

International Engineering Research Journal

STUDY & SIMULATION OF CO₂ REFRIGERATION SYSTEM

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Abstract

Carbon dioxide is not new to refrigeration. In the mid-nineteenth century carbon dioxide was used as refrigerant and its use increased steadily, reaching highest in the 1920s. Its use declined with the introduction of chlorofluorocarbons (CFCs) that operated at much low pressures. Use of CO₂ continued, but mainly in cascade systems for industrial and process applications. Recently, keen interest has been shown in CO₂ as a refrigerant by vending machine manufacturers. There are also possibilities for other light commercial refrigeration applications, as well as for residential air conditioning also including the industrial sector applications that involves cold storage.

Keywords—compressor; trans-critical; sub-critical; cascade system; critical point.

1. Introduction

Carbon Dioxide (CO₂) is a component of our atmosphere that is essential to life. It has no ozone depletion potential and global warming potential, so CO₂ has no regulatory liability, as do HFCs. Other principal benefits of CO₂ are that it is a natural substance; it is cheaper, easily available, not poisonous in any common concentration, and non-flammable. Currently it is not an easy matter for decision makers in commercial refrigeration to make a definitive choice when it comes to refrigerants and system type. The sector has been in the environmental spotlight in recent years, especially as the leakage studies have revealed the true effects of HFC emissions. Considerable reductions in emissions are certainly possible, but they do require some changes.

The challenge that is being faced in implementing this cycle is the trans-critical conditions of the gas which is very difficult to control. But surely it has its advantages as well as disadvantages. The major challenges in CO₂ refrigeration involve the relatively high working pressures.

R-744 (CO₂) is a leading option for environmental reasons, and it can be a winner for power consumption as developments of component technology and application methods continue to reveal potential performance gains. Good experience has been gained with different system configurations over past years, particularly in central and northern Europe. The confidence resulting from this experience ensures that CO₂ will be a long-term option in the foreseeable future.

2. Problem Statement

In early days of refrigeration the two refrigerants most commonly used were ammonia and carbon dioxide. Both were problematic - ammonia is toxic and carbon dioxide requires extremely high pressure (In

between 30 and 200 atmospheres) to operate in a refrigeration cycle, and since it operates on a trans-critical cycle the compressor outlet temperature is extremely high (around 160°C). When Freon 12 was successfully discovered it totally took over as the refrigerant of choice. It is an extremely stable, non-toxic fluid, which does not interact with the compressor lubricant, and operates at pressures always higher than atmospheric pressure.

Unfortunately when the refrigerant leaks and makes its way up to the ozone layer the ultraviolet radiation breaks the molecule releasing the highly active chlorine radicals, which helps in depletion of ozone layer. Freon 12 has since been banned from its usage on a global scale, and has been essentially replaced by chlorine free R134a (tetra-fluoro-ethane) not as stable as Freon 12, however it does not have ozone depletion characteristics.

Recently, however, the international scientific consensus is that Global Warming is caused by human energy related activity, and various man made substances are defined on the basis of a Global Warming Potential (GWP) with reference to carbon dioxide (GWP = 1). R134a has been found to have a GWP of 1300 and in Europe, within a few years, automobile air conditioning systems will be discarded from its use as a refrigerant. The new hot topic is use of carbon dioxide as a refrigerant. The previous two major problems of high pressure and high compressor temperature are found in fact to be advantageous. The very high pressure of a cycle results in a high fluid density throughout the cycle, allowing miniaturization of the systems for the same heat pumping power requirements.

3. Trans-critical Cycle

Although R-744 has poor thermodynamic properties with regard to energy efficiency in a traditional reverse refrigeration cycle, it is

considered as an efficient solution to global warming. This is because it contains characteristics beneficial to the environment and can be obtained from industrial process waste based on a vision of sustainable production. When compared with conventional refrigerant fluids, the most notable property of R-744 is its low critical temperature of 31°C.

Refrigerants with lower critical temperatures than the ambient temperature tropical regions cannot use the condensation process for heat rejection. Instead, the refrigeration cycle becomes trans-critical located between sub-critical and super-critical temperatures. As per, the greatest difference between cycles using R-744 and R134a for example, is in the line following compression. In the conventional system, the phase change (vapor – liquid) known as liquid condensation, occurs after compression. However, in the trans-critical cycle gas is highly overheated and cooled, without phase change, in a gas cooler.

