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Study & Simulation of Co2 Refrigeration System

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ABSTRACT

Carbon-dioxide is not new to refrigeration. Its use began in the mid-nineteenth century and steadily increased, reaching a peak in the 1920s. Its use declined with the introduction of chlorofluorocarbons (CFCs) that operated at much lower pressures. Use of CO_2 continued, but chiefly in cascade systems for industrial and process applications. Recently, strong interest has been shown in CO_2 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.

I. INTRODUCTION

Carbon-Di-Oxide (CO_2) is a component of our atmosphere that is essential to life. It has no ozone depletion potential and insignificant global warming potential, so CO_2 has no regulatory liability, as do HFCs. There is no need to account for the amount used, and it does not need to be reclaimed. Other principal benefits of CO_2 are that it is a natural substance; it is cheap, readily 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. For the last decade, many refrigerant options and system architectures have appeared both on paper and in practice. The sector has been in the environmental spotlight in recent years, especially as the leakage studies have revealed the true effects of HFC emissions in centralized systems. Considerable reductions in emissions are certainly possible, but they do require changes.

The challenge that is being faced in implementing this cycle is the trans-critical conditions of the gas which are very difficult to control. But it surely has its advantages and disadvantages as well. Those in detail are being discussed below as a part of the project that we carry out. The major challenges in CO_2 refrigeration involve the relatively high working pressures.

R-744 (CO_2) 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 many years, particularly in central and northern Europe. The confidence resulting from this experience ensures that CO_2 will be a long-term option in the foreseeable future.

II. PROBLEM STATEMENT

In the early days of refrigeration the two refrigerants in common use were ammonia and carbon dioxide. Both were problematic - ammonia is toxic and carbon dioxide requires extremely high pressures (from around 30 to 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 (dichloro-diflouro-methane) was newly 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 somewhat higher than atmospheric, so that if any leakage occurred, air would not leak into the system, thus one could recharge without having to apply vacuum.

Unfortunately when the refrigerant does ultimately leak and make its way up to the ozone layer the ultraviolet radiation breaks up the molecule releasing the highly active chlorine radicals, which help to deplete the ozone layer. Freon 12 has since been banned from usage on a global scale, and has been essentially replaced by chlorine free R134a (tetra-flouro-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 barred from using R134a as a refrigerant. The new hot topic is a return to 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 cycle pressure results in a high fluid density throughout the cycle, allowing miniaturization of the systems for the same heat pumping power requirements.

III. TRANS-CRITICAL CYCLE

Although R-744 has poor thermodynamic properties with regard to energy efficiency in a traditional reverse refrigeration cycle, it is considered an efficient solution to anthropic 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 (Cecchinato et al., 2009). When compared with conventional refrigerant fluids, the most notable property of R-744 is its low critical temperature of 30.98^{0}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 subcritical and supercritical temperatures (DanFoss, 2008). 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.

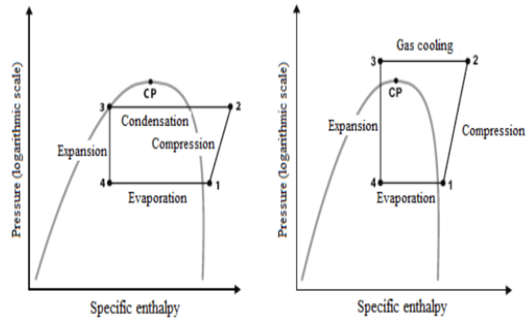

Figure 1: Pressure -Enthalpy Diagram for the subcritical cycle with R134a and trans-critical cycle with R-744

This is a unique characteristic of the R-744 cycle, where gas at supercritical temperatures is cooled in a gas cooler rather than a condenser. The heat discharge process occurs at constant pressure and above the critical point. The temperature keeps varying between the inlet and the 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.

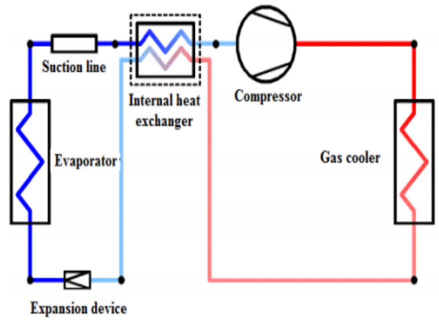

Figure 2: Diagram of a trans-critical single stage compression refrigeration cycle

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 o f 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.

