THERMODYNAMIC EFFICIENCY EVALUATION OF A LOW PRESSURE TURBO EXPANDER CRYOGENIC CYCLE BASED ON EXERGY ANALYSIS

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Thermodynamic analysis, using the exergy or entropy methods, is usually carried out for better insight into the physical meaning of the losses encountered in a cryogenic plant. From the results of such analysis, it is possible to evaluate the individual efficiencies of the various processes and to identify those calling for an improvement. It is also possible to determine thermodynamic efficiency of the cycle as a whole. The technique involves determination of entropy changes or exergetic losses in each of the processes making up the cycle.

Based on the exergy analysis, it has been possible to evaluate specific work requirement, overall thermodynamic efficiency, Specific cooling capacity, work requirement per kg of liquid nitrogen product and coefficient of Performance of the Turbo expander cryogenic cycle using hydrogen and helium as the refrigerant.

Key words*: Exergy analysis, Turbo expander cycle, Cryogenic, Liquid nitrogen*

1. Introduction

The concept of exergy is extremely useful for the purpose of finding inefficiencies and losses for any process with the proper application of the principles of Second Law of Thermodynamics [1].Energy efficiency analyses and optimizes chemical processes for the effective use of energy and resource materials [2].Exergy can be defined as the maximum work that is available in any station in the system relative to the surrounding condition. As such an exergy analysis can provides pertinent information on the effective energy used in the process along with the exact location and magnitude of the exergy loss in the equipment for improving overall efficiency [3-4].

Yong et al. [5] in their experiment perform the exergy analysis and exergy losses of two process i.e. cold heat supply method and distillation column method that generates nitrogen and identified criteria to select the optimal process and equipment design. Sieve tray method and structured packing method for the distillation were taken into consideration for the analysis. Consequently, the process conditions were determined for maximizing the exergy efficiency and minimizing the energy loss for each operation mode. Acikkalp et al. [6] carried out an exergy analysis on producing argon, nitrogen, and oxygen with a daily capacity of 250 tons. The energy and exergy values for each point defined in the system were obtained. By using these values, thermodynamic evaluations for both the whole system and also its components were made. The efficiency values of energy and exergy, the values of energy losses and exergy destruction rates, the EIP (energetic improvement potential rate), ExIP (exergetic improvement potential rate), and the production of entropy values were found as 0.453, 0.79, 4368.475 kW, 10535.875 kW, 2391.535 kW, 3800.485 kW, and 35.347 kW/K, respectively. The energy and exergy efficiencies of the plant were found to be 45.3% and 13.1% respectively. Sapali et al. [7] in his work, simulated medium purity oxygen cryogenic plant by using Aspen plus computer code [8] for gasification application and found out specific power consumption. It is also an attempt to develop a model of biomass/coal gasifier with oxygen and steam as oxidizing agent by using Aspen plus. In addition to this exergy analysis of cryogenic ASU and biomass gasifier is also performed. Wang et al. [9] reports specific distribution of exergy destruction and the influences of mixed refrigerant composition concentration on the performance of the NGEMR (Mixed refrigerant process using natural gas expansion) liquefaction process. The result shows that the exergy destruction in the system is mainly caused by expansion turbines and compressors. For a certain high pressure of the MRC (Mixed Refrigeration Cycle) under a fixed pressure ratio, there is a corresponding optimal certain mixed refrigerant composition, under which the minimum exergy destruction of the liquefaction process could be obtained. Li Yao et al. [10] carried out the exergy analysis of a cryogenic air distillation plant and quantified the exergy loss in the various plant sections. An exergy calculation software program for an air separation process is developed and the detailed exergy calculation and analysis for an actual 40,000 m^3/h air separation unit are performed. Yasuki et al. [11] investigated a novel cryogenic air separation process to reduce energy consumption. H. Dong et al. [12] proposed a basic Stirling cycle cryogenic generation process for LNG cold energy recovery. The energy and exergy changes in thermodynamic process, both of LNG regasification process and Stirling cycle operation process are calculated. The effects of some key parameters on performance of the basic process, such as LNG vaporization pressure and ambient temperature, are also investigated. Tuo et al. [13] in their paper presented a novel combined cycle of air separation and natural gas liquefaction. The idea is that natural gas can be liquefied meanwhile gaseous or liquid nitrogen and oxygen are produced in one combined cryogenic system. Cycle simulation and exergy analysis are performed to evaluate the process revealing the influence of the crucial parameter, i.e., flow rate ratio through two stages expander, heat transfer temperature difference, its distribution and consequent exergy loss. Exergy analysis is carried out on Pressure swing adsorption cycles for the production of oxygen from air by Banerjee et al. [14] for equilibrium separations using the method of exergy analysis. The optimum operating point is determined and individual component losses are identified. A modified cycle incorporating a pressure equalization step has been found to be superior in terms of its power requirements. The methodology of exergy analysis provides a rational criterion for determination of the optimal operating parameters for a specified configuration and for comparing different configurations. Agarwal et al. [15] identified exergy analysis inefficiencies in the distillation system for an efficient cryogenic air separation plant producing large tonnage quantities of nitrogen. Van der Ham et al. [16] have evaluated two cryogenic processes; one with two distillation columns and another with three distillation columns, using exergy analysis. They found that the three column design is superior to two column design in term of exergy destruction by 12%.

