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Determination of Cooling Effect of R12 and R404A

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Abstract: Purposes of this research as to determine cooling effect of Freon 12 (R12) and R404 A in a cold store built specifically for this research. Inlet and outlet temperature of the evaporator and condenser, compressor suction and compression pressures were measured and cooling coefficients were calculated. Required flow rate of R12 quite larger than that of R404 A because of its less evaporative specific heat. High flow rate increases compression work and consequently compressor will use more energy. Used energy per unit cooling of R12 was higher than R404 A. Capacity of condenser for R12 refrigerant was too much in spite of small cooling load. Evaporator pressure of R404 A was higher than R12. This caused to increase the vapour density and to establish more cooling effect. Suction and compression pressures of the R404 A were higher than Freon 12. High suction and compression pressure decrease compressor life. Cooling effect of system showed superiority of R404 A.

Key words: Cooling system, R12, R404A, performance of the cooling system

Introduction

Not only the production but also the protection of agricultural crops has vital importance for human life. Agricultural products that don't have resistance for decaying and spoiling have to be marketed as quickly as possible otherwise have to be disposed. Most important factor for marketing agricultural products at good price is the protection of its taste and other specifications. It is only possible by storing. The cold store is one of the widely used storage system for agricultural products. Storage of an agricultural product depends on the cooling system and package box.

Most storage facilities use mechanical refrigeration to control storage temperature. The cooling system consists of a condenser, compressor, evaporator, refrigerant, expansion valve and lines. Environmental effect of refrigerants protected using of some refrigerants and caused to product new refrigerants.

The optimization of a jet cooling system using refrigerant mixtures as substitutes of pure refrigerants has been investigated. A steady-state simulation program, for given temperatures of the sources, integrating simple models of each component has been developed. A Peng-Robinson equation of state assuming equality of the fugacities of the two phases was used to model the thermodynamic properties of the vapour and liquid-vapour equilibrium. The refrigerants investigated in this study are: the pure refrigerants R142b, R152a, RC318, R124, R134a, R22 and the binary refrigerants R22/RC318, R22/R142b, R22/R124, R22/R152a, R22/R134a, R134a/R142b, R152a/R142b and R134a/R152a. Results show that the use of a binary mixture does not always increase the performance of system. Generally, when the mixture is strongltropic (e.g.: R22/RC318), the cooling efficiency of the system decreases. However, when the mixture is mildly zeotropic (e.g. R134a/R142b) or almost esotropic (e.g. R134a/R152a), efficiency and energetic efficiency increase (Boumaraf and Lallemand,1999).

For the cold storage of agricultural products at temperatures of 2-4°C in India, a solar-hybrid cooling system has been developed, using solar energy from flat plate collectors and the waste heat of a genset, operated with producer gas. A commercially available low temperature (80-90°C) adsorption cooling system for air-conditioning application had to be modified for operation at cooling temperatures below the freezing point of water. Methanol instead of water was investigated as a refrigerant. Because of the inferior thermodynamic properties of methanol and the lower operation temperature, the efficiency is reduced. Calculations showed that the COP for a commercial adsorption cooling system is about 30% when operating the system with methanol/silicagel at a chilled water temperature of T sub 0 equals minus 2°C, a heating water temperature of T sub h equals 85°C and a condenser temperature of T sub c equals 30°C. A laboratory test facility with heating and cooling capacities of about 2 kW was set up. By comparative experiments with water and methanol, the calculations could be validated, though the specific power density is only 1/10 of the commercial system (Oertel and Fischer, 1998).