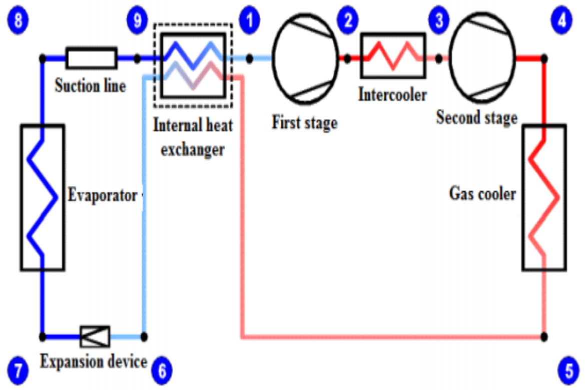

Figure 3: Diagram of a two-stage trans-critical compression refrigeration cycle

A significant portion of refrigeration installations operate between evaporation and condensation temperatures whose difference varies from 50ºC to 80ºC. Although such a marked temperature difference presents a series of operational problems, it also imposes a search for non-trivial solutions. 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 (Stoecker and Jabardo, 2002). Common trans-critical cycles have elevated compressor discharge temperatures in the range of 100ºC, 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 intercooling. This reduces compressor discharge temperatures in the second stage and produces lower gas cooler outlet temperatures, thereby providing greater refrigeration capacity. Energy efficiency is an essential requirement for any substitute of traditional refrigerant liquids. Thus, R-744 can only be successful in operating conditions of more sophisticated cycles and/or more effective components in order to compensate for the lower energy efficiency in the trans-critical cycle of reference (Cecchinato et al., 2009).

Intercooling aims 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, intercooling is carried out at a relatively high temperature to facilitate cooling by the ambient air. However, in refrigeration systems, intercooling occurs at somewhat low refrigerant temperatures, which requires an additional cost. Intercooling 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 (Stoecker and Jabardo, 2002). In two-stage air compression, optimal intermediate pressure corresponds to the geometric mean between aspiration pressure (suction) and discharge (gas cooler), as shown in Eq. (1) (Stoecker and Jabardo, 2002). 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 (Agrawal et al., 2007), which is a crucial choice parameter (Cecchinato et al., 2009).

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.

- R-744 systems operate sub critically when the condensing temperature is below 87.8°F
- 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- to transcritical.

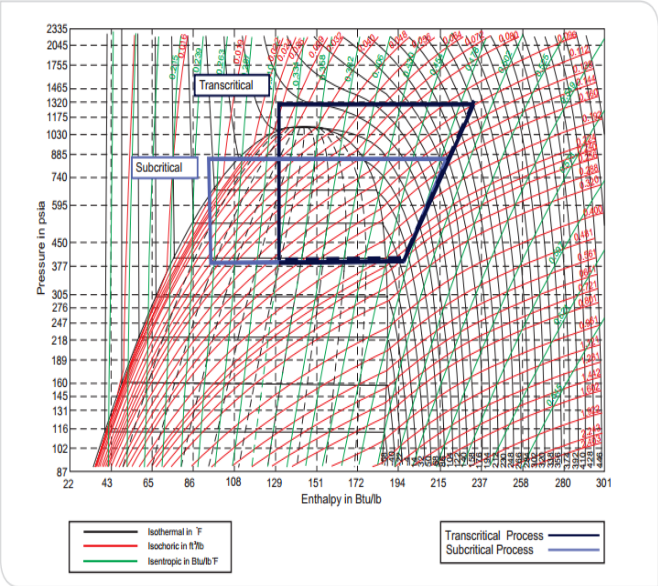

Figure 4: R-744 pressure enthalpy chart showing subcritical and trans-critical systems

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.

IV. 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 bu 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:

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

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 reciever. 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 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.