Several exergy analyses at low temperature for refrigeration and air conditioning are also available in the literature. Saravanan et al. [17] carried out energy and exergy analysis using a mathematical model to conclude that inlet air wet bulb temperature is the most important parameter than inlet water temperature and the performance of wet cooling tower remains unaffected with the variation of dead state properties. Thermodynamic analysis for refrigerating machines and heat pumps using entropy-cycle method is reported by Morosuk et al. [18]. Kalaiselvam et al. [19] reported the exergy analysis and exergual features of three refrigerants i.e. R22, R417A, and R407C used in scroll compressor for air conditioning system. Thermo dynamical behavior, irreversibility, interaction between the system and the refrigerant in terms of pressure drop and heat transfer, are used for calculation of exergy. It is observed that coefficient of performance of R407C is 2.41% less than R22. Wang et al. [20] carried out exergy and energy analysis of heat pump using blended R744/R32 refrigerant. It is reported by them that performance at blending concentration ratio of the refrigerant (15/85 by mass) is better than the performance of R22.

A cryogenic plant accomplishes its assigned duty by performing a sequence of simple processes which together make up a particular thermodynamic cycle. In designing a cryogenic plant it is first necessary to select an appropriate cycle and then to make necessary calculations. These calculations are carried out in order to determine heat flows, refrigerant mass and state variables at all characteristic points. Many variables such as stream pressures, temperature levels , temperature difference in heat exchangers, expander efficiency, heat in leak from the surroundings, pressure drop etc. are usually specified in advance and if properly chosen all of these variables will lead to an optimum efficiency .

The present study applies exergy analysis on a unique closed loop low pressure refrigeration cycle with perfect gas such as hydrogen or helium as the refrigerant. The cold produced in the refrigeration cycle is utilized for liquefaction of purified nitrogen which passes through a separate passage in heat exchanger i.e. condenser. Details of thermodynamic analysis and performance study of the cycle is necessary for optimum design of such plants.

2. Description of the proposed Turbo Expander Cryogenic Cycle

The proposed unique low pressure cryogenic cycle for nitrogen gas liquefaction is presented in Fig. 1. Gases such as helium or hydrogen are used as the refrigerant in the closed loop refrigerating cycle which produces the required cooling effect for liquefaction of nitrogen gas in the cryogenic temperature range.