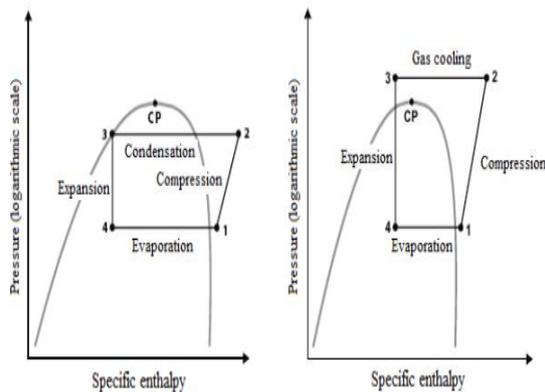


Fig 1: Pressure -Enthalpy Diagram for the subcritical cycle with R134a and trans-critical cycle with R-744.

This is a unique characteristic of the R-744 cycle, where gas at super-critical temperatures is cooled in a gas cooler rather than in a condenser. The heat discharge process occurs at constant pressure and above the critical point. The temperature keeps varying between the inlet and outlet of the gas cooler. The ultimatum is that the COP of the cycle is increased when the liquid outlet temperature of the gas cooler is lower.

The use of an internal heat exchanger in a cycle improves the heat transfer rate there by increasing the performance characteristics of the refrigeration cycle. The following figures represent the trans-critical simple stage and two stage cycles.

Bibliographic research determined that the type of expansion device used differs amongst various authors. According to Madsen et al. (2005), capillary tubes are recommended in applications where evaporation pressure is constant and the discharge temperature from the gas cooler varies no more than ± 10 K from the initial project condition. Capillary tube is more flexible to sudden and ambient temperature changes and it can provide good control of optimal high pressure, offering maximum COP and greater energy efficiency. Combine it with a two-stage cycle, the efficiency rating is off the chart and shoots up very highly. A significant portion of refrigeration

installations operate between evaporation and condensation temperatures whose difference varies from 50°C to 80°C. Although such a marked temperature difference presents a series of operational problems, it also imposes a search for non-trivial solutions.

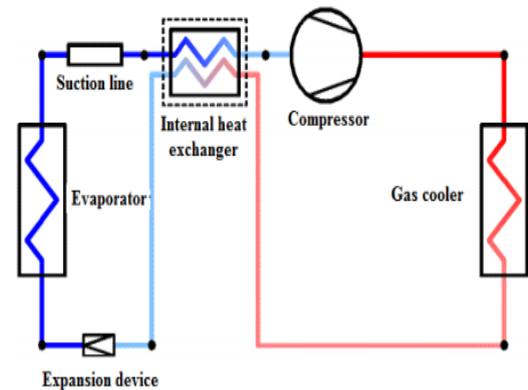


Fig 2: Diagram of a trans-critical single stage compression refrigeration cycle.

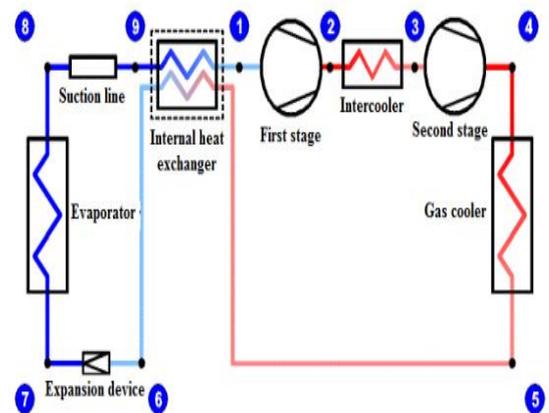


Fig 3: Diagram of a two-stage trans-critical compression refrigeration cycle.

One of these is multistage compression, which implies an increase in initial installation cost in relation to simple stage compression. In contrast, the use of multiple stages reduces some of the problems resulting from high temperature differences, as well as lowering compression power. Common trans-critical cycles have elevated compressor discharge temperatures in the range of 100°C, primarily owing to the need to achieve high pressures in the gas cooler (approximately 95 bar). A solution adopted to improve performance in trans-critical cycles with R-744 is the use of two-stage compression with inter-cooling. This reduces compressor discharge temperatures in the second stage and produces lower gas cooler outlet temperatures, thereby providing greater refrigeration capacity.

Inter-cooling helps to decrease overheating of the refrigerant as it leaves the low pressure stage. Initially, it may seem that this cooling would result in reduced total compression. In compressed air, inter-cooling is carried out at a relatively high temperature to facilitate cooling by the ambient air. However, in refrigeration systems, inter-cooling

occurs at slow refrigerant temperatures, which requires an additional cost. Inter-cooling does not imply a significant reduction in compression power, and can even increase it.

The main reason for applying this process is to limit refrigerant temperature at compressor discharge. In installations containing reciprocating compressors, high discharge temperatures may compromise compressor lubrication and reduce the useful life of discharge valves. In two-stage air compression, optimal intermediate pressure corresponds to the geometric mean between aspiration pressure (suction) and discharge (gas cooler). According to Yang et al. (2007), optimal intermediate pressure for the two-stage trans-critical cycle can be estimated through the classic two stage air compression formula. An iterative method must be used in order to determine actual optimal intermediate pressure, which is a crucial choice parameter.