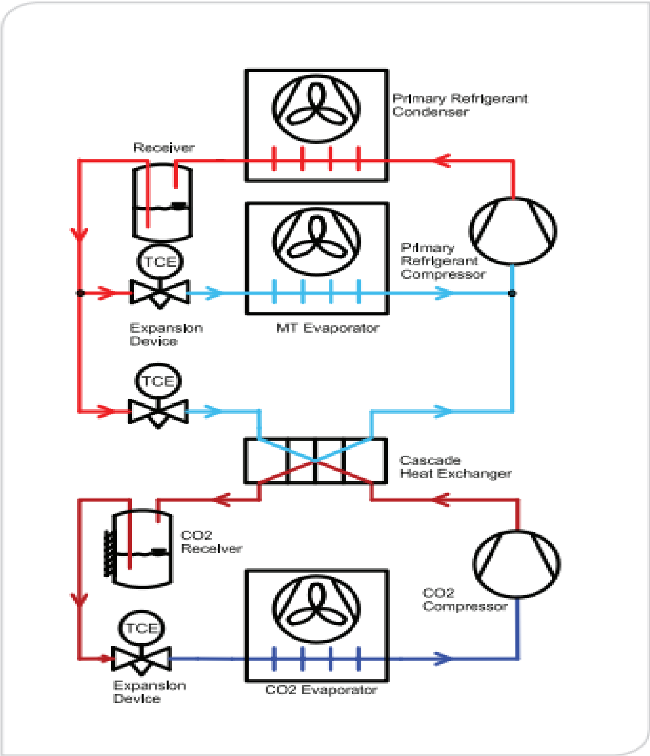

Figure 5: R-744 Simple Cascade System

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 system 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,

$T < 68\ ^0F$ - System will be sub-critical.

$68\ ^0F < T < 77\ ^0F$ - System will be trans-critical.

V. 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

- Refrigerant Used: R134a (Tetra-flouro-ethane)
- Condenser Temperature: 35 C
- Super-heating: 10 K
- Evaporator Temperature:-20 ^{0}C
- Sub-cooling: 2 K
- Isentropic Efficiency: 0.7
- Cooling Capacity: 100 kW
- Heat Loss Factor: 10%
- Pressure losses in the suction and discharge lines are taken as 0.5 K.
- There is no heat exchanger in the system.
- 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,

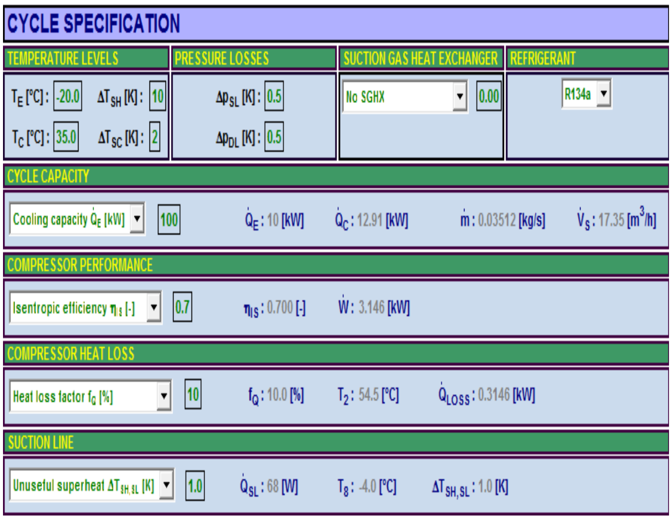

Figure 6: Sample problem Input screen

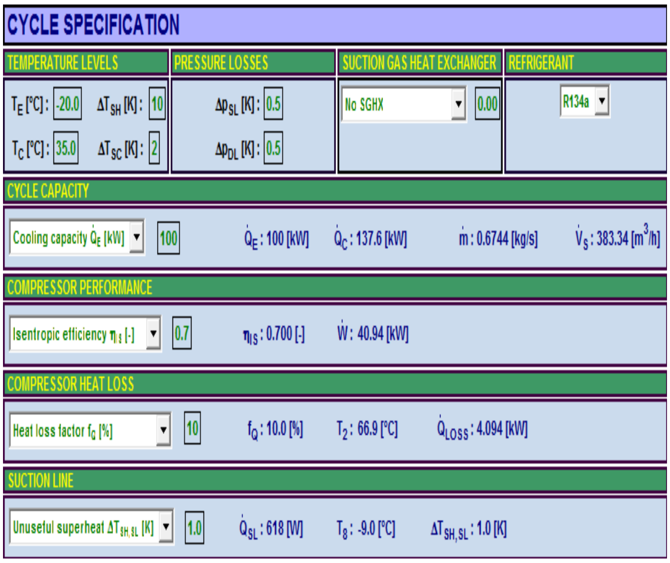

Figure 7: Solution to the sample problem

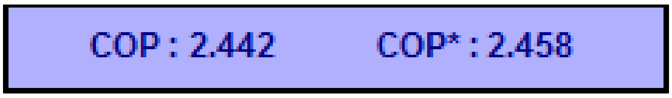

Figure 8: COP of the system

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

A. Actual CO2 Cycle

In this segement 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 the COP of the system varies with repect 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

B. Variation of Evaporator Temperature (T_1)

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.

