



Thermodynamic (energy-exergy) analysis of pre-cooled linde system

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Abstract

The present studies concern on detailed thermodynamic (energy and exergy analyses) of Pre cooled Linde cryogenic system up to their sub component level. This parametric study is conducted to investigate the effects of variation of various system input parameters such as pressure ratio, expander mass flow ratio, compressor output temperature on different performance parameters like COP, work input, liquefaction rate, specific heat and exergy. The numerical computations have been carried out for Claude system are study with six different gases for liquefaction like oxygen, argon, methane, fluorine, air and nitrogen respectively. It was observed that methane has better thermal performances (first law performance i.e. COP) and exergetic efficiency (second law performance) as compared to other gases. The total net work done decreasing.

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Keywords: Thermodynamics Analysis, Collin Cryonic System, Energy-Exergy Analysis, First and second Law Analysis

1. Introduction

The term “cryogenic” is derived from the Greek word Kryos which means cold or frost. It is frequently applied to very low temperature refrigeration applications such as in the liquefaction of gases and in the study of physical phenomenon at temperature approaching absolute zero.

The first low temperature refrigeration system was primarily developed for the solidification of carbon dioxide and the liquefaction and subsequent fractional distillation of gases such as air, oxygen, nitrogen, hydrogen and helium. Cryogenic process to liquefy air which is further extent to extract various particular gas like oxygen, nitrogen, iron etc. Always various analyses is done to identify the loop hole of process and to rectify it to their upper level. electro caloric cooling is a transiting to new cooling principle's is critical and one of the most promising alternatives may be [1]. Various particular part are taken under study to increase overall performance of cryogenic system e.g. A good exergetic design of a heat exchanger would allow for an increase in the global efficiency of the process, by defining a thermodynamic cycle in which the exergetic losses would be limited [2] apart from this other parts

like expander, mass ratio and input variables are considered to improve cryo systems.

Pre-cooled Linde Hampson system is advanced stage old simple Linde Hampson system. In this system vapour compression evaporator having R134 a refrigerant is taken as additional equipment to fill the gap of simple Linde system. In Fig.1 (a&b) shows the detail block diagram and T-S diagram of Pre-cooled Linde system. Vapor compression system lower down the temperature of gases coming through the compressor which is desirable for better performance of system. The method of cooling the gas after the compression or before the entrance to the heat exchanger is called as precooling. The precooling limit of the precooling cycle is governed by the boiling point of refrigerant at its suction pressure this increment in the yield is dependent on the The change in enthalpy values from $(h_d - h_a)$ of the refrigerant. Refrigerant flow rate (m_r) . the heat of compression Q_R can be obtained by using 2nd Law for an isothermal compression.