The rapid technology development occurring in commercial vapor compression cycles reflects the growing concern over the greenhouse effect and ozone depletion issues relating to energy conversion and the chemical substances used in them. The efficiency potential of the vapor cycle is receiving collateral interest in the aircraft community where its potential to significantly improve cooling efficiency over the currently used reverse Brayton air cycles will have significant benefits in payload/range and operating efficiencies as heat loads and fuel prices increase. This paper describes the analytical design optimization of a new microprocessor-based, variable-speed, vapor compression cooling system which uses a binary nonazeotropic refrigerant mixture to provide a significant efficiency improvement to aircraft environmental control. Areas discussed include the thermodynamic and thermophysical modeling of the working fluid, cycle synthesis and the performance-physical characteristic optimization process. Design tools developed for this effort are readily adaptable to heating/cooling systems for domestic and industrial application - space, ground-based, or other. Although the system design was based on fully halogenated chlorofluorocarbon substances, subsequent work to identify blends and to quantify benefits of new working fluids that have acceptably low ozone depletion potential are reported upon. The paper also describes the status of system development scheduled to conclude with a flight test (Desai ***et al.***, 1991).

In this research; cooling effects of the R12 and R404A were investigated in cooling system and cold store. A cold store was designed and built for this aim. Then Freon12 and R404 A refrigerants were loaded consecutively. Observations and measurements were done to realise the objective.

Materials and Methods

Cold store

Dimensions of cold storage were 4.52 m in length × 1.90 m in width × 2.22 m in height (Fig. 1). The volume of the cold store was 19.07 m^3.

The cold store was made of reinforced concrete and isolated by using Foamglass. Foamglass was placed on the wall using an adhesive material produced as a side of asphalt. Then a coverage material formed steel rush was placed on the wall surface to cover the foamglass. The steel rush material was covered with plaster. Finally ceramics were used to cover the wall. Foamglass was put on the ceiling of the cold store and then both surfaces were plastered with cement. The floor of the cold store was covered with 3 Foamglass and ceramics.

Cooling system

A compressor was located in the corridor (Fig. 1.). Natural air was used to cool the cold store. A condenser, a condenser fan, a fluid hopper and a drier were gathered as a group inside the compressor chassis. Evaporator was put on the ceiling of the cold storage with steel screws. Thermostatic expansion valve was fitted in the evaporator inlet and on the high pressure side of the compressor. A control panel which includes a thermometer-thermostat, a manometer and an humidity gauge were placed on the front wall of the cold store.

The compressor was a hermetic type and its capacity was 10460 kJ h^{-1}. An electrical motor of 1.49 kW powered the compressor.

The condenser was cooled by air. It consists of copper pipes, aluminum plates and an axial type fan. Its capacity was 12552 kJ h^{-1}.

The evaporator was made of copper pipes and outside surface covered with aluminum plates. An axial fan was located on the back of the evaporator to distribute the cooled air into the cold store. An electrical heater was used to defrost. The capacity of evaporator was 10460 kJ h^{-1}.

Refrigerant R404A

Chloro-floro-carbons (CFC) have been used approximately for 60 years because of its good cooling specifications. Poisonous value of CFC is low, flame resistance is high and anticorrosive specifications. It is used in many fields.

The composition of the R404A refrigerant as a percentage of weight is 44% HFC-125, 52% HFC 143a and 4% HFC 134a.

R404A refrigerant classified as a high-pressure refrigerant can be used in commercial applications of medium and low temperature cooling systems instead of R502. Cooling performance of the R404A refrigerant is too close to that of the R502. R404A can work at low compressor outlet temperature and this specification extends compressor life. R404A can be used in some systems designed for R502 and R22 refrigerants but that cooling system has to be changed for R404A. In this case, compressor oil should be changed three times using polyol-ester based (Dagsoz, 1981).

