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**Conversion of Prototypes
at Electro Still Co.**

UNIDO PROJECT NUMBER
MP/IRA/97/196

CONTRACT NUMBER 98/071P

FINAL REPORT

OCTOBER 1998

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SECTION I

Synopsis

UNIDO's Project no. MP/IRA/97/196 was nominated by UNIDO to phase out ODS in the first group of Iranian Commercial Refrigerator Manufacturers. This project will be implemented in four major companies, Electrossteel, Zagross II, Yakh saran, and Yakh Chavan.

General Background

This contract was achieved based on the UNIDO's request for proposal no. P.98/32 dated 24 February 1998, our proposal no. ANA/PRP/ES/01 and relevant terms of reference prepared by UNIDO and the requirement of Islamic Republic of Iran. The project will phase out, use of CFC11 and CFC12 for the production of Commercial Refrigerators in Four Refrigerator Manufacturers in Iran. The redefinition of the existing refrigerator models in these companies (Electrosteel, Zagross II, Yakh saran, and Yakhchavan) covers activities such as calculation and refrigeration system components selection



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Electro Steel Mfg. Co. Background

Electro Steel is a private company located in the Mashahad employing 150 personnel, it was established in 1977 and manufactures a wide range of commercial refrigeration equipment.

The company is wholly Iranian owned and all products are manufactured and sold in Iran.

- Iceland-type display case, 6 and 12 m length
- Display cabinets for bakery products
- Open-type multi-deck cabinet, up to 3 m length, 2 m high
- Sales chilled display cabinet
- Chest freezers, 3 models
- Drinking water cooler, 3 models
- Milk coolers, 2,500 to 6,000 liters
- Sandwich panels, 2.7 to 12m length
- Cold chambers and cold stores



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PROJECT OBJECTIVE

The objective of this project is to eliminate the use of CFC-11 and CFC-12 in the production of commercial refrigeration equipment at Electro Steel through conversion to the use of HFC 134a as refrigerant for the cooling system and Cyclopentane as blowing agent for the polyurethane insulation foam.

The Aim of Project

The aim of the project is design, calculation and drafting for model redefinition of eight different models of commercial refrigerators and freezers as follows, i.e., to calculate the cooling capacity and selection of the cooling units of all models so that they could run the new ozone friendly refrigerant R134a instead of the ODP active CFC-12.

- 1) Ice-land Display one model.
- 2) Display Cabinet for Bakery
- 3) Open Type Multi-deck Cabinet
- 4) Sales Chilled Display Cabinet
- 5) Chest Freezers
- 6) Drinking Water Cooler
- 7) Milk Cooler
- 8) Cold Chamber

The Scope of the Contract

- A study has been made of eleven models manufactured by Electro Steel on;



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- 1) Dimensional specifications
 - 2) Type and insulation thickness
 - 3) Refrigerating unit component details
 - 4) Working performance
 - 5) Energy Consumption
 - 6) Optimization of R134a Refrigerant
-
- Selection of **HFC-134a** compatible components.
 - Redesigning of the complete refrigeration circuit.
 - Specifying necessary change in the cooling system if required.
 - Preparation of one prototype per model.

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Refrigeration Load Calculation for Water Cooler and Chest Freezer and Display cases

Refrigeration load consist of three individual components:

- 1- Transmission load;
Heat transfer through side walls by conduction
- 2 - Product load;
Heat Removed from and produced by the products which are stored.
- 3 - Internal load;
Heat produced by internal sources such as lights, fan or heaters;
- 4 - Infiltration load
Heat gains associated with air entering the refrigerated space and door opening and etc.;

In this section , the above mentioned components will be discussed separately to analyze and extract the most useful and practical equipment's.

Transmission Load

Heat gain through walls of a refrigerated space depends on cabin Temperature, liner, insulation and cabin conductivity and also the surrounded ambient air. In other word, there are four different resistance opposing heat flows between cabin space and ambient air as given in resistance circuit.

$$T_{\text{refrigerator}} \longleftarrow R_{\text{liner}} + R_{\text{insulation}} + R_{\text{cabin}} + R_{\text{ambient}} \longleftarrow T_{\text{ambient}}$$

Considering the above mentioned resistance, R_l , R_c and R_a are not comparable in magnitude with R_i (Insulation resistance) and so can be neglected in our calculations. Therefore, the resultant circuit and related equations is.

$$R = \frac{x}{KA} \quad \text{Heat Resistance}$$

$$Q_{TL} = \frac{\Delta T}{R} \quad \text{Heat Transfer}$$



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Where:

x = Insulation Thickness, mm

K = Insulation Conductivity, $Wmm/m^2 \cdot C$

A = Outside Area, m^2

ΔT = Temperature difference ($T_a - T_c$), C

If the insulation thickness of side walls, back panels, top, bottom and door are different. Heat transfer for each part can be calculated separately and then summed for freezer and refrigerator compartments as necessary, heat transfer for each compartment should be calculated separately and then added together.

Product Load

Heat removed from products (meat, fruits, vegetables, water and etc.) to reduce temperature from receiving to storage temperature is known as product load. Following steps can be taken to calculated of product loads.

1 - Heat removed from initial temperature (T_i) to storing temperature (T_{rs}) in refrigerator compartment is;

$$Q_{rs} = \dot{M} C (T_i - T_{rs})$$

Where:

\dot{M} = Mass of product, Kg / h

C = Specific heat of product, Kcal / Kg

2 - Heat removed from initial temperature (T_i) to freezing temperature (T_f) is

$$Q_{df} = \dot{M} C (T_i - T_f)$$

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Where:

\dot{M} = Mass of product, Kg / h

C = Specific heat of product above freezing point, Kcal / Kg

3 - Latent heat of fusion for products is equal to;

$$Q_L = \dot{M} h$$

Where h = Latent heat of product, Kcal / Kg

4 - Heat removed from freezing temperature (T_f) to final storage temperature (T_{fs}) is;

$$Q_{bf} = \dot{M} C_{bf} (T_f - T_{fs})$$

Where:

C_{bf} = Specific heat of products below freezing temperature.

For upright freezers or chest freezer, total product load is

$$Q_{pl} = Q_{af} + Q_l + Q_{bf}$$

For storage products to some lower temperatures above freezing temperature in refrigerator display cases compartment is;

$$Q_{pl} = Q_{rs}$$

Internal Load

Electrical energy dissipated in the refrigerated space such as lights, fan motors, heaters, should be calculated as appropriate depending on type of display cases and other products.

Infiltration Load

Infiltration air load is the heat transfer due to exchanging of refrigerated air with ambient caused by opening of the door or leakage through the gasket area and /or



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open top freezer of show cases. Infiltration load is one of the most important load components.

Total Refrigeration load

As it was mentioned before, transmission load (Q_{tl}), product load (Q_{pl}) and internal load (Q_{il}) can be calculated separately. For infiltration load (air exchange through doorways or gasket leakage), we have to take into account that depending on the type of models we have to consider different amount of heat gain, or a percentage of amount of the above mentioned components. (Transmission load, product load and internal load). For example;

$$\underline{Q_{TL} = 1.20 (Q_{TL} + Q_{PL} + Q_{IL})}$$

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Refrigeration Load Calculation for different type of Water Coolers

Water cooler cabinet usually consist of a sheet metal housing built around a steel framework, inside this sheet housing there is usually a condensing unit, located near the floor, and above this is the water-cooling mechanism. The latter is the only part insulated (foamed plastic) from the room. The insulation is usually specially formed and between one and one half inches and two inches thick. These cabinets are made in such a way that one or more sides may be easily removed to gain access to the interior. The basin of the water cooler is generally made of porcelain-coated cast iron, porcelaicoated- steel, or stainless steel. Heat exchangers are frequently used on water coolers. These make use of the low temperature of waste water and the suction line to pre-cool the fresh water line to the evaporator coil.

Self-are of two types,

- 1- Bottle Type.
- 2- Tap water type

The bottle cooler usually uses a 20 to 25 liter bottle of water inverted on the top of the cabinet. Overflow and drain water are stored in a container built the cabinet. These coolers use air-cooled condensing units exclusively. They are used where water and drains are not available or where available the plumbing insulation may be expensive.

Water cooler using a plumbing supply and drain connection, must be installed according the relevant approved standards. The plumbing should be concealed, a hand shutoff valve should be installed in the fresh water line. Drain pipe at least 1 ½ inches in diameter provided, and rubber opening must be above the drain in such a way as to eliminate the chance for accidental siphoning of the drain water back into the fresh water system. The tap water models use variety of evaporator coil wrapped around the water-cooling tank.

Temperatures of the cooling water are variable depending on the persons who are drinking the water. We consider 10 C for the temperature of drinking water, while our inlet temperature is considered 24 C.

In large business establishment, in office buildings, or in factories, multiple water cooler, instead of individual ones, are popular. These coolers have one large condensing unit supplying many bubblers and these may be of many different types.



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Water cooler is a device that usually is used in the public area to supply cold drinking water to the customers and different people. The appliance is mainly used in the Airports, Railways Station, Coach Terminals, Banks, Offices, Parks, and etc. therefore, it is hard to specify an standard for cold water consumption during the day from the water cooler.

We consider three refrigeration load components that should be taken into our consideration.

- 1- Heat gain by heat transmission from, main water storage tank wall insulation.
- 2- Heat removed from water entering to the water tank at the initial refrigeration system operating condition, (water stored in storage tank during the night, with normal ambient temperature) which is divided by 24 hrs.
- 3- Heat removed from Drinking Water flow that are consumed during designated operating hours " \dot{M} "

The problem of determining the refrigeration load of a water-cooled installation is basically a specific heat and heat leakage problem combination. The water is cooled to temperature which vary upward from about 4 degree centigrade, and the amount heat removed from the water to cool it to a predetermined temperature is simple specific heat problem. The water, being maintained at these low temperature, results in a heat leakage from room into the water, and this part involves the heat leakage portion of installation.

$$\underline{Q_1 = m C \Delta T}$$

Where:

Q₁ Total heat removed from total drinking water tank volume capacity (lit.) during specific period, related to compressor cooling capacity power in Watts, at initial compressor start up, and early in the morning. When the water temperature is 30 C.

m total weight of water in the water cooler storage tank in Kg. Considering that one liter of water at 24 C is equal to approximately one Kg.

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C Specific heat factor of water in Kcal/Kg °C

T Temperature difference ($T_i - T_c$), where, T_i is inlet water temperature, and T_c is final cooled water.

$$Q_2 = \dot{M} C \Delta T$$

Q₂ Total heat removed from total drinking water flow (lit.) during specific period, 16 hours. In Kcal.

\dot{M} total weight of water flow during 16 hours. in Kg.

C Specific heat factor of water in Kcal/Kg °C

T Temperature difference ($T_i - T_c$), where, T_i is inlet water temperature, and T_c is final cooled water temperature.

$$Q_3 = UA \Delta T$$

Where:

Q₃ Total Leak, gained through side wall of drinking water storage tank by conduction in Kcal..

U Heat Resistance Coefficient Factor in Kcal/Sq. mt. C

A Total Area which heat is transmitted by. In Sq. Mt.

T Temperature difference ($T_a - T_c$), where, T_a is ambient temperature, and T_c is final cooled water temperature.



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Section III

Heat Leak Load Calculation of Prototypes

Calculation of refrigeration load is the basis for selecting system equipment. First step is selection of a suitable compressor with cooling capacity comparable to calculated load, then a capillary tube should be selected so that the compressor and tube fix a balance point at the desired evaporating temperature, also two evaporator and condenser should be selected to balance compressor capacity.

Compressor selection

Assuming 16 hours daily operating time for the compressor, the calculated refrigeration load will be modified to:

$$Q_c = \frac{Q_{TL \times 24}}{16} = 1.5 Q_{TL}$$

Where :

Qc = required cooling capacity

Saturation Properties Comparison

Temp C	R12					R134a				
	P Kpa	Entholpy Kj/Kg	Entholpy Kj/Kg	Sp.Vol Lit/Kg	Sp.Vol Lit/Kg	P Kpa	Entholpy Kj/Kg	Entholpy Kj/Kg	Sp.Vol Lit/Kg	Sp.Vol Lit/Kg
		hf	hg	Vf	Vg		hf	hg	Vf	Vg
-30	100.41	172.81	338.14	0.672	159.37	84.36	61.51	277.208	0.7100	0.2219
-26	118.72	176.38	339.96	0.677	136.28	101.65	66.56	212.96	0.7171	0.1868
-22	139.53	179.96	341.78	0.682	117.16	121.62	71.63	281.86	0.7243	0.1570
-18	163.04	183.56	343.58	0.688	101.24	144.56	76.72	284.19	0.7318	0.1313
-14	189.50	187.18	345.36	0.694	87.89	170.76	81.84	286.52	0.7396	0.1138
-10	219.12	190.82	347.13	0.700	76.64	200.51	86.98	288.85	0.7475	0.0941
-6	252.14	194.47	348.88	0.706	67.11	234.13	92.162	291.18	0.7558	0.0843
-4	270.01	196.31	349.75	0.709	62.89	252.49	94.76	292.35	0.7600	0.0784
-2	288.82	198.15	350.61	0.712	58.99	271.94	97.377	293.522	0.7643	0.0730
0	308.61	200.00	351.47	0.715	55.38	292.52	100.00	294.68	0.7687	0.0681
2	329.40	201.85	352.33	0.719	52.04	314.27	102.63	295.35	0.7732	0.0635
4	351.24	203.71	353.17	0.722	48.94	337.24	105.28	297.01	0.7777	0.0592
6	374.14	205.57	354.02	0.726	46.07	361.47	107.93	298.017	0.7823	0.0552
8	398.15	207.44	354.85	0.729	43.40	387.01	110.60	299.33	0.7870	0.0520
10	423.30	209.32	355.68	0.733	40.91	413.90	113.29	300.49	0.7918	0.0482



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Load Calculation of Water Cooler Model EL-WC18

1-

$$Q_1 = m C \Delta T$$

Where:

Q₁ Total heat removed from total drinking water tank volume capacity (lit.) during specific period, related to compressor cooling capacity power in Watts, at initial compressor start up, and early in the morning. When the water temperature is 24 C.

m total weight of original water in the water cooler storage tank in Kg. Considering that one liter of water at 24 C is equal to approximately one Kg.

$$M = 4.5 \text{ liter} = 4.5 \text{ Kg.}$$

C Specific heat factor of water in Kcal/Kg °C = 1

T Temperature difference (Ti - Tc), where, Ti is inlet water temperature, and Tc is final cooled water.

$$T_i = 24 \text{ °C and } T_c = 10 \text{ °C}$$

$$T_i - T_c = 24 - 10 = 14 \text{ °C}$$

$$Q_1 = m C T = 4.5 \times 1 \times 14 = 63 \text{ Kcal} = 63 \times 1.163 = 73.3 \text{ Watts/24 hrs}$$

$$Q_1 = 73.3 / 16 \text{ water cooler operating time per day} = 4.6 \text{ Watts}$$

$$\underline{\underline{Q_1 = 4.6 \text{ Watts}}}$$



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2-

$$Q_2 = \dot{M} C \Delta T$$

Q_2 Total heat removed from total drinking water flow (lit.) during specific period, 16 hours. In Kcal.