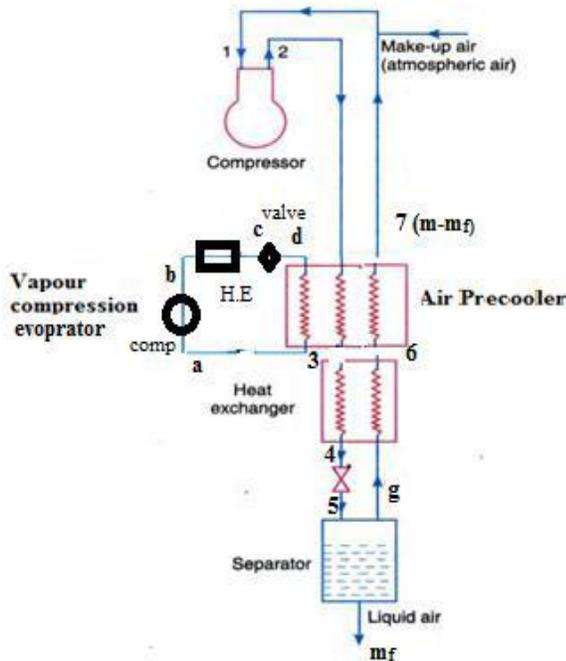


Figure 1(a): Schematic of Pre-cooled linde system

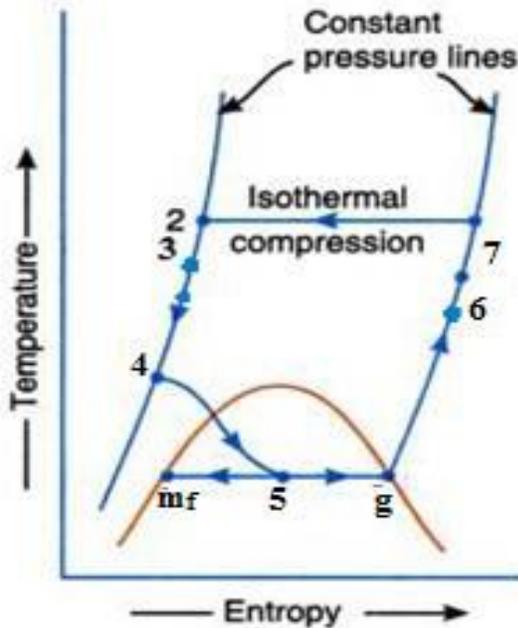


Figure 1(b): T-S diagram of Pre-cooled linde system

2. Mathematic Modelling of "Pre-Cooled Linde Hampson Cycle"

Working Fluid Properties

R\$ = 'oxygen'

$$P_1 = 1.013, T_1 = 298, T_1 = T_0, P_2 = 100, T_2 = T_1$$

$$P_f = P_1, x_0 = 0, x_1 = 1, T_f = T_g,$$

Refrigerant Properties R134 a

$$P_a = 1.013$$

$$P_b = 10.013 \text{ Adiabatic Compression}$$

$$P_c = P_b, T_a = 300, T_b = 373, T_c = 300, h_a = 278, h_b = 334.9, h_c = 89.07$$

"Boiling Temperature of Refrigerant"

$$T_d = 247.1$$

Control Vol having 2 HT, J-T, Separator

"Energy Balance by First Law"

$$\frac{m_r}{m} = r, r = 0.08$$

$$m_r * h_d + m * h_2 = m_r * h_a +$$

$$(m - m_f) * h_1 + m_f * h_f$$

$$y = \frac{m_f}{m}$$

Control Vol having HT, J-T, Separator

T, Separator

$$m * h_3 = m_f * h_f + (m - m_f) * h_6$$

$$y_{\max} = \frac{h_6 - h_3}{h_6 - h_f}$$

$$W_{\text{total}} = W_{c1} + W_{c2}$$

Compressor work per liquefaction of system

$$E_{\text{in}} = E_{\text{out}}$$

$$E_{\text{in}} = m * h_1 - W_{c1}$$

$$E_{\text{out}} = m * h_2 - Q_R$$

$$Q_R = m * T_1 * (s_2 - s_1)$$

Compressor work of ref system

$$m_r * h_a - W_{c2} = m_r * h_b$$

$$-\frac{W_{\text{total}}}{m_f} = Z$$

$$-\frac{W_{\text{total}}}{m} = \text{abs} \left(\frac{T_1 * (s_1 - s_2) - (h_1 - h_2) + r * (h_b - h_a)}{m} \right)$$

$$-\frac{W_i}{m} = T_1 * (s_1 - s_f) - (h_1 - h_f)$$

$$-\frac{W_i}{m_f} = K$$

$$\text{FOM} = \frac{Z}{K}$$

$$\text{COP} = \left(\frac{h_1 - h_f}{W_{\text{total}}} \right)$$

$$\text{Eta}_{2\text{nd}\%} = \text{abs} \left(\left(\frac{(h_f - h_1) - T_0 * (s_f - s_1)}{W_{\text{total}}} * m_f \right) * 100 \right)$$

