

# **R744 (CO<sub>2</sub>) Refrigerant Cooling Systems, the Finned Tube R744 (CO<sub>2</sub>) Evaporator and Gas Cooler Design Principles**

**Fatih KASAP**  
Mechanical Engineer  
R&D Engineer

**Hasan ACÜL**  
Mechanical Engineer, NPDP  
R&D Manager

**Hatice CANBAZ**  
Mechanical Engineer  
R&D Engineer

**Selim ERBİL**  
Mechanical Engineer

## **ABSTRACT**

R744 (CO<sub>2</sub>) was used in the early stages of the refrigeration industry, but it lost the competition with halocarbon refrigerants because of its high operating pressure and the loss of capacity and coefficient of performance when rejecting heat near or above the critical point. Because the negative effect of halocarbon refrigerants on environment, that R744 (CO<sub>2</sub>) started to used recently. New heat exchanger technology and system components allow R744 (CO<sub>2</sub>) to reach competitive efficiency levels for transcritical cycle especially in northern countries and for sub-critic cascade cycle in southern countries.

Recently, CO<sub>2</sub> has been increasingly studied for application as the primary refrigerant in transcritical mobile air conditioners, vending machines, supermarkets, cold rooms, food production and process industry, industrial ice cream machineries and heat pumps.

This article is prepared to give general technical information on R744 (CO<sub>2</sub>) cooling systems, CO<sub>2</sub> evaporators and gas coolers.

**Keywords :** CO<sub>2</sub>, R744, Transcritical CO<sub>2</sub> Cycle, Subcritical CO<sub>2</sub> Cycle, Gas Cooler, Natural Refrigerant

## **1. INTRODUCTION**

Throughout the 20th century, economic operations occurring within the triangle of industrialisation-growth-development, led by developed countries have resulted in a pollution that the environment, the habitat of man cannot endure. Soil, water, and air are faced with a level of pollution that threatens human life. Countries that have begun to grasp the magnitude of the problem have commenced seeking permanent solutions [1]. New standards and norms are being developed and implemented each year in this field. To be able to adapt to the developments, companies that are associated with the process are forced to develop technologies, products and manufacturing (process) methods conforming to the new laws and standards.

Like many branches of industry, the air conditioning sector is affected by these developments. Efforts are underway to develop environmentally friendly refrigerants for cooling systems to replace chlorofluorocarbons (CFC) and hydrochlorofluorocarbons (HCFC) which accelerate global warming, harm the ozone layer, and have a negative influence on the environment. Hydrofluorocarbons (HFC) are synthesized from natural gases such as ethane and methane that do not contain chlorine, replacing chlorine with hydrogen, and are consequently described as relatively more environment friendly [2,3]. In addition to CO<sub>2</sub>, natural refrigerants such as isobutane and propane cannot be used in large batch

systems due to their flammable and explosive nature. The R717 (NH<sub>3</sub>, Ammonia) gas on the other hand have such disadvantages as their effects on copper alloys, and the risk of toxicity in large scale systems.

Natural refrigerants that do not have an effect on global warming and do not harm the ozone layer are without a doubt the most significant alternatives in this process. The CO<sub>2</sub> gas presents itself as an effective solution at this point. CO<sub>2</sub> has been widely used in the developing stages of the cooling industry. However, it has been replaced by halocarbon refrigerants due to the drop in coefficient of performance that emerge due to heat transfer around and above the critical point, as well as high operating pressures [4]. The harmful effects of halocarbon refrigerants on the environment have in turn led to a return to CO<sub>2</sub> as an alternative, natural refrigerant. The current machine and exchanger technology, and system control elements have enabled CO<sub>2</sub> to reach competitive levels in terms of efficiency particularly in northern countries through the transcritical cycle, and in the southern countries through the subcritical cascade cycle. Due to its low critical temperature of 31.06°C and high critical pressure of 73.8 bars, R744 (CO<sub>2</sub>) has some additional technical requirements to other conventional refrigerants.

## 2. R744 (CO<sub>2</sub>) AS REFRIGERANT

The use of R744 (CO<sub>2</sub>) as refrigerant was first suggested in 1850 by the Alexander Twining of Britain. Yet the first CO<sub>2</sub> refrigerated system was built toward the end of the 1860's by the American Thaddeus S.C. Lowe. It was widely used in ships and industry until the 1920's due to its clean and reliable nature. With the advent of CFC's in 1928, and due to the reasons mentioned above CO<sub>2</sub> was gradually pushed out of the market, and was no longer used in the 1950's. The increasing restrictions on CFC's toward the end of the 20<sup>th</sup> century have led to new pursuits, causing old natural refrigerants such as CO<sub>2</sub> and NH<sub>3</sub> to be revisited. Proffesor Gustav Lorentzen's patenting of the transcritical CO<sub>2</sub> cycle in 1990 was a milestone in this subject [5].

### 2.1 ENVIRONMENTAL PROPERTIES OF CO<sub>2</sub>

There are two basic concepts regarding the environmental properties of refrigerants.

1. ODP (the Ozone Depletion Potential): Denotes the damage that a gas can have on the ozone layer. It is expressed as a relative figure [6].
2. GWP (the Global Warming Potential): It is a measure of the impact of a greenhouse gas on the environment over a specific time interval in relation to CO<sub>2</sub> , taking the GWP of CO<sub>2</sub> as 1 [6].

CO<sub>2</sub> has an ozone depletion potential of zero (ODP=0) and has very low direct impact on global warming (GWP=1). Table 1 shows the environmental properties of common refrigerants.

Table 1. Enviromental Properties of Various Widely Known Refrigerants [6]

Refrigerant	Refrigerant Classification	ODP	GWP
R 11	CFC	1	4600
R 12	CFC	0,82	10600
R 22	HCFC	0,034	1700
R 134a	HFC	0	1300
R 410a	HFC	0	1980
R 404a	HFC	0	3780
R 407c	HFC	0	1650
R 507a	HFC	0	3850
<b>R744 (CO<sub>2</sub>)</b>	<b>Naturel Refrigerant</b>	<b>0</b>	<b>1</b>
R 717 (NH <sub>3</sub> )	Naturel Refrigerant	0	<1

ODP: (R11=1 admitted) GWP: (CO<sub>2</sub>=1 admitted)

Another concept exists in the literature in addition to ODP and GWP: the Total Equivalent Warming Impact (TEWI). The Total Equivalent Warming Impact (TEWI) denotes the direct or indirect global warming impact of the refrigerant and the system of which it is part. The impact that the refrigerant has on the environment is the direct impact. The CO<sub>2</sub> emission that occurs during the supply of energy to the cooling system of which it is a part is the indirect impact. The sum of the two impacts is defined as the Total Equivalent Warming Impact (TEWI). CO<sub>2</sub> as a refrigerant is effective in reducing total equivalent warming impact in cooling systems [7].

## 2.2 CO<sub>2</sub> SAFETY

Refrigerants are collected in 2 groups with respect to toxicity. Class A rating indicates no evidence of toxicity below 400 ppm. Class B rating indicates evidence of toxicity below 400 ppm. CO<sub>2</sub> is included in Class A, which indicates no toxicity at concentrations below 400 ppm [8].

Refrigerants are divided in 3 classes in terms of flammability. Class 1 indicates a refrigerant which will not burn in the flammability test under conditions of 21°C and 101 kPa pressure. Class 2 indicates refrigerants that is mildly flammable and produces a combustion heat of less than 19 kJ/kg under conditions of 21°C and 101 kPa pressure. Class 3 indicates refrigerants that is highly flammable and produces a combustion heat of more than 19 kJ/kg under conditions of 21°C and 101 kPa pressure. CO<sub>2</sub> is rated as Class 1, as inflammable [8].

Although CO<sub>2</sub> has been described as non-toxic, it has some physical effects on people depending on its concentration in the air. When the volume concentration of CO<sub>2</sub> in the air is between 2 and 3%, rapid breathing and mild headache may be experienced. The Immediately Dangerous to Life or Health (IDLH) concentration has been fixed at 4% and the lethal concentration has been declared as 10%. In practice the %5 limit for CO<sub>2</sub> volume in the air is a quite reasonable figure [9]. Consequently designs should be based on these limits, and the damage on living things in case of an accident or leak should be minimised. Table 2 shows the safety classification of common refrigerants.

Tablo 2. Safety Classification of Various Widely Known Refrigerants [8]

Refrigerant	R 11	R 12	R 22	R 134a	R 410a	R 404a	R 407c	R 507a	<b>R744 (CO<sub>2</sub>)</b>	R 717 (NH <sub>3</sub> )
<b>Toxicity</b>	A	A	A	A	A	A	A	A	<b>A</b>	B
<b>Flammable</b>	1	1	1	1	1	1	1	1	<b>1</b>	2

### 2.3 THERMOPHYSICAL PROPERTIES OF CO<sub>2</sub>

The properties of the refrigerant are very important in designing a refrigeration system. CO<sub>2</sub> is known to have different properties than other conventional refrigerants. In comparison, the most remarkable property of CO<sub>2</sub> is its low critical temperature (31.06°C) and high critical pressure (73.8 bar). Another limiting factor in the application of CO<sub>2</sub> is its high triple point that occurs at -56.6°C and 5.1 bar pressure. Also in comparison with conventional refrigerants (CFC, HCFC, HFC, HC), it has a fairly high steam pressure and volumetric heat transfer capacity (22545 kJ/m<sup>3</sup> at 0°C) [9].