\dot{M} total weight of water flow during 16 hours. in Kg. = H x N x M
for office building with 20 employees, where:

H = Total Water Cooler Usage Time (Hours) = 16

N = Number of person at the office = 20

M = 400 cc of water per person per hour (including waste water)

$$\dot{M} = 0.4 \times 20 \times 16 = 128 \text{ lit.}$$

C Specific heat factor of water in Kcal/Kg °C = 1

T Temperature difference ($T_i - T_c$), where, T_i is inlet water temperature, and T_c is final cooled water temperature.

$$T_i = 24 \text{ °C and } T_c = 10 \text{ °C}$$

$$T_i - T_c = 24 - 10 = 14 \text{ °C}$$

$$Q_2 = m C T = 128 \times 1 \times 14 = 1792 \text{ Kcal} = 1792 \times 1.163 = 2084 \text{ Watts/16 hrs}$$

$$Q_2 = 2084/12 \text{ compressor operating time per day} = 173 \text{ Watts}$$



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$$\underline{Q_2 = 173.7 \text{ Watts}}$$

3-

$$Q_3 = UA\Delta T$$

Where:

Q₃ Total Leak, gained through side wall of drinking water storage tank by conduction in Kcal..

U Heat Resistance Coefficient Factor in Kcal/Sq. mt. C

$$U = \frac{1}{\frac{x}{K}} = \frac{1}{\frac{0.05}{0.019}} = 0.38 \text{ Kcal} / \text{m}^2 \cdot \text{°C}$$

A The water storage tank is considered as cylinder with 32 cm diameter and 12.5 cm height. therefore total area which heat is transmitted by. in Sq. Mt. is calculated as follow.

$$A_1 = A_2 \text{ bottom ant top area} = 3.14 \times 6.5 \times 6.5 = 133 \text{ square cm. } 0.0133 \text{ sq. m.}$$

$$A_3 = \text{Cylinder side wall} = 3.14 \times 13 \times 34 = 1388 \text{ sq. cm} = 0.1388 \text{ sq. m.}$$

Where; $A_1 = A_2 =$ bottom and top surface area of the storage tank are the same,

$$A = A_1 + A_2 + A_3 = 0.0133 + 0.0133 + 0.1388 = 0.1654 \text{ sq. m.}$$

T difference ($T_a - T_c$), where, T is ambient temperature, and T_c is final cooled water temperature.

$$T_a = 30 \text{ °C and } T_c = 10 \text{ °C}$$

$$T_a - T_c = 30 - 10 = 20 \text{ °C}$$

$$Q_3 = UA \Delta T = 0.38 \times 0.1654 \times 20 = 1.25 \text{ Watts}$$



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$$Q3 = 4.35 \text{ Watts}$$

$$Q_t = Q1 + Q2 + Q3 = 4.7 + 173.7 + 1.26 = 179.7 \text{ Watts}$$

$$Q_{\text{Grand Total}} = Q_t + 10\% \text{ safety factor} = 197.6$$

Compressor R134a, Model FR6G (with total cooling capacity 225 watts at -10 C evaporating temperature) manufactured by DANFOSS, is selected as a suitable compressor to replace R12 compressor model FR6B.

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Load Calculation of Water Cooler Model ELC- WC50C

1-

$$Q_1 = m C \Delta T$$

Where:

Q₁ Total heat removed from total drinking water tank volume capacity (lit.) during specific period, related to compressor cooling capacity power in Watts, at initial compressor start up, and early in the morning. When the water temperature is 24 C.

m total weight of original water in the water cooler storage tank in Kg. Considering that one liter of water at 24 C is equal to approximately one Kg.

$$M = 15 \text{ liter} = 15 \text{ Kg.}$$

C Specific heat factor of water in Kcal/Kg °C = 1

T Temperature difference ($T_i - T_c$), where, T_i is inlet water temperature, and T_c is final cooled water.

$$T_i = 24 \text{ °C and } T_c = 10 \text{ °C}$$

$$T_i - T_c = 24 - 10 = 14 \text{ °C}$$

$$Q_1 = m C T = 15 \times 1 \times 14 = 210 \text{ Kcal} = 210 \times 1.163 = 244.23 \text{ Watts/24 hrs}$$

$$Q_1 = 244 / 16 \text{ water cooler operating time per day} = 15.2 \text{ Watts}$$

$$\underline{\underline{Q_1 = 15.2 \text{ Watts}}}$$

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2-

$$Q_2 = \dot{M} C \Delta T$$

Q₂ Total heat removed from total drinking water flow (lit.) during specific period, 16 hours. In Kcal.

\dot{M} In a heavy manufacturing factory, like a foundry employs 50 men for a period of 8 hours, the water load per day would be

\dot{M} = number of employee X working hour X amount of water used and wasted per person

H = Working Hour = 8

N = Number of number of employee = 50

M = Kg weight of water used per person per hour = 1/8 US Gal 0.472 lit

$$\dot{M} = H \times N \times M = 8 \times 50 \times 0.472 = 189 \text{ lit.}$$

C Specific heat factor of water in Kcal/Kg °C = 1

T Temperature difference (Ti - Tc), where, Ti is inlet water temperature, and Tc is final cooled water temperature.

$$T_i = 24 \text{ °C and } T_c = 10 \text{ °C}$$

$$\underline{T_i - T_c = 24 - 10 = 14 \text{ °C}}$$

$Q_2 = m C T = 189 \times 1 \times 14 = 2646 \text{ Kcal} = 2646 \times 1.163 = 3077 \text{ Watts/8 hrs.}$ we consider maximum 80% compressor running time.



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$$Q_2 = 4949/6 \text{ compressor operating time per day} = 512.9 \text{ Watts}$$

$$\underline{Q_2 = 512.9 \text{ Watts}}$$

3-

$$Q_3 = UA\Delta T$$

Where:

Q₃ Total Leak, gained through side wall of drinking water storage tank by conduction in Kcal..

U Heat Resistance Coefficient Factor in Kcal/Sq. mt. C

$$U = \frac{1}{\frac{x}{k}} = \frac{1}{\frac{0.005}{0.019}} = 0.38 \text{ Kcal} / \text{m}^2 \cdot \text{°C}$$

A The water storage tank is considered as cylinder with 32 cm diameter and 14 cm height. therefore total area which heat is transmitted by. in Sq. Mt. is calculated as follow.

$$A_1 = A_2 \text{ bottom and top area} = 3.14 \times 13 \times 13 = 530 \text{ square cm. } 0.0530 \text{ sq. m.}$$

$$A_3 = \text{Cylinder side wall} = 3.14 \times 26 \times 28 = 2286 \text{ sq. cm} = 0.2286 \text{ sq. m.}$$

Where; $A_1 = A_2$ = bottom and top surface area of the storage tank are the same,

$$A = A_1 + A_2 + A_3 = 0.0530 + 0.0530 + 0.2286 = 0.3346 \text{ sq. m.}$$

T difference ($T_a - T_c$), where, T is ambient temperature, and T_c is final cooled water temperature.

$$T_a = 30 \text{ °C and } T_c = 10 \text{ °C}$$

$$T_a - T_c = 30 - 10 = 20 \text{ °C}$$

$$Q_3 = UA \Delta T = 0.38 \times 0.3346 \times 20 = 2.54 \text{ Watts}$$



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$$Q_3 = 2.54 \text{ Watts}$$

$$Q_t = Q_1 + Q_2 + Q_3 = 15.2 + 512.9 + 2.54 = 530.6 \text{ Watts}$$

$$Q_{\text{Grand Total}} = 530$$

Compressor R134a, Model Danfoss SC15G (with total cooling capacity 565 watts at -10 C evaporating temperature CECOMAF Standard with condensing temperature 55 C and liquid temperature 55 C) manufactured by DANFOSS, is selected as a suitable compressor to replace R12 compressor model SC12B.



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Load Calculation of Front Two Taps Water Cooler Model ELC- WC50T

1-

$$Q_1 = m C \Delta T$$

Where:

Q₁ Total heat removed from total drinking water tank volume capacity (lit.) during specific period, related to compressor cooling capacity power in Watts, at initial compressor start up, and early in the morning. When the water temperature is 24 C.

m total weight of original water in the water cooler storage tank in Kg. Considering that one litter of water at 24 C is equal to approximately one Kg.

$$M = 72.5 \text{ liter} = 72.5 \text{ Kg.}$$

C Specific heat factor of water in Kcal/Kg °C = 1

T Temperature difference (Ti - Tc), where, Ti is inlet water temperature, and Tc is final cooled water.

$$T_i = 24 \text{ °C and } T_c = 10 \text{ °C}$$

$$T_i - T_c = 24 - 10 = 14 \text{ °C}$$

$$Q_1 = m C T = 72.5 \times 1 \times 14 = 1015 \text{ Kcal} = 1015 \times 1.163 = 1180 \text{ Watts/24 hrs}$$

$$Q_1 = 1180 / 16 \text{ water cooler operating time per day} = 73.8 \text{ Watts}$$

$$\underline{\underline{Q_1 = 73.8 \text{ Watts}}}$$



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2-

$$Q_2 = \dot{M} C \Delta T$$

Q₂ Total heat removed from total drinking water flow (lit.) during specific period, 16 hours. In Kcal.

\dot{M} In a heavy manufacturing factory, like a foundry employs 50 men for a period of 8 hours, the water load per day would be

\dot{M} = number of employee X working hour X amount of water used and wasted per person

H = Working Hour = 8

N = Number of number of employee = 50

M = Kg weight of water used per person per hour = 1/8 US Gal 0.472 lit

$$\dot{M} = H \times N \times M = 8 \times 50 \times 0.472 = 189 \text{ lit.}$$

C Specific heat factor of water in Kcal/Kg °C = 1

T Temperature difference (Ti - Tc), where, Ti is inlet water temperature and Tc is final cooled water temperature.

$$T_i = 24 \text{ }^\circ\text{C} \text{ and } T_c = 10 \text{ }^\circ\text{C}$$

$$\underline{T_i - T_c = 24 - 10 = 14 \text{ }^\circ\text{C}}$$

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$Q_2 = m C T = 189 \times 1 \times 14 = 2646 \text{ Kcal} = 2646 \times 1.163 = 3077 \text{ Watts/8 hrs.}$ we consider maximum 80% compressor running time.

$Q_2 = 4949/6$ compressor operating time per day = 512.9 Watts

$Q_2 = 512.9 \text{ Watts}$

3-

$Q_3 = UA\Delta T$

Where:

Q_3 Total Leak, gained through side wall of drinking water storage tank by conduction in Kcal..

U Heat Resistance Coefficient Factor in Kcal/Sq. mt. C

$$U = \frac{1}{\frac{x}{K}} = \frac{1}{\frac{0.005}{0.019}} = 0.38 \text{ Kcal} / \text{m}^2 \cdot \text{°C}$$

A The water storage tank is considered as cylinder with 32 cm diameter and 14 cm height. therefore total area which heat is transmitted by. in Sq. Mt. is calculated as follow.

$A_1 = A_2$ bottom ant top area = $50 \times 50 = 2500$ square cm. 0.2500 sq. m.

$A_3 = A_4 = A_5 = A_6$ side wall = $50 \times 33 = 1650$ sq. cm = 0.1560 sq. m.

Where; $A_1 = A_2 =$ bottom and top surface area of the storage tank are the same,

$A = A_1 + A_2 + A_3 + A_4 + A_5 + A_6 = 0.2500 + 0.2500 + 4 \times (0.1650) = 1.16 \text{sq. m.}$

T difference ($T_a - T_c$), where, T is ambient temperature, and T_c is final cooled water temperature.

$T_a = 30 \text{ °C}$ and $T_c = 10 \text{ °C}$



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$$T_a - T_c = 30 - 10 = 20 \text{ }^\circ\text{C}$$

$$Q_3 = UA \quad T = 0.38 \times 1.16 \times 20 = 8.8 \text{ Watts}$$

$$Q_3 = 8.8 \text{ Watts}$$

$$Q_t = Q_1 + Q_2 + Q_3 = 73.8 + 512.9 + 8.8 = 595.5 \text{ Watts}$$

$$Q_{\text{Grand Total}} = 595$$

Compressor R134a, Model Danfoss SC15G (with total cooling capacity 565 watts at $-10 \text{ }^\circ\text{C}$ evaporating temperature CECOMAF Standard with condensing temperature $55 \text{ }^\circ\text{C}$ and liquid temperature $55 \text{ }^\circ\text{C}$) manufactured by DANFOSS, is selected as a suitable compressor to replace R12 compressor model SC12B.

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Chest Freezer Model ELS-CF450

a) Transmission Load Calculation

Dimension

	Dimension Cm.	Area (sq. mt.)	Insulation Thickness mm
Side Walls	2 x (71x70)	0.9940	65
Front & Back Panel	2 x (132x70)	1.8480	65
Chest Door	71 x 132	0.9372	65
Bottom Floor	71 x 132	0.9372	65

Insulation Type: Pu Foam Cyclopentane expanded blowing PU foam

Cyclopentane Foam Thermal Conductivity: 0.019 W /mt.C

Temperature Difference: $(\Delta T) = 43 - (-18) = 61$ C

Ambient Temperature = 43 C

Freezer Air Temperature = - 18 C

Calculation :

$$Q_{TL} = Q_{\text{side Walls}} + Q_{\text{Bottom}} + Q_{\text{Top}}$$

$$Q = U A (T_a - T_r)$$

$$U = \frac{1}{X_1 / K_1 + X_2 / K_2 + \dots}$$

Where :

U = Heat Resistance Coefficient Factor

K₁ = Foam Thermal Conductivity

X₁ = Foam Thickness



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Note : Due to the short thickness of cabinet out side panel (0.6 mm) and plastic inner liner (1.5 mm) heat resistance of these materials have been considered negligible.