First Heat Exchanger (HX1) analysis

TypeHX1\$ = 'counterflow'

$$\text{epsilon}_{\text{HX1}} = 0.85$$

$$T_{c0} = 292, T_{h1} = T_1, T_{10} = T_{c0}$$

$$m_{h\text{HX1}} = m, m_{c\text{HX1}} = m - m_f$$

$$T_{h0} = T_3, T_{c1} = T_9$$

$$C_{h\text{HX1}} = m_{h\text{HX1}} * \text{cp}_{\text{fluidHX1}}$$

$$C_{c\text{HX1}} = m_{c\text{HX1}} * \text{cp}_{\text{fluidHX1}}$$

$$Q_{\text{HX1}} = C_{h\text{HX1}} * (T_{h1} - T_{h0})$$

$$Q_{\text{HX1}} = C_{c\text{HX1}} *$$

$$q_{\max_HX1} = C_{\min_HX1} * (T_{h_i} - T_{c_i})$$

$$\epsilon_{HX1} = \frac{q_{HX1}}{q_{\max_HX1}}$$

$$Ntu_{HX1} = HX(\text{TypeHX}_{1\$}, \epsilon_{HX1}, C_{h_{HX1}}, C_{c_{HX1}}, 'Ntu')$$

$$Ntu_{HX1} = (G_{HX1})/C_{\min_HX1}$$

$$Ed_{HX1} = \left(T_0 * ((s_2 + s_9) - (s_3 + s_{10})) \right)$$

Second Heat Exchanger (HX_3) analysis

$$\text{TypeHX}_{3\$} = \text{'counterflow'}$$

$$\epsilon_{HX3} = 0.85$$

$$m_{h_{HX3}} = m$$

$$m_{c_{HX3}} = m - m_f$$

$$C_{h_{HX3}} = m_{h_{HX3}} * cp_{\text{hot fluid}_{HX3}}$$

$$C_{c_{HX3}} = m_{c_{HX3}} * cp_{\text{cold fluid}_{HX3}}$$

$$q_{HX3} = C_{h_{HX3}} * (T_4 - T_5)$$

$$q_{HX3} = C_{c_{HX3}} * (T_9 - T_g)$$

$$q_{\max_HX3} = C_{\min_HX3} * (T_4 - T_g)$$

$$\epsilon_{HX3} = \frac{q_{HX3}}{q_{\max_HX3}}$$

$$Ntu_{HX3} = HX(\text{TypeHX}_{3\$}, \epsilon_{HX3}, C_{h_{HX3}}, C_{c_{HX3}}, 'Ntu')$$

$$Ntu_{HX3} = (G_{HX3})/C_{\min_HX3}$$

$$Ed_{HX3} = \left(T_0 * \left(\frac{(s_4 + s_g)}{(s_5 + s_9)} \right) \right)$$

In Non-ideal gas any variable can be defined by two other dependent variable on them

$$a_{\text{non-ideal gas}} = f(x(b, c))$$

Table: 1 Variable Table (Pre-cooled Linde system)

Variable (a)	Gas	Variable (b)	Variable (c)
h_0	R\$	T_0	P_1
h_1	R\$	T_1	P_1
h_2	R\$	T_2	P_2
s_0	R\$	T_0	P_1
s_1	R\$	T_1	P_1
s_2	R\$	h_2	P_2
s_f	R\$	x_f	T_f
T_f	R\$	x_0	P_1
h_f	R\$	x_f	T_f
s_g	R\$	x_1	P_1
h_g	R\$	x_1	T_g
s_3	R\$	T_3	P_2
h_3	R\$	T_d	P_2
s_9	R\$	T_9	P_1
s_{10}	R\$	T_{10}	P_1
$cp(\text{fluid})_{HX1}$	R\$	T_2	P_2
$cp(\text{cf})_{HX1}$	R\$	T_8	P_1
C_{\min}	-	$C_{\text{hot}_{HX1}}$	$C_{\text{cold}_{HX1}}$
$cp(\text{hf})_{HX3}$	R\$	T_3	P_2
$cp(\text{cf})_{HX3}$	R\$	$T_g + 1$	P_1
C_{\min}	-	$C_{\text{hot}_{HX3}}$	$C_{\text{cold}_{HX3}}$
h_6	R\$	h_d	P_1
s_4	R\$	T_4	P_2
s_5	R\$	T_5	P_2