Many R-744 systems operate above the critical point some or all of the time. This is not a problem, the system just works differently.

1. R-744 systems operate sub critically when the condensing temperature is below 87.8°F
2. R-744 systems operate trans-critically when the “gas cooler exit temperature” is above 87.8°F and the evaporating temperature is below 87.8°F.

A capacity drop also occurs with HFC systems when the ambient temperature increases, but the change is not as great as it is with R-744 when the change is from sub-critical to trans-critical.

The pressure enthalpy chart in Figure 7 shows an example of a simple R-744 system operating sub-critically at a low ambient temperature and trans-critically at a higher ambient temperature. The chart shows that the cooling capacity at the evaporator is significantly less for trans-critical operation.

It is important that appropriate control of the high side (gas cooler) pressure is used to optimize the cooling capacity and efficiency of the trans-critical cycle. For example: Increasing the high side pressure will increase the cooling capacity of the cycle. This is covered more in the further parts of the report.

4. Cascade System

R-744 is the low-stage refrigerant in the cascade system in which the R-744 is always sub-critical. The heat rejected by the condensing R-744 is absorbed by the evaporating high-stage refrigerant. The high-stage refrigerant is a conventional refrigerant usually a HFC or HC type of system. In which case it is termed as a hybrid cascade system. In some systems the R-744 is used both in the high stage as well as the low stage system. In such cases the low stage system is usually sub-critical and the high stage is usually trans-critical at higher ambient conditions.

The cascade system comprises of:

1. The low-stage, which provides the cooling for the load. It uses R-744 and is always sub-critical.

2. The high-stage, which absorbs heat from the condensing R-744 at the cascade heat exchanger. This stage is usually a HFC or HC. But in case, R-744 is used here too, then this stage will always be in trans-critical state.

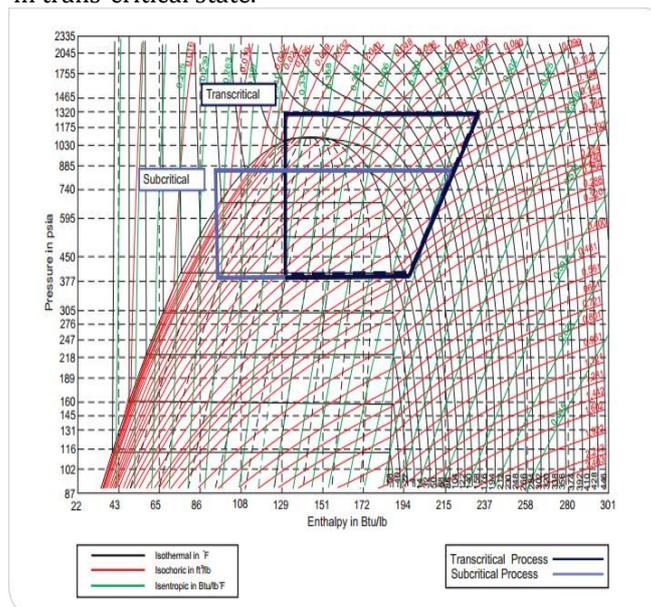


Fig 4: R-744 pressure enthalpy chart showing subcritical and trans-critical systems.

Within the cascade heat exchanger, the evaporating high-stage refrigerant absorbs the heat rejected by the condensing R-744. The R-744 condensing temperature is maintained below the critical point. The high-stage is usually simple, closed loop system. It is controlled by the pressure in the lower stage receiver. The working of the entire system is just like the normal refrigeration system using conventional refrigerants like tetra-flouro ethane and others. The only difference here is that the operating pressures are very very high since the R-744 is involved. Therefore controlling this system is very necessary and essential for the utility of proper need. Hybrid cascade being more feasible but for maximum advantageous usage, having R-744 in both the higher stage as well as lower stage would be very beneficial.

The following shows the example of a simple cascade system using R-744 as the lower stage refrigerant and the higher stage using a HFC or HC. In other terms, it's a hybrid cascade.

In this case, the high stage provides cooling for the medium temperature load as well as removing heat from the condensing R-744 in the low-stage at the cascade heat exchanger. The high stage refrigerant used in this sytem is usually a Hydro-flouro-carbon (HFC) or HC (hydro-carbon) in which case it is termed as a hybrid system. The ambient temperatures ranges are as follows:

Let, T be the usage temperature.

If,

- $T < 68 \text{ F}$ - System will be sub-critical.
- $68 \text{ }^\circ\text{F} < T < 77 \text{ }^\circ\text{F}$ - Sytem will be trans-critical.