The gas is isothermally compressed from 0.1 MPa (1 atm.) and ambient temperature (T_1) to a higher pressure at about 0.6 MPa (6atm.) in the compressor C and work performed by compressor is W_C. The gas is then admitted to a heat exchanger at temperature $T_2 = T_1$. The gas comes out from the heat exchanger at point 3 and its temperature drops down to T_3. Past the heat exchanger the gas is admitted in a turbo expander E, where it expands to a low pressure while performing some work W_e. The pressure ratio of the expander is adjusted in such a way so that the refrigerant gas does not reach its dew point at the expander exit at the prevailing low pressure. The cold q_C gas goes to the condenser, low temperature condenser (LTC) where the refrigerant gas absorbs heat q_C from the pure nitrogen gas and which in turn gets liquefied. The temperature of refrigerant is raised from T_4 corresponding to point 4 to T_5 (point 5). The return stream of refrigerant gas is heated in the heat exchanger to a state corresponding to point 1' by absorbing heat from the forward stream (compressed hot gas). At the warm end of the heat exchanger, the difference in temperature between forward and return streams is $\Delta T_h = T_1 - T_{1'}$ as per Fig.2 (b). The heat exchanger, expander, the condenser are all insulated from the surroundings.

The purified Nitrogen is fed to the condenser through a pressure regulating valve at a pressure slightly above the atmospheric pressure so that gas can flow through its passage. The liquid nitrogen from the condenser passes to the liquid nitrogen storage vessel from which liquid nitrogen could be drawn through the delivery line as shown in (Fig. 1). Boil off nitrogen gas from the storage vessel is returned to the condenser for optimizing the efficiency.

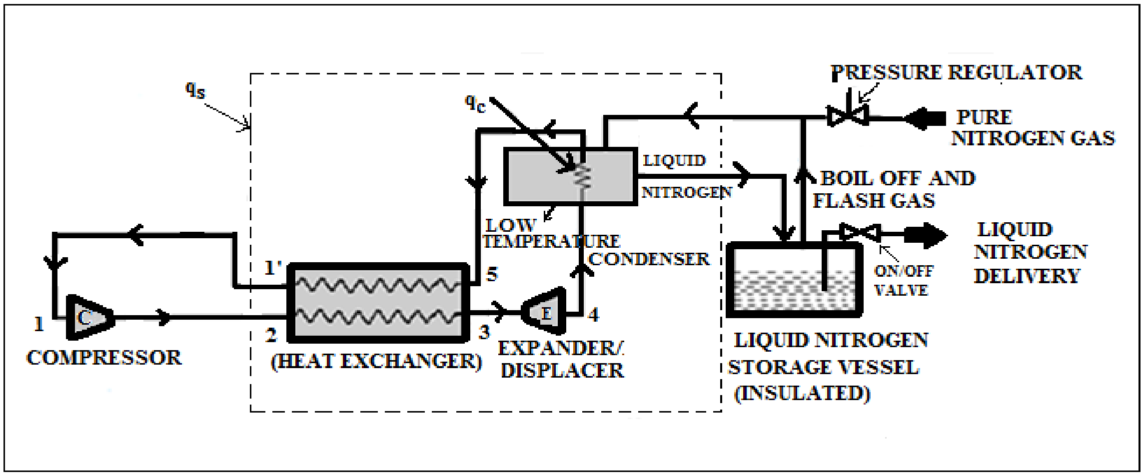

Figure 1: Simple Expander cryogenic cycle for nitrogen liquefaction

3. Thermodynamic analysis of the Turbo Expander Cryogenic Cycle

There is a marked discrepancy in performance between ideal and real cycles because any processes occurring in real conditions are irreversible to one degree or another. This discrepancy manifests itself above all in the quantity of energy required to transport heat from the lower to the higher temperature level that is for operation of cryogenic plant. The energy needed by real cycles is higher than the energy requirement by ideal one and it is because of the losses due to irreversibility.
In case of real cycles, total energy needed is expressed by