Therefore:

1- $Q_{\text{SideWalls}} = [U A (T_a - T_f)]$

T_a = Ambient Temperature

T_f = Freezer air Temperature

$$U = 1 / (0.065/0.019) = 0.29 \text{ W/ sq.m C}$$

$$A = 0.9940 \text{ Sq. Mt.}$$

$$T_a = 43 \text{ C}$$

$$T_f = - 18 \text{ C}$$

$$Q_{\text{SideWalls}} = 0.29 \times 0.9940 \times 61 = 17.6 \text{ Watts}$$

$$Q_{\text{SideWalls}} = 17.6 \text{ Watts}$$

2 - $Q_{\text{Front \& Back Walls}} = [U A (T_a - T_f)]$

T_a = Ambient Temperature

T_f = Freezer air Temperature

$$U = 1 / (0.065/0.019) = 0.29 \text{ W/ sq.m C}$$

$$A = 1.8480 \text{ Sq. Mt.}$$

$$T_a = 43 \text{ C}$$

$$T_f = - 18 \text{ C}$$

$$Q_{\text{Front \& Back Walls}} = 0.29 \times 1.8480 \times 61 = 32.7 \text{ Watts}$$

$$Q_{\text{Front \& Back Walls}} = 32.7 \text{ Watts}$$

3 - $Q_{\text{Top door}} = [U A (T_a - T_f)]$

T_a = Ambient Temperature



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T_f = Freezer air Temperature

$$U = 1 / (0.065 / 0.019) = 0.29 \text{ W/ sq.m C}$$

$$A = 0.9372 \text{ Sq. Mt.}$$

$$T_a = 43 \text{ C}$$

$$T_f = - 18 \text{ C}$$

$$Q_{\text{Top door}} = 0.29 \times 0.9372 \times 61 = 16.6 \text{ Watts}$$

$$Q_{\text{Top door}} = 16.6 \text{ Watts}$$

4 - $Q_{\text{Bottom}} = [U A (T_a - T_f)]$

T_a = Ambient Temperature

T_f = Freezer air Temperature

$$U = 1 / (0.065 / 0.019) = 0.29 \text{ W/ sq.m C}$$

$$A = 0.9372 \text{ Sq. Mt.}$$

$$T_a = 55 \text{ C}$$

$$T_f = - 18 \text{ C}$$

$$Q_{\text{Bottom}} = 0.29 \times 0.9372 \times 73 = 20.6 \text{ Watts}$$

$$Q_{\text{Bottom}} = 19.8 \text{ Watts}$$

Total Heat Leaks;

$$Q_{\text{TL}} = 17.6 + 32.7 + 16.6 + 19.8 = 86.7 \text{ watts}$$

$$Q_{\text{Total Heat Leaks}} = 86.7 \text{ Watts}$$

b) Product Loads;

Through our knowledge, experience and facts and figures of calculation of Electro Still products we found out that calculation of product loads for each individual model depends upon many factors that we could mention briefly as follows:



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- Product design;
- Product style;
- Company policy;
- Useful internal volume;
- Type of evaporator;
- Type of cellar compartment;
- Freezer volume;
- Culture of customer;
- Country of origin and etc.;

We consider 10 kg of Ice Making per hour capability for this model of chest freezer, with respect to the size of the model.

$$\text{Ice Making Capacity} = 10 \text{ kg} \times 1 \times (15 - 0) \times 1.163 = 174.45 \text{ Watts}$$

c) Heat gain through infiltration;

Total heat gain through infiltration (door opening, and gasket) are considered to 10 % of total heat gain by conduction and heat removed from products, therefore;

$$\text{Heat gain by infiltration} = 0.1 \times (\text{total heat leaks})$$

$$\text{Heat gain by infiltration} = 0.1 \times (86.7) = 8.7 \text{ Watts}$$

Total Cooling Capacity Required are calculated as follows;

$$Q_{\text{Grand Total}} = Q_{\text{Heat Leaks}} + Q_{\text{Ice Making}} + Q_{\text{Infiltration}}$$

$$Q_{\text{Grand Total}} = 91.2 + 174.25 + 8.7 = 269.7 \text{ Watts}$$

$$Q_{\text{Grand Total}} = 269.7 \text{ Watts}$$

The Suitable Compressor selected for this Model is Danfoss SC18G with a total 280 watts cooling capacity, at - 25 C evaporating temperature at CECOMAF test condition, (55 degree condensing and liquid temperature)



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Chest Freezer Model ELS-CF750

a) Transmission Load Calculation

Dimension

	Dimension Cm.	Area (sq. mt.)	Insulation Thickness mm
Side Walls	2 x (72.5x82)	1.1890	65
Front & Back Panel	2 x (189x82)	3.0995	65
Chest Door	72.5 x 189	1.370	65
Bottom Floor	72.5 x 189	1.370	65

Insulation Type: Pu Foam Cyclopentane expanded blowing PU foam

Cyclopentane Foam Thermal Conductivity: 0.019 W /mt.C

Temperature Difference: $(\Delta T) = 43 - (-18) = 61$ C

Ambient Temperature = 43 C

Freezer Air Temperature = - 18 C

Calculation :

$$Q_{TL} = Q_{side\ Walls} + Q_{Bottom} + Q_{Top}$$

$$Q = U A (T_a - T_f)$$

$$U = \frac{1}{X_1 / K_1 + X_2 / K_2 + \dots}$$

Where :

U = Heat Resistance Coefficient Factor

K₁ = Foam Thermal Conductivity

X₁ = Foam Thickness



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Note : Due to the short thickness of cabinet out side panel (0.6 mm) and plastic inner liner (1.5 mm) heat resistance of these materials have been considered negligible.

Therefore:

1- $Q_{\text{SideWalls}} = [U A (T_a - T_f)]$

T_a = Ambient Temperature

T_f = Freezer air Temperature

$$U = 1 / (0.065/0.019) = 0.29 \text{ W/ sq.m C}$$

$$A = 1.189 \text{ Sq. Mt.}$$

$$T_a = 43 \text{ C}$$

$$T_f = - 18 \text{ C}$$

$$Q_{\text{SideWalls}} = 0.29 \times 1.189 \times 61 = 21 \text{ Watts}$$

$$Q_{\text{SideWalls}} = 21 \text{ Watts}$$

2 - $Q_{\text{Front \& Back Walls}} = [U A (T_a - T_f)]$

T_a = Ambient Temperature

T_f = Freezer air Temperature

$$U = 1 / (0.065/0.019) = 0.29 \text{ W/ sq.m C}$$

$$A = 3.0995 \text{ Sq. Mt.}$$

$$T_a = 43 \text{ C}$$

$$T_f = - 18 \text{ C}$$

$$Q_{\text{Front \& Back Walls}} = 0.29 \times 3.0995 \times 61 = 54.8 \text{ Watts}$$

$$Q_{\text{Front \& Back Walls}} = 54.8 \text{ Watts}$$

3 - $Q_{\text{Top door}} = [U A (T_a - T_f)]$

T_a = Ambient Temperature

T_f = Freezer air Temperature



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$$U = 1 / (0.065/0.019) = 0.29 \text{ W/ sq.m C}$$

$$A = 1.370 \text{ Sq. Mt.}$$

$$T_a = 43 \text{ C}$$

$$T_f = - 18 \text{ C}$$

$$Q_{\text{Top door}} = 0.29 \times 1.370 \times 61 = 24.2 \text{ Watts}$$

$$Q_{\text{Top door}} = 24.2 \text{ Watts}$$

4 -
$$Q_{\text{Bottom}} = [U A (T_a - T_f)]$$

T_a = Ambient Temperature

T_f = Freezer air Temperature

$$U = 1/(0.065/ 0.019) = 0.29 \text{ W/ sq.m C}$$

$$A = 1.370 \text{ Sq. Mt.}$$

$$T_a = 55 \text{ C}$$

$$T_f = - 18 \text{ C}$$

$$Q_{\text{Bottom}} = 0.29 \times 1.37 \times 73 = 29 \text{ Watts}$$

$$Q_{\text{Bottom}} = 29 \text{ Watts}$$

Total Heat Leaks;

$$Q_{\text{TL}} = 21 + 54.8 + 24.2 + 29 = 129 \text{ watts}$$

$$Q_{\text{Total Heat Leaks}} = 129 \text{ Watts}$$

b) Product Loads;

Through our knowledge, experience and facts and figures of calculation of Electro Still products we found out that calculation of product loads for each individual model depends upon many factors that we could mention briefly as follows:



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- Product design;
- Product style;
- Company policy;
- Useful internal volume;
- Type of evaporator;
- Type of cellar compartment;
- Freezer volume;
- Culture of customer;
- Country of origin and etc.;

We consider 10 kg of Ice Making per hour capability for this model of chest freezer, with respect to the size of the model.

$$\text{Ice Making Capacity} = 24 \text{ kg} \times 1 \times (15 - 0) \times 1.163 = 418.7 \text{ Watts}$$

c) Heat gain through infiltration;

Total heat gain through infiltration (door opening, and gasket) are considered to 10 % of total heat gain by conduction and heat removed from products, therefore;

$$\text{Heat gain by infiltration} = 0.1 \times (\text{total heat leaks})$$

$$\text{Heat gain by infiltration} = 0.1 \times (129) = 13 \text{ Watts}$$

Total Cooling Capacity Required are calculated as follows;

$$Q_{\text{Grand Total}} = Q_{\text{Heat Leaks}} + Q_{\text{Ice Making}} + Q_{\text{Infiltration}}$$

$$Q_{\text{Grand Total}} = 129 + 418.7 + 13 = 460.7 \text{ Watts}$$

$$Q_{\text{Grand Total}} = 561 \text{ Watts}$$

The Suitable Compressor selected for this Model is Danfoss SC18/18G with a total 560 watts cooling capacity, at - 25 C evaporating temperature at CECOMAF test condition, (55 degree condensing and liquid temperature)

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Chest Freezer Model ELS-CF220

a) Transmission Load Calculation

Dimension

	Dimension Cm.	Area (sq. mt.)	Insulation Thickness mm
Side Walls	2 x (54.5x78.5)	0.8556	65
Front & Back Panel	2 x (101x78.5)	1.5857	65
Chest Door	101 x 54.5	0.55	65
Bottom Floor	72.5 x 132	0.55	65

Insulation Type: Pu Foam Cyclopentane expanded blowing PU foam

Cyclopentane Foam Thermal Conductivity: 0.019 W /mt.C

Temperature Difference: $(\Delta T) = 43 - (-18) = 61$ C

Ambient Temperature = 43 C

Freezer Air Temperature = - 18 C

Calculation :

$$Q_{TL} = Q_{\text{side Walls}} + Q_{\text{Bottom}} + Q_{\text{Top}}$$

$$Q = U A (T_a - T_r)$$

$$U = \frac{1}{X_1 / K_1 + X_2 / K_2 + \dots}$$

Where :

U = Heat Resistance Coefficient Factor

K₁ = Foam Thermal Conductivity

X₁ = Foam Thickness

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Note : Due to the short thickness of cabinet out side panel (0.6 mm) and plastic inner liner (1.5 mm) heat resistance of these materials have been considered negligible.

Therefore:

1- $Q_{\text{SideWalls}} = [U A (T_a - T_f)]$

T_a = Ambient Temperature

T_f = Freezer air Temperature

$$U = 1 / (0.065/0.019) = 0.29 \text{ W/ sq.m C}$$

$$A = 0.8556 \text{ Sq. Mt.}$$

$$T_a = 43 \text{ C}$$

$$T_f = - 18 \text{ C}$$

$$Q_{\text{SideWalls}} = 0.29 \times 0.8556 \times 61 = 15.1 \text{ Watts}$$

$$Q_{\text{SideWalls}} = 15.1 \text{ Watts}$$

2 - $Q_{\text{Front \& Back Walls}} = [U A (T_a - T_f)]$

T_a = Ambient Temperature

T_f = Freezer air Temperature

$$U = 1 / (0.065/0.019) = 0.29 \text{ W/ sq.m C}$$

$$A = 1.5857 \text{ Sq. Mt.}$$

$$T_a = 43 \text{ C}$$

$$T_f = - 18 \text{ C}$$

$$Q_{\text{Front \& Back Walls}} = 0.29 \times 1.5857 \times 61 = 28. \text{ Watts}$$

$$Q_{\text{Front \& Back Walls}} = 28 \text{ Watts}$$

3 - $Q_{\text{Top door}} = [U A (T_a - T_f)]$

T_a = Ambient Temperature

T_f = Freezer air Temperature



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$$U = 1 / (0.065/0.019) = 0.29 \text{ W/ sq.m C}$$

$$A = 0.550 \text{ Sq. Mt.}$$

$$T_a = 43 \text{ C}$$

$$T_f = - 18 \text{ C}$$

$$Q_{\text{Top door}} = 0.29 \times 0.550 \times 61 = 9.7 \text{ Watts}$$

$$Q_{\text{Top door}} = 9.7 \text{ Watts}$$

4 -
$$Q_{\text{Bottom}} = [U A (T_a - T_f)]$$

T_a = Ambient Temperature

T_f = Freezer air Temperature

$$U = 1 / (0.065 / 0.019) = 0.29 \text{ W/ sq.m C}$$

$$A = 0.55 \text{ Sq. Mt.}$$

$$T_a = 55 \text{ C}$$

$$T_f = - 18 \text{ C}$$

$$Q_{\text{Bottom}} = 0.29 \times 0.550 \times 73 = 20.6 \text{ Watts}$$

$$Q_{\text{Bottom}} = 11.6 \text{ Watts}$$

Total Heat Leaks;

$$Q_{\text{TL}} = 15.1 + 28 + 9.7 + 11.6 = 64.4 \text{ watts}$$

$$Q_{\text{Total Heat Leaks}} = 64.4 \text{ Watts}$$

b) Product Loads;

Through our knowledge, experience and facts and figures of calculation of Electro Still products we found out that calculation of product loads for each individual model depends upon many factors that we could mention briefly as follows:

- Product design;
- Product style;

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- Company policy;
- Useful internal volume;
- Type of evaporator;
- Type of cellar compartment;
- Freezer volume;
- Culture of customer;
- Country of origin and etc.;

We consider 10 kg of Ice Making per hour capability for this model of chest freezer, with respect to the size of the model.

$$\text{Ice Making Capacity} = 6 \text{ kg} \times 1 \times (15 - 0) \times 1.163 = 104.7 \text{ Watts}$$

c) Heat gain through infiltration;

Total heat gain through infiltration (door opening, and gasket) are considered to 10 % of total heat gain by conduction and heat removed from products, therefore;

$$\text{Heat gain by infiltration} = 0.1 \times (\text{total heat leaks})$$

$$\text{Heat gain by infiltration} = 0.1 \times (64.4) = 6.4 \text{ Watts}$$

Total Cooling Capacity Required are calculated as follows;

$$Q_{\text{Grand Total}} = Q_{\text{Heat Leaks}} + Q_{\text{Ice Making}} + Q_{\text{Infiltration}}$$

$$Q_{\text{Grand Total}} = 64.4 + 104.7 + 6.4 = 175.5 \text{ Watts}$$

$$Q_{\text{Grand Total}} = 175.5 \text{ Watts}$$

The Suitable Compressor selected for this Model is Danfoss SC12G with a total 183 watts cooling capacity, at - 25 C evaporating temperature at CECOMAF test condition, (55 degree condensing and liquid temperature)



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Transmission load calculation
Model Up Right Refrigerator Show Case
Model ELS-1889V

Dimensions

	Dimension Cm.	Area (sq. mt.)	Insulation Thickness
Side Walls	2x(86 x 200)	3.44	50
Back Panel	200 x 195	3.9	50
Top Roof	195 x 86	1.68	50
Bottom Floor	195 x 86	1.68	50
Door	200 x 195	3.9	50

Insulation Type: Pu Foam

Cyclopentane Foam Thermal Conductivity: 0.019 W /mt.C

Temperature Difference: (ΔT) = 43 - 10 = 33 C

Ambient Temperature = 33 C

Refrigerator Air Temperature = 0 C

The refrigerator is designed to be used for keeping fresh lamb containing 67 % of lean. The initial fresh lamb temperature is considered to be 24 C. the specific heat of lamb containing 67% lean is 0.76 Kcal/hr kg. C.

Calculation :

$$Q_{TL} = Q_{SW} + Q_{BP} + Q_{BOTTOM} + Q_{TOP} + Q_{DOOR}$$

$$Q = U A (T_a - T_r)$$



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$$U = \frac{1}{X_1 / K_1 + X_2 / K_2 + \dots}$$

Where :

U = Heat Resistance Coefficient Factor

K₁ = Foam Thermal Conductivity

X₁ = Foam Thickness

Note : Due to the short thickness of cabinet out side panel (0.6 mm) and plastic inner liner (1.5 mm) heat resistance of these materials have been considered negligible.