3. Result and Discussion

Fig.2 illustrates the variations in COP and second law efficiency with compressor pressure. It has been observed that methane has the highest COP and second law efficiency among the other gases, i.e. 1.615 at 40bar and 47.27% at 220 bar. Fig.3 illustrates the variations in liquefaction mass flow rate with the compressor pressure from 40 to 220bar. It has been noticed that methane has the highest liquefaction mass flow rate, i.e. 0.07248kg/s to 0.3444kg/s followed by other considered gases

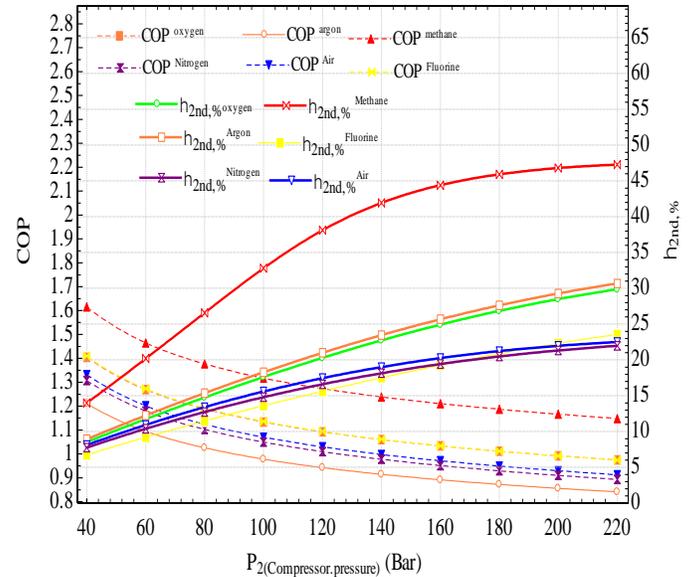


Figure 2: Variation in COP and second law efficiency with the compressor pressure