As can be seen from the CO<sub>2</sub> phase diagram, there are three conditions where two phases are in equilibrium: Solid-Gas, Solid-Liquid, and Liquid-Gas equilibrium states. CO<sub>2</sub> can only exist as a solid or gas under atmospheric pressure. The liquid phase is not possible at this pressure. Above this temperature, CO<sub>2</sub> sublimates, transitioning to the gaseous phase. 5.2 bar and -56.6°C are the conditions where all states of CO<sub>2</sub> are in equilibrium. The three states of the substance are in equilibrium under these conditions. CO<sub>2</sub> reaches the critical point at 31.1°C. At this point, the densities of the liquid and gas phases are equal and the difference between the two phases disappears. A new phase emerges as a result. This phase of matter is called the supercritical phase. Figure 1 shows the CO<sub>2</sub> Phase Diagram, while Figure 2 shows the CO<sub>2</sub> In P-h Diagram.

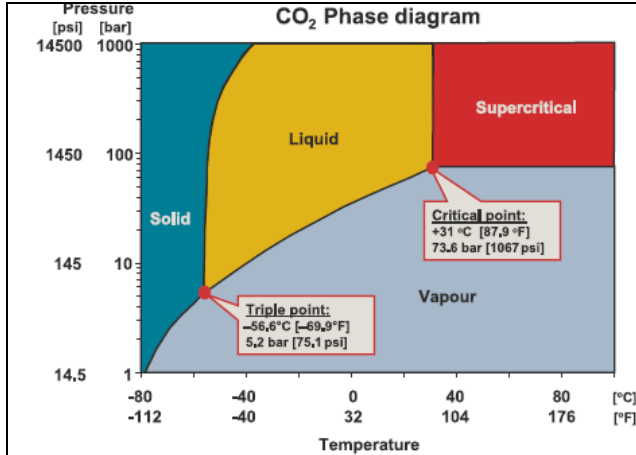
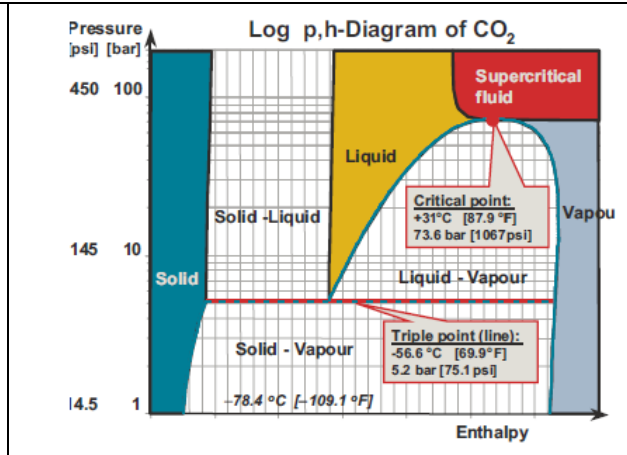


Figure 1. CO<sub>2</sub> Phase Diagram [10]



Şekil 2. CO<sub>2</sub> In P – h Diagram [10]

As shown in Table 3, CO<sub>2</sub> has desirable thermophysical properties and lower viscosity, higher volumetric capacity, higher thermal conductivity, and higher steam density than similar halocarbon refrigerants.

Table 3. Thermophysical Properties of Various Widely Used Refrigerants [11]

Refrigerant	$T_{crit}$ , (°C)	$P_{crit}$ , (bar)	Liquid Phase Density $\rho_f$ , (kg/m <sup>3</sup> ) [a]	Gas Phase Density (kg/m <sup>3</sup> ) [b]	Specific Heat Capacity $c_p$ , (kJ/kg) [a]	Volumetric Capacity (kJ/m <sup>3</sup> ) [a]	Thermal Conductivity $k$ , (W/m.K) [a]	Dynamic Viscosity $\mu$ , (mPa.s) [a]
R 11	198	44,1	1536,9	2,36	0,85	450,76	0,09	0,5
R 12	112	42,2	1400,1	17,185	0,93	2636,52	0,62	0,25
R 22	96,2	49,9	1285,7	20,41	1,16	4205,28	0,09	0,22
R 134a	101,1	40,6	1298,9	13,9	1,3	2773,75	0,09	0,27
R 410a	72,13	49,3	1175	28,82	1,5	6566,35	0,1	0,16
R 404a	72	37,3	1154,8	29,91	1,3	4953,99	0,07	0,18
R 407c	86,74	46,2	1240,8	18,86	1,4	3973,24	0,01	0,21
R 507a	70,6	37,05	1161,1	30,98	1,37	5055,32	0,072	0,18
<b>R744 (CO<sub>2</sub>)</b>	<b>31</b>	<b>73,7</b>	<b>934,26</b>	<b>94,148</b>	<b>2,5</b>	<b>22089,00</b>	<b>0,11</b>	<b>0,101</b>
R 717 (NH <sub>3</sub> )	132,3	113,3	640,28	3,31	4,41	4192,51	0,56	0,172

**Açıklamalar:** [a] For saturated liquid at -1,1°C; [b] For saturated vapor at -1,1°C

## 2.4 COST OF THE CO<sub>2</sub> REFRIGERANT

Table 4 shows a comparison of the prices of common refrigerants. It is seen that CO<sub>2</sub> as a natural refrigerant has a much lower cost than conventional refrigerants.

Table 4. Cost Comparison of Widely Known Refrigerants [12]

R134a	R404a	R407c	R410a	R507	<b>R744 (CO<sub>2</sub>)</b>	R717 (NH <sub>3</sub> )
9,38 €/kg	8,85 €/kg	10,68 €/kg	9,58 €/kg	19,53 €/kg	0,52 €/kg	1,3 €/kg

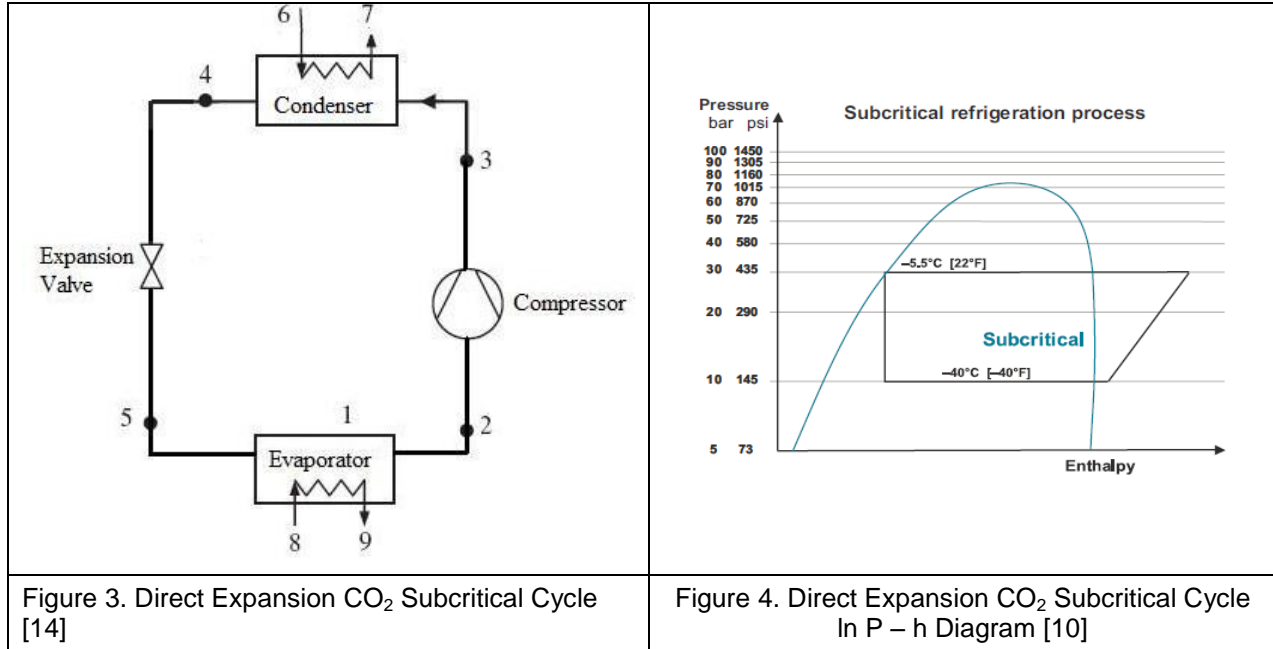
## 3. CO<sub>2</sub> BASIC REFRIGERATION CYCLES

CO<sub>2</sub> refrigeration systems have two types of basic cycles: Subcritical and Transcritical. In the cycle known as Subcritical, the pressure is below the critical point throughout the cycle. In the Transcritical cycle, heat transfer to the environment occurs above the critical point.