Therefore:

1- $Q_{\text{Side Walls}} = [U A (T_a - T_r)]$

T_a = Ambient Temperature

T_r = Refrigerator air Temperature

$$U = 1 / (0.050 / 0.019) = 0.38 \text{ W/ Sq. Mt. C}$$

$$A = 3.44 \text{ Sq. Mt.}$$

$$T_a = 33 \text{ C}$$

$$T_r = 0 \text{ C}$$

$$Q_{\text{Side Walls}} = 0.38 \times 3.44 \times 33 = 43 \text{ Watts}$$

$$Q_{\text{Side Walls}} = 43 \text{ Watts}$$

2 - $Q_{\text{Back panel}} = [U A (T_a - T_r)]$

T_a = Ambient Temperature

T_r = Refrigerator air Temperature

$$U = 1 / (0.050 / 0.019) = 0.38 \text{ W/ Sq. Mt. C}$$



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$$A = 3.9 \text{ Sq. Mt.}$$

$$T_a = 33 \text{ C}$$

$$T_r = 0 \text{ C}$$

$$Q_{\text{Back panel}} = 0.38 \times 3.9 \times 33 = 49 \text{ Watts}$$

$$Q_{\text{Back panel}} = 49 \text{ Watts}$$

3 - $Q_{\text{Top}} = [U A (T_a - T_r)]$

T_a = Ambient Temperature

T_r = Refrigerator air Temperature

$$U = 1 / (0.050/0.019) = 0.38 \text{ W/ Sq. Mt C}$$

$$A = 1.68 \text{ Sq. Mt.}$$

$$T_a = 45 \text{ C}$$

$$T_r = 0 \text{ C}$$

$$Q_{\text{Top}} = 0.38 \times 1.68 \times 45 = 28.7 \text{ Watts}$$

$$Q_{\text{Top}} = 28.7 \text{ Watts}$$

4 - $Q_{\text{Bottom}} = [U A (T_a - T_r)]$

T_a = Ambient Temperature

T_r = Refrigerator air Temperature

$$U = 1 / (0.050/0.019) = 0.38 \text{ W/Sq. Mt C}$$

$$A = 1.68 \text{ Sq. Mt.}$$

$$T_a = 33 \text{ C}$$

$$T_r = 0 \text{ C}$$

$$Q_{\text{Bottom}} = 0.38 \times 1.68 \times 33 = 21 \text{ Watts}$$

$$Q_{\text{Bottom}} = 21 \text{ Watts}$$

5 - $Q_{\text{Door}} = [U A (T_a - T_r)]$



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According to ASHREA handbook of refrigeration and Standard Mechanical Engineering Handbook published by Mc Graw Hill Book Co. the thermal conductivity for Tempered glass and air are as follows. The refrigerator door cabinet is considered to be 5 mm Double Tempered Glass with 15 mm still air between two side glasses.

- a) Thermal conductivity for Tempered Glass at 366 degree of Kelvin is equal to 1 W/(m. K)
- b) Thermal Conductivity for air at 0 degree centigrade and 1 atmosphere pressure is equal 0.024 W/(m.K)

T_a = Ambient Temperature
 T_r = Refrigerator air Temperature
Glass thickness = 5 mm

$$U = 1 / [(0.005/1) + (0.015/0.024) + (0.005/1)] = 1.57 \text{ W/ Sq. Mt. C}$$

$A = 3.9 \text{ Sq. Mt.}$
 $T_a = 33 \text{ C}$
 $T_r = 0 \text{ C}$

$$Q_{\text{Door}} = 1.57 \times 3.9 \times 33 = 202.7 \text{ Watts}$$

$$Q_{\text{Door}} = 202.7 \text{ Watts}$$

Total Heat Leaks;

$$Q_{\text{TL}} = 43 + 49 + 28.7 + 21 + 202.7 = 209.9$$

$$Q_{\text{Total Heat Leaks}} = 344 \text{ Watts}$$



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b) Product Loads;

$$Q = m C (T_{im} - T_{fm})$$

Where;

Q = Total heat removed from fresh lamb meat. Per 24 hr

M = Total mass of lamb meat = 200 Kg.

C = 0.76 Kcal/24hrs. Kg. C

T_{im} = Lamb meat initial temperature = 24 C

T_{fm} = Lamb meat final temperature = 0 C

$$Q = 200 \times 0.76 \times (24-0) = 3648 \text{ Kcal per 24 hrs}$$

The total compressor operating time is considered 20 hours per day equals;

$$Q = 3648 / 20 = 182.4 \text{ Kcal/hr} = 212 \text{ Watt}$$

$$Q_{\text{Misc}} = Q_{\text{Electricity}} + Q_{\text{Infiltration}} + Q_{\text{Door Opening}} + Q_{\text{Etc.}}$$

According to our experience we can consider 20% of total heat removed from products and heat leaks through refrigerator walls for Q_{misc}. therefore Total Cooling Capacity required is calculated as follows;

$$Q_{\text{Misc}} = 20\% (Q_{\text{Heat Leak}} + Q_{\text{Product Load}}) = 20\% \times (344 + 212) = 111.2$$



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$$Q_{\text{Grand Total}} = Q_{\text{Heat Leaks}} + Q_{\text{Product Loads}} + Q_{\text{Misc.}}$$

$$Q_{\text{Grand Total}} = 344 + 212 + 111 = 634 \text{ Watts}$$

$$Q_{\text{Grand Total}} = 667 \text{ Watts}$$

The Suitable Compressor selected for this Model is Danfoss SC12/12G with a total 680 watts cooling capacity, or Compressor Model Danfoss SC15/15G with a total 870 watts cooling capacity at - 15 C evaporating temperature at CECOMAF test condition, (55 degree condensing and liquid temperature)



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Transmission load calculation
Upright Refrigerator Show Case
Model ELS-1982

Dimensions

	Dimension Cm.	Area (sq. mt.)	Insulation Thickness
Side Walls	2x(82 x 200)	3.28	50
Back Panel	200 x 200	4	50
Top Roof	200 x 82	1.64	50
Bottom Floor	200 x 82	1.68	50
Door	200 x 200	4	50

Insulation Type: Pu Foam

Cyclopentane Foam Thermal Conductivity: 0.019 W /mt.C

Temperature Difference: (ΔT) = 33 - 10 = 33 C

Ambient Temperature = 43 C

Refrigerator Air Temperature = 0 C

The refrigerator is designed to be used for keeping fresh lamb containing 67 % of lean. The initial fresh lamb temperature is considered to be 24 C. the specific heat of lamb containing 67% lean is 0.76 Kcal/hr kg. C.

Calculation :

$$Q_{TL} = Q_{SW} + Q_{BP} + Q_{BOTTOM} + Q_{TOP} + Q_{DOOR}$$

$$Q = U A (T_a - T_r)$$



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$$U = \frac{1}{X_1 / K_1 + X_2 / K_2 + \dots}$$

Where :

U = Heat Resistance Coefficient Factor

K₁ = Foam Thermal Conductivity

X₁ = Foam Thickness

Note : Due to the short thickness of cabinet out side panel (0.6 mm) and plastic inner liner (1.5 mm) heat resistance of these materials have been considered negligible.

Therefore:

1- $Q_{\text{Side Walls}} = [U A (T_a - T_r)]$

T_a = Ambient Temperature

T_r = Refrigerator air Temperature

$$U = 1 / (0.050 / 0.019) = 0.38 \text{ W/ Sq. Mt. C}$$

$$A = 3.28 \text{ Sq. Mt.}$$

$$T_a = 33 \text{ C}$$

$$T_r = 10 \text{ C}$$

$$Q_{\text{Side Walls}} = 0.38 \times 3.28 \times 23 = 26.7 \text{ Watts}$$

$$Q_{\text{Side Walls}} = 26.7 \text{ Watts}$$

2 - $Q_{\text{Back panel}} = [U A (T_a - T_r)]$

T_a = Ambient Temperature

T_r = Refrigerator air Temperature

$$U = 1 / (0.050 / 0.019) = 0.38 \text{ W/ Sq. Mt. C}$$



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$$A = 4 \text{ Sq. Mt.}$$

$$T_a = 33 \text{ C}$$

$$T_r = 10 \text{ C}$$

$$Q_{\text{Back panel}} = 0.38 \times 4 \times 23 = 49 \text{ Watts}$$

$$Q_{\text{Back panel}} = 35 \text{ Watts}$$

3 - $Q_{\text{Top}} = [U A (T_a - T_r)]$

T_a = Ambient Temperature

T_r = Refrigerator air Temperature

$$U = 1 / (0.050/0.019) = 0.38 \text{ W/ Sq. Mt C}$$

$$A = 1.64 \text{ Sq. Mt.}$$

$$T_a = 45 \text{ C}$$

$$T_r = 10 \text{ C}$$

$$Q_{\text{Top}} = 0.38 \times 1.64 \times 35 = 22 \text{ Watts}$$

$$Q_{\text{Top}} = 22 \text{ Watts}$$

4 - $Q_{\text{Bottom}} = [U A (T_a - T_r)]$

T_a = Ambient Temperature

T_r = Refrigerator air Temperature

$$U = 1 / (0.050/0.019) = 0.38 \text{ W/Sq. Mt C}$$

$$A = 1.64 \text{ Sq. Mt.}$$

$$T_a = 33 \text{ C}$$

$$T_r = 10 \text{ C}$$

$$Q_{\text{Bottom}} = 0.38 \times 1.64 \times 23 = 14 \text{ Watts}$$

$$Q_{\text{Bottom}} = 14 \text{ Watts}$$

5 - $Q_{\text{Door}} = [U A (T_a - T_r)]$



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- c) Thermal conductivity for Tempered Glass at 366 degree of Kelvin is equal to 1 W/(m. K)
- d) Thermal Conductivity for air at 0 degree centigrade and 1 atmosphere pressure is equal 0.024 W/(m.K)

T_a = Ambient Temperature
 T_r = Refrigerator air Temperature
Glass thickness = 5 mm

$$U = 1 / [(0.005/1) + (0.015/0.024) + (0.005/1)] = 1.57 \text{ W/ Sq. Mt. C}$$

$A = 4 \text{ Sq. Mt.}$
 $T_a = 33 \text{ C}$
 $T_r = 10 \text{ C}$

$$Q_{\text{Door}} = 1.57 \times 4 \times 23 = 144 \text{ Watts}$$

$$Q_{\text{Door}} = 144 \text{ Watts}$$

Total Heat Leaks;

$$Q_{\text{TL}} = 41 + 35 + 22 + 14 + 144 = 256$$

$$Q_{\text{Total Heat Leaks}} = 256 \text{ Watts}$$



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b) Product Loads;

$$Q = m C (T_{im} - T_{fm})$$

Where;

Q = Total heat removed from mixed fresh fruit and vegetables. Per 24 hr

M = Total mass of fresh fruit and vegetables = 100 Kg.

C_{Lettuce} = 4.060x 0.23885 = 0.969 Kcal/24hrs. Kg. C

C_{Spinach} = 4.000x 0.23885 = 0.955 Kcal/24hrs. Kg. C

C_{Radish} = 4.060x 0.23885 = 0.969 Kcal/24hrs. Kg. C

C_{Apple} = 3.78x 0.23885 = 0.903 Kcal/24hrs. Kg. C

C_{Cherries} = 3.78x 0.23885 = 0.903 Kcal/24hrs. Kg. C

C_{Average} = (0.969+0.955+0.969+0.903+0.903)/5 = 0.934 Kcal/24hrs. Kg. C

T_{iv} = Fresh Fruit and Vegetable initial temperature = 25 C

T_{fv} = Fresh Fruit and Vegetable final temperature = 10 C

Q = 300 x 0.934 x (25-10) = 4903 Kcal per 24 hrs

The total compressor operating time is considered 20 hours per day equals;



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$$Q = 4903 / 20 = 245. \text{ Kcal/hr} = 285 \text{ Watt}$$

$$Q_{\text{Misc}} = Q_{\text{Electricity}} + Q_{\text{Infiltration}} + Q_{\text{Door Opening}} + Q_{\text{Etc.}}$$

According to our experience we can consider 20% of total heat removed from products and heat leaks through refrigerator walls for Q_{misc} . therefore Total Cooling Capacity required is calculated as follows;

$$Q_{\text{Misc}} = 20\% (Q_{\text{Heat Leak}} + Q_{\text{Product Load}}) = 20\% \times (256 + 285) = 108$$

$$Q_{\text{Grand Total}} = Q_{\text{Heat Leaks}} + Q_{\text{Product Loads}} + Q_{\text{Misc.}}$$

$$Q_{\text{Grand Total}} = 256 + 285 + 108 = 649 \text{ Watts}$$

$$Q_{\text{Grand Total}} = 649 \text{ Watts}$$

The Suitable Compressor selected for this Model I Danfoss SC21G with a total 680 watts cooling capacity at -10°C evaporating temperature at CECOMAF test condition, (55 degree condensing and liquid temperature)

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Show Case Model ELS-260-3F **Heat Load Calculation**

Heat Load

Total heat load consists of the amount of heat to be removed from cabinet during a certain period. This is dependent on two main factors:

- 1- Heat Leakage Load
- 2- Heat usage of Product load.

The heat leakage load or heat transfer load is total amount of heat that leaks through the walls, windows, ceiling, and floor of the cabinet.

The Heat usage or product load, is the sum of the heat load of cooling the contents to cabinet temperature, cooling of air changes, removing respiration heat from fresh or live vegetables, meat, removing heat released by electric lights, electric motors and the like, and heat given off by people entering and/or working in the cabinet.

Heat Leakage Variables

There are five factors, which affect heat leakage.

- 1- Time. The longer the period of time, the more heat will leak through a certain wall. The standard time unit used for computation is the 24 hour period in refrigeration situation.
- 2- Temperature difference. The temperature difference is an important factor in heat leakage into the cabinet.
- 3- Insulation Thickness. The thicker the insulation, the less heat flow through it.
- 4- Kind of Insulation. The kind of insulation or the material used, is an important consideration in the construction of the cabinet.



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5- Area of the Cabinet. Just as size of a pipe determination how much water will flow through it. So the more the area through which heat may leak the greater the heat flow will be. The common unit used for determining the heat flow is square meter of area.

Determination Usage Load, Product Load

The total heat load of refrigerator cabinet, in addition to being dependent upon the heat leaking through walls and windows is also affected by the heat to be removed from articles in the cabinet, the air change and other sources of heat. This heat is called heat usage, or product load, and it is caused by changes of air in the cabinet, by products to be cooled, by lights and motors which may be used inside the box, and by the occupancy of the box.

Refrigerator equipment manufacturers have developed a standard where by you may obtain a fairly accurate estimate of the usage heat load. The method is as follows;

The cabinet is classified as the type of service to be performed. Under this classification come florist's cabinets, grocery boxes, normal market coolers, fresh meat cabinets, and restaurant and short order boxes. This load depends in detail upon the following basic factors:

- 1- Temperature difference between exterior and interior of cabinet.
- 2- Volume of cabinet (internal).
- 3- Type of Service.
- 4- Time.

It is impossible to calculate, using a typical installation and determine the amount of food put into the refrigerator, how many times the door is opened, and for how long a period of time the employees are inside the cabinet. This is a laborious process and, unless carefully performed, discrepancies are bound to appear in the results.

The usual procedure to determine the usage heat load is as follows:

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- 1- The temperature difference is the same value as that used for heat leakage into the cabinet.
- 2- Volume of the cabinet is computed from inside diameters.
- 3- Next determine under what type service the cabinet is being used, such as average, heavy or long storage. The service load is also dependent on cabinet size, the smaller the cabinet the more heat load is caused by products. A meat market, for instance, may be one in either a small neighborhood store or it may be in a supermarket.
- 4- Time, 24 hour. There would, of course, be considered variation in the amount of heat to be removed from the cabinet content. From the table we can find the load for three conditions.

