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Saving Energy by Exergetic Analysis of MTP Process Refrigeration System

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Abstract
The exergetic analysis is a tool that has been used successfully in many studies aiming a more rational energy consumption to reduce the cost of processes. With this analysis, it is possible to perform an evaluation of the overall process, locating and quantifying the degradation of exergy. This paper applies exergy approach for analyzing the heat exchanger network design and refrigeration of MTP process. For this purpose, the behavior of some industrial processes and refrigeration cycle with propylene refrigerant has been investigated by exergy method. The equations of exergy destruction and exergetic efficiency for main components such as compressors, heat exchangers and expansion valves were studied. A specified section of propylene recovery unit with its refrigeration cycle has been simulated to perform the exergy analysis. Adding a new valve with optimum pressure drop according to process constraints and an expander results in high performance of multi-stream heat exchanger, saving cold utility consumption, using excess heat of the process section, and low required compression power (in the refrigeration section) which are the most important characteristics of the proposed configuration. Results show that the annual profit reaches 4.3 %.

Keywords
MTP process, Refrigeration system, Energy saving, Exergy analysis.

1. Introduction

In process plants such as petrochemical plants and refineries, which are the major energy consumers, heat recovery of heat exchanger network is an important issue (Sun, Luo, and Zhao,2015) because heat exchanger networks play an important role in heat integration.

Normally, a sub-ambient temperature process, such as propylene plant cold-end, comprises of three major parts including the process, the heat exchanger network and the refrigeration system.

Depleting energy resources, increasing environmental concerns, and energy prices are the major impetuses to improve heat integration in existing process plants (Sreepathi and Rangaiah,2014).

In order to analyze systems that involve heat and power, the consideration of heat loads and thermal gradients in a process is not sufficient. So, exergy analysis is introduced. The exergy analysis is a tool that allows identifying and quantifying inefficient equipment in a system involving heat and power, not only in terms of heat loads (quantity of energy), but also in terms of temperature and pressure gradients (energy quality) with respect to the ambient conditions.

Propylene production from methanol is economically important but is highly sensitive to energy because it requires refrigeration at low temperatures. So, continuing development of the methods to reduce net power to provide this refrigeration is important in the petrochemical industry.

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Exergetic analysis of the refrigeration system in ethylene and propylene production process was investigated by Fabrega, Rossi, and Angelo (2010). Results have shown that exergy of refrigeration system of the process can be reduced by about 13%.

Exergy analysis of multistage cascade low temperature refrigeration systems in olefin plants was done by Mafi, Naeynian, and Amidpour (2009). They developed an expression for minimum work requirement for the refrigeration systems of olefin plants. They have shown that exergetic efficiency of chosen olefin refrigeration plants is 30.8%, so they have demonstrated a suggestion for increasing the efficiency of refrigeration plant.

Liquefied natural gas (LNG) plant is investigated for reducing energy consumption by Mortazavi et al. (2012). For this purpose, genetic algorithm (GA) from Matlab optimization tool box is used to optimize propane pre-cooled mixed refrigerant (C3-MR) LNG plant. Results of MCR cycle optimization show that power saving is as high as 13.28%. Propane cycle optimization causes power saving by about 17.16%.

Ghorbani et al. (2013) have investigated minimizing the work consumed in refrigeration cycle of ethylene production. The objective was to provide improvements through mixed working fluids instead of pure working fluid in cryogenic section of low temperature processes with a view to decrease the power consumption for providing the same refrigeration duty. Results show that power consumption in the new refrigeration cycle configuration can be decreased by about 22.3 %.

Mafi et al. (2014) have researched mixed refrigerant cycles of olefin plant cryogenic section. The objective of this paper was to present a methodology for finding the optimal MRC configuration for providing refrigeration in a certain low temperature process. By considering the ECC and GCC charts of these cycles, it is concluded that using multi-stream heat exchangers in the cycle configuration will lead to better matching between hot and cold composite curves.

Energy and exergy analysis of the process has been developed for hydrocarbon recovery process by Mehrpooya et al. (2015). They have presented a new method for refrigeration of NGL recovery that needs only two multi heat exchangers. Results show that compressor work for refrigeration section can be reduced by 38.57 %.

Ghorbani, Hamedi, and Amidpour (2016) have probed nitrogen rejection unit with LNG and NGL co-production processes. In this paper, mixed fluid cascade natural gas liquefaction process of NGL LNG and NGL co-production is assessed through the exergy and exergo-economic analysis methods. After replacing one of the vapor compression cycles with a water-ammonia absorption refrigeration cycle, their exergy destruction cost was much higher than other devices due to the high value of fuel cost in compressor; the fourth compressor had the highest exergy destruction cost (5750307$/hr) and one of the shell and tube exchanger in the absorption refrigeration cycle had the lowest exergy destruction cost (2.033 $/hr).