$$W = W_{\min} + T_0 \sum \Delta S \qquad \textbf{(1)}$$

Where, $T_0 \sum \Delta S$ is the excess work required in addition to minimum work ($W_{\min}$)
$W_{\min}$, is min work required in case of ideal cycle. T_0 is the temperature of the surroundings and $\sum \Delta S$ is the total entropy changes of all bodies taking part in a given process. This excess work does not produce any useful effect in the final analysis it is all converted to heat which is rejected to the surroundings. Entropy is a quantitative characteristic of irreversibility, a fact which enables it to be used in evaluating energy losses and efficiency.
The value of ΔS in expression **(1)** for each particular process (piece of equipment) of a cryogenic system depends on the magnitude of losses. For analysis of cryogenic cycles, in addition to the minimum and excess work, also certain coefficients which give a measure of the efficiency of a particular cycle are also required. These are the specific amount of cooling q_C the specific energy, the coefficient of performance (C.O.P) the thermodynamic efficiency η_t, the fraction liquefied x etc.
For the thermodynamic analysis of the cycle and for finding the energy balance equation the (Fig. 1) could be reduced to a simple expander cycle as shown in (Fig. 2 (a)) and it T-S diagram is shown in (Fig. 2(b)). This simple expander cycle is designed to produce the required refrigeration for liquefaction of nitrogen gas.

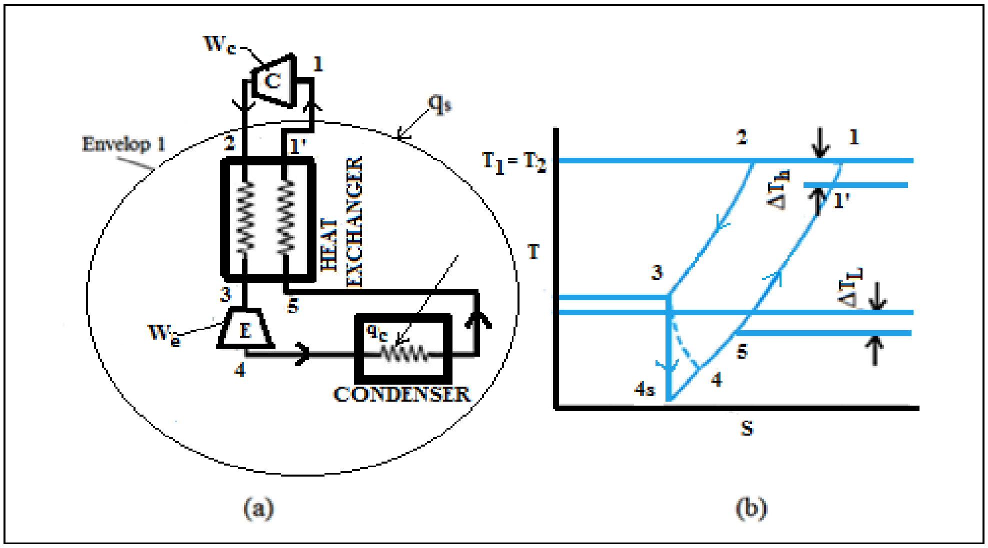

Figure 2: (a) Simple expander cryogenic cycle (b) T-S Diagram

4. Energy balance equations and equations for certain coefficients

For writing the energy balance equation an envelope-1 is considered (Fig. 2(a)).

Now for mass flow rate of M kg/hr. the energy balance equations can be written by neglecting variations in Kinetic and Potential energy of the system. The energy balance around the closed loop has the form

$$Mh_2 + M(h_5 - h_4) + Mq_S = Mh_{1'} + MW_e \qquad \textbf{(2)}$$

h_1, h_2, h_3, h_4, h_5, are the specific enthalpies of the refrigerant gas at point 1', point 2, point 4 and point 5 respectively. q_S is the heat load due to imperfect insulation, W_e is the expander work and is equal to $h_S\eta_S$ where η_S adiabatic efficiency of the expander. , h_S is the enthalpy difference in an ideal expander and is equal to isentropic enthalpy difference between points 3and 4s in the T-S diagram.