After the total volume of the box has been calculated, the load for each cubic foot is determined by referring to the table, if the cabinet appears to have average product with a temperature difference of 60 F., and has a volume of 300 cu. ft. the amount of heat to be removed from each cubic feet will be 78 BTU per 24 hours. We multiply this value by the total volume in cubic feet, and a fairly accurate estimate of the product load may be obtained. The table gives the heat usage over a period of 24 hours as this time is the established standard.

$$\text{Heat Usage} = \text{Usage BTU} \times \text{Volume in cu. ft.}$$

Through our knowledge, experience and facts and figures we believe that calculation of product loads for each individual model depends upon many factors that we could mention briefly as follows:

- Product design;
- Product style;
- Company policy;
- Useful internal volume;
- Type of evaporator;
- Type of cellar compartment;
- Freezer volume;
- Culture of customer;
- Country of origin and etc.;



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Transmission load calculation

Show Case Model ELS-2603F

Dimensions

	Dimension Cm.	Area (sq. mt.)	Insulation Thickness mm
Side Walls	2x(144x90)	2.592	50
Upper Front Panel	72x260	1.872	15 (glass and air)
Lower Front Panel	72x260	1.872	50
Back Panel	144x260	3.744	50
Top Panel	80x260	2.080	50
Bottom Panel	90x260	2.340	50

Insulation Type: Pu Foam

Cyclopentane Foam Thermal Conductivity: 0.019 W /mt.C

Temperature Difference: $(\Delta T) = 43 - 10 = 33 \text{ C}$

Ambient Temperature = 33 C

Refrigerator Air Temperature = 10 C

The refrigerator is designed to be used for keeping fresh lamb containing 67 % of lean. The initial fresh lamb temperature is considered to be 24 C. the specific heat of lamb containing 67% lean is 0.76 Kcal/hr kg. C.

Calculation :

$$Q_{TL} = Q_{SW} + Q_{BP} + Q_{Bottom} + Q_{Top} + Q_{Upper Door} + Q_{Lower front Panel}$$

$$Q = U A (T_a - T_r)$$

$$U = \frac{1}{X_1 / K_1 + X_2 / K_2 + \dots}$$

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Where :

U = Heat Resistance Coefficient Factor

K₁ = Foam Thermal Conductivity

X₁ = Foam Thickness

Note : Due to the short thickness of cabinet out side panel (0.6 mm) and plastic inner liner (1.5 mm) heat resistance of these materials have been considered negligible.

Therefore:

1- $Q_{\text{Side Walls}} = [U A (T_a - T_r)]$

T_a = Ambient Temperature

T_r = Refrigerator air Temperature

$$U = 1 / (0.050/0.019) = 0.38 \text{ W/ Sq. Mt. C}$$

$$A = 2.592 \text{ Sq. Mt.}$$

$$T_a = 33 \text{ C}$$

$$T_r = 10 \text{ C}$$

$$Q_{\text{Side Walls}} = 0.38 \times 2.592 \times 23 = 22.6 \text{ Watts}$$

$$Q_{\text{Side Walls}} = 22.6 \text{ Watts}$$

2 - $Q_{\text{Back panel}} = [U A (T_a - T_r)]$

T_a = Ambient Temperature

T_r = Refrigerator air Temperature

$$U = 1 / (0.050 / 0.019) = 0.38 \text{ W/ Sq. Mt. C}$$

$$A = 3.744 \text{ Sq. Mt.}$$

$$T_a = 33 \text{ C}$$

$$T_r = 10 \text{ C}$$



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$$Q_{\text{Back panel}} = 0.38 \times 3.744 \times 23 = 32.7 \text{ Watts}$$

$$Q_{\text{Back panel}} = 32.7 \text{ Watts}$$

3 - $Q_{\text{Top}} = [U A (T_a - T_r)]$

T_a = Ambient Temperature

T_r = Refrigerator air Temperature

$$U = 1 / (0.050/0.019) = 0.38 \text{ W/ Sq. Mt C}$$

$$A = 2.080 \text{ Sq. Mt.}$$

$$T_a = 33 \text{ C}$$

$$T_r = 10 \text{ C}$$

$$Q_{\text{Top}} = 0.38 \times 2.080 \times 23 = 18.2 \text{ Watts}$$

$$Q_{\text{Top}} = 18.2 \text{ Watts}$$

4 - $Q_{\text{Bottom}} = [U A (T_a - T_r)]$

T_a = Ambient Temperature

T_r = Refrigerator air Temperature

$$U = 1 / (0.050/0.019) = 0.38 \text{ W/Sq. Mt C}$$

$$A = 2.340 \text{ Sq. Mt.}$$

$$T_a = 45 \text{ C}$$

$$T_r = 10 \text{ C}$$

$$Q_{\text{Bottom}} = 0.38 \times 2.340 \times 35 = 31.1 \text{ Watts}$$

$$Q_{\text{Bottom}} = 31.1 \text{ Watts}$$

5 - $Q_{\text{Lower Front Panel}} = [U A (T_a - T_r)]$

T_a = Ambient Temperature

T_r = Refrigerator air Temperature



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$$U = 1 / (0.050/0.019) = 0.38 \text{ W/Sq. Mt C}$$

$$A = 1.872 \text{ Sq. Mt.}$$

$$T_a = 33 \text{ C}$$

$$T_r = 10 \text{ C}$$

$$Q_{\text{Lower Front Panel}} = 0.38 \times 1.872 \times 23 = 16.4 \text{ Watts}$$

$$Q_{\text{Bottom}} = 16.4 \text{ Watts}$$

6 - $Q_{\text{Upper Door}} = [U A (T_a - T_r)]$

According to ASHREA handbook of refrigeration and Standard Mechanical Engineering Handbook published by Mc Graw Hill Book Co. the thermal conductivity for Tempered glass and air are as follows. The refrigerator door cabinet is considered to be 5 mm Double Tempered Glass with 15 mm still air between two side glasses.

e) Thermal conductivity for Tempered Glass at 366 degree of Kelvin is equal to 1 W/(m. K)

f) Thermal Conductivity for air at 0 degree centigrade and 1 atmosphere pressure is equal 0.024 W/(m.K)

T_a = Ambient Temperature

T_r = Refrigerator air Temperature

Glass thickness = 5 mm

$$U = 1 / [(0.005/1) + (0.015/0.024) + (.005/1)] = 1.57 \text{ W/ Sq. Mt. C}$$

$$A = 1.872 \text{ Sq. Mt.}$$

$$T_a = 33 \text{ C}$$

$$T_r = 10 \text{ C}$$

$$Q_{\text{Upper Door}} = 1.57 \times 1.872 \times 23 = 67.6 \text{ Watts}$$

$$Q_{\text{Door}} = 67.6 \text{ Watts}$$

Total Heat Leaks;



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$$Q_{TL} = 22.6 + 32.7 + 18.2 + 31.1 + 16.4 + 67.4 = 189$$

$$Q_{\text{Total Heat Leaks}} = 189 \text{ Watts}$$

b) Product Loads;

$$Q = m C (T_{im} - T_{fm})$$

Where;

Q = Total heat removed from mixed fresh fruit and vegetables. Per 24 hr

M = Total mass of fresh fruit and vegetables = 100 Kg.

C_{Lettuce} = 4.060 x 0.23885 = 0.969 Kcal/24hrs. Kg. C

C_{Spinach} = 4.000 x 0.23885 = 0.955 Kcal/24hrs. Kg. C

C_{Radish} = 4.060 x 0.23885 = 0.969 Kcal/24hrs. Kg. C

C_{Apple} = 3.78 x 0.23885 = 0.903 Kcal/24hrs. Kg. C

C_{Cherries} = 3.78 x 0.23885 = 0.903 Kcal/24hrs. Kg. C

C_{Average} = (0.969 + 0.955 + 0.969 + 0.903 + 0.903) / 5 = 0.934 Kcal/24hrs. Kg. C

T_{iv} = Fresh Fruit and Vegetable initial temperature = 25 C

T_{iv} = Fresh Fruit and Vegetable final temperature = 10 C

Q = 150 x 0.934 x (25 - 10) = 2101 Kcal per 24 hrs

The total compressor operating time is considered 20 hours per day equals;

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$$Q = 2101 / 20 = 105 \text{ Kcal/hr} = 122 \text{ Watt}$$

$$Q_{\text{Misc}} = Q_{\text{Electricity}} + Q_{\text{Infiltration}} + Q_{\text{Door Opening}} + Q_{\text{Etc.}}$$

According to our experience we can consider 20% of total heat removed from products and heat leaks through refrigerator walls for Q_{misc} . therefore Total Cooling Capacity required is calculated as follows;

$$Q_{\text{Misc}} = 20\% (Q_{\text{Heat Leak}} + Q_{\text{Product Load}}) = 20\% \times (189 + 122) = 62.2$$

$$Q_{\text{Grand Total}} = Q_{\text{Heat Leaks}} + Q_{\text{Product Loads}} + Q_{\text{Misc.}}$$

$$Q_{\text{Grand Total}} = 189 + 122 + 62.2 = 320 \text{ Watts}$$

$$Q_{\text{Grand Total}} = 373 \text{ Watts}$$

The Suitable Compressor selected for this Model is FR11G with a total 385 watts cooling capacity at -10°C evaporating temperature at CECOMAF test condition, (55 degree condensing and liquid temperature)



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Transmission load calculation

Show Case Model ELS-220-3F

Dimensions

	Dimension Cm.	Area (sq. mt.)	Insulation Thickness mm
Side Walls	2x(144x90)	2.592	50
Upper Front Panel	72x220	1.584	15 (glass and air)
Lower Front Panel	72x220	1.584	50
Back Panel	144x220	3.168	50
Top Panel	80x220	1.76	50
Bottom Panel	90x220	1.98	50

Insulation Type: Pu Foam

Cyclopentane Foam Thermal Conductivity: 0.019 W /mt.C

Temperature Difference: (ΔT) = 43 - 10 = 33 C

Ambient Temperature = 33 C

Refrigerator Air Temperature = 10 C

The refrigerator is designed to be used for keeping fresh lamb containing 67 % of lean. The initial fresh lamb temperature is considered to be 24 C. the specific heat of lamb containing 67% lean is 0.76 Kcal/hr kg. C.

Calculation :

$$Q_{TL} = Q_{SW} + Q_{BP} + Q_{Bottom} + Q_{Top} + Q_{Upper Door} + Q_{Lower front Panel}$$

$$Q = U A (T_a - T_r)$$



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$$U = \frac{1}{X_1 / K_1 + X_2 / K_2 + \dots}$$

Where :

U = Heat Resistance Coefficient Factor

K₁ = Foam Thermal Conductivity

X₁ = Foam Thickness

Note : Due to the short thickness of cabinet out side panel (0.6 mm) and plastic inner liner (1.5 mm) heat resistance of these materials have been considered negligible.

Therefore:

1- $Q_{\text{Side Walls}} = [U A (T_a - T_r)]$

T_a = Ambient Temperature

T_r = Refrigerator air Temperature

$$U = 1 / (0.050 / 0.019) = 0.38 \text{ W/ Sq. Mt. C}$$

$$A = 2.592 \text{ Sq. Mt.}$$

$$T_a = 33 \text{ C}$$

$$T_r = 10 \text{ C}$$

$$Q_{\text{Side Walls}} = 0.38 \times 2.592 \times 23 = 22.6 \text{ Watts}$$

$$Q_{\text{Side Walls}} = 22.6 \text{ Watts}$$

2 - $Q_{\text{Back panel}} = [U A (T_a - T_r)]$

T_a = Ambient Temperature

T_r = Refrigerator air Temperature

$$U = 1 / (0.050 / 0.019) = 0.38 \text{ W/ Sq. Mt. C}$$



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$$A = 3.168 \text{ Sq. Mt.}$$

$$T_a = 33 \text{ C}$$

$$T_r = 10 \text{ C}$$

$$Q_{\text{Back panel}} = 0.38 \times 3.168 \times 23 = 27.7 \text{ Watts}$$

$$Q_{\text{Back panel}} = 27.7 \text{ Watts}$$

3 - $Q_{\text{Top}} = [U A (T_a - T_r)]$

T_a = Ambient Temperature

T_r = Refrigerator air Temperature

$$U = 1 / (0.050/0.019) = 0.38 \text{ W/ Sq. Mt C}$$

$$A = 1.76 \text{ Sq. Mt.}$$

$$T_a = 33 \text{ C}$$

$$T_r = 10 \text{ C}$$

$$Q_{\text{Top}} = 0.38 \times 1.76 \times 23 = 15.4 \text{ Watts}$$

$$Q_{\text{Top}} = 15.4 \text{ Watts}$$

4 - $Q_{\text{Bottom}} = [U A (T_a - T_r)]$

T_a = Ambient Temperature

T_r = Refrigerator air Temperature

$$U = 1 / (0.050/0.019) = 0.38 \text{ W/Sq. Mt C}$$

$$A = 1.98 \text{ Sq. Mt.}$$

$$T_a = 45 \text{ C}$$

$$T_r = 10 \text{ C}$$

$$Q_{\text{Bottom}} = 0.38 \times 1.98 \times 35 = 26.3 \text{ Watts}$$

$$Q_{\text{Bottom}} = 26.3 \text{ Watts}$$

5 - $Q_{\text{Lower Front Panel}} = [U A (T_a - T_r)]$



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T_a = Ambient Temperature

T_r = Refrigerator air Temperature

$$U = 1 / (0.050 / 0.019) = 0.38 \text{ W/Sq. Mt. C}$$

$$A = 1.584 \text{ Sq. Mt.}$$

$$T_a = 33 \text{ C}$$

$$T_r = 10 \text{ C}$$

$$Q_{\text{Lower Front Panel}} = 0.38 \times 1.584 \times 23 = 13.8 \text{ Watts}$$

$$Q_{\text{Lower Front Panel}} = 16.8 \text{ Watts}$$

6 - $Q_{\text{Upper Front Door}} = [U A (T_a - T_r)]$

According to ASHREA handbook of refrigeration and Standard Mechanical Engineering Handbook published by Mc Graw Hill Book Co. the thermal conductivity for Tempered glass and air are as follows. The refrigerator door cabinet is considered to be 5 mm Double Tempered Glass with 15 mm still air between two side glasses.

g) Thermal conductivity for Tempered Glass at 366 degree of Kelvin is equal to 1 W/(m. K)

h) Thermal Conductivity for air at 0 degree centigrade and 1 atmosphere pressure is equal 0.024 W/(m.K)

T_a = Ambient Temperature

T_r = Refrigerator air Temperature

Glass thickness = 5 mm

$$U = 1 / [(0.005/1) + (0.015/0.008) + (0.005/1)] = 1.57 \text{ W/ Sq. Mt. C}$$

$$A = 1.584 \text{ Sq. Mt.}$$

$$T_a = 33 \text{ C}$$

$$T_r = 10 \text{ C}$$

$$Q_{\text{Upper Door}} = 1.57 \times 1.584 \times 23 = 57.2 \text{ Watts}$$

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$$Q_{\text{Door}} = 57.2 \text{ Watts}$$

Total Heat Leaks;

$$Q_{\text{TL}} = 22.6 + 13.8 + 57.2 + 27.7 + 15.4 + 26.3 = 163$$

$$Q_{\text{Total Heat Leaks}} = 163 \text{ Watts}$$

b) Product Loads;

$$Q = m C (T_{\text{im}} - T_{\text{fm}})$$

Where;

Q = Total heat removed from mixed fresh fruit and vegetables. Per 24 hr

M = Total mass of fresh fruit and vegetables = 100 Kg.

