refrigerant expands from a high pressure it creates a low temperature while its expansion from a low pressure does not satisfy the refrigeration requirement, which is in turn compensated for by the increase in refrigerant flow rate. This conflicting effect between \(\text{N}_2\) cycle operating pressures and \(\text{N}_2\) flow rate makes an apparently simple problem complex and demands a systematic approach for optimization.

7.3. Optimization of the \(\text{N}_2\) discharge pressure

The discharge pressure was varied between the feasible range obtained from Fig. 8. The point where the approach temperature became 3 °C was selected as the optimized value for the discharge pressure. Note that the interaction of the discharge pressure within the feasible range on other variables was small enough to carry out non simultaneous optimization of the variables and did not affect the overall system optimality. Fig. 9 shows that the increase in discharge pressure brings the system to a feasible range (MITA = 3 °C), and there is a sharp decrease in the approach temperature once the discharge pressure crosses 100 bar, the reason for which was explained in section 7.1. One point (100 bar) in the discharge pressure line satisfied the design constraint in the current system condition, which was selected as the optimal value.

Fig. 7. Feasible range of \(\text{N}_2\) suction pressures.
7.4. Optimization of N2 suction pressure

The optimal discharge pressure value of 100 bar was obtained from Fig. 9, and the optimal suction pressure was determined using this value. Fixing the discharge pressure to 100 bar, the suction pressure was varied and the point, where the approach temperature constraint satisfies, was selected. The optimal suction pressure obtained was 6.3 bar from Fig. 10. After 3 °C, there was a sharp change in the direction of the discharge pressure, which also decreases the compression energy demand but in an infeasible range and was discarded. The pressure values obtained above were used to determine the optimal CO2 degree of superheating and the nitrogen flow rate in the second optimization step.

7.5. Optimization of N2 flow rate

Considering Figs. 7 and 8, a feasible operating range of N2 flow rates was obtained. Below 5 kg/h, the approach temperature was negative, which is physically unrealizable, and beyond 6 kg/h (both suction and discharge pressure), the approach temperature reached the maximum value, and there was no change with increasing pressure.

The approach temperature vs. N2 flow rate graph for most advantageous N2 suction and discharge pressure are plotted in Fig. 11 to determine the optimal N2 flow rate. The N2 flow rate that satisfies the approach constraint at both pressure levels was selected as the optimum.

7.6. Optimization of CO2 degree of superheat

To determine the optimal CO2 degree of superheating, the values of the decision variables obtained above were fixed to their respective values and CO2 was then varied. The value where approach temperature reached 3 °C was selected as the optimal value. Figs. 12 and 13 show the effect of a variation of CO2 degree of superheat on the approach temperature and compression energy demand (in Fig. 12, CO2 cooling temperature). At −49 °C, there was a sharp change in the approach temperature and an increase in the
The degree of superheat had no added benefit, even though the compression energy demand decreased, but in an infeasible region. Therefore, \(-49 \, ^\circ\text{C}\) was selected as the optimum for the \(\text{N}_2-\text{CO}_2\) expander system.

8. Optimization results and discussion

The optimization procedure presented in section 7 is illustrated with an algorithm in Fig. 14. The steps in Fig. 15 must be followed sequentially for system optimization. The reason for using the sequential approach, where pressure levels are optimized first, lies in the fact that the operating pressure is the main source of refrigeration in the expander-based cycles unlike the MR system, where MR composition also has a significant effect on the system optimality. Expanding the refrigerant from a high pressure creates more cold energy or refrigeration than changing the flow rate and other temperature levels. Therefore, optimization of the pressure levels followed by the refrigerant flow rate and temperature levels has consistent results, as supported by the simulation.

The optimization algorithm presented was applied to the base case presented in Table 4. The results revealed 15% improvement in the compression power requirement and a 7% increase in the refrigerator exergy efficiency. The improvement in system efficiency can be explained by analyzing Figs. 15 and 16, which shows the hot and cold composite curves drawn for the base and optimized case presented in Table 4, respectively. The area between the composite curves represents the amount of irreversibility involved in heat transfer. The higher area involved in the base case represents inefficient heat transfer, and a higher than the required heat transfer driving force in the perspective optimized case shows smaller area, hence more efficient heat transfer. Provided the heat transfer driving force is higher than the required potential, efforts must be made to make it as small as possible to improve the system efficiency (Shukri, February 2004).
The optimization method presented in this paper for a single expander N₂—CO₂ NG liquefaction process was inspired by the knowledge of the system, which presents windows of opportunities for possible improvement. Seeing thorough those windows, an opportunistic optimization algorithm was devised for the N₂—CO₂ example. On the other hand, with a slight modification, the developed algorithm can be generalize easily for processes involving expanders, such as a single and dual N₂ expander, dual expander N₂—CO₂ process and will be discussed in future issues.

Although the present optimization methodology was not as exhaustive as the conventional methods that work with specific sets of mathematical rules for objective improvements, it has the benefits of process in-sight over conventional mathematical optimization methods. Currently, the methodology was carried out manually but some extracted general rules can be programmed to be executed independently and will be addressed in future editions.

Certainly, the overall design optimization of any process not only includes the operating cost and capital cost. For example, a high heat transfer area gives a small approach temperature in a LNG exchanger but eventually increases the capital cost, which in turns increases the total annualized cost and the overall system economics. Therefore, balance between the capital and operating cost is needed. For that, a multi-objective optimization problem must be solved. This would be a challenging task and requires considerable endeavor, which will be addressed in future. The development of this work is believed to provide a solid groundwork for this future thrust.

### 9. Conclusions

N₂—CO₂ expander is an important NG liquefaction cycle, which has attracted recent attention because of the inherent safety in the operation of offshore plants. Unlike mixed refrigerant cycles, the high energy demand in the N₂—CO₂ expander cycle is an unusual concern. To address this issue, optimization was performed for the N₂—CO₂ expander process. First, the decision variables were identified and component sensitivity analysis was performed to determine their normal operating ranges. An opportunistic optimization algorithm was devised using process knowledge and with rigorous simulations, the N₂—CO₂ expander process was optimized for compression energy requirements. The optimization results revealed energy savings of 15% compared to the base case and an increase in the process exergy efficiency. The strong point of the proposed opportunistic optimization algorithm is that it can be generalized easily for any type of expander process of the NG liquefaction process. This methodology can commence from any point that does not depend on good initial estimates and improves consistently with the aid of process insight.

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