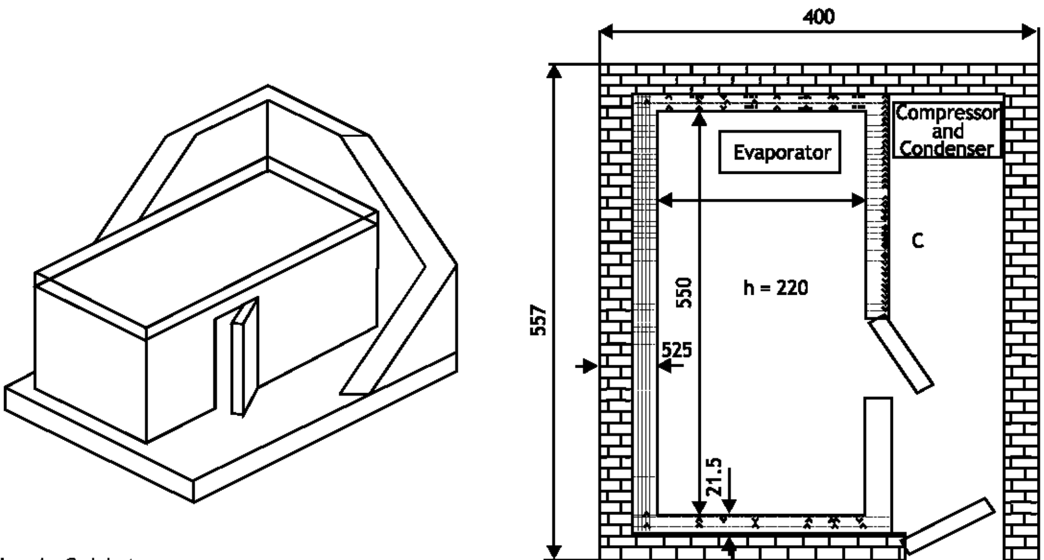

Fig. 1: Cold store

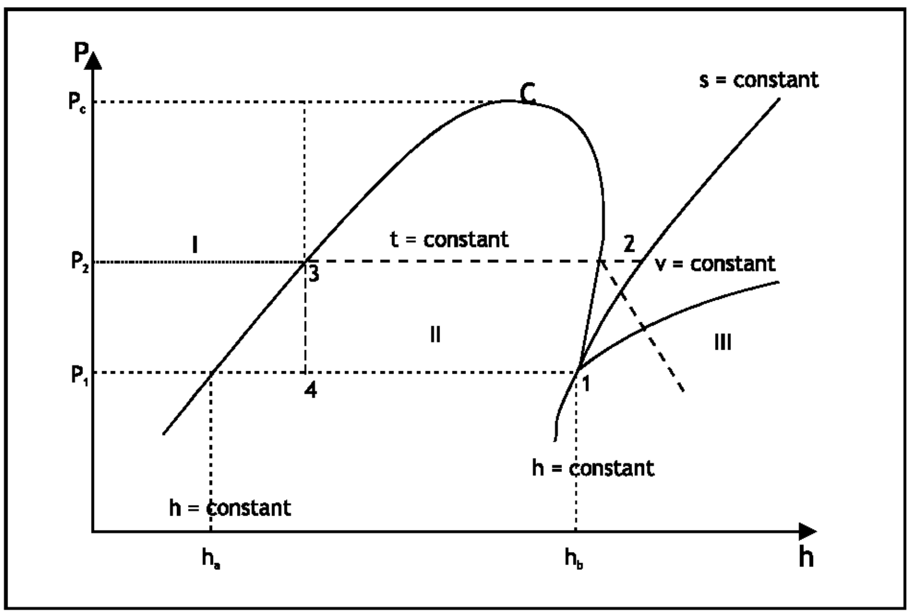

Fig. 2: Pressure-heat enthalpy diagram (Dagsoz, 1981)

Refrigerant R12 (Freon-12-CCl_2F_2)

R-12 is a very popular refrigerant. It is a colourless, almost odourless liquid with a boiling point of -29°C at atmospheric pressure. It is non-toxic, non-corrosive, non-irritating and non-flammable.

R-12 has a relatively low latent heat value. In the smaller refrigerating machines, this is an advantage. The large amount of refrigerant circulated will permit this use of less sensitive and more positive operating and regulating mechanisms. It is used in reciprocating, rotary and large centrifugal compressors. It operates at a low but positive head and back pressure and with a good volumetric efficiency.

R-12 ha a pressure 183 kPa at -15°C and a pressure of 745 kPa at 30°C. The latent heat of R-12 at -15°C is 159 J g^{-1} (Althouse ***et al.***, 1992).

Refrigerants which include Chlorine (Cl) (R12 and R22) were abandoned in last decade because of negative effect on Ozone layer.