C_{Lettuce} = 4.060 x 0.23885 = 0.969 Kcal/24hrs. Kg. C

C_{Spinach} = 4.000 x 0.23885 = 0.955 Kcal/24hrs. Kg. C

C_{Radish} = 4.060 x 0.23885 = 0.969 Kcal/24hrs. Kg. C

C_{Apple} = 3.78 x 0.23885 = 0.903 Kcal/24hrs. Kg. C

C_{Cherries} = 3.78 x 0.23885 = 0.903 Kcal/24hrs. Kg. C

C_{Average} = (0.969 + 0.955 + 0.969 + 0.903 + 0.903) / 5 = 0.934 Kcal/24hrs. Kg. C

T_{iv} = Fresh Fruit and Vegetable initial temperature = 25 C

T_{fv} = Fresh Fruit and Vegetable final temperature = 10 C

$$Q = 100 \times 0.934 \times (25 - 10) = 1398 \text{ Kcal per 24 hrs}$$



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The total compressor operating time is considered 20 hours per day equals;

$$Q = 1868 / 20 = 93.4 \text{ Kcal/hr} = 111.4 \text{ Watt}$$

$$Q_{\text{Misc}} = Q_{\text{Electricity}} + Q_{\text{Infiltration}} + Q_{\text{Door Opening}} + Q_{\text{Etc.}}$$

According to our experience we can consider 20% of total heat removed from products and heat leaks through refrigerator walls for Q_{misc} . therefore Total Cooling Capacity required is calculated as follows;

$$Q_{\text{Misc}} = 20\% (Q_{\text{Heat Leak}} + Q_{\text{Product Load}}) = 20\% \times (163 + 111.4) = 55$$

$$Q_{\text{Grand Total}} = Q_{\text{Heat Leaks}} + Q_{\text{Product Loads}} + Q_{\text{Misc.}}$$

$$Q_{\text{Grand Total}} = 163 + 111.4 + 55 = 329 \text{ Watts}$$

$$Q_{\text{Grand Total}} = 329 \text{ Watts}$$

The Suitable Compressor selected for this Model is Danfoss FR10G with a total 320 watts cooling capacity, at -10 C evaporating temperature at CECOMAF test condition, (55 degree condensing and liquid temperature)



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Transmission load calculation

Show Case Model ELS-1219F

Dimensions

	Dimension Cm.	Area (sq. mt.)	Insulation Thickness mm
Side Walls	2x(137x84)	2.3	50
Glass Front Panel	193x111	2.14	(Double Glass and 15 mm Still Air)
Lower Front Panel	193x80	1.54	50
Back Panel	193x137	2.64	50
Bottom Panel	193x84	1.62	50

Insulation Type: Pu Foam

Cyclopentane Foam Thermal Conductivity: 0.019 W /mt.C

Temperature Difference: $(\Delta T) = 43 - 10 = 33 \text{ C}$

Ambient Temperature = 33 C

Refrigerator Air Temperature = 10 C

The refrigerator is designed to be used for keeping fresh lamb containing 67 % of lean. The initial fresh lamb temperature is considered to be 24 C. the specific heat of lamb containing 67% lean is 0.76 Kcal/hr kg. C.

Calculation :

$$Q_{TL} = Q_{SW} + Q_{BP} + Q_{Bottom} + Q_{Top} + Q_{Upper Door} + Q_{Lower front Panel}$$

$$Q = U A (T_a - T_r)$$

$$U = \frac{1}{X_1 / K_1 + X_2 / K_2 + \dots}$$



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Where :

U = Heat Resistance Coefficient Factor

K₁ = Foam Thermal Conductivity

X₁ = Foam Thickness

Note : Due to the short thickness of cabinet out side panel (0.6 mm) and plastic inner liner (1.5 mm) heat resistance of these materials have been considered negligible.

Therefore:

1- $Q_{\text{Side Walls}} = [U A (T_a - T_r)]$

T_a = Ambient Temperature

T_r = Refrigerator air Temperature

$$U = 1 / (0.050/0.019) = 0.38 \text{ W/ Sq. Mt. C}$$

$$A = 2.3 \text{ Sq. Mt.}$$

$$T_a = 33 \text{ C}$$

$$T_r = 10 \text{ C}$$

$$Q_{\text{Side Walls}} = 0.38 \times 2.3 \times 23 = 20.1 \text{ Watts}$$

$$Q_{\text{Side Walls}} = 20.1 \text{ Watts}$$

2 - $Q_{\text{Back panel}} = [U A (T_a - T_r)]$

T_a = Ambient Temperature

T_r = Refrigerator air Temperature

$$U = 1 / (0.050 / 0.019) = 0.38 \text{ W/ Sq. Mt. C}$$

$$A_{\text{Heat transferable area}} = 2.64 / 2 = 1.32 \text{ Sq. Mt.}$$

$$T_a = 33 \text{ C}$$

$$T_r = 10 \text{ C}$$



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$$Q_{\text{Back panel}} = 0.38 \times 1.3 \times 23 = 11.4 \text{ Watts}$$

$$Q_{\text{Back panel}} = 11.4 \text{ Watts}$$

3 - $Q_{\text{Front Site Glass and Top}} = [U A (T_a - T_r)]$

According to ASHREA handbook of refrigeration and Standard Mechanical Engineering Handbook published by Mc Graw Hill Book Co. the thermal conductivity for Tempered glass and air are as follows. The refrigerator door cabinet is considered to be 5 mm Double Tempered Glass with 15 mm still air between two side glasses.

- i) Thermal conductivity for Tempered Glass at 366 degree of Kelvin is equal to 1 W/(m. K)
- j) Thermal Conductivity for air at 0 degree centigrade and 1 atmosphere pressure is equal 0.024 W/(m.K)

T_a = Ambient Temperature

T_r = Refrigerator air Temperature

Glass thickness = 5 mm

$$U = 1 / [(0.005/1) + (0.015/0.024) + (.005/1)] = 1.57 \text{ W/ Sq. Mt. C}$$

$$A = 2.14 \text{ Sq. Mt.}$$

$$T_a = 33 \text{ C}$$

$$T_r = 10 \text{ C}$$

$$Q_{\text{Front Site Glass and Top}} = 1.57 \times 2.14 \times 23 = 77.3 \text{ Watts}$$

$$Q_{\text{Top}} = 77.3 \text{ Watts}$$

4 - $Q_{\text{Bottom}} = [U A (T_a - T_r)]$

T_a = Ambient Temperature

T_r = Refrigerator air Temperature



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$$U = 1 / (0.050 / 0.019) = 0.38 \text{ W/Sq. Mt C}$$

$$A = 1.62 \text{ Sq. Mt.}$$

$$T_a = 45 \text{ C}$$

$$T_f = 10 \text{ C}$$

$$Q_{\text{Bottom}} = 0.38 \times 1.62 \times 35 = 21.5 \text{ Watts}$$

$$Q_{\text{Bottom}} = 21.5 \text{ Watts}$$

Total Heat Leaks;

$$Q_{\text{TL}} = 20.1 + 77.3 + 11.4 + 32.7 + 21.5 = 144.5$$

$$Q_{\text{Total Heat Leaks}} = 130 \text{ Watts}$$

b) Product Loads;

$$Q = m C (T_{im} - T_{fm})$$

Where;

Q_{TP} = Total heat removed from mixed fresh fruit, vegetables and fresh lamb meet. Per 24 hr

M_1 = Total mass of fresh fruit and vegetables = 150 Kg. Fruit and Vegetables

$$C_{\text{Lettuce}} = 4.060 \times 0.23885 = 0.969 \text{ Kcal/24hrs. Kg. C}$$

$$C_{\text{Spinach}} = 4.000 \times 0.23885 = 0.955 \text{ Kcal/24hrs. Kg. C}$$

$$C_{\text{Radish}} = 4.060 \times 0.23885 = 0.969 \text{ Kcal/24hrs. Kg. C}$$

$$C_{\text{Apple}} = 3.78 \times 0.23885 = 0.903 \text{ Kcal/24hrs. Kg. C}$$



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$$C_{\text{Cherries}} = 3.78 \times 0.23885 = 0.903 \text{ Kcal/24hrs. Kg. C}$$

$$C_{\text{Average}} = (0.969 + 0.955 + 0.969 + 0.903 + 0.903) / 5 = 0.934 \text{ Kcal/24hrs. Kg. C}$$

$$T_{iv} = \text{Fresh Fruit and Vegetable initial temperature} = 25 \text{ C}$$

$$T_{fv} = \text{Fresh Fruit and Vegetable final temperature} = 10 \text{ C}$$

$$Q_{\text{Fresh Fruit and Vegetables}} = 150 \times 0.934 \times (25 - 10) = 2101 \text{ Kcal per 24 hrs}$$

$$M_z = \text{Total mass of lamb meat} = 100 \text{ Kg.}$$

$$C = 0.76 \text{ Kcal/24hrs. Kg. C}$$

$$T_{im} = \text{Lamb meat initial temperature} = 24 \text{ C}$$

$$T_{fm} = \text{Lamb meat final temperature} = 10 \text{ C}$$

$$Q_{\text{Fresh Lamb Meat}} = 150 \times 0.76 \times (28 - 10) = 2052 \text{ Kcal per 24 hrs}$$

$$Q_{\text{Total Product}} = Q_{\text{Fresh Lamb Meat}} + Q_{\text{Fresh Fruit and Vegetables}} = 2052 + 2101 = 3697$$

The total compressor operating time is considered 20 hours per day equals:

$$Q_{\text{Total Product}} = 4062 / 16 = 253 \text{ Kcal/hr} = 295 \text{ Watt}$$

$$Q_{\text{Misc}} = Q_{\text{Electricity}} + Q_{\text{Infiltration}} + Q_{\text{Door Opening}} + Q_{\text{Etc.}}$$

According to our experience we can consider 40% of total heat removed from products and heat leaks through refrigerator walls for Q_{misc} for this type of product. therefore



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Total Cooling Capacity required is calculated as follows;

$$Q_{\text{Misc}} = 30\% (Q_{\text{Heat Leak}} + Q_{\text{Product Load}}) = 40\% \times (130 + 295) = 127$$

$$Q_{\text{Grand Total}} = Q_{\text{Heat Leaks}} + Q_{\text{Product Loads}} + Q_{\text{Misc.}}$$

$$Q_{\text{Grand Total}} = 130 + 295 + 127 = 552 \text{ Watts}$$

$$Q_{\text{Grand Total}} = 552 \text{ Watts}$$

The Suitable Compressor selected for this Model is Danfoss SC15G with a total 565 watts cooling capacity, or Compressor Model Danfoss SC18G with a total 680 watts cooling capacity at -10°C evaporating temperature at CECOMAF test condition, (55 degree condensing and liquid temperature)



Activities

Training

A course was conducted at the counterpart premises. Ten technical staffs consist of the following personnel participated in the course.

1- Mr. Sbaghian	Technician
2- Mr Ghorbani	Technician
3- Mr. Fallah	Technician
4- Mr. Amid	Technician
5- Mr. Behrooz	Technician
6- Mr. Nezhad Ali	Technician
7- Mansoori	Technician
8- Marvian	Engineer
9- Imani	Engineer
10- Mohamadzadeh	Engineer
11- Miss Barati	Engineer

The main topics of the course are:

- 1- Ozone Layer
- 2- Montreal Protocol
- 3- Implementation of CFC phase out projects by UN agencies
- 4- CFC alternatives
- 5- An Introduction to R134a refrigerants
- 6- Conversion of prototypes into ozone layer friendly refrigerants
- 7- Refrigeration load calculation
- 8- Component selection
- 9- Testing Prototypes
- 10- Trial Tests
- 11- Obtaining data from the test results
- 12- Test results analysis
- 13- Trial production
- 14- Mass production



The Ozone Layer

The following subjects were discussed about ozone layer during the course.

Oxygen can exist in two stable forms. The normal form is gaseous oxygen, sometimes referred to as dioxygen but normally just oxygen or O_2 . Two oxygen atoms form each molecule. It is the gas which makes up about one **fifth** of air and is essential for almost all earthly life.

The other form is ozone. In trioxygen or O_3 , each molecule is made up of 3 oxygen atoms. This form is less stable than O_2 but nevertheless occurs naturally. It is produced when O_2 is split into single, radical O atoms which are unstable and will bond with whatever is available. On some cases this will be an O_2 molecule and so O_3 is re-formed.

This happens most frequently in the upper atmosphere near the equator where the wavelength of the incident sunlight gives the correct amount of energy to split the oxygen molecules. Ozone can also form where the high voltage discharges take place, again because into two (O) radicals. Ozone gives the characteristic smell associated with high voltage switch gear.

Most atmospheric ozone exists in the stratosphere between altitude of 15 km and 50 km. It is thinly dispersed throughout this region. If all the ozone were "squashed" down to normal atmospheric pressure at sea level it would form a layer only 2 to 3 mm thick.

Ozone Layer as a UV Light Filter

The importance of this "layer" is as a filter of ultra- violet (UV) light. When UV light strikes an ozone molecule it breaks to form an O_2 molecule and a radical O with an associated release of energy.



The UV is absorbed in this process. The radical O released may combine with an O₂ molecule to re-form ozone, or with another radical O to form NO₂, or with any other gases which are available in the stratosphere.

Although the ozone is thinly dispersed it stops all the UV of certain frequencies and drastically reduces the infiltration of other frequencies. These frequencies are beneficial to life and are sometimes used for sterilisation. In fact one of the reasons that life on earth first developed in the sea is because there was no atmospheric oxygen levels or ozone to remove the UV light. The planet therefore remained sterile except for the levels in the oceans where the UV could not fully penetrate.

The filtering out of the UV light is crucial to life on our planet. If the existing balance is shifted there will be drastic effects on plant and animal life.

CFC's

The term CFC is short for Chlorofluorocarbon. It refers to a series of compounds developed in the late 1920's which display an unparalleled range of qualities for use as refrigerants.

The molecules are based on either methane, CH₄, or ethane, C₂H₆. One or more of the hydrogen, H atoms is replaced by chlorine, Cl or fluorine, F.

The CFC refrigerants quickly replaced most other refrigerants with the exception of ammonia, NH₃ (R717) in the majority of applications.

- The properties which led to this replacement were:
- Very low toxicity



- Non flammability
- Chemical stability
- Good thermodynamic properties (there is a CFC refrigerant suitable for almost all normal applications)
- Good behaviour with oils
- Comparatively low price and ready availability
- Suitability for use with copper pipework

This combination of properties are also made CFC's the ideal propellants for aerosols and agents for foam blowing. They are also excellent de-greasers and so are used extensively in the electronics industry for component cleaning.

For many years CFCs were thought to be totally environmentally friendly. However one of the properties which makes them good as refrigerants etc. is leading to the current problem of ozone depletion.

Effect of CFC's on Ozone Layer

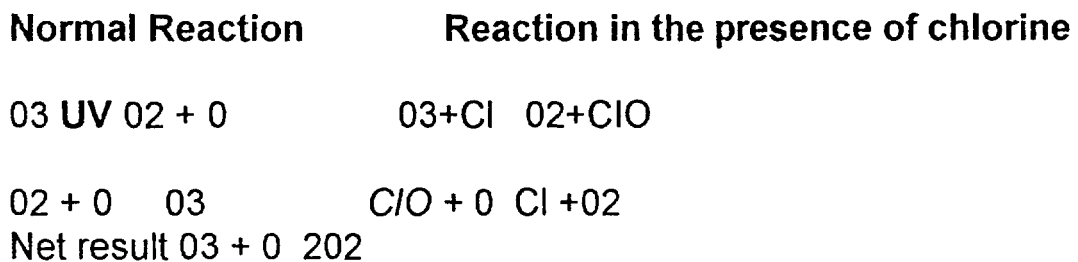
The long term chemical stability of the CFC's means that once they are released they remain in the atmosphere for many years. At low altitudes this is no problem. However, when they reach the upper atmosphere CFC'S, like ozone, are broken down by ultra violet light.

When CFC's are split in this way, it results in the release of free, radical chlorine atoms, which interfere with the normal formation of ozone.



When ozone O₃ is split by ultra violet into gaseous oxygen O₂ and radical oxygen atom O, to form chlorine oxide ClO. The chlorine atom will then only "give up" its oxygen atom to another radical oxygen atom. In this way the presence of chlorine in the upper atmosphere is reducing the number of radical oxygen atoms available for the formation of ozone. Each chlorine atom is freed at the end of this process and so has the potential to stop the normal formation of many thousands of ozone molecules without itself being consumed.