The total cooling capacity is given by

$$q_C = M(h_5 - h_4) \qquad \textbf{(3)}$$

Now, substituting this term in equation (2) we get

$$Mh_2 + q_C + Mq_S = Mh_{1'} + Mh_S\eta_S \qquad \textbf{(4)}$$

Again
$$h_{1'} = h_1 - C_p\Delta T_h$$

Where, C_p is Specific heat of gas and ΔT_h is temperature difference at the warm end of the heat exchanger, and h_1 is the specific enthalpy of the gas at point 1.

Therefore, the equation (4) takes the form

$$Mh_2 + q_C + Mq_S = M(h_1 - C_P\Delta T_h) + Mh_S\eta_S \qquad \textbf{(5)}$$

A Rearranging equation (5) we get,

$$Mh_S\eta_S + M(h_1 - h_2) = q_C + MC_P\Delta T_h + Mq_S$$

Or,

$$M(h_S\eta_S + h_1 - h_2) = q_C + M(C_P\Delta T_h + q_S) \qquad \textbf{(6)}$$

For 1 kg of compressed gas (refrigerant) the equation (6) reduces to

$$h_S\eta_S + h_1 - h_2 = q_C + (C_P\Delta T_h + q_S)$$

which on rearranging becomes

$$q_C = (h_1 - h_2 + h_S\eta_S) - (q_S + C_P\Delta T_h) \qquad \textbf{(7)}$$

Where q_C is now gives the specific cooling capacity. $h_1 - h_2 = \Delta h_{t_1}$ is the specific refrigeration produced by the cycle owing to the isothermal throttle expansion of the compressed gas and $h_S\eta_S$ gives the specific refrigeration produced due to expansion of the gas in the expander . $(q_S + C_p\Delta T_h)$ is the amount of specific refrigeration losses in the cycle. Ignoring the loss due to gas leaks, the specific work requirement of the cycle is

$$W = W_C - W_e\eta_m = (\frac{RT_1}{\eta_t})\ln r_C - h_S\eta_S\eta_m \qquad \textbf{(8)}$$

Where, r_C is the compression ratio of the compressor, η_t is the isothermal efficiency of the compressor, η_m is the mechanical efficiency of the expander (the fraction of the compression work returned to the cycle) and R is the gas constant. The compression ratio, $r_C = \frac{P_2}{P_1}$ is connected to the pressure ratio of the expander by a relation $r_C = jr_e$, where j is the reduced pressure drop coefficient, given by

$$j = \frac{1 + \Delta P_3 / \Delta P_4}{1 - \Delta P_4 / P_4} \qquad \textbf{(9)}$$

Where, $\Delta P_3 / \Delta P_4$ and $\Delta P_4 / P_4$ are the dimensionless pressure drops for the forward and return stream respectively and $r_e = P_3 / P_4$.The temperature difference across the low temperature condenser is given by $\Delta T_L = T_L - T_5$,where T_L is the constant temperature in the low temperature condenser.

In theoretical cases, T_L = 0 i.e. T_L becomes equal to T_5. For each value of T_L, there exist optimal cycle variables such that the coefficient of performance or the thermodynamic efficiency of the cycle η_t is maximum and optimization of variables is necessary.

5. Exergetic analysis of the expander cycle

Various techniques involves in determining the entropy changes or exergetic losses in each of the processes for making up the cycle.

Step – 1: Increase in entropy in the compression stage

The increase in entropy of the system upon compression of 1 kg of working fluid in the compressor C is

$$\Delta S_C = \frac{R}{\eta_t}\ln r_C - R\ln r_C = (\frac{1}{\eta_t} - 1)R\ln r_C \qquad \textbf{(10)}$$

Where, R is the gas constant, r_C is the compression ratio, η_t is the isothermal efficiency of the compressor.