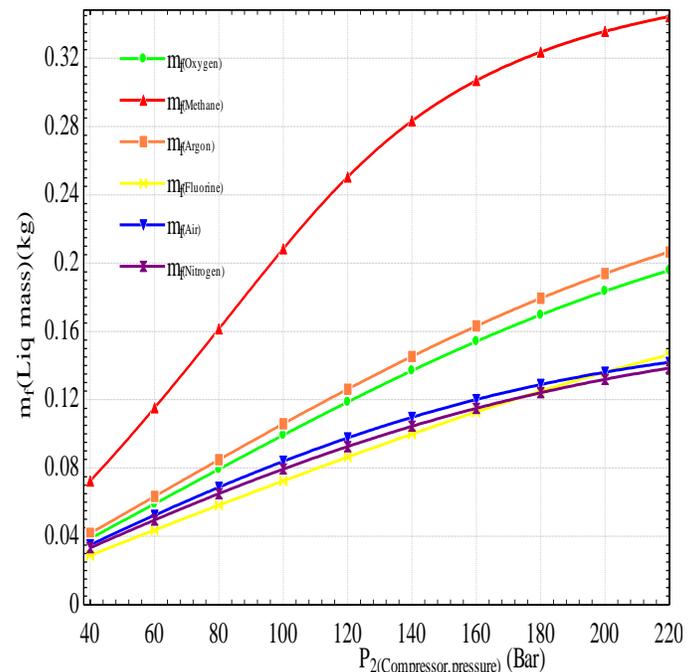


Figure 3: Variation in liquefaction mass flow with compressor pressure

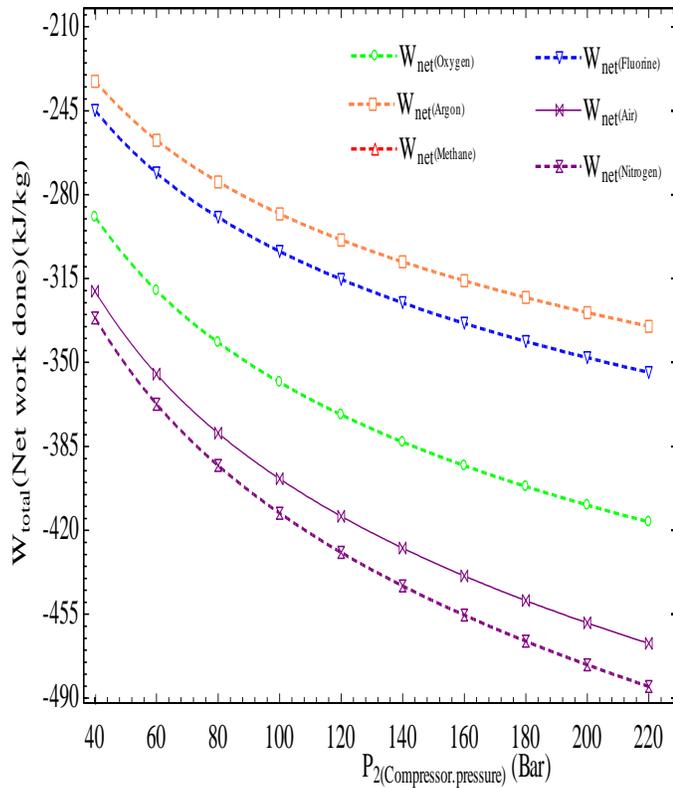


Figure 4: Variation in total net work done with compressor pressure

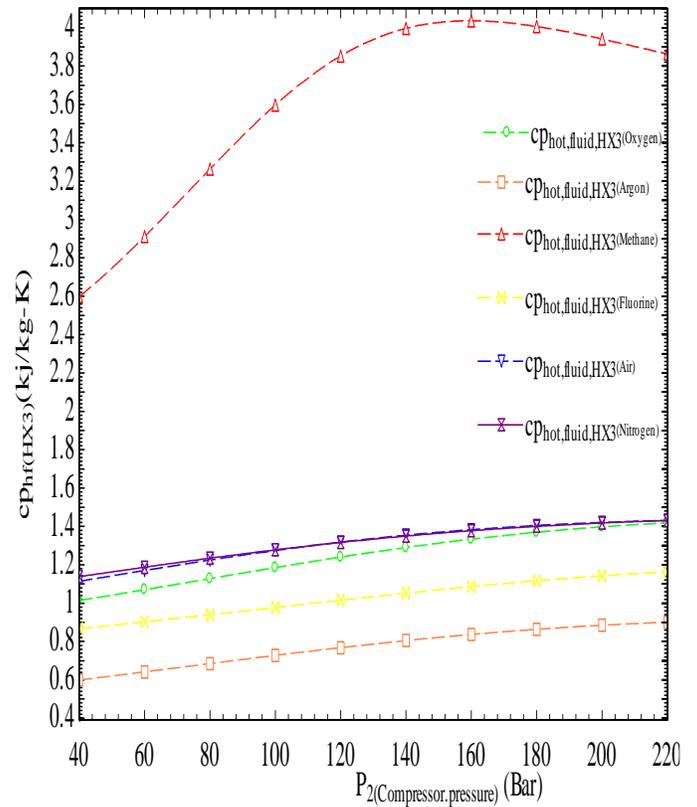


Figure 6: Variation in specific heat of hot fluid in HX3 with compressor pressure