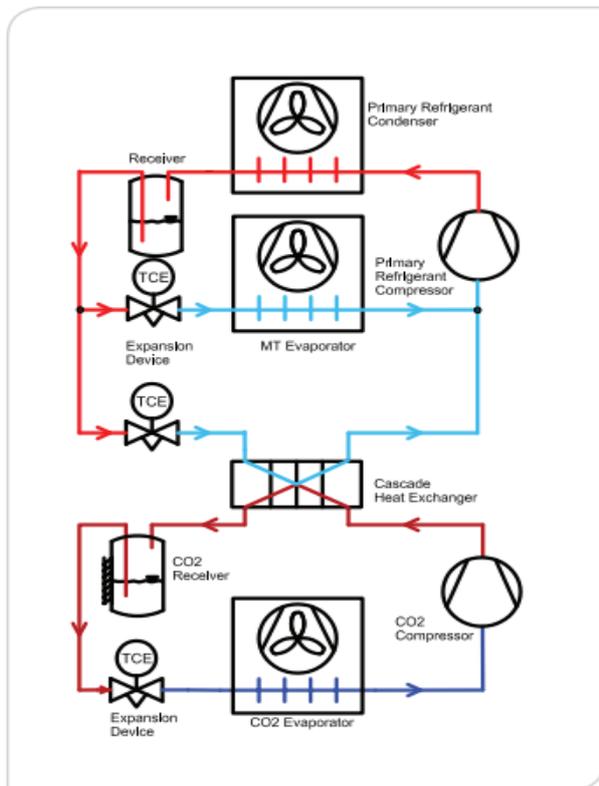


Fig 5: R-744 Simple Cascade System.

5. Software Analysis

This is the stage of the project where the simulation model is being created. The software used for this purpose is **CoolPack V1.5**. It is a refrigeration simulation utility tool that incorporates the different refrigeration cycles known to man along with all types of refrigerants available. It is a uttermost important tool in the field of refrigeration and air conditioning as it used to test the feasibility of the cycle by plotting the different charts that help define it.

Given the details of a one stage refrigeration cycle.

1. Refrigerant Used: R134a (Tetra-flouro-ethane)
2. Condenser Temperature: 35 °C
3. Super-heating: 10 K
4. Evaporator Temperature:-20 °C
5. Sub-cooling: 2 K
6. Isentropic Efficiency: 0.7
7. Cooling Capacity: 100 kW
8. Heat Loss Factor: 10%
9. Pressure losses in the suction and discharge lines is taken as 0.5 K.
10. There is no heat exchanger in the system.
11. Un-useful superheat = 1 K.

The solution procedure is to navigate to the cycle specs screen and input the values as given in the data. The following screenshot gives that. Values in the green colour are the input values and the one in grey are the predefined values before entering the desired values. The main aim is to find out the COP. So once the values have been entered, the next step is to calculate. Select the calculate and the COP turns out to be,

CYCLE SPECIFICATION			
TEMPERATURE LEVELS	PRESSURE LOSSES	SUCTION GAS HEAT EXCHANGER	REFRIGERANT
T_E [°C]: -20.0 ΔT_{SH} [K]: 10	ΔP_{SL} [K]: 0.5	No SGHX 0.00	R134a
T_C [°C]: 35.0 ΔT_{SC} [K]: 2	ΔP_{DL} [K]: 0.5		
CYCLE CAPACITY			
Cooling capacity \dot{Q}_E [kW]: 100	\dot{Q}_E : 10 [kW]	\dot{Q}_C : 12.91 [kW]	\dot{m} : 0.03512 [kg/s] \dot{V}_S : 17.35 [m ³ /h]
COMPRESSOR PERFORMANCE			
Isentropic efficiency η_1 [-]: 0.7	η_1 : 0.700 [-]	\dot{W} : 3.146 [kW]	
COMPRESSOR HEAT LOSS			
Heat loss factor f_c [%]: 10	f_c : 10.0 [%]	T_2 : 54.5 [°C]	\dot{Q}_{LOSS} : 0.3146 [kW]
SUCTION LINE			
Unuseful superheat $\Delta T_{SH,SL}$ [K]: 1.0	\dot{Q}_{SL} : 68 [W]	T_g : -4.0 [°C]	$\Delta T_{SH,SL}$: 1.0 [K]

Fig 5(a): Sample problem Input screen

CYCLE SPECIFICATION			
TEMPERATURE LEVELS	PRESSURE LOSSES	SUCTION GAS HEAT EXCHANGER	REFRIGERANT
T_E [°C]: -20.0 ΔT_{SH} [K]: 10	ΔP_{SL} [K]: 0.5	No SGHX 0.00	R134a
T_C [°C]: 35.0 ΔT_{SC} [K]: 2	ΔP_{DL} [K]: 0.5		
CYCLE CAPACITY			
Cooling capacity \dot{Q}_E [kW]: 100	\dot{Q}_E : 100 [kW]	\dot{Q}_C : 137.6 [kW]	\dot{m} : 0.6744 [kg/s] \dot{V}_S : 383.34 [m ³ /h]
COMPRESSOR PERFORMANCE			
Isentropic efficiency η_1 [-]: 0.7	η_1 : 0.700 [-]	\dot{W} : 40.94 [kW]	
COMPRESSOR HEAT LOSS			
Heat loss factor f_c [%]: 10	f_c : 10.0 [%]	T_2 : 66.9 [°C]	\dot{Q}_{LOSS} : 4.094 [kW]
SUCTION LINE			
Unuseful superheat $\Delta T_{SH,SL}$ [K]: 1.0	\dot{Q}_{SL} : 618 [W]	T_g : -9.0 [°C]	$\Delta T_{SH,SL}$: 1.0 [K]

Fig 5(b) : Solution to the sample problem.

COP : 2.442 COP* : 2.458

Fig 5(b) : COP of the system.

Going by our objective of the project, the main aim is to increase the COP of the refrigeration cycle that we have been given. Heading towards the next segment, implementation of the actual cycle.

5.1. Actual CO₂ Cycle:

In this segment we check the values of the actual CO₂ cycle by entering the values related to the cycle that we follow. The approach we follow and the concept we are going to be working on is how does the COP of the system vary with respect to the change in the following parameters:

1. Variation of the evaporator temperature.
2. Variation of the compressor discharge temperature.
3. Variation of the gas cooler outlet temperature.

Heading on forward to the first parameter that we are working on. It is to be noted that in each of the parameters we will be considering 2 cases.

Case 1: With Heat Exchanger.
Case 2: Without Heat Exchanger.

5.1.1 Variation of Evaporator Temperature (T₁):

The evaporator temperature is the value of the temperature that tells that the evaporation of the refrigerant takes place at that temperature. In other terms the value indicates the present temperature of the evaporator while producing the required refrigeration effect that is needed.

The following screen is what appears while entering the values and specification of the cycle that we assume. The screenshot will be shown in only this example and rest of the examples, the process remains same. Only the value of the specific parameter has to be changed. The values that we enter are as follows:

1. Evaporator Temperature: -10 C
2. Superheating: 5K
3. Gas Cooler Pressure: 10 MPa
4. Gas Cooler Outlet Temp: 30 C
5. Cooling Capacity : 10 kW
6. Effectiveness of the HE : 0.6

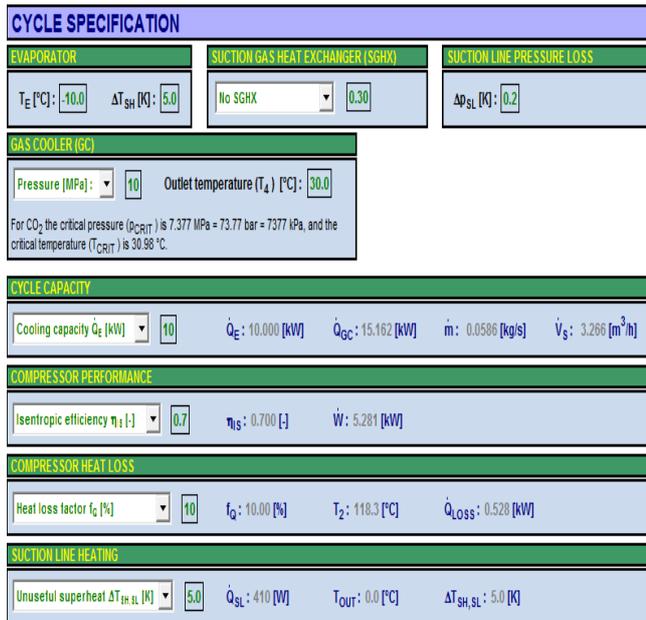


Fig 6: Variation of Evaporator Temperature.

Tabulations:

Case 1: At P₂ = 8 MPa

Table No 1: Tabulation of Changes in Evaporator Temperature Case 1.

T _{evap} (°C)	Without Heat Exchanger			With Heat Exchanger		
	R (kJ/kg)	W (kJ/kg)	COP	R (kJ/kg)	W (kJ/kg)	COP
10	35.9	4.6	7.8	174.6	23.4	7.46
0	51.1	9.2	5.55	201.1	42.5	4.73
-5	57.5	12.3	4.67	212.2	54.5	3.89

Case 2: At P₂ = 9 MPa

Table No. 2: Tabulation of Changes in Evaporator Temperature Case 2

T _{evap} (°C)	Without Heat Exchanger			With Heat Exchanger		
	R (KJ/kg)	W (KJ/kg)	COP	R (kJ/kg)	W (kJ/kg)	COP
10	91.9	12.3	10	91.9	12.3	10
0	105.2	19.5	0	105.2	19.5	0

Case 3: At P₂ = 10 MPa

Table No. 3: Tabulation of Changes in Evaporator Temperature Case 3

T _{evap} (°C)	Without Heat Exchanger			With Heat Exchanger		
	R (kJ/kg)	W (KJ/kg)	COP	R (kJ/kg)	W (kJ/kg)	COP
10	121.7	17.8	10	121.7	17.8	10
0	133.9	26.5	0	133.9	26.5	0
-5	138.8	31.6	-5	138.8	31.6	-5

Case 4: When P₂ = 11 MPa

Table No. 4: Tabulation of Changes in Evaporator Temperature Case 4.

T _{evap} (°C)	Without Heat Exchanger			With Heat Exchanger		
	R (kJ/kg)	W (kJ/kg)	COP	R (kJ/kg)	W (kJ/kg)	COP
10	132.4	21.6	10	132.4	21.6	10
0	144.3	31.1	0	144.3	31.1	0

5.1.2 Variation of Compressor Discharge Pressure(P₂):

Compressor discharge pressure is defined as the discharge line pressure of the compressor after pressurizing the refrigerant. The values of the other parameters are kept the same as mentioned above before the start of the first parameter variation.

Tabulations:

Table No. 5: Tabulation of Changes in Compressor Discharge Pressure.

P ₂ (bar)	Without Heat Exchanger			With Heat Exchanger		
	R (kJ/kg)	W (kJ/kg)	COP	R (kJ/kg)	W (kJ/kg)	COP
8	35.9	4.6	8	35.9	4.6	8
9	91.9	12.3	9	91.9	12.3	9
10	121.7	17.8	10	121.7	17.8	10
11	132.4	21.6	11	132.4	21.6	11
12	139.5	24.4	12	139.5	24.4	12
13	144.6	27.2	13	144.6	27.2	13

5.1.3 Variation of Gas Cooler Outlet Temperature:(T₃)

The gas cooler outlet temperature is defined as the value of temperature of the refrigerant that comes out of the gas cooler after the refrigerant is being transformed to low temperature liquid.

Tabulations:

Case 1: When P₂ = 8 MPa

Table No. 6: Tabulation of Changes in Gas Cooler Outlet Temperature Case 1.

T ₃ (°C)	Without Heat Exchanger			With Heat Exchanger		
	R (kJ/kg)	W (kJ/kg)	COP	R (kJ/kg)	W (kJ/kg)	COP
40	35.9	4.6	40	35.9	4.6	40
45	18.6	2.5	45	18.6	2.5	45
50	6.2	0.9	50	6.2	0.9	50

Case 2: When P₂ = 9 MPa

Table No. 7: Tabulation of Changes in Gas Cooler Outlet Temperature Case 2

T ₃ (°C)	Without Heat Exchanger			With Heat Exchanger		
	R (kJ/kg)	W (KJ/kg)	COP	R (kJ/kg)	W (kJ/kg)	COP
40	91.9	12.3	40	91.9	12.3	40
45	49.8	7.5	45	49.8	7.5	45
50	29.4	4.7	50	29.4	4.7	50

Case 3: When P₂ = 10 MPa

Table No. 8: Tabulation of Changes in Gas Cooler Outlet Temperature Case 3.

T ₃ (°C)	Without Heat Exchanger			With Heat Exchanger		
	R (kJ/kg)	W (kJ/kg)	COP	R (kJ/kg)	W (kJ/kg)	COP
40	121.7	17.8	40	121.7	17.8	40
45	89.7	14.3	45	89.7	14.3	45
50	58.0	10.2	50	58.0	10.2	50

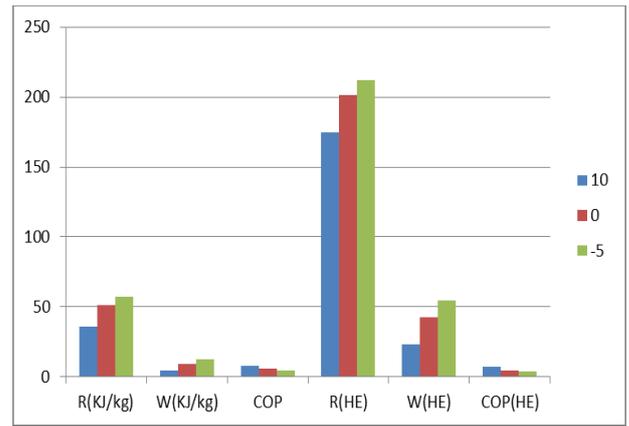
The observations and trends that are observed while analysing the variation in parameters. The next step is plotting the bar graphs of the observed trends in the parameters that have been varied. In order to do so, we need to come to rough conclusion regarding the tabulations. How has the COP varied on changing what, this is a worthy question that comes into the mind. These tabulations have been taken manually without the use of any other software or analysis tool.

6. Results & Discussion

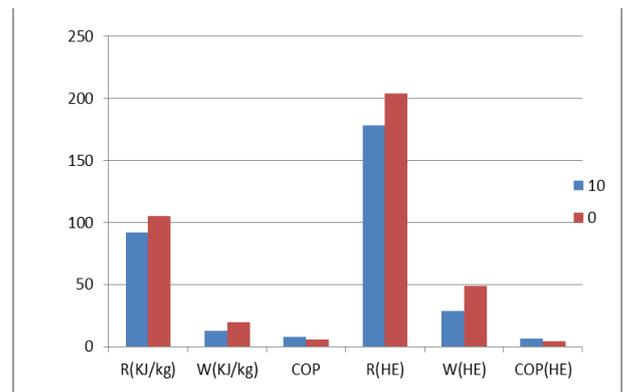
6.1 Results:

In this segment we plot the bar graphs on the observed trend of each varying parameter. This is to be done on an excel sheet. These are the results of the bar graph that have been attained and formulated.

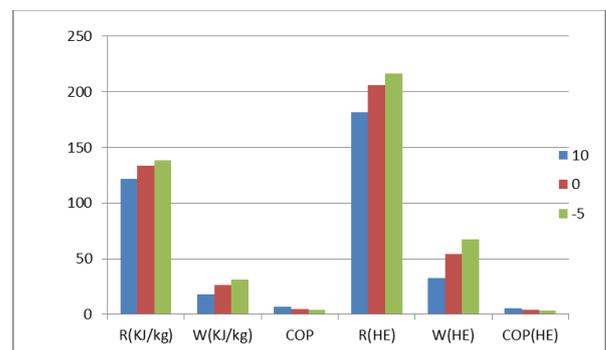
Case1:Varying the Evaporator Temperature:(T_{evap})



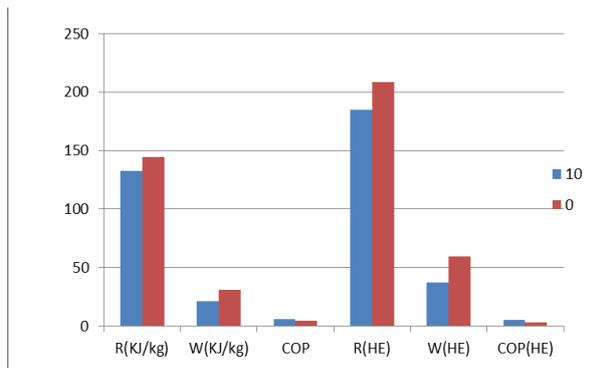
Graph1: Bar graph of change in evaporator temperature Case 1.



Graph 2: Bar graph of change in evaporator temperature Case 2.

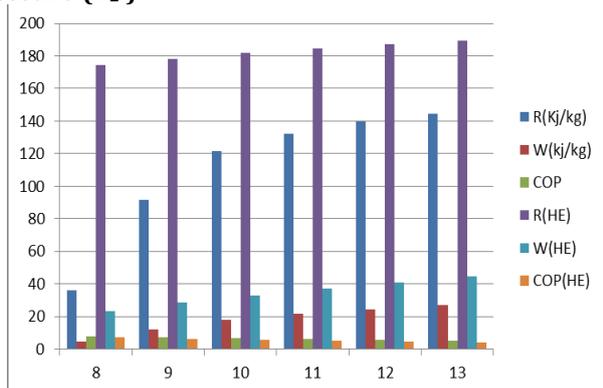


Graph 3: Bar graph of change in evaporator temperature Case 3.



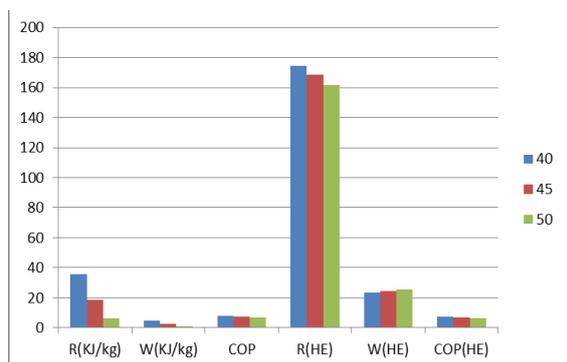
Graph 4: Bar graph of change in evaporator temperature Case 4.