Step 2: Increase in entropy in the turbo expansion stage

The increase in entropy of the system upon compression of 1 kg of refrigerant in the adiabatic turbo expander (E) is

$$\Delta S_e = S_4 - S_3 + (h_3 \eta_S / T_0)(1 - \eta_m) \qquad \textbf{(11)}$$

Where, S_3 and S_4 is the specific entropy of the working fluid (refrigerant) before and after the expander and it is assumed $T_0 = T_1$ (ambient temperature). η_S and η_m are the adiabatic efficiency and mechanical efficiency of the expander respectively and h_s is the enthalpy difference in an ideal expander. The last term on the right-hand side of equation **(11)** represents an additional increase in the entropy of the system due to loss of some of the work delivered by the expander, if any. For an ideal gas, equation **(11)** may be re-written.

$$\Delta S_e = C_p \{ \frac{\gamma - 1}{\gamma} \ln r_C + \frac{T_3 - T_4}{T_0}(1 - \eta_m) - \ln \frac{T_3}{T_4} \} \qquad \textbf{(12)}$$

As for ideal gas we have,

$$h_S \eta_S = C_P (T_3 - T_4)$$

$$\eta_S = \frac{(T_3 - T_4)}{(T_3 - T_{4s})}$$

$$S_4 - S_3 = S_4 - S_{4s} = C_P \ln \frac{T_4}{T_{4s}}$$

$$and, \frac{T_4}{T_{4s}} = r_e = \frac{\gamma - 1}{\gamma (T_4 / T_3)}$$

Step – 3: Increase in entropy in the heat exchanger stage

If pressure drop is negligible and the flow rates of the forward and return streams are the same, then the increase in entropy of the system due to heat transfer for 1 kg of refrigerant gas at $T_2 = T_1 = T_o$ is

$$\Delta S_{HE} = C_p [\ln(1 - \frac{\Delta T_h}{T_1})(1 + \frac{\Delta T_C}{T_5}) - \frac{\Delta T_C - \Delta T_h}{T_1}] \qquad \textbf{(13)}$$

Where $\Delta T_C = T_3 - T_5$ and $\Delta T_h = T_1 - T_{1'}$.

If there is a pressure drop, an irrecoverable loss also occurs in the heat exchanger resulting in an increase in entropy which can be approximately determined, taking it equal to the increase in entropy upon the throttle expansion of an ideal gas.

$$\Delta S_{pd} = \frac{\gamma - 1}{\gamma} C_p \ln j \qquad \textbf{(14)}$$

Where, reduced pressure drop coefficient is given by

$$j = \frac{1+\Delta P_3 / \Delta P_4}{1-\Delta P_4 / P_4}$$

Total increase in entropy in the heat exchanger stage is

$$\Delta S_H = \Delta S_{HE} + \Delta S_{pd} \qquad \textbf{(15)}$$

Step – 4: Increase in entropy in the low temperature chamber

The increase in entropy due to interaction of 1 kg of refrigerant with the material being cooled in the low temperature chamber is

$$\Delta S_L = C_P(\ln\frac{T_5}{T_4} - \frac{T_5 - T_4}{T_L}) \qquad \textbf{(16)}$$

Where, T_L is the temperature of the material being cooled.