In chemical terms the reactions are as follows:



There has been a measured increase in the amount of free chlorine in the atmosphere from 2 to 3 parts per billion. Scientists now hold the opinion that CFC releases are responsible for the increase in chlorine level, is responsible for ozone depletion.

The Damage Done

There is considerable debate as to how much damage has been done. Measurements of the quantities of stratospheric ozone have only been carried out since the mid 1950's. The instrumentation used has increased in sophistication during this time and so the accuracy of measurement has improved. It is therefore difficult to say what is a normal, natural fluctuation in ozone and what is impending disaster.



There is similar thinning taking place over the Arctic but the weather system behaves slightly differently there and it has not yet been referred as a hole.

Increased Global Warming (The Greenhouse Effect)

CFC's (including HCFC's and others) are also contributing to an increase in global warming. Commonly known as the "Greenhouse Effect". This is a totally separate environmental issue.

Global warming, is a natural phenomenon, the average global temperature is about +15 degrees Celsius supporting life throughout most of the planet. Without **natural** global warming by gases in the atmosphere the average temperature would probably be around - 18 degrees Celsius. Life as we know it would be unable to exist/

The activities of **man** are adding to the natural effect and causing an imbalance. This is sometimes referred to as "ANTHROPOGENIC GLOBAL WARMING"

The atmosphere allows solar radiation through to earth. This is mostly visible light predominantly in the yellow part of the visible spectrum. When this energy strikes the surface of the earth it is absorbed and some of it is re-radiated chiefly in the infra red part of the spectrum. Radiation of this wave length is absorbed and reflected by dense molecules (the so called greenhouse gases) in the lower part of the atmosphere, (troposphere).

This keeps the earth warmer than it otherwise would be. Some of the infra red radiated by the earth escapes 'into space and a natural balance between energy coming in and that going- out to space.

This cycle has been linked to the way a greenhouse works, hence the name "Greenhouse Effect". In fact a greenhouse mostly works



in a different way, by not allowing hot air warmed by the sun to rise and escape.

Anthropogenic Global Warming refers to the extra warming of the earth and atmosphere caused by gases released by man and his activities. If the average temperature of the world is increased by only a few degrees there will be drastic changes in climate and a rise in sea levels.

CFC's and related chemicals are only part of this problem the main greenhouse gases are carbon dioxide (CO₂), CFC's methane, nitrous oxide. They are each thought to contribute to the problem by the following percentages:

Carbon Dioxide	55%
CFC's 11 & 12	17%
Other CFC's	7%
Methane	15%
Nitrous Oxide	7%

The quantities of CFC's released are small compared to other greenhouse gases such as carbon dioxide, CO₂ from power stations. However each Kg of CFC's has many thousands of times the effect of each Kg Of CO₂ SO in consideration of the environmental impact of CFC's this issue should also be addressed.

Ozone Depletion and "Greenhouse" Effect

If there is an increase in the amount of UV light reaching earth there may be an increase in skin cancers. "United states Environmental Protection Agency has calculated that for each 1% decrease in the concentration of ozone in the stratosphere there will be a 5% increase in non-malignant skin cancers each year.

UV light is also one of the major causes of cataracts of the eye; any increase in UV at ground level may increase the occurrence of this



debilitating problem, not only humans, but also domestic, farm and wild animals.

Plants are very sensitive to the amount and quality of sunshine which they receive. An appreciable increase in the UV portion of sunlight could devastate plant life. Most at risk will be the hybrid, high yield varieties of wheat, soya beans etc. Which form a substantial part of mans diet.

It has also been established that changes in UV radiation levels have a serious effect on **Antarctic phytoplankton**. Any great reduction in the numbers of these single cell plants is effectively knocking out the first link in the food chain for all the southern oceans with inevitable consequences for half the worlds marine life.

If depletion of the ozone layer continues then **food production** is bound to be affected and world famine could be the end result, along with the medical problems mentioned earlier.

If there is no reduction in the emissions of greenhouse gases there is likely to be drastic changes in **weather patterns**.

Some effects of this are likely to be:

Increases in **sea level** due to thermal expansion and melting glaciers (not polar ice caps). Predictions of how much rise there will be vary widely but in every case huge areas now marginal, unproductive.

Increased frequency of **hot weather** will put great stress on crops and livestock.

If the changes are rapid, species of plants and animals may be **unable to migrate** fast enough to tolerable climate from their present ecosystems and so perish.



Forests may also be unable to adopt quickly enough.

Water **availability** and quality could be affected by quite small shifts in climate. No one really knows how rainfall patterns will change.

Changing **sea temperatures** may cause redistribution of fish and shifts in major currents.

More heat waves would cause more related deaths and possibly encourage the **spread of diseases** such as malaria.

The predictions widely. Like the ozone issue, no one really knows the most likely outcome, as **man has never played with such a large chemistry set before.**

Both ozone depletion and the greenhouse effect may have dire consequences for the planet. Even if the worst predictions prove incorrect we should surely make an effort to avert even the possibility of global catastrophe.

The Montreal Protocol,

FLUOROCARBON CONTROL AND ALTERNATIVES

Once the effects of fluorocarbon gases on the ozone layer were fully appreciated, negotiations were carried out under the supervision of the United National Environment Programme (UNEP) and the world Meteorological Organisation to limit their production.

Leading industrialised countries agreed on a CFC production control protocol. This agreement has become known as the Montreal Protocol. It was originally signed by 27 countries in September 1987 in Montreal. The protocol was then revised and agreed by about 60 countries at the London meeting in June 1990. By February 1991,



70 countries had signed the protocol, with China expected to sign later that year. The next revision of the Protocol took place in November 1992 in Copenhagen. Currently, nearly 150 countries have signed the protocol.

The main features of the protocol as updated in November 1992 were:

CFC's

1. By 1st January 1994 production should be cut by 75% of 1986 levels.
2. By January 1996 production phased out.

The main controlled substances are CFCs 11, 12, 113, 114, 115. This also effect blends such as R502 (being a mixture of CFC 115 and HFC 22). Solvents and fire extinguishing agents were also controlled.

Trade restrictions on controlled substances, for example import and export bans to and from non-signatory countries, were also agreed. Other trade restrictions were agreed in principle.

HCFC's

Schedule of phase out:

- 35% by 2004
- 65% by 2010
- 90% by 2015
- 99.5 % by 2020
- 100% by 2030

Agreements were also made on their application, selection, emission, recovery, recycling and destruction.



European Union Regulations

Although signatories of Montreal Protocol, the countries of the European Union have drawn up more stringent regulations, broadly similar to the Protocol in that they restrict production and

trade of "controlled substances". The European Regulations have an accelerated phased-out timetable.

Under the December 1992 revision of the regulations for CFC's the European Union phase out targets are:

CFC's

Date	Percentage production of 1986 levels
01/01/94	15%
01/01/95	Total phase-out

HCFCs

Starting approximately in June 1995 and in stages at 1 January 1996, 1 January 1998 and 1 January 2000. Restrictions have also been placed on new equipment using R22 as a refrigerant.

The controlled substances are the same as for the Montreal Protocol. Both the Montreal Protocol and European Regulations provide for regular revision and updating based on scientific data on the environment.



In 1986 the EU countries produced about 440,00 tonnes of CFCs, 70% of which were consumed by the EU countries. This does not allow for substances that were imported. By 1993 European consumption of fluorocarbons had decreased to 120,00 tonnes.

Ozone depletion and global warming potential of fluorocarbons

Refrigerants have been rated according to their potential to damage the environment. These ratings are called Ozone Depletion Potential (ODP) and Global Warming Potential (GWP). The higher the number, the greater their potential to cause damage.

ODPs are based on a "bench mark" of 1.0 for R 11.

GWP's as shown below are based on a bench mark of Carbon Dioxide $CO_2=1$ and have been taken from UNEP/WMO scientific assessment 1994/1995.

The above represents the most common refrigerants. Manufacturers and suppliers will O'ive data for each of their products which are too numerous to include here.

Refrigerant replacement categories

There are three main categories when it comes to replacement refrigerants. These are:

1. Drop-ins: Refrigerant that can be substituted into an existing system without any work being required apart from very minor servicing, such as the replacement of a refrigerant filter drier.
2. Retrofittable refrigerants: refrigerants that can be substituted into an existing plant but only after certain changes have been made, such as substitution of a new type of lubricating or a modification of compressor speed.



3. **Non retrofittable refrigerants:** Refrigerants that cannot be used in existing equipment even with major modifications because of different operating pressures, materials, incompatibility and other potential problems.

European Standards

CEN Standard

Member states of the EEC have technical representatives working on a "CEN" standard for refrigerating systems and heat pumps.

CEN can be loosely translated as the "European Committee for Standardisation". CEN 182 European Standard for Refrigerating Systems and Heat Pumps, is expected to be adopted in 1991/1992. This is a comprehensive standard covering all aspects of a refrigeration system from design, installation etc., through to maintenance and operation. One complete section of the standard will deal with Refrigerant Safety and Environmental requirements, Recovery, Reuse and Disposal.

The foreword of this section is likely to contain a statement urging all personnel involved in anyway with refrigerants to take steps to eliminate emissions. It will probably declare that this should be an objective throughout the design, installation and use of the equipment.

Briefly, to comply with the Standard, refrigerants will have to be handled safely, not emitted to atmosphere and recovered, recycled and reclaimed. This will have to be undertaken only by competent persons.

Cecomaf Recommendations



The European Committee of Manufacturers of Refrigeration Equipment - CECOMAF - have published a document called "Reduction of Chloro-Fluoro - Carbon Emissions from Refrigerant Systems (see appendix 2B)

This publication deals with all aspects of refrigeration work from design through to end user instructions. It covers:

Refrigerant Piping - The need for greater use of brazed unions - fitting of isolating valves using flexible hoses for connections to pressure controls and minimising vibration in pipe work which is one of the biggest causes of leaks.

Leak Testing - The need to use dry nitrogen and soapy water, and the helium method to leak test new or opened systems and to check systems at regular intervals.

Fluorocarbon Users and Their Response

Refrigeration Industry

All people active in the refrigeration industry have an obligation to protect the environment against CFC emissions. Every effort must be made to save the CFC's that are presently held in refrigeration systems.

In the short term a reduction in CFC consumption can only be achieved by reducing refrigerant losses from existing systems. Estimates on the extent of losses from existing systems vary but it is generally accepted that between 30% and 40% of refrigerant consumption is used for servicing. This represents operational leakage's.



It is important that leak testing and servicing techniques improve. Also there will be a greater need for the recovery and recycling of refrigerants.

R22 is likely to become more widely used in the short term.

The industry will also have to look closely at the possibility of using Ammonia more Widely.

To help minimise CFC emissions it is important that all staff now involved with refrigeration work are trained to show a proven ability in controlling CFC emissions. It is also vital that contractors show they are doing all they can to make their employees aware of the need to minimise the release of CFC'S.

To this end, it is important that end users only engage contractors, whose staff employ the latest techniques for protecting the environment.

Aerosol Manufacturers

CFC 11 and 12 have been used in the aerosol industry since 1946. They are mainly used as a propellant. When pressurised inside a sealed can, together with the product they provide pressure to force the product out of the can.

The aerosol industry has switched to using hydrocarbon gases, mainly pentane and butane, as the propellant.

Pentane and Butane will not necessarily totally replace CFC's in all aerosols; in fact, some people see them only as medium term alternatives, the main problem being their flammability.

Dimethyl-Ether - DME - and related chemicals are related to the simple hydrocarbons propane and butane. They have been used in small quantities as aerosol propellants for many years but their use



has been limited by their relatively high cost. But as the aerosol industry looks for alternatives to CFC's the cost of DME is likely to come down. Like the hydrocarbons- DME - contains no chlorine and is therefore ozone friendly.

The British Aerosol Manufacturers Association has announced that 90% of CFC's will be replaced in their products by the end of 1989.

Compressed Gases

Compressed air or a number of similar gases can be used as aerosol propellants.

These gases include oxygen, nitrogen, nitrous oxide, and carbon dioxide. All these gases are ozone friendly and cheaper than CFC'S. They do have limitations in their use though. The major problem is that they tend to produce wet sprays and can lose pressure faster as they are used up, quicker than either hydrocarbons or CFC'S.

Manufactures are looking at ways of overcoming these problems. One method being investigated is the "bag-in-a-can". The product is placed in a flexible bag connected to a valve and surrounded by compressed gas. One advantage of this method is that the product doesn't have to be able to mix with the propellant.

Non-chemically Propelled Aerosols

Aerosols do not have to "powered" by a chemical. Old fashioned, manually operated systems such as scent sprays and pump dispensers work perfectly well.

Future trends

Of the new generation of CFC's being produced HCFC, 123 and HFC 134a are thought to be good candidates for aerosols.



Foam Manufacturers

CFC's are widely used as blowing agents for a variety of plastic foams. The main types of foam which uses CFC's are:

Rigid polyurethane used in building and construction
Refrigerator and freezer insulation

Rigid PU foam is closed cell foam blown with carbon dioxide. CFC I I used to modify the density of the foam and to provide cooling during the curing stage of manufacture. In some cases CFC I I can be replaced with methylene chloride or water, but the introduction of CM (Combustion Modified) foams has meant an increasing reliance on the non flammable CFC I 1. Several companies are actively involved with the development of activated carbon recovery systems. These units are expensive but able to recover 40% of the blowing agent used.

Polystyrene Foams for packing

Packing foams were traditionally blown with CFC 12 or a mixture of CFC's I I and 12, producing extra extruded sheets of foam which were then vacuum formed into meat trays, egg boxes and hamburger cartons.

Electronics and other cleaning applications

CFC 113 is used mainly as a cleaner and degreaser in the manufacture of printed circuit boards. It is used because during soldering, deposits of grime and metals are left on the board and have to be removed. CFC 113 is favoured because of its good solvent power and compatibility with the electronics attached to the board: it is non-flammable and has a low human toxicity level.



One alternative to CFC 113 is Trichloroethane. It is a more powerful solvent which means it will clean away more flux and grease but will attack some sensitive plastic components. Trichloroethane is not completely ozone benign, therefore it is seen only as a short term alternative.

The future for electronics applications may be in the use of water based cleaners (i.e.: water plus detergent), although this may not be the total answer and more research needs to be done to find fully acceptable alternatives.

In dry cleaning CFC 113 is mainly used. HCFC 123 is likely to be an alternative in this field, but there is no readily available alternative at present.

Recovery of Used Refrigerants

To help minimise the loss of CFC's to the atmosphere when working on systems, every effort should be made to recover the refrigerant for future use. The refrigerant should be decanted into specially supplied cylinders. These cylinders are then returned to the appropriate chemical company, so the refrigerant can be reclaimed or recycled.

The chemical companies will only accept cylindrical-containing one refrigerant, for recycling. Mixtures cannot be recycled but will be collected for safe disposal, the cost of disposal will probably be passed on by the manufacturer to the contractor.

Also on the market, are Recovery and Reclaim stations. These machines will draw the refrigerant out from an existing system - separate any oil and other contaminant - dry the refrigerant and then store it. The refrigerant can then be put back into the system after the repair has been completed.