Heat transfer and condensation specifications of R12 are good in the condenser. R12 is a solvent for oil and this specification prevents reducing heat transfer by accumulating oil between heat transition surfaces in the condenser and evaporator. Required flow rate is high because vaporisation heat of R12 is low. But this is not an important disadvantage for small cooling systems; on the contrary, this property is preferred to make easy to control of flow rate. Required cylinder volume per cooling cycle is not too bigger than R22, R500 and R-717 in the big cooling systems because vapour density of R12 is too high. Using the R12 has to be abandoned according to the International agreement due to determined calendar because R12 is one of the refrigerants that affect ozone layer adversely (Ozkol, 1999).

Determination of the cooling load (heat balance)

Total amount of heat in a cold store includes transition heat (q_1), infiltration heat (q_2), product heat (q_3), heat created by other sources such as workers, illumination, airing (q_4) and unknown and unexpected heat (q_5) (Ozkol, 1999). The cooling load was calculated for apple, cherry, sour cherry and grape because these fruits are widely growing in the Thrace Region. Equation 1 was used to calculate the cooling load for each fruit (Ozkol, 1999)

$$QK = (24/20)^* \, (q_1+q_2+q_3+q_4+q_5) \quad (1)$$

QK = Cooling load (kJ day^{-1})

24/20 = Time factor

Calculated maximum cooling load was used to determine capacity of cooling system elements.

Calculation of selection criteria of the cooling system components

Schematic diagram of theoretical cooling cycle is given in Fig. 2. Evaporator specific heat load, evaporator vaporisation capacity, amount of refrigerants, condenser specific heat load, condenser condensation capacity, compression heat of compressor, effective power and theoretical power were calculated by using pressure-heat enthalpy diagram (Dagsoz, 1981; Savas, 1987).

The capacity of compressor should be equal to the removed heat from cold store for a unit time. Compressor capacity should be determined according to maintenance, repairing and idle time of the compressor for a day. Daily total working hours mean working period of the compressor. This period depends on cold storage temperature, evaporator input and output temperatures and daily defrost period. Daily total working hours of the compressor period change between 14 and 22 h. The compressor working period was selected as 20 h in this research because different type agricultural products could be stored at 1-3°C (Genceli, 1987).

Compression work, theoretical compression heat, theoretical and practical power of the compressor were calculated by Equations 2, 3, 4 and 5 (Ozkol, 1999).

$$L = (h_2 - h_1) \tag{2}$$

L = Compression work (kJ h^{-1})
h_1 = Heat enthalpy value for point 1 in Fig. 2. (kJ kg^{-1})
h_2 = Heat enthalpy value for point 2 in Fig. 2. (kJ kg^{-1})
Amount of refrigerant per hour was calculated as follows;

$$G = \frac{QK}{h_1 - h_3} \tag{3}$$

QK = Cooling load of the system (kJ h^{-1})
h_3 = Heat enthalpy value for point 3 in Fig. 2. (kJ kg^{-1})
G = Flow rate of refrigerant (kg h^{-1})

$$q_k = G.L \tag{4}$$

q_k = Theoretical compression heat (kJ h^{-1})
G = Amount of refrigerants (kg h^{-1})

$$Wt = \frac{q_k}{860} \tag{5}$$

Wt = Theoretical power of compressor (kW)
860 = Converting coefficient

$$Wp = Wt. \frac{1}{\eta_i} . \frac{1}{\eta_m} \quad (6)$$

W_p = Practical power of compressor (kW)

η_i = Indicator efficiency (85%)

η_m = Mechanical efficiency (85%)

The required electrical motor power was calculated by using Equations 6 for triangle-star connection.

$$W_m = 1.3\ W_p \quad (7)$$

Wm = Electrical motor power (kW)

Evaporator specific heat load was calculated by using Equation 8 and determined heat enthalpy values from Mollier Diagram (pressure-heat diagram) due to vaporisation and condensation temperatures of R12 and R 404A (Ersoydan, 1967; Genceli, 1997)

$$Qb = h_1 - h_3 \quad (8)$$

q_b = Evaporator specific heat load (kJ kg^{-1})

q_b should be positive because of $h_1 > h_3$. The cooling system gain heat between point 1 and 3 in Fig. 2.