Step –5: Increase in entropy due to irreversibility of the process 1' -1

The temperature difference of the warm end of the heat exchanger gives rise to a small additional loss due to the irreversibility of the process 1' -1 and the corresponding increase in entropy is approximately given by

$$\Delta S_{add} = \frac{1}{2} C_p (\frac{\Delta T_h}{T_1})^2 \qquad \textbf{(17)}$$

If the entropy changes are known, it is possible to determine the exergetic losses,

$$L_i = T_0 \Delta S_i \qquad \textbf{(18)}$$

or,

the reduced exergetic losses,

$$T_0 \frac{\Delta S_i}{W} = \overline{L_i} \qquad \textbf{(19)}$$

in the various steps of the cycle, where W is the specific work requirement of the cycle. The reduced exergetic losses of the various steps of the cycles are given by

Step1: $$\frac{T_0 \Delta S_C}{W} = \overline{L_C} \qquad \textbf{(20)}$$

Step2: $$\frac{T_0 \Delta S_e}{W} = \overline{L_e} \qquad \textbf{(21)}$$

Step3: $$\frac{T_0 \Delta S_H}{W} = \frac{T_0(\Delta S_{HE} + \Delta S_{pd})}{W} = \overline{L_H} \qquad \textbf{(22)}$$

Step4: $$\frac{T_0 \Delta S_L}{W} = \overline{L_L} \qquad \textbf{(23)}$$

Step5: $$\frac{T_0 \Delta S_{add}}{W} = \overline{L_{add}} \qquad \textbf{(24)}$$

Therefore, total reduced exergetic losses in all steps making up the cycle is given by

$$\sum \bar{L_i} = \bar{L_C} + \bar{L_e} + \bar{L_H} + \bar{L_L} + \bar{L_{add}} \qquad \textbf{(25)}$$

And, thermodynamic efficiency of the cycle is given by

$$\eta_t = 1 - \sum \bar{L}_i \quad \textbf{(26)}$$

6. Conditions

Based on the following standard condition, cycle calculation and exergetic analysis could be carried out to find out the overall thermodynamic efficiency, co-efficient of performance of the cycle in addition to specific work requirement and specific cooling capacity.

1) Pressure drop coefficient j = 1.15
2) Compressor isothermal efficiency η_t = 0.6
3) Expander adiabatic efficiency η_S = 0.8
4) η_m = 0, as work of compression is not returned to the cycle
5) The temperature difference at the warm end of heat exchanger ΔT_h = 4K and that at the cold end ΔT_C = 5K = (T_3 - T_5)
6) Ambient temperature $T_0 = T_1 = T_2$ = 300 K (normal average ambient temperature)
7) Nitrogen is assumed to enter the condenser at 1.2 atm. (0.11 MPa) i.e. just slightly above atmospheric pressure so that it can flow through the condenser.
8) Condenser efficiency 80%

7. Result and Discussion

Based on the above recommended values and for an expander ratio of 5.22 (operating pressure 0.1 MPa to 0.6MPa), various essential parameters are computed are given in Table 1.

Table 1: Parameters used for exergy calculation based on turbo expander refrigeration cycle in the cryogenic range for two refrigerants.

Parameter	Hydrogen expander cycle	Helium expander Cycle
T_0=T_1=T_2	300K	300K
$T_{1'}$	296K	296K
ΔT_h=T_1-$T_{1'}$	4K	4K
T_3	135K	135K
T_4	75K	75K
T_5=T_L	130K	130K
ΔT_c = T_3-T_5	5K	5K
R	4157 J/kg/K	2078.5 J/kg/K
C_P	10510 J/kg	5.2*10^3J/kg
ɣ	1.41	1.66

ɣ is the specific heat ratio of the refrigerant gases.

Specific cooling capacity (q_C) and specific work requirement (W) are computed by applying equation **(7)** and equation **(8)**. Work requirement per kg of liquid nitrogen (W_L) are also calculated using enthalpies from T-S diagram and for condenser efficiency of 80% as nitrogen liquefaction cooling load is easily available by any standard methods. All these thermodynamic analysis are presented in Table 2.

