Exergy analysis of cascade refrigeration system working with refrigerant pairs R41-R404A and R41-R161

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Abstract. In this paper, a comparative study of vapour cascade refrigeration system performance with R41 – R161 and R41 – R404A refrigerant couples have been presented to find out best suited refrigerant between R161 and R404A in the high temperature circuit. A mathematical model has been developed using the basic equations of refrigeration system for the calculation of basic performance parameters energetically and exergetically. Few parameters have been assumed to carry out the simulation. Simulated results show that R41 – R161 gives better results than R41 – R404A in all aspects, i.e. compressor power, COP, exergy destruction and exergetic efficiency wise. Exergetic analysis reveals that maximum and minimum losses for R41 – R161 occurs in the high temperature circuit compressor and evaporator respectively. Therefore, R161 can be a promising refrigerant to R404A due to its better thermophysical and environmental properties.

1. Introduction
Discovery of ozone holes in the atmospheric stratosphere by the synthetic CFC and HCFC refrigerants initiated the Montreal Protocol and other amendments resulting in permanent ban of those refrigerants. Hence, low GWP and zero ODP refrigerants are considered for the replacement of those. Refrigeration plays a significant role to maintain very low temperature for different purposes. According to IIR, about 15% of the world’s total energy is being consumed by the refrigeration and air conditioning system [1]. But, while operating between very high temperature difference of evaporator and condenser cooling capacity and COP decreases and even compressor breaks down. These problems can be overcome by using multistage refrigeration system or cascade refrigeration system (CRS) [2].

Different researchers have investigated on cascade refrigeration system to find out best suitable refrigerant pairs according to the desired condition. Kilicarslan [3] investigated on different types of cascade refrigeration system using R134a and found an average decrease in compressor power and increase in COP compared to single stage system. In another study, Kilicarslan and Hosoz [4] found that COP increased and the irreversibility decreased with the increase in evaporator temperature whereas, COP decreased and the irreversibility increased when condenser temperature was increased. Cabello et al. [5] concluded from their study that R152a could replace R134a without any system modifications in the low temperature circuit (LTC). Lee et al. [6] reported that the optimal condenser temperature in the cascade condenser increased linearly with the increase in condenser, evaporator and temperature difference in the cascade condenser. Liang et al. [7] used R41-R404A and R23-R404A as refrigerant combination in their numerical work and found that R41-R404A refrigerant couple gave better performance than R23-R404A. Mohammadi and Ameri [8] simulated a cascade refrigeration system and found maximum irreversibility in generator and the first stage compressor. Park et al. [9]
investigated on a cascade refrigeration system using R410A and R134a as refrigerants both theoretically and experimentally. They experienced a similar trend from both the studies. Korshetti et al. [10] pointed R161 as a promising alternative to R22 due to its better thermodynamic and environmental properties.

In this paper, authors made an attempt to compare the performance of two cascade refrigeration systems working with R41–R404A and R41–R161 as refrigerant combinations. A mathematical model has been developed using engineering equation solver (EES) [11] software based on standard energy and exergy equations for refrigeration system.

2. Cycle Description

Figure 1(a) shows the schematic diagram of a typical cascade refrigeration system and the corresponding p – h plot has been shown in figure 1(b). Two basic vapour compression refrigeration cycles, LTC and HTC are coupled with each other by means of a cascade condenser. The cascade condenser behaves like a condenser in the LTC and an evaporator in the HTC. Refrigerant R41 is chosen to be the LTC refrigerant, whereas R161 and R404A are considered as the working fluids separately in HTC. Comparisons of different thermo physical and environmental properties [12] of those three refrigerants used in this investigation are shown in table 1.

![Schematic Diagram and (b) corresponding p-h plot of cascade refrigeration system](image_url)

In LTC, evaporator absorbs the heat $Q_1$ from the refrigerated space at evaporation temperature $T_E$ and is passed to the LTC compressor. $W_1$ amount of work is supplied to run the compressor and the vapour refrigerant enters into the cascade condenser coming out of the compressor. In the cascade condenser, the LTC refrigerant at $T_{LC}$ temperature rejects the heat $Q_{CC}$ which is then absorbed by the HTC refrigerant at the temperature of $T_{HE}$. After taking the heat in the cascade condenser HTC refrigerant gets evaporated and enters into the HTC compressor. Again, $W_2$ amount of work is done on the system to compress the vapour refrigerant to the condenser pressure. In the condenser, the vapour refrigerant gets condensed by rejecting $Q_2$ amount of heat at condenser temperature $T_C$. Temperature difference between the LTC condenser and HTC evaporator is represented by $\Delta T$, which is one of the most important parameters to influence the performance of a cascade refrigeration system.

| Refrigerant | Molecular mass (gm/mole) | Critical temperature (°C) | Boiling Point (°C) | ASHRAE safety code | ODP | GWP |
|-------------|--------------------------|---------------------------|-------------------|--------------------|-----|-----|
| R41         | 34.03                    | 44.1                      | -78.1             | A2                 | 0   | 97  |
| R161        | 48.06                    | 102.2                     | -37.1             | A3                 | 0   | 12  |
| R404A       | 97.6                     | 72.1                      | -46.6             | A1                 | 0   | 3800|

Figure 1(a). Schematic Diagram and (b) corresponding p-h plot of cascade refrigeration system

Table 1. Thermo physical Properties of R41, R161 and R404A [12]
3. Mathematical Model
A numerical model has been developed based on the following assumptions. Pressure drop and heat loss in the pipe line are neglected. No subcooling is considered and superheating is taken as effective heating. All the components are at steady state condition. Temperature difference between the hot and cold fluid in the cascade heat exchanger is assumed to be 5ºC. Other basic input parameters values considered for the present analysis are presented in table 2.