### 3.1 SUBCRITICAL CYCLE

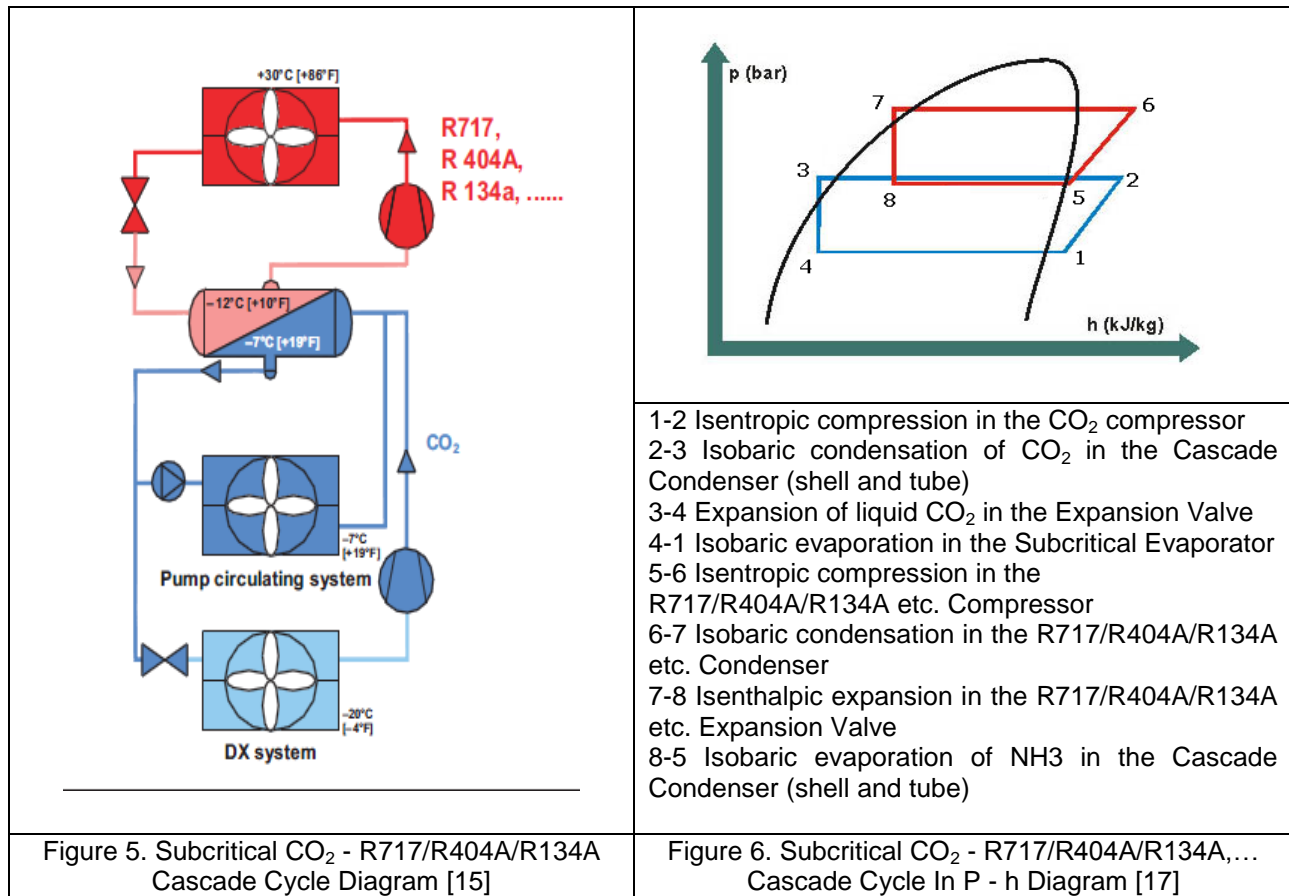
#### 3.1.1 Direct Expansion CO<sub>2</sub> Subcritical Cycle

The subcritical cycle is the most commonly used system in the refrigeration industry. All temperatures and pressures are below the critical point, and above the triple point. The single stage subcritical CO<sub>2</sub> cycle is a fairly simple system. However, it has some disadvantages such as a limited temperature range and high pressure. The low condensation temperature makes heat transfer to the environment more difficult. The operating pressure is around 60 bar [13].



#### 3.1.2 CO<sub>2</sub> Subcritical Cascade Cycle

The CO<sub>2</sub> cascade system is a refrigeration cycle which consists of two separate refrigerants that do not intermingle. One of the two cycles is usually the compression cycle and the cycle where heat transfer to the outside occurs. In these systems, the most widely used refrigerant is R717 (NH<sub>3</sub>, ammonia), which is followed by R404A and other suitable HFC refrigerants [15]. The operating pressure in this type of system is about 40-45 bar [16]



A different feature of cascade systems is that while CO<sub>2</sub> condenses in the heat exchanger known as the cascade condenser, the refrigerant on the high temperature side evaporates. This type of heat exchanger can be plate type or shell and tube type [16]

## 3.2 CO<sub>2</sub> TRANSCRITICAL CYCLE

### 3.2.1 CO<sub>2</sub> Transcritical Cycle

CO<sub>2</sub> has a low critical temperature of 31.06 °C and a high critical pressure of 73.8 bar. As is the case in conventional steam compression cycles, it is not possible for the refrigerant to reject heat to the atmosphere by condensing in the condenser. Heat transfer to the environment in the supercritical area, as the temperature of gaseous CO<sub>2</sub> drops without condensation. Cycles that occur in this way are called "transcritical CO<sub>2</sub> cycles" [6]. Hence, the gas cooler replaces the condenser in the system. In the Transcritical CO<sub>2</sub> cycle, the compressed CO<sub>2</sub> in the compressor rejects its heat to the atmosphere in the gas cooler [18].

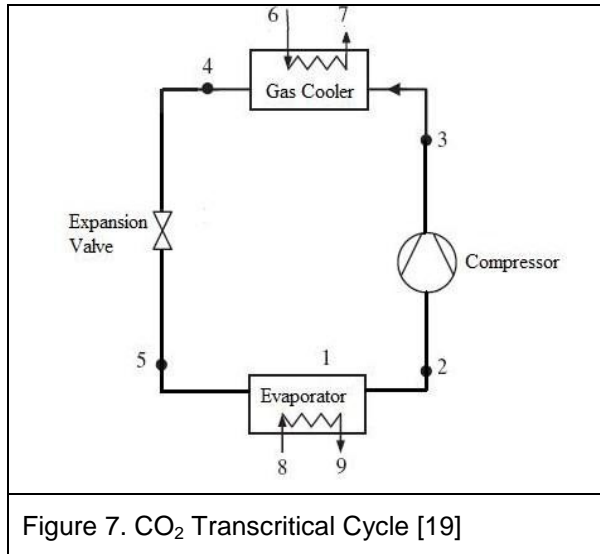


Figure 7. CO<sub>2</sub> Transcritical Cycle [19]

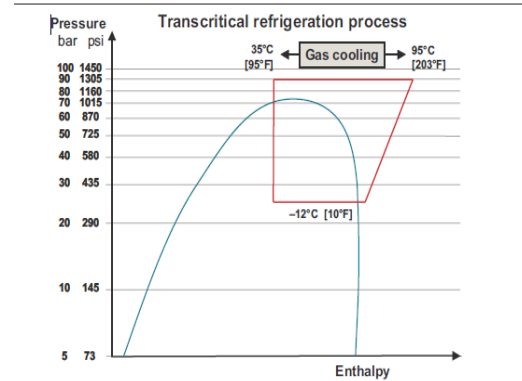


Figure 8. CO<sub>2</sub> Transcritical Cycle In P – h Diagram[10]

### 3.2.1.1 Simple One Stage Transcritical Cycle with Internal Heat Exchanger

An internal heat exchanger is used to increase efficiency in transcritical systems. To increase the EER of the exchanger system, this lowers the gas cooler outlet temperature while serving as an accumulator to prevent liquid leaking to the compressor, ensuring that there is gas at the evaporator outlet.