Evaporator capacity should be equal to the total heat load removed. This cooling load is taken by evaporator and given to the refrigerants. Vaporisation temperature should be below 10°C and 15°C according to the cold storage temperature. Vaporisation temperature was taken as -10°C because The Thrace Region was located mild climate region in the World and storage temperature was chosen as 3°C.

$$Q_b = QK = G.\ q_b \quad (9)$$

Q_b = Evaporator capacity (kJ h^{-1})

Condenser specific heat load was calculated as follows for R12 and R404 A by using Mollier Diagram.

$$q_y = h_3 - h_2 \quad (10)$$

q_y = Condenser specific heat load (kJ kg^{-1})

q_y should be negative because of $h_2>h_3$. Cooling system looses heat between point 2 and point 3 in Fig. 2.

$$Q_y=G.q_y \tag{11}$$

Q_y = Condenser capacity (kJ h^{-1})

The capacity of condenser is chosen as equal to the amount of heat losses per hour and extra 15% of this value.

$$Q_y=1.15.QK \tag{12}$$

Main purpose of the cooling systems is to remove heat as possible as big from the cold source. Cooling efficiency (ε) was calculated by using Equation 13 (Erol, 1993).

$$\varepsilon=\frac{h_1-h_4}{h_2-h_1} \tag{13}$$

h_4= Heat enthalpy value for point 4 (kJ kg^{-1})

The capacities of the cooling system components were determined considering the highest total cooling load.

Measurement of temperatures and pressures for cooling system

Evaporator inlet and outlet temperatures, condenser inlet and outlet temperatures and compressor suction and discharge pressures were measured for R12 and R404A refrigerants. Measurements were done at every 1°C between +11°C and -3°C. Pressures were used to determine enthalpy values by using pressure-heat enthalpy diagram for each refrigerant (Anonymus, 2001; Genceli, 1997).

Measurement limits of low-pressure manometer were varied between -1 and 8.3 kg/cm^2 (being equal to 100 kPa and 830 kPa, respectively). Measurement limits of high-pressure manometer were varied between 0 and 34 kg/cm^2 (being equal to 0 kPa and 3400 kPa, respectively). Accuracy of the low and high-pressure manometer was ±1%.

Measurement limits of the temperature sensors ranged between -20°C and +100°C. Accuracy of the temperature sensors was ±1%.

Coefficient of performance was calculated as follows for each temperature degree for R12 and R404A refrigerant (Ozkol, 1999). The experiments were carried out for both refrigerants, respectively. Outside temperature and relative humidity values were also given in Table 3 and Table 4.

$$COP = \frac{T_2}{T_1 - T_2} \qquad (14)$$

COP = Coefficient of performance

T_1 = Condensation temperature (°C)

T_2 = Vaporisation temperature (°C)

Results and Discussions

Cooling load and selection criteria of compressor, condenser and evaporator

Results of the transition heat caused by heat lost through the wall, the floor and the ceiling of the cold store (q_1), infiltration heat caused by entering air whose heat enthalpy value was higher than inside air (q_2), heat created by agricultural products (q_3), heat created by other heat sources as human, lightning, motor etc.(q_4), heat created by unknown and unexpected sources (q_5) and total cooling load (QK) are given in Table 1.

Total cooling loads were calculated as 9652.849 kJ h^{-1} for apple, 9590.007 kJ h^{-1} for cherry, 10392.206kJ h^{-1} for sour cherry and 7755.682 kJ h^{-1} for grape. Cooling system components were selected considering the highest total cooling load, which is 10392.206kJ h^{-1} for sour cherry.

Amount of the refrigerant, theoretical compression work and heat, theoretical and effective compressor power, evaporator specific heat, evaporator capacity, condenser specific heat, condenser capacity and cooling efficiency are given in Table

The cold store was designed to work with R12 and R404A. Consequently, the cooling system elements were selected for working with both refrigerants.

According to the results; the highest amount of refrigerant was 89.34 kg h^{-1}, the effective compression power was 0.748 kW, the evaporator capacity was 10392.2 kJ h^{-1} and the condenser capacity was 12336.04 kJ h^{-1}. The cooling efficiency (ϵ) was calculated for R12 and R404 A as 5.34 and 6.30, respectively.

Table 1: Energy balance for cooled fruits

	Apple		Sour cherry		Cherry		Grape	
	KJ day^{-1}	KJh^{-1}	KJ day^{-1}	KJ h^{-1}	KJ day^{-1}	KJ h^{-1}	KJ day^{-1}	KJ h^{-1}
q_1	49216.157	2050.673	53183.344	2215.972	53183.344	2215.972	53183.344	2215.972
q_2	735.433	30.643	781.398	32.558	781.398	32.558	781.398	32.558
q_3	111047.744	4626.989	105933.859	4413.910	120519.283	5021.636	72582.492	3024.270
q_4	14507.015	604.458	14465.175	602.715	14465.175	602.715	14465.175	602.715
q_{top}	175506.352	7312.764	174363.777	7265.157	188949.201	7872.883	141012.410	5875.517
q_5	17550.635	731.276	17436.377	726.515	18894.920	787.288	14101.241	587.551
Q_T	193056.987	8044.041	191800.155	7991.673	207844.122	8660.171	155113.652	6463.068
QK		9652.849		9590.007		10392.206		7755.682

Table 2: Selection criteria for cooling system

Fruit	Refrigerant	Compressor amount of refrigerant (kg h^{-1})	Evaporator theoretical compression work (kJ kg^{-1})	Condenser theoretical compression heat (kJ kg^{-1})	Theoretical compression power (kW)	Effective power (kW)	Specific heat (kJ kg^{-1})	Capacity (kW)	Specific heat (kJ kg^{-1})	Capacity (kW)	Colling efficiency (ε)
Apple	R12	82.988	21.756	1805.568	0.501	0.693	116.315	9652.848	138.07	11458.32	5.346
	R404A	76.596	19.999	1506.253	0.418	0.578	126.022	9652.848	146.02	11184.67	6.300
Kiraz	R12	82.448	21.756	1793.811	0.498	0.689	116.315	9590.004	138.07	11383.76	5.346
	R404A	76.097	19.999	1521.917	0.422	0.584	126.022	9590.004	146.02	11111.80	6.300
Vişne	R12	89.345	21.756	1943.861	0.540	0.748	116.315	10392.200	138.07	12336.04	5.346
	R404A	82.463	19.999	1649.224	0.458	0.634	126.022	10392.200	146.02	12041.38	6.300
Üzüm	R12	66.678	21.756	1450.702	0.501	0.693	116.315	7755.680	138.07	9206.36	5.346
	R404A	61.542	19.999	1230.811	0.342	0.472	126.022	7755.680	146.02	8986.45	6.300

Table 3: Measured temperature and pressure values for R12 refrigerant

	Temperature of cold storage (°C)														
Values	11	10	9	8	7	6	5	4	3	2	1	0	-1	-2	-3
Evaporator input temperature (°C)	0.3	-0.7	-2.7	-3.9	-5.0	-6.9	-6.0	-6.9	-7.0	-9.8	-11.5	-12.1	-14.5	-15.7	-18
Evaporator output temperature (°C)	9.1	8.4	7.4	6.5	5.7	4.5	3.4	2.5	1.5	0.7	-0.1	-0.5	-2.2	-3	-3.7
Condenser input temperature (°C)	42.2	44.7	43.2	43.2	43.1	40.5	37.2	35.4	37.6	37.7	37.2	37.5	32.4	31.8	29.1
Condenser output temperature (°C)	22.5	21.9	21.4	21.2	20.6	20.3	14.6	14.5	15.2	15.2	15.2	16.0	12.4	12.5	10.4
Compressor suction pressure (kPa)	125.0	120.0	120.0	112.0	110.0	100.0	100.0	85.0	80.0	80.0	70.0	70.0	60.0	60.0	58.0
Compressor discharge pressure (kPa)	800.0	795.0	790.0	79.0	760.0	790.0	780.0	725.0	710.0	710.0	700.0	730.0	720.0	730.0	720.0
COP	12.3	12.0	11.2	10.7	10.5	9.8	13.0	12.4	12.0	10.5	9.8	9.3	9.6	9.1	9.2

Table 4: Measured temperature and pressure values for R404A refrigerant

	Temperature of cold storage (°C)														
Values	11	10	9	8	7	6	5	4	3	2	1	0	-1	-2	-3
Evaporator input temperature (°C)	-3.7	-5	-7	-8.8	-10.6	-10.9	-11.6	-13.1	-13.3	-14	-14.7	-15.8	-14.3	-15.1	-15.8
Evaporator output temperature (°C)	9	6.9	5.7	4.7	3.2	2.4	1.6	0.4	-0.5	-1.4	-2.5	-3.5	-4.3	-5.5	-6.2
Condenser input temperature (°C)	13	12.9	18.7	20.6	18.4	19.5	31.8	43.2	42.8	41.4	39.7	39.9	39.5	42.4	43.8
Condenser output temperature (°C)	14.9	16.6	13.9	13.2	13.1	13.6	14.7	15.3	14.7	14.7	13.8	12.9	14	14.8	15.2
Compressor suction pressure (kPa)	200.0	140.0	170.0	150.0	170.0	160.0	150.0	140.0	160.0	170.0	170.0	150.0	170.0	150.0	150.0
Compressor compr. pressure (kPa)	1100.0	1060.0	1080.0	1070.0	1140.0	1100.0	1040.0	860.0	1071.0	1100.0	1100.0	1100.0	1070.0	1050.0	1050.0
COP	14.5	12.4	12.7	12.0	11.1	10.7	9.9	9.2	9.3	9.0	9.1	9.0	9.1	8.6	8.3

Temperatures and pressures results

The evaporator inlet and outlet temperatures, the condenser inlet and outlet temperatures, the compressor suction and compression pressures are given in Table 3 and Table 4 for R12 and R404A refrigerants, respectively.

The inlet and outlet temperatures of R404A refrigerant in the condenser and evaporator were generally higher than that of refrigerant R12.

Suction and compression pressures of the R12 were lower than the R404A refrigerant. Reason of this result is low vaporisation heat for R12 (for example 145.9 kJ kg^{-1} for +11°C) according to the R404A for example 160.4 kJ kg^{-1} for +11°C.

According to the COP results, it changed at a large scale for each refrigerants. COP of R12 was better than R404A at cold storage temperatures (+4......1°C) for 15.5°C outside temperature.

In Thrace Region, apple, cherry, sour cherry and grape are widely produced. A cold store was designed to store these fruits for the maximum cooling load calculated for sour cherry storage as 10392.206 kJ h^{-1}. The cooling system elements, the compressor, the evaporator and the condenser were selected considering the maximum cooling load.

According to the results; required flow rate of refrigerant R12 (89.34 kg h^{-1}) was greater than that of R404A because the evaporation specific heat of R12 was lower than that of R404A. High flow rate of refrigerant increases compression work and consequently compressor uses more energy.

Required compressor power for R12 refrigerant (0.748 kW) was higher than R404A (0.634 kW) because the theoretical compression heat of R12 was higher than that of R404A.

Although the same evaporator capacity, the amount of specific heat absorbed by the evaporator from the cold store for R12 and R404A were different from each other, being 116.315 kJ kg^{-1} and 126.022 kJ kg^{-1}, respectively.

The amount of specific heat removed from the condenser by air for R12 and R404A were 138.07 kJ kg^{-1} and 146.02 kJ kg^{-1}, respectively. Capacity of the condenser for R12 was much greater because of its smaller specific heat.

As for the cooling efficiency (ϵ) of the system, R404A had superiority over R12 because the specific heat of R404A was higher and the theoretical compression work was lower than that of R12.

Evaporation pressure of R404A was higher than R12. This caused to increase the vapour density and to establish more cooling effect.

Suction and compression pressures of R404A were higher than Freon 12. High suction and compression pressure decrease compressor life.

According to these results and considering its good specifications known, R404A is suggested as a refrigerant.

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