Inventory of Direct and Indirect GHG Emissions from Stationary Air Conditioning and Refrigeration Sources, with Special Emphasis on Retail Food Refrigeration and Unitary Air Conditioning

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Final Report

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ABSTRACT

This project, *Inventory of Direct and Indirect GHG Emissions from Stationary Air Conditioning and Refrigeration Sources, with Special Emphasis on Retail Food Refrigeration and Unitary Air Conditioning*, contains two parts; the first one addresses the refrigerant inventory and greenhouse gas (GHG) emissions of Californian stationary refrigeration equipment. The inventory method follows the Intergovernmental Panel on Climate Change (IPCC) guidelines 2006 and is based on activity data describing the installed base of refrigeration equipment, refrigerants in use, and emission factors depending on the activity sector. The second part addresses the energy consumption of commercial refrigeration and evaluates technical options to improve energy efficiency and their associated costs. The report evaluates the possible energy gains as well as greenhouse gas emission abatement in the commercial refrigeration sector.

EXECUTIVE SUMMARY

Background
To meet the goals of AB 32 (the Global Warming Solutions Act of 2006) the Air Resources Board (ARB) identified and approved two early action measures to reduce direct emissions of high global warming potential (GWP) refrigerant from stationary refrigeration and air conditioning equipment that contribute to climate change. To inform development of these early action measures an inventory of direct greenhouse gas (GHG) emissions from stationary air conditioning and refrigeration equipment, and indirect GHG emissions created by power consumed during the operation of refrigeration equipment, was needed. ARB contracted with ARMINES to:

1) Estimate the refrigerant inventory and emissions of the stationary refrigeration and air conditioning sector in California; and
2) Estimate indirect greenhouse gas emissions from commercial and industrial refrigeration equipment, and evaluate current energy consumption and possible energy saving strategies achieved by integrating high-efficiency technical options in the commercial refrigeration sector.

Methods
The calculation method used for refrigerant inventory follows the Tier 2a method recommended by the Intergovernmental Panel on Climate Change (IPCC) guidelines in 2006, which is a “bottom-up” method based on the description of refrigeration equipment by sector. The sectors taken into account are: commercial refrigeration, industrial refrigeration, and air conditioning (including chillers). The installed base of equipment is described for each year and throughout the lifetime of the equipment, which can vary from 10 years to more than 40 years. In addition to annual emission estimates, the installed amount of refrigerant in equipment, or “banks” have been estimated to produce reliable projections of future refrigerant emissions through 2020.

ARMINES evaluated new refrigeration architectures and technologies that could result in a lower overall carbon footprint (total warming impact) from a combination of less refrigerant required in a system and potentially less energy required to operate the system.

For energy consumption of commercial refrigeration systems installed in California, the report describes in detail the commercial outlets where refrigeration systems are in use: Thirteen different categories of stores are defined, including supermarkets, grocery stores, restaurants, hotels, and gas stations. Commercial refrigeration systems are classified in three main categories depending on the size, the technology and their energy consumption: stand-alone equipment, condensing units, and supermarket centralized systems. In order to assess the energy consumption of each type, more than 100 detailed visits to stores were conducted in order to evaluate the types of equipment used in commercial refrigeration.

The annual energy consumption of all commercial refrigeration systems have been estimated by evaluating the energy use of the 28 different types of equipment used in these systems. The calculation method developed takes into account the equipment types, their operating conditions, and the outdoor temperatures of the eight climatic zones of California.

Three main technical options have been evaluated for energy saving in supermarkets:
1) Installing glass doors on medium-temperature display cases;

2) New technologies for auxiliary components (efficient lighting, efficient fan motors, better control of defrosting devices), and
3) Better control of the refrigeration system condensing pressure.

Energy savings from using these technical options have been estimated for a typical supermarket and for a typical small grocery store in California.

Results

Annual GHG emissions in 2004 from stationary refrigerant sources in California were estimated at 16 million metric tons of carbon dioxide equivalents ($MMTCO_2E$). Air-conditioning is the dominant source of these emissions, accounting for 55 percent of the total, with the remaining from commercial refrigeration (19 percent), chillers (19 percent), and industrial refrigeration (7 percent).

An additional 185 $MMTCO_2E$ of potential emissions exists in the installed base, or "bank" of refrigerant in equipment. The majority of the bank, 57 percent, is represented by HCFC-22, and another 40 percent is comprised of hydrofluorocarbons (HFCs). The production of HCFC-22 will be phased out beginning in 2010, and its replacement with higher-global warming HFCs could double the global warming impact from this sector if emission rates remain the same (HCFC-22 has a GWP of 1,500; and a likely replacement is R-404A, with a much higher GWP of 3,260).

Equipment changes in commercial refrigeration lead to a potential three-fold reduction in the carbon footprint of grocery stores, supermarkets, and food-related businesses by replacing current centralized systems (which tend to be very emissive) with more leak-tight indirect systems. In indirect systems, the high-GWP refrigerant is contained in a much smaller circuit in the machinery room and a heat transfer fluid with a negligible GWP provides the coldness in the display cases located in the sales area.

Annual energy consumption of commercial refrigeration systems in California is significant, accounting for up to 8 percent of all electrical usage in the state, with year 2004 consumption estimated at 20,200 GWh (giga-watt hours; or billion-watt hours) for all Californian commercial refrigeration. Grocery supermarkets used 5,300 GWh, or 25 percent of the total commercial refrigeration energy usage. Current energy consumption can be decreased 30 percent for supermarkets, and decreased 20 percent for the entire commercial sector, by using currently available best technologies for stand-alone equipment, condensing units, and supermarket centralized systems. The payback period from applying best technologies vary from three months to less than four years.

Conclusions

This study establishes that refrigerant GHG emissions from stationary refrigerant sources in California are significant, and will most likely increase through 2020 without changes made to the types of equipment technologies used. Progress has been made by some commercial chains, and refrigerant emissions have been lowered from an average of about 30% per year to about 15%, but this progress needs to be consolidated and furthered.

Indirect systems using CO_2 as heat transfer fluid is a technical option decreasing the refrigerant charge required four to eight-fold, and radically changes future refrigerant emissions with very significant GHG reductions. In parallel with possible refrigeration system evolution, several technical options are available to limit the energy consumption of commercial refrigeration systems. Significant energy gains are realized from the redesign of display cases by adding transparent doors for all operating temperatures, the use of LED lighting, and the adoption of efficient technologies for fan motors and defrosting.

INTRODUCTION

High global warming potential (GWP) greenhouse gas (GHG) emissions are expected to grow substantially over the next several decades, primarily from the use of ozone-depleting substance (ODS) substitutes, such as hydrofluorocarbons (HFCs). The introduction of ODS substitutes to the refrigeration and air-conditioning (A/C) sectors is the major driver of this growth. Additionally, although they are being phased out, ODS (many of which have high GWPs) are currently used in older refrigeration and air-conditioning equipment

There are two significant sources of GHG emissions from the refrigeration and air-conditioning sectors, direct emissions (high GWP refrigerant emissions due to leaks, routine servicing, intentional or accidental venting, and end-of-life refrigerant reclamation and/or venting), and indirect emissions (CO_2 emissions resulting from energy generated to operate the equipment). The purpose of this study is to quantify direct GHG emissions from the refrigeration and air-conditioning sectors in California, and to quantify indirect GHG emissions from the retail food sector.

The study is based on a field survey of commercial outlets to evaluate the numbers and the types of all refrigerating equipment of the commercial sector. Refrigeration equipment can be classified in three families: stand-alone equipment, condensing units, and centralized systems. They differ in size, refrigerant charge, refrigerant type, emissions, and energy efficiency. The survey is necessary to inventory refrigerants and to assess the energy consumption.

For the refrigerant inventory, the CEP (Center for energy and Processes) has developed a long experience for the French inventories but also at the global level. RIEP database (Refrigerant inventories and emission previsions) has been developed; refrigerant banks of 62 countries and eight regions are stored and updated every 3 years. Specific studies have been carried out for the U.S. and the U.S. EPA “vintaging model” has been compared with RIEP [Pal 04]. The database is filled with data either bought from marketing studies or coming from interviews of experts from the refrigeration industry, as well as field survey and data collection from technical papers, specialized magazines, and web sites. For California, the data are either derived from U.S. numbers, California statistics or direct survey.

For energy consumption, based on the field survey a typical “averaged” supermarket is defined. Knowing the total number of supermarkets, the total energy consumption can be derived for California as well as the energy savings associated with the implementation of energy efficient technical options. Moreover, small commercial stores using refrigeration systems have also to be described because all stand-alone equipment such as vending machines, ice cube machines, soda fountains, and movable display cases constitute a large stock of equipment with low energy efficiency.

A calculation tool has to be developed in order to calculate energy consumption of all refrigeration systems based on their technical description, their operating conditions, and the outdoor temperatures of the eight climatic zones of California. The calculation will be performed on a hour-by-hour basis for all the year and for each type of system.

The report is structured in two parts: the first one addresses refrigerant banks and emissions of stationary refrigeration system including AC systems, and the second part develops the analysis of energy consumption of the commercial refrigeration sector, evaluates the energy gains associated with several technical options, and makes the TEWI (Total Equivalent Warming Impact) comparison of direct and indirect systems in order to evaluate direct and indirect emissions of those refrigeration systems.

Part 1 - REFRIGERANT INVENTORY AND EMISSIONS FOR CALIFORNIAN STATIONARY REFRIGERATION SYSTEMS

1 Method of calculation

The calculation method is described in Annex 2.

2 Database for commercial refrigeration in California

The bank of refrigerants contained in grocery supermarkets is calculated based on the last 30 years. The bank represents the cumulated market of refrigerants filled in new refrigerating systems year after year, taking into account the average lifetime of the store (30 years for a supermarket) and the lifetime of the refrigeration system (15 years for centralized systems) before remodeling. The average lifetime of refrigeration systems is thus 15 years. However, calculations are performed taking into account an extinction curve of the equipment around this average value.

Calculations are performed from year 1990 to 2004, and forecasts are simulated until 2020. In order to initiate calculations as of 1990, the database has to include data from 1960 to 2020, assuming an average 30-year lifetime of grocery supermarkets and other small stores.

2.1 Grocery supermarkets

2.1.1 Number of stores in California

The number of grocery supermarkets in California from 1960 to 2020 must be evaluated. Statistical data issued from the U.S. Bureau of Census are used. For a given year, different numbers have been found depending on the source. The definition of supermarket store reported in the U.S. Bureau of Census is as follows:

"Supermarkets and other general-line grocery stores: establishments commonly known as supermarkets, food stores, grocery stores, and food warehouses, primarily engaged in the retail sale of a wide variety of grocery store merchandise. Customers normally make large, volume purchases from these stores."

Numbers of grocery supermarkets in the State of California are available for a few years. More data are available for USA, and can be helpful to evaluate California numbers, by means of ratio based on GDP and population. Those ratios are used to fill the database back to 1960.

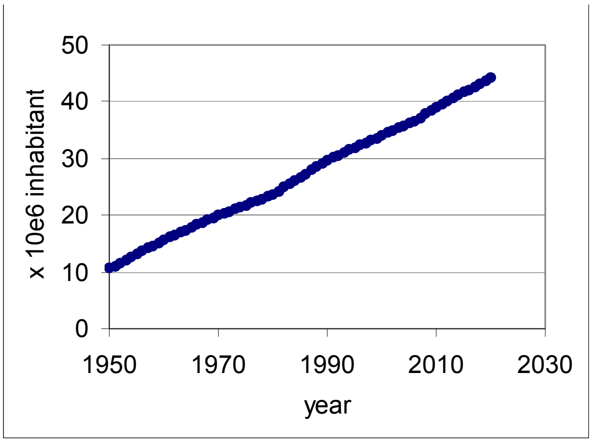


Figure 1.1 Population growth in California.

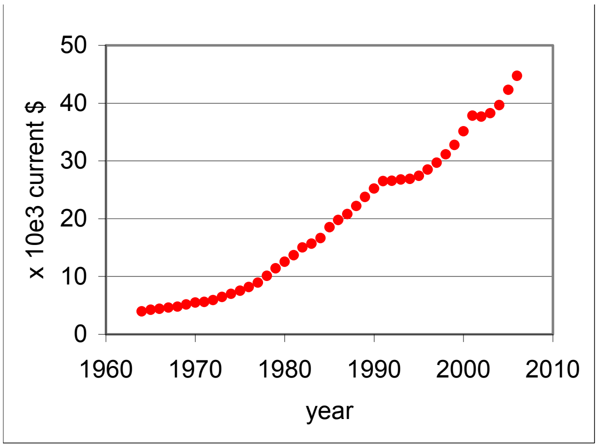


Figure 1.2 GDP/inhab. growth in California.

Figure 1.3 draws the evolution of supermarket number in California, based on U.S. Bureau of Census data for U.S. and GDP ratio (California / USA). The first supermarkets were built after the Second World War. 1950 is taken as the initial year. The average area of supermarket is known for a few years [FMI07]. A ratio of the total sales area to population was introduced and extrapolated for all years concerned by this study (see Figure 1.4).

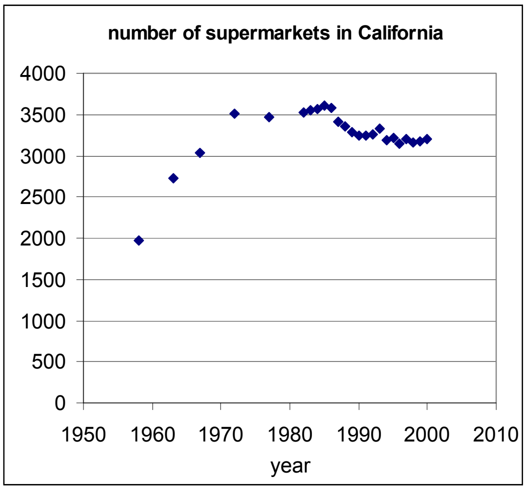


Figure 1.3 Number of supermarkets in California. Data from U.S. Bureau of Census.

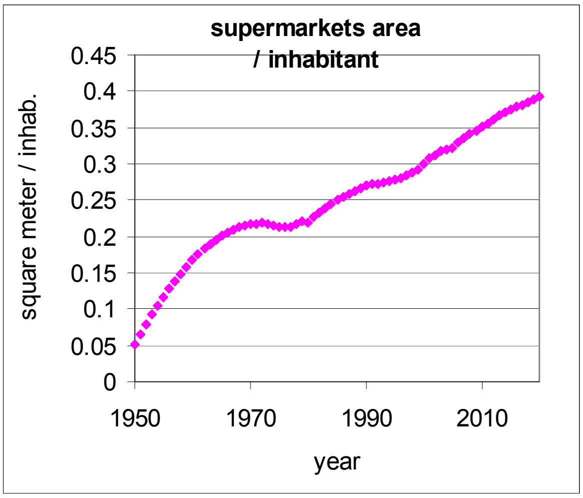


Figure 1.4 Average sale area/per inhabitant growth from 1950 to 2020.

Multiplying this ratio (sale area / inhab.) by the population, the total sales area in California is derived from 1950 to 2020 as shown in Figure 1.5.

From the evolution of the supermarket number in California, openings of new stores per year can be determined. The average lifetime of a supermarket store is generally 30 years before replacement or significant remodeling.

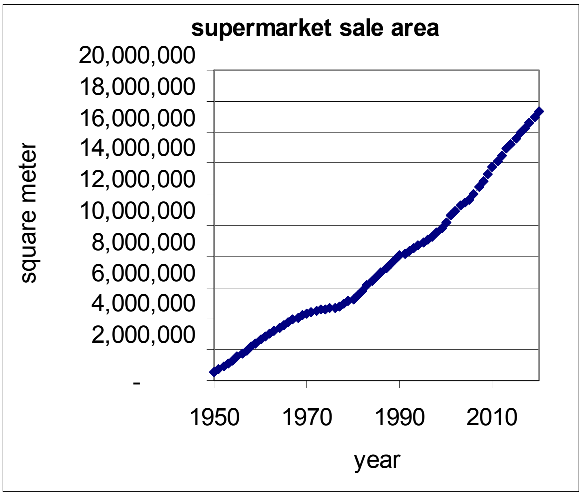


Figure 1.5 Supermarket total sale area in California.

2.1.2 Average area and refrigerant charge

The average area in 2004 is 4,400 m^2 (47,360 ft^2) but this value was not constant throughout the last 50 years. Newly opened stores, including supermarkets, have a sales area larger than this average value.

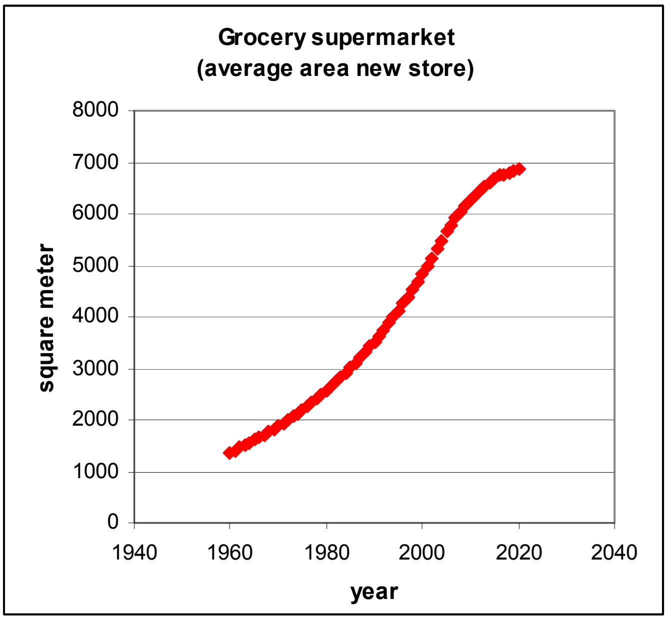


Figure 1.6 Average area of grocery supermarkets.

Figure 1.6 depicts the evolution of the sales area of grocery supermarkets as of 1950. This value is the typical area of stores classified by vintage, meaning that the average value for all supermarkets is lower than the current vintage area.

For example it is assumed that the average sales area of supermarkets in California, whatever their date of opening, is 4400 m^2 in 2004. But a new supermarket opened in 2004 has an average area of 5000 m^2.

The refrigerant charge has been determined from the calculation of the cooling capacity of the centralized refrigeration system. The typical layout of the grocery supermarket has been established after the field survey. Each and every display case has been characterized by its cooling capacity and temperature level. Ratios of refrigerant charge per kW of cooling capacity are known for centralized systems. Table 1.1 summarizes the results of calculations used for energy consumption calculations. Table 1.2 gives the ratio of refrigerant charge / cooling capacity for the different technologies of refrigeration systems commonly found in grocery supermarkets.

Table 1.1 Cooling capacity of each refrigeration system for a typical grocery supermarket.

Cooling Capacity	*Medium temperature*	*Low temperature*	*Total*
Centralized System (kW)	193	152	345
Condensing Units (kW)	18	14	32
Stand-alone (kW)	13.0	2	15
Total	**224**	**168**	**392**

Table 1.2 Ratios for refrigerant charge.

Refrigerant charge / Cooling Capacity	*Medium temperature*	*Low temperature*
Centralized System in direct expansion	2.8 kg/kW	5.5 kg/kW
Centralized System with secondary loop	0.8 kg/kW	1.2 kg/kW
Condensing Units	1.4 kg/kW	2.4 kg/kW

Stand-alone equipment are not included in this table because the information for the refrigerant charge is known (from the manufacturer) for each type of display case considered.

Using the cooling capacity and the refrigerant charge ratios, the total refrigerant charge for a supermarket is determined. Table 1.3 gives the results. Currently in California, it is considered that the use of secondary loop systems or CO_2 cascade is not significant enough to be considered. On the contrary, forecasts will be performed in order to evaluate the impact of a widespread use of secondary loop systems on refrigerant emissions.

Table 1.3 Refrigerant charge in a grocery supermarket.

Refrigerant charge	*Medium temperature*	*Low temperature*	*Total*
Centralized System	540 kg	837 kg	1377 kg

During the field survey, in most of supermarket stores visited, it was impossible to visit the machinery room. Nevertheless, some machinery rooms of a few supermarkets have been visited with the store manager. A first cross checking has been done with refrigerant charge indicated in one of these supermarkets. For confidentiality reasons, the brand name and the location of the reference supermarket are not mentioned. Two compressor racks for medium temperature display cases and storage room are charged with 408,15 kg of R-507A each. The compressor racks for low-temperature equipment are charged with 408,15 kg. This supermarket is representative of the typical grocery supermarkets described and the refrigerant charge is 3 x 450 kg, nearly the same (Table 1.3) as the one defined using the refrigerant ratio (Table 1.2) and the cooling capacity (Table 1.1). Those ratios have been elaborated on a number of field surveys made in the U.S. and in Europe.

For calculations in RIEP, the average refrigerant charge is related to the supermarket sale area and a coefficient of refrigerant charge per square meter is defined. This coefficient, 0.36 kg/m^2 for year 2004, is the expression of the share of refrigerated (and frozen) food in a grocery supermarket. The change in food consumption habits during the last 50 years had an impact on the refrigeration ratio for a given sales area.

The assumption is made that the ratio of refrigerated sales area has increased by 50% between 1960 and 1990 (see Figure 1.7).

In parallel, the average area of grocery supermarket was increasing (see Figure 1.6), meaning that the refrigerant charge in a typical supermarket was 200 kg in 1960, and is 1,600 kg today.

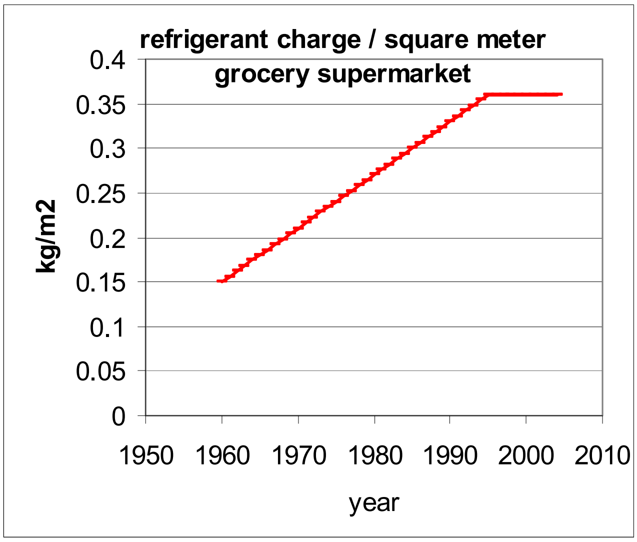


Figure 1.7 Refrigerant charge ratio (kg/m^2).

2.1.3 Refrigerant emission rates

Refrigerants emissions are of two types: fugitive emissions during all the lifetime of the refrigeration equipment, including accidental emissions (rupture of pressure valve or liquid line), and end-of-life emissions when refrigerant recovery is not performed carefully (if performed).

The fugitive emission rate is an average value taking into account refrigerant losses of different types. This emission rate is established based on refrigerant annual consumption for a given store: the refrigerant quantity refilled annually in the system compensates refrigerant emissions.

In RIEP, emission rate of commercial centralized systems is usually estimated at **30% per year**, except when other emission rates are verified by the refrigerant invoices of commercial companies. This value is a conservative one, emissions could vary from 10 to 30% [GAG97], [IPC05], [TOC06]. Complementary data are needed but are not easily disclosed by commercial chains. Based on answers to a survey launched by the Center for Energy and Processes (CEP) for this study, one Californian commercial chain has explained how they have reduced their annual emission rates from more than 30% down to 18%. Those two values (18 and 30%) will be used as lower and higher thresholds for the evaluation of emissions (see Section 2.3.3) and refrigerant demand (see Section 2.3.1).

Refrigerant recovery at end of life has increased in the past 10 years due to the phase-out of CFCs. Figure 1.8 presents the refrigerant recovery efficiency evolution assumed for RIEP calculations.

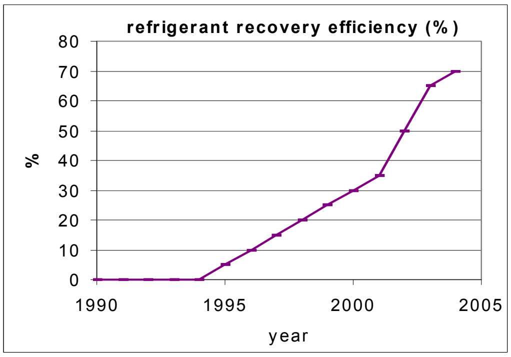


Figure 1.8 Refrigerant recovery efficiency at end of life.

2.1.4 Refrigerants in use

The refrigerant market share follows the regulations. Since 1995, CFCs are no longer used in new refrigeration systems. CFCs have been replaced first by HCFC blends for retrofitting of existing systems and in new systems. HCFCs will be banned in new refrigeration systems as of 2010. In the U.S., HFCs began to be used in centralized systems in 1999. Refrigerant shares are presented in Figure 1.9.

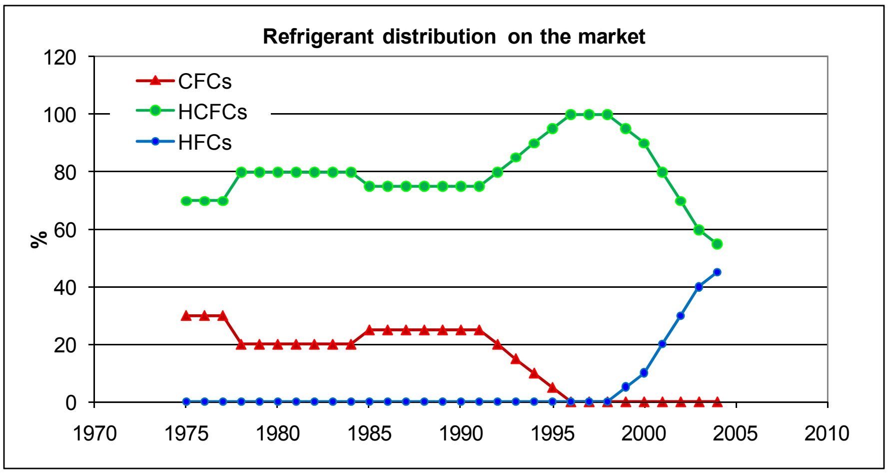


Figure 1.9 Refrigerant market shares (new equipment and remodeling).

Remodeling of the refrigeration system is supposed to be done at mid-lifetime of the store, after 15 years. After 1995, this step is typically the time to retrofit CFCs refrigerant to HCFC blends. Retrofit of CFCs was supposed to be achieved in 2004.

2.2 Small stores

2.2.1 Number of stores in California

U.S. Bureau of Census data have been mainly used to evaluate the number of the different stores in California since 1960.

The same methodology is applied to each type of store when statistical data are not available. For small stores like bakeries, butcheries, fishmongers, and convenience stores another point has been considered: these small stores were common before the growth of supermarkets. For the last 20 years, the numbers of these small stores has been decreasing. They are replaced by larger sales area stores such as mini-markets and supermarkets.

Stores, where refrigeration equipment is used, are summarized in Table 1.4. Those stores were described in Section 1.1 of “Part 2 – Energy consumption of the commercial refrigeration sector in California”.

Table 1.4 Stores reported in the database.

Type of stores	*Number of stores in 2004*
Grocery stores (food dedicated)	3,370
Minimarkets	4,693
Pharmacies	4,846
Convenience stores	2,317
Liquor stores	3,466
Butcheries, Pork-butcheries	763
Fishmonger stores	184
Bakeries and Pastries	5,512
Small size Gas Stations	5,453
Large size Gas Stations	1,818
Hotels	5,458
Motels	5,817
Bars and Restaurants	66,306
Stand-alone equipment studied independently of their location of use	***Number of units in use in 2004***
Carbonated Soda Fountains	23,040
Vending machines	500,000

2.2.2 Refrigerant charge and type of refrigerant

The field survey has allowed identifying the typical layout of each small store using refrigerating equipment. This refrigerating equipment is not connected to a centralized system in the machinery room like in supermarkets. Stand-alone equipment, display cases, and walk-in coolers connected to one or several condensation units are the typical technologies met in those stores. Table 1.5 presents the typical refrigerant charge evaluated for each type of store, based on the field survey.

Table 1.5 Refrigerant charges.

Type	*Refrigerant charge in stand-alone equipments (kg)*	*Refrigerant charge in condensing units (kg)*
Bakeries	2.65	2.4
Bars & restaurants	2.5	19.3
Vending machines	0.3	-
Butcheries	0.3	7.0
Center gas stations	3.45	16.7
Convenience stores	4.1	27.3
CSD fountains	0.3	-
Fishmonger stores	0.6	7.0
Grocery supermarkets	7.4	58.9
Hotels	3.4	19.3
Liquor stores	5.8	17.6
Mini-markets	9.6	99.2
Motels	1.45	0
Pharmacies	2.35	34.8
Small gas stations	0.8	3.2

Note: Grocery supermarkets are considered in the list of stores because the use of stand-alone equipment in the sales area is significant. Moreover some walk-in coolers are not connected to the centralized system, but run with independent condensing units.
Stand-alone equipment and condensing units are not similar to centralized system in terms of energy efficiency, refrigerant type, and emissions. In order to be more accurate in the evaluation of refrigerant

emissions, refrigerating equipment has been sorted and calculated by technology: stand alone equipment, condensing unit, and centralized system.