Case2: Varying the Compressor Discharge Pressure:(P_2)

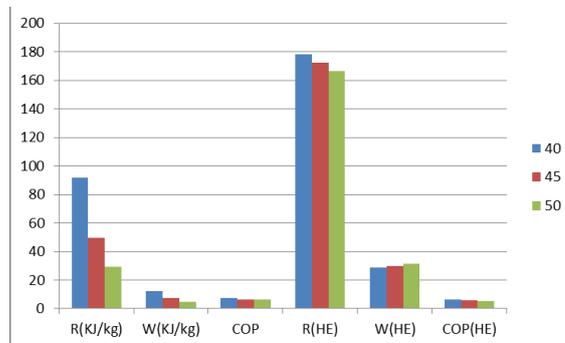


Graph 5: Bar graph of change in Discharge Pressure.

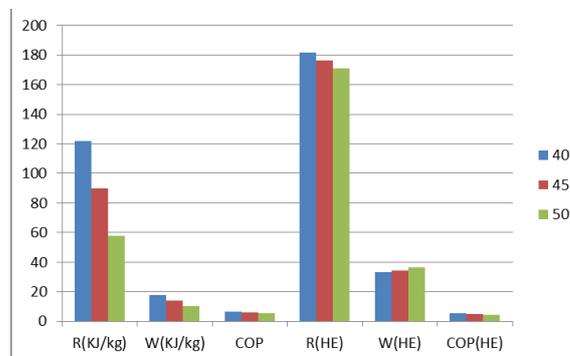
Case3: Varying the Gas Cooler Outlet Temperature:(T_3)



Graph 6: Bar graph of change in gas cooler outlet temperature Case 1.



Graph 7: Bar graph of change in gas cooler outlet temperature Case 2.



Graph 8: Bar graph of change in gas cooler outlet temperature Case 3.

6.2 Discussion:

From the previous results that are in the last segment, we have concluded and drawn out the trends that have been observed. The COP in each case has been turning out have a different outcome. The trends observed are as follows:

Case 1: Varying T_{evap}

The effectiveness or efficiency of the heat exchanger is been kept 0.6 and the gas cooler discharge temperature has been kept as 40 °C. The inferences that have been drawn in this case when varying the evaporator temperature are as follows:

1. The refrigeration effect (R) increases with the increase in evaporator temperature.
2. The Work done (W) also increases.
3. The COP of the system decreases as the work done increases.
4. (COP)_{With HE} < (COP)_{Without HE}

Case 2: Varying P_2

The effectiveness or efficiency of the heat exchanger is been kept 0.6 and the gas cooler discharge temperature has been kept as 40 °C. The inferences that have been drawn in this case when varying the compressor discharge pressure are as follows:

1. The refrigeration effect (R) increases with the increase in compressor discharge pressure.
2. The Work done (W) also increases.
3. The COP of the system decreases as the work done increases.
4. (COP)_{With HE} < (COP)_{Without HE}

Case 3: Varying T_3 :

The effectiveness of the heat exchanger is kept the same of 0.6. But the saturation temperature of the evaporator is now kept as 10°C . The summarized inferences of the observations have been listed below. They are:

1. The refrigeration effect (R) increases.
2. The net work done (W) decreases without heat exchanger.
3. The net work done (W) increases with heat exchanger.
4. The COP of the system decreases.
5. $(\text{COP})_{\text{With HE}} < (\text{COP})_{\text{Without HE}}$

CONCLUSION:

- (1) High pressure CO_2 systems were already developed by the end of 19th century and used in many of the refrigeration applications.
- (2) In the last 15 years, this technology was identified as an environmental friendly solution in commercial refrigeration based on low GWP (Global Warming Potential) of R-744 and resulting in low **Total Equivalent Warming Impact (TEWI)** compared to HFC refrigerants.
- (3) The applications are mainly cold storage and super-market retail systems.
- (4) Carbon-di-oxide refrigeration provides a viable option to deal with today's and future environmental challenges.
- (5) Taking a variety of factors in to account and consideration, the conclusion drawn from the implementation of this project is that the use of carbon-di-oxide can prove to be the beginning of a new era of refrigeration.
- (6) The refrigerant is an old one, but if it can attain modern methods of cycle implementation, it surely can achieve new standards in the refrigeration and air conditioning industry.

- (7) There is a real advantage in developing a R-744 competent team giving a competitive edge, taking up on the industry standards and leading to more end user respect and consumer satisfaction.
- (8) This would benefit the entire refrigeration sector as a whole since the entire specifications and developments would be in accordance with the norms of the Kyoto Protocol.

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