

Figure 1(b): T-S diagram of Dual compressor Linde system

2. Use of entropy generation method for computing exergy for finding irreversibility in the system

Whole part of heat energy can never be converted completely into work, there some part of energy which used and second which get waste, the useful part of energy that is able to convert into heat is called available energy or exergy and unavailable part which get destroyed is called unavailable energy or exergy. As the first law of thermodynamics state that the energy is always conserved but the content of that energy which is capable of producing useful work is not constant that is exergy. The maximum useful work or exergy at a particular state is a composite property depending upon the state of system and surrounding. A dead state having Zero exergy that is equilibrium state. The exergy analysis allows us to identify and quantify the sites with the losses of exergy, and therefore showing the direction for the minimization of exergy losses to approach the reversible COP.

R. Agrawal, et.al [1] carried out exergy analysis for efficient cryogenic nitrogen generators: Gadhiraaju Venkatarathnam [2], also carried out Simulation of cryogenic processes, J. Rizk, M. et.al, [3] carried out exergy optimization of a cryogenic air separation unit, Yasuki Kansha, et.al [4], developed novel cryogenic air separation process based on self-heat recuperation. R.L. Cornelissen, [5], carried out the energy-exergy analysis of cryogenic air separation system.

From literature it noticed that exergy efficiency depend upon mainly upon the inlet condition of the system but which inlet condition best suit for a particular type of the system except to increase the whole system efficiency stress are done on particular parts of system From the literature review, it conclude that every part of system has its own and equal importance because ones effect on another whether it is small

or big create a lot of difference in proper analysis of system. Ignoring one small system due less effect can put research gap in complete thermodynamic analysis of system that why it quite important take all parts of system as one and finding out the every part impact on another to calculate right equation for high output. Therefore following objectives of present investigations are thermodynamic (energy-exergy) analysis of considered systems and finding exergy destruction in each and there individuals components and to suggestion for reducing exergy destruction losses in whole systems and there components. The effects of pressure ratio and gas outlet temperature of compressor on various energy- and exergy-based performance parameters are investigated considering all six gases as the gas being liquefied. Mathematical

3. Modelling of Dual Compressor Linde System

$$\begin{aligned}
 R\$ &= \text{' Gas' } \\
 m_4 &= 1 \\
 m_4 &= m_7 + m_{12} \\
 T_0 &= 298 \\
 T_1 &= 300 \text{ K} \\
 T_4 &= T_3 \\
 T_3 &= \frac{T_2 + T_{13}}{2} \\
 P_1 &= 1.013 \\
 P_2 &= \frac{P_4}{5} \\
 P_4 &= 80
 \end{aligned}$$

3.1 Analysis of Compressor

$$\begin{aligned}
 Q_1 &= m_{10} * (h_2 - h_1) \quad (12) \\
 W_{c1} &= \left(m_{10} * ((h_2 - h_1) - T_2 * (s_2 - s_1)) \right) \\
 Ed_{comp1} &= \left(m_{10} * T_2 * (s_1 - s_2) - \left(Q_1 * \left(\frac{T_0}{T_2} \right) \right) \right) \\
 Q_2 &= m_4 * (h_4 - h_3) \\
 W_{c2} &= \left(m_4 * ((h_4 - h_3) - T_4 * (s_4 - s_3)) \right) \\
 Ed_{comp2} &= \left(m_4 * T_4 * (s_3 - s_4) - \left(Q_2 * \left(\frac{T_0}{T_4} \right) \right) \right) \\
 W_{Net} &= W_{c2} + W_{c1} \\
 COP &= \frac{h_1 - h_{10}}{W_{Net}}
 \end{aligned}$$

3.2 Analysis of heat exchanger

$$\begin{aligned}
 P_5 &= P_4 \\
 m_h &= m_4 \\
 m_c &= m_{12} \\
 C_h &= m_h * cp_{hot\ fluid_{HX}} \\
 C_c &= m_c * cp_{Cold\ fluid_{HX}} \\
 q_{hx} &= C_h * (T_4 - T_5) \\
 q_{hx} &= C_c * (T_{13} - T_{12})
 \end{aligned}$$

"TypeHX\$ = ' counterflow'"

$$q_{max} = C_{min} * (T_4 - T_{12})$$

$$\epsilon = \frac{q_{HX}}{q_{max}}$$

$$\epsilon = .85$$

$$Ntu = HX(\text{TypeHX}\$, \epsilon, C_h, C_c, 'Ntu')$$

$$Ex_{in_{HX}} = m_4 * ((h_4 - h_5) - (T_0 * (s_4 - s_5)))$$

$$Ex_{out_{HX}} = m_{12} * ((h_{12} - h_{13}) - (T_0 * (s_{12} - s_{13})))$$

$$Ed_{HX} = ((Ex_{in_{HX}}) - (Ex_{out_{HX}})) \quad (33)$$

3.3 Thermodynamic Analysis of Valve

$$h_5 = h_6$$

$$Ex_{in_{val}} = (h_5 - h_0) - T_0 * (s_5 - s_0)$$

$$Ex_{out_{val}} = (h_6 - h_0) - T_0 * (s_6 - s_0)$$

$$Ed_{val1} = (Ex_{in_{val}} - Ex_{out_{val}})$$

"Analysis of seperator"

$$m_4 * h_6 = ((m_7 * h_7) + (m_{12} * h_{12}))$$

$$x_7 = 0$$

$$x_{12} = 1$$

$$Ed_{sep1} = \left(T_0 * \left(\frac{((m_{12} * s_{12}) - (m_4 * s_6)) + \left(\frac{(m_{12} * h_{12}) - (m_7 * h_7)}{T_{12}} \right)}{T_{12}} \right) \right)$$

3.4 Thermodynamic Analysis of heat exchanger

$$m_{12} * h_{12} + m_{10} * h_{10} = m_{12} * h_{13} + m_{10} * h_{11}$$

"Analysis of Valve"

$$h_7 = h_8$$

$$Ex_{in_{val2}} = (h_7 - h_0) - T_0 * (s_7 - s_0)$$

$$Ex_{out_{val2}} = (h_8 - h_0) - T_0 * (s_8 - s_0)$$

$$Ed_{val2} = \text{abs}(Ex_{in_{val2}} - Ex_{out_{val2}})$$

3.5 Thermodynamic Analysis of separator

$$m_7 * h_8 = ((m_9 * h_9) + (m_{10} * h_{10}))$$

$$m_7 = m_9 + m_{10}$$

$$m_9 = m_f$$

$$m_{10} = m_g$$

$$x_9 = 0$$

$$x_{10} = 1$$

$$Ed_{sep2} = \text{abs} \left(T_0 * \left(\frac{((m_{10} * s_{10}) - (m_7 * s_8)) + \left(\frac{(m_{10} * h_{10}) - (m_9 * h_9)}{T_{10}} \right)}{T_{10}} \right) \right)$$

$$Ed_{comp1\%} = \left(\frac{Ed_{comp1}}{Ed_{DualCompLinde}} \right) * 100$$

$$Ed_{comp2\%} = \left(\frac{Ed_{comp2}}{Ed_{DualCompLinde}} \right) * 100$$

$$Ed_{HX\%} = \left(\frac{Ed_{HX}}{Ed_{DualCompLinde}} \right) * 100$$

$$Ed_{val1\%} = \left(\frac{Ed_{val1}}{Ed_{DualCompLinde}} \right) * 100$$

$$Ed_{sep1\%} = \left(\frac{Ed_{sep1}}{Ed_{DualCompLinde}} \right) * 100$$

$$Ed_{val2\%} = \left(\frac{Ed_{val2}}{Ed_{DualCompLinde}} \right) * 100$$

$$Ed_{sep2\%} = \left(\frac{Ed_{sep2}}{Ed_{DualCompLinde}} \right) * 100$$

$$Eta_{2nd\%} = \text{abs} \left(\left(m_9 * \frac{(h_9 - h_1) - T_0 * (s_9 - s_1)}{W_{Net}} \right) * 100 \right)$$

$$Ed_{DualCompLinde} = Ed_{comp1} + Ed_{comp2} +$$

$$Ed_{HX} + Ed_{val1} + Ed_{val2} + Ed_{sep1} + Ed_{sep2}$$

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

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

Table 1: Variable Table (Dual Compressor 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_3	R\$	T_3	P_2
h_3	R\$	T_3	P_2
s_4	R\$	h_4	P_2
h_4	R\$	T_4	P_2
$cp(hf)_{HX}$	R\$	T_4	P_4
$cp(cf)_{HX}$	R\$	T_{13}	P_2
C_{min}	-	$C_{hot_{HX}}$	$C_{cold_{HX}}$
h_7	R\$	X_7	P_2
s_7	R\$	X_7	P_2
s_6	R\$	h_6	P_2
X_6	R\$	h_6	P_2
X_8	R\$	h_8	P_1
T_6	R\$	h_6	P_2
s_5	R\$	h_5	P_4
T_5	R\$	h_5	P_4
T_8	R\$	h_8	P_1
s_8	R\$	h_8	P_1
h_9	R\$	X_9	P_1
s_9	R\$	X_9	P_1
h_{10}	R\$	X_{10}	P_1
s_{10}	R\$	h_{10}	P_1
T_{10}	R\$	h_{10}	P_1
h_{12}	R\$	X_{12}	P_2
s_{12}	R\$	h_{12}	P_2
T_{12}	R\$	h_{12}	P_2
h_{13}	R\$	T_{13}	P_2
s_{13}	R\$	T_{13}	P_2

The effects of pressure ratio and gas outlet temperature of compressor on various energy- and exergy-based performance parameters are investigated considering all six gases as the gas being liquefied.