An Introduction to R134a refrigerants

- Use and Applications
- Hazards, Health Care and Safety Precaution
 - Potential Hazards
 - Asphyxiation
 - Burns
 - De-composition in Naked Flames
 - Narcotic Sensation
 - Heart Effects
 - Safety Precaution
 - Goggles and rubber gloves
 - Workshop Ventilation
- Dehydration
 - Moisture effects to the refrigeration system
 - Maximum allowable moisture in refrigeration system
- Refrigeration System Evacuation
- Vacuum
- Refrigerant/refrigeration oil compatibility
- Leak Testing
- Cleanliness
- Refrigerant Recovery
- Refrigerant Recycling
- Refrigerant Reclamation
- Affect of R134a to system component
 - Capillary Tube
 - Compressor
 - Filter Drier
 - Condenser and Evaporator

Conversion of prototypes into ozone layer friendly refrigerants

- Preliminary Design of a new product
 - Overall Dimension
 - Refrigeration load calculation
 - Model Type



- Refrigeration System (No-Frost or Defrost)
- Internal Design of the products
- Material Used
- Air Circulation
- Type of Insulation
- Number and type of door
- Type and material of Door Gasket
- Operational Test
- Performance Test

- Renovation of a Product
 - Design Style
 - Material
 - Insulation
 - Energy Consumption
 - Internal Volume
 - Refrigeration System
 - Noise Level
 - Compressor
 - Performance Test

- Modification and Engineering Changes
 - Refrigeration System Optimisation
 - Performance Test
 - Major and Minor changes to the foam thickness, inner liner, door, cabinet and minor adjustment to the refrigeration system circuits
 - Material Changes, ABS, Polystyrene and etc.

- Conversion
 - Refrigeration load Calculation
 - Component Selection
 - Refrigeration System Adjustment
 - Compressor
 - Capillary Tube
 - Thermostat



- Condenser
- Evaporator
- Refrigerant Charge
- Evacuation
- Leak Test
- Environmental Test
 - Performance Test
 - ISO standard Test

Refrigeration load calculation

- Dimensional Studies
- Heat Transfer through side walls
- Product Load
- Infiltration, door opening Etc.
- Heat gain by miscellaneous apparatus, such as lights, electromoter, fans and etc.
- Calculation of Total Cooling Capacity
- Selection Compressor Cooling Capacity
- Component selection
- Testing Prototypes
- Trial Tests
- Obtaining data from the test results
- Test results analysis
- Trial production
- Mass production

Selection of Components

With respect to the refrigeration load calculations and refrigeration circuit design of each models. Following components were selected.

1- Compressors



- a) Water Chiller Model ELC-WC18,
Danfoss FR6G
- b) Water Chiller Model ELC-WC50C,
Danfoss SC15G
- c) Water Chiller Model ELC-WC50T,
Danfoss SC15G
- d) Chest Freezer Model ELS-CF450,
Danfoss SC18G
- e) Chest Freezer Model ELS-CF750,
Danfoss SC18/18G
- f) Chest Freezer Model ELS-CF220,
Danfoss SC12G
- g) Show Case Model EIS-1889V
Danfoss SC12/12G
- h) Show Case Model ELS-1982
Danfoss SC21G
- i) Show Case Model ELS-260-3F
Danfoss FR11G
- j) Show Case Model ELS-220-3F
Danfoss FR10G
- k) Show Case Model ELS-1219F
Danfoss SC15G

2- Refrigerant Charge

The refrigerant charges for different prototype models are varied. We tried to keep refrigerant charge weight same as R12 refrigerant models, for water chillers and chest freezer with total cooling capacity of less than 300 watts at the first step of testing prototypes we considered same refrigerant charges as R12. But for bigger size of Chest Freezers and Show Cases, we preferred to test the prototypes one by one at different conditions. Therefore we recommend that all prototypes should be charged as follows:

- l) Water Chiller Model ELC-WC18, Refrigerant charge
= 210 grams



- m) Water Chiller Model ELC-WC50C, Refrigerant charge = 600 grams
- n) Water Chiller Model ELC-WC50T, Refrigerant charge = 600 grams
- o) Chest Freezer Model ELS-CF450, 10% reduction of refrigerant weight = 480 grams
- p) Chest Freezer Model ELS-CF750, 10-15% reduction of refrigerant weight = 720 grams
- q) Chest Freezer Model ELS-CF220, Refrigerant charge = 380 grams
- r) Show Case Model EIS-1889V, 10-15% reduction of refrigerant weight = 1200 grams
- s) Show Case Model ELS-1982, 10-15% reduction of refrigerant weight = 720 grams
- t) Show Case Model ELS-260-3F, 10-15% reduction of refrigerant weight = 630 grams
- u) Show Case Model ELS-220-3F, 10-15% reduction of refrigerant weight = 630 grams
- v) Show Case Model ELS-1219F, 10-15% reduction of refrigerant weight = 630 grams

3- Capillary Tube Length

The length of capillary tube depending on internal refrigerant circuit volume should be adjusted, to correspond to the adequate delivery of refrigerant liquids into the refrigeration circuit. Therefore we recommend an increase of ten to fifty percent of capillary tube length.

4- Evaporator and Condenser

According to our experience of making prototypes and test results evaluation achieved in several project we did not changed the size of evaporator and condenser. Any changes must be done

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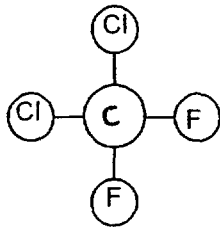
prior to making prototypes and at R12 refrigeration circuit optimisation programme.

Prototype Making

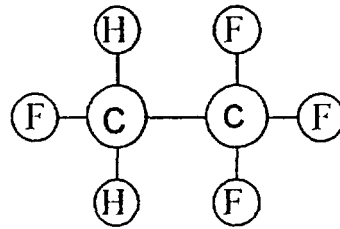
The prototypes have been manufactured successfully in accordance with the existing design and drawings and also advises given to the engineering department of the factory.

HFC134a, Refrigerant Use and Application

HFC134a is a HydroFluoroCarbon (CH_2FCF_3) to replace CF_2Cl_2 , ChloroFluoroCarbon (CCl_3F),



CCl_2F_2 (R12)



CH_2FCF_3 (R134a)

Hazards, Health Care and Safety Precaution

Potential Hazards:

Asphyxiation, (Suffocating)

Since R134a is heavier than air it tends to concentrate on the floor, this situation may cause breathable air leading to dizziness, nausea, and fainting.

Burns

When liquid refrigerant R134a contacts the skin or eyes it boils off at -26.5 C , therefore it causes severe burns to the skin and possible blindness if it contact with the cornea.

Decomposition in naked flames

Although R134a is non flammable, and toxic decomposition products. It may be given off when contacts with naked flames. Acid fumes which are irritant are given off (released) first, and then phosgene which is highly toxic.

Narcotic sensation (effects)

At high concentration, normally in excess of 15% by volume, exposure to the vapors can cause a feeling of well being, or can be dangerously misleading.

Heart Effects

At high concentration, normally in excess of 15% by volume, exposure to the vapors can cause irregularities in heart beat, which in severe cases it can cause heart attack and possibly death.

Safety Precaution

Proper use of personal protective equipment such as; Goggles and Rubber Gloves help to reduce the hazards.

Workshop Ventilation also ensures the reduce of risks to health by losses of refrigerant into the workplace.

Cylinders should be stored in well ventilated areas, out of direct sunlight and away local heat sources. The cylinders should not restrained and not standing in water.

Adequate warnings should be displayed to warn people of their presence.

When handling refrigerants from systems that have suffered electrical burn out, be aware that refrigerant and oils may be acidic and thus pose a risk to health.

Dehydration

It is important to realize that moisture in a refrigeration system is directly the major cause of several other symptoms.

Moisture can be classified as visible (in form of water liquid) and invisible (in form of water vapor).

Water vapor can be found everywhere, in all solids, liquids and gases.

Content of water vapor in air is expressed as relative humidity. This gives the greatest trouble in refrigeration system.

A single drop of water may look harmless, but to a refrigeration system it is a monster, the number one enemy to be combated by refrigeration service specialists.

Moisture can get into a system easily and is hard to get out.

Moisture effects to the refrigeration system

1. "Freeze ups", moisture will be picked up by the refrigerant and become entrained in refrigerant line in a fine point of expansion. The ice crystal will retard or stop the flow of the refrigerant, causing a reduction or complete stoppage of cooling.
2. As the expansion valve warms, due to the lack of refrigerant, the ice melts and moisture returns to the expansion valve and once more builds an intermittent cooling.
3. Moisture can induce Corrosion, which can present serious trouble because often the effects of corrosion are not apparent until the real damage has occurred.

4. Moisture can alone in the form of water cause rust after a period of time. However, moisture plus refrigerant creates much more corrosion trouble.
5. Max. Allowable moisture in a system is **50 PPM**
6. Refrigerant such as R12 containing chlorine will slowly hydrolyze with water and form hydrochloric acid, which greatly increases the corrosion of metals.
7. Heat increases the rate of corrosion due to acids because at higher temperature the acid forming process is accelerated. This acid will attack all kind of materials.
8. Refrigeration oil presents another problem caused by moisture. Refrigerant oil is an exception to the rule that "Oil and water don't mix" in fact, refrigerant oil has an close relation with moisture and will absorb it rapidly if left open to the atmosphere.
9. Water change into acid emulsifies with refrigerant oil, the two forming a mixture of exceedingly fine globules. This effect is called "Slugging" of the oil and greatly reduces its lubricating ability.

10. Corrosion becomes troublesome from the operating standpoint when the metallic surface is eaten away and solid, detachable product is formed. This formation is commonly known as "sludge".
11. Sludge exists as slimy liquids, fine powders, granular solids or sticky solids and causes a variety of problems. They can plug fine strainers, expansion valves and capillary tubes. And because they usually contain acids they corrode whatever they stick to, accelerating damage.
12. To eliminate moisture it is necessary to take precaution and actions which will assure a moisture-free system. And one of the most effective ways to eliminate moisture from a system is through the use of a high vacuum pump.

System Evacuation

1. Reducing the system pressure sufficiently to allow the heat from the atmosphere to vaporize any moisture left in the system is necessary before carrying out the charging process.

2. A system opened for any type of repair must be completely evacuated to remove air and moisture. The new refrigerant HFC134a is very much Hygroscopic and one of the lesser known factors regarding the use of HFC 134a is its tolerance to contaminants. It is believed that an HFC134a system will be somewhat less tolerant to contamination than CFC 12 which has always been relatively tolerant in range of adverse conditions such as moisture, air and dirt.

Always evacuate a system when:

1. Replacing compressor, condenser, drier, evaporator etc...
2. The system has no refrigerant.
3. The refrigerant becomes contaminated.
4. The refrigerant oil is changed.

It is recommended to use Vacuum Pump with 8 cubic meter per hour or more to obtain necessary vacuum pressure, for evacuating the system.

There are three basic parameters to consider for selection of vacuum pump.

1. Free air displacement, e.g. C.F.M. or cubic meter per hour,
2. Vacuum Performance, e.g. mm Hg, TORR etc...
3. Water Vapor tolerance, e.g. max. allowable water content in the system, contamination and employment of vacuum pumps purely for dehydration work.

Vacuum

As we have stressed, refrigerant is sensitive to moisture in the system. To understand how water behaves and how to dry out a system, following natural laws must be understood. The boiling point of water varies depending on pressure. Pressure below atmosphere is called vacuum, and can be measured in:

Millibar (Mbar), Torr with reference to Mercury (Hg) height and Microns.

Atmospheric pressure (1 Atm) = $1 \text{ Kp/Cm}^2 = 0,98 \text{ Bar}$
 $14,22 \text{ Lb/In}^2$ (79% N₂ and 20% O₂)

To obtain evaporation of water/moisture at room temperature, pressure must be 15 MBar only, means a vacuum of $1000-15=985$ Mbar.

Freezing point for moisture/water at 0 C deg. at 6,11 Mbar equal $760 \times 6,11 / 1000 = 4.643$, mmHg = 4,64 = 4640 microns.

Refrigerant / Refrigeration Oil Compatibility;

R134a is not miscible with the traditional mineral oils that were popular with CFC refrigerants.

Polyester lubricants must be used, the compressor will be already charged with polyester lubricant.

R134a is a very hygroscopic and polyester lubricants are approximately 100 times more hygroscopic than mineral oils.

Leak and leak testing;

Because of R134a physical property, it easily leaks in comparison with R12. That is, it leaks more easily and therefore joints have to be carefully inspected for leaks before the refrigerant is charged into the system.

Traditional leak detectors search for Chlorine, and react against it. Therefore, they could not be used for R134a refrigeration system, because there are no Chlorine in R134a refrigerant. R134a is fluorinated Hydrocarbon.

The system should be leak tested prior to charging the refrigerant.

Oxygen free Nitrogen can be used to provide the pressure, with joints tested using soap and water.

Refrigerant Recovery

when refrigerators are to be serviced it is necessary to remove or recover the refrigerant from them.

Note that the capillary tube slows the process of recovery and must be taken into account.

Refrigerant Recycling

this is the process of cleaning the refrigerant by filtration and separation to remove oil, acids and moisture to improve its quality prior to re-use.

Refrigerant Reclamation

This is the process of returning the refrigerant to its original virgin condition in a specially built machine. Before and after processing the refrigerant would be analyzed in a specialist laboratory.

Capillary Tubes

The length of capillary tube vary from 1mt to 6 mt and inside diameter from 0.5 to 2 mm.

Liquid refrigerant enters the capillary tube from the condenser, and as it flows through the tube the pressure drops because of friction and acceleration of the refrigerant.

Some of the liquid flashes to vapor as refrigerant flows through the tube.

Capillary tubes are often installed in heat exchange with the suction line (sub-cooled) to retard the flashing of refrigerant flowing through the capillary tube.

Some Advantage and Disadvantage of Capillary Tube

Advantage:

No moving parts allow pressure equalization during off cycle so that low start torque motor can be used.

Disadvantage:

- 1) Not adjustable to changing load conditions, susceptible to clogging or blockage by foreign matter.
- 2) The mass of refrigerant charge is critical within close limit, it means that capillary tube is only suitable for hermetically sealed systems manufactured in carefully controlled environments and to high quality standards.

Cleanliness

Cleanliness of plant is associated with the experience of the initial installation and its subsequent need for servicing other than routine.

The procedure for new R134a plant are more stringent than those used for R12, attention being drawn, particularly to good evacuation and charging.

It is vital that the system is not contaminated with Chlorinated residues from charging manifolds and vacuum pumps previously used with R11 or R12, or any refrigerant containing chlorine.

The safe level of chlorinated residues is in the region of 200 PPM, but every endeavor should be made to eliminate traces of chlorine to zero.

In application where the plant is to be converted from R12 to R134a the situation is more critical as the installation will contain chlorine and possible chlorine residues.

Condenser and Evaporator Performance

R134a compared with R12 has a better thermal conductivity, higher latent heat and higher coefficients of heat transfer.

The specific volume of the vapor however, is greater as the evaporation temperature falls below about + 5 C.

The consequence of these properties is:

1. A reduction of in capacity of about 10-25 % at LBP if an R12 compressor displacement is used, due to the higher specific volume.
2. At HBP compressor there should be an improvement in performance due to the higher thermal properties and lower viscosities (possibly up to 10% at 10% evaporating).
3. Similarly there should be an improvement in performance at LBP if compressor with higher displacement is used.

4. With a new installation/ optimization of the evaporator and condenser should be considered to achieve the best balance with compressor, according to the experience if proper compressor is being selected there is no need to change condenser and evaporator capacity.
5. The effect of the lubricant on the above properties is however, still under review and the tendency is to keep the same evaporator to the condenser as selected for R12

Pressure drops in both evaporators and condensers will be less than R12

Filter Drier

Due to the hygroscopic property of R134a it is necessary to use a filter drier stronger than R12 drier with 1.4 capacity more, all manufacturers have recommended the replacement of 10 gram conventional drier with XH7 while 15 gram conventional drier should be replaced with XH9 new specially designed drier.