Table 2: Result of thermodynamic Analysis

Calculated parameters	**Hydrogen expander cycle**	**Helium expander Cycle**
Specific work requirement	3724.17 kJ/kg	1862.086 kJ/kg
Specific Cooling capacity	825.56 kJ/kg	334.20 kJ/kg
Work requirement per kg of liquid nitrogen product	3665.25 kJ	4527.06 kJ

Increase in entropies in compression stage, expansion stage, heat transfer stage, due to pressure drop stage, condenser stage, due to irreversibility of the process (1-1') stage are computed by using equation **(10)** to equation **(17)**. The values thus formed for entropies increases are given in Table 3.

Table 3: Various entropy increases per kg of refrigerants used are calculated

Increase of entropy in different section	**Hydrogen expander cycle**	**Helium expander Cycle**
Increase in entropy in the compression stage **(** ΔS_C **)**	4965.56 J/kg/K	2482.78 J/kg/K
Increase in entropy in the expander **(** ΔS_e **)**	874.95 J/kg/K	1404 J/kg/K
Increase in entropy due to heat transfer **(**ΔS_{HE}**)**	214.26 J/kg/K	106 J/kg/K
Increase in entropy due to pressure drop **(**ΔS_{pd} **)**	427.12 J/kg/K	288.95 J/kg/K
Increase in entropy in the heat exchange stage**,** **(**$\Delta S_H = \Delta S_{HE} + \Delta S_{pd}$**)**	641.39 J/kg/K	394.95 J/kg/K
Increase in entropy in the low temperature condenser **(** ΔS_L **)**	1334.95 J/kg/K	655.2 J/kg/K
Increase in entropy due to irreversibility of the process(1-1') **(** ΔS_{add} **)**	0.934 J/kg/K	0.46 J/kg/K

The reduced exergetic losses at various stages are evaluated by employing equation **(20)** to equation **(24)**. Total reduced exergetic losses are found by equation **(25)**. Finally, the overall thermodynamic efficiency of the cycle is estimated by equation **(26)**. All these values for exergetic losses and overall thermodynamic efficiency are given in Table 4.

Table 4: Reduced exergetic losses of two refrigerant cycles and overall thermodynamic efficiency

Reduced exergetic losses ($\overline{L_i}$)	**Hydrogen expander cycle**	**Helium expander Cycle**
$\overline{L_C}$	0.399	0.400
$\overline{L_e}$	0.070	0.226
$\overline{L_H}$	0.051	0.063
$\overline{L_L}$	0.107	0.106
$\overline{L_{add}}$	0.000075	0.000074
$\sum \overline{L_i}$	0.627	0.795
Overall thermodynamic efficiency, η_t **= 1-** $\sum \overline{L_i}$	0.373 (37.3%)	0.205 (20.5%)

8. Conclusion

It is found that the overall thermodynamic efficiency of the expander cycle for hydrogen and helium are as 37.3% and 20.5% respectively For the same adiabatic efficiency of the expander, a higher temperature drop (T_3-T_4) during expansion of helium gas is encountered but for simplicity of calculation, a same temperature drop of 75K is considered for both hydrogen and helium gases. If the actual temperature drop is considered from T-S diagram, then overall thermodynamic efficiencies will be almost same for both the gases. As helium gas is not easily available, hydrogen gas is recommended as the refrigerant for the refrigeration cycle.

From exergetic analysis of the low pressure turbo refrigeration cycle, it is observed that most exergetic losses occurred in case of compression stage followed by expansion stage, heat exchanger stage for both the hydrogen and helium expander cycle. Thermodynamic efficiencies of the cycle depend on the magnitude of the exergetic losses. Higher the exergetic losses lesser is the overall thermodynamic efficiency of the cycle. It is found that the overall thermodynamic efficiency of the expander cycle for hydrogen and helium are as 37.3% and 20.5% respectively.

The entire analysis clearly indicates the piece of equipment requiring improvement in efficiencies as being reflected by high exergetic losses. Therefore, present exergy analysis will help design an efficient of low pressure turbo refrigeration cycle for liquefaction of cryogenic gases using hydrogen as a refrigerant.

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