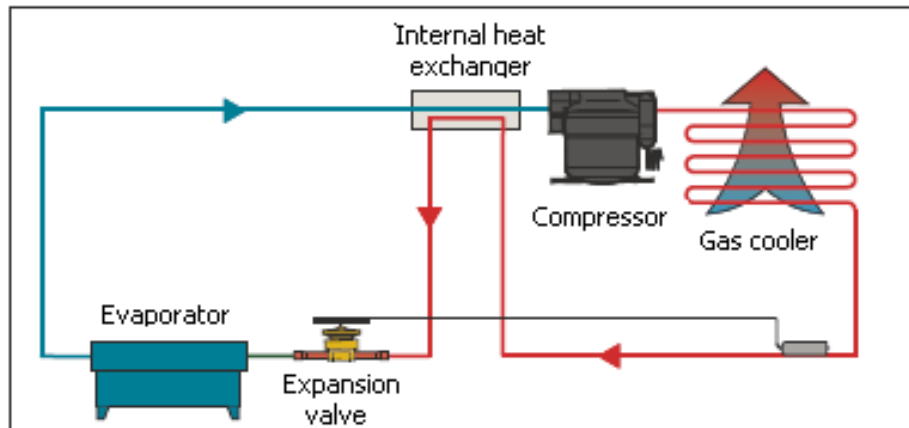


Figure 9. One Stage Transcritical Cycle with Internal Heat Exchanger



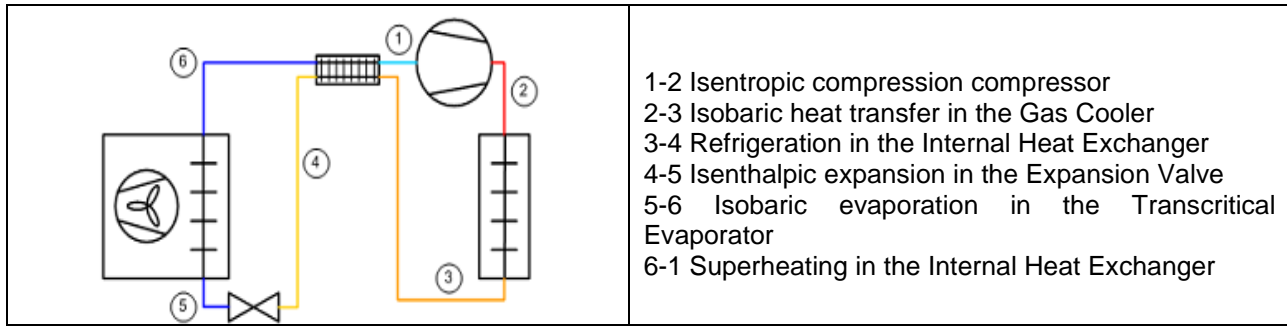


Figure 10. One Stage Transcritical Cycle with Internal Heat Exchanger [20]

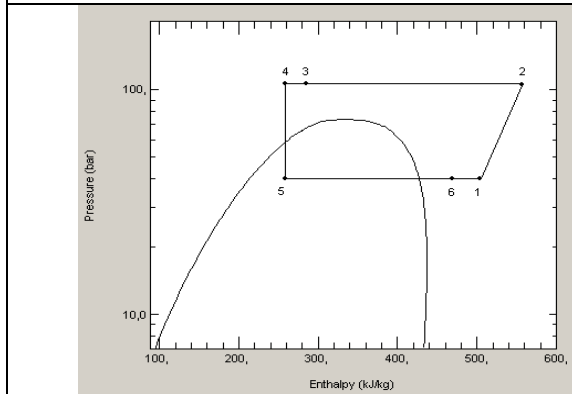


Figure 11. CO<sub>2</sub> Transcritical Cycle with Internal Heat Exchanger In P – h Diagram [21]

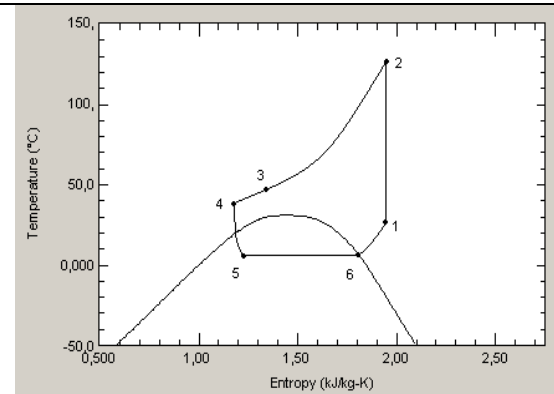
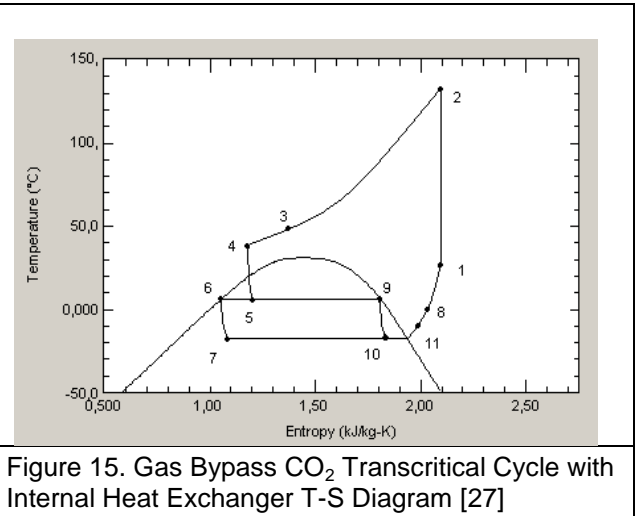
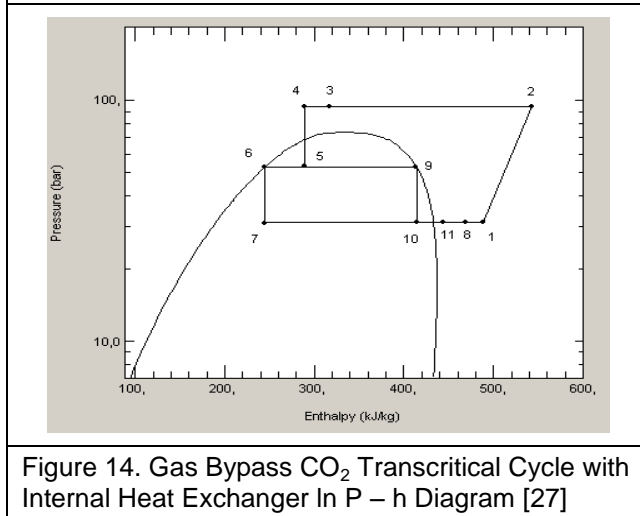
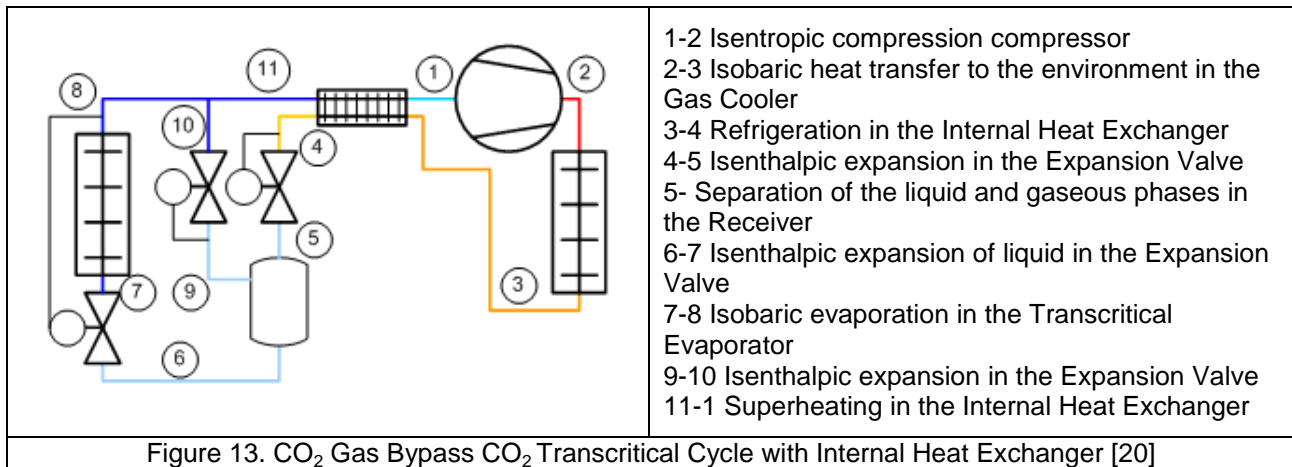


Figure 12. CO<sub>2</sub> Transcritical Cycle with Internal Heat Exchanger T-S Diagram [21]

### 3.2.1.2 Simple Gas Bypass Transcritical Cycle with Internal Heat Exchanger

Gas bypasses are also included in designs to increase the efficiency of transcritical systems. Unlike conventional systems, a second expansion valve is added to the system. While the first expansion occurs from the gas cooler to the liquid tank, the liquid and gas states occur here. Afterwards, the liquid is once again brought to the desired evaporation temperature. In these systems, the dryness at the evaporator inlet is decreased due to the second expansion, thus increasing the EER of the system, while the connection of the system with the outside environment is reduced since a liquid tank is used in the intermediate pressure stage. The system also includes a balancing valve. Since this valve regulates the pressure level in the liquid tank, the temperature before the expansion valve, and therefore the inlet enthalpy of the evaporator are regulated.



#### 4. COMPARISON OF CO<sub>2</sub> REFRIGERANT SYSTEMS WITH OTHER GASEOUS SYSTEMS AND SYSTEM EFFICIENCY

Studies conducted in recent years on CO<sub>2</sub> Transcritical Refrigeration systems indicate that this cycle is more profitable particularly in temperate and cold climates. The reason that stresses this opinion is the fact that the system EER is higher in regions with lower outdoor ambient temperatures. The TESCO supermarket chain in the UK and the FAKTA supermarket chain in Denmark are leaders in Europe in terms of CO<sub>2</sub> Transcritical Refrigeration applications. Even in periods where summers are warm, the energy expended in CO<sub>2</sub> Transcritical refrigeration systems in these countries have been observed to be close to conventional systems. The energy saving comparisons of the R404A single stage system and the CO<sub>2</sub> Transcritical system in major European cities based on average annual temperatures are shown below on the geographic map of Europe. [22]

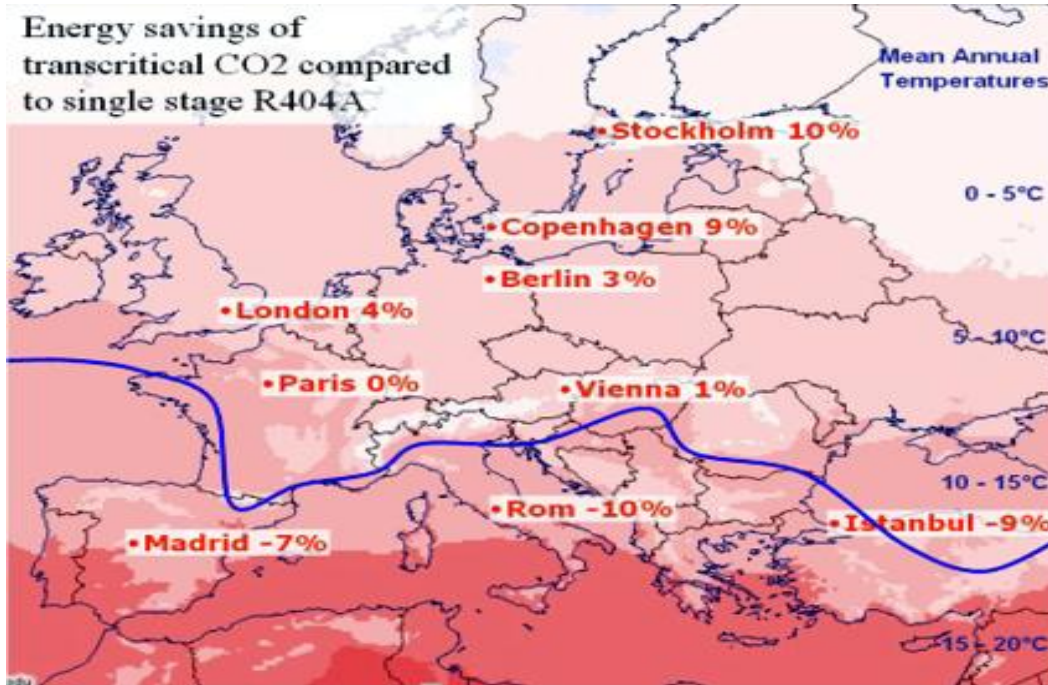


Figure 16. Comparison of the R744 (CO<sub>2</sub>) Transcritical Refrigeration System Efficiency with the R404A gas in the European region [22]

R744 refrigeration systems have been shown to be more efficient in temperate and cold climates with medium and low temperatures. The use of cascade systems is recommended in regions with very warm climates [23, 27].

Efforts to achieve higher efficiency coefficients and increases in efficiency in supermarket systems that combine deep-freezing with normal refrigeration are in progress. Furthermore, it is predicted that efforts to increase efficiencies through evaporative condenser systems in warm climates will intensify in the years to come [27].

## 5. BASIC ELEMENTS OF CO<sub>2</sub> REFRIGERANT REFRIGERATION SYSTEMS

### 5.1 CO<sub>2</sub> COMPRESSORS

Today, the advances in the technology of refrigeration with CO<sub>2</sub> have led many firms to offer compressors to the market. Compressors that are suited for both subcritical and transcritical are available in rotor, hermetic, semi-hermetic, spiral, spiral lobe, and centrifugal types. Transcritical compressors can be selected as single stage or double stage depending on the application. These compressors are also available with economizer equipment [24].

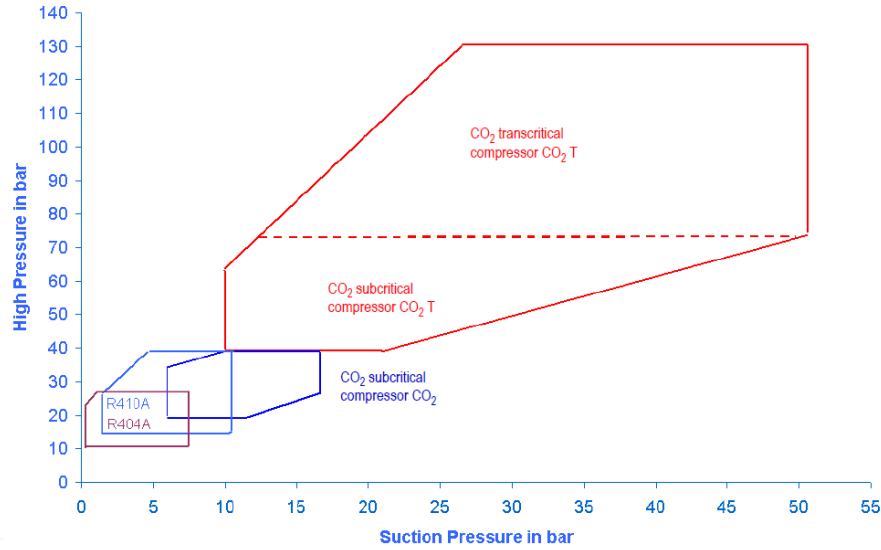


Figure 17. Comparison of operating pressures [25]

The advantages and disadvantages of compressors used in R744 refrigeration systems are as follows:

Table 5. Advantages and Disadvantages of R744 Compressors [24]	
Advantages	Disadvantages
<ul style="list-style-type: none"> <li>• Low compression ratio and high volumetric efficiency</li> <li>• Low piston displacement</li> </ul>	<ul style="list-style-type: none"> <li>• Discharge line temperature increases too much in very low temperature applications.</li> <li>• Substantial reinforcement is required for the outer shell and other components, due to high operating pressures.</li> </ul>

## 5.2 CO<sub>2</sub> COMPATIBLE LUBRICANTS

Two types of synthetic lubricants are used in CO<sub>2</sub> transcritical refrigerant systems: miscible and non-miscible. Poly-alpha-olefin (PAO) is a non-miscible lubricant and generally flows on liquid CO<sub>2</sub>. Polyester (POE) lubricants are used as miscible lubricants, but since these lubricants are highly sensitive to water, they should be stabilized in oil. Detailed information has been provided in the following table [10].

Table 6. Comparison of Lubricants [10]		
Lubricant Type	PAO	POE
Solubility	Low	High
Hydrolysis	Low	High sensitivity to water
Oil Separator	High degree of filtration required (Active carbon and multi stage)	Simple as in HFC/ HCFC systems
Oil Feed	Required for oil drainage from low temperature drip pan	Simple as in HFC/ HCFC systems
Disadvantages	Special requirements in oil separator and feed	Stabilization in oil and sensitivity to water

### 5.3 CO<sub>2</sub> EXPANSION VALVES

The expansion process is definitely used in the gas cooler outlet of the system in order to convey the CO<sub>2</sub> gas to the evaporator at fixed enthalpy. The vanes that implement this process are as follows: [26]

- Capillary Tube or Orifice
- Automatic Expansion Valve
- Thermostatic Expansion Valve
- Electronic Expansion Valve

You can find the advantages and disadvantages of valves in Table 10.

Table 7. Comparison of Expansion Valves [10]				
	Capillary Tube / Orifice	Mechanic Expansion Valve	Thermostatic Expansion Valve	Electronic Expansion Valve
<b>Advantage</b>	Very simple and cheap.	Reaction depends on varying capacity.	Reaction depends on the varying outside temperature.	Provides full control and optimization to the system.
<b>Disadvantage</b>	Operates optimally only under specified conditions.	Has only one set value; does not react sufficiently to variations in outside temperature.	Does not optimize according to varying capacity.	Costly and complex systems.

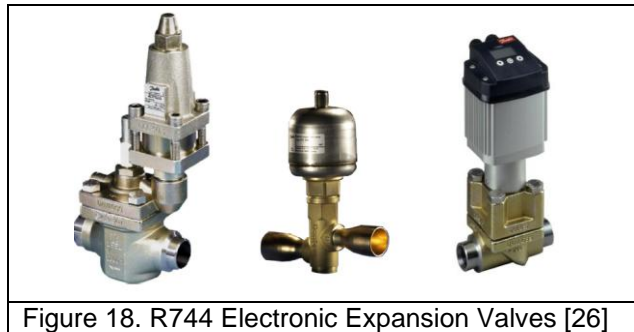


Figure 18. R744 Electronic Expansion Valves [26]

### 5.4 FINNED TUBE CO<sub>2</sub> HEAT EXCHANGERS

Studies were conducted by the Friterm A.Ş. R&D Department within the scope of a scientific study supported by TÜBİTAK, under the main heading of environmentally friendly technologies on designs for evaporators and gas coolers to be used in CO<sub>2</sub> subcritical and transcritical refrigeration systems. Within the scope of this study, software was developed for the design of CO<sub>2</sub> heat exchangers and a test laboratory was set up for the same purpose. During the course of the study, software data were compared to test laboratory data and the compatibility of software results to test results was achieved. The following information involves a general discussion of the data obtained in the course of this study. [27]

### 5.4.1 CO<sub>2</sub> EVAPORATORS

Finned lock CO<sub>2</sub> evaporators can be used in both subcritical and transcritical cycles. The data required for design are discussed in three basic sections:

1. Air side design data
2. CO<sub>2</sub> side design data
3. Heat exchanger design data

The screenshot displays the 'Evaporator Rating' module interface. It is divided into three main sections: Input, Coil Side, and Outputs.

**Input Section:**

- Air Side:** Atmospheric Pressure (101325 Pa), Air Flow (4718.19 m<sup>3</sup>/h), Inlet Temp. (9.96 °C), Humidity (49.23 %).
- Tube Side:** Fluid Name (CO<sub>2</sub>), Refrigerant Flow (100 kg/h), Evaporation Temp. (1.92 °C), Condensing Temp. (10 °C), Super Heating Temp. (5 °C), Sub Cooling Temp. (1 °C).

**Coil Side Section:**

- Geometry:** F 32x28-3/8 C S, Tube (Copper, 0.65 mm), Fin (Aluminium, 0.1 mm), Fin Distance (4 mm).
- Dimensions:** Length (1600 mm), Number of Tubes (20), Number of Rows (6), Number of Circuit (3).
- Manifold In/Out:** 12.7 mm / 12.7 mm.

**Buttons:** Calculate, Save, Print, Draw Diagram.

**Outputs Section:**

- Air Side:** Capacity (10.3 kW), Outlet Air Temp. (3.76 °C), Outlet Rel. Hum. (75.4 %), Air Vel. (1.29 m/s), Air Side Pressure Drop (21.95 Pa).
- Tube Side:** Fluid Flow (171.3222 kg/h), Tube Side Press. Drop. (0.355 bar), Inlet Enthalpy (223 kJ/kg), Outlet Enthalpy (438.8 kJ/kg).
- Coil Dimensions:** Exchange Surface (82.72 m<sup>2</sup>), Din (12.7 mm), Dout (12.7 mm), Dry Weight (51.58 kg), LDmaks (1730 mm), LR (1660 mm), HR (685 mm), LH (635 mm), TH (165 mm), TR (250 mm).

**Diagram:** A schematic diagram of the evaporator coil showing the flow of air and refrigerant, with labels for dimensions and flow directions.

Figure 19. FRTCOILS CO<sub>2</sub> Evaporator Module Interface [27]

#### 5.4.1.1 Air Side Design Data

1. **Atmospheric pressure:** Atmospheric pressure must be known to determine the air pressure dependent physical properties of air. Here the 'Altitude' values can also be defined instead of atmospheric pressure.
2. **Air flow:** The air flow that can accommodate the required capacity must be known. Air speed and mass flow values can also be defined.

3. **Inlet temperature:** The ambient air inlet temperature where the refrigerant will operate must be known.
4. **Relative Humidity:** The relative humidity of the ambient air inlet temperature where the refrigerant will operate must be known. Wet bulb temperature can also be defined instead of relative humidity in this section.

#### 5.4.1.2 CO<sub>2</sub> Side Design Data

1. **Refrigerant Flow:** If the refrigerant flow is known, the outlet conditions of the evaporator can easily be calculated. In this case the superheating degree can be determined according to the flow. If it is not known, the superheating degree must be specified in addition to the other data listed below, that need to be defined.
2. **Evaporation temperature:** The evaporation temperature that will make up for the difference between the ambient air temperature that is specified according to the products and refrigerant usage requirements and the evaporation temperature must be known.
3. **Dryness Fraction:**
  - In the transcritical R744 (CO<sub>2</sub>) Refrigeration Cycle, The dryness fraction of the refrigerant must be known in order to determine the properties of the refrigerant entering the evaporator. This value depends on the operating pressure and outlet temperature of the gas cooler that serves the function of heat rejection in the cycle.

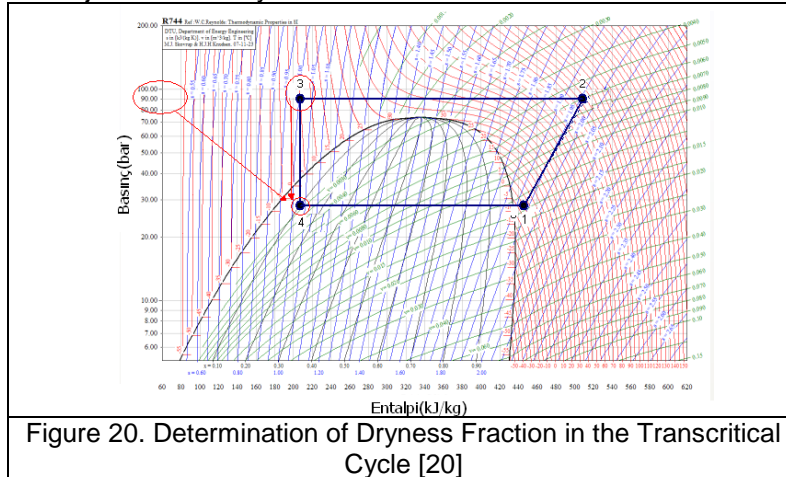
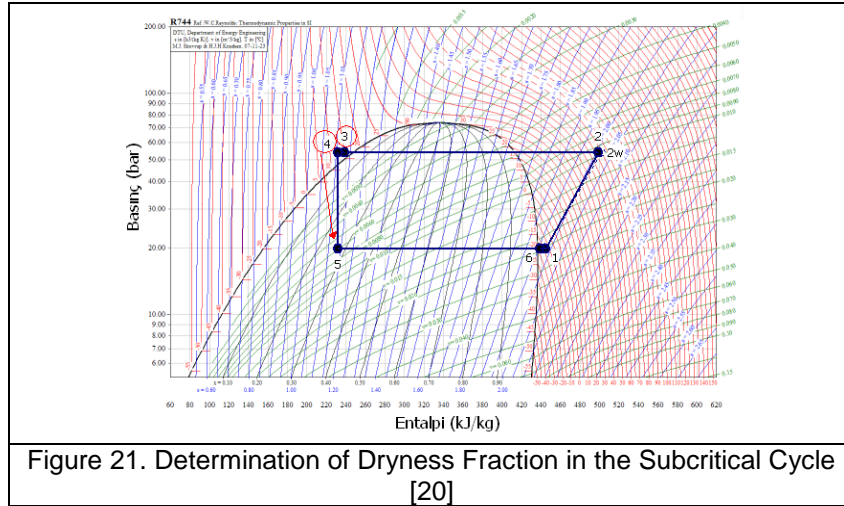


Figure 20. Determination of Dryness Fraction in the Transcritical Cycle [20]

- In the Subcritical R744 (CO<sub>2</sub>) Refrigeration Cycle, the design Condensation temperature of the system and the supercooling degree must be known.





4. **Superheating Degree:** The superheating degree must be specified for the outlet conditions of the evaporator to be known.
5. **Supercooling Degree:** It is the difference between the condensation temperature defined in the subcritical CO<sub>2</sub> refrigeration cycle and the temperature before the expansion valve.

#### 5.4.1.3 Heat Exchanger Design Data

Finned tube heat exchanger design data are given below:

- Coil geometry that determines tube diameter and arrangements as well as distance between tubes and rows
- Fin arranged length
- Tube number
- Row number
- Pass number or circuit number
- Fin pitch
- Fin material
- Fin thickness
- Collector diameter (If known)
- Distributor tubes' diameter and dimension
- Tube material
- Tube thickness

The data defined above under three main headings allow the design and computation of the finned tube CO<sub>2</sub> heat exchanger.

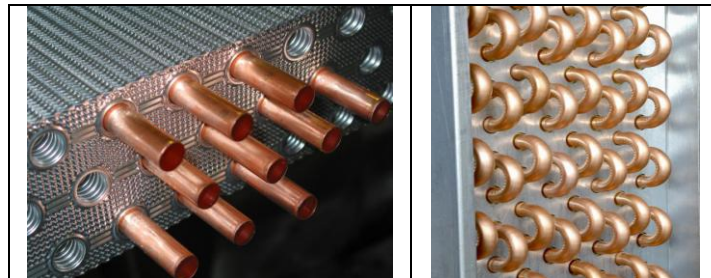




Figure 22. Appearance of copper tubes and curves [27]

There are some important factors that require attention in the design of the CO<sub>2</sub> heat exchanger. For instance, the operating pressures in CO<sub>2</sub> evaporators are higher than operating pressures of evaporators operating in conventional refrigeration systems. For this reason, the thicknesses of copper tubes used must be increased.

Another point is that CO<sub>2</sub> has higher specific heat, higher transfer coefficient and lower viscosity than gases such as R404A, R410A, R134A, etc. This means that the CO<sub>2</sub> gas causes lower pressure loss at the same mass flow for both gases. Another issue that is observed in the design of the CO<sub>2</sub> heat exchanger is that tube diameters can also be made smaller due to high volumetric efficiency.

CO<sub>2</sub> evaporator prototypes designed within this framework and according to data specified above have been tested in the balanced ambient calorimeter laboratory.



Figure 23. CO<sub>2</sub> Evaporator Product Test in the Laboratory [27]



Figure 24. CO<sub>2</sub> refrigerant preparation unit [27]

The conditions specified for testing CO<sub>2</sub> evaporators are presented in Table 11.

Table 8. Evaporator test conditions [27]

Test	Tested Products	CO <sub>2</sub> Evaporation Temperature (°C)	CO <sub>2</sub> Mass Flow (kg/h)	Air Inlet Temperature (°C)	Air Relative Humidity (%)
1	Prototype 1	1,58	193,11	12,02	45,17
2	Prototype 2	1,25	164,068	10	48,79
3		-7,08	122,829	-0,01	50,20
4	Prototype 3	-12,39	76,492	-0,01	41,50
5	Prototype 4	-2,74	110,705	15,03	27,70

Test results obtained for evaporator designs were analysed in the FRTCOILS CO<sub>2</sub> software that had been developed. The experimental results that were obtained can be seen in Table 12.

Table 9. CO<sub>2</sub> Evaporator test results [27]

Test	Tested Products	Experimental Results Capacity (kW)
1	Prototype 1	11,503
2	Prototype 2	9,667
3		7,684
4	Prototype 3	4,665
5	Prototype 4	6,989

Compatibility between the obtained test results and the software that was developed was achieved.

#### 5.4.2 CO<sub>2</sub> GAS COOLERS

Finned tube CO<sub>2</sub> gas coolers are used in the transcritical cycle. The data required for design is grouped under three basic sections as in the evaporators: air side, CO<sub>2</sub> side, and heat exchanger design data.

**CO2 Test**

**Input**

**Air Side**

Atmospheric Pressure: 101325 Pa  
Air Flow: 6860 m3/h  
Inlet Temp.: 25 C  
Humidity: 25.81 %

**Tube Side**

Fluid Name: CO2  
Inlet Temp.: 84.92 C  
Fluid Flow: 227.47 kg/h  
CO2 Pressure: 8500 kPa

**Coil Side**

Geometry: F 25x22-5/16 C S  
Length: 1140 mm  
Tube: Copper 0.9 mm  
Fin: Aluminium 0.1 mm  
Fin Distance: 2.5 mm  
Number of Tubes: 20  
Number of Rows: 3  
Number of Circuit: 2  
Manifold In/Out: 21.3 mm / 21.3 mm

Calculate Save Print Draw Diagram

**Outputs**

Air Side		Coil Dimensions	
Capacity	:: 13.9 kW	Exchange Surface	29.58 m2
Outlet Air Temp.	:: 31.12 C	Din	21.3 mm
Outlet Rel. Hum.	:: 18.1 %	Dout	21.3 mm
Air Vel.	:: 3.29 m/s	Dry Weight	19.87 kg
Air Side Pressure Drop	:: 61.98 Pa	LDmak	1260 mm
		LR	1200 mm
		HR	558 mm
		LH	508 mm
		TH	66 mm
		TR	110 mm

**Tube Side**

Fluid Flow: 227.47 kg/h  
Fluid Outlet Temp.: 28.74 C  
Fluid Speed: 5.662 m/s  
Tube Side Press. Drop: 2.86 bar

**Schematic Diagram**

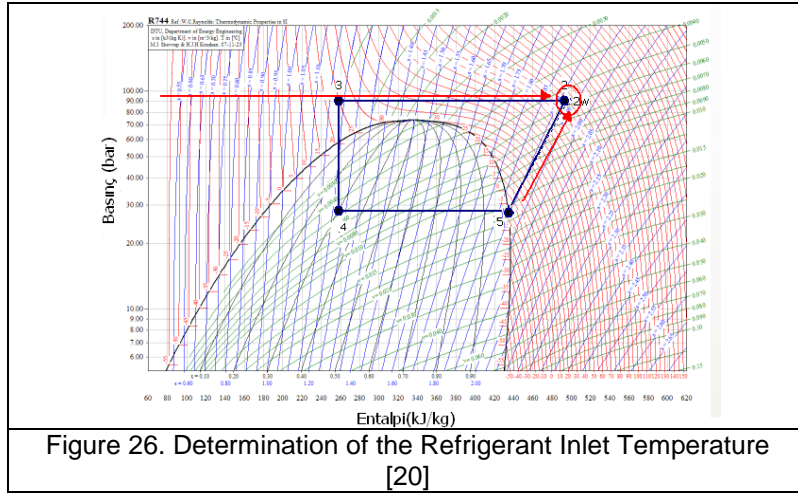
Figure 25. FRTCOILS CO<sub>2</sub> Gas Cooler Module Interface [27]

#### 5.4.2.1 Air Side Design Data

1. **Atmospheric pressure:** Atmospheric pressure must be known to determine the air pressure dependent physical properties of air. Here the 'Altitude' values can also be defined instead of atmospheric pressure.
2. **Air flow:** The air flow that can accommodate the required capacity must be known. Air speed and mass flow values can also be defined.
3. **Inlet temperature:** The ambient air inlet temperature where the gas cooler will operate must be known.
4. **Relative Humidity:** Although the humidity of outside air is not an influential parameter, it must be known.

#### 5.4.2.2 CO<sub>2</sub> Side Design Data

1. **Refrigerant Inlet temperature:** The refrigerant inlet temperature must be specified in relation to operating pressures and compressor efficiency.



2. **Refrigerant Flow:** If the refrigerant flow is known, the outlet conditions of the gas cooler can be calculated. If it is not known, the CO<sub>2</sub> gas cooler outlet temperature must be specified in addition to the other data listed below, that need to be defined.
3. **Refrigerant Outlet Temperature:** In cases where the refrigerant flow is not known, the outlet temperature of the gas cooler must definitely be specified. The desired air inlet temperature in designs is within 3-5 K of the CO<sub>2</sub> outlet temperature.
4. **CO<sub>2</sub> Pressure:** Operating pressure must be specified in relation to ambient temperature and compressor data in gas cooler designs. Furthermore, the EER of the system is influenced not only by the compressor discharge line pressure but also by other parameters such as ambient and gas cooler outlet temperature. Therefore an optimization study should be conducted in the system design phase.

#### 5.4.2.3 Heat Exchanger Design Data

The design data for the finned tube heat exchanger have been presented below.

- Coil geometry that determines tube diameter and arrangements as well as distance between tubes and rows
- Fin arranged length
- Tube number
- Row number
- Pass number or circuit number
- Fin pitch
- Fin material
- Fin thickness
- Collector diameter (If known)
- Distributor tubes' diameter and dimension
- Tube material
- Tube thickness

The data defined above under three main headings allow the design and computation of the finned tube CO<sub>2</sub> heat exchanger.

CO<sub>2</sub> gas cooler prototypes designed within this framework and according to data specified above have been tested in the balanced ambient calorimeter laboratory.



Figure 27. CO<sub>2</sub> Gas Cooler Product [27]

The gas cooler that was designed was tested at the Friterm A.Ş. R&D Test laboratory. Test conditions are presented in Table 13.

Table 10. Gas Cooler Test Conditions [27]					
Test	CO <sub>2</sub> Inlet Pressure (Mpa)	CO <sub>2</sub> Inlet Temperature (°C)	CO <sub>2</sub> Mass Flow (kg/h)	Air Inlet Temperature (°C)	Air Mass Flow (m <sup>3</sup> /h)
1	7,647	64,78	281,879	25,03	6840
2	9,2	84,92	227,47	25	6840

Gas cooler test results were analysed in the FRTCOILS CO<sub>2</sub> software that had been developed. The experimental results that were obtained can be seen below.

Tablo 11. CO <sub>2</sub> Gas Cooler Test Results [27]	
Test	Experimental Results Capacity (kW)
1	13,215
2	14,535

The prototype tests conducted in the laboratory supported software studies. Compatibility between the results obtained from the software that had been developed and experimental data was achieved. Consequently, the software that allows the design of finned tube CO<sub>2</sub> evaporator and gas cooler was created.

## CONCLUSION

For along time, CFC and HCFC refrigerants have been widely used in refrigerant systems that are an absolute necessity in all areas of life, from the past to the future. Since these chemical gases cause a greenhouse effect, they exacerbate the problem of global warming. For this reason, a search has begun for new alternatives in refrigerants. Among natural refrigerants that are in the limelight, CO<sub>2</sub> applications are becoming more widespread. The ozone depletion factor of the R744 gas is zero, and its global warming potential is one. Aside from being very cheap, it is also easy to procure. Its most significant properties are its high volumetric cooling capacity and its thermal properties. This facilitates a reduction in

the intake and discharge lines of the tubing installation as well as in tube diameters of the finned tube heat exchangers that are used. Apart from all of these significant advantages, R744 (CO<sub>2</sub>) has some additional technical requirements to other conventional refrigerants due to its low critical temperature of 31.06°C and high critical pressure of 73.8 bars.

To make sure that the very high critical pressure as compared to other gases does not create problems particularly in terms of safety, the mechanical strength of the system equipment used should be increased. The low critical temperature makes operation of the system's discharge line difficult particularly in countries with temperate and warm climates. This is eliminated by cascade system applications and optimization of gas cooler applications above the critical temperature.

As can be seen from the studies above, the selection of the system that is compatible with design conditions becomes important. Particularly in more temperate and colder climates, subcritical and transcritical systems operate with higher efficiency. In warm climates on the other hand, CO<sub>2</sub> cascade applications are more efficient.

Software development efforts required for the design of evaporators and gas coolers –among the most important components of the system- were conducted. Software studies were supported by prototype tests conducted in the laboratory. The results obtained from the software that had been developed were observed to be compatible with the experimental data. In consequence, the software that allows the design of finned tube CO<sub>2</sub> evaporator and gas cooler was created.

## REFERENCES

- [1] YALÇIN A. Z., Küresel Çevre Politikalarının Küresel Kamusal Mallar Perspektifinden Değerlendirilmesi, Balıkesir Üniversitesi Sosyal Bilimler Enstitüsü Dergisi, Cilt 12, Sayı 21, ss.288-309, Haziran 2009
- [2] BULGURCU, H., KON, O., İLTEN, N., “Soğutucu Akışkanların Çevresel Etkileri İle İlgili Yeni Yasal Düzenlemeler Ve Hedefler”, VIII. Ulusal Tesisat Mühendisliği Kongresi, 2007
- [3] ÖZKOL, N.; “Uygulamalı Soğutma Tekniği”, TMMOB Makine Mühendisleri Odası Yayın No: 115, Nisan 1999, Ankara.
- [4] 2006 Ashrae handbook-refrigeration (SI)
- [5] PEARSON A., Carbon dioxide—new uses for an old refrigerant, 2005
- [6] Thermophysical Properties R744, International Institute of Refrigeration, 2003
- [7] IIR 15th Informatory Note on Refrigerants, February 2000
- [8] Classification of Refrigerants, International Institute of Refrigeration, 2001
- [9] KIM M.H., PETTERSEN, J. ve BULLARD C.W., “Fundamental process and system design issues in CO<sub>2</sub> vapour compression systems”, Progress in Energy and Combustion Science 30, 119–174, 2004
- [10] Danfoss Refrigeration and Air Conditioning Division, Food Retail CO<sub>2</sub> Refrigeration Systems, Danfoss A/S (RA Marketing/MWA), DKRCE.PA.R1.A1.02 / 520H3371, February 2009
- [11] LEMMON, E. W., HUBER, M. L., MCLINDEN, M. O., NIST Standard Reference Database 23, Version 8.0, Physical and Chemical Properties Division, 2007
- [12] DuPont - Cantaş AŞ ve Güneş Gaz Ltd. Şti. Mayıs 2010
- [13] EVANS C. L., BOHN Group-Product Management, CO<sub>2</sub> Unit Coolers for Supermarket Refrigeration Systems, September 2009
- [14] <http://www.cartage.org.lb/en/themes/sciences/physics/thermodynamics/basicthermodynamics/Refrigeration/Refrigeration.htm>
- [15] Danfoss Refrigeration and Air Conditioning Division, CO<sub>2</sub> Refrigerant for Industrial Refrigeration, Danfoss A/S (RA Marketing/MWA), DKRCI.PZ.000.C1.02 / 520H2242, September 2007
- [16] ONBAŞIOĞLU H., An Overview of Vapour Compression System Using Natural Refrigerant CO<sub>2</sub>, İSKİD ACV&R Journal of Turkey, Vol:2, pp: 48-64, 2010
- [17] <http://www.swep.net>



- [18] SARKAR, J., BHATTACHARYYA, S., RAMGOPAL, M., Optimization of a Transcritical CO<sub>2</sub> Heat Pump Cycle for Simultaneous Cooling and Heating Applications, International Journal of Refrigeration, 27, 2004.
- [19] BAYRAKÇI, H.C., ÖZGÜR, A.E., AKDAĞ, A.E., “Aynı soğutma Yüğü için CO<sub>2</sub>’li ısı pompalarının enerji sarfiyatlarının karşılaştırılması”, IX.Ulusal Tesisat Mühendisliği Kongresi
- [20] <http://simple-one-stage-co2-cycle.software.informer.com>
- [21] FARTAJ, A., TING, D.S.K., YANG, W.W., “Second law analysis of the transcritical CO<sub>2</sub> refrigeration cycle”, Energy Conversion and Management Vol:45, 2004
- [22] Danfoss Teknik Dökümanları
- [23] CECCHINATO, L., CORRADI, M., “Transcritical carbon dioxide small commercial cooling applications analysis”, International Journal of Refrigeration, Vol: 34, 2011.
- [24] ZIMMERMANN, A., “Carbon Dioxide as the Refrigerant in Supermarket, CO<sub>2</sub> Compressors”, 10th Short Course in Supermarket Refrigeration, 2010
- [25] Bock Kompresör Teknik Dökümanları
- [26] The Danfoss CO<sub>2</sub> Gas Cooler Solution
- [27] Friterm A.Ş. Teknik Dökümanları

#### YAZAR ÖZGEÇMİŞİ:

Fatih KASAP was born in Karabük in 1986. He graduated from the Mechanical Engineering Department of the Istanbul Technical University in 2009. He is currently pursuing his postgraduate studies in the department of Thermal Fluids of the same University. He completed his Undergraduate and Post Graduate theses within the Friterm A.Ş. R&D department. He is currently employed as a Test Laboratory and Product Development Engineer at the R&D department of Friterm A.Ş..

#### YAZAR ÖZGEÇMİŞİ:

Hasan ACÜL was born in Ayvalık in 1976. He graduated from the Mechanical Engineering Department of the Yıldız Technical University in 1999. He received his master's degree at the Department of Science and Technology Strategies of the Gebze Institute of Technology. He has worked as an engineer, coordinator and manager in construction sites, sales, production and R&D departments of various firms operating in Heating, Cooling and Air Conditioning (HVAC) industry. He is currently R&D Department Manager at Friterm A.Ş., where he has been working since 2003. He is also coordinator and trainer of Friterm Corporate Academy.

Hasan Acül has The New Product Development Professional (NPDP) certification, developed and offered by the Product Development & Management Association (PDMA). He has also Tübitak/Tüside “R&D Professional” Certificate.

Hasan Acül is a member of the executive board of the Kartal District of Istanbul Chamber of Mechanical Engineers. Besides he is a member of the International Product Development and Management Association (PDMA), the Istanbul project Management Association, and the Association of Ship Modellers. He is married, and has a daughter.

#### YAZAR ÖZGEÇMİŞİ:

Hatice CANBAZ was born in Kocaeli in 1985. She graduated from the Mechanical Engineering Department of the Yıldız Technical University in 2008. She is currently pursuing her postgraduate studies in the department of Thermal Process of the same University. She has worked as a R&D engineer at Friterm A.Ş. where she began working after receiving her graduate degree for nearly two years, and has the Tübitak/Tüside “Product Development Specialist” Certificate.

**YAZAR ÖZGEÇMİŞ:**

Selim ERBİL was born in Ankara in 1986. He graduated from the Mechanical Engineering Department of the Yıldız Technical University in 2010. He is pursuing his postgraduate studies at the Istanbul Technical University Energy Institute. He completed his Undergraduate dissertation within the Friterm A.Ş. R&D department. He is currently employed by a firm operating in the automotive industry, as a product development and design engineer.