4. Result and Discussion

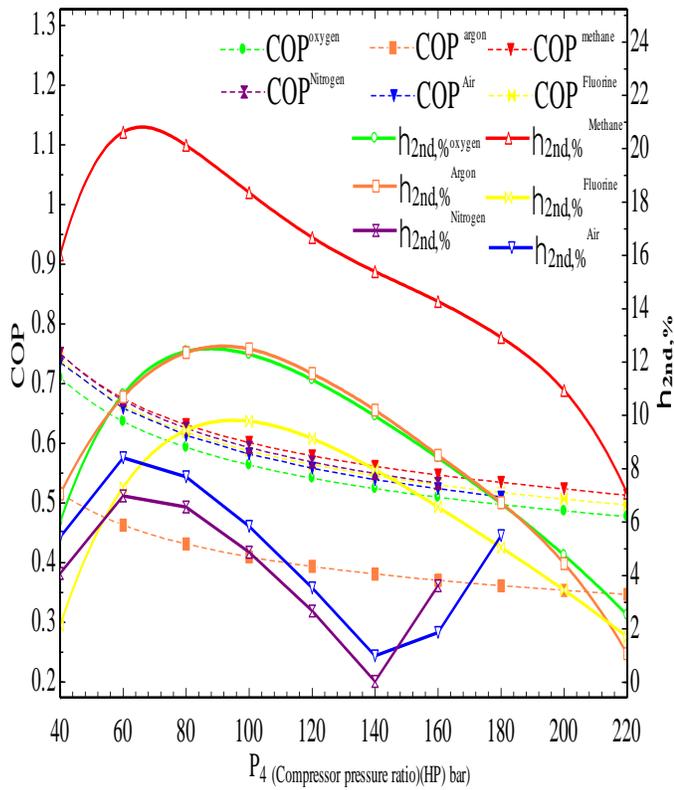


Figure 2(a): COP and second law efficiency versus high pressure compressor ratio

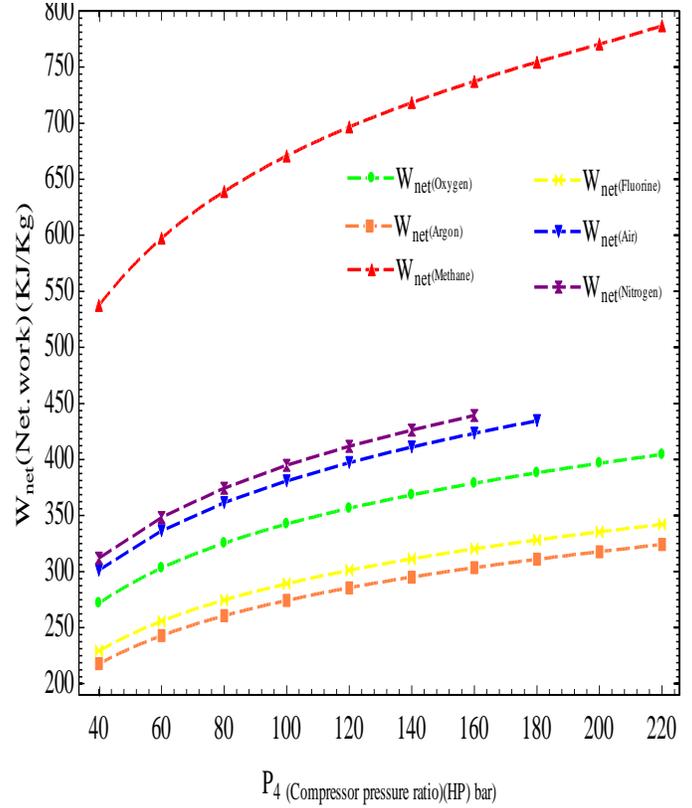


Figure 3: Net work done versus high pressure compressor ratio

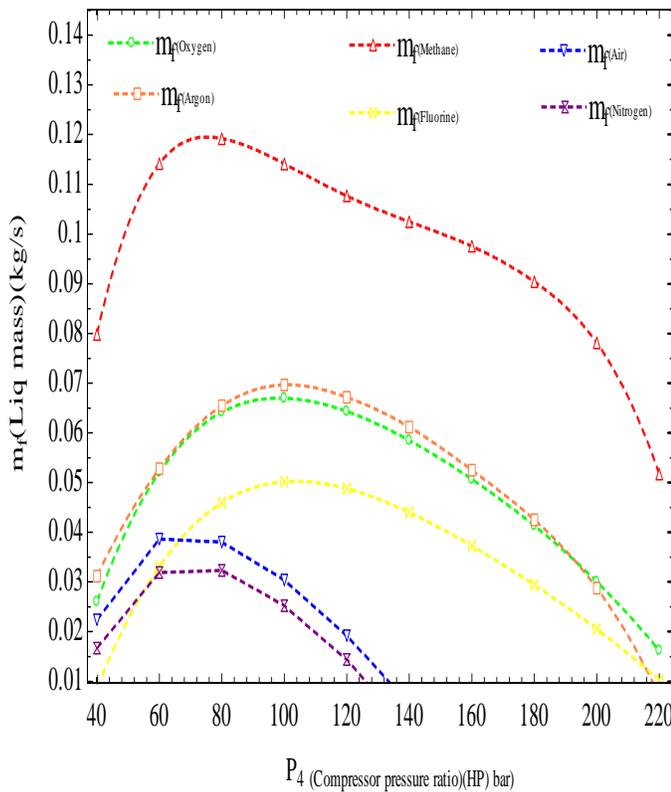


Figure 2(b): Liquefaction rate versus high pressure compressor Ratio

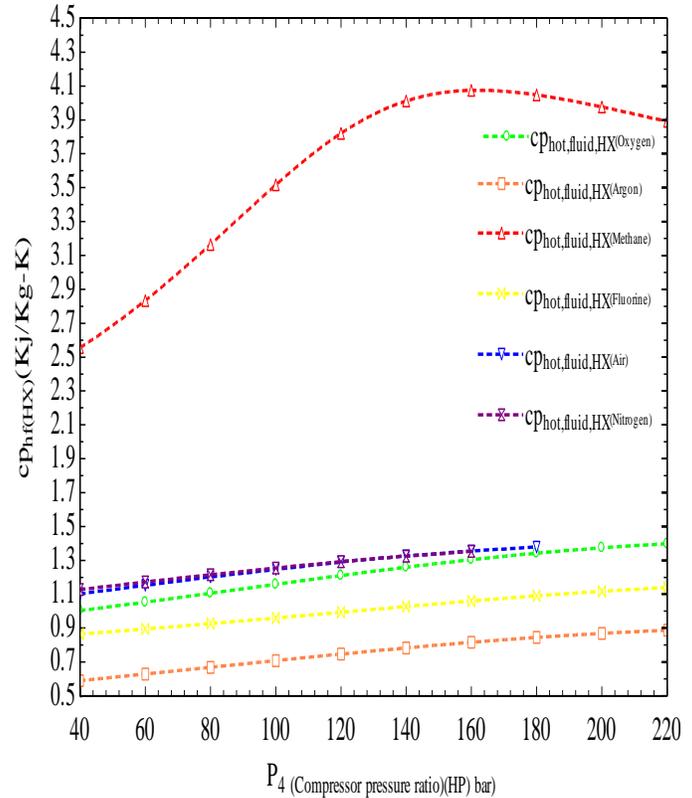


Figure 4: Specific heat of hot fluid of heat exchanger versus high pressure compressor ratio

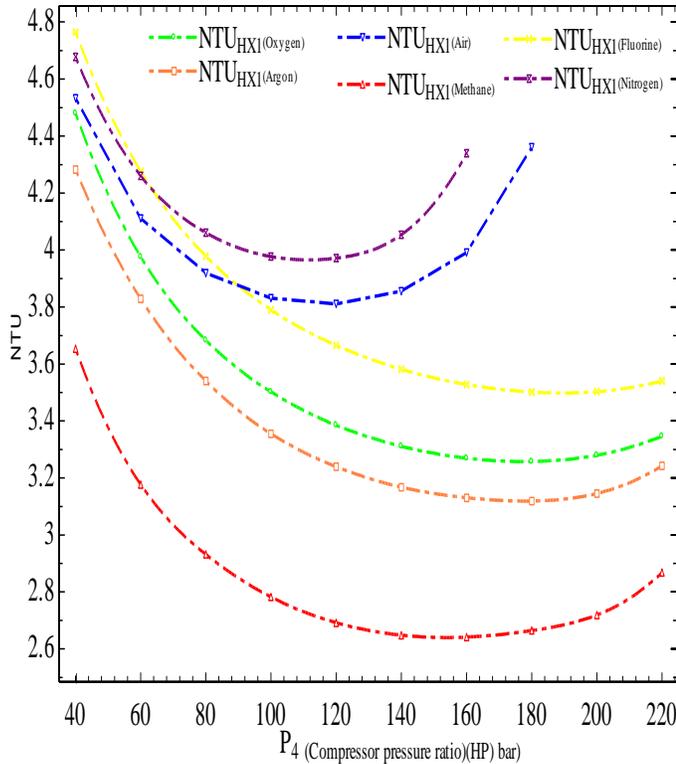


Figure 5: NTU versus high pressure compressor ratio

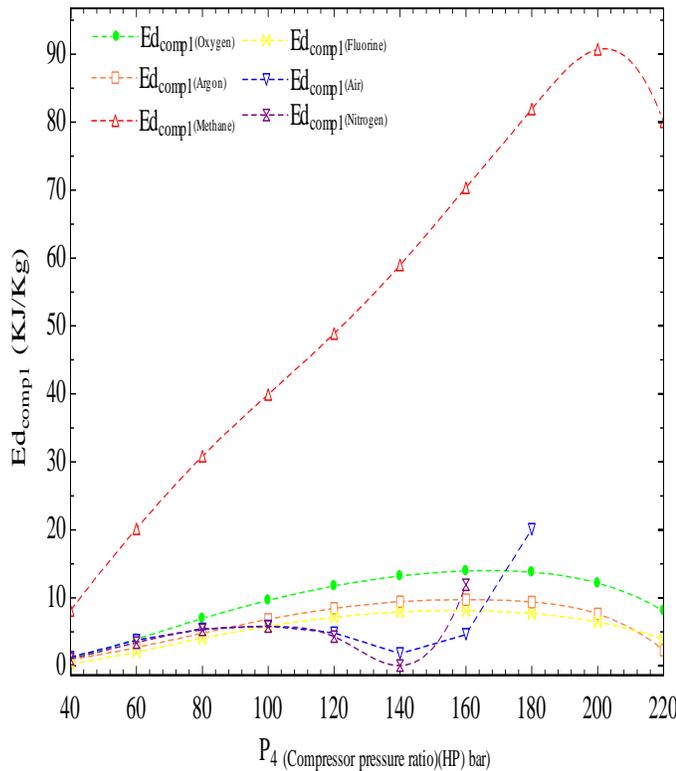


Figure 6: Exergy destruction rate versus high pressure compressor ratio

In fig.2 the optimum pressure ratio 0.4 gases like nitrogen, air and methane is 60PR while for the fluorine oxygen, argon it is 80PR the air nitrogen second law efficiency first increasing up to 60PR but after further increase in pressure ratio it show decrement up to 140PR. While other gases achieving after their optimum PR they show continuous decrement or fall in second law efficiency. While in COP case all gases show decrease in COP by increasing the PR. In fig.3 the liquefaction rate start decreasing after achieving their optimum PR but gases like air and nitrogen show huge decrease in liquefaction rate but after 140PR they show almost negligible rate of liquefaction fig. 4 show net work done for liquefaction of gases with increase in pressure ratio. It is very important factor to know how much work done is required for liquefaction to cost optimization and system design parameters. From thermodynamic analysis, it depicts that almost all gases show increase in work done as the pressure ratio of the system increases. For Dual Linde compressor system argon show least work requirement while gases like methane and nitrogen require highest work done for liquefaction specific heat of hot side fluid play very important role in analysis part of system, In fig.4 show the specific heat of all gases show increment with increase in PR while specific heat in case of methane gas show first increase up to 160PR. Then it starts decreasing with in very less marginal rate.

Number of transfer unit (NTU) help in the design of heat exchanger, the size of heat exchanger can be predicting by assessing the NTU of heat exchanger. Fig. 5 show NTU variation of various gases with respect to increasing PR. Except air and nitrogen all for gases first decreases up to 120PR then starts increasing at a very marginal rate by further increasing in PR.

Fig.6 in dual compressor system the destruction in high pressure and low pressure are different at the PR140 the exergy destruction rate is minimum for five gases except methane in these gases the destruction first increases up to 160PR and then decreases. Whereas in gases like air and nitrogen show very sharp decrement at 140PR methane gas show a straight slope of increase in exergy destruction up to peak at 200PR then further increase in PR show decrease in exergy destruction. Fig. 7 show high pressure compressor exergy destruction rate with respect to PR.

The exergy destruction is continuously increasing for all six gases with increase of PR the trend of exergy destruction in heat exchanger is shown in fig. 8 the air and nitrogen gas show highest rate of destruction among other gases.

The destruction rate for said gases first decrease up to 100PR and then increase again by further increase in PR. Methane gas show very sharp decrement in exergy destruction up to 160PR then became constant and start rise at very low rate at 200PR. Argon show lowest exergy destruction rate in heat exchanger, while fluorine and oxygen are continuous decrease by increasing PR and become constant in range of 140 to 220PR

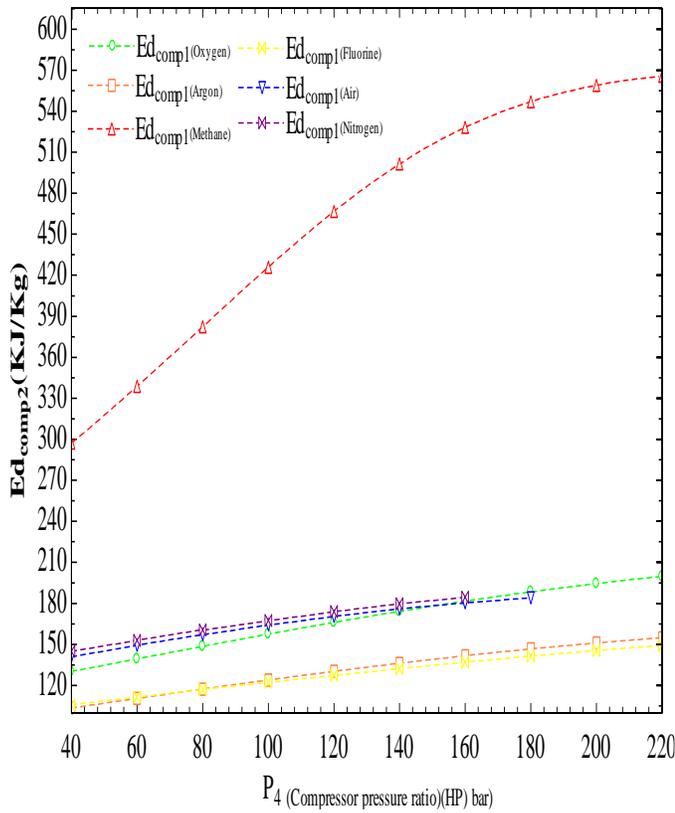


Figure 7: Exergy Destruction rate of High pressure compressor versus high pressure compressor ratio

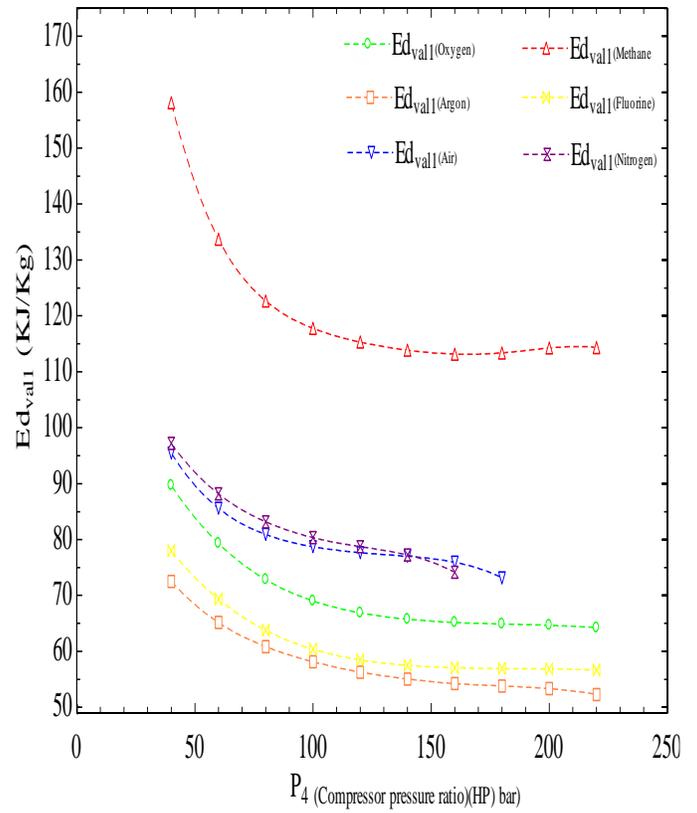


Figure 9: Exergy destruction of expansion valve versus high pressure compressor ratio

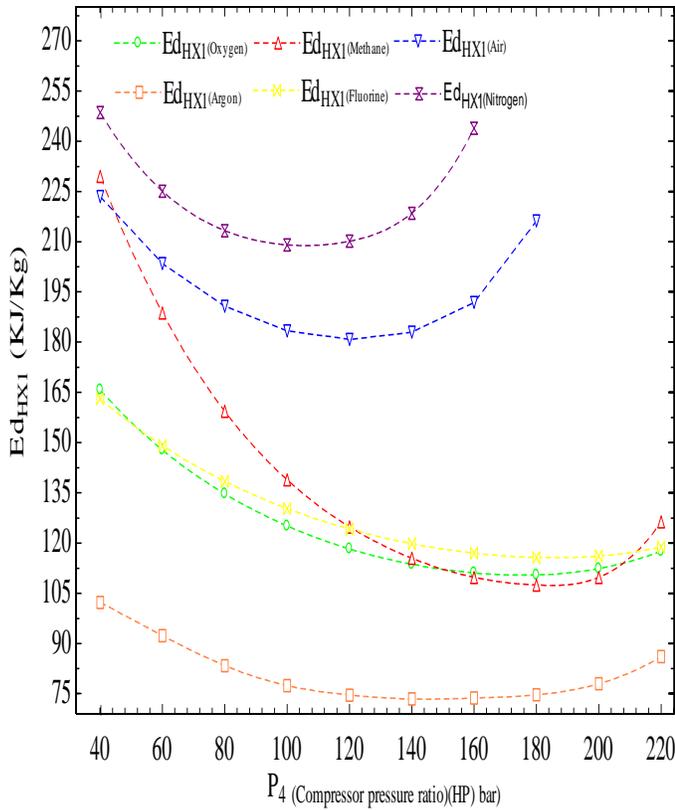


Figure 8: Exergy destruction rate of heat exchanger versus high pressure compressor ratio

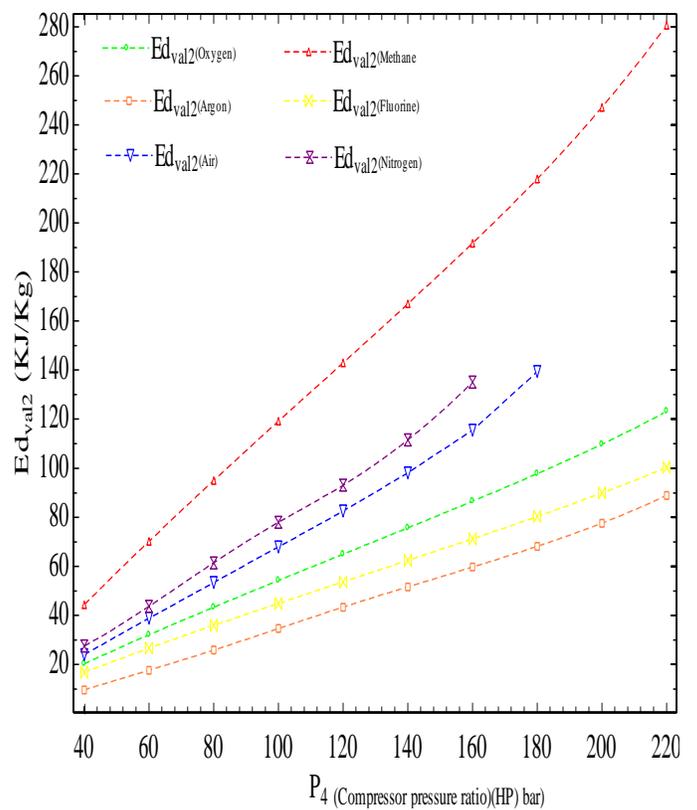


Figure 10: Exergy Destruction rate of second expansion valve versus compressor pressure ratio

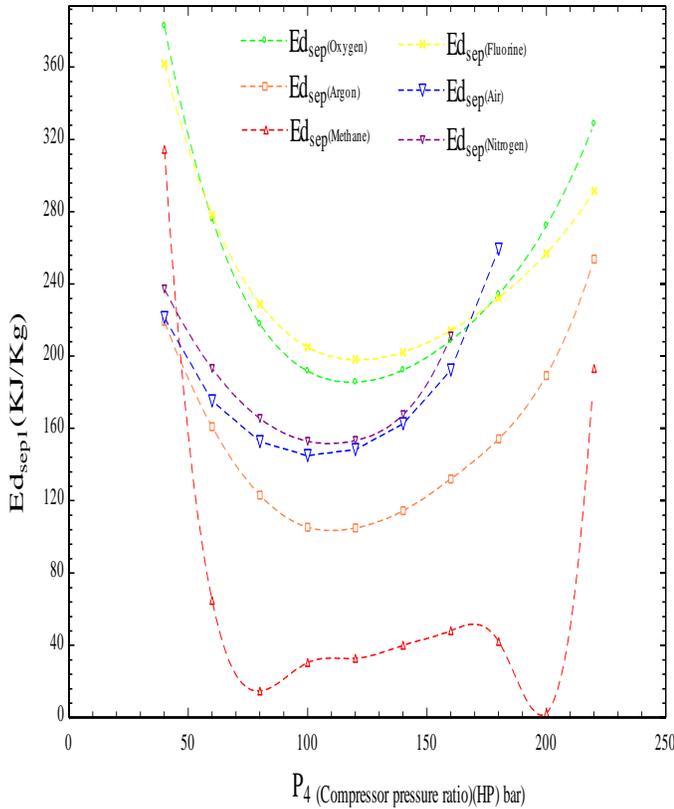


Figure 11: Exergy destruction rate of first separator versus high pressure compressor ratio

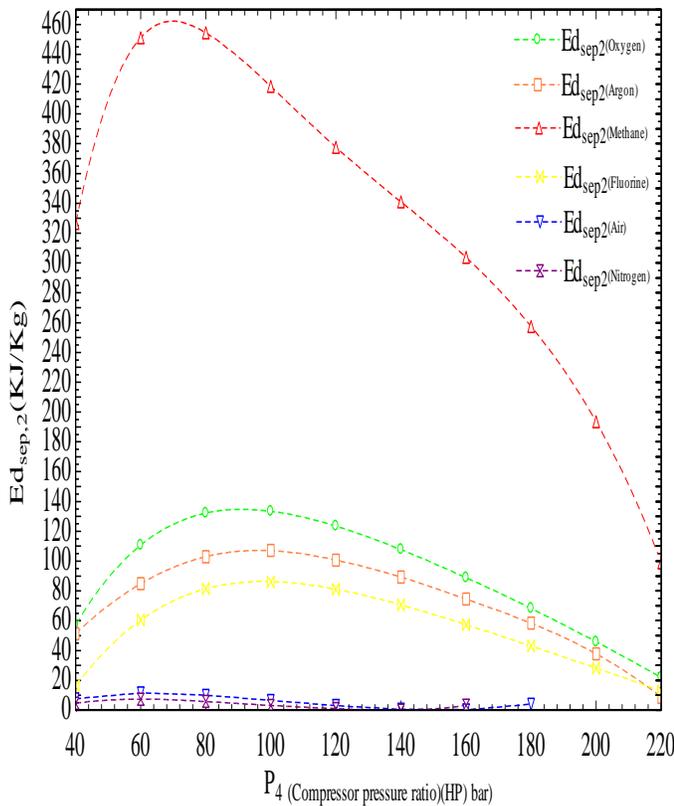


Figure 12 Exergy destruction rate of second separator versus high pressure compressor ratio

Dual compressor system contain two expansion valve (V1 and V2), Valve V1 work on the high temperature while V2 is work on at lowest temperature. Fig. 9 and 10 show exergy destruction rate of gases with respect to PR (Pressure ratio) the destruction in valve 1 the destruction rate for gases first decrease up to 80 PR then they become constant, while on the other hand the lowest temperature working valve V2 the destruction rate of exergy is high and it increase with increase in PR of system. Methane gas in both valve show highest rate of destruction among all six gases. Fig. 11 & 12 show exergy destruction rate in separator 1 and 2. Dual compressor system having two separator at different temperature level. In separator 1, the trend of exergy destruction for five gases except methane show decreasing trend in the range of 40-100 PR but after this range the exergy destruction in separator start increasing again in methane gas case separator 1 show unusual behaviour it decrease at very fast rate and become minimum at 80PR then increase again with small rate up to 180PR then again decrease, At 200PR it show almost negligible destruction for methane. In separator 2 the rate of destruction increase up to 80PR for all six gases and then decreases in further increases in PR.

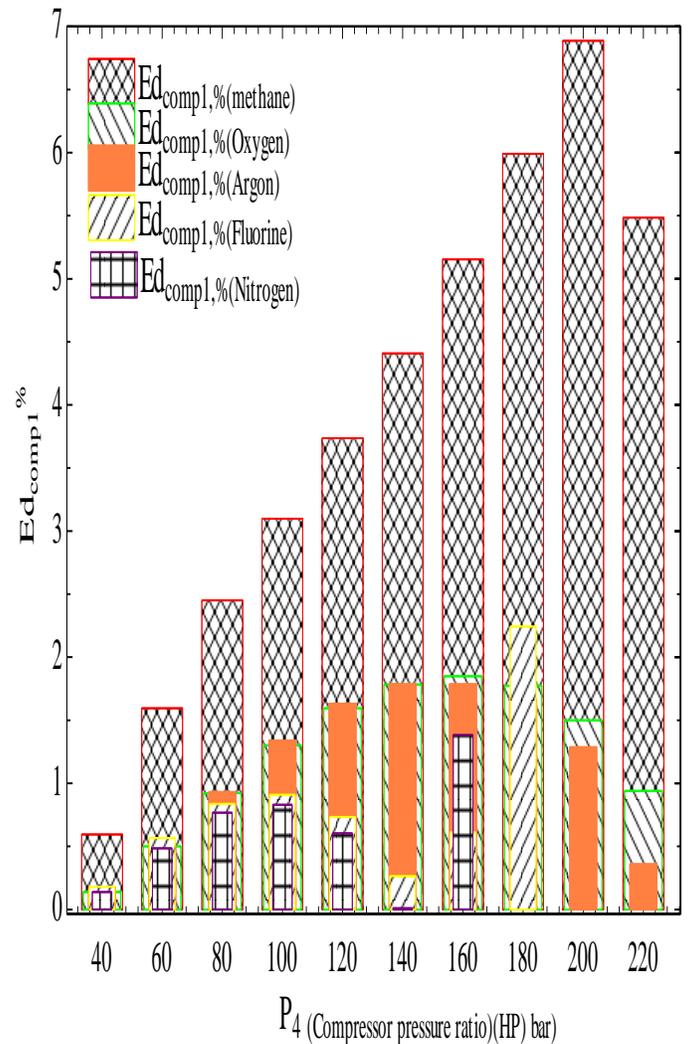


Figure 13: Exergy destruction of Compressor 1 % versus high pressure compressor ratio

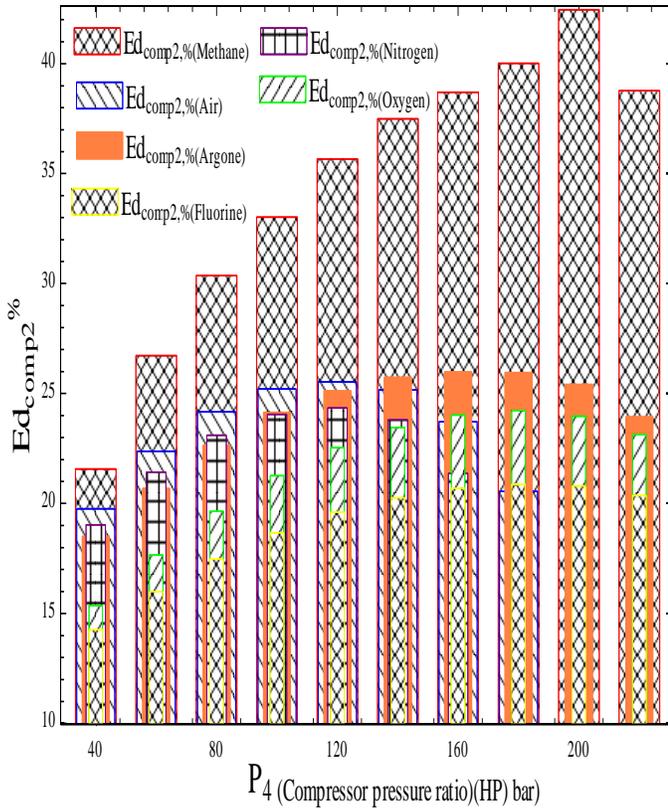


Figure 14: Exergy destruction of Compressor 2 % versus high pressure compressor ratio

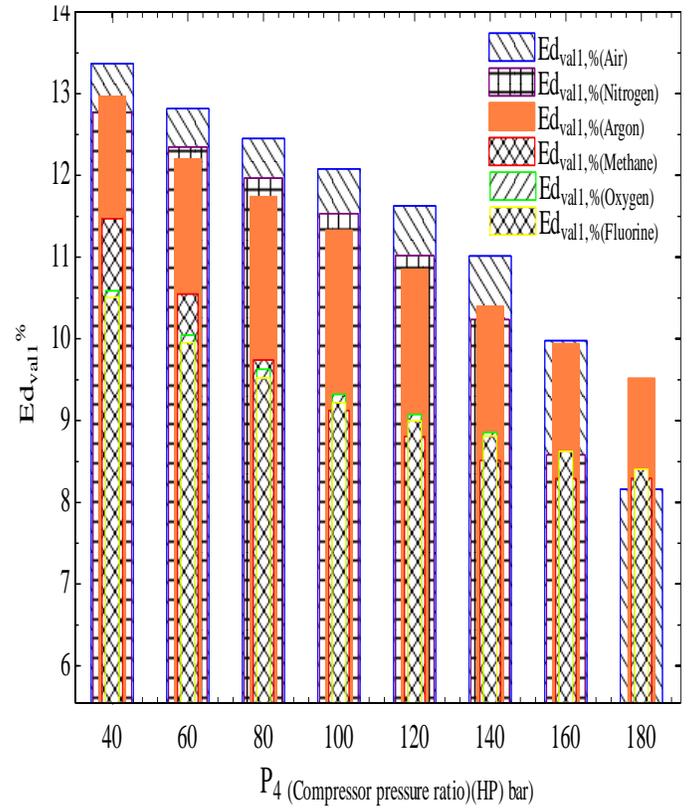


Figure 16: Exergy destruction of expansion valve 1 % versus high pressure compressor ratio

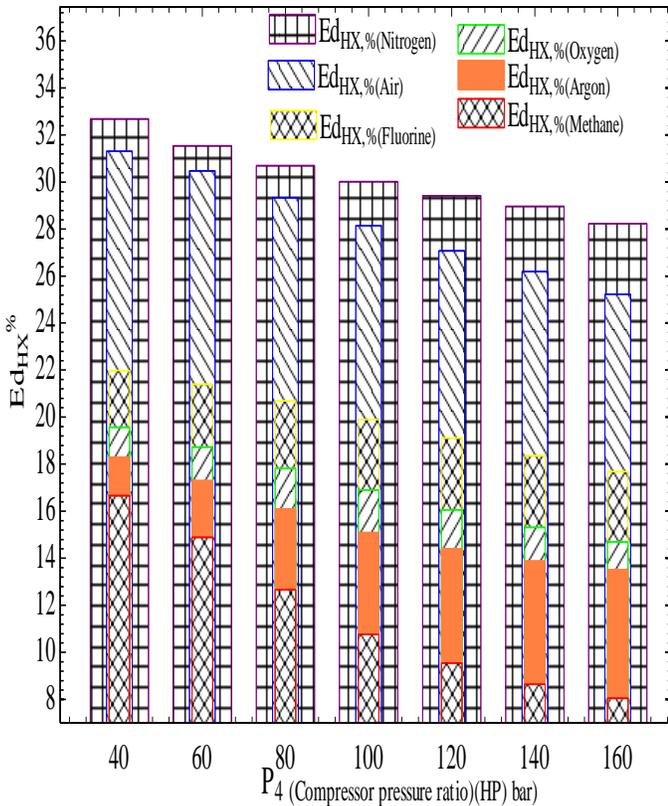


Figure 15: Exergy destruction of heat exchanger 1% versus high pressure compressor ratio

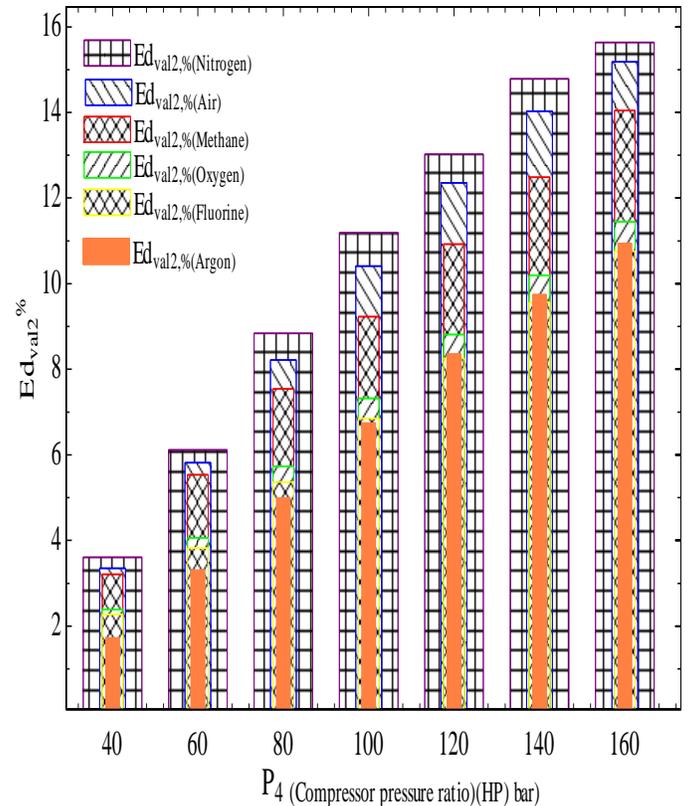


Figure 17: Exergy destruction of expansion valve 2 % versus high pressure compressor ratio

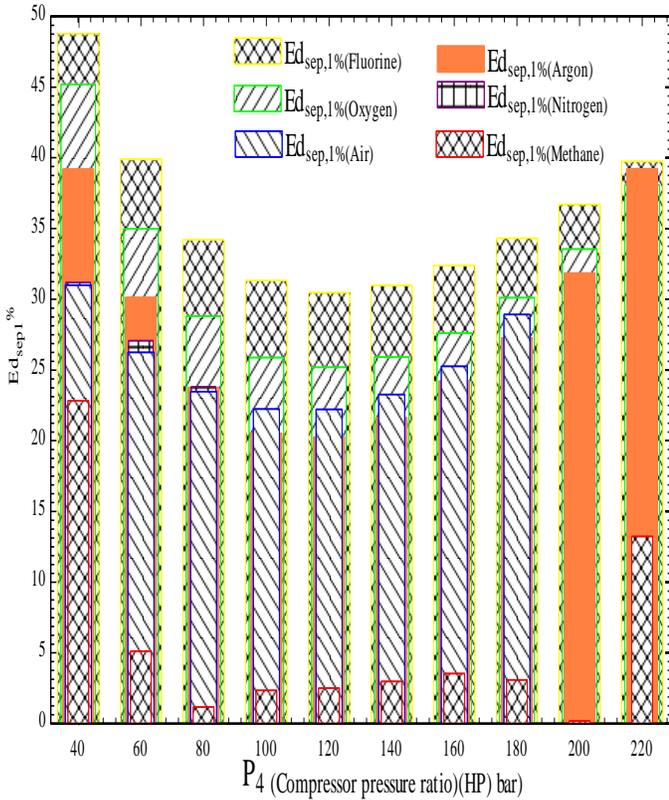


Figure 18: Exergy destruction of Separator 1 % versus high pressure compressor ratio

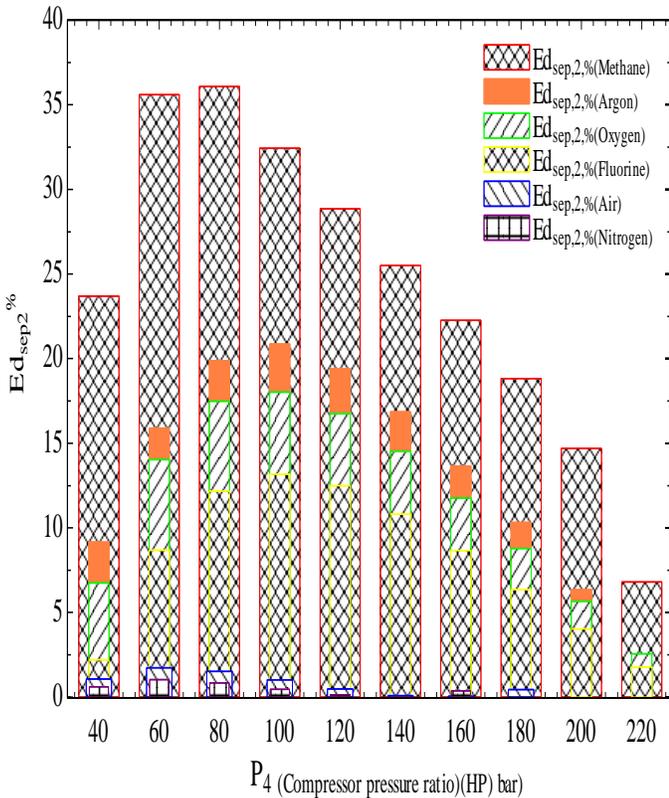


Figure 19: Exergy destruction of separator 2 % versus high pressure compressor ratio

From Fig.13-19 show the exergy destruction in percentage form of every component with six gases for easy understanding. In fig.13 show methane has highest rate of destruction followed by oxygen, argon, fluorine and nitrogen. While, in fig 14 compressor 2 methane has highest oxygen show lowest percentage of exergy destruction. In heat exchanger nitrogen has highest followed by air, fluorine, oxygen and methane shown in fig 15. Fig. 16-17 the valve 1 show air is the highest percentage destruction and fluorine has the least value. In fig.18 separator 1 show fluorine has a highest destruction of exergy, while methane has the least value in the PR range of 40 to 80. Fig.19 show separator 2 percentage destruction form in this methane gas show highest and nitrogen has the least value.

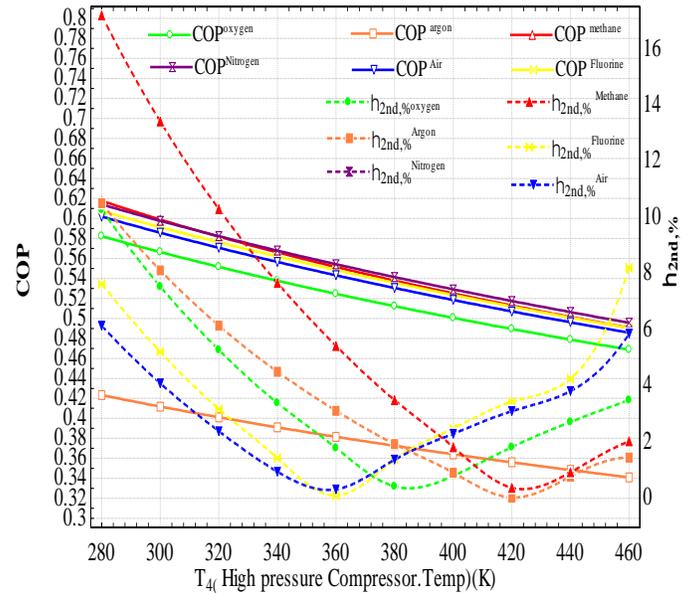


Figure 20: COP and second law efficiency versus high pressure compressor temperature

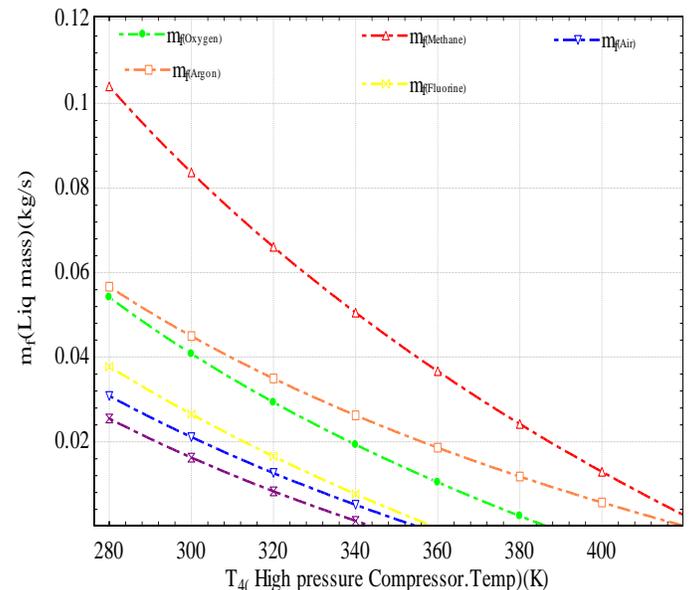


Figure 21: Liquefaction rate versus high pressure compressor temperature

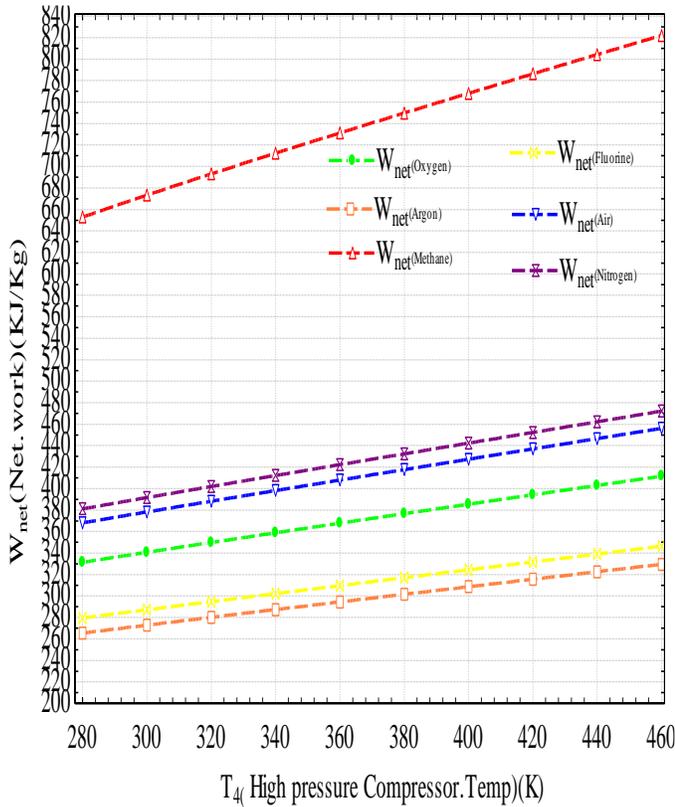


Figure 22: Net work done versus high pressure compressor temperature

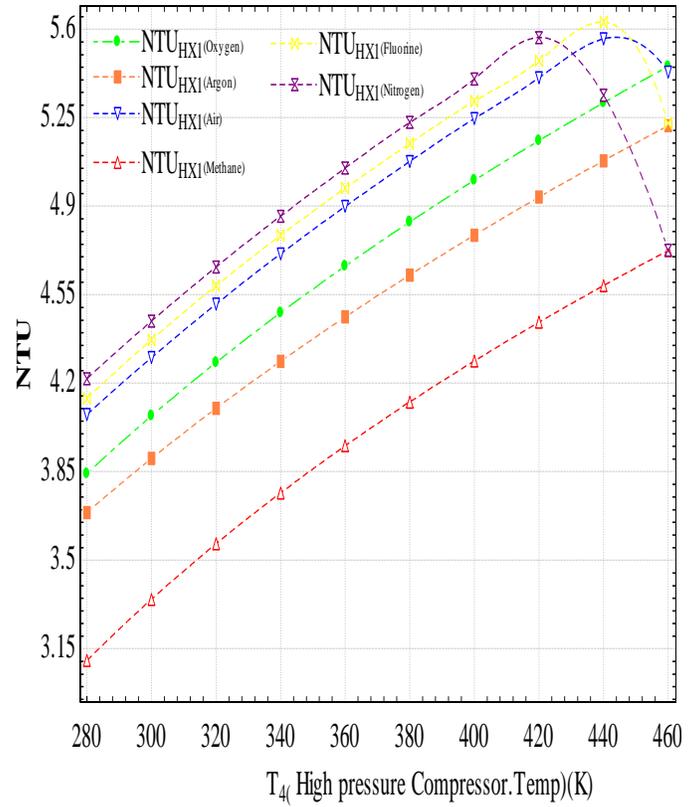


Figure 24: NTU versus high pressure compressor temperature

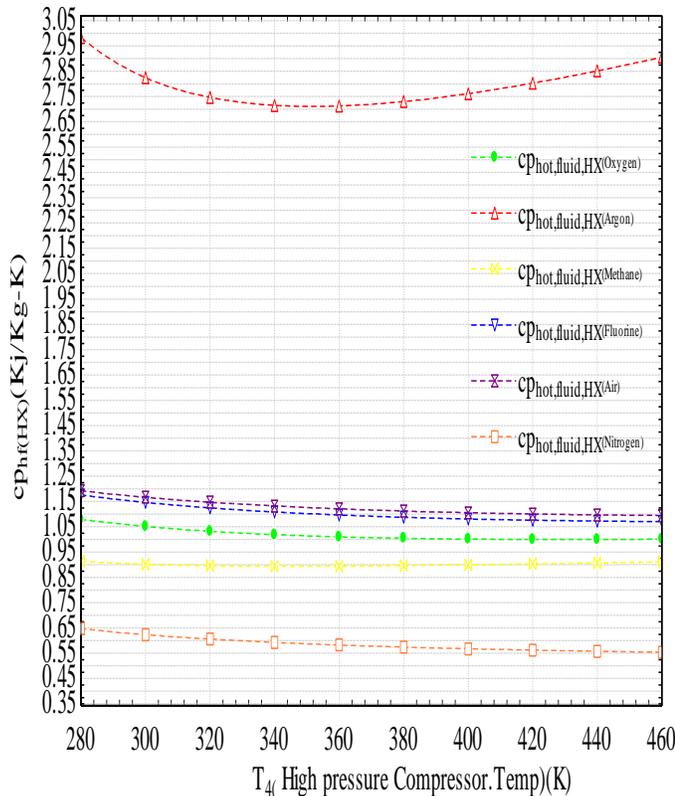


Figure 23: Specific heat of heat exchanger versus high pressure compressor temperature

High pressure outlet temperature of high pressure compressor affect the overall performance of the system. Fig.20 show the effect of temperature variation on COP and second law efficiency of dual compressor system from analysis, it is noticed that COP of gas are continuously decreasing with increasing outlet temperature of high pressure compressor the mean temperature of all gases is the lowest temperature, i.e. 280K, but trend of decreasing COP and second law efficiency for methane gas is highest as compare to other gases. The lowest range of working temperature for methane and argon is 420K and, for air and fluorine is 360K. While, oxygen show poor range of working temperature at 380K. Fig. 21 show liquefaction rate variations with increasing outlet compressor temperature. The liquefaction also affected by increasing temperature range. From graph study, it has been noticed that with increase of temperature the liquefaction rate decreasing drastically. Gases like fluorine air and nitrogen cannot be liquefied, if compressor outlet temperature increasing beyond 340K, and for oxygen, this temperature would be 380K. Fig.22 show the work requirement for liquefaction either increase continuously with increase of compressor temperature. So it is desirable that compressor outlet temperature should be minimum. Fig.23 show specific heat of gases in heat exchanger is also affected by the increasing outlet temperature of compressor. In this fig., it noticed that as we increase the temperature the specific heat of five gases is decreasing at a very minimum rate. While, in case of methane it decrease first up to 340k and then start increasing again by further increase in compressor outlet temperature. Fig. 24

show NTU variation of heat exchanger with outlet temperature of compressor. The graph analysis states that there is increase in NTU value with increase in high pressure compressor temperature up to 420K. But after that it start decreasing for all gases, methane gas show lowest NTU with highest NTU variations range in 4.2 to 5.6

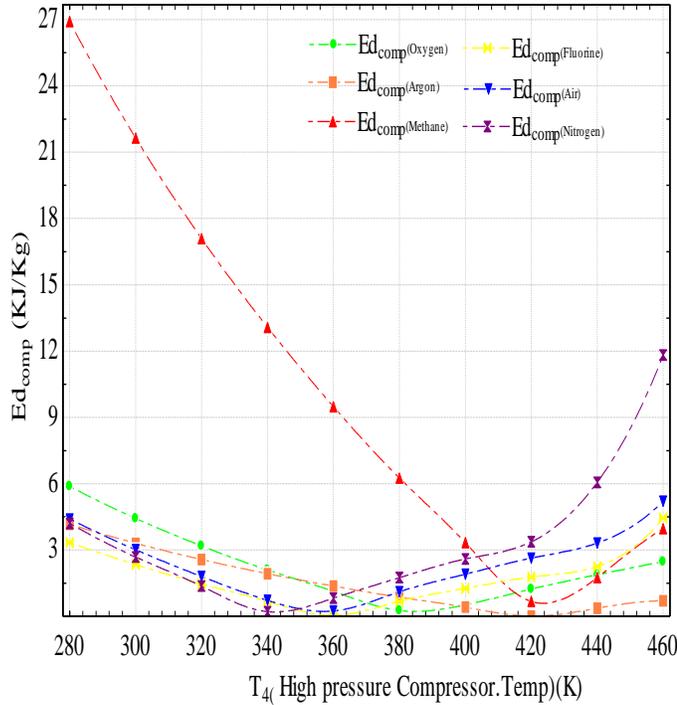


Figure 25: Exergy destruction in compressor 1 versus high pressure compressor temperature

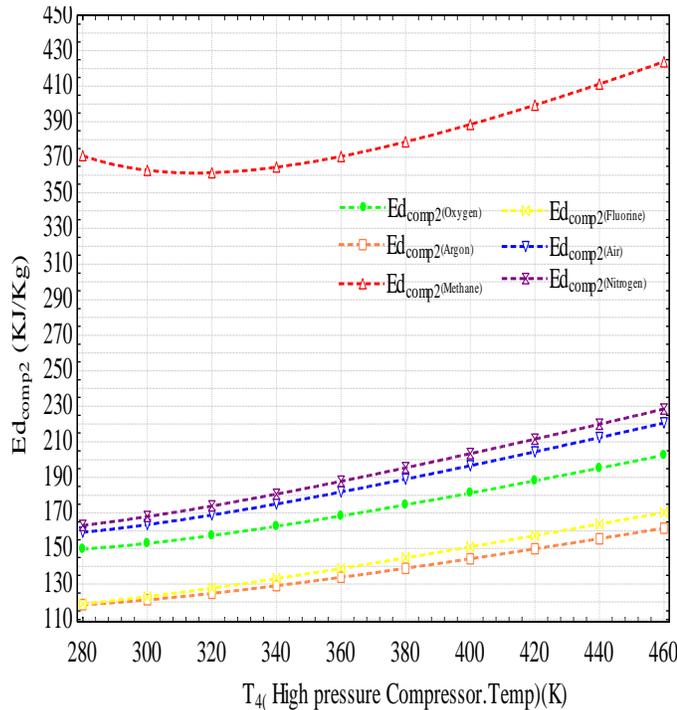


Figure 26: Exergy destruction in compressor 2 versus high pressure compressor temperature

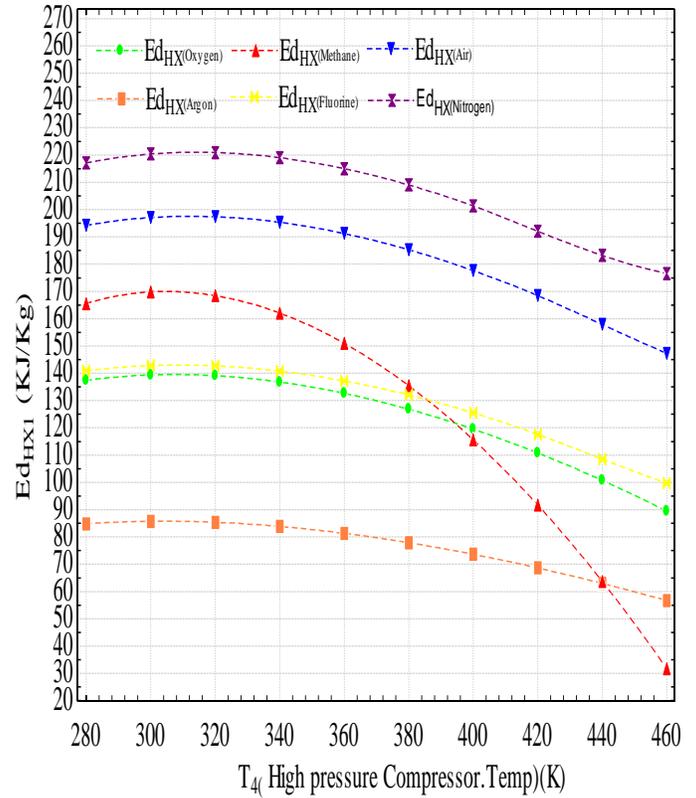


Figure 27: Heat exchanger exergy destruction versus high pressure compressor temperature

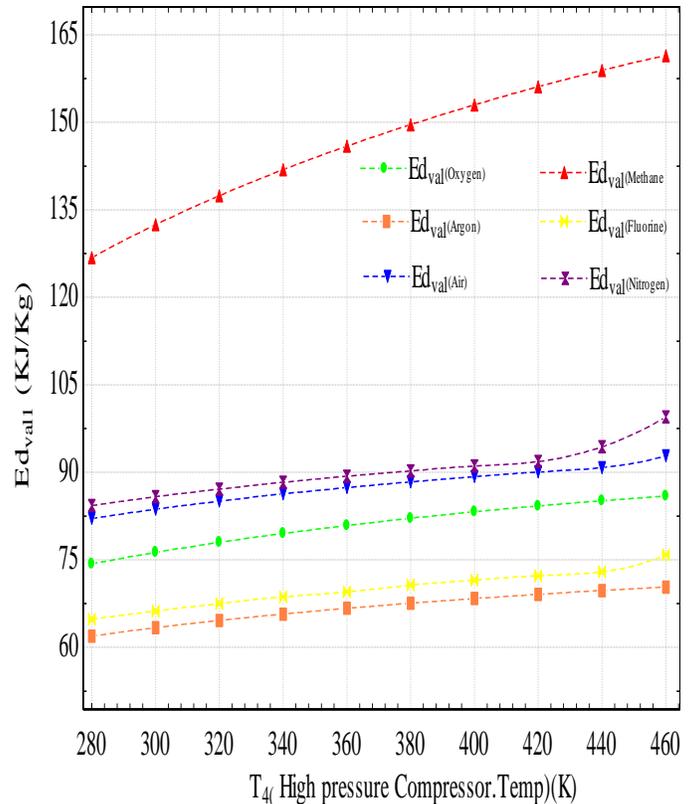


Figure 28: Exergy destruction in expansion valve 1 versus high pressure compressor temperature

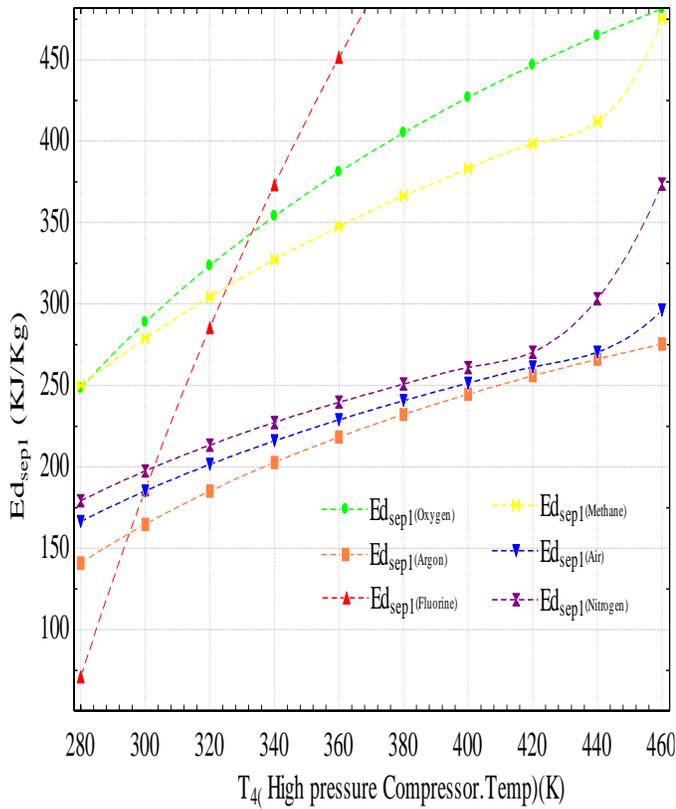


Figure 29: Exergy destruction in separator 2 versus high pressure compressor temperature

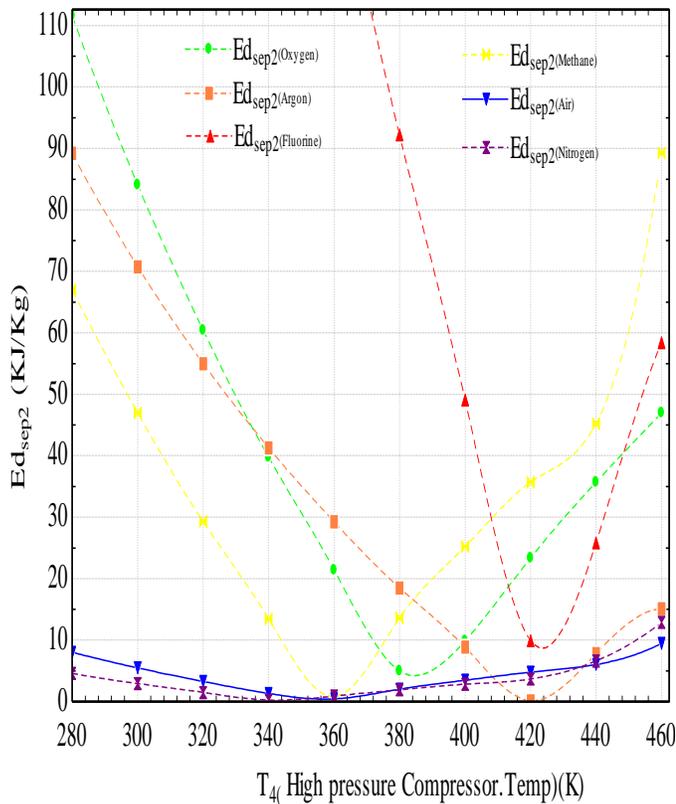


Figure 30: Exergy destruction in separator 2 versus high pressure compressor temperature

The low pressure and high pressure compressor have exergy destruction in each other with variation in high compressor outlet temperature. Fig.25 and 26 show exergy destruction of low pressure and high pressure compressor with temperature variations. The rate of exergy destruction is start decreasing with increase of outlet compressor temperature. For gas air and nitrogen, it is up to 350K. Where this decrement for oxygen is 380K. Methane and argon show the decrement up to 420K, then their destruction rate rise up again by further increase in temperature of compressor. The exergy destruction of high pressure compressor for all gases are increases with increase in compressor temperature.

Fig.27 show exergy destruction rate in heat exchanger for different gases with variation in compressor outlet temperature, the exergy destruction for all gases almost constant up to 360K. But onwards this temperature is start decreasing. Methane gas show huge dip in exergy destruction rate up to 340K. Fig.28 show that with increase in compressor temperature, the rate of exergy destruction in HX also increases for all gases. Fig. 29 show that in high temperature separator, the exergy destruction trend for all six gases with respect to variation in high pressure compressor temperature. The argon, air and nitrogen show increment up to 420K. After this temperature increment in exergy destruction rate increases, the oxygen and fluorine gas show increasing curve in slightly parabolic nature. While, methane gas show exceptionally high rate of exergy destruction with increase in compressor outlet temperature. Fig.30 show that in separator 2, exergy destruction is first decreases then increase with increase of outlet compressor temperature.

5. Conclusion

Exergy analysis of Dual Compressor Linde System with different gases are evaluated on the basis of pressure ratio, compressor outlet temperature, and expander mass flow ratio. Following results are concluded from study.

- (1) During off design condition, performance of cycle does not hamper within the specific range of cyclic pressure ratio, for particular considered system there is always appropriate operating pressure ratio range for each working gas on which system work better
- (2) Dual Compressor Linde system are compared on the basis of performance parameters at different pressure ratio, form the data observation it observed that heat exchanger help in achieving more refrigerant effect which is in turn optimize the performance of the system.
- (3) During PR increase, there is an imbalance in mass flow of forward and return stream of heat exchanger HX. Second law efficiency with the help of increasing pressure ratio which variant and create specific heat imbalance to overcome the mass imbalance.
- (4) Variation in expander mass flow has highly influence the refrigeration effect of expander and overall performance of system. Optimum range of expander flow fraction (τ) producing refrigeration effect is 0.55 to 0.7. Liquid

production rate is highly influenced by refrigeration effect of expander.

- (5) Inlet temperature of expander also plays an important factor to determine the refrigeration effect while other parameters in the system are constant. As the mass flow fraction increases through expander the output temperature of expander T_e also decreases which in turn lower the inlet temperature of input temperature of $T_{in EXP}$.
- (6) In all gases methane gas show highest performance parameters in most of system while argon show lowest.

References

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