In this study, the MTP process of Lurgi technology, energy consumption and utility consumption of this process are investigated. At the beginning, the operation of cascade refrigeration system of a typical propylene plant for cooling of deethanizer feed is described. Equations have been studied for each component of the refrigeration system. Then, exergy analysis is applied to calculate exergetic efficiency and exergy destruction for each component. This paper suggests some methods for decreasing of refrigeration compressors work. In this work, exergy analysis has been used for improving exergetic efficiency of the refrigeration system, decreasing cold utility consumption and investment.

2. Materials and Methods

2.1. MTP refrigeration cycle

In this section, the cascade refrigeration system of MTP deethanizer feed is analyzed that consists of propylene refrigeration system, multi-heat exchanger, drums and Joule–Thomson expansion valve.

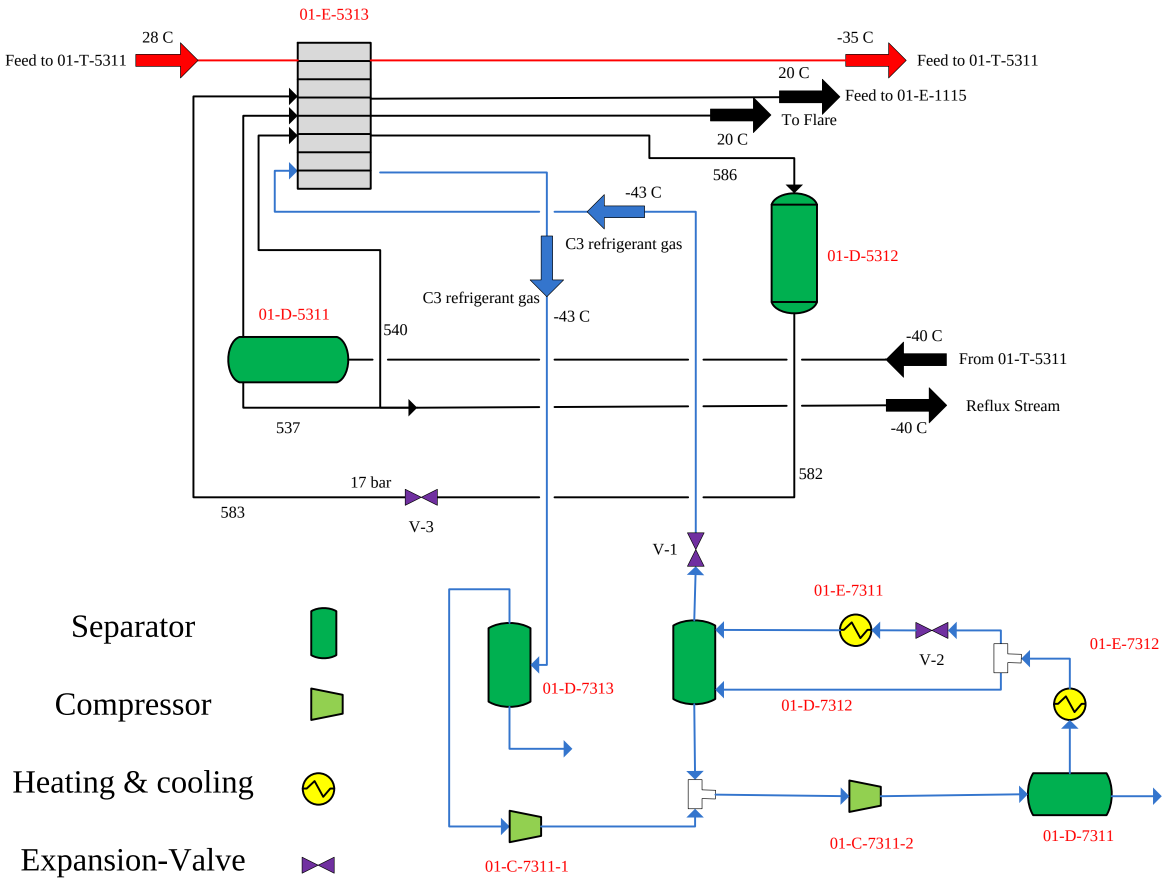

Figure.1. Flow diagram of propylene refrigeration cycle and cooling of deethanizer feed section

Figure.1 shows the flow diagram of propylene refrigeration cycle and cooling of deethanizer feed section in propylene production plant which is designed for capacity of 15200 $^{Kg}/_{hr}$ and comprises of five process units, including Reaction, Regeneration, Gas Separation, Compression, and Drying and Purification.

In the reaction section, methanol is entered into the process through the inlet gas stream at 25 ˚C and ambient pressure. In this section, after pre-heating to 275 ˚C at 16 bar, methanol is converted to dimethyl ether via DME Reactor (01-R-1111). High conversion and extraordinary selectivity cause dimethyl ether to be produced without any purification which then entered to adiabatic MTP Reactor (01-R-1511) directly, where dimethyl ether is converted to propylene and by-products (such as propane, butane, pentene, pentane, hexane, hexane and heavy hydrocarbons) at high temperatures of about 480 °C and ambient pressure. Propylene together with other by-products is sent to the next section for separation and purification.

Propylene refrigerant system supplies propylene at two temperature levels: -43 °C and -15 °C. A fraction of produced propylene in the MTP process that was sent to Buffer 01-D-7311 is used as propylene refrigeration unit via the propylene product pump. The propylene leaves the Propylene Buffer Tank in liquid state. It is cooled with cooling water in the Propylene sub-cooler 01-E-7312. A part of cooled propylene is sent to propylene separator 01-D-7312 as liquid state. The other part of refrigerant propylene, after depressurizing via expansion valve to 8.9 bar and evaporated in the users at 15 °C, is sent as vapor to the propylene separator 01-D-7312. The vapor phase from Separator 01-D-7312 is recycled to the suction side of the propylene Compressor 01-C-7311-1. The liquid phase from 01-D-7312 is depressurized via expansion valve to 1.2 bar and it is directed to the users (01-E-5313) at -43 °C in the deethanizer column section and then entered to the propylene separator 01-D-7313 as vapor phase. The propylene separator outlet vapor (01-D-7313) is fed to the 1st stage of propylene compressor 01-C-7311-1. The vapor is compressed in two stages, and then it is condensed in the propylene condenser 01-E-7311. Finally it is sent back to the propylene buffer 01-D-7311. Deethanizer column (01-T-5311) is used to separate ethane from propane and propylene at -40°C and 19 bar. The output stream from bottom of debutanizer column is entered to the dehexanizer column at 172 °C and 23 bar (01-T-5411) to separate C_{6^-} from C_7^+ hydrocarbons for gasoline production. The vaporous C_2 top product is liquefied by propylene refrigeration and then it is split into two streams. One of two streams is returned to the deethanizer column as reflux stream and the other stream is entered to LNG exchanger (01-E-5313). This stream is used for cooling the deethanizer feed, so, it reduces refrigerant consumption. Part of heated output stream from LNG exchanger is sent to the MTP Reactor as recycled stream and the remaining output stream is sent to the flare.

2.2. Simulation and selection of the equation of state for the system

The simulation of the entire plant is done by using HYSYS simulator. Equations of state are one of the key parameters in simulators.There are papers and books that deal with the differences between present equations of state. The fluid package chosen in the simulator for the determination of thermodynamic properties is PRSV equation of state (Campbell, Lilly, and Maddox 1992).
After simulating the process via HYSYS; it is linked to MATLAB to evaluate the efficiency and the lost work by changing the Refrigeration cycle. Figure.2 shows the steps involved in the information transfer in the sensitive analysis.

Figure.2. MTP plant analysis approach

2.3. Exergy analysis

Exergy analysis combines the first and second laws of thermodynamics, and is a powerful tool for analyzing both the quantity and quality of energy utilization. Exergy is defined as the maximum work possible to obtain from a system during a process that brings this system into equilibrium with environmental as reference state (Kotas,1985).
According to physical exergy is equal to the maximum amount of work obtainable when the stream of substance is brought from its initial state to the environmental state which is defined by equation (1.

$$e_{ph1} - e_{ph2} = (h_1 - h_2) - T_0 (s_1 - s_2) \quad (1)$$

where h and s are the enthalpy and entropy, respectively, T_0 is the reference environmental temperature, h_0 and s_0 are the corresponding properties at the dead state.
Based on exergy balance between input and output streams, exergetic efficiency and exergy loss for equipments in various refrigeration systems and cooling of deethanizer feed section are described below.

2.3.1. Exergy balance for process equipment

2.3.1.1. Compressors

A gas compressor is a mechanical device that increases the pressure of the fluid and it is able to transport the fluid through a pipe. If the compressor work is reversible, there will be no exergy destruction. It means that irreversibility can be totally eliminated, which

results in minimizing consumption work of compressor.
In the absence of heat transfer, the exergy balance for one multistage compressor is obtained with the following Equation 2:

$$\mathrm{I}=E_{\mathrm{x_i}}-E_{\mathrm{x_o}}+\mathrm{w_c}=\sum\left(\dot{\mathrm{m}}.e\right)_i-\sum\left(\dot{\mathrm{m}}.e\right)_0+\mathrm{w_c} \tag{2}$$

In Equation (2), increasing of exergy is equal to e_0-e_i and $\mathrm{w_c}$ is the actual power input.

$\dot{\mathrm{m}}$ is the molar flow and e is the exergy of streams. So exergetic efficiency of a compressor is defined as Equations (3) or (4):

$$\eta=\frac{\sum\left(\dot{\mathrm{m}}.e\right)_0-\sum\left(\dot{\mathrm{m}}.e\right)_i}{w_c} \tag{3}$$

$$\eta=1-\frac{I}{w_c} \tag{4}$$

2.3.1.2. Valve

Expansion valve is usually used in gas liquefaction plants and thermal plants such as refrigerators and heat pumps. Expansion processes occur mostly at below ambient temperature. The primacy purpose of such expansion processes is the production of cooling effect (Kotas, 1985).
It is known that expansion valves are essentially isenthalpic devices with no work interaction, so the irreversibility of the throttling process can be obtained from the exergy balance by neglecting heat transfer with the surroundings is obtained from Equation (5:

$$e_1^{\Delta T}+e_1^{\Delta P}=e_2^{\Delta T}+e_2^{\Delta P}+I \tag{5}$$

$e^{\Delta T}$ and $e^{\Delta P}$ are defined as Equations (5 and(6, respectively:

$$e^{\Delta T}=\left[-\int_T^{T_0}\frac{T-T_0}{T}dh\right]_P \tag{6}$$

$$e^{\Delta P}=\left(h-h_0\right)-T_0\left(s-s_0\right) \tag{7}$$

So exergy efficiency of expansion valve can be obtained from Equation (5:

$$\eta=\frac{e_0^{\Delta T}-e_i^{\Delta T}}{e_i^{\Delta P}-e_0^{\Delta P}} \tag{8}$$

If the valve is not an expansion valve and it is just a control valve, exergy efficiency for this type of valve is obtained by Equation (9:

$$\eta=\frac{E_{\mathrm{x_o}}}{E_{\mathrm{x_i}}} \tag{9}$$

2.3.1.3. Heat Exchangers

There are four categories of heat exchangers in the process and refrigeration system. The most complicated heat exchangers are LNG or multi-stream heat exchanger.
At last, for multi-heat exchanger, the irreversibility and exergy efficiency based on the above explanations and neglecting heat transfer with the surrounding are Equations**Error! Reference source not found.** and(11.

$$I=E_{X_{in}}-E_{X_{out}}=\sum[(\dot{m}_e e_e)+(\dot{m}_c e_c)+(\dot{m}_h e_h)]_{in}-\sum[(\dot{m}_e e_e)+(\dot{m}_c e_c)+(\dot{m}_h e_h)]_{out} \tag{10}$$

$$\eta=\frac{\sum[(\dot{m}_e e_e)_{out}-(\dot{m}_e e_e)_{in}]_c+[(\dot{m}_c e_c)_{out}-(\dot{m}_c e_c)_{in}]}{\sum[(\dot{m}_e e_e)_{in}-(\dot{m}_e e_e)_{out}]_h+[(\dot{m}_h e_h)_{in}-(\dot{m}_h e_h)_{out}]} \tag{11}$$

Subscripts in, out, c, h and e stand for inlet, outlet, cooler, heater and process stream, respectively.

3. Results and discussions

3.1. Exergy analysis and minimizing compressor power

The exergy analysis of low temperature refrigeration system and cooling of deethanizer feed section of MTP plant was studied in the present study to evaluate the amount of exergy destruction and exergetic efficiency for each equipment.

3.1.1. Compressors

The exergy destruction and exergy efficiency of the compressor according to Equation (3) or (4) are shown in Table 1.

Table 1. Exergy destruction and exergy efficiency of compressor

	C-7311-1	C-7311-2
Power(kW)	1920	5525
Lost(kW)	462.3	1179.4
η	0.769	0.787

3.1.2. Heat exchangers

The exergy destruction and exergetic efficiency of the multi-heat exchangers according to Equations **Error! Reference source not found.** and (11 are shown in Table 2.

Table 2. Exergy destruction and exergy efficiency of multi-heat exchanger

	E-5313
Duty (kW)	3076
Lost (kW)	621.75
η	0.8212

The results show that compressors have high amount of exergy destruction, so their performance shall be improved. In the following, some practical ways are suggested to improve the exegetic efficiency of the refrigeration compressors through pressure drop of process stream and employing a turbo-expander. The amount of exergy loss of each series devices is illustrated in Figure.3.

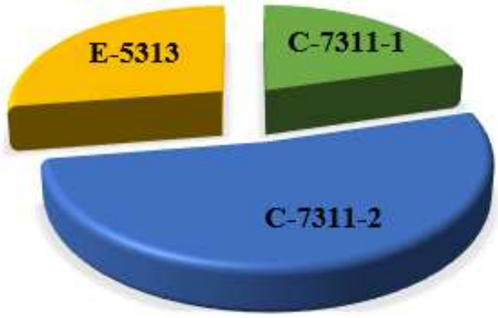

Figure.3. Diagram of exergy loss percentage of each series devices

3.1.3. Sensitivity analysis

Sensitivity analysis and changing of operating parameters are efficient methods in order to determine the behavior and improve exergy loss of equipment.

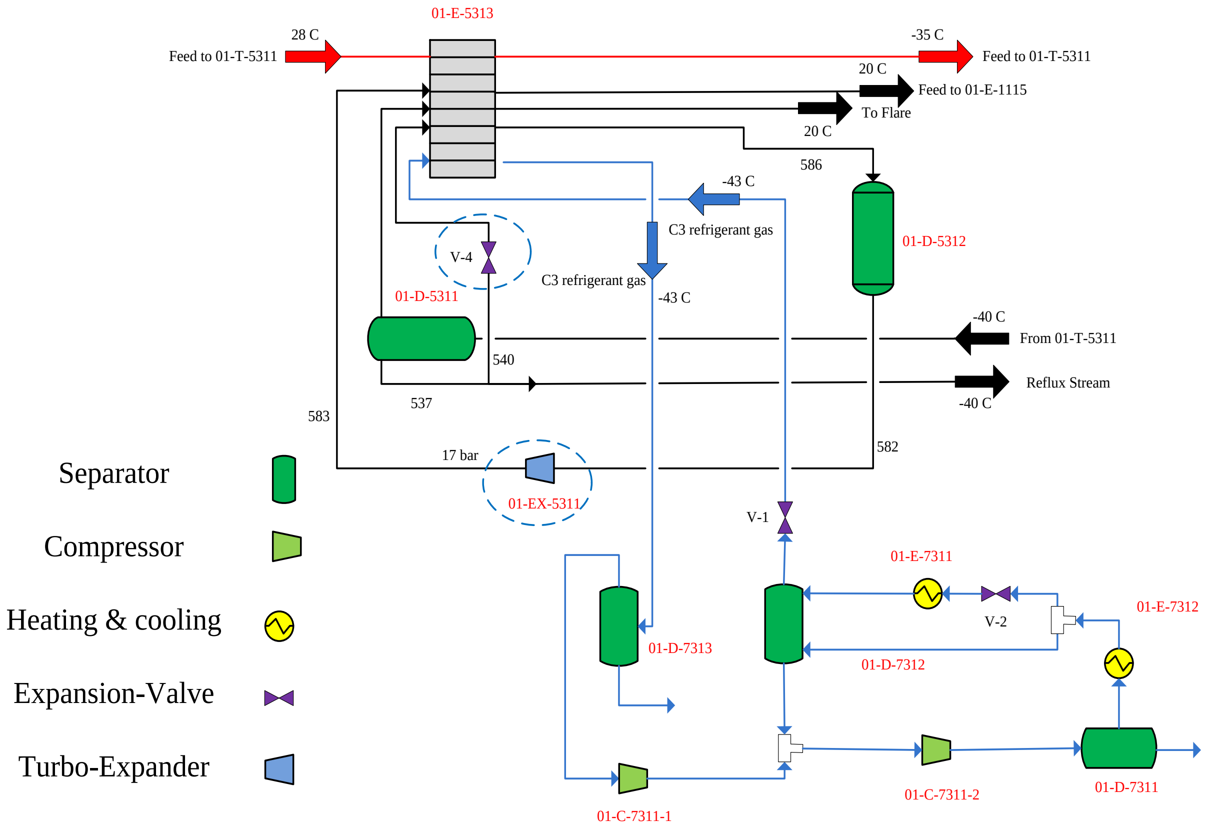

Figure.4. Flow diagram of new design of propylene refrigeration cycle and cooling of deethanizer

The reduction of power consumption of compressors in the refrigeration cycle can be achieved by reducing the necessary cooling via refrigerant. One of the available solutions is to use the energy of low temperature process streams. As can be seen in Figure.1, hot process stream is cooled from 28 °C to -35 °C during four stages. Cooling stages consist of top product of deethanizer in three levels of temperature and one propane refrigerant stream in multi heat exchanger (01-E-5313). First, the C_2^-recycled stream coming from 01-D-5312 A/B is heated up to 20 °C and is directed to 01-E-1115. In the second stage, the C_2^- purge coming from 01-D-5311 is heated to 20 °C and is sent to the flare line. Third, heat of feed deethanizer is used to evaporate the C_2^- recycle sent from 01-D-5311 to 01-D-5312 A/B and finally, it is cooled by propylene refrigerant. One of the parameters that affect the reduction of the required propane refrigerant is the changing of the inlet process streams pressure drop which has no effect on the performance of other equipments. Figure.4 shows the novel design of cooling deethanizer column feed. It is clear that pressure drop of a stream decreases the stream temperature. So,by adding a new Joule-Thomson valve on the 540 stream line, the stream is partially vaporized and its temperature would be decreased. Then, by adding an expander on 582 process stream for pressure drop, the 583 stream temperature and pressure are decreased to -12.1^{0}C and 17 bar. Figure.5 illustrates the variations of exergy loss as a function of pressure drop for LNG exchanger and Joule-Thomson valve, while figure Figure.**6** shows variations of exergy loss and compressor work as a function of pressure drop compressor work of refrigeration cycle when the pressure drop of new valve on 540 stream line changes from 0 kPa (industrial design) to 1900 kPa (new design).

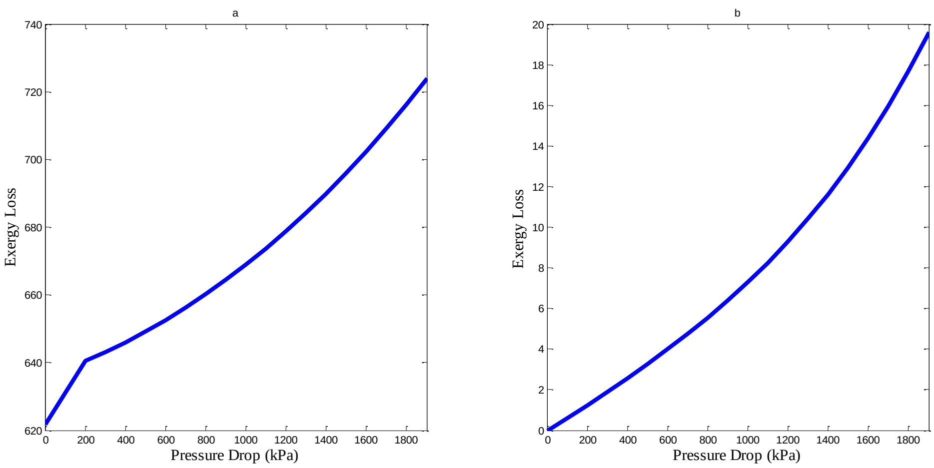

Figure.5. Variation of exergy loss a) LNG exchanger (01-E-5313), b) Joule-Thomson valve (V-4) versus valve (V-4) pressure drop

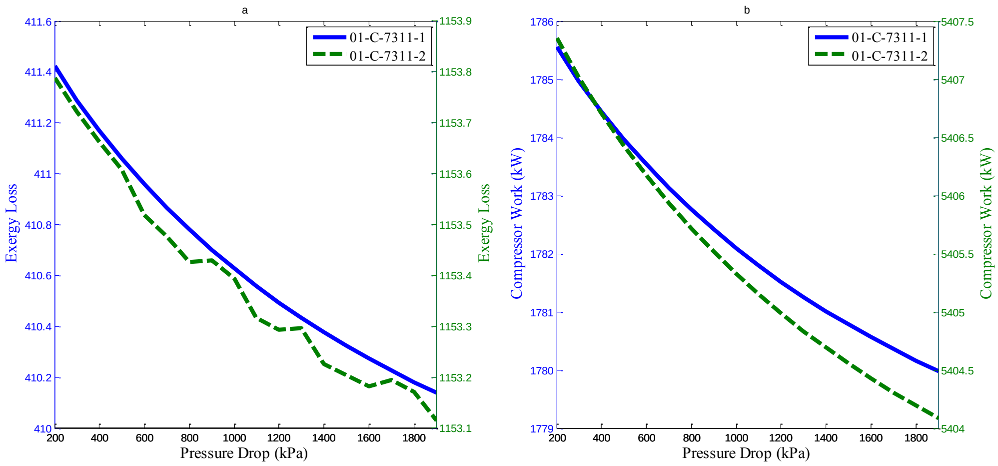

Figure.6. Variations of a) exergy loss, b) compressor work of refrigeration cycle versus valve (V-4) pressure drop

As can be seen from Figure.5, when the valve pressure drop is increased, the exergy loss of LNG exchanger and new valve (V-4) increases gradually to 1200 kPa, then exergy increases dramatically, while Figure.5 shows compressors work and its exergy loss have different behavior and they decrease in lower slope to 1200 kPa, then variation of compressor work and exergy loss are approximately constant. In this case, we should set valve operating pressure drop to 1200kPa.

According to Figure.4, a turbo-expander is used instead of valve on 582 stream line (V-3), which can decrease the compressor work of refrigeration cycle. So, the work produced by the turbo-expander should be considered. Turbo-expander outgoing stream temperature is much lower than that of the expansion valve. Further decrease of turbo-expander outgoing stream temperature reduces the amount of required propylene refrigerant, so that the compressor work of refrigeration cycle would be decreased. Specifications of the cold process, feed and C_3 refrigeration streams are summarized in Table 3.

According to Table 3 and pervious discussion, the amount of required propylene refrigerant for turbo-expander is lower than expansion valve. By applying optimum pressure drop of new valve and using a turbo-expander according to operational conditions (e.g. outgoing stream pressure of turbo-expander cannot be lower than 17.0 bar), optimized refrigeration cycle is calculated and listed in Table 4 and Table 5. Parameters of turbo-expander

	Exergy loss (kW)	Turbo-Expander Work (kW)	Capital Cost ($/year)	The total annual profit ($/year)
01-EX-5311	13.9	36.2	469.6	13808.3

Capital cost of adding a new turbo-expander is obtained by using equation **Error! Reference source not found.**:

$$PEC = (Nelson\ Cost\ Index/\ 1961.6) \times$$
$$(-1900 + 820 \times (Wrk)^{0.8}) \qquad (12)$$

Nelson cost index is equal to 2553. Equation (12) shows capital cost of turbo-expander for one year.

$$\text{Capital Cost (\$/year)} = \text{PEC} \times \text{CRF} \qquad (12)$$

In above equation, CRF is capital recovery factor which is calculated using equation (13):

$$CRF = \frac{i_{eff}\,(1+i_{eff}\,)^{BL}}{(1+i_{eff}\,)^{BL} - 1} \qquad (13)$$

BL is plant economic life that is considered 25 years and ieff is average annual rate of the cost of money that is equal to 10%.

Table 3. Specifications of process streams and refrigeration cycle shown in Figure.4

	Feed Stream	V-4 streams	582-583 With valve	582-583 With turbo-expander	C_3 refrigeration streams
Input temperature (° C)	31	540: -40	7	7	-43.4
Output temperature (° C)	-35	541: -45.3	-1.9	-12.1	-43.4
Input pressure (bar)	21.5	540: 36.4	24.4	24.4	1.2
Output pressure (bar)	19.2	541: 24.4	17.0	17.0	1.1
Input vapor fraction (%)	100	540: 0.0	100	100	30
Output vapor fraction (%)	2.4	541: 7.8	100	100	100
Flow rate (kmoleh^{-1})	688	301	261.6	261.4	With valve: 599.5 With turbo-expander: 589.2

Table 4. The effect of pressure drop of new valve and turbo-expander on refrigeration cycle compressors work

	01-C-7311-1			01-C-7311-2		
	Industrial design	**New design**	**Reduction rate (%)**	**Industrial design**	**New design**	**Reduction rate (%)**
Exergy loss (kW)	445	406.6	8.6	1178.0	1151.4	2.3
Compressor Work (kW)	1920	1765	8.1	5525	5395	2.4
The total annual profit ($/year)		61134.5			51274.1	

Table 5. Parameters of turbo-expander

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BL is plant economic life that is considered 25 years and i_{eff} is average annual rate of the cost of money that is equal to 10%.

Figure.7 shows the rate of compressors work before and after of refrigeration cycle, by considering the simulated process and actual operational conditions.

According to Table 4 and Table 5. Parameters of turbo-expander

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$$CRF = \frac{i_{eff}\,(1+i_{eff})^{BL}}{(1+i_{eff})^{BL} - 1} \quad (13)$$

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, the exergy loss of refrigeration cycle compressors has been decreased in comparison with the primitive design. Compressor exergy loss represents the amount of additional work that compressors must consume for compression of stream to the desired pressure. So, decreasing the compressor exergy loss will result in lower compressor power consumption. As can be seen in Table 4, by increasing the process integration with refrigeration cycle, the exergy loss and power consumption of compressors refrigeration cycle has significantly decreased. The amount of power consumption for compressor 01-C-7311-1 and 01-C-7311-2 are decreased by about 155 kW and 130 kW, respectively. Since the production work in turbo-expander is low (Table 5. Parameters of turbo-expander

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01-EX-5311	13.9	36.2	469.6	13808.3

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), it can also be used for driving pumps motor.

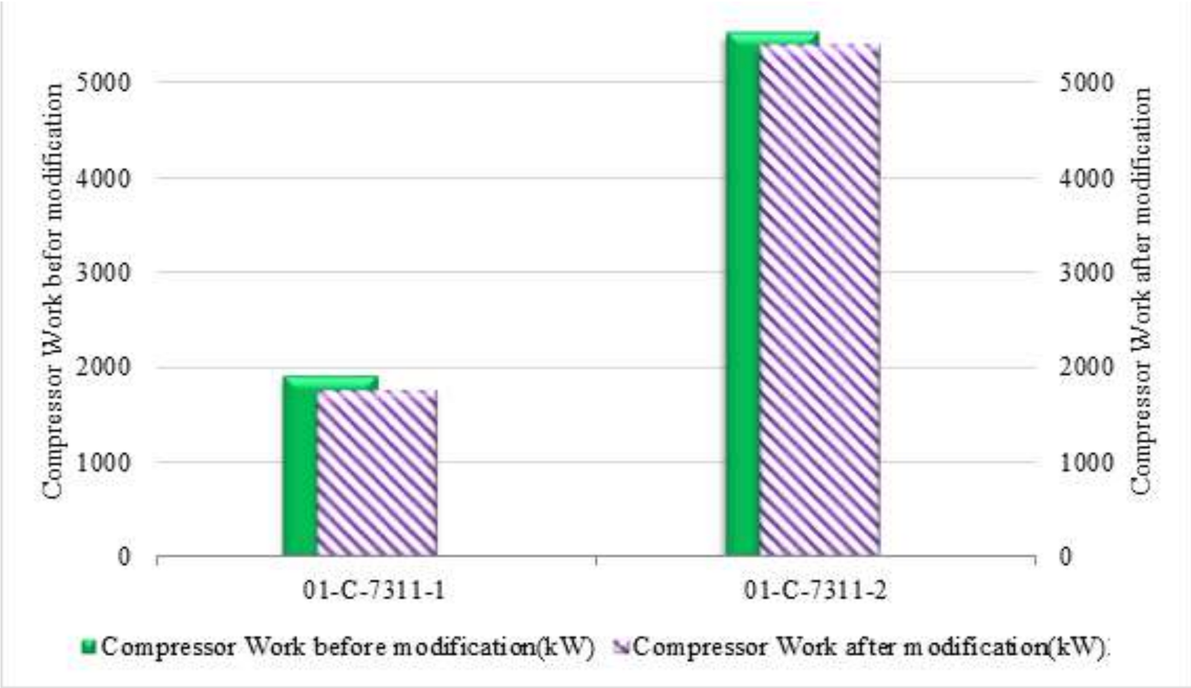

Figure.7. Rate of compressors work before and after of refrigeration cycle

4. Conclusions

This paper presents an exergetic analysis of the propylene production and its refrigeration cycle. The exergy method used here is found to be a powerful tool in optimizing the performance of such complex process units. The equations of exergy efficiency and exergy destruction for each component of refrigeration system are investigated. In this paper, a basic practical way including exergy analysis is used. For improving exergetic efficiency and decreasing exergy loss of the refrigeration system and due to operation constraints, new operational conditions for cooling of deethanizer feed and propylene cycle equipments are proposed. It is found that the refrigeration cycle with excess expansion valve and a turbo-expander surpasses the refrigeration cycle without turbo-expander in terms of the net power consumption and the total irreversibility. After integration and optimization, the net power of refrigeration cycle is 7123.8 that is reduced by about 4.3 % relative to Lurgi design. The results show that the total annual profit is decreased about 126216.9 $ for each year. By adding an expansion valve and a turbo-expander it was possible to produce power for rotating motor of pumps and also use excess heat in the process for preheating the deethanizer feed.

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Nomenclature

I	Irreversibility (kW)
Ex	Exergy (kJ/kgmole)
T	Temperature (°C)
P	Pressure (bar)
S	Entropy (kJ/kgmole°C)
H	Enthalpy (kJ/kgmole)
X	Component mole fraction
m^0	Flow rate (kgmole/hr)
Q	Heat duty (kW)
W	Work transfer rate (kW)
$\dot{E}$	Rate of exergy

Greek letters

η	Exergy efficiency

Subscripts

i	Inlet
o	Outlet
sh	Shaft
a	Air
c	Cold
h	Hot

Superscripts

ΔP	Pressure component
ΔT	Thermal component
°	Standard condition
Ph	Physical

Abbreviations

MTP	Methanol to propylene
J-T	Joule- Thomson valve
E	heat exchanger
C	Compressor
V	Expansion valve
D	Flash drum