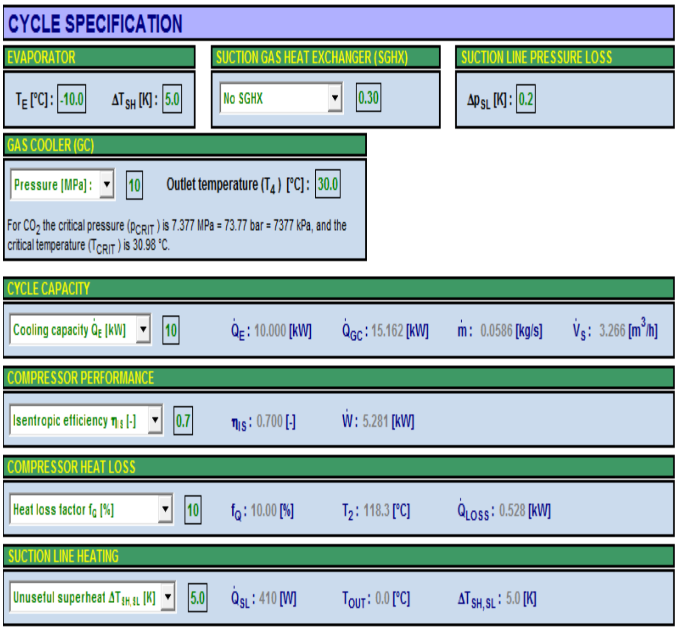

Figure 9: Variation of Evaporator Temperature

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

Tabulations:

Case 1: At P_2 = 8 MPa

Table 1: Tabulation of Changes in Evaporator Temperature Case 1

T_{evap} (^{0}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 P_2 = 9 MPa

Table 2: Tabulation of Changes in Evaporator Temperature Case 2

T_{evap} (^{0}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 P_2 = 10 MPa

Table 3: Tabulation of Changes in Evaporator Temperature Case 3

T_{evap} (^{0} 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 P_2 = 11 MPa

Table 4: Tabulation of Changes in Evaporator Temperature Case 4

T_{evap} (^{0}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

C. 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.

Table 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

D. 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.

Case 1: When P_2 = 8 MPa

Table 6: Tabulation of Changes in Gas Cooler Outlet Temperature Case 1

T_3 (^{0}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 P_2 = 9 MPa

Table 7: Tabulation of Changes in Gas Cooler Outlet Temperature Case 2

T_3 (^{0}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 P_2 = 10 MPa

Table 8: Tabulation of Changes in Gas Cooler Outlet Temperature Case 3

T_3 (^{0}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.

VI. RESULTS AND DISCUSSION

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.

Case 1: Varying the Evaporator Temperature(T_{evap})