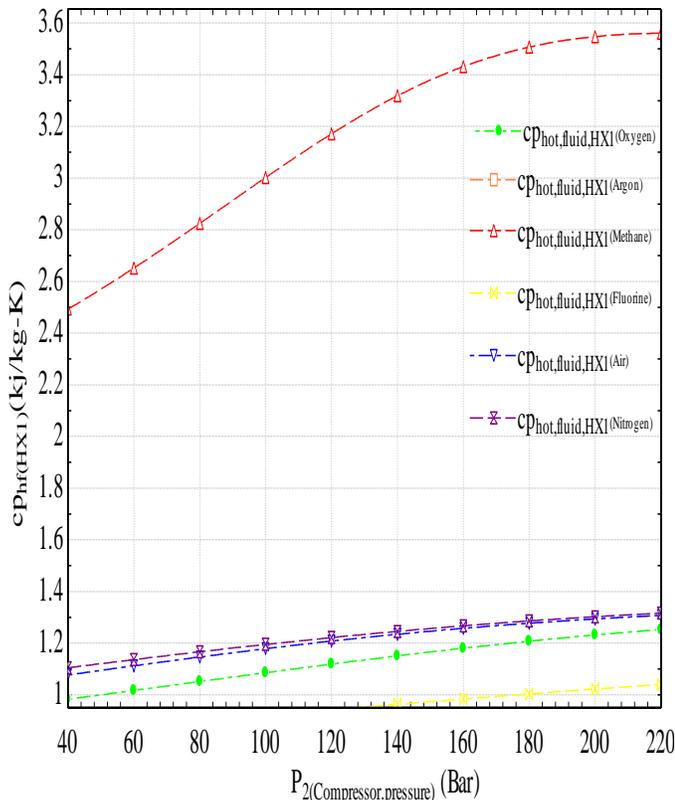


Figure 5: Variation in specific heat of hot fluid in HX1 with compressor pressure

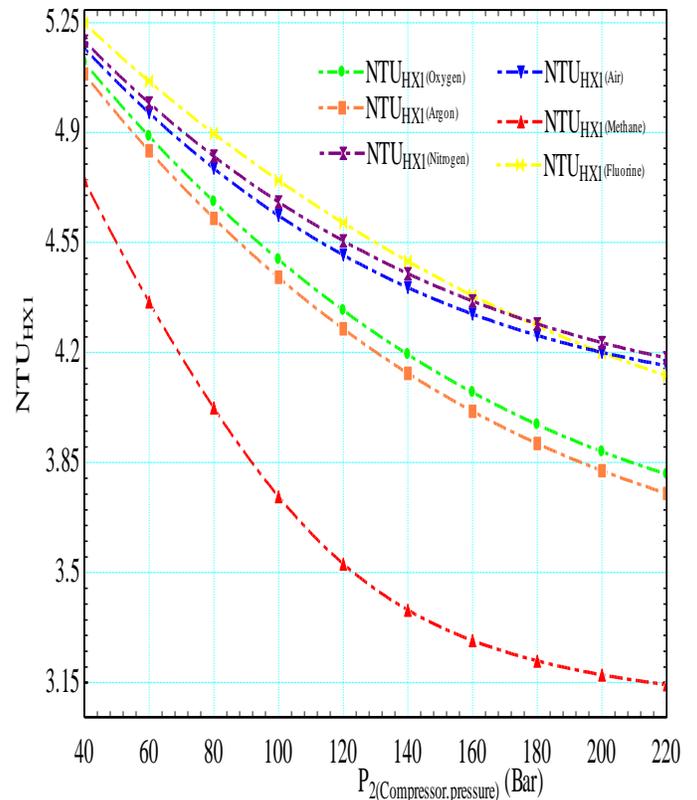


Figure 7: Variation in NTU in HX1 with compressor pressure

Table 2(a) Various performance parameters on optimum Temperature (300K) and P₂ (100 bar)

Gases	COP	Eff. η _{2nd} %	Edest (HX1) (kj/kg)	Edest (HX3) (kj/kg)
Oxygen	1.136	17.54	2.38	39.48
Argon	0.9814	18.21	2.424	31.58
Methane	1.321	32.65	9.414	87.63
Fluorine	1.139	13.54	1.296	49.8
Air	1.072	15.52	2.368	190
Nitrogen	1.051	14.73	2.086	9.818

Table 2(b) Various performance parameters on optimum Temperature (300K) and P₂ (100 bar)

Gases	COP	Efficiency η _{2nd} %	m_f kg/s	FOM
Oxygen	1.136	17.54	0.09928	0.566
Argon	0.9814	18.21	0.1059	0.5814
Methane	1.321	32.65	0.2084	0.2084
Fluorine	1.139	13.54	0.07244	0.5349
Air	1.072	15.52	0.0839	0.5405
Nitrogen	1.051	14.73	0.07938	0.5387

Table 2(c) Various performance parameters on optimum Temperature (300K) and P₂ (100 bar)

Gases	COP	Efficiency η _{2nd} %	NTU (HX1)	NTU (HX3)
Oxygen	1.136	17.54	4.496	3.518
Argon	0.9814	18.21	4.439	3.387
Methane	1.321	32.65	3.742	2.718
Fluorine	1.139	13.54	4.748	3.757
Air	1.072	15.52	4.636	3.956
Nitrogen	1.051	14.73	4.679	3.911

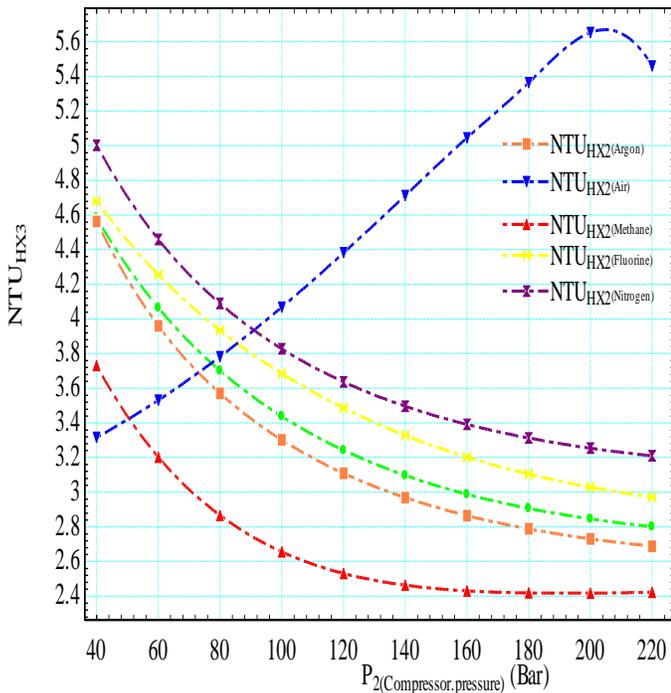


Figure 8: Variation in NTU in HX3 with compressor pressure

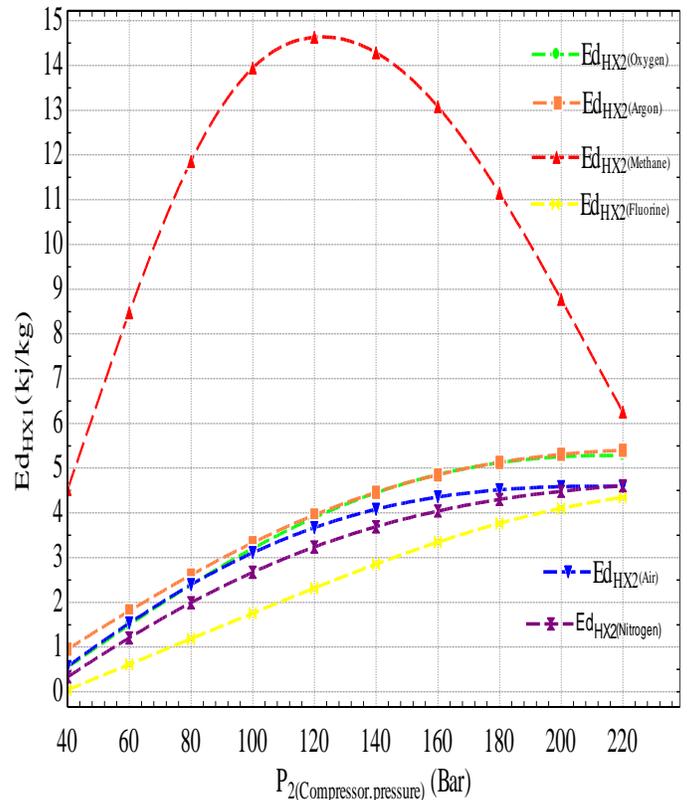


Figure 9: Variations in exergy destruction in HX1 with compressor pressure

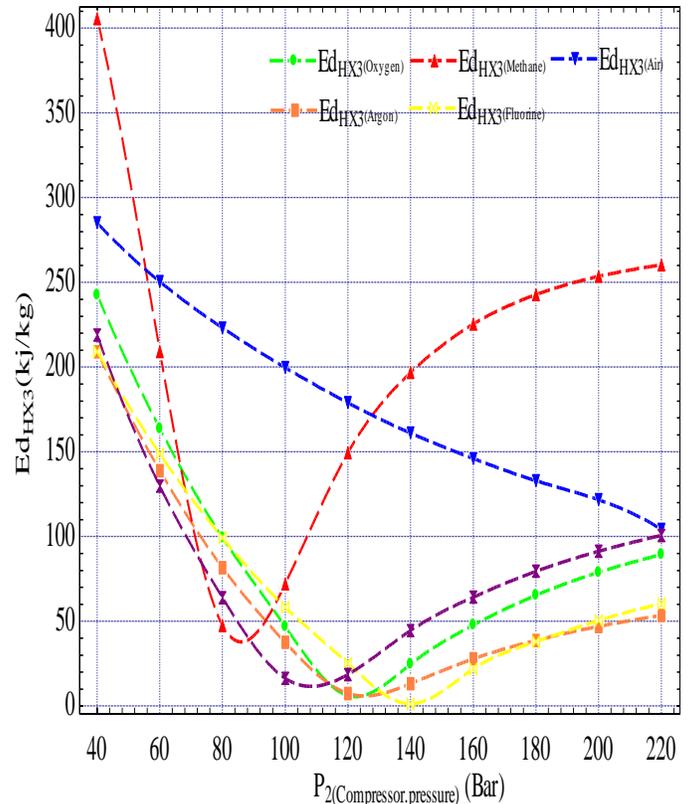


Figure 10: Variation in exergy destruction in HX3 with compressor pressure

The optimum performance parameters at optimum temperature at optimum pressure are shown in Table-2 respectively.

Fig.4 shows the variation in net work done with the compressor pressure and it has been observed that total net work done decreases with the compressor pressure, and argon has the highest value among other gases. Fig.5 illustrates the specific heat of hot fluid in HX1 and methane has the largest value of specific heat among the other gases i.e. 3.561kJ/kg-K at 220bar. Fig.6 demonstrates that methane has the highest value of specific heat of hot fluid in HX3 among the other considered gases, i.e. 4.036Kj/kg-K at 160bar. Fig.7 shows that NTU in HX1 continuously decreasing and fluorine has the highest value of NTU among the other gases, i.e. 4.747 at 40bar. Fig.8 shows the NTU in HX3 and it has been seen that all the gases has the decreasing trend except air between the prescribed the temperature range of 40bar to 220bar. Fig.9 illustrates the exergy destruction rate in HX1 and it has been seen that methane shows the highest value of exergy destruction rate, and its graph first increasing then shows the decreasing trend. Alternatively, all the other gases shows the increasing trend. Fig.10 shows the exergy destruction rate in HX3 and it has been seen that methane has the highest value, and it is decreasing first up to a certain pressure and then increasing followed by other considered gases

4. Conclusions

- Methane has the highest first law efficiency (i.e. COP) and second law efficiency (i.e. exergetic Efficiency) among the other gases.
- Methane has the highest liquefaction mass flow rate, followed by other considered gases.
- The total net work done decreases with the compressor pressure, and argon has the highest value among other gases.
- NTU in the third heat exchanger (HX3) for all gases has the decreasing order.
- Methane shows the highest value of Exergy destruction rate in first heat exchanger (HX1).
- Methane has the highest value of specific heat of hot fluid in HX3 among the other considered gases.
- Exergy destruction rate of methane in the third heat exchanger (HX3) has the highest value, and it is decreasing first up to a certain pressure and then increasing followed by other considered gases.
- Total net work done decreases with the compressor pressure, and argon has the highest value among other gases.

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