Table 2. Basic assumption for the simulation

| Parameters       | Values | Parameters       | Values | Parameters       | Values |
|------------------|--------|------------------|--------|------------------|--------|
| Q_{eva} (kW)     | 10     | T_E              | -60ºC to -30ºC | T_0    | 25ºC             |
| \eta_{C,LTC}     | 80%    | T_C              | 40ºC   | Superheating in LTC | 5ºC   |
| \eta_{C,HTC}     | 80%    | \Delta T         | 5ºC    | Superheating in HTC | 5ºC   |

Based on those assumptions, the mass flow rate in the LTC can be calculated as [7]:

\[ \dot{m}_1 = \frac{Q_1}{h_1 - h_4} \]  \( (1) \)

Compressor power in the LTC and heat load in cascade heat exchanger can be written as:

\[ W_1 = \dot{m}_1 (h_2 - h_1) \eta_{C,LTC}, \quad Q_{cc} = \dot{m}_1 (h_2 - h_3) = \frac{Q_1 (h_2 - h_3)}{(h_1 - h_4)} \]  \( (2, 3) \)

Mass flow rate and compressor power in the HTC can be expressed as:

\[ \dot{m}_h = \frac{Q_{cc}}{h_3 - h_4}, \quad W_2 = \dot{m}_h (h_8 - h_3) \eta_{C,HTC} \]  \( (4, 5) \)

Total compressor power, heat load in the condenser and COP of the system are as follows:

\[ W_T = W_1 + W_2, \quad Q_2 = \dot{m}_h (h_8 - h_7), \quad \text{COP} = \frac{Q_1}{W_T} \]  \( (6, 7, 8) \)

Exergy loss in different components in the system can be expressed as:

Evaporator:

\[ ED_{eva} = EX_4 - EX_1 + Q_1 \times \left(1 - \frac{T_0}{T_E}\right) \]  \( (9) \)

Compressor and expansion device in LTC:

\[ ED_{comp,1} = EX_1 - EX_2 + W_1, \quad ED_{exp,1} = EX_3 - EX_4 \]  \( (10, 11) \)

Compressor and expansion device in HTC:

\[ ED_{comp,h} = EX_5 - EX_6 + W_2, \quad ED_{exp,h} = EX_7 - EX_8 \]  \( (12, 13) \)

Cascade heat exchanger and condenser:

\[ ED_{cas} = EX_2 + EX_8 - EX_3 - EX_5, \quad ED_{cond} = EX_6 - EX_7 + Q_2 \times \left(1 - \frac{T_0}{T_C}\right) \]  \( (14, 15) \)

Total exergy destruction of the system and can be written as:

\[ ED_{total} = ED_{eva} + ED_{comp,1} + ED_{exp,1} + ED_{comp,h} + ED_{exp,h} + ED_{cas} + ED_{cond} \]  \( (16) \)
Exergetic efficiency of the system can be expressed as:

$$\eta_{ex} = \frac{W_T - ED_{total}}{W_T}$$  \hspace{1cm} (17)

Percentage of exergy loss in each component can be expressed as:

$$\delta_i = \frac{ED_i}{ED_{total}}$$  \hspace{1cm} (19)

4. Results and Discussions

Thermodynamic analysis of cascade refrigeration system using R41-R404A and R41-R161 as refrigerant pairs has been carried out to evaluate the optimal performance of the system. Figure 2 shows the variation of optimal condensing temperature in the LTC with the evaporator temperature when the condenser temperature in the HTC is fixed at 45°C. It shows that the optimal condenser temperature in LTC increases with the increase in evaporator temperature for both R41 – R404A and R41 – R161 systems. It is evident from the figure that the optimal condensing temperature for R41 – R161 is lower than that of the R41 – R404A system. The optimal saturation temperatures of the LTC condenser for each corresponding evaporator temperatures have been used to find out the optimal performance of the system.

The effect of evaporator temperature on the discharge temperatures of both the compressors has been shown in figure 3. It is seen from the figure that the discharge temperature of the compressor decreases with the increase in evaporator temperature. It is evident from the figure that the HTC compressor discharge temperature with refrigerant pair of R41 – R161is higher than that with R41 – R404A pair, whereas, the LTC compressor discharge temperature is less for R41 – R161 pair compared to that of R41 – R404A. The difference in compressor discharge temperatures between R161 and R404A in HTC is found to increase from 18.6°C to 34.2°C for the evaporator temperature range of -30°C to -60°C.

The variation of total compressor work with evaporator temperature at optimal condition has been depicted in figure 4. Figure shows that the total compressor power requirement decreases when evaporator temperature increases. Less power is needed to run the compressor when the refrigerant R161 is used in the HTC. Differences in required compressor power between both the systems have been calculated and are noted to be 8.59% and 7.47% less at evaporator temperature of -60°C and -30°C respectively. It can be attributed to the lower compressor power requirement in the LTC for R41 - R161 system due to its lower optimal condensation temperature in the LTC.
The effect of evaporator temperature on the optimal COP of the system has been presented in figure 5. It is evident from the figure that the COP of both the systems increases with the increase in evaporator temperature. Also, COP of the system is found to be slightly higher when R161 is used as refrigerant in the HTC instead of R404A due to lower compressor power and constant cooling load. The differences in COP between the investigated refrigerant pairs have been calculated for different evaporator temperatures. The maximum and minimum differences are noted to be 9.38% and 8.08% for evaporator temperatures of -60°C and -30°C respectively.

Figure 6 shows the effect of evaporator temperature on total exergy destruction of the system. It can be observed from the figure that the total exergy loss from the system decreases with the increase in evaporator temperature. It is also observed that exergy loss is slightly higher for R41-R404A refrigerant pair compared to R41 – R161. Differences in exergy loss have been calculated and are found to be 17.03% and 15.96% lower for R41 – R161 system for evaporator temperatures of -60°C and -30°C respectively.

Comparison of optimum exergetic efficiency of the system between R41-R161 and R41-R404A are presented in figure 7. Figure shows that exergetic efficiency increases with the increase in evaporator temperature and the system with refrigerant R41-R161 shows relatively higher exergetic efficiency compared to that of R41-R404A. The exergetic efficiency at the -30°C evaporator temperature for both the systems has been calculated and they are found to be 59.47% and 55.38% for R41-R161 and R41-R404A systems respectively.

Table 3 presents the percentage of exergy destruction in each component of the system for both the refrigerant combinations at all the investigated evaporator temperature and at optimal LTC condenser temperature corresponding to the evaporator temperature. The values are taken keeping the condenser at constant temperature of 40°C. Results depict that maximum exergy loss is occurring in HTC compressor and minimum in LTC evaporator at all evaporator temperature.

5. Conclusions
The following conclusions can be drawn from the simulation work on cascade refrigeration system. The refrigerant pair R41-R161 shows better energetic and exergetic performance than R41-R404A combination. Total compressor power is less when system is working with R41 – R161. Higher level of COP and exergetic efficiency is achieved when R41 – R161 is used in the system instead of R41 – R404A. Exergy loss is also found to be marginally less for R41 – R161 system in comparison with R41 – R404A system. Finally, it can be concluded that R41 – R161 system can be used as a replacement of R41 – R404A system due to its better performance and relatively less GWP value by taking care of the flammability issue of R161.
Table 3. Proportion of exergy losses in different part of the system

| Exergy loss percentage | T_e (°C) |
|-----------------------|----------|
|                       | -60      | -55 | -50 | -45 | -40 | -35 | -30 |
| δ_Comp_l              | R1 – R161 | 16.3| 16.16| 16.27| 16.29| 16.69| 16.52| 16.81 |
| δ_Comp_l              | R1 – R40A| 16.77| 17.3 | 17.79| 18.24| 18.64| 18.5 | 18.21 |
| δ_evap_l             | R1 – R161 | 0.245| 0.3054| 0.3631| 0.4146| 0.4569| 0.4877| 0.5065 |
| δ_evap_l             | R1 – R40A| 0.2034| 0.254 | 0.3026| 0.3463| 0.3824| 0.4092| 0.4257 |
| δ_exp_l               | R1 – R161 | 10.67| 9.739 | 9.298| 8.821| 8.886| 8.342| 8.379 |
| δ_exp_l               | R1 – R40A| 14.78| 14.68 | 14.56| 14.42| 14.25| 13.28| 12.24 |
| δ_Comp_h             | R1 – R161 | 23.6 | 24.72| 25.49| 26.3 | 26.66| 27.53| 27.88 |
| δ_Comp_h             | R1 – R40A| 19.82| 20.13 | 20.44| 20.76| 21.08| 22.01| 23.02 |
| δ_exp_h               | R1 – R161 | 22.9 | 23.62 | 23.62| 23.6 | 22.79| 22.75| 21.88 |
| δ_exp_h               | R1 – R40A| 25.76| 25.42 | 25.06| 24.69| 24.97| 25.7 |       |
| δ_cond_h             | R1 – R161 | 5.907| 6.064 | 6.006| 5.95 | 5.673| 5.622| 5.349 |
| δ_cond_h             | R1 – R40A| 1.806| 1.836 | 1.871| 1.912| 1.959| 2.057| 2.166 |
| δ_cas                | R1 – R161 | 20.38| 19.38 | 18.95| 18.62| 18.85| 18.75| 19.21 |
| δ_cas                | R1 – R40A| 20.87| 20.38 | 19.97| 19.64| 19.4 | 18.77| 18.24 |

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