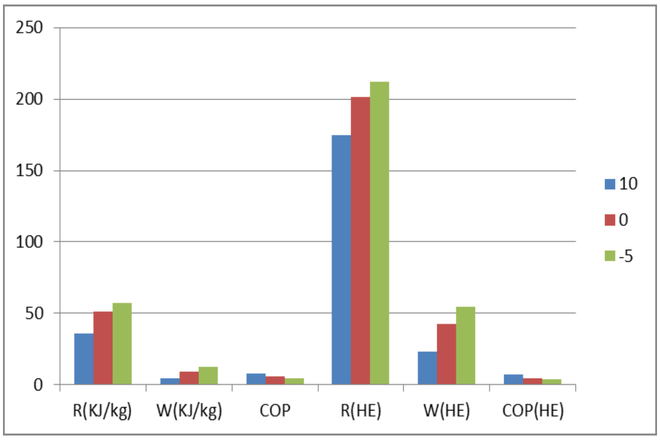

Figure 10: Bar graph of change in evaporator temperature Case 1

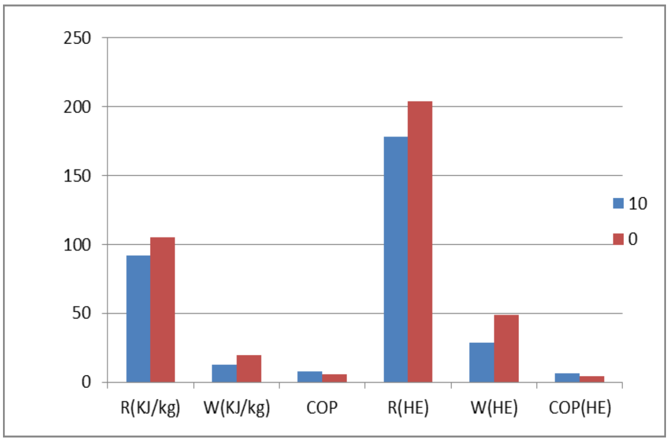

Figure 11: Bar graph of change in evaporator temperature Case 2

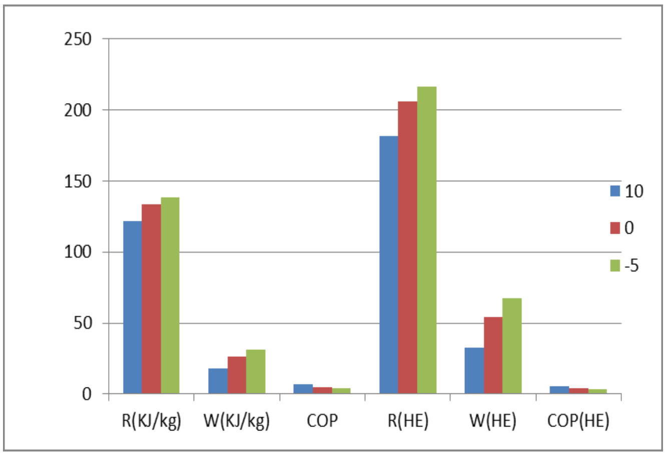

Figure 12: Bar graph of change in evaporator temperature Case 3

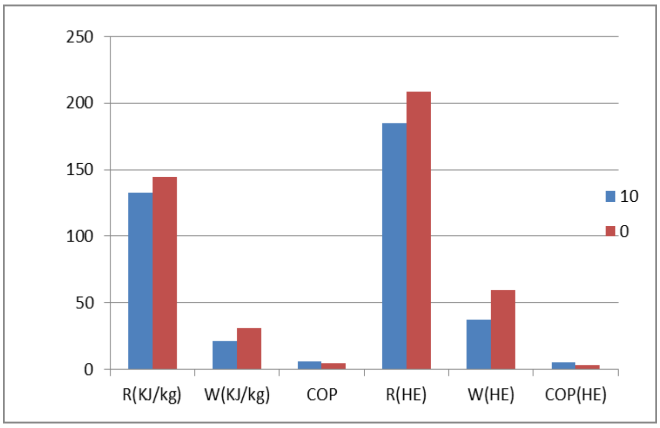

Figure 13: Bar graph of change in evaporator temperature Case 4

Case 2: Varying the Compressor Discharge Pressure:(P_2)

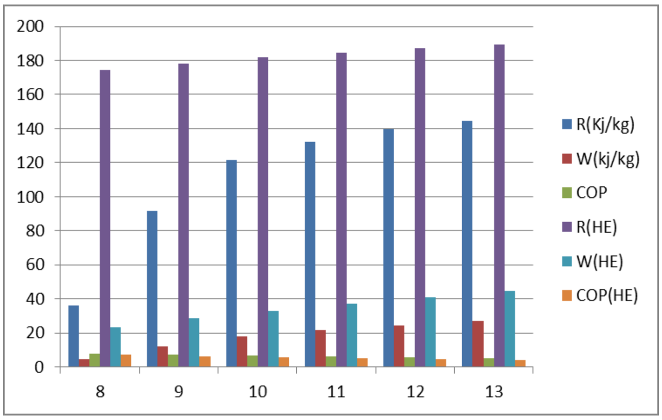

Figure 14: Bar graph of change in Discharge Pressure

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

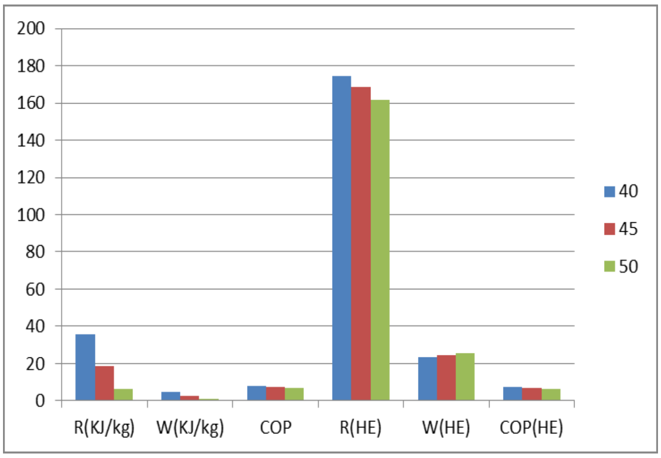

Figure 15: Bar graph of change in gas cooler outlet temp Case 1

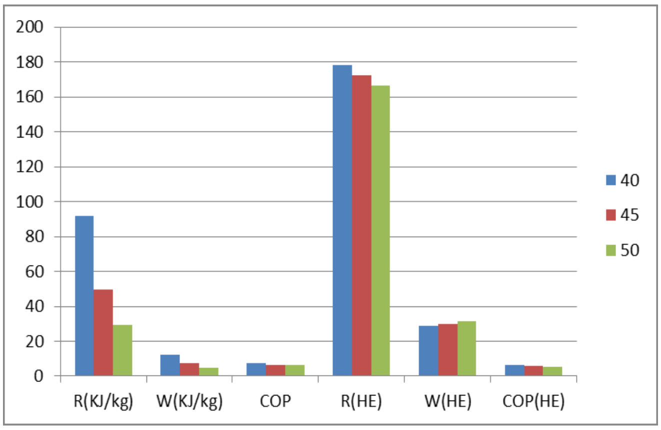

Figure 16: Bar graph of change in gas cooler outlet temp Case 2

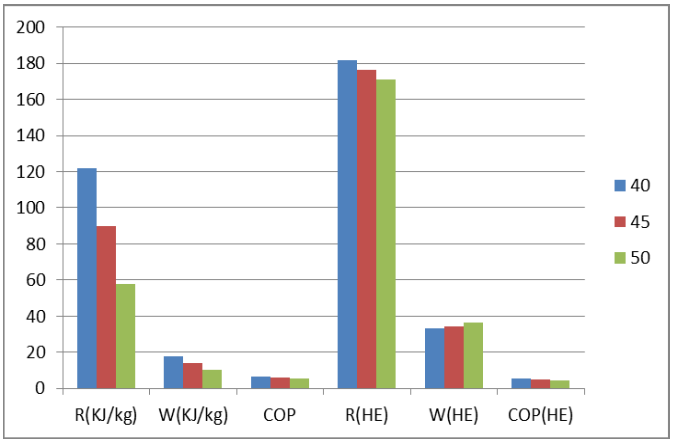

Figure 17: Bar graph of change in gas cooler outlet temp Case 3

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 ^{0}C. The inferences that have been drawn in this case when varying the evaporator temperature are as follows:

- The refrigeration effect (R) increases with the increase in evaporator temperature.
- The Work done (W) also increases.
- The COP of the system decreases as the work done increases.
- $(COP)_{\text{With HE}} < (COP)_{\text{Without HE}}$

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 ^{0}C. The inferences that have been drawn in this case when varying the compressor discharge pressure are as follows:

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

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 ^{0}C. The sumarrized inferences of the observations have been listed below. They are:

- The refrigeration effect (R) increases.
- The net work done (W) decreases without heat exchanger.
- The net work done (W) increases with heat exchanger.

- The COP of the system decreases.
- (COP) $_{\text{With HE}}$ < (COP) $_{\text{Without HE}}$

VII. CONCLUSION

- High pressure CO_2 systems were already developed by the end of 19^{th} century and used in many of the refrigeration applications.
- 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.
- The applications are mainly cold storages and super-market retail systems.
- Carbon-dioxide refrigeration provides a viable option to deal with todays and future environmental challenges.
- 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-dioxide can prove to be the beginning of a new era of refrigeration.
- 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.
- 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.
- 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.

VIII. REFERENCES

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