2.2.3 Refrigerant emission rate

Stand-alone equipment is characterized by short refrigerant circuit, the compressor and the condenser being integrated in the cabinet. Tube and fittings are usually brazed, which helps reduce fugitive emissions. Because of a small unitary refrigerant charge (less than 2 kg in most cases), refrigerant recovery at end of life is not done on stand-alone equipment.

Condensing units are more emissive systems. The refrigerant charge can be significant, for example in mini-markets, where the number of display cases is high. Recovery at end of life should be improved.

Table 1.6 Emission rates and recovery efficiency.

Type	*Emission rate (%)*	*Recovery efficiency (%)*
Stand-alone equipment	1	0
Condensing units	15	30

2.3 Refrigerant inventory from 1990 to 2004

2.3.1 Refrigerant demand

The activity data for the commercial sector is the number of all refrigeration systems: stand-alone equipment, condensing units, and centralized systems. The uncertainties are relatively low and can be estimated in the range of ± 2% due to the data availability. For emission factors, uncertainties are low for stand-alone equipment and emissions are also low, but for centralized systems, uncertainties are high and emission factors are strongly dependent on the containment policies of commercial companies. The refrigerant demand, related to the refrigerant charge needed for servicing is evaluated by 2 thresholds:

- High threshold where refrigerant emissions from centralized systems is evaluated at 30%, and
- Low threshold where emissions are fixed at 18% (see Section 2.1.3).

The refrigerant demand is the addition of refrigerant needs for servicing of all refrigerating systems in use, and the refrigerant needs for first charge of new refrigerating systems. Figure 1.10 and Figure 1.11 present the lower and higher threshold for the refrigerant demand in centralized systems. Figure 1.12 and Figure 1.13 give the refrigerant demand, by type, respectively for condensing units, and stand-alone equipment.

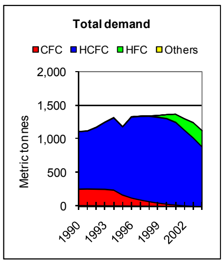


Figure 1.10 Total demand in centralized systems – Lower threshold.

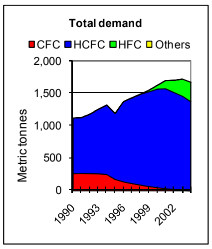


Figure 1.11 Total demand in centralized systems – Higher threshold.

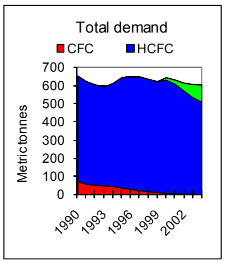


Figure 1.12 Refrigerant demand of condensing unit systems.

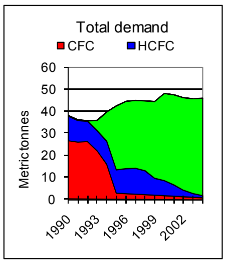


Figure 1.13 Refrigerant demand of stand-alone equipment.

Depending on the technology (centralized, condensing unit, and stand alone) the refrigerant demand is substantially different. As indicated on Figure 1.16, centralized systems, with 1,700 tonnes per year, represent nearly 72% of the refrigerant demand in the commercial refrigeration sector.
Note: 1,7000 t/yr correspond to the lower emission threshold.

Refrigerant demand for stand-alone equipment is mainly for new equipment sold on the market, because for this technology, servicing needs are low. In terms of refrigerant distribution, HFCs are mainly dedicated to stand-alone equipment. In other technologies (centralized and condensing units) refrigerant needs are high for servicing, because of high emission rates of these systems (18 to 30% in centralized systems and 15% in condensing units).

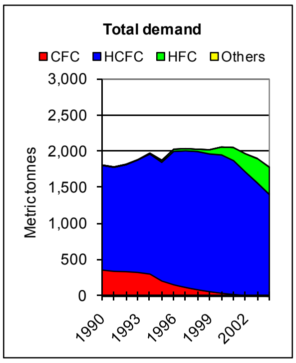


Figure 1.14 Total demand in commercial refrigeration – Lower threshold.

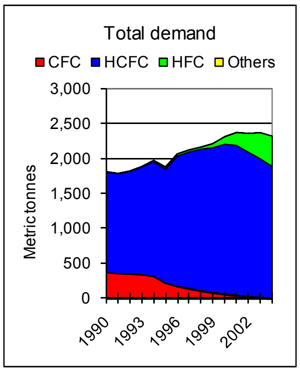


Figure 1.15 Refrigerant demand in commercial refrigeration – Higher threshold.

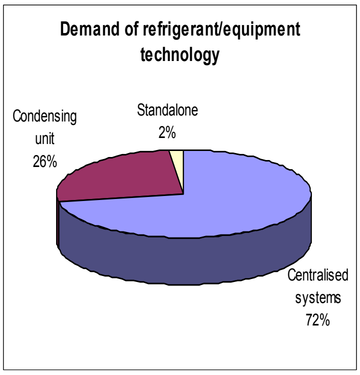


Figure 1.16 Distribution by refrigeration equipment technology – Higher threshold.

HCFC demand, mainly R-22, represents 72% of the refrigerant market. The commercial refrigeration sector demand of HCFCs in 2004 is evaluated between 1,500 and 2,000 tonnes.

2.3.2 Refrigerant bank charged in refrigeration equipment

In the commercial sector, the refrigerant bank is the total amount of refrigerant charged in all refrigeration systems in use, whatever their vintage. Figure 1.17 to Figure 1.19 present the refrigerant bank, by family, respectively in centralized systems, condensing units, and stand-alone equipment.

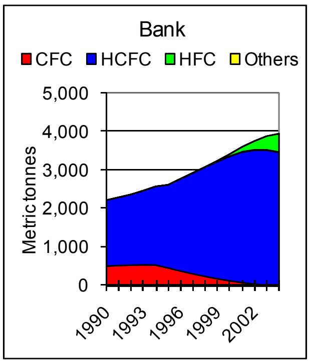


Figure 1.17 Refrigerant bank in centralized systems (grocery supermarkets).

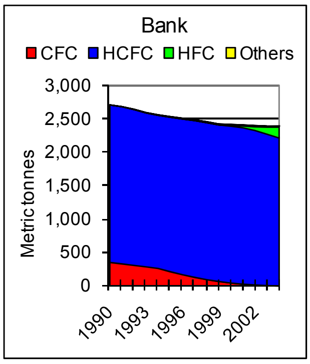


Figure 1.18 Refrigerant bank in condensing unit systems.

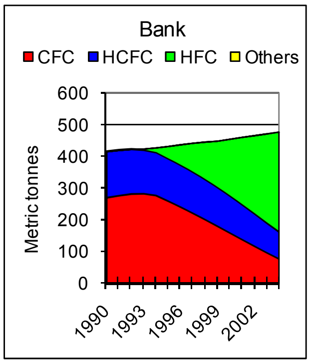


Figure 1.19 Refrigerant bank in stand-alone equipment.

In California, the bank of refrigerants in centralized systems in supermarkets is nearly 4,000 tonnes in 2004, and 90% of this bank are HCFCs. The introduction of HFC on the market, in new equipment, began in 1999. In condensing units, the bank of refrigerant is around 2,500 tonnes, but is not growing any longer.

Stand-alone equipment, working with small refrigerant unitary charges, were filled mainly with R-12 before 1992. No retrofit is performed on those systems. In 2004, the remaining bank of R-12 in stand-alone equipment is estimated at 500 tonnes. R-12 has been replaced by R-134a, which is in the main refrigerant in use, today, in stand-alone equipment.

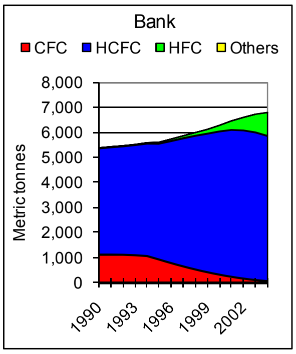


Figure 1.20 Refrigerant bank in commercial refrigeration.

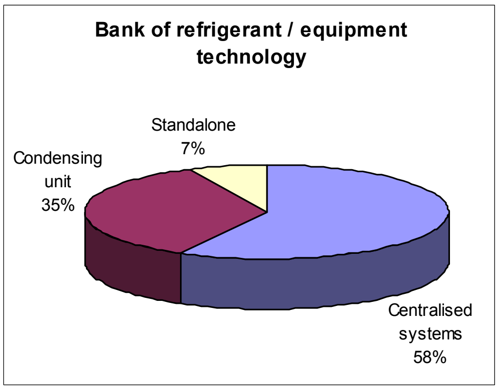


Figure 1.21 Distribution by refrigeration equipment technology.

In the commercial refrigeration sector, the total refrigerant bank is estimated around 6,800 tonnes in 2004, mainly constituted of HCFCs. 58% of the refrigerant bank is filled in centralized systems in supermarkets.

2.3.3 Refrigerant emissions

Emissions represent both fugitive losses, and end-of-life emissions. Figure 1.22 to Figure 1.25 present the refrigerant emissions, by type, respectively from centralized systems (lower and higher thresholds), condensing units, and stand-alone equipment.

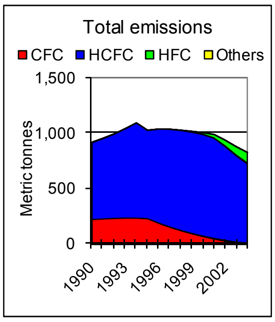


Figure 1.22 Total emissions in centralized systems – Lower threshold.

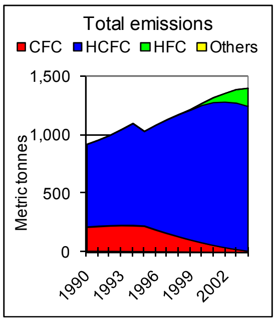


Figure 1.23 Total emissions in centralized systems – Higher threshold.

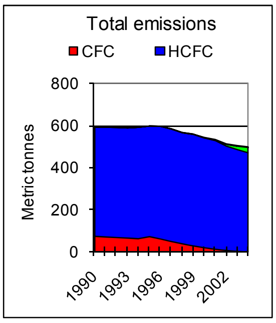


Figure 1.24 Refrigerant emissions in condensing unit systems.

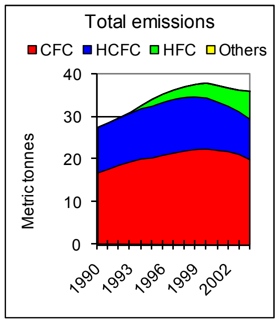


Figure 1.25 Refrigerant emissions in stand-alone equipment.

Lifetime of these refrigeration systems is estimated to be 15 years, in average. For centralized systems, the real lifetime of the system is usually longer. But from1990 to 2020, many retrofit operations have been and will be done, due to the phase-out of CFCs, which will be followed by the phase-out of HCFCs. When the system is renewed, most of the time, refrigerant handling leads to emissions.

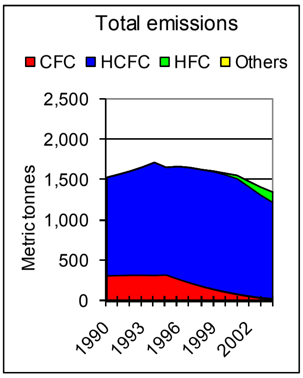


Figure 1.26 Total emissions in commercial refrigeration – Lower threshold.

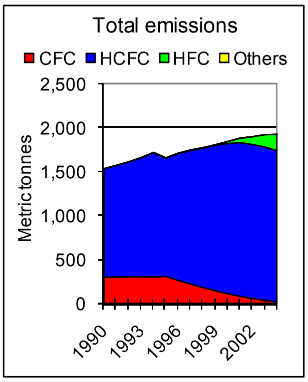


Figure 1.27 Total emissions in commercial refrigeration – Higher threshold.

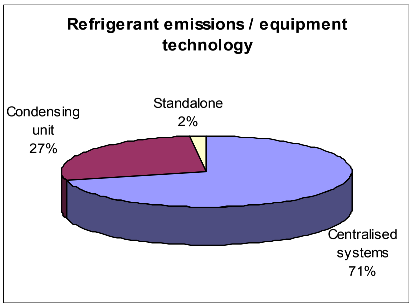


Figure 1.28 Distribution of emissions by refrigeration equipment technology.

Total emissions from the commercial refrigeration sector in California are estimated between 1,400 and 1,800 metric tonnes in 2004, more than 70% of these emissions are coming from centralized systems.

2.3.4 CO_2 equivalent emissions of refrigerants

Refrigerant emissions expressed in CO_2 equivalent are based on GWP values from the IPCC Second Assessment Report. Figure 1.29 to Figure 1.32 present the refrigerant emissions in CO_2 equivalent values, respectively from centralized systems (lower and higher thresholds), condensing units, and stand-alone equipment.

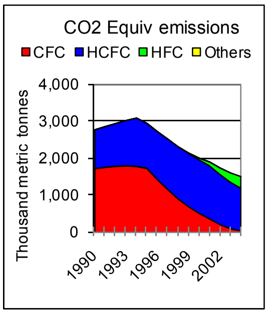


Figure 1.29 CO_2 equiv. emissions in centralized systems – Lower threshold.

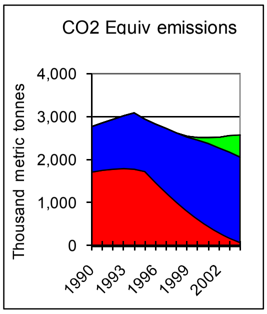


Figure 1.30 CO_2 equiv. emissions in centralized systems – Higher threshold.

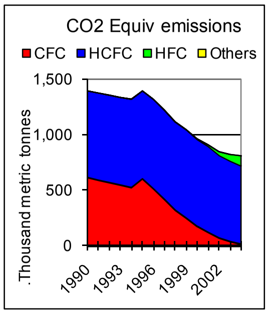


Figure 1.31 CO_2 emissions in condensing unit systems.

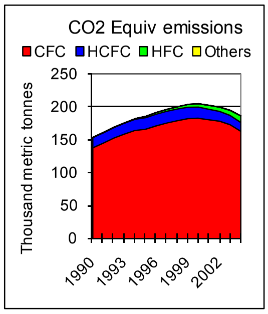


Figure 1.32 CO_2 emissions in stand-alone equipment.

From 1990 to 1995, CFC emissions represent around 20% of refrigerant emissions in centralized systems in supermarkets. Because of its high GWP (GWP R-12: 8600), emissions of this CFC represent more than 55% of CO_2 equivalent emissions.

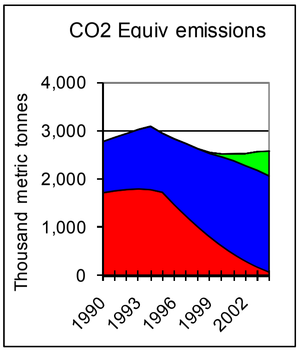


Figure 1.33 CO_2 equiv. emissions in commercial refrigeration – Higher threshold.

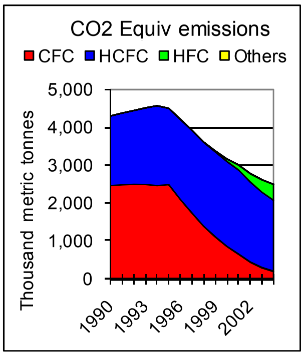


Figure 1.34 CO_2 equiv. emissions in commercial refrigeration – Lower threshold.

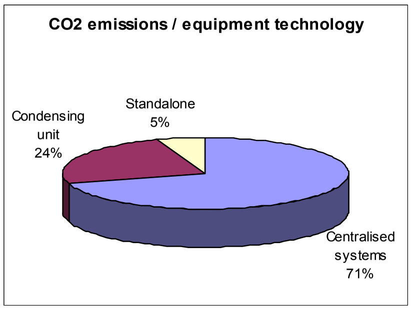


Figure 1.35 Distribution by refrigeration equipment technology.

Total CO_2 equivalent emissions have been decreasing since 1995 because of the R-12 phase-out. In 2004, CO_2 equivalent emissions in commercial refrigeration are estimated between 3.5 and 3.6 million metric tonnes of CO_2 depending on the emission factor of centralized systems.

2.4 Scenarios and projections to 2020

2.4.1 Assumptions for scenarios

Three scenarios have been simulated to evaluate the impact of technical changes and policies on refrigerants.

2.4.1.1 Scenario 1: business as usual (BAU)

No significant changes are considered. The regulation organizing the phase-out of HCFCs after 2010 is taken into account. The use of secondary loop systems is not accelerated. Emission rate and recovery efficiency are kept at the same level. There is no significant effort to retrofit R-22 systems.

2.4.1.2 Scenario 2: large introduction of secondary loop systems

Indirect refrigeration systems decrease the refrigerant charge and minimize potential refrigerant leakage. Indirect systems have many forms: complete indirect system, partial indirect system, and indirect cascade system. Water solutions have long been used as heat transfer fluid (HTF). Other very promising developments are phase-change HTF, mainly CO_2.

Starting in 2008, secondary loop systems are progressively introduced. The use of CO_2 as HTF is technically possible for both medium and low-temperature systems. For now, this technology is only used for low-temperature systems, where the pressure is limited to 1.2 MPa in operating conditions.

Assumptions made for the simulations in Scenario 2 are as follows:
- 75% of new refrigeration systems are built with a CO_2 secondary loop for low-temperature applications
- 50% of new refrigeration systems dedicated to medium-temperature are secondary loop systems with water solutions of glycols
- Secondary loop systems have an emission rate of 10%, as a result of improved refrigerant containment in the machinery room

- The refrigerant charge is reduced (see Table 7.2) with secondary loop systems, R-404A is the refrigerant used in the machinery room
- R-22 retrofit with R-422A or equivalent intermediate HFC blends starts in 2008 and is totally done in 12 years
- Recovery efficiency is progressively increased to 80%
- The emission rate on new centralized systems is progressively reduced from 30% to 20% thanks to improved leak tightness of components, improvement in the leak search and data reporting when refrigerant losses are observed.

2.4.1.3 Scenario 3: introduction of low GWP refrigerants, reduction in cooling capacity and refrigerant charge

Recent research on refrigerants has lead to new molecule developments resulting in to reach very low GWP refrigerant. In 5 to 10 years, refrigerant blends with GWP lower than 500 will possibly be available for low-temperature application.

Simulations on energy consumption, when all display cases are closed, have shown significant decrease of the refrigeration needs. This scenario is evaluated in Scenario 3.

Assumptions taken for the simulations in scenario 3 are:
- Identical to scenario 2, except the choice of R-404A in secondary loop system. A new refrigerant blend, called BLD1 (blend 1) with a GWP of 500, is introduced progressively on the market, beginning in 2012. It replaces R-404A and R-507A in new refrigeration systems.
- The cooling capacity is cut by nearly 40%: all open display cases are replaced by display cases equipped with glass doors. The replacement of old display cases is done in 15 years, starting in 2008.

2.4.2 Refrigerant bank filled in refrigeration equipment

2.4.2.1 Case of centralized systems in supermarket only

Figure 1.36 to Figure 1.38 present, for each scenario, the refrigerant bank changes from 2000 to 2020 in centralized systems in supermarkets.

In Scenario 1, business as usual, the refrigerant bank continues to grow, and reaches 6,000 tonnes in 2020. In 2010, HCFC bank is still more than 50% of the total bank.

In Scenario 2, the introduction of secondary loop systems starting in 2008, allows reversing the growth of the bank. In 2020, around 2,000 tonnes of refrigerants are avoided compared to the BAU scenario.

In Scenario 3, both secondary loop systems, and reduction of the refrigeration needs in the sales area (glass doors) have permitted to divide the BAU bank of refrigerants by a factor 2, at 3,000 tonnes, in 2020.

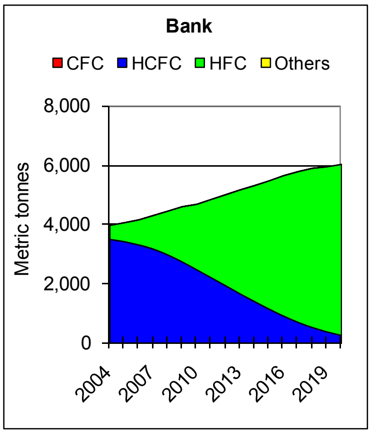


Figure 1.36 Scenario 1 Refrigerant bank changes in centralized systems.

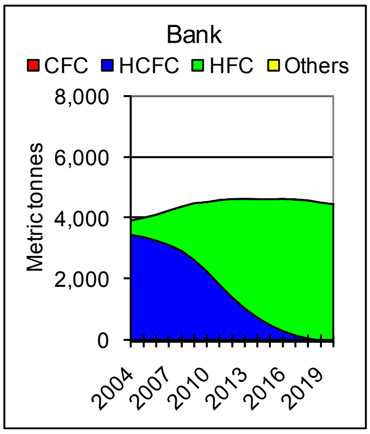


Figure 1.37 Scenario 2 Refrigerant bank changes in centralized systems.

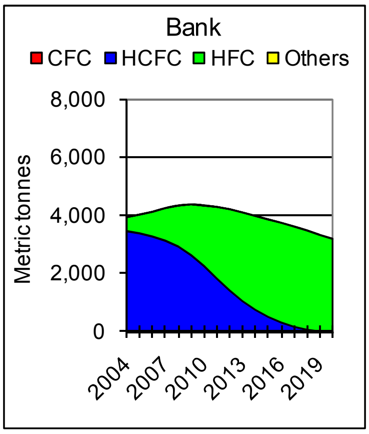


Figure 1.38 Scenario 3 Refrigerant bank changes in centralized systems.

2.4.2.2 Commercial refrigeration sector, including small stores

Figure 1.39 to Figure 1.41 present, for each scenario, the refrigerant bank changes from 2000 to 2020 in the commercial refrigeration sector.

In the business as usual scenario, the total bank of commercial refrigeration sector reaches 9,000 tonnes in 2020.

In Scenario 2, the impact of secondary loop introduction in supermarkets is less significant in relative value, because of the refrigerant bank in condensing units. Technically, the use of secondary loop systems is possible in small stores in replacement of condensing units, but the uptake of the technology is relatively slow. In the commercial refrigeration sector, including small stores, the impact of measures taken in Scenarios 2 and 3 are significant. In Scenario 3, the refrigerant bank reduction is 30% in 2020.

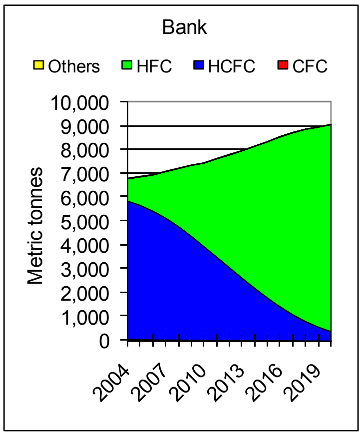


Figure 1.39 Scenario 1 - Refrigerant bank changes in commercial refrigeration.

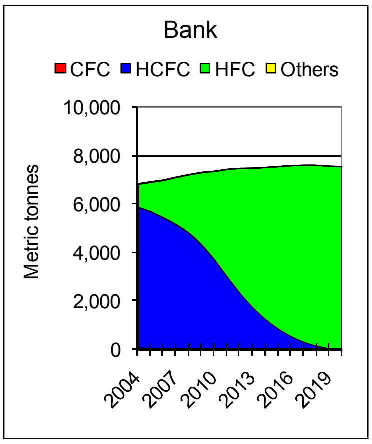


Figure 1.40 Scenario 2 - Refrigerant bank changes in commercial refrigeration.

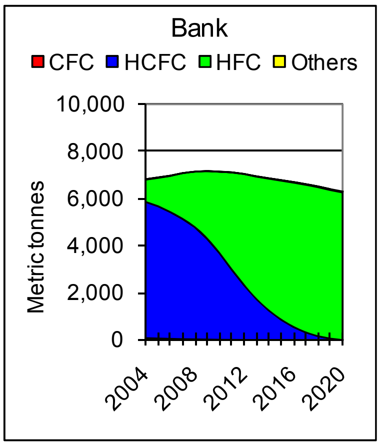


Figure 1.41 Scenario 3 - Refrigerant bank changes in commercial refrigeration.

2.4.3 Refrigerant emissions

2.4.3.1 Centralized systems in supermarkets only

The impact of technology changes, moving to secondary loop systems, is both on the refrigerant charge and on the fugitive emission rate. As shown on Figure 1.43 (Scenario 2) and Figure 1.44 (Scenario 3), the level of HFC emissions is divided by 2 (Scenario 2) and divided by a factor 3 in Scenario 3.

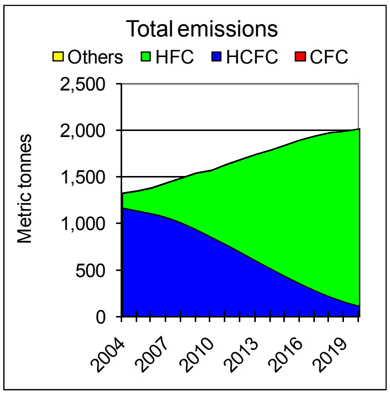


Figure 1.42 Scenario 1 - Refrigerant emission changes in centralized systems.

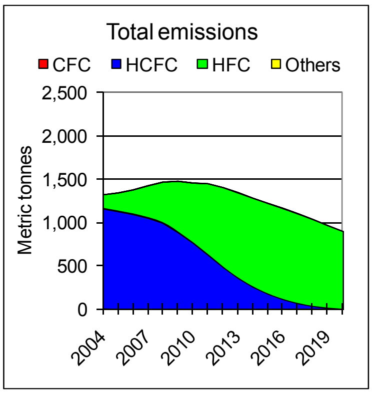


Figure 1.43 Scenario 2 - Refrigerant emission changes in centralized systems.

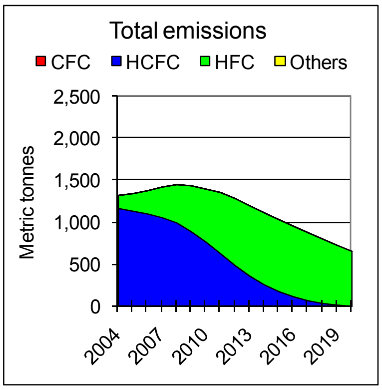


Figure 1.44 Scenario 3 - Refrigerant emission changes in centralized systems.

2.4.3.2 Commercial refrigeration sector, including small stores

In Scenarios 2 and 3, improvements of leak tightness and of recovery efficiency have also been considered for condensing units and stand-alone equipment. Figure 1.45 to Figure 1.47 present results in emission reductions for the commercial refrigeration sector, taking into account all technologies.

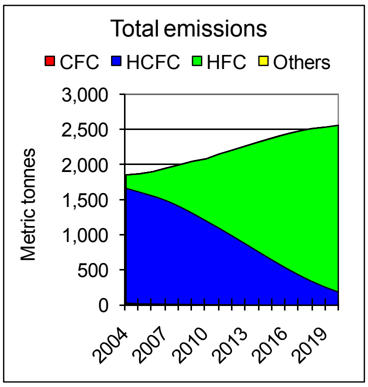


Figure 1.45 Scenario 1 - Refrigerant emission changes in commercial refrigeration.

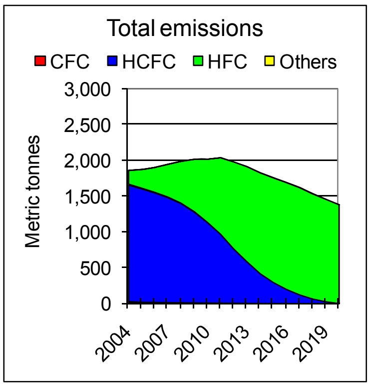


Figure 1.46 Scenario 2 - Refrigerant emission changes in commercial refrigeration.

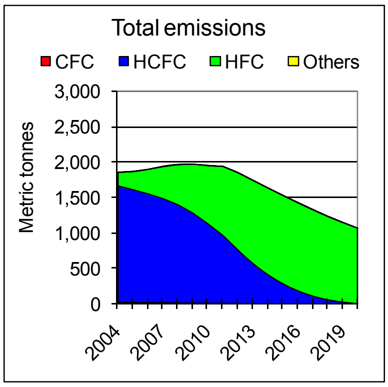


Figure 1.47 Scenario 3 - Refrigerant emission changes in commercial refrigeration.

In the business as usual scenario, the level of refrigerant emissions is above 2,500 tonnes per year in 2020. In Scenario 2, after the introduction of secondary loop systems in supermarkets, refrigerant emissions are limited to 1,400 tonnes in 2020. When the cooling capacity is decreased (Scenario 3), in addition of a secondary loop system, refrigerant emissions are lower: 1,000 tonnes in 2020.

2.4.4 Refrigerant CO_2 equivalent emissions

2.4.4.1 Centralized system in supermarkets only

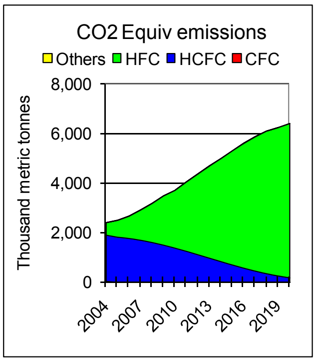


Figure 1.48 Scenario 1 - CO_2 emission changes in centralized systems.

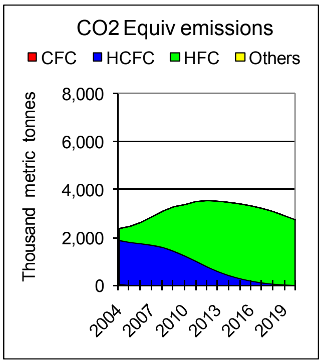


Figure 1.49 Scenario 2 - CO_2 emission changes in centralized systems.

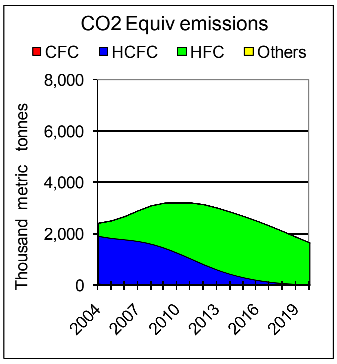


Figure 1.50 Scenario 3 - CO_2 emission changes in centralized systems.

R-507A and R-404A have the highest GWP of HFC refrigerants currently used, with GWPs of 3300, and 3260 respectively. The phase-out of HCFCs that had lower GWPs has an impact on CO_2 equivalent emissions. In scenario 1, which is the current scenario for year 2000 to 2008, a minimum of CO_2 equivalent emissions is observed in 2004. After this date, the wide use of R-404A has a negative impact on CO_2 equivalent emissions. Those emissions could reach 6.2 million tonnes in 2020, more than the values met in the period of use of CFCs (from 1990 to 1994) (see Figure 1.48).

2.4.4.2 Commercial refrigeration sector, including small stores

The commercial refrigeration sector includes all centralized systems, condensing units, and standalone equipment in supermarkets and all small stores.

Changes in refrigerants, with low-GWP blends, could take place beginning in 2012, but would most likely be used only in new refrigeration systems. The simulation of scenario 3 does not consider a retrofit of existing systems with R-404A. Nevertheless, the reduction in CO_2 equivalent emissions is significant: less than 2.5 million tonnes in 2020, instead of 8 million in scenario 1 under business as usual.

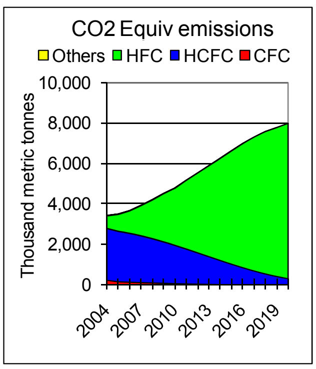


Figure 1.51 Scenario 1 - CO_2 emission changes in commercial refrigeration.

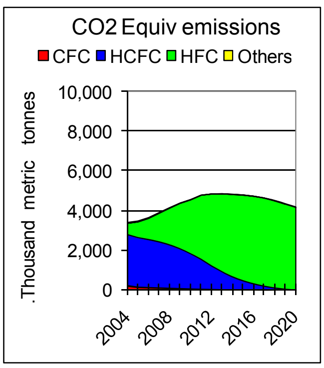


Figure 1.52 Scenario 2 - CO_2 emission changes in commercial refrigeration.

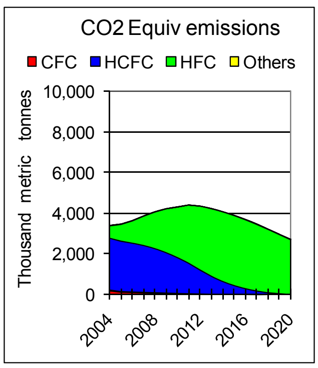


Figure 1.53 Scenario 3 - CO_2 emission changes in commercial refrigeration.

2.4.5 HCFC recovery and refrigerant demand

In 2010, the production of virgin HCFCs is banned. In developed countries, the use of HCFCs is still possible with recycled fluids, but the demand will have an impact on the refrigerant prices. Figure 1.54 and Figure 1.55 give an evaluation of HCFC demand for servicing in commercial refrigeration, and, in parallel, the total amount of HCFCs recovered after end of life or retrofit operation.

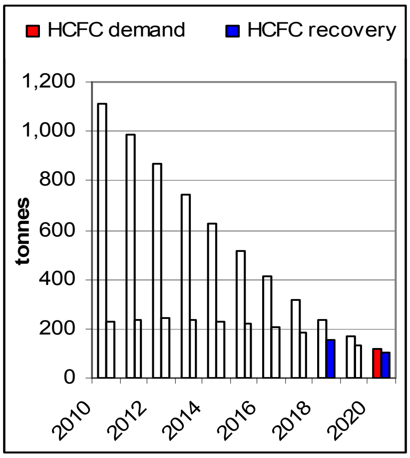


Figure 1.54 Scenario 1 - R-22 demand and recovery.

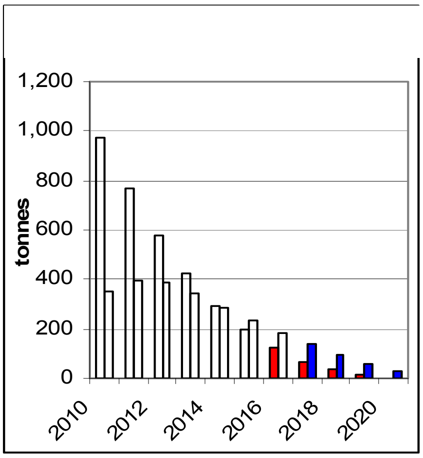


Figure 1.55 Scenario 2 & 3 - R-22 demand and recovery.

In Scenario 1, no retrofit has been considered before the end of life of systems, or before the renewing period (15 years). Figure 1.54 shows clearly the lack in refrigerant for the period from 2010 to 2020: the recovery of R-22 is nearly 200 tonnes per year, when the refrigerant needs for servicing are 1,100 tonnes in 2010 and 600 tonnes in 2015. Without any leak tightness improvement, and retrofit policy, HCFC needs will exceed available recycled refrigerants and thereby the change to intermediate blend has to be highly accelerated.

In Scenarios 2 and 3 (Figure 1.55), retrofits of HCFC installations start in 2008 and generate additional refrigerant on the market after recycling. The demand for HCFCs is covered by recovery from 2013 to 2020. Before 2013, needs of R-22 for servicing cannot be covered by refrigerant recovery from the commercial sector only.

3 Unitary Air Conditioning and chillers

This section covers two sub-domains:
- air-to-air stationary air-conditioning systems and
- chillers

Chillers are used for climate comfort and in industrial processes. Chiller manufacturers consider that about 2/3 of large chillers manufactured have been installed for climate comfort. The two equipment sub-domains and even the eight categories of air/air AC systems, and the two categories of chillers exist as specific parts in the RIEP database. These categories have been merged into one category when it comes to markets, banks, emissions, etc… in order to comply with the IPCC reporting format and limit the amount of tables and figures.

3.1 Air-to-air systems

3.1.1 Data sources and detailed calculation method

Data for sales and production of new equipment are not available for California. Therefore, a ratio (*Population of California/Population of USA*) is applied to available statistics of the USA from Building Services Research and Information Association (BSRIA) [BSR02], [BSR05].

$$Market\ California\ (year\ i) = Market\ USA\ (year\ i) * Ratio\ (year\ i) \quad (1.1)$$

$$Production\ California\ (year\ i) = Production\ USA\ (year\ i) * Ratio\ (year\ i) \quad (1.2)$$

This source includes 8 categories of air-to-air systems:
- Portable/Moveable
- Window
- Splits (Ductless < 5 kW)
- Splits (Ductless > 5 kW)
- Indoor Packaged
- Ducted Splits < 17 kW
- Ducted Splits > 17 kW
- Roof tops.

In the RIEP database, the eight different categories are calculated separately, and one global methodology is applied to all categories. Differences in refrigerant charge and choices are described the following sections.

Based on data from reference sources, the equipment production and markets are calculated, taking into proper account exports and imports of equipment. Knowing the average charge and the refrigerant type selected for each eight air-to-air AC category, the annual refrigerant quantity charged in new equipment is calculated. This also applies to the total refrigerant charge of the equipment exported.
- With data on the annual refrigerant market and the equipment lifetime, the Californian refrigerant bank can be determined.
- Using the fugitive emission rate of each category, the annual refrigerant servicing market of a country is determined.
- Refrigerant emissions (fugitive and at end of life) can be derived from the refrigerant bank while using data on the equipment lifetime.

3.1.2 Installed base of AC unitary systems

Based on data available for each category [TOC03] [TOC07] [BSR02] [BSR05], the average refrigerant charge can be established (see Table 1.7). These values correspond to average values if one uses information on the typical shares of refrigerating capacities (which is directly related to the refrigerant charge) within the different categories.

Table 1.7. Characteristics of the eight categories of air-to-air AC equipment.

Type	*Charge (kg)*	*Life (years)*
Portable/Moveable	0.5	10
Splits (Ductless <5kW)	1	15
Splits (Ductless > 5kW)	7.5	15

Indoor Packaged	5.5	15
Window	0.7	12
Roof Top	5	20
Ducted Splits < 17 kW	3.5	15
Ducted Splits > 17 kW	9	15

Table 1.7 shows as well the average lifetime used to establish the law of end of life of equipments given in the following figures.

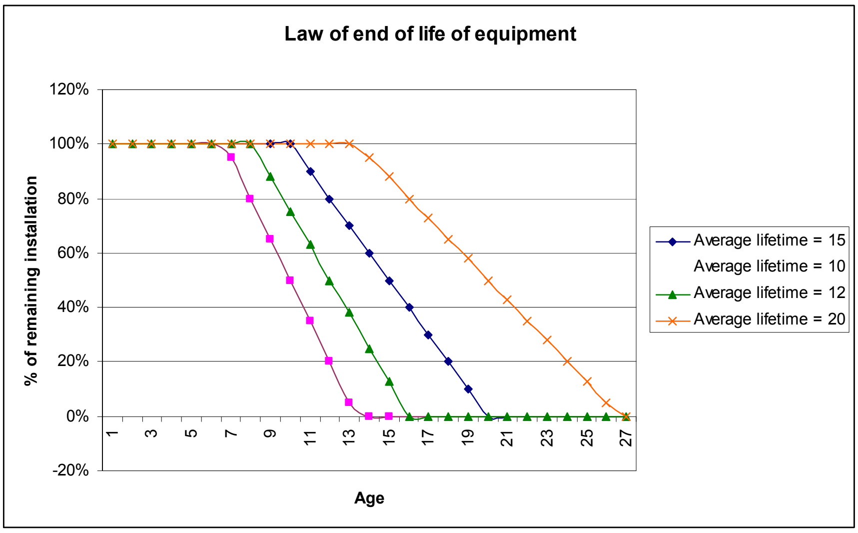


Figure 1.56 Law of end of life equipment (cf. Annex 3).

End of life curves given above are applied to the market of corresponding AC unitary system. The installed base for eight categories is given in Table 1.8.

Table 1.8 Installed base of stationary air conditioners.

Year	Portable	Splits (Ductless <5kW)	Splits (Ductless > 5kW)	Indoor Pack	Window	Roof Top	Ducted Splits < 17kW	Ducted Splits > 17kW
1990	11,820	39,290	18,011	418,596	3,598,786	785,746	3,288,974	332,330
1991	12,680	42,460	19,464	452,398	3,875,957	856,228	3,544,786	359,164
1992	13,580	45,750	20,972	487,456	4,160,071	929,383	3,807,006	386,999
1993	14,460	49,120	22,516	523,378	4,447,229	1,004,331	4,081,997	415,517
1994	15,420	52,660	24,136	561,036	4,748,529	1,082,784	4,366,097	445,414
1995	16,380	56,310	25,809	599,925	5,059,829	1,163,782	4,662,766	476,289
1996	17,340	60,020	27,761	639,127	5,384,157	1,250,271	5,005,851	508,741
1997	18,300	62,810	30,959	678,525	5,734,143	1,336,451	5,342,049	540,454
1998	19,260	65,750	34,311	718,222	6,097,200	1,422,595	5,667,471	572,244
1999	20,240	68,790	37,876	758,109	6,480,214	1,509,996	5,997,946	618,732
2000	21,260	71,930	41,619	798,209	6,856,471	1,598,626	6,335,109	665,973
2001	22,620	75,810	44,204	835,867	7,065,471	1,682,353	6,658,249	706,990
2002	24,340	81,930	45,617	870,958	7,525,829	1,760,888	7,026,360	741,621
2003	26,400	89,620	45,851	903,462	8,031,314	1,834,038	7,368,209	769,781
2004	28,520	97,950	46,095	940,395	8,410,257	1,907,035	7,740,046	800,630

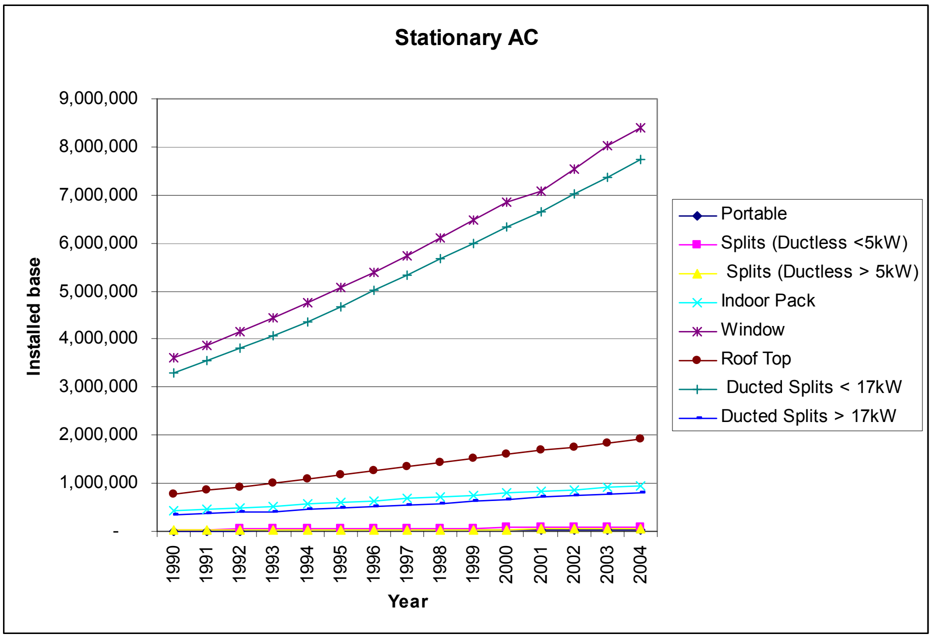


Figure 1.57 Installed base of stationary air conditioners in number of units.

3.1.3 Analysis of the uptake of R-410A

The phase-out of HCFC in brand new equipment in Europe and Japan has led to the introduction of HFC blends on the market: R-410A and R-407C.

In the U.S., the use of R-22 in new equipment is allowed until year 2010. Little interest has been shown by U.S. manufacturers to use R-407C in new equipment, because R-22 is still widely used and readily available. Nevertheless, the use of R-410A, a higher pressure refrigerant and nearly azeotrope blend, makes it possible to reduce the compressor size and increase the compactness of the AC unit.

R-410A units were introduced on the market in 1999 but were limited to small size units. R-410A compressors are specially designed for high operating pressures, up to 40 bar (580 Psi abs.). At first, the range of R-410A compressor capacity was limited to 25 kW cooling capacity. Later on in 2004, as a result of new research and development, the average cooling capacity of a roof-top unit with R-410A reached.

In the U.S. and California, the share of R-410A used in brand new equipment approached 10% in 2004. However, a rapid growth is forecasted and R-410A will be the major refrigerant used in stationary AC.

3.1.4 Fugitive emission rate and recovery efficiency

Table 1.9 summarizes the fugitive emission rate and the recovery efficiency at end of life for stationary equipment. The recovery efficiency is directly related to the refrigerant charge. Namely, the higher the refrigerant charge, the higher the recovery rate. This may sound optimistic, but corresponds to the thresholds in a number of regulations, i.e., recovery is made mandatory above a certain refrigerant charge value.

Table 1.9 Fugitive emission rates and recovery efficiency in 2004.

	Portable	*Ductless Splits<5 kW*	*Ductless Splits>5 kW*	*Indoor Packaged*	*Window*	*Roof Top*	*Ducted Splits<17 kW*	*Ducted Splits>17kW*
Annual emission rate	2%	5 %	10%	5%	2%	5%	5%	5%
Recovery efficiency	0%	0%	30%	50%	0%	70%	50%	70%

3.2 Chillers

3.2.1 Data sources and calculation method

An additional parameter to take into account when dealing with chillers is the refrigerant charge ratio, which is used here in the calculation process. The refrigerant charge ratio depends on the type of chiller: volumetric (reciprocating, screw, and scroll) or centrifugal. These two categories exist as specific separate parts within the RIEP database.

Data for California are not available. Market data are derived from the BSRIA marketing study [BSR05]. A ratio (*Population of California/Population of USA*) is applied to the U.S. available statistics. The average cooling capacity of centrifugal chillers is 3 MW and the average cooling capacity of a volumetric chiller is 350 kW. Table 1.10 shows the chiller market in 2004.

Table 1.10 Chiller market in 2004 [BSR05].

Chillers market	*USA*	*California*
Centrifugal	3402	418
Other Chillers	14980	1845

3.2.2 Installed base of chillers

The U.S. installed base of chillers is the derivation of chiller market for the last 20 years, taking into account the extinction curve of these units.

Table 1.11 U.S. installed base of chillers, from 1990 to 2004.

Year	*Centrifugal chillers*	*Other chillers (volumetric)*
1990	9,309	13,553
1991	9,465	13,942
1992	9,781	14,389
1993	10,430	14,941
1994	11,360	15,687
1995	12,268	16,310
1996	13,122	17,240
1997	13,571	18,175
1998	13,571	18,833
1999	13,748	19,247
2000	13,982	19,737
2001	14,145	20,295
2002	14,246	20,886
2003	14,281	21,500
2004	14,309	22,175

3.2.3 Analysis of the R-123 uptake

Refrigerants used in centrifugal chillers (CFC-11, CFC-12, HFC-134a, and HCFC-123) are significantly different from those used in volumetric chillers (HCFC-22, HFC-134a, R-410A, and R-717 (ammonia)). For these two categories, refrigerant types are analyzed below.

Table 1.12 Centrifugal chillers, refrigerant distribution in market of brand new equipment.

Year	*% of R-134a*	*% of R-11*	*% of R-12*	*% of R-123*
1990	0	70	30	0
1991	0	70	30	0
1992	5	70	25	0
1993	10	50	20	20
1994	20	30	10	40
1995	40	0	0	60
1996	40	0	0	60
1997	40	0	0	60
1998	40	0	0	60
1999	40	0	0	60
2000	40	0	0	60
2001	40	0	0	60
2002	40	0	0	60
2003	40	0	0	60
2004	50	0	0	50

Table 1.13 Volumetric chillers, refrigerant distribution in market of brand new equipment.

Year	*% of R-134a*	*% of R-717*	*% of R-22*	*% of R-410A*
1990	0	2	98	0
1991	0	2	98	0
1992	0	2	98	0
1993	1	2	97	0
1994	2	2	96	0
1995	3	2	95	0
1996	3	2	95	0
1997	3	2	95	0
1998	3	2	95	0
1999	3	2	95	0
2000	3	2	95	0
2001	3	2	95	0
2002	3	2	95	0
2003	3	2	95	0
2004	3	2	90	5

3.2.4 Refrigerant charge, emission factor, and recovery efficiency

The refrigerant charge corresponding to the refrigerating capacity varies significantly with the technology and the liquid density. The ratio of refrigerant charge per one kilowatt of refrigerating capacity is shown in Table 1.14 for centrifugal chillers.

Table 1.14 Charge / cooling capacity ratio for centrifugal chillers [TEA04].

Refrigerant	*CFC-11*	*CFC-12*	*HCFC-123*	*HFC-134a*
Charge/cooling capacity ratio (kg/kW)	0.28	0.35	0.40	0.36

When considering volumetric chillers, two technical options are available for the evaporator, either dry expansion evaporator (dry Hex) or flooded evaporator used with centrifugal chillers. As shown in Table 1.15, the evaporator technology influences the refrigerant charge per kW of refrigerating capacity.

Table 1.15 Charge / cooling capacity ratio for volumetric chillers [TEA04].

Refrigerant and chiller type	*Evaporator type*	*kg/kW*
HCFC-22 and HFC-134a screw and scroll	Dry Hex	0.27
R-410A and R-407C scroll	Dry Hex	0.27
HCFC-22 and HFC-134a screw	Flooded	0.35
HCFC-22 reciprocating	Dry Hex	0.26
R-717 screw or reciprocating	Dry Hex	0.04 to 0.2
R-717 screw or reciprocating	Flooded	0.2 to 0.25

Table 1.16 Annual emission rate for centrifugal chillers.

Centrifugal chillers	*R-134a*	*R-11*	*R-12*	*R-22*	*R-123*
Annual emission rate (%)	5	10	10	10	3

Table 1.17 Recovery efficiency.

Chiller type	*Centrifugal chillers*	*Volumetric chillers*
Recovery Efficiency (%)	80	50
Annual emission rate (%)	(See Table 1.16)	5

3.3 Results of calculations: refrigerant bank and emissions

3.3.1 Refrigerant bank

Figure 1.57 and Figure 1.59 present refrigerant bank evolution from 1990 to 2004 in stationary air conditioning and chillers.

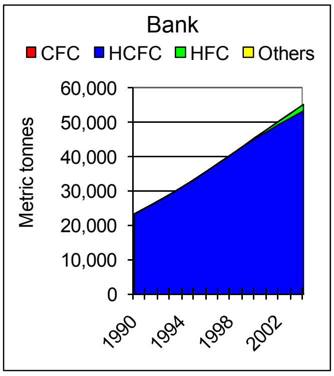


Figure 1.58 Refrigerant bank in stationary AC.

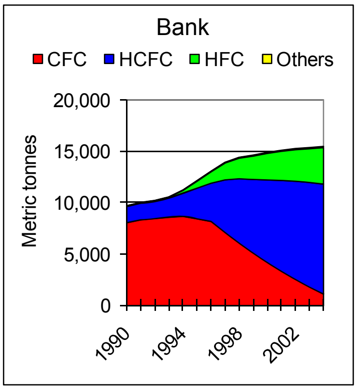


Figure 1.59 Refrigerant bank in chillers.

R-22 dominates refrigerant bank in stationary AC sector where a remarkable growth rate is observed between years 1990 and 2004. In 2004, the installed base of stationary equipment is filled with more than 53,000 tonnes of R-22. Chillers contribute to 16,000 tonnes of refrigerant bank, with HCFCs accounting for 80% of the chiller refrigerant bank in 2004. The primary HCFCs used in chillers are R-123 and R-22. CFCs are still used in some chillers, but this use will be null in a few years, as older systems are replaced or retrofitted.

3.3.2 Refrigerant emissions

Figure 1.60 and Figure 1.61 illustrate refrigerant emissions evolutions from 1990 to 2004 in stationary air conditioning and chillers.

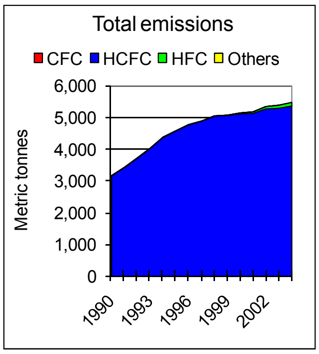


Figure 1.60 Refrigerant emissions from stationary AC.

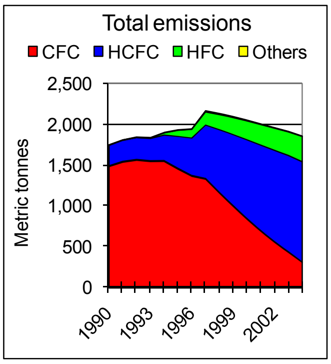


Figure 1.61 Refrigerant emissions from chillers.

Following the trend of the bank, the emissions in stationary AC sector reach 5,500 tonnes in 2004. CFC emissions from chillers are still significant, with nearly 500 tonnes in 2004.

3.3.3 Refrigerant CO_2 equivalent emissions

Figure 1.62 and Figure 1.63 present refrigerant CO_2 equivalent emissions from 1990 to 2004 in stationary air conditioning and chillers. CO_2 equivalent emissions from chillers were significant in the beginning of the 1990s due to the high GWP of R-12 and R-11 compared to R-134a and R-22. In the stationary AC sector, CO_2 equivalent emissions total 8 million of CO_2 metric tonnes.

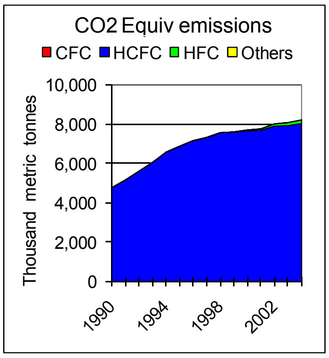


Figure 1.62 CO_2 equivalent emissions from stationary AC.

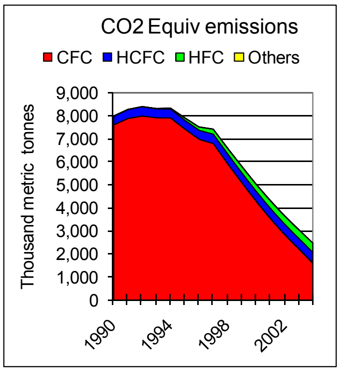


Figure 1.63 CO_2 equivalent emissions from chillers.

3.4 Scenarios to forecast refrigerant emission from unitary air conditioners and chillers

3.4.1 Scenarios assumptions

3.4.1.1 Business as usual scenario (scenario 1): Scenario 1: Business As Usual

Usual practices and emission rates are kept unchanged for the next 16 years. The recovery efficiency is not improved. Nevertheless, the regulation corresponding to the refrigerant phase-out is taken into account for refrigerant replacement. No effort is made to perform the retrofit of HCFCs during this period.

3.4.1.2 Scenario 2

Some improvements are made to reduce equivalent CO_2 emissions of refrigerants:

- The system leak tightness is improved by choosing more reliable components
- The recovery efficiency is improved at servicing and end of life. Recovery begins on low charge equipment where it was not done before
- Technologies permitting to reduce the refrigerant charge are chosen (compactness, dry heat exchanger)
- Lower GWP refrigerants are preferred when possible
- Retrofit of R-22 starts from 2012 and continues for a period of 12 years.

3.4.1.3 Scenario 3: Partial phase-out of high GWP HFCs

Efforts are made with the same improvements mentioned in the second scenario. Technological options are chosen in order to decrease refrigerant charge and GWP when possible. A new blend is introduced with a low GWP (100). In 2020, this blend will cover 50% of market of brand new equipment.

Table 1.18 Fugitive emission for different scenarios.

Emission rate (%)	*2004*	*2020 – Scenario 1*	*2020 - Scenario 2*	*2020 - Scenario 3*
Portable	2	2	2	2
Splits (Ductless < 5 kW)	5	5	5	5
Splits (Ductless > 5 kW)	10	10	8	8
Indoor Packaged	5	5	5	2
Window	2	2	2	2
Roof Top	5	5	5	5
Ducted Splits < 17 kW	5	5	5	5
Ducted Splits > 17 kW	5	5	5	5

The level of emission rate is low in stationary AC equipment, compared to commercial and industrial refrigeration. Unitary AC systems are compact and most of components are molded. The number of fittings is limited and the sensitivity to leaks is low.

Table 1.19 Recovery efficiency.

Recovery efficiency (%)	*2004*	*2020 – Scenario 1*	*2020 - Scenario 2*	*2020 - Scenario 3*
Portable	0	0	30	30
Splits (Ductless < 5 kW)	0	0	30	30
Splits (Ductless > 5 kW)	30	30	50	50
Indoor Packaged	50	70	70	80
Window	0	0	30	30
Roof Top	70	70	70	80
Ducted Splits < 17 kW	50	50	70	70
Ducted Splits > 17 kW	70	70	70	70

Recovery of refrigerant at the end of life is not performed on low charge equipment such as windows, portable or split systems. In scenarios 2 and 3, it is considered that refrigerant recovery starts in 2010 for these small unitary systems.

Table 1.20 Refrigerant charge ratio.

Ratio kg/kW	*2004*	*2020 – Scenario 1*	*2020 - Scenario 2*	*2020 - Scenario 3*
Centrifugal chillers	0.3	0.3	0.25	0.25
Volumetric chillers	0.35	0.35	0.3	0.3

Flooded evaporators have a large content of refrigerant while dry expansion heat exchangers are more adapted to reduce the refrigerant charge. Moreover, air cooled condenser can be built with micro-channel flat tubes to limit the refrigerant charge.

3.4.2 Refrigerant bank

Figure 1.64 and Figure 1.65 present refrigerant bank evolution from 1990 to 2004 in stationary air conditioning and chillers. No significant change in the bank growth is observed for different scenarios. In 2010, new systems are no longer filled with HCFCs. R-22 is replaced by HFCs, especially R-410A.

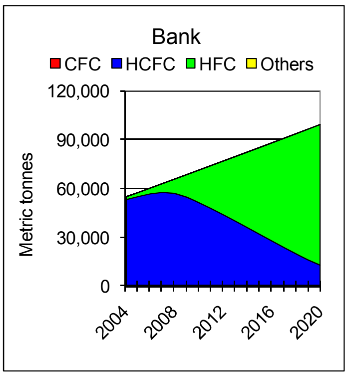


Figure 1.64 Refrigerant bank in stationary AC.

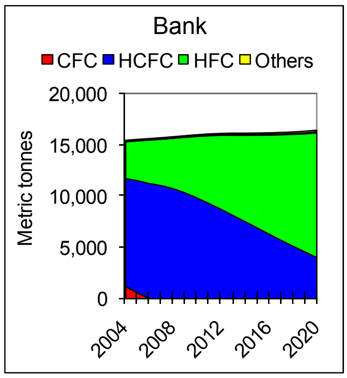


Figure 1.65 Refrigerant bank in chillers.

Figure 1.64 shows that the bank growth in 2020 in stationary AC units is still high and could reach 100,000 tonnes. In chillers the bank stabilizes at 15,000 tonnes. The market growth of chillers is absorbed by the decrease of the refrigerant charge in new chillers.

3.4.3 Refrigerant emissions

Figure 1.66 to Figure 1.68 present refrigerant emissions evolution from 2004 to 2020 in the stationary air-conditioning sector.

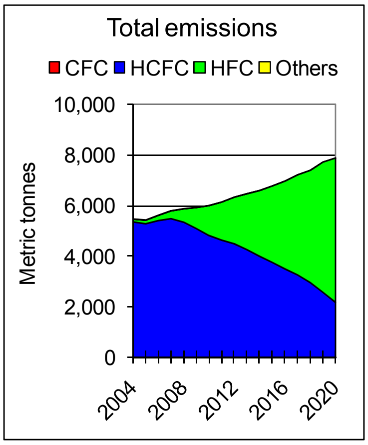


Figure 1.66 Scenario 1, emissions from stationary AC.

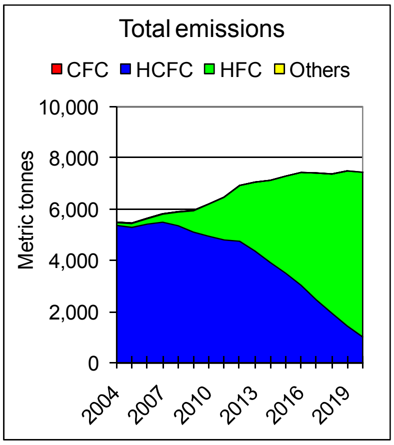


Figure 1.67 Scenario 2, emissions from stationary AC.

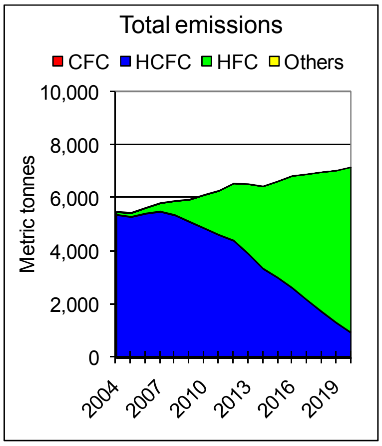


Figure 1.68 Scenario 3, emissions from stationary AC.

In 2004, the level of refrigerant emissions (fugitive and end of life) is 5,500 metric tonnes for stationary AC and consists mainly of HCFCs. In the BAU scenario, refrigerant emissions reach 8,000 in 2020 due to the bank increase.

In Scenarios 2 and 3, improved recovery of refrigerant is considered at the equipment end of life. Nevertheless, the side effect of R-22 retrofit causes a rapid increase in HCFC emissions after 2010. Retrofit of R-22 generates anticipated end of life emissions. In 2020, HCFC emissions are lower and the total amount of refrigerant release to the atmosphere is limited to 7,500 metric tonnes for scenario 2 and 7,000 for Scenario 3. Figure 1.69 to Figure 1.71 present refrigerant emissions evolution from 2004 to 2020 in chillers.

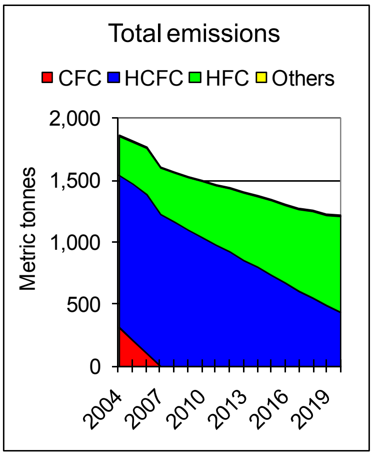


Figure 1.69 Scenario 1, emissions from chillers.

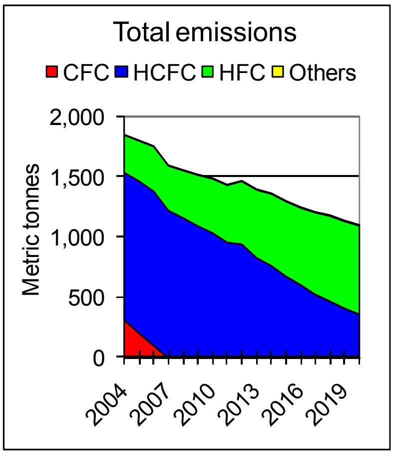


Figure 1.70 Scenario 2, emissions from chillers.

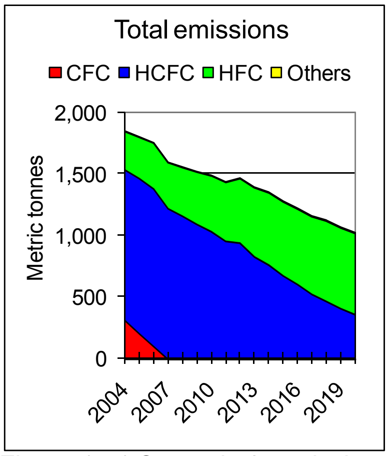


Figure 1.71 Scenario 3, emissions from chillers.

The new generation of chillers, filled with R-134a and R-123 are more leak tight than CFC chillers. These improvements, along with a better recovery efficiency at end of life, lead to a reduction in refrigerant emissions of 1,000 metric tonnes in 2020.

3.4.4 Refrigerant CO_2 equivalent emissions

Figure 1.72 to Figure 1.74 present CO_2 equivalent emissions evolution from 2004 to 2020 in the stationary air-conditioning sector.

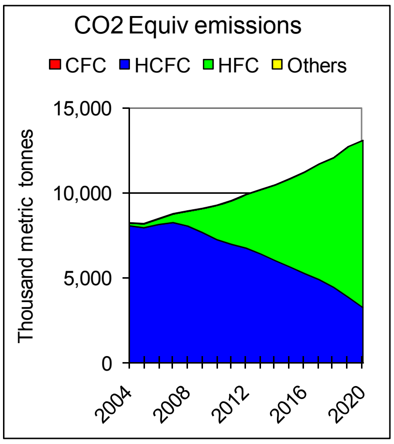


Figure 1.72 Scenario 1, CO_2 equiv. emissions in stationary AC.

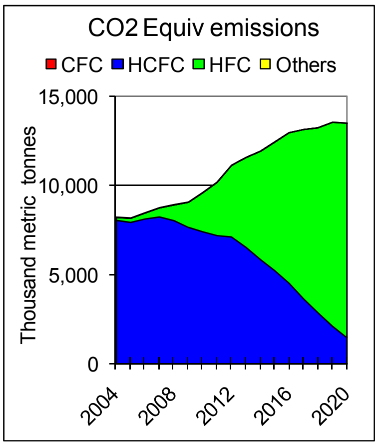


Figure 1.73 Scenario 2, CO_2 equiv. emissions in stationary AC.

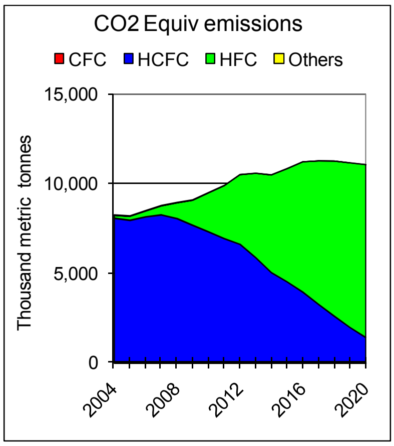


Figure 1.74 Scenario 3, CO_2 equiv. emissions in stationary AC.

R-22 GWP is 1500 and R-410A GWP is nearly 2000. The introduction of R-410A on the market has an impact on CO_2 equivalent emissions in stationary AC. As shown in scenario 2, the retrofit of R-22 increases CO_2 equivalent emissions because of higher GWP of HFCs. In 2020, the level of emission reaches 13 million metric tonnes CO_2 equivalent. In scenario 3, a new refrigerant blend with a very low GWP is assumed to be available in 2012, and filled in 50% of new equipment in 2020. Therefore, CO_2 equivalent emissions in 2020 will decrease to 11 million metric tonnes CO_2 equivalent. Figure 1.75 to Figure 1.77 present CO_2 equivalent emissions evolution from 2004 to 2020 in chillers.

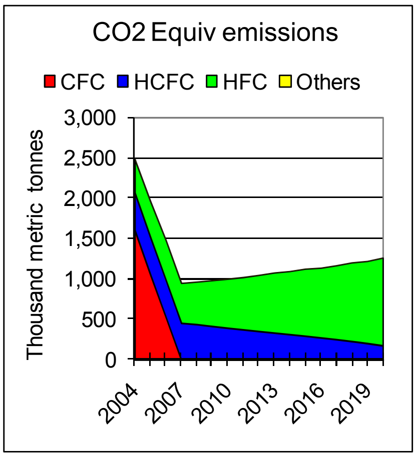


Figure 1.75 Scenario 1, CO_2 equivalent emissions in chillers.

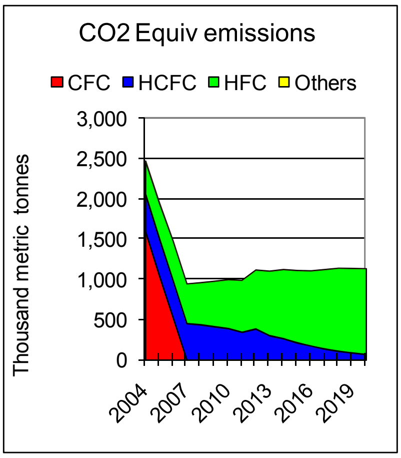


Figure 1.76 Scenario 2, CO_2 equivalent emissions in chillers.

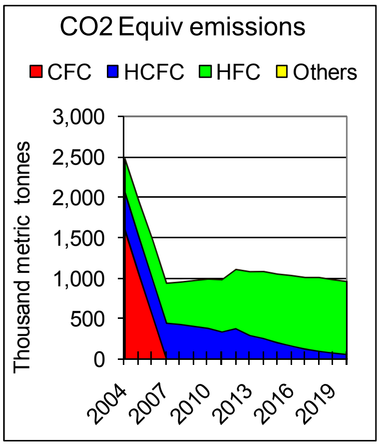


Figure 1.77 Scenario 3, CO_2 equivalent emissions in chillers.

The Scenario 3 observation concerns the continuing phase-out of CFCs used in centrifugal chillers. The phase-out will decrease CO_2 equivalent emissions. R-123 has a low GWP, and R-134a GWP is “only” 1300 (compared to R-12 (GWP 8600) and R-11 (GWP 4600)). In Scenario 3, the level of CO_2 equivalent emissions is limited to 1 million metric tonnes in 2020.

3.5 Refrigerant demand and recovery

Figure 1.78 and Figure 1.79 present the comparison of R-22 demand recovery for the period from 2010 to 2020 in both stationary AC and chillers sectors.

Figure 1.78 HCFC demand and recovery in business as usual scenario.

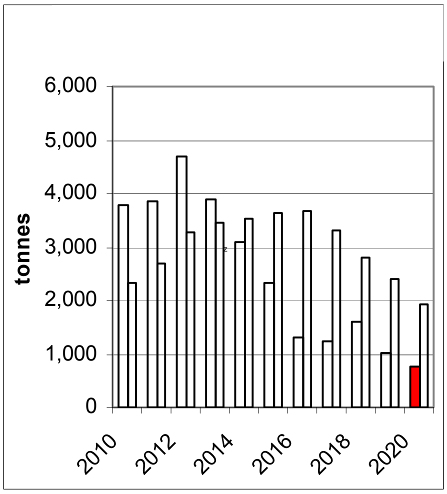


Figure 1.79 HCFC demand and recovery in scenario 2.

In the Business As Usual (BAU) scenario (see Figure 1.78), HCFC demand reaches twice the recovery of HCFCs for the year 2013. This figure shows that the recovery is not sufficient to feed the market for servicing demand.

In scenario 2 (see Figure 1.79), the retrofit of R-22 in large capacity equipment starts in 2010, for a 15-year period. Between the years 2010 and 2013, R-22 recovery covers almost 2/3 of the servicing demand, then after 2014 the recovery exceeds the demand.

Scenario 2 shows that R-22 phase-out should not be a problem in the stationary and chiller air conditioning sectors if the recovery is efficiently carried out in these sectors and the R-22 retrofit is performed.

4 Refrigerant emissions from the industrial sector

This section describes refrigerant use, banks, and emissions from the non-retail food industry, cold storage facilities, and industrial process cooling.

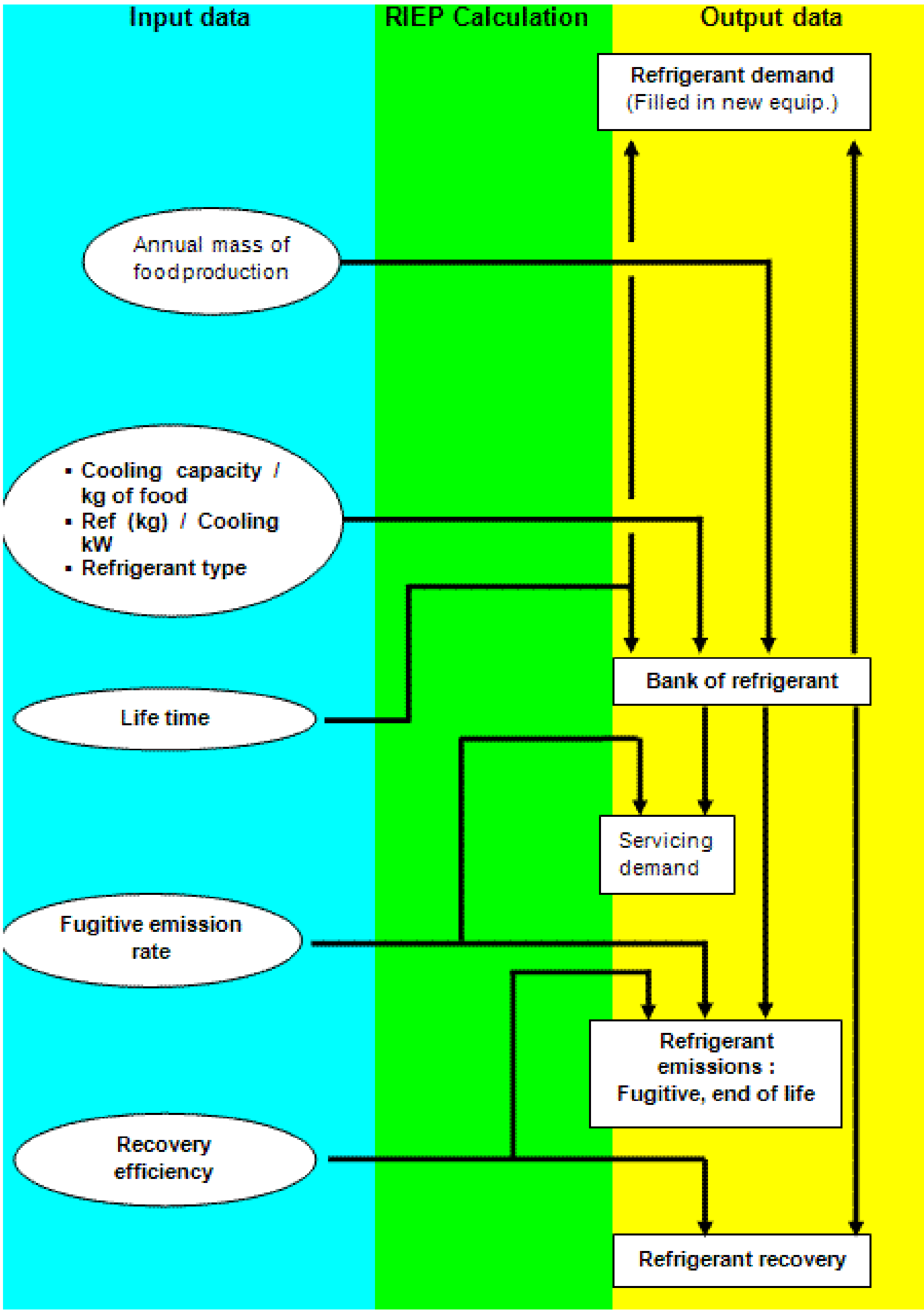


Figure 1.80 Calculation steps for emission forecasts in the food industry.

4.1 Food industry and cold storage

Cooling and freezing processes in the food industry are applied to all types of meat, to dairy products, wines, beers, soft drinks, frozen food, and chocolate. Flake ice is used for the cooling of fresh fish.

4.1.1 Calculation method - RIEP

The methodology is based on food products. This choice has been made since the "Food and Agricultural Organization (FAO) updates the database every year for food products produced and sold in each country. These data are needed for the "Inventories of the worldwide fleets of refrigerating and air-conditioning equipment in order to determine refrigerant emissions".

Vegetables and fruits are taken into account in the cold storage and warehouse calculations. This choice has been made due to the very large difference that exists between crops and refrigerated vegetables and fruits. A calculation performed for the storage volume avoids large overestimates.

Food domains taken into account are those of major importance: meat, dairy products, wines, beers, fishes, and frozen food. Cold storage is taken into account via two different routes:

- at the process facilities using a ratio between the cooling process and the properties of the product; it is therefore integrated in the cooling capacity dedicated to products,
- for general cold storage purposes, where needs are calculated separately.

For the Californian inventory, chocolate processes are not taken into account due to lack of information, but their contribution to the total bank and emissions is not significant. The only available information found for the U.S. is the following [SCN08]:

***Shipments** (million lbs)*	
1994	*2.859*
1995	*3.02*
1996	*3.1*

The consumption (*Consumption = Shipments + Imports – Exports*) [SCN08] is also given for another three years.

The global methodology (see Figure 1.80) used to determine the refrigerant inventories and emissions is based on data available for the production of all types of refrigerated and frozen food. The different food products are cooled or frozen at production sites, transported in refrigerated transport means, and then possibly stored in general warehouses. So, the food production data is used to establish the refrigerating equipment installed in the food industries.

The calculation steps are as follows:

- Analysis of the usual process design of a slaughterhouse, dairy, brewery, etc… to determine the installed refrigerating capacity
- Definition of typical ratios of refrigerant charge referenced to the refrigerating capacity and the temperature level
- Definition of the type of refrigerants selected, which selection depends on the temperature level and on the type of country
- Calculation of the refrigerant bank

- Calculation of the refrigerant demand for new equipment (based on the equipment lifetime and the bank)
- Determination of national or regional emission factors applicable to the bank, yielding emissions.

4.1.2 Calculation of the cooling capacity and the refrigerant charge

Calculations are performed for the seven following sub-domains:

1. meat industry
2. dairy industry
3. wine and beers
4. flake ice for fresh fish
5. frozen food
6. warehouses
7. soft drinks

Annex 4 presents detailed calculations for each sub-domain where the ratio between the cooling capacity and the annual production of a given product (kW/kg) is defined. Figure 1.81 summarizes the methodology and describes the relation between the annual production of refrigerated and frozen food and the refrigerant bank for all sub-domains.

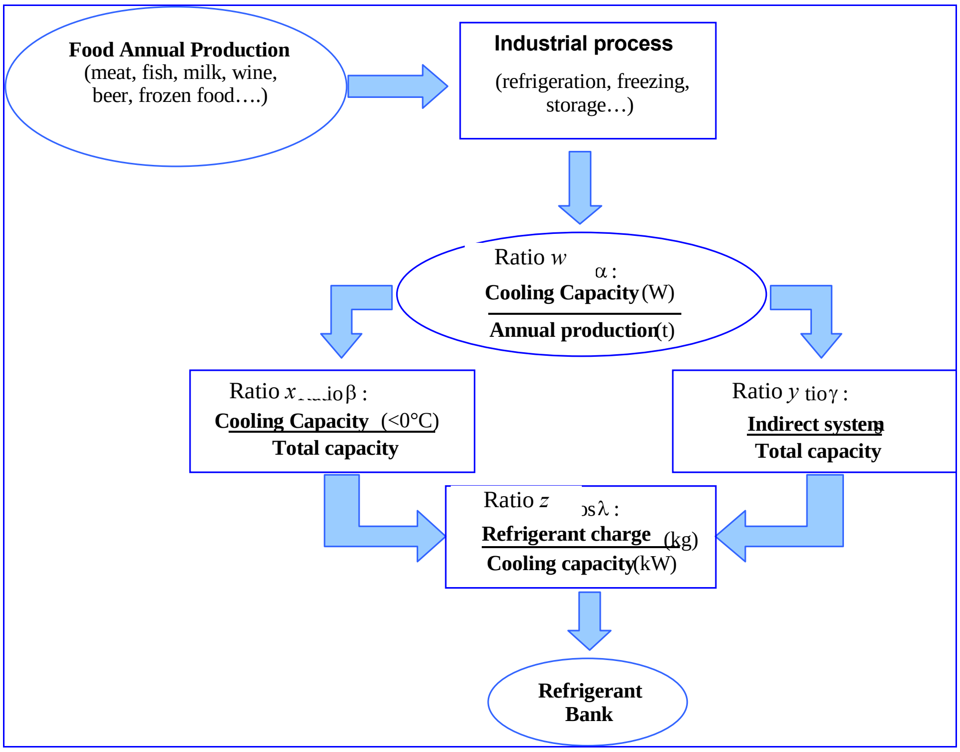


Figure 1.81 Methodology for each type of refrigerated/frozen food industry.

The detailed studies of the processes applied in each sub-domain (see Annex 4) allows determining the ratio between the total cooling capacity and the annual mass production of each sub-domain:

$$w = \text{Cooling Capacity (W) / Annual production (t)} \quad (1.3)$$

Taking into account the division the cooling capacities for low and medium temperatures, which are again dependent on the cooling process, for each sub-domain the ratio x *is defined:*

$$x = \text{Low Temp. Cooling Capacity } (<0°C) / \text{Total cooling capacity} \quad (1.4)$$

Depending on the sub-domain and the technology, the indirect systems form a different part in the total. *y* is defined as the ratio of the cooling capacity of indirect systems and the total cooling capacity (direct + indirect):

$$y = \text{Cooling Capacity of indirect systems} / \text{Total capacity} \quad (1.5)$$

The ratio *z* refers the refrigerant charge to the cooling capacity while considering the temperature level, and the technology (indirect or not).

$$z = \text{Refrigerant charge (kg)} / \text{Cooling capacity (kW)} \quad (1.6)$$

4.1.2.1 The cooling capacity per mass of product and by level of temperature

Ratios defined in Equations (9.1) and (9.2) are presented in Table 1.21 (see also Annex 4 for a justification).

Table 1.21 Ratios *w* and *x* for the different sub-domains.

	W **Cooling Capacity / mass of processed product**	***x*** **Low T Cooling Capacity / Total cooling capacity**
Meat industry	43 W/t	0.3
Dairy industry	12.9 W/t	0.2
Wine and beers	20.5 W/t	0
Flake ice for fresh fish	11.9 W/t	1
Soft drinks	4 W/t	0
Frozen food	35.8 W/t	1
Warehouses	33 W/m^3	0.7

The freezing capacity for meat is included in the calculations for the amount of frozen products globally. The cooling capacity in the meat industry is only defined for the production of fresh meat.

4.1.2.2 The Cooling capacity of indirect systems

Values of the *y* ratio as defined in Equation (1.5) are given in Table 1.22 for each sub-domain, for the year 2004. This ratio is year dependent.

Table 1.22 Ratio of indirect systems in new equipment in 2004.

Industry	*y = Cooling Capacity of indirect systems / Total capacity*
Meat industry	0.15
Dairy industry	0.3
Wine and beers	0.15
Flake ice for fresh fish	0
Soft drinks	1
Frozen food	0.25
Warehouses	0.15

4.1.2.3 The refrigerant charge

Values of the *z* ratio as defined in Equation (1.6) are given in Table 1.23. This ratio is given for medium and low temperature, for both direct and indirect systems. Because of lower liquid density, these ratios have to be divided by a factor of 2 for ammonia.

Table 1.23 Refrigerant charge referred to the cooling capacity.

System	*z = Refrigerant charge (kg) / Cooling capacity (kW)*
Med Temp. Direct system	*5.5*
Low Temp. Direct system	*8.8*
Med Temp. Indirect system	*1*
Low Temp. Indirect system	*1.5*

4.1.3 Type of refrigerants

Ammonia (R-717) is widely used in the domain of industrial refrigeration (50% to 60% in U.S.).

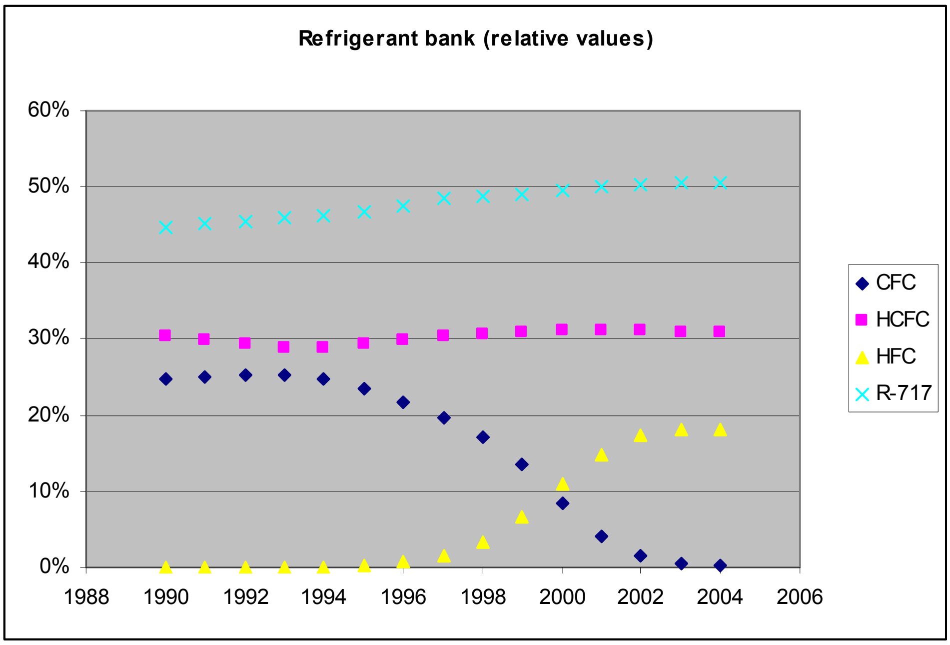


Figure 1.82 Evolution of refrigerant bank in industrial refrigeration.

4.1.4 Other characteristics

The average lifetime is used to establish the law of end-of-life of equipment.

Table 1.24 Complementary data necessary to perform the RIEP calculations.

Year 2004	*California*
Average equipment lifetime	*30 years*
Annual fugitive emission rate	*10%*
Percentage of charge emitted before servicing	*30%*
Recovery efficiency	*70%*

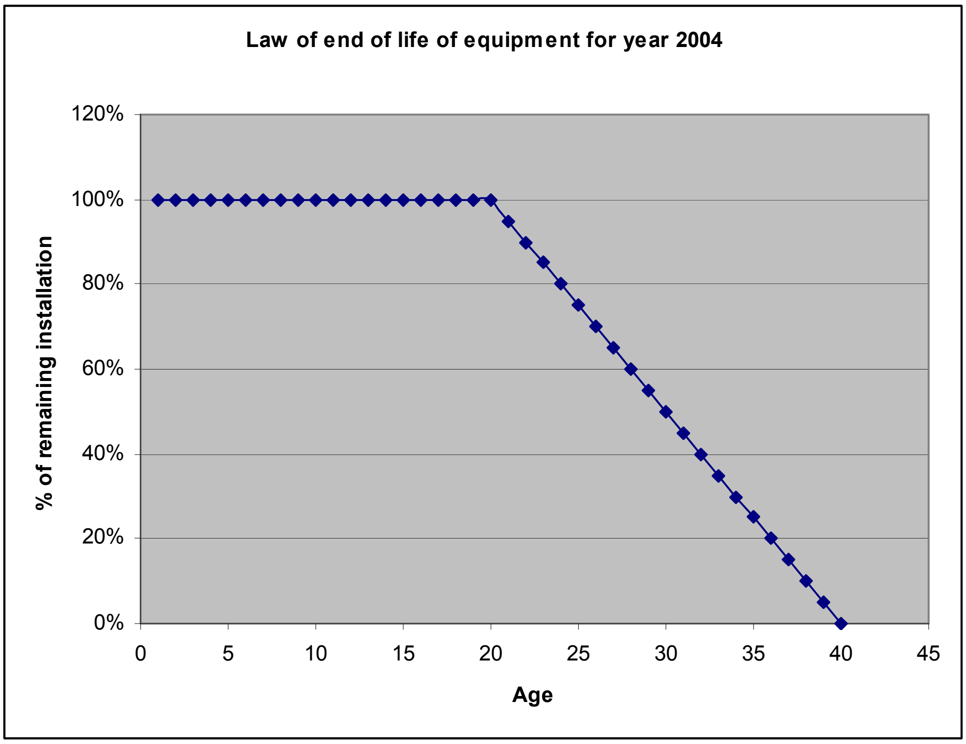


Figure 1.83 Law of end of life of refrigeration equipment.

4.1.5 Production data for food products

4.1.5.1 Production data is available for the following sub-domains.

Meat industry

Data on production of meat is available for the USA from the FAO [FAO08] and USDA [USD01]. For California, such data is available from the United States Department of Agriculture (USDA) for many years, except for chicken meat. A compilation of both sources allowed establishing the total meat production for California.

Table 1.25 Meat production in the U.S. and California.

Meat production (t)	*1990*	*2004*
USA	39,747,784	49,966,545
California	2,058,102	2,637,806

Dairy industry

The dairy industry covers the production of milk, butter, cheese, and cream. California is a leading state in milk production, covering approximately 20% of the total national milk production.

Both United States Department of Agriculture (USDA) [USD02] and FAO [FAO08] sources were used in order to compute the total industry dairy production.

Historical data for the U.S. are available. For those unavailable values for the state of California, ratios for cheese, butter, and cream production over the USA production were established (eg.

Cheese Production U.S. (t) /Milk Production U.S. (t)) were established, and applied to the milk production in California in order to compute the cheese and other unavailable dairy products historical statistics for California.

Table 1.26 Dairy production.

Food Production		*1990*	*2004*
Milk production (t)	*USA*	67,005,118	77,534,358
	California	9,501,399	16,540,246
Cheese production (t)	USA	2,748,516	4,024,793
	California	317,054	903,928

Wine and beers

California produces more than 90 percent of total U.S. wine production [WIN]. Production data is available for California from the Wine Institute [WIN] for years 1986 to 2004. The FAO provides statistics for the USA starting from the year 1961. Both sources were used to compute the historical production for California.

Table 1.27 Wine production.

Wine production (t)	*1995*	*2004*
USA	1,654,354	2,304,817
California	1,502,967	2,070,724

Information on the beer production is available for the USA from the FAO [FAO08] and the Beer Institute [BEI08]. The total USA production of beer is about 23 million of metric tons. The Beer Institute gives data for California for many years. In 2004, the California beer production accounts for about 11% of the global U.S. beer production. This ratio is almost constant for years 1999 to 2006, and was therefore applied to estimate historical values for California.

Flake ice for fresh fish

In the calculations, flake ice production is directly linked to the daily catch of sea and river fish. NOAA's National Marine Fisheries Service provides statistics on domestic commercial landings [NMF07] for the USA and California, and the FAO gives global production statistics.

Refering to the NOAA's data, the fish landing in California accounted for 31% of the total U.S. landings in the year 1950. In the year 2005, the share of California in the total U.S. landings decreased to reach 5%, while the state of Alaska became the most important producer with 60% of the total U.S. landings. Both sources were therefore used to calculate the production of California.

Table 1.28 Fish production.

Fish production (t)	*1990*	*2004*
USA	6,096,539	5,972,841
California	435,438	233,633

Frozen food

The FAO provides the quantity of frozen food produced in the USA, i.e. 13.8 million of metric tonnes for the year 1990 and 16.3 for the year 2004. Based on the U.S. Bureau Census, about 14% of the USA frozen food industries are in California. Due to lack of information regarding the quantity produced in California, this ratio was adopted.

Warehouses.

The volume of refrigerated warehouses is available from the USDA for the USA and for California for many years. Linear interpolation is done for the years that are not given by the USDA.

Table 1.29 Warehouse capacity.

Net refrigerated warehouses capacity (m^3)	*1990*	*2004*
USA	55,578,891	73,086,802
California	6,015,404	9,137,563

Soft drinks

The U.S. Bureau Census gives the total number of establishments engaged in manufacturing soft drinks and artificially carbonated waters for the USA and for California. The ratio *Industries in California/Industries in USA* = 11% is applied to the total volume of carbonated soft drinks (CSD) manufactured by the U.S. industry, given by the Beverage Disgest [BED07].

Table 1.30 Carbonated soft drinks production.

	CSD volume (192 billion oz cases)		**CSD volume (t)**	
	1990	*2004*	*1990*	*2004*
USA	*6.1308*	*7.84384*	33,371,171	41,619,252
California	*0.674388*	*0.8628224*	3,670,829	4,578,118

4.1.6 Milk tanks

Milk tanks are installed on farms and represent a sub-domain included in the dairy domain. The calculations are specific, based on the number of milkings, which enables to define the storage volume of the milk tank. Knowing the storage volume, it is possible to define the refrigerating capacity and the refrigerant charge; the method and additional data are presented in Annex 4.

4.1.6.1 Average refrigerant charge

The average charge of refrigerant is 2.09 kg/m^3 of storage.

4.1.6.2 Characteristics

Table 1.31 Characteristics of milk tanks.

Average Lifetime (years)	*Annual Emissions (%)*	*Recovery Efficiency (%)*	*Charge Emitted before Servicing (%)*
15	5	50	30

4.1.6.3 Refrigerant type

Table 1.32 Refrigerant distribution on the market.

Year	*% of R-12 in Market*	*% of R-404A in Market*	*% of R-22 in Market*
1990	20	0	80
1991	10	0	90
1992	0	0	100
1993	0	0	100
1994	0	0	100
1995	0	0	100
1996	0	0	100
1997	0	0	100
1998	0	0	100
1999	0	10	90
2000	0	20	80
2001	0	20	80
2002	0	20	80
2003	0	20	80
2004	0	20	80

4.2 Industrial process cooling

4.2.1 Data sources and detailed calculations

Refrigerating needs in industrial processes other than food processing are multiple. They cover a broad range of temperatures. Two types or categories of refrigerating equipment have been defined and analyzed.
Chillers operating at temperature above 0°C. A large amount of this equipment is bought "from the shelf", and one only needs to define the capacity and the level of temperature. These chillers cover 55% of the refrigerating capacity needs.
Refrigerating systems particularly designed for low-temperature applications, where the process specifications are well taken into account.

In order to avoid double counting, chillers are not taken into account in this section because available data for the chiller production and the chiller demand normally merge all different chillers types, i.e., the ones for comfort cooling and the ones for industrial processes (see Section 3.2.1).

In this section only the low-temperature refrigerating systems installed in the chemical industry are considered for the following reasons:

- it is a more significant domain (except food) for low-temperature applications,
- the important industrial domains such as tire manufacturing, electronics, etc. use chillers only to cover their space cooling needs.

A thorough analysis of the installed base of a chemical company (under confidentiality agreement) has enabled the development of a typical scheme of an industrial production site. Based on this study, the low-temperature cooling capacity has been projected to all other chemical manufacturers, in order to obtain a first estimate. Characteristics are presented in Table 1.33:

Table 1.33 Refrigerant charge and cooling capacity for a chemical plant.

	Medium temperature	***Low temperature***
Cooling capacity	55%	45%
Ratio (kg/kW)	2.3	5.5
Refrigerant charge	40%	60%

Even if many installations may have a lifetime longer than 30 years - taking into account the big overhauls -, the lifetime of equipment is considered to be 15 years, i.e., the time before a significant maintenance takes place.

Table 1.34 Other characteristics of typical refrigerating systems installed in chemical plants.

Life (years) Before remodeling	*Annual Emissions (%)*	*Recovery Efficiency (%)*	*Charge Emitted before Servicing (%)*
15	10	50	30

Based on available information collected from the websites of the main chemical companies, operating globally, the French inventory for chemical industries has been developed. The Californian inventory for this sector was then derived from the French one by applying GDP ratios.

Table 1.35 describes the evolution of refrigerants in use for the new refrigerating systems installed in the chemical industry.

Table 1.35 Refrigerant distribution in industrial processes.

Year	*% of R-404A*	*% of R-11*	*% of R-12*	*% of R-22*	*% of R-717*
1990	0	4	34	60	2
1991	0	4	34	60	2
1992	0	4	34	60	2
1993	0	3	34	61	2
1994	0	1	25	72	2
1995	0	0	0	98	2
1996	0	0	0	98	2
1997	0	0	0	98	2
1998	0	0	0	98	2
1999	0	0	0	98	2
2000	0	0	0	95	5
2001	0	0	0	95	5
2002	5	0	0	90	5
2003	10	0	0	85	5
2004	10	0	0	85	5

4.3 Results of calculations: refrigerant banks, and emissions

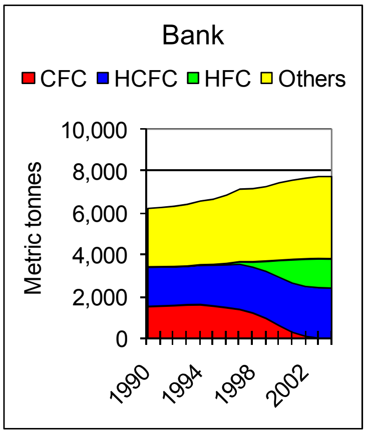


Figure 1.84 Refrigerant bank in industry.

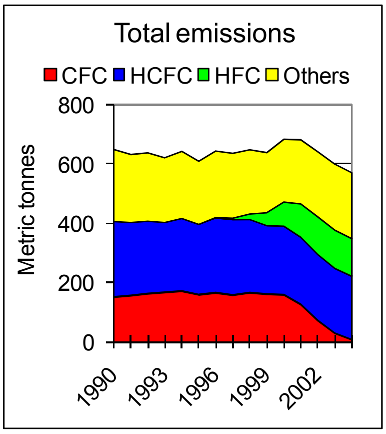


Figure 1.85 Refrigerant emissions in industry.

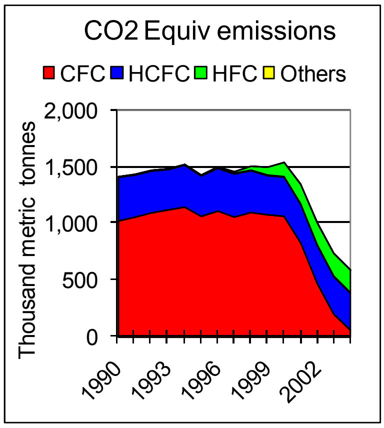


Figure 1.86 CO_2 equivalent emissions in industry.

Ammonia is widely used in industrial refrigeration. The bank of ammonia approaches 4,000 metric tonnes in 2004 (shown in Figure 1.84 as "Others"). Emissions of HCFC totals near 250 metric tonnes in 2004. The phase-out of CFCs, when started in 1996 with the retrofit of existing systems, continues to have a high impact on reducing CO_2 equivalent emissions.

5 Overall refrigerant bank and emissions

The mobile air-conditioning sector as well as the refrigerated transport sector, out of the scope of the present report, have been established nevertheless in order to have an overall picture of refrigerant inventories in California. Moreover, the inventories have also been done at the U.S. level in order to compare refrigerant demands per refrigerant as derived from this work and the refrigerant sales per refrigerant as possibly known by refrigerant manufacturers. No data have been declared so far by those manufacturers.

5.1 Refrigerant bank

The refrigerant bank is presented here by refrigerant types. The type “others” is ammonia mainly used in the food industry.

- Figure 1.87 shows the evolution of the refrigerant bank per type of refrigerants. It is clearly seen that the refrigerant bank follows the same growth as the population and GDP shown in Figure 1.1 and Figure 1.2.
- In 2004, the refrigerant bank in California is estimated at 116,000 metric tonnes and it was about 65,000 tonnes in 1990.
- CFCs are not in wide use anymore, with insignificant contribution in 2004, but R-22 and HFCs have been continuously used since 1995.
- The dominant refrigerant in the overall bank is R-22 that constitutes 57% of the bank.

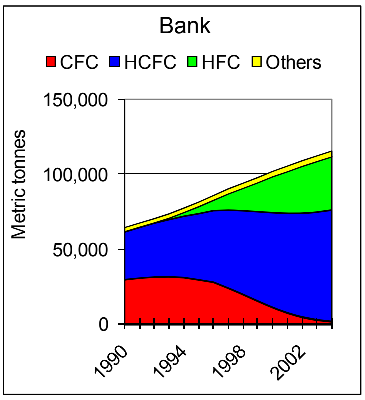


Figure 1.87 California refrigerant bank per refrigerant types

When analyzing the repartition of refrigerants in the different sectors, stationary air conditioning constitutes clearly the dominant sector with nearly 55,000 metric tonnes and so 47% of the total bank of refrigerants. Within this sector, rooftops constitute the dominant part of air-to-air systems.

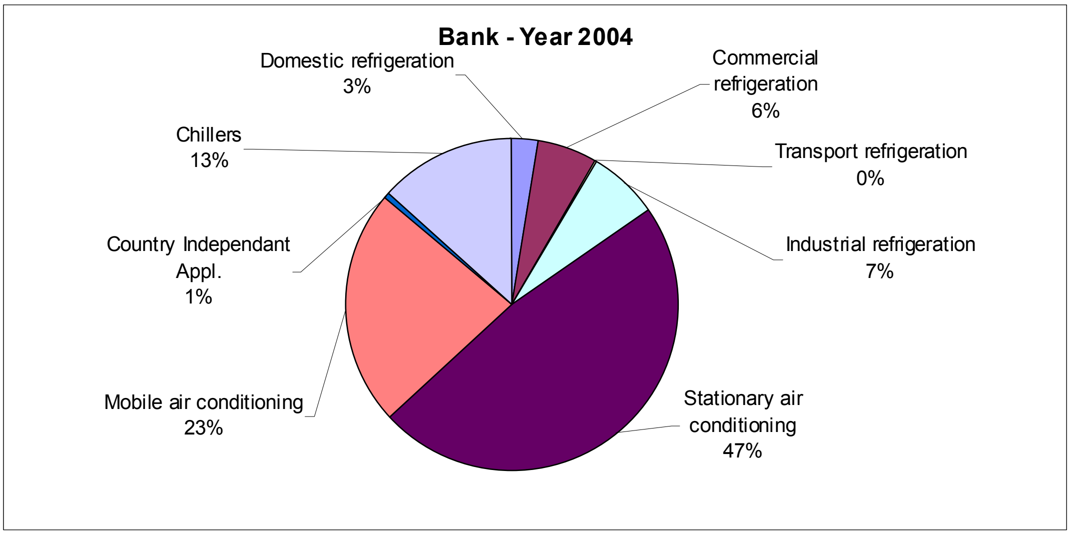


Figure 1.88 Structure of the refrigerant bank per sectors.

Note: the "country independent application" corresponds to refrigerated containers coming in California and are serviced. However, international refrigerated containers are difficult to attribute to countries.

5.2 Refrigerant emissions

Refrigerant emissions are defined for each sector and more precisely for each type of equipment as presented in Section 2 of the "Part1 - Refrigerant inventory and emissions for stationary systems" and in Annex 2.

- From 1990 to 1997 CFCs represent 50% of the emissions, and the other 50% being mainly HCFCs.
- From 2000, the emissions are decreasing based on the assumption of recovery at end of life (see section 9.3.4) and to the fact that new equipment requires less servicing during the first years of use. A mature servicing market with new refrigerant requires at least 15 years of operation.
- The replacement of CFCs and of some HCFCs by HFCs and a better initial containment explain the decreasing trend.

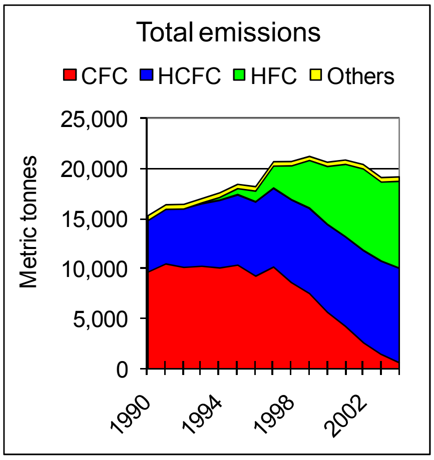


Figure 1.89 Refrigerant emissions per refrigerant type

Figure 1.90 Refrigerant emissions per application sectors shows the impact of different emission factors depending on the application sector: mobile AC that represents 23% of the bank contributes to 45% of the overall emissions. This sector has been more emissive, thanks to the use of small cans for servicing and of emissive components as the shaft seal of the open type compressor.

On the contrary Stationary AC systems whose emission factors are lower, represent 47% of the overall bank, and only 29% of the overall emissions. The main progresses have to be made for the recovery at end of life, even if lessons learnt from the field are necessary to understand what are the components to be improved in term of leak tightness.

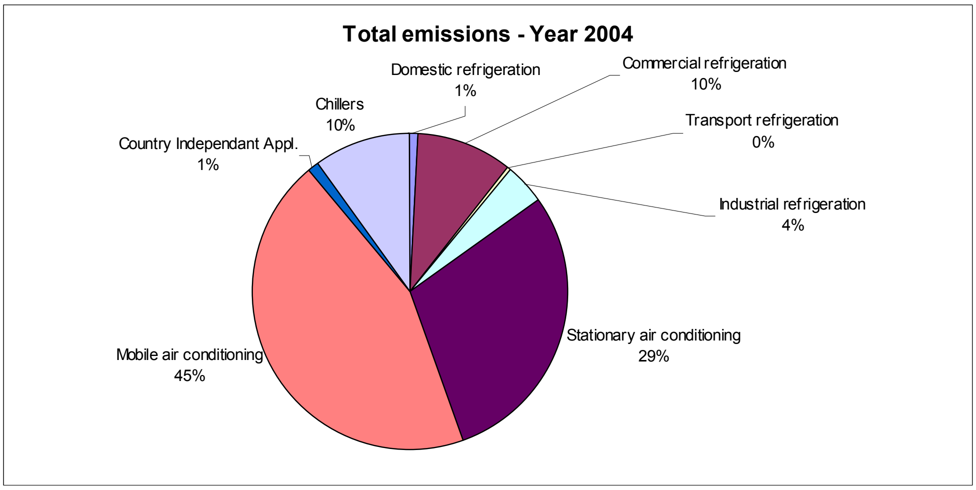


Figure 1.90 Refrigerant emissions per application sectors.

5.3 CO_2 Equivalent emissions

When looking at Figure 1.91, the paramount factor is the change from CFCs having high GWPs and specially R-12 (8,100) to HFCs mainly R-134a (1,300) resulting in a dramatic decrease of Equivalent CO_2 emissions. It is obvious that the absence of accounting of CFCs and HCFCs in the climate convention is not physically justified.

- Until 2002, the CO_2 equivalent emissions are dominated by R-12 with its high GWP and it is mainly associated with mobile air conditioning systems, domestic refrigeration and small commercial equipment.
- Note: a systematic and efficient recovery policy at end of life of equipment could have limited a very significant effect on climate change.
- In 1997, emissions reach a maximum at nearly 92 million metric tonnes CO_2 equivalent.
- In 2004, emissions drop down to 31 million metric tonnes CO_2 equivalent.

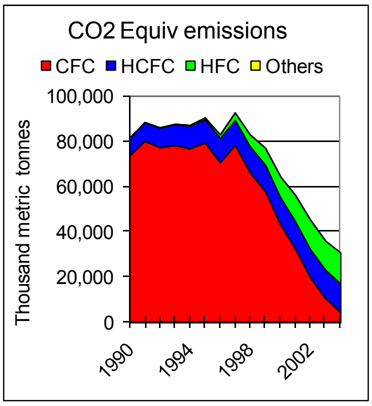


Figure 1.91 Refrigerant emissions expressed in CO_2 equivalent.

- The mobile air conditioning (MAC) sector is the first domain with 47% of CO_2 equivalent emissions in 2004, then comes stationary AC with 27% of global emissions.
- Commercial refrigeration is 6% of the refrigerant bank, but 11% of CO_2 equivalent emissions. It has to be noted that if R-404A (GWP = 3,260) is going to replace systematically R-22 (GWP = 1,500) the increase in CO_2 equivalent will be twice the one of R-22 in the next 10 years.

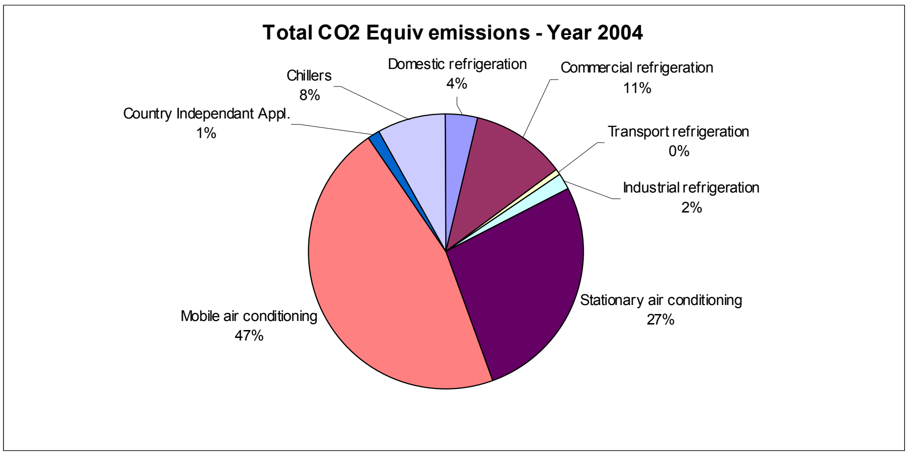


Figure 1.92 Refrigerant emissions per sectors and expressed in CO_2 equivalent.

When comparing Figure 1.90 and figure 1.92, due to the nearly nil contribution of CFCs in 2004 in the refrigerant emissions, the repartition of emissions expressed in CO_2 equivalent is not that different compared to the emissions expressed in refrigerant tonnes.

5.4 Refrigerant recovery

Figure 1.93 presents the recovered quantities from refrigeration systems at end of life, based on the assumptions taken in this report. Refrigerant distributors have provided no data.

- Refrigerant recovery is considered to be effective since 1996, with phase out of CFCs.
- Refrigerant recovery is estimated to be around 2,300 metric tonnes in California in 2004.
- These quantities include refrigerant recovered and recycled on site and refrigerant recovered and regenerated at the manufacturer plant. It is acknowledged that the need of CFCs and HCFCs have been and are motivations to recover for reuse and keep the old refrigerating systems in operation with the refrigerants no longer available on the market.

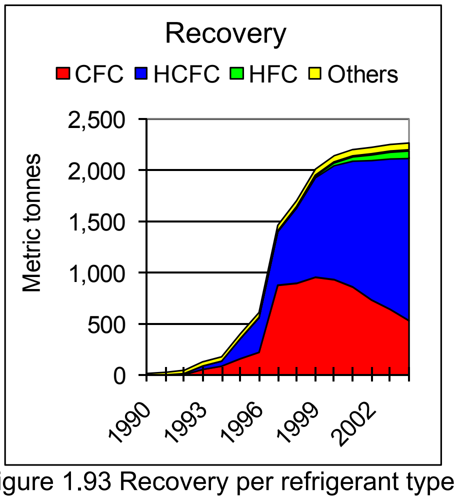


Figure 1.93 Recovery per refrigerant type.

Part 2 – ENERGY CONSUMPTION OF THE COMMERCIAL REFRIGERATION SECTOR IN CALIFORNIA

1 Description of commercial refrigeration and stores

1.1 Store categories using refrigeration equipment

Commercial refrigeration equipment is used in different types of stores, for cooled beverage delivery, and food preservation at medium or low temperature. Refrigerating equipment numbers and technologies differ significantly with store types.

Each type or category of store is characterized by a typical structure defined by the average sales surface area, the number of refrigeration equipment, and the length of refrigerated cases. Global numbers are established based on Californian statistical data or ratios taken from overall USA numbers.

In order to define a typical store layout, a field study has been carried out in the state of California over a large number of stores, brands, and sale products. Based on the field survey and technical literature analyses, sixteen categories of stores using refrigerating equipment are identified. A total number of 122 stores have been visited during the survey. Table 2.1 presents these categories as well as well the corresponding visited number. Complete list with brand name is reported in Annex 1.

Table 2.1 Store categories based on field survey

Type	*Number of stores visited and described*
Grocery supermarkets	54
Minimarkets	3
Pharmacies	10
Convenient stores	12
Liquor stores	5
Butcheries, Pork-butcheries	4
Fishmonger stores	2
Bakeries and Pastries	1
Small size Gas Stations	14
Large size Gas Stations	4
Hotels	8
Motels	5
Bars and Restaurants	1
Carbonated Soda Fountains	-
Vending machines	-
Total	122

Note: Carbonated soda fountains and vending machines are refrigerating equipment studied independently. They are used in many different stores.

1.1.1 Grocery stores or grocery supermarkets

This category gathers two subcategories: grocery stores established primarily for the retailing of food, and large supermarkets that store products other than food, such as clothing or household items. However, since they present the same sales area dedicated to food retailing, these two families are merged in one category referred to as grocery supermarkets. A total number of 54 groceries have been visited. Main brands are Albertsons, Ralphs, Wholefood, Safeway, Walmart, Target, and Costco Wholesale. Those stores present an average sales area of 4,400 m².

1.1.2 Minimarkets

The mainly visited brands are Smart &Final, Foods co. The sales area varies between 300 m² and 1,000 m².

1.1.3 Convenience stores

A convenience store is a small store or shop often located along busy roads. The main visited brands are seven/eleven and AM/PM stores, as well as local stores. An average sales area of 150 m² resulted from survey data processing (visited convenience stores presented sales area varying between 100 and 300 m²).

1.1.4 Liquor stores

A liquor store is the American and Canadian name for a type of convenience stores, which specializes in the sale of alcoholic beverages in the countries where its consumption is regulated. This category presents an average sales area identical to a convenience store. However, a category is dedicated to liquor stores because survey data processing demonstrated that installed refrigeration equipment and systems differ from those found in usual convenience stores.

1.1.5 Pharmacies

The pharmacy is a retail shop where medicine and other articles are sold. The main visited brands are: Walgreens, CVS pharmacy, and Rite aid. The sales area varies from 600 m² to 1,000 m². An average sales area of 800 m² is therefore chosen for this category.

1.1.6 Gas stations

A filling station, fueling station, gas station, service station or petrol station is a facility that sells fuel and lubricants for motor vehicles. Most of the visited gas stations had convenience stores selling food and beverages of different sizes. Therefore, two categories are dedicated to gas stations according to the store size and heat load. A first category includes small gas stations, and another one includes mid-size gas stations and gas stations related to commercial centers (for example, Walmart Gas station). The principal brands present in the survey are: 76, Chevron, Mobil, Exxon, and Arco.

1.1.7 Hotels

Hotels of different sizes have been visited during the survey, starting from 1-story to 12-story hotels. The principal brands visited are Best Western, Hilton, Marriott, Crowne Plaza, and

Holiday Inn. In order to cover the wide range of hotels, a typical hotel lay–out is defined in terms of room numbers. The typical room number is estimated based on the US Census numbers for hotels and hotel rooms and is found equal to 100 rooms. For a hotel description, the kitchen description is also taken into account.

1.1.8 Motels

The data processing concerning motels is identical to that presented in the hotel section. It is not appropriate to merge these two categories mainly because of significant differences in their kitchen refrigeration features. The visited motel chains are: America's Best Value Inn, Super Motel, and Comfort Inn Sunset.

1.1.9 Bars and restaurants

Bars and restaurants have refrigerating equipment for food conservation and beverage cooling. During the survey, it was not easy to access this equipment for a technical description. The layout of the hotel, which has a restaurant and a bar, is quite similar in terms of refrigeration equipment, except for the ice dispenser at each floor.

1.1.10 Bakeries

Bakeries primarily produce bread and related products, which are then transported to numerous selling points throughout a region. They normally sell beverages and snacks. An average sales area of 125 m² is estimated based on survey data processing.

1.1.11 Butcheries

Butcheries are stores dedicated to prepare meats and other related goods for sale. Several butcheries have been visited (El Cochinito Meat Market, Economy Meat, Veronica Meat Market, Meat Market Carniceria Latina). This category presented an average area of approximately 125 m².

1.1.12 Fishmonger Stores

A fishmonger store sells fish and seafood. This category presents an average area identical to butcheries.

1.1.13 Vending machines

After the data processing, it was more appropriate to group the vending machines in one category to avoid double counting.

1.1.14 Carbonated Soda Fountains (CSD Fountains)

Data related to CSD fountains have been processed identically to vending machines data, and a category is dedicated to group them.

1.2 Identification of refrigeration systems

1.2.1 Refrigeration systems

Three main technologies of refrigeration systems are used in stores: stand-alone equipment, condensing units, and centralized systems [LIT96].

Stand-alone or plug-in equipment is a display case where the refrigeration system is integrated into the cabinet and the condenser heat is rejected to the sales area of the supermarket. The purpose of plug-in equipment is to display ice cream or cold beverages such as beer or soft drinks.

Condensing units are small-size refrigeration equipment with one or two compressors and a condenser installed on the roof or in a small machine room. Condensing units provide refrigeration to a small group of cabinets installed in convenience stores and small supermarkets.

Centralized systems consist of a central refrigeration unit located in a machine room. There are two types of centralized systems: direct and indirect systems. In a direct system (DX), racks of compressors in the machine room are connected to the evaporators in the display cases and to the condensers on the roof by long pipes. In an indirect system, the central refrigeration unit cools a heat transfer fluid (HTF) that circulates from the evaporator in the machinery room and the display cases in the sales area.

Centralized direct systems are the dominant technology in the US and globally for supermarkets.

1.2.2 Refrigerated cabinets and rooms

Refrigerating equipment is sorted under 3 cabinet technologies: stand-alone equipment or self contained system (SA), display cases (DC) and walk-in coolers (WI).

Display cases and walk-in cabinets can be connected either to centralized system or to condensing unit depending on the equipment size and on the store category, whereas stand-alone equipment are by definition self-contained refrigerating systems.

For each cabinet technology, different types or designs have been identified based on the survey feedback. Technical characteristics and thermal equations have been issued for each type.

1.3 Survey of current refrigeration cases

A survey of 115 stores has been performed from June to November 2007 in order to collect data on existing store structures and types of refrigeration systems and cabinets. The results of this survey provided an abundance of information and allowed estimates to be made of current electricity consumption for the operation of refrigeration cabinets either as direct consumption by the cabinets (lighting, fans, anti-sweat heaters, defrosting) or by refrigeration compressors and condenser fans in order to provide refrigeration to these cabinets. The survey was performed for both remotely operated cases (DC and WI) and self-contained refrigerated equipment (SA).

1.3.1 Survey contents and data collection

Display cases (DC) and stand-alone equipment (SA) include low temperature single-deck, low temperature multi-deck, low and medium temperature glass door, medium temperature single-deck, medium temperature multi-deck, service cases, and specialty cases. Specifications include the make and model, case length, blown air temperature, saturated suction temperature, and all are included in the database.

The product display has been divided into the following categories: dairy, deli, meat product, beverage, bakery, frozen food, and ice cream. In many instances, a cabinet can be used for several of these products. Where the product displayed affects the operating temperatures or heat loads, a separate entry (in the data base) for the case is provided for each product. If the specified temperatures and heat loads are identical for multiple products, the products are noted in the description.

Survey data have been collected and grouped as a function of the refrigeration cabinet type. Hence, surveys are presented separately for display cases, stand-alone equipment, walk-in and storage rooms.

During the survey, store data have been recorded and included store's brand name, location and average sales area. For the presently manufactured refrigeration equipment, the following information was also collected:

- Brand name of the equipment manufacturer
- Equipment model number: *ex:* for a TRUE equipment, GDM–35
- Temperature level (medium, low)
- Equipment position: horizontal, vertical, semi-vertical
- Open or closed type equipment
- For closed type, the number of doors is recorded whereas for open type, the total length of the equipment is estimated.
- Equipment capacity and dimensions: capacity in cf or liters, height, width and length.
- Refrigerant type and charge.
- Product type (dairy, deli, bakery, salads, floral, meat, drinks, ice cream…).

The purpose of remote or self-contained refrigerated display cases in a store is to provide temporary storage for perishable foods prior to sale. Most of the design characteristics and general shape and layout of display cases are based on marketing specifications and constraints. The configuration of display cases falls into essentially four different categories.

- **Tub**: The tub case is often used for the storage and display of frozen foods and meats. Tub cases operate at a very uniform temperature and require the lower refrigeration capacity per foot of any display case type. The primary disadvantage of the tub is a low product storage volume per square foot of sales area.

- **Open-front multi-deck**: This case type possesses the largest storage volume per square foot of floor area, because of the use of an upright cabinet and shelves. Refrigeration capacity required for multi-deck cases is very high, including a large latent load portion due to the entrainment of ambient air in the air curtain passing over the opening of the case.

- **Glass door reach-in**: The reach-in case has glass doors over the opening of the case; these must be opened for product removal and stock. Reach-in cases are used in supermarkets primarily for frozen foods, because of their ability to contain the cold refrigerated air, which reduces the "cold aisle" problem.

- **Single-deck or service**: Open single-deck cases are commonly used for display of fresh meat products. The service display case is a single-deck case equipped with sliding doors in the back for access by serving people and a glass front to show product to customers. Cases of this type are commonly seen in the deli and meat departments of supermarkets.

Display cases have been developed and refined for specific merchandising applications, and cases of each type listed above exist specifically for the storage and display of specific food types.

To define a refrigeration equipment baseline, survey data have been processed based on technical data of leading refrigeration cabinet manufacturers in the United States. Data processing showed capacities and dimensions found in the stores that could be different from data gathered on websites of equipment manufacturers. To take into account these differences, interpolations have been made on the refrigeration capacity as well as the input power.

When an equipment description is identical to a model listed in the Table (except for its manufacturer), input power and refrigerant data are directly applied to the studied equipment.

1.3.2 Stand-alone equipment

One objective of the survey is to define baseline stand-alone equipment models depending on description parameters stated above. Leading manufacturers of stand-alone equipments are: True Manufacturing, Beverage Air, and Hussmann Corporation. The stand-alone cases listed have been categorized into 23 models presented in Table 2.2, each model having a number starting from 1 to 23.

1.3.3 Display cases

Leading refrigeration equipment manufacturers of display cases are: Hussmann Corporation, Hill Phoenix, Tyler Refrigeration Corporation, and Kysor Warren. Fourteen baseline display cases are defined in Table 2.3, each model having a number starting from 1 to 14.

1.3.4 Walk-ins

Selections include storage walk-ins, walk-in boxes with glass doors (e.g., dairy, beverage and floral boxes), preparation areas that may be fully enclosed or have one side open to the sales area, and other perimeter zones that are air conditioned from the refrigeration system (e.g., bakery prep areas, pharmacy, etc.). Specifications include the make and model (for components in the library), size, temperature, location, reach-in doors, walk-in doors, heat load, lighting, evaporator coils, defrost type and control, fans, and internal loads. For walk-in (WI), 5 baseline categories are found and listed in Table 2.5.

Table 2.2 Baseline stand-alone equipments list.

ID	Brand Name	Model	T° Level	Open/ Closed	Position	Number of doors	Capacity (liters)	Length (m)	Height (m)	Width (m)	Product type	Refrigerant	Refrigerant charge (g)	Cooling capacity (HP)	Product Temp (°C)	DTEvap
SA_01	(TRUE)	GDM-7	Medium	closed	Vertical	1	200	0.8	1	0.6	drinks, salads, deli, dairy	R-134a	300	0.2	2	9
SA_02	(TRUE)	GDM-10	Medium	closed	Vertical	1	300	0.8	1.4	0.65	drinks, salads, deli, dairy	R-134a	300	0.2	2	9
SA_03	(TRUE)	GDM-23	Medium	closed	Vertical	1	600	0.8	2	0.7	drinks, salads, deli, dairy	R-134a	300	0.33	2	9
SA_04	(TRUE)	GDM-35	Medium	closed	Vertical	2	1000	1.6	2	1	drinks, salads, deli, dairy	R-134a	300	0.5	2	9
SA_05	(TRUE)	GDM-72	Medium	closed	Vertical	3	2000	2.4	2	2	drinks, salads, deli, dairy	R-134a	550	0.5	2	9
SA_06	(TRUE)	TCGR-50	Medium	closed	S-Vertical	2	800	1.6	1.2	1.3	salads, deli, bakery	R-134a	300	0.5	2	9
SA_07	Hussman	SHM	Medium	open	S-Vertical	0		1.8	1.3	0.9	drinks, salads, deli, dairy	R-134a	800	0.5	2	9
SA_08	(TRUE)	TAC-48	Medium	open	Vertical	0	1000	1.22	2	1.2	drinks, salads, deli, dairy	R-404A	1000	1	2	9
SA_09	Hussman	DDS8	Medium	open	Vertical	0		2.4	2.5	2.1	drinks, salads, deli, dairy	R-22	1400		2	9
SA_10	(TRUE)	THF-41FL	Low	closed	Horizontal	2	300	1.6	1	0.8	frozen	R-134a	1200	0.75	-18	11
SA_11	(TRUE)	GDM-23F	Low	closed	Vertical	1	600	0.8	2	0.7	frozen	R-404A	600	0.75	-18	11
SA_12	(TRUE)	GDM-35F	Low	closed	Vertical	2	1000	1.6	2	1	frozen	R-404A	1000	1	-18	11
SA_13	(TRUE)	GDM-72F	Low	closed	Vertical	3	2000	2.4	2	2	frozen	R-404A	2500	1.5	-18	11
SA_14	Bev_Air	FC-45	Medium	closed	Vertical	2	1200	1.6	2	0.7	Floral	R-134a	600	0.33	2	9
SA_15	Hussman	ISMGG	Medium	open	Horizontal	0		1.7	1	1	deli, dairy	R-22	2300		2	9
SA_16	Hussman	ISFGG	Low	open	Horizontal	0		1.7	1	1	frozen	R-22	2300		-18	11
SA_17	(TRUE)	GDM41/45	Medium	closed	Vertical	2	1200	1.6	2	0.75	drinks, salads, deli, dairy	R-134a	500	0.5	2	9
SA_18	(TRUE)	GDM 12	Medium	closed	Vertical	1	340	0.8	1.6	0.7	drinks, salads, deli, dairy	R-134a	250	0.2	2	9
SA_19	Bev_Air	MT12	Medium	closed	Vertical	1	400	0.8	1.6	0.61	boissons	R-134a	300	0.2	2	9
SA_20	Bev_Air	UR30	Medium	closed	Vertical	1	700	0.8	0.9	0.7	boissons	R-134a	200	0.2	2	9
SA_21	LEER	Model 45	Low	closed	Vertical	2	1200	1.6	2	1.2	Ice maker	R-134a	300	0.33	-25	9
SA_22	Bev_Air	DD68	Medium	closed	Horizontal	2	800	1.6	1	0.7	Beer	R-134a	300	0.33	2	9
SA_23	(TRUE)	GSM	Medium	closed	Vertical	1	700	0.8	2	0.8	Tour Patissier	R-134a	300	0.5	2	9

Table 2.3 Baseline display cases list.

ID	Brand Name	Model	T° Level	Open/ Closed	Position	Number of doors	Length (m)	Height (m)	Width (m)	Cooling Capacity (W/m)	Refrigerant charge (kg/m)	Product type	Product Temp (°C)	DTEvap
DC-01	TYLER	N6D	Medium	open	Vertical		1.2;1.8;2.4;3.6	2	1	1222	1.55	Dairy, dely, produce, juice	4	13
DC-02	HUSSMAN	RM	Medium	closed	Vertical	1;2;3	0,8	2.12	1.11	312	0.52	Dairy, deli, beverages	2	11
DC-03	HUSSMAN	E2	Medium	open	S-Vertical		1.2;1.8;2.4;3.6	1.24	1.2	909	0.34	Deli, pizza, ploral, juice	4	13
DC-04	HUSSMAN	SMGV	Medium	closed	S-Vertical		1.2;1.8;2.4;3.6	1.32	1.1	202	0.28	Meat, delicatessen	2	11
DC-05	TYLER	NM/NMG	Medium	open	Horizontal		1.8;2.4;3.6	0.96	1	416	0.39	Meat, deli, Produce	2	11
DC-06	HUSSMAN	F6L	Low	open	Vertical		1.2;1.8;2.4;3.6	2	1.2	1688	0.66		-18	13
DC-07	HUSSMAN	RLN	Low	closed	Vertical	2;3;4;5	0,8	2.12	2	484	0.51	Frozen	-18	13
DC-08	TYLER	NFNG	Low	closed	Horizontal		2.4;3.6	0.9	1	352	0.42	Frozen	-18	13
DC-09	TYLER	NPW	Medium	open	Horizontal		2.4;3.6	0.8	1.8	747	0.76	Produce	5	14
DC-10	HUSSMAN	P2X	Medium	open	Vertical		1.2;1.8;2.4;3.6	1.9	1	875	0.57	Fruits, produce	5	14
DC-11	HUSSMAN	F2XLG	Low	open	S-Vertical		1.2;1.8;2.4;3.6	1.4	1.2	1972	0.66	Frozen meat, seafood	-18	13
DC-12	HUSSMAN	DSFM	Medium	closed	S-Vertical		1;2;3;4;	1.2	1.06	481	0.49	Seafood	2	11
DC-13	Hussman	FI	Low	open	Horizontal		2,4;3,6	0.91	1.471	462	0.5	Frozen	-18	16
DC-14	ZeroZone	RMCP30FL	Medium	closed	Vertical	2;3;4;5	0,8	2.1	0.9	1174	0.64	FLORAL	5	14

Table 2.4 Baseline walk-in cases list.

ID	T° level	Open/ Closed	ReachIn (0)/ ColdStorage (1)	Number of doors	Length (m)	Height (m)	Width (m)	ProductDH (kJ/kg)	Product mass flow (kg/m3 24h)	Product Temp (°C)
WI-01	Medium	Open	0		1	2.5	3	45	20	2
WI-02	Medium	Closed	0	12	1	2.5	3	45	20	2
WI-03	Low	Closed	0	12	1	2.5	3	55	15	-20
CR-01	Medium	Closed	1		1	3.5	6	45	20	2
CR-02	Low	Closed	1		1	3.5	6	55	15	-20

Once the baseline refrigeration equipment, self-contained or remote refrigerated equipment are described, it is possible to draw typical layouts for the 16 store categories defined in Section 1.1. The next section presents the grocery supermarket layout based on equipment listed in Table 2.2 to Table 2.5.

1.4 Typical grocery store lay-out

A representative grocery supermarket layout is shown in Figure 2.1. Refrigerated fixtures are located throughout the store, because of the large amount of perishable food products that are sold. These fixtures fall into 3 categories, stand-alone equipment, display cases, and walk-in storage coolers. Stand-alone equipment and display cases are located on the sales floor and are designed to refrigerate food products while providing a place to merchandise them. Walk-in coolers are used to store food products during the time period between receiving the product and placing the product out for sale.

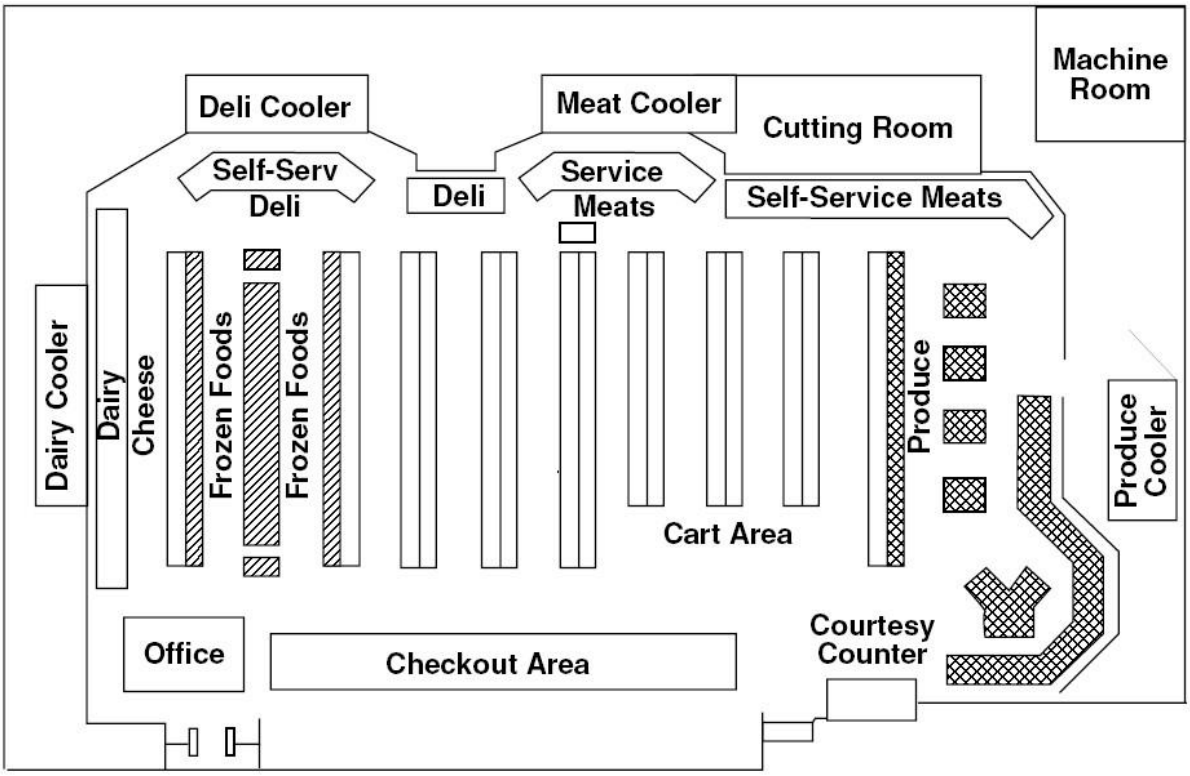


Figure 2.1 Lay out of the refrigerated fixtures in a supermarket [ORN04].

A typical arrangement of refrigerating equipment in a grocery is shown in Figure 2.1. Display cases, of a variety of configurations and products, are generally used in the sales area and are located at the periphery of the store near their associated walk-ins. The survey data processing enabled the definition of a typical grocery refrigeration configuration presented in Table 2.6, Table 2.7, and Table 2.8.

Table 2.5 Self-contained refrigerating equipments found in a grocery store.

Stand Alone equipments		
Description	**Model**	**Number of equipment**
Medium temperature self contained closed vertical case for drinks salads, deli and dairy with a capacity of 200 liters	SA-1	3
Medium temperature self contained closed vertical case for drinks salads, deli and dairy with a capacity of 300 liters	SA-2	2
Medium temperature self contained closed vertical case for drinks salads, deli and dairy with a capacity of 600 liters	SA-3	1
Medium temperature self contained open vertical case for drinks salads, deli and dairy with a capacity of 1000 liters	SA-8	1
Medium temperature self contained open vertical case for drinks salads, deli and dairy with a capacity of 2000 liters	SA-9	1
Medium temperature self contained open horizontal case for deli and dairy with a capacity of 1500 liters	SA-15	1
Medium temperature self contained closed vertical case for drinks salads, deli and dairy with a capacity of 1200 liters	SA-17	1
Medium temperature self contained closed vertical case for drinks salads, deli and dairy with a capacity of 340 liters	SA-18	1
Self contained Ice maker with a capacity of 1200 liters	SA-21	1

Table 2.6 Display cases equipments found in a grocery store.

Display cases equipments		
Description	**Model**	**Length (m)**
Medium temperature Open-front multi-deck vertical display case for dairy, deli, juice and drinks	DC-1	75
Medium temperature Glass door reach-in multi-deck vertical display case for dairy, deli, juice and drinks	DC-2	7
Medium temperature Open-front single-deck semi-vertical display case for deli, pizza floral and juices	DC-3	15
Medium temperature Glass door reach-in single-deck semi-vertical display case for meat and delicatessen	DC-4	20
Medium temperature Open Tub case for meat and delicatessen	DC-5	10
Low temperature Open-front multi-deck vertical display case for frozen products	DC-6	4
Low temperature Glass door reach-in multi-deck vertical display case for frozen products	DC-7	86
Medium temperature Open Tub case for produce	DC-9	17
Medium temperature Open-front multi-deck vertical display case for produce	DC-10	27
Medium temperature Open-front single-deck semi-vertical display case for seafood	DC-12	5
Low temperature Open Tub case for frozen products	DC-13	17
Medium temperature Glass door reach-in multi-deck vertical display case for floral	DC-14	3

Table 2.7 Walk-in and cold rooms found in a grocery store.

Walk In and Storage Rooms		
Description	**Model**	**Length (m)**
Medium temperature Open-front multi-deck walk in for dairy, deli, juice and drinks	WI-1	4
Medium temperature Glass door reach-in multi-deck walk in for dairy, deli, juice and drinks	WI-2	12
Low temperature Glass door reach-in multi-deck walk in for dairy, deli, juice and drinks	WI-3	12
Medium temperature Cold Storage room	CR-1	60
Low temperature Cold Storage room	CR-2	18.5

1.5 Typical layout of small stores

Similarly to grocery stores, the survey data processing enabled the definition of a typical refrigeration layout for each of the 15 categories defined previously. These layouts are described in Table 2.8, Table 2.9, and Table 2.10.

Table 2.8 Self-contained refrigerating equipment distribution for different store categories.

Stand Alone	SA-1	SA-2	SA-3	SA-4	SA-5	SA-6	SA-7	SA-8	SA-9	SA-10	SA-11	SA-12	SA-13	SA-14	SA-15	SA-16	SA-17	SA-18	SA-19	SA-20	SA-21	SA-22	SA-23
Grocery	3	2	1	0	0	0	0	1	1	0	0	0	0	0	1	0	1	1	0	0	1	0	0
Large Supermarket	2	2	0	3	0	0	0	0	0	2	1	0	0	2	0	1	1	0	0	0	2	0	0
Pharmacy	1	1	0	0	0	0	0	0	0	1	0	0	0	0	0	0	0	0	1	0	1	0	0
Convenience	0	0	0	2	0	0	0	0	1	2	0	0	0	0	0	0	0	0	1	0	1	0	0
Liquor Store	0	1	0	1	1	0	0	0	0	3	0	0	0	0	0	0	1	0	1	0	1	0	0
Minimarket	1	2	1	0	0	0	0	1	0	1	1	0	1	0	1	0	0	0	0	0	2	0	0
Small Gas Stat	0	0	1	0	0	0	0	0	0	0	0	0	0	0	0	0	1	0	0	0	0	0	0
Center Gas Stat	1	0	0	1	0	0	0	0	0	1	1	0	0	0	0	0	1	0	1	0	1	0	0
Hotel	2	0	0	1	0	0	0	0	0	0	0	1	0	0	0	0	0	0	0	0	4	1	0
Motel	0	0	0	0	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	3	0	0
Butchery	0	0	0	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Fishmonger Store	0	0	0	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	0	0
Bakery	0	0	0	2	1	0	0	0	0	1	0	0	0	0	0	0	0	0	0	0	0	0	1
Restaurants Bar	2	0	0	1	0	0	0	0	0	0	0	1	0	0	0	0	0	0	0	0	1	1	0
Vending Machine	0	0	0	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Soda Fountain	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	0	0

Table 2.9 Distribution of refrigerated display cases for different store categories.

Display Case	DC-1	DC-2	DC-3	DC-4	DC-5	DC-6	DC-7	DC-8	DC-9	DC-10	DC-11	DC-12	DC-13	DC-14
Grocery	75	7	15	20	10	4	86	0	17	27	0	5	17	3
Large Supermarket	0	10	8	0	6	0	11	0	0	1	0	0	1	0
Pharmacy	0	1	0	0	0	0	3	0	0	0	0	0	0	0
Convenience	1	0	0	2	0	0	1	0	0	3	0	0	0	0
Liquor Store	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Minimarket	4	0	0	0	0	0	3	0	0	2	0	0	2	0
Small Gas Stat	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Center Gas Stat	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Hotel	0	0	0	3	0	0	0	0	0	0	0	0	0	0
Motel	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Butchery	0	0	2.5	2.5	0	0	0	0	0	0	0	0	0	0
Fishmonger Store	0	0	5	0	0	0	0	0	0	0	0	0	0	0
Bakery	0	0	0	5	0	0	0	0	0	0	0	0	0	0
Restaurants Bar	0	0	0	3	0	0	0	0	0	0	0	0	0	0
Vending Machine	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Soda Fountain	0	0	0	0	0	0	0	0	0	0	0	0	0	0

Table 2.10 Distribution of walk-in and cold storage rooms for different store categories.

Walk IN	WI-1	WI-2	WI-3	CR1	CR2	Sales Area (m^2)
Grocery	4	12	12	60	19	2500
Large Supermarket	0	6	9	27.5	9	8500
Pharmacy	3	8	4	0	0	800
Convenience	0	9	2	0	0	150
Liquor Store	0	14	0	0	0	153
Minimarket	0	14	21	27.5	9	1145
Small Gas Stat	0	2	0	0	0	25
Center Gas Stat	0	7	2	0	0	100
Hotel	0	0	0	6	3	
Motel	0	0	0	0	0	
Butchery	0	0	0	3	0	125
Fishmonger Store	0	0	0	3	0	125
Bakery	0	0	0	0	0	125
Restaurants Bar	0	0	0	6	3	
Vending Machine	0	0	0	0	0	
Soda Fountain	0	0	0	0	0	

2 Energy consumption of refrigerating systems in California

2.1 Method for energy consumption calculation

Supermarkets represent one of the largest energy-intensive building groups in the commercial sector, consuming 2 to 3 million kWh annually per store [BAX03a]. Several studies have shown that annual electricity consumption ranges from 1 to 1.5 million kWh per store for refrigeration [LIT96]. A typical electricity usage of a grocery in the U.S. shows that 39% is used for refrigeration, 23% for lighting, 11% for cooling, 4% for ventilation, 13% for heating, and 10% for miscellaneous applications (cooking, water heating, …) as shown in Figure 2.2.

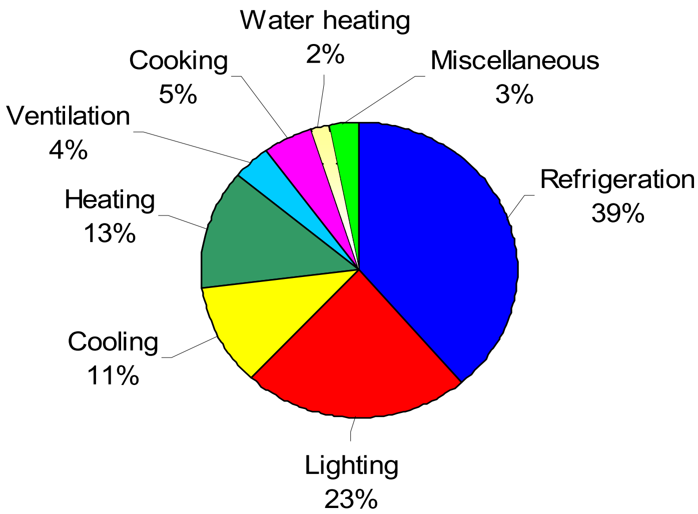


Figure 2.2 Typical electrical energy usage in a grocery store in USA [LIT96].

Recent field tests tend to confirm that this figure is still a good estimate. Data from a field test in a 50,000 ft²-store in Southern California indicate annual usage of about 1,500,000 kWh for all refrigeration including case lights, fans, heaters, etc [ORL04].

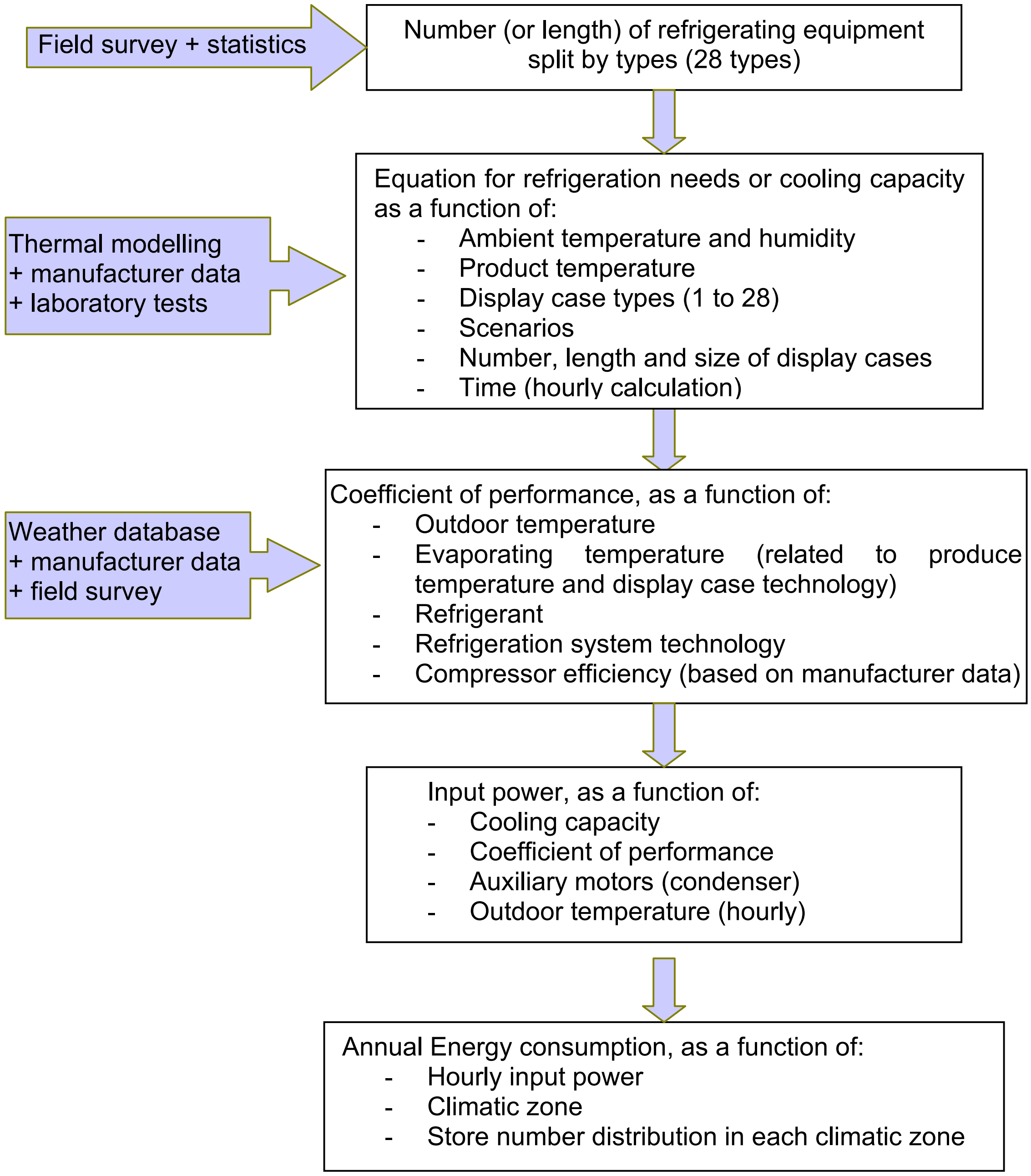
The approach for energy consumption calculation in commercial refrigeration, detailed in this report, is qualified as “bottom – up approach”. In order to simulate energy efficiency improvement of refrigeration equipment, each element in the energy consumption chain has to be considered and described in detail.

The energy consumption calculation is based on the evaluation of heat loads, hour by hour, on a given year, taking into account weather conditions (temperature and humidity) of the 8 California climatic zones.

Each type of store (16 families) has been calculated independently, when the layout of refrigeration equipment in each store has been issued.

2.1.1 Energy consumption calculation

The algorithm here below presents the method for energy consumption calculation, which is applied to each type of store.



The cooling capacity of display cases is provided by a large vapor compression refrigeration system. The operating characteristics and energy requirements of the refrigeration system are directly related to the refrigeration capacity necessary to maintain the display case temperature.

There are two main temperature levels in supermarkets: medium temperature for preservation of chilled food and low temperature for frozen products. Chilled food is maintained between 1°C and 14°C, while frozen food is kept between -12°C to -18°C. The evaporation temperature, for a medium-temperature system, varies between –15°C and -5°C, and for a low-temperature system, the evaporation temperatures are in the range of –30°C to -40°C. Variations in temperature are dependent upon products, display cases and the chosen refrigeration system [LIT96].

2.1.2 Heat load, refrigeration capacity

The heat load gained by the case is the amount of heat that must be removed from the display case in order to maintain the product in the case at the desired storage temperature. The refrigeration capacity is equal or superior to the heat loads to maintain the product temperature. The refrigeration capacity of a display case is most often given at a specific blown air temperature at the outlet of the evaporator, since this value is easier to measure (and control) than the temperature of the stored product. The standard rating condition to specify the refrigeration capacity of a display case is for operation in an indoor environment with a 75°F dry-bulb temperature and a relative humidity of 55 percent. The heat loads of a refrigerated cabinet are coming from convection, conduction, radiation, and advection.

2.1.2.1 Conduction

Ambient heat that passes through the walls of the display case is intercepted by the air flowing around the perimeter of the display case.

2.1.2.2 Radiation

Thermal radiation heat transfer occurs between the interior of the display case and the surrounding ambient environment.

2.1.2.3 Convection (air entrainment)

The air curtain passing across the opening of the display case mix with and entrain part of the surrounding ambient air, which is then returned to the case evaporator. The heat load due to the entrained air consists of both sensible and latent heats. Ambient air entrainment occurs in all display case types, but represents the largest portion of the heat load for open, multi-deck cases.

2.1.2.4 Internal loads

Heat is generated by the use of electric energy in the display case for the following auxiliaries:

- Lights: fluorescent light features are installed in the display cases for illumination of the product. Heat from the ballasts may also enter the case if the ballast is installed in the refrigerated portion of the case.
- Fan motors: the electric energy associated with the fans used to circulate air around the display case.
- Anti-sweat heaters: are installed in glass doors and on other surfaces that operate at a temperature below the ambient dew-point temperature. If heaters are not installed, condensation and possibly frost will form on these surfaces.

  The contribution of each load source will vary according to display case type. The heat load of open multi-deck display cases is dominated by air entrainment. Internal electric loads represent a significant portion of the heat load of reach-in frozen food cases. For single-deck and tub cases, radiation heat transfer accounts for a large fraction of the heat loads.

The impact of each of these thermal loads on the refrigeration capacity depends on the case type. For example, air infiltration is the most significant portion of heat loads for open, multi-deck cases, while radiation is the largest part of the heat load for tub-type cases. The door anti-sweat heaters represent a major share of the heat load for frozen food door reach-in cases.

- Defrosting: the conditions of air in cold storage rooms or in display cases affect the refrigerating capacity of the evaporator. At surface temperature lower than the dew point temperature of air, the water vapor contained in the humid air will condense on surfaces, and at surface temperature lower than 0°C frost will deposit on the surfaces. The frost formation that is seen on evaporator surfaces is an important factor in the operation of refrigeration systems. Without periodic removal, the frost will accumulate and eventually block the airflow passages of the evaporator, resulting in loss of cooling capacity. The usual operation for supermarket refrigeration systems is to defrost the display cases on a scheduled basis. Several different methods are employed for defrosting: off-cycle defrosting, electric defrosting.

2.1.2.5 Off-Cycle defrosting

Refrigeration to the case is shut off and the evaporator warms above the melting temperature of the frost. This method is commonly used for display cases operating at the highest blown air temperatures (34 to 37°F), because frost loading is relatively small. Off-cycle defrosting is also used where the product is not sensitive to air temperature change, such as milk and other dairy products. For frozen food or meat, off-cycle defrosting is not appropriate.

2.1.2.6 Electric defrosting

Electric heaters are installed at the inlet of the evaporator so that the circulated air can be heated. The warm air passes through the evaporator where it provides the heat needed to melt the frost. Although it is the most energy consuming application, electric defrosting remains used in all refrigeration systems and is considered the most reliable defrosting method.

Defrosting has a significant impact both for energy consumption and product temperatures because of the air and product temperature rises during defrosting and has to be lowered quickly after defrosting leading to a significant overcapacity for rapid “pull down” of temperatures. If not performed correctly, the product can be damaged. The number of defrosting cycles required for a refrigeration case depends on its type. Open, multi-deck display cases will require several (3 to 6), while tub and reach-in cases normally have only one defrosting per day. Defrosting schedule is normally controlled by a time clock that initiates defrosting for each case at specific times each day.

2.1.3 Thermal modeling of display cases

The average air temperature, inlet and return air temperatures, evaporating temperature, electrical data for fans, heating wires, defrosting heaters and light, coil volume, diameter of tubs and heat loads at 22°C – 65% RH and at 25°C – 60% RH for each cabinet have been put into a database. The heat loads in display cases are dependent on indoor conditions in the supermarket; a higher indoor temperature and relative humidity increase the cooling demand and the energy requirement. An energy balance of an open vertical cabinet is shown in Figure

2.3 where heat losses from infiltration, radiation, conduction, lighting, the fan, heating wires, and defrosting are presented.

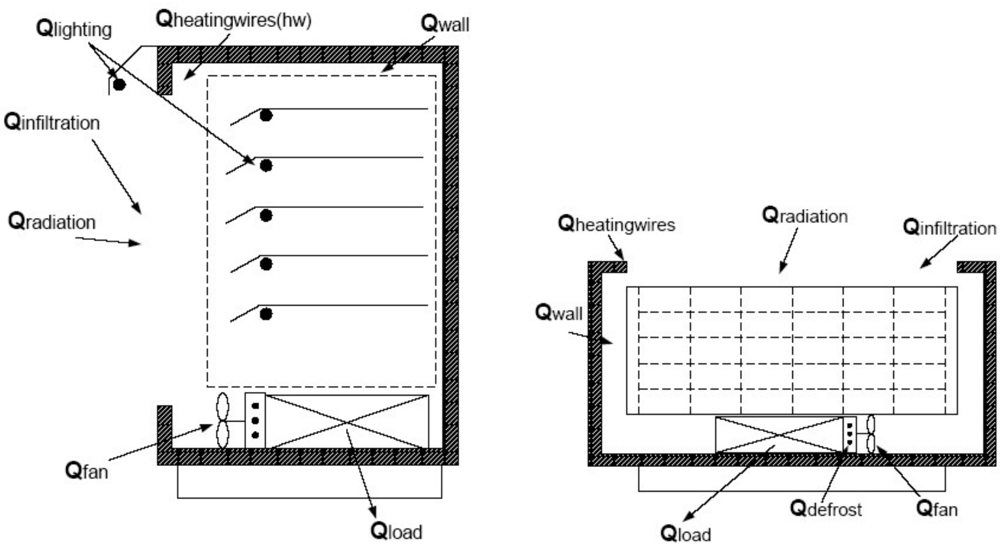


Figure 2.3 Energy balance of vertical (left) and horizontal (right) display cases.

The following Equations state the expressions of different loads accounted for in the cooling load calculations as a function of the display case and the store temperature.

Starting with the conduction load expressed in Equation (2.1):

$$\dot{Q}_w = U_{case} A_{case} \left(T_{store} - T_{case} \right) \tag{2.1}$$

Where $U_{case} = \dfrac{1}{\dfrac{1}{h_i} + \dfrac{t}{k} + \dfrac{1}{h_o}}$ and A_{case} corresponds to the total surface of the cabinet exchanging by conduction with the surrounding store ambience. h_i and h_o represent the inner and outer convective heat exchange coefficients respectively, t and k the insulation thickness and thermal conductivity respectively.

The radiation load is also considered and can be evaluated applying Equation (2.2):

$$\dot{Q}_{radiation} = \frac{\sigma \left(T_w^4 - T_{case}^4 \right)}{\left[\left(\dfrac{1-\varepsilon_w}{\varepsilon_w A_w} \right) + \left(\dfrac{1}{A_w F_{case,w}} \right) + \left(\dfrac{1-\varepsilon_{case}}{\varepsilon_{case} A_{case}} \right) \right]} \tag{2.2}$$

Where the subscript w refers to the store wall, σ Stefan Boltzmann's constant and ε the surface emissivity.

The infiltration load depends on the amount of store air entrained in through frozen-food cabinets. This amount is usually expressed as a ratio of the cabinet blown airflow rate. The percentages of the store air entrapment into the cabinets and freezer rooms are found in literature or evaluated through extensive measurements and parametric analyses for specific blown air velocities. For instance, for an open cabinet, air entrapment is taken equal to 7 or 8% of the blown airflow rate, while it is approximately equal to 1% for closed cabinets. Equation (2.3) states the infiltration load expression:

$$\dot{Q}_{infiltration} = \rho_{air} V_{air,ent} \left(h_{store} - h_{case} \right) \quad (2.3)$$

Where the subscript air,ent refers to entrapped air, V the volumetric air flow rate and h air enthalpy at store or case temperature.

Dissipations of heat from installed equipment should also be taken into account such as lamps and ballasts ($\dot{Q}_{lighting}$), fan motors ($\dot{Q}_{fan}$), anti-sweat heaters ($\dot{Q}_{heating-wires}$) and extra heat from defrost ($\dot{Q}_{Defrost}$). The total load is obtained by summing all of the above evaluated quantities as expressed in Equation (2.4):

$$\dot{Q}_{load} = \dot{Q}_{wall} + \dot{Q}_{infiltration} + \dot{Q}_{radiation} + \dot{Q}_{lighting} + \dot{Q}_{fan} + \dot{Q}_{heating-wires} + \dot{Q}_{Defrost} \quad (2.4)$$

The load calculation of a stand-alone equipment is identical to a closed cabinet display load calculation.

2.1.4 Thermal modeling of cold storage room

The dimensioning of refrigeration capacity for cold storage is defined by four factors: heat transmission, exchange of air, cooling or freezing of products and internal heat generation [GRA03]. Heat transmission through walls, floor, and ceiling is dependent on the overall heat transfer coefficient and the temperature difference between the room and the surroundings. The heat transmission has been defined as shown in Equation (2.5):

$$\dot{Q}_{cond} = \Sigma \left(U_{CR} A_{CR} \left(T_{store} - T_{CR} \right) \right) \quad (2.5)$$

The exchange of air in cold rooms depends on the frequency of door openings and the size of the room. The exchange of air increases the heat load of the room. The influence of incoming air in the room can be calculated from Equation (2.6):

$$\dot{Q}_{airex} = \rho_{air} \dot{V}_{airex} \left(h_{store} - h_{CR} \right) \quad (2.6)$$

Where $\dot{V}_{airex}$ is an average volume flow of incoming air that is defined in (Granryd 2003) as presented in Equation (2.7):

$$\dot{V}_{airex} = V_{CR} \frac{nd}{24.3600} \quad (2.7)$$

nd is the number of air exchanges in the room per 24 hours. Temperatures and the frequency of door openings influence the number of air exchanges. Results from experiments are presented in Table 2.11 [GRA03].

Table 2.11 Number of air exchanges [ARI05].

Room Volume (m^3)	*Air exchanges - Medium temp. room (°C)*	*Air exchanges – Low temp. room (°C)*
7	38	30
10	31.5	24.5
20	21.5	17
40	14.5	11.5
100	9	7
500	3.5	2.7
1000	2.5	2.7
3000	1.35	1.05

The enthalpy difference for freezer rooms has been assumed to be 45 [kJ/kg], which is the average between the enthalpies of different products at temperatures -15°C and –18°C. Similarly, the enthalpy difference for the cold room has been assumed to be 55 [kJ/kg], which is the average between the enthalpies of different products at temperatures 17°C and 1°C. The mass flow has been assumed to be 20 kg/m^3 per 24 hours [ARI05] for cold rooms and 15 kg/m^3 per 24 hours for freezer rooms. Internal heat generation from lighting and people also affects the heat load of the cold room. The heat generated by lighting has been assumed to be 15 W/m^2 and the heat from people to be 200 W.

2.1.5 Coefficient of performance

Many factors have an impact on the coefficient of performance (COP):

- Level of temperature for product or beverage conservation
- Temperature differences at the condenser and evaporator coils
- Compressor efficiency
- Type of refrigerant
- Configuration of the refrigeration system (one or two compression stage, sub-cooling or not).

For energy calculation along the year, the coefficient of performance has been considered as a function of these variable values. The expression of the COP for the theoretical cycle of Carnot is:

$$COP_c = \frac{Te}{Tc - Te} \tag{2.8}$$

Where *Te* and *Tc* are respectively evaporating temperature and condensing temperature expressed in Kelvin. These temperatures are linked to product temperature and ambient temperature. Product conservation temperature is supposed to be fixed along the year. Typical temperature difference at the evaporator and the condenser are presented in Table 2.12.

Table 2.12 Temperature difference in heat exchangers.

Difference of temperature in heat exchangers	*DTev evaporator*	*DTcd condenser*
Centralized System / medium temperature	15 K	12 K
Centralized System / low temperature	17 K	10 K
Condensing Units / medium temperature	15 K	12 K
Condensing Units / low temperature	17 K	12 K
Stand-alone equipment / medium temperature	15 K	15 K
Stand-alone equipment / low temperature	15 K	15 K

In centralized systems, the evaporating temperature is the same for all display case connected to the same compressor rack. The evaporating temperature varies usually between –10°C and –12°C for medium temperature display cases. Product temperature conservation ranges from 0°C to +10°C. A thermostatic expansion valve (TxV) controls usually the refrigerant feeding of the evaporator. The superheat control is not optimized and leads to poor efficiency of the evaporator coil. In consequence, temperature difference between air and refrigerant is high.

Because of pressure drops in the compressor rack suction lines, the pressure at the suction port is lower, and the equivalent saturating temperature is between –12°C and –15°C. The airflow rate on the display-case evaporator is low, for noise reduction.

For calculations, the evaporating temperature is fixed at **–13°C** for medium-temperature compressor rack, and –35°C for low-temperature compressor rack.

The coefficient of performance can be expressed with the theoretical Carnot COP and the cycle efficiency. The cycle efficiency depends mainly on the compressor efficiency, the system design, and the refrigerant properties. For each type of refrigeration system (centralized, condensing units, stand alone), cycle efficiencies have been calculated for the refrigerant in use. Compressor efficiencies have been taken from manufacturer data (Copeland, Carlyle)

Table 2.13 Cycle efficiency (COP / COPc).

Refrigeration system	*Cycle efficiency*
Centralized System / medium temperature	45 %
Centralized System / low temperature	42 %
Condensing Units	40 %
Stand-alone equipment	25 %

The additional power consumption of the condenser fan is integrated in the cycle efficiency. In centralized systems, the input power for condenser ventilation is 7% of compressor rack input power for medium-temperature system, and 8% of compressor rack input power for low-temperature system [BIG02, FAY00].

In supermarkets, the condensation pressure is controlled to a minimum level to keep the pressure sufficiently high so that the thermo expansion valves (TxV) correctly feed the evaporators. In wintertime, with low outdoor temperature, it is possible to reduce the condensing pressure, taking advantage of a lower pressure ratio for compressors. This is possible when expansion valves are designed for a wider range of pressure differences. The impact on the cycle efficiency, when the head pressure control is activated, has been taken into account for the different scenarios of energy consumption.

Finally the equation of the coefficient of performance is a function of the outdoor temperature:

$$COP = \frac{Tev}{\left(Text + DT_{cd}\right) - Tev} \cdot \eta c \qquad (2.9)$$

with *DTcd* (temperature difference at the condenser) as functions of the out door temperature (*Text*), the floating head pressure control (*FHPC*), the technology of the refrigerating system (*Tech*), and the level of the evaporating temperature (*Tev*).

$$DTcd = f\left(Text; FHPC; Tech; Tev\right) \qquad (2.10)$$

The cycle efficiency (η_c) is a function of the technology (*Tech*) and the level of temperature for centralized systems (*Tev*) (see Table 2.13). Two other variables are considered for the cycle efficiency: the floating head pressure control (*FHPC*) and the outdoor temperature (*Text*).

Because ventilation is reduced when outdoor temperature is low, in wintertime for example, the additional input power of the condenser fan decreases. The cycle efficiency includes the additional power input of the condenser fans, which varies according to the outdoor temperature. In consequence, the cycle efficiency must be correlated to the outdoor temperature and the head pressure control.

$$\eta c = f(Text; FHPC; Tech; Tev) \tag{2.11}$$

2.1.6 Store distribution in climatic zones

Calculations for energy consumption are done for different climatic conditions. The outdoor temperature has an impact on the coefficient of performance of the refrigerating system. Depending on the climatic zone of the supermarket and the other stores, the hourly variation of temperature will have an impact on the energy consumption.

Eight climatic zones have been defined in California. Temperature variations during one typical year are known and registered hour by hour in weather stations. The distribution of stores and supermarkets in the different climatic zones has been done proportionally to the population living in each zone.

Figure 2.4 presents the average temperature (24 hours averaged) for three weather stations in California.

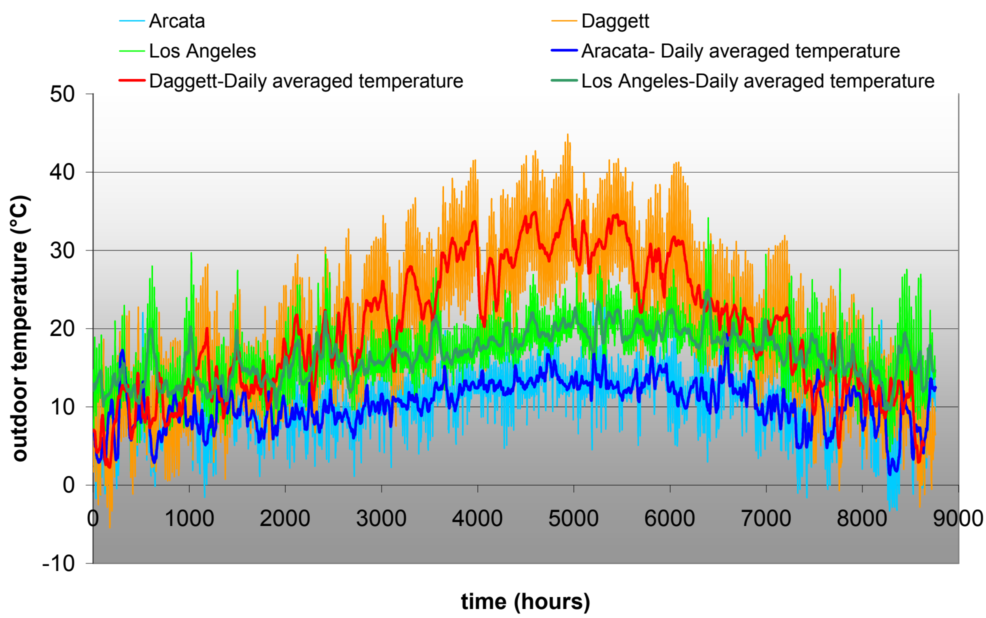


Figure 2.4 Temperature evolution in one year – 3 weather station measurements.

Figure 2.5 presents the 8 climatic zones, and the population distribution in these zones. Los Angeles, with 36% of the population, has the greatest number of inhabitants. The distribution of the different stores studied in the commercial refrigeration sector is approximately the same as the population distribution among climatic zones shown.

The population distribution is computed based on population estimates given per air basin and counties by the U.S Census Bureau. The temperature curves are obtained from the thermal engineering software TRNSYS (The Transient Energy System Simulation Tool) for all the climatic zones defined in Figure 2.5.

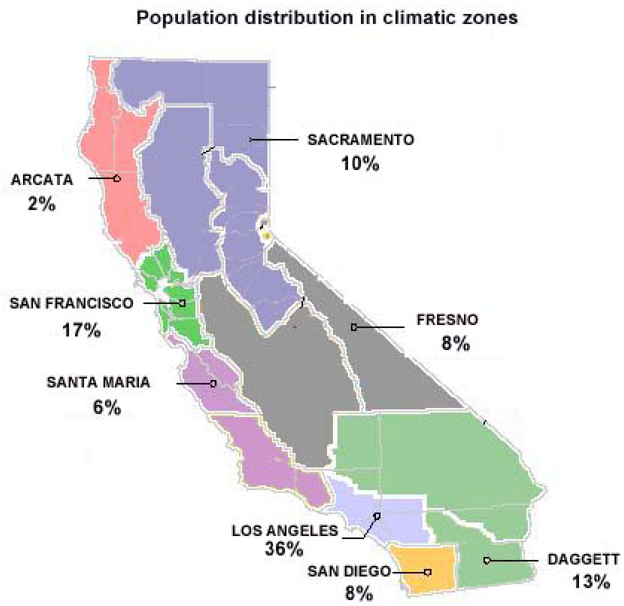


Figure 2.5 Climatic zones and population distribution.

2.2 Energy Savings

The quest for increased energy efficiency and the phase-out of ozone depleting substances (ODS) have changed the refrigeration system design for some new supermarkets. Many research laboratories as well as commercial chains or equipment manufacturers have assessed a great potential for energy efficiency improvement as well as limitation of greenhouse gas emissions. Energy saving technologies such as heat recovery, floating head pressure, defrosting control, energy efficient lighting, high efficiency motors, and efficient control have been implemented in many supermarkets to reduce energy consumption. The objective of these technical options is to develop energy-efficiency solutions in refrigerated cases, while enhancing food safety without hampering merchandizing facets.

2.2.1 Heat recovery systems

The refrigeration system in a supermarket always rejects heat to the environment through the condensers. It is possible to use the rejected heat for heating the store in the winter season. Heat rejected from condensers can also be used to heat service water and premises in cold climates, which is a good measure to improve energy usage in new efficient refrigeration cycle design. Heat recovery leads to a reduction in costs and in the usage of fossil fuels for heating.

2.2.2 Floating head pressure

A drawback of the heat recovery system is the high condensing temperatures that increase the energy consumption of the refrigeration system. In the so-called floating head pressure condensing systems, the condensing temperature follows the outdoor temperature. The system is implemented with electronic expansion valve (or multiple-orifice expansion valve) operating over a wide range of pressure differences and allowing for low condensing temperature at low ambient temperatures. A reduction of condensing temperatures increases the coefficient of performance of refrigeration systems.

2.2.3 Installing glass doors in open cases

Display cases commonly carry large heat loads, especially vertical open display cabinets. The reason is that this kind of cabinet displays a large amount of food on a small surface in the store with a large open front area. The heat and moisture exchanged between the products in the cabinet and the store environment affect the heat load, defrost and condensation, on walls and products. Infiltration causes about 60 to 70% of the cooling load for a typical open vertical display cabinet [ARI05].

The heat loads associated with the glass door reach-in case are normally less than those of the multi-deck, but greater than for the tub case. Glass door cases are, however, equipped with anti-sweat electric heaters in the doors to prevent fogging.

Installing glass doors in display cases reduces the infiltration and energy consumption of the cabinets. The reason for the absence of the doors in a display case is to avoid placing an obstacle between the customer and the product, which may hinder the customer impulse to purchase a new product. Results from a laboratory test that evaluated glass doors on a open five-deck display case show a reduction of the total cooling load of the case by 68% [FAR02].

2.2.4 Hot Gas Defrost

Discharge refrigerant gas is piped from the compressor rack to the display case where the refrigerant is condensed by melting the frost. The piping is arranged so that the liquid refrigerant is returned to the compressor rack for distribution to other display cases in the system. Hot gas defrosting is the fastest method to remove frost and tends to have the least impact on case air and product temperatures. Hot gas is the most costly defrosting method to implement because of the extensive piping and controls needed.

2.3 Results for energy consumption

2.3.1 Energy consumption in the commercial refrigeration sector

Results for energy consumption are presented first for supermarkets only, and second for all the commercial refrigeration sector, including small stores and vending machines.

2.3.1.1 Results for grocery supermarkets

One typical grocery supermarket

Before deriving the calculation for California State, one typical supermarket located in the Los Angeles (LA) climatic zone is presented. Table 2.14 gives the cooling capacity distribution, for medium and low temperature systems, and for each technology of display cases and walk-in coolers.

Table 2.14 Cooling capacity in a typical grocery supermarket.

Cooling Capacity	*Medium temperature*	*Low temperature*	*Total*
Centralized System (kW)	193	152	345
Condensing Units (kW)	18	14	32
Stand-alone (kW)	13	2	15
***Total** (kW)*	**224**	**168**	**392**

90% of the total cooling capacity of refrigeration equipment is the centralized system. Stand-alone display cases represent a total of 12 kW of cooling capacity, which is 3 % of the refrigeration capacity in the grocery supermarket.

Refrigerant charge is around 1300 kg including stand-alone equipment and condensing units. Centralized system represents 90% of the total amount. The evaluation of the refrigerant charge is presented in detail in Section 2.2.2 of the "Part 1 - Refrigerant inventory and emissions for stationary systems".

Energy consumption for one supermarket in LA climatic zone

Energy consumption is calculated hour by hour, for each climatic zone. Results for one supermarket in LA climatic zone are presented in Table 2.15.

Table 2.15 Annual energy consumption for 1 grocery supermarket in LA climatic zone.

Grocery supermarket (GWh/year)	***Centralized System***	***Condensing Units***	***Stand-alone***
Compressor for refrigeration	0.827	0.098	0.041
Auxiliary components	0.504	0.071	0.027
AC additional energy consumption	0	0	0.013
Total (GWh)	1.331	0.169	0.081
		1.581 GWh	

The energy consumption of refrigeration compressors is 0.827 GWh/year. Auxiliary components (fans, lighting, anti-sweat heaters, and defrosting heaters) totalize 0.504 GWh. As mentioned by different studies, (Wal03, Bax03, ORL04, Lit96) field tests on energy consumption measurement in a supermarket are in general agreement, with consumption numbers in the same order of magnitude.

Derivation to California State

Taking into account the different climatic zones, and the distribution of the stores in these zones, the energy consumption is derived to California (Table 2.16).

Table 2.16 Annual energy consumption for grocery supermarkets in California.

Grocery supermarkets in CA (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	2,810	334	137
Auxiliary components	1,692	237	92
AC additional energy consumption	0	0	39
Total (GWh)	4,502	571	268
		5,341 GWh	

Annual energy consumption evaluation of commercial refrigeration equipment in grocery supermarkets, including auxiliary electric loads, is 5,341 GWh in California. 84% is due to centralized systems, and 5% of the total energy consumption is due to stand-alone display cases.

2.3.1.2 Results including the other small stores using refrigeration equipment

Table 2.17 Annual energy consumption for commercial refrigeration sector in California, small stores included.

Total commercial ref. (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	2,810	3,831	4,548
Auxiliary components	1,692	2,364	3,802
AC additional energy consumption	0	0	1,181
Total (GWh)	4,502	6,196	9,531
		20,228 GWh	

When all types of stores are added to grocery supermarkets, the annual energy consumption grows to 20,228 GWh. The share by technology of refrigeration equipment is presented on Figure 2.6.

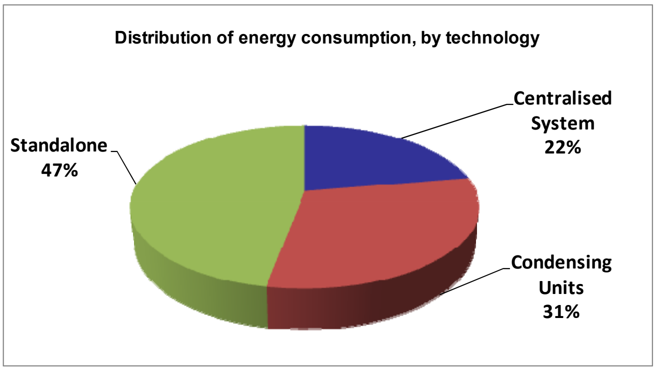


Figure 2.6 Distribution of energy consumption by technology, in commercial refrigeration.

Because of the high number of stand-alone equipment in small stores, including vending machines, and the poor efficiency of their refrigerating systems, this technology represents the largest share of energy consumption in the commercial refrigeration sector. It uses approximately 10 TWh per year in California. Centralized systems, only used in supermarkets, are more energy efficient and represent 22% of the global energy consumption.

2.3.2 Technical options for energy savings

Five technical options for energy savings have been evaluated.

- **Technical option 1:** night curtains are installed on each open display case. The ambient air induction is reduced, and the thermal load on the refrigeration system decreases. Night hours have been considered from 10 pm to 4 am. Moreover, during night hours, lighting is off in all display cases.
- **Technical option 2:** all medium-temperature open display cases (except for vegetables and flowers) are replaced by glass door display cases. Ambient air induction is significantly reduced (by factor 7), decreasing the thermal load of the display case and the energy consumption of the refrigeration system.
- **Technical option 3:** auxiliary components are replaced by new technologies, with improved energy efficiency (LED lighting, DC current fan, high efficiency heater…)
- **Technical option 4:** the floating head pressure control is done on every centralized system in supermarkets. Depending on the climatic zone, the impact is more or less significant on the annual energy consumption.

- **Technical option 5:** three options are combined: 100% glass door + high efficiency electrical components + floating head pressure control.

2.3.2.1 Technical option 1: night curtain installed on every open display case

Grocery supermarkets

Table 2.18 presents the results for one typical supermarket in LA climatic zone. Energy savings, thanks to night curtains installed on every open display case in a supermarket is 92 MWh/year, 5.82% of the energy consumption without night curtain.

Table 2.18 Energy consumption for one grocery supermarket – Technical option 1.

Operating Mode	Night curtains		
Grocery supermarket (GWh/year)	**Centralized System**	**Condensing Units**	**Stand-alone**
Compressor for refrigeration	0.745	0.096	0.038
Auxiliary components	0.497	0.072	0.028
AC additional energy consumption	0	0	0.013
Total	1.242	0.168	0.079
		1.489	
	Energy Savings (GWh ; %)	0.092	5.82%

Table 2.19 Energy consumption for all supermarkets in California – Technical option 1.

Operating Mode	Night curtains		
Grocery supermarkets in CA (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	2,533	327	128
Auxiliary components	1,666	240	94
AC additional energy consumption	0	0	37
Total	4,200	567	259
		5,027	
	Energy Savings (GWh ; %)	313	6%

Deriving the calculation for all supermarkets in California, the energy saving associated to the installation of night curtain is 313 GWh/year.

All commercial refrigeration equipment (small stores and supermarkets) in California

Considering now all refrigeration equipment, the additional savings is limited and evaluated between 313 and 351 GWh, because most of stand-alone equipment is already closed with glass doors and the night curtain technical option has no effect on this equipment.

Table 2.20. Energy consumption in commercial refrigeration sector – Technical option 1.

Operating Mode	Night curtains		
Total commercial ref. (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	2,534	3,797	4,532
Auxiliary components	1,666	2,365	3,805
AC additional energy consumption	0	0	1,178
Total	4,200	6,162	9,515
		19,877	
	Energy Savings (GWh ; %)	351	2%

2.3.2.2 Technical option 2: Open display cases closed with glass doors

The impact of night curtain is limited to 6 hours in the night when the coefficient of performance increases due to reduced outdoor temperatures. Closing all open display cases with doors represents a more radical change in display case technology to reduce energy consumption.

Grocery supermarkets

Table 2.21 Energy consumption for one grocery supermarket – Technical option 2.

Operating Mode	Add doors		
Grocery supermarket (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	0.539	0.084	0.034
Auxiliary components	0.611	0.074	0.031
AC additional energy consumption	0	0	0.011
Total	1.15	0.158	0.076
		1.384	
	Energy Savings (GWh ; %)	0.197	12.46%

Closing all the display cases, the energy savings for one year is 12.5%: 200 MWh per supermarket.

Table 2.22 Energy consumption for all supermarkets in California – Technical option 2.

Operating Mode	Add doors		
Grocery supermarkets in CA (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	1,812	281	114
Auxiliary components	2,050	250	105
AC additional energy consumption	0	0	32
Total	3,862	531	251
		4,644	
Energy Savings (GWh ; %)		697	13.05%

Deriving the scenario to California, the energy saving is nearly 0.7 TWh per year.

All commercial refrigeration equipment (small stores and supermarkets) in California

Table 2.23 Energy consumption in commercial refrigeration sector – *Technical option 2.*

Operating Mode	Add doors		
Total commercial ref. (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	1,812	3,412	4,518
Auxiliary components	2,050	2,374	3,816
AC additional energy consumption	0	0	1,173
Total	3,862	5,786	9,507
		19,155	
Energy Savings (GWh ; %)	1,073		5.305%

Most of stand-alone equipment is already equipped with glass doors. The impact on energy savings is significant mainly in supermarkets. Nevertheless, for the complete commercial refrigeration sector, the energy savings are 5.3 % compared to the baseline.

2.3.2.3 Technical option 3: Cabinet lighting, anti-sweat heater and ventilation: low energy consuming technologies

Grocery supermarkets

Table 2.24 Energy consumption for one grocery supermarket – Technical option 3

Operating Mode	Eco for auxiliary components		
Grocery supermarket (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	0.698	0.078	0.038
Auxiliary components	0.358	0.047	0.023
AC additional energy consumption	0	0	0.013
Total	1.056	0.125	0.074
		1.255	
Energy Savings (GWh ; %)		0.326	20.62%

Auxiliary components are energy consumers, first by their own electrical load, and second by the additional heat load to the display case. This additional heat load increases the cooling capacity of the refrigeration system, and its energy consumption.

Improved technologies are available to retrofit lighting, ventilation, and anti-sweat heaters. Technical option 3 gives the range of energy savings if all auxiliary components were replaced.

Table 2.25 Energy consumption for all supermarkets in California – Technical option 3.

Operating Mode	Eco for auxiliary components		
Grocery supermarkets CA (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	2,345	261	128
Auxiliary components	1,200	159	77
AC additional energy consumption	0	0	37
Total	3,545	420	241
		4,206	
Energy Savings (GWh ; %)		1,134	21.24%

Energy savings, thanks to high-efficiency auxiliary components, is around 21% compared to the baseline. In California, the annual savings are 1.2 TWh for this technical option.

All commercial refrigeration equipment (small stores and supermarkets) in California

Table 2.26 Energy consumption in commercial refrigeration sector – Technical option 3.

Operating Mode	Eco for auxiliary components		
Total commercial ref. (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	2,345	3,058	4,112
Auxiliary components	1,200	1,488	3,125
AC additional energy consumption	0	0	1,062
Total	3,545	4,546	8,298
		16389	
Energy Savings (GWh ; %)		3,840	19.0%

Technical option 3 is applied to each type of display cases. Stand-alone equipment can benefit of the technical changes. Deriving the scenario to California, for commercial refrigeration sector, energy savings for one year are 3.84 TWh, nearly 19% of the baseline consumption.

2.3.2.4 Technical option 4: Floating head pressure on centralized systems

Grocery supermarkets

Table 2.27 Energy consumption for one grocery supermarket – Technical option 4.

Operating Mode	Floating head pressure (Eco)		
Grocery supermarket (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	0.778	0.089	0.041
Auxiliary components	0.504	0.071	0.027
AC additional energy consumption	0	0	0.013
Total	1.282	0.16	0.081
		1.523	
Energy Savings (GWh ; %)		0.058	3.67%

Table 2.27 presents the energy consumption for one supermarket in LA climatic zone, when floating head pressure control is activated. The interest of this control, and the energy savings associated are strongly dependent on the temperature changes during the year. In a climatic zone where maximum and minimum temperatures are not far from each other, the interest of a floating head pressure is limited. In LA climatic zone, the energy saving is 3.7%. The derivation of energy consumption in California, taking into account 8 climatic zones, give a better result with 5% of energy savings thanks to the floating head pressure control.

Table 2.28 Energy consumption for all supermarkets in California – Technical option 4.

Operating Mode	Floating head pressure (Eco)		
Grocery supermarkets in CA (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	2,583	296	137
Auxiliary components	1,692	237	92
AC additional energy consumption	0	0	39
Total	4,275	534	268
		5,077	
Energy Savings (GWh ; %)		264	5%

All commercial refrigeration equipment (small stores and supermarkets) in California

Table 2.29 Energy consumption in commercial refrigeration sector – Technical option 4.

Operating Mode	Floating head pressure (Eco)		
Total commercial ref. (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	2,583	3,400	4,548
Auxiliary components	1,692	2,364	3,802
AC additional energy consumption	0	0	1,181
Total	4,275	5,764	9,531
		1,9570	
Energy Savings (GWh ; %)		658	3.251%

Floating head pressure cannot be applied to stand-alone equipment, which are located in an air-conditioned area. The overall impact of this technical option, on complete commercial refrigeration sector is lowered to 3.3%, representing 0.66 TWh per year.

2.3.2.5 Technical option 5: three options combined

The last scenario is the combination of three technical options: closing all open display cases, low energy consuming components, and floating head pressure.

Grocery supermarkets

In a typical supermarket, located in LA climatic zone, the annual energy savings is 37%, meaning 811 MWh saved (see Table 4.17).

Table 2.30 Energy consumption for one grocery supermarket – Technical option 5.

Operating Mode	Add doors + eco Aux + FHP		
Grocery supermarket (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	0.489	0.071	0.031
Auxiliary components	0.442	0.05	0.026
AC additional energy consumption	0	0	0.01
Total	0.931	0.121	0.067
		1.119	
Energy Savings (GWh ; %)		0.462	29.22%

Table 2.31 Energy consumption for all supermarkets in California – Technical option 5.

Operating Mode	Add doors + eco Aux+Eco FHP		
Grocery supermarkets in CA (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	1,621	235	103
Auxiliary components	1,482	168	86
AC additional energy consumption	0	0	29
Total	3,103	403	218
		3,724	
Energy Savings (GWh ; %)		1,617	30.273%

For all supermarkets in California, the maximum energy savings are 1.62 TWh per year when all technical options are applied.

All commercial refrigeration equipment (small stores and supermarkets) in California

Table 2.32 Energy consumption in commercial refrigeration sector – Technical option 5.

Operating Mode	Add doors + eco Aux+Eco FHP		
Total commercial ref. (GWh/year)	*Centralized System*	*Condensing Units*	*Stand-alone*
Compressor for refrigeration	1,621	2,955	4,080
Auxiliary components	1,482	1,495	3,135
AC additional energy consumption	0	0	1,052
Total	3,103	4,450	8,267
		15,820	
Energy Savings (GWh ; %)		4,409	21.8%

For the complete commercial refrigeration sector, the maximum energy savings is 4.41 TWh per year, totalizing 22% of the baseline consumption. Overall annual energy consumption is evaluated at 15.8 TWh, and stand-alone equipment consumes more than half of this value.

2.3.2.6 Summary

Energy savings in supermarkets

Figure 2.7 presents the comparison of the energy savings, related to the baseline, for different technical options applied to grocery supermarkets in California.

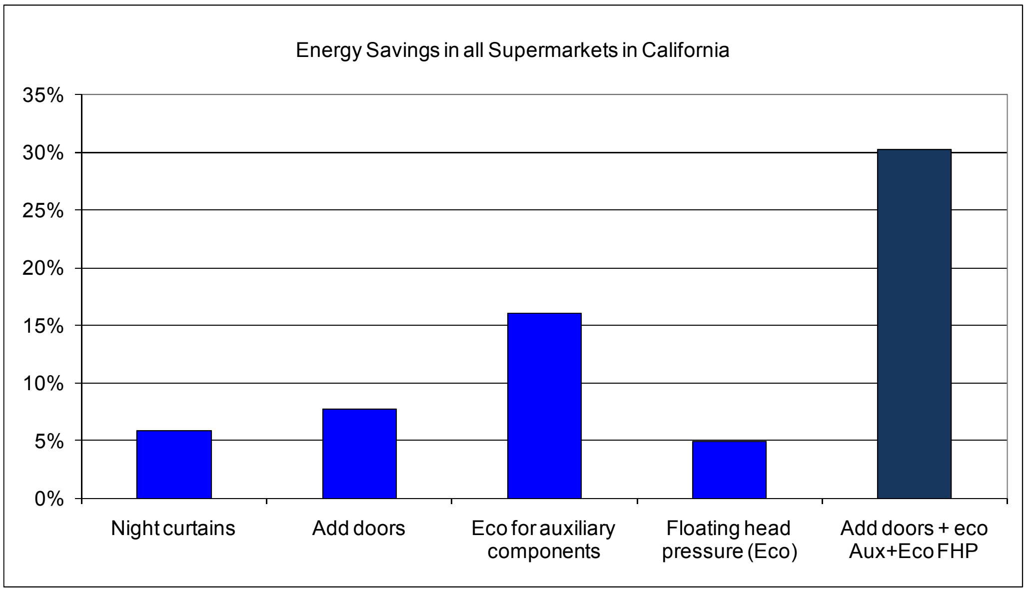


Figure 2.7 Energy savings / technical options applied in Californian supermarkets.

- Night curtains installed on open display cases have a limited impact on the energy consumption. Night period is short in time in supermarkets (6 hours only) and during this period, the coefficient of performance of the refrigerating system is improved thanks to quite low outdoor temperature, lowering the condensation temperature.
- Floating head pressure control is interesting in climatic zones with wide temperature differences between day and night. Near the coast, where the temperature is more stable, the interest of this system is limited.
- In supermarkets, most of display cases are open, and heat loads due to air induction is around 70% of the total load. Closing the display cases decreases the cooling capacity and the energy consumption of the compressor racks. 8 % of energy saving are possible with this change in technology.
- The other elements for energy consumption are the auxiliary components. High energy efficiency technologies exist and could reduce by 16% the energy consumption.
- All options applied together lead to 30% of energy savings in supermarkets

Energy savings in small stores (condensing units and standalone equipment)

Figure 2.8 presents the comparison of the energy savings, related to the baseline, for different technical options applied to small stores in California. Stand-alone equipment and condensing units are the two refrigeration technologies used in small stores.

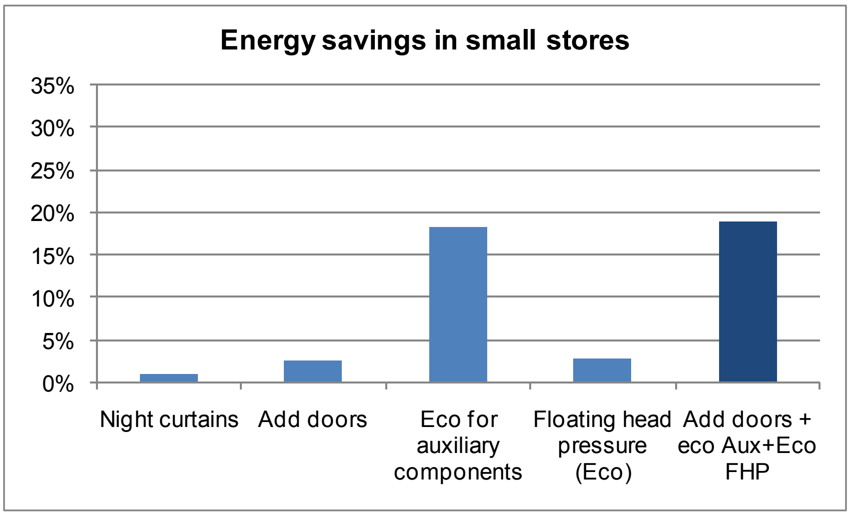


Figure 2.8 Energy savings / technical options applied in Californian small stores.

Most of stand-alone equipment is closed with glass doors (vending machines for example). Options of night curtains and closing the display cases are not applied on this stand-alone equipment. The impact on energy consumption is low.

Progresses to save energy on stand-alone equipment must be focused on auxiliary components and compressor efficiency, which is very poor today.

Energy savings in commercial refrigeration sector, all types of stores

Figure 2.9 presents the comparison of the energy savings, related to the baseline, for different technical options applied to all types of stores using refrigeration equipment in California, whatever the technology.

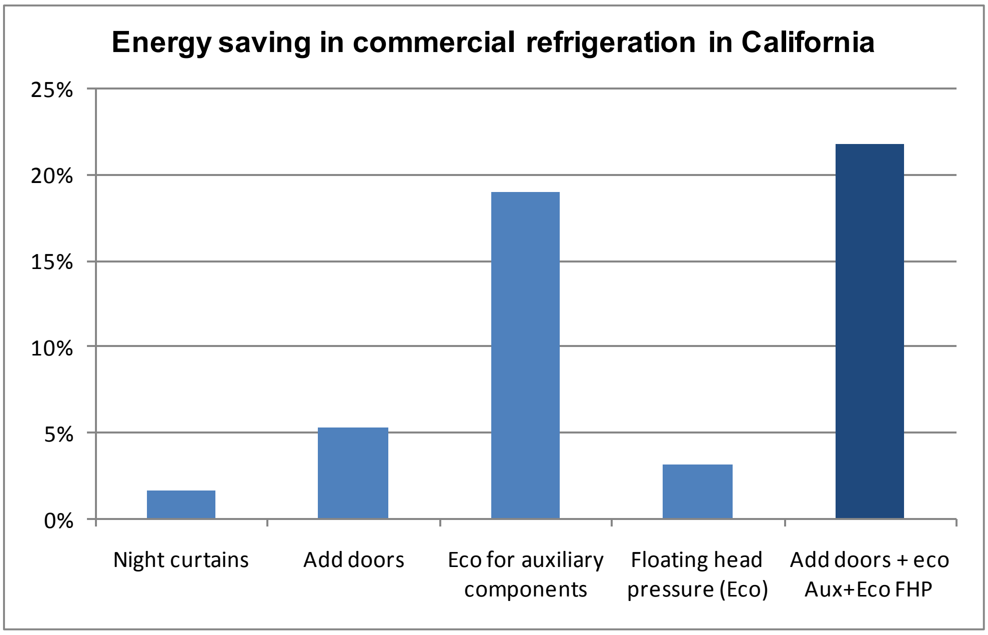


Figure 2.9 Energy savings / technical options applied in commercial sector.

Figure 2.10 presents for technical option 5, where all technical options are applied, the distribution in energy consumption by technology of refrigerating system.

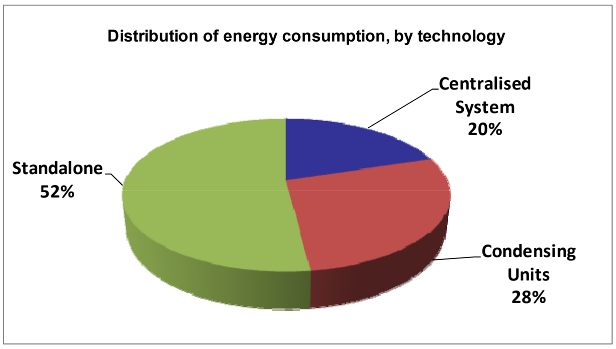


Figure 2.10 All technical options combined: energy consumption distribution.

It appears clearly that stand-alone equipment, by their high numbers in every type of stores, consumes more than 50% of the total energy consumption in commercial refrigeration sector. The poor efficiency of small hermetic compressor, and sometime heat exchanger designs not adapted, lead to a poor cycle efficiency (25%). Technical options to reduce energy consumption are more effective for centralized system. In technical option 5, the energy consumption of centralized systems is cut by nearly 40% compared to the baseline.

2.4 General approach for Life Cycle Cost Analysis

2.4.1 Calculation method

The Life Cycle Cost (LCC) is the total customer cost over the lifetime of the equipment, including purchase cost and operating cost (including energy cost). Future operating costs are discounted to the time of purchase and summed over the lifetime of the equipment. Inputs to the LCC analysis are categorized as follows:

- inputs for establishing the purchase cost, otherwise known as the total installed cost, and
- inputs for calculating the operating cost (i.e., energy, maintenance, and repair costs).

Life-cycle cost is defined by Equation (2.12):

$$LCC = IC + \sum_{t=1}^{N} \frac{OC_t}{(1+r)^t} \qquad (2.12)$$

Where
LCC life-cycle cost ($),
IC total installed cost ($),
N lifetime of equipment expressed in years,
Σ sum over the lifetime, from year 1 to year N,
OC operating cost ($),
r discount rate (4.76% [DOE07]),
t year for which operating cost is being determined.
Because most of data used to conduct the LCC analysis are collected in 2008, all costs are expressed in US $ (2008).

The LCC analysis is performed for different efficiency levels and LCC difference between the baseline equipment and equipment with higher efficiency level is evaluated. A distribution of LCC differences is then generated to determine the mean LCC difference.

2.4.1.1 Total installed cost

The primary inputs for establishing the total installed cost are: the baseline manufacturer selling price, mark-ups and sales tax, and the installation price.

2.4.1.2 Baseline manufacturer selling price

Baseline manufacturer selling price is the price charged by the manufacturer to either a wholesaler or customer for equipment meeting existing minimum efficiency (or baseline) standards. The manufacturer selling price includes a markup that converts the cost (i.e., the manufacturer cost) to a manufacturer selling price. Standard-level manufacturer selling price increase: standard-level manufacturer selling price increase is the incremental change in manufacturer selling price associated with producing equipment at each of the higher standard levels.

Markups and sales tax

Markups and sales tax convert the manufacturer selling price into a customer price.

Installation price

The installation price is the cost to the customer of installing the equipment. The installation price represents all costs required to install equipment but does not include the marked-up customer equipment price. The installation price includes labor, overheads, and any miscellaneous materials and parts. Thus, the total installed cost equals the customer equipment price plus the installation price and is defined by Equation (2.13):

$$IC = EP + InstC \tag{2.13}$$

where

EP equipment price (i.e., customer price for the equipment only), expressed in $,

InstC the installation cost or the customer price to install equipment (i.e., the cost for labor and materials), also expressed in $.

The equipment price includes the manufacturing cost of equipment multiplied by different markups. A first markup, “the baseline manufacturer markup”, converts the manufacturing cost to a manufacturer selling price, which is the price charged by manufacturers to either a wholesaler/distributor or a very large customer for existing equipment. All associated retail markups and applicable sales tax markup together are then multiplied and expressed as the “overall markup”. The overall markup in turn is multiplied by a “baseline manufacturer selling price” to attain the price paid by the customer as stated in Equation (2.14):

$$EP = OMU \times BMU \times MFC \tag{2.14}$$

Where

OMU Overall markup.

BMU Baseline manufacturer markup

MFC Manufacturing cost

The installation cost is the price to the customer of labor and materials (other than the actual equipment) needed to install the refrigeration equipment. Installation costs were derived for commercial refrigeration equipment from data provided by the DOE based on RS Means Mechanical Cost Data.3. RS Means provides estimates on person-hours required to install commercial refrigeration equipment and labor rates associated with the type of crew required to install the equipment [DOE07].

The installation cost is then calculated by multiplying the number of person-hours by the corresponding labor rate. Since labor rates vary significantly from one region to another, the regional variability is taken into account and is expressed in terms of cost indices for 50 states as shown in Table 2.33.

The total installed cost is therefore expressed as shown in Equation (2.15):

$$IC_{CA} = OMU \times BMU \times MFC + InstC_{USA} \times \frac{II_{CA}}{II_{USA}} \tag{2.15}$$

where II represents the cost installation index and CA refers to California. This method is applied to display cases and self contained categories defined in section 1.4.

Table 2.33 Installation cost indices (national average value = 100).

State	Index	State	Index	State	Index
Alabama	65.4	Kentucky	73.2	North Dakota	67.0
Alaska	117.1	Louisiana	60.9	Ohio	103.0
Arizona	79.1	Maine	76.9	Oklahoma	67.3
Arkansas	53.7	Maryland	92.1	Oregon	115.3
California	123.8	Massachusetts	123.1	Pennsylvania	128.5
Colorado	88.3	Michigan	112.3	Rhode Island	120.9
Connecticut	111.7	Minnesota	122.8	South Carolina	42.6
Delaware	125.1	Mississippi	41.6	South Dakota	40.1
Dist. of Columbia	97.7	Missouri	104.0	Tennessee	75.2
Florida	64.8	Montana	80.9	Texas	66.7
Georgia	67.3	Nebraska	83.7	Utah	76.6
Hawaii	126.6	Nevada	113.1	Vermont	73.6
Idaho	78.5	New Hampshire	91.9	Virginia	70.8
Illinois	129.1	New Jersey	132.3	Washington	109.8
Indiana	91.7	New Mexico	78.3	West Virginia	93.5
Iowa	85.6	New York	166.3	Wisconsin	99.3
Kansas	75.0	North Carolina	46.4	Wyoming	56.4

2.4.1.3 Operating cost

The operating cost includes the equipment energy consumption, repair cost associated with component failure, and maintenance cost for equipment operation as expressed in Equation (2.16) :

$$OC = EC + RC + MC \tag{2.16}$$

OC Operating cost, expressed in $,
EC Energy cost associated with operating the equipment, in $,
RC Repair cost associated with component failure, in $,
MC Service cost for maintaining equipment operation, in $,
Several primary inputs are needed to evaluate the operating cost such as: the lifetime, discount rate, electricity prices, and electricity price trends.

Equipment energy consumption

The equipment energy consumption is the site energy use associated with the use of commercial refrigeration equipment. Although there are potentially some minor interactive effects on the overall heating and cooling of the building, for purposes of the ANOPR (Advanced

Notice for Proposed Rulemaking), the LCC analysis includes only the use of electricity by the equipment itself. This approach is consistent with most other DOE equipment efficiency rulemakings.

Maintenance costs

The maintenance cost is the cost to the consumer associated with general maintenance, such as checking and maintaining refrigerant charge levels, cleaning heat exchanger coils,... Annualized maintenance costs for commercial refrigeration equipment were taken from DOE reports based on RS Means Facilities Maintenance & Repair Cost Data [DOE07]. RS Means provides estimates on the person-hours, labor rates, and materials required to maintain commercial refrigeration equipment.

Maintenance costs include both preventive activities and lighting maintenance. For commercial display cases, preventive maintenance activities expected to occur on a semi-annual basis include the following actions: cleaning evaporator coils, drain pans, fans and intake screens; lubricating motors; inspecting door gaskets and seals, and lubricating hinges; cleaning condenser coils; checking refrigerant pressures and compressor oil as necessary; checking starter panels and controls; and checking defrost system operation. However, these activities were not broken into separate line-item maintenance activities since no detailed data were available.

A single figure of $156/yr (in 2008 $) for preventive maintenance activities is applied for all commercial refrigeration (DOE value). Moreover, preventive maintenance costs remain constant as equipment efficiency increases since no data were available to indicate how maintenance costs vary with equipment efficiency level.

Lamp replacements and other lighting maintenance activities are considered apart from preventive maintenance and are required for commercial refrigeration equipment. Because the lighting configurations can vary by equipment class and efficiency level, the relative maintenance cost are estimated for each case type and lighting technology. The frequency of failure and replacement of individual lighting components are estimated based on DOE report [DOE07], then an annualized maintenance cost is defined as the sum of the total lighting maintenance costs (in 2008 $) over the estimated equipment lifetime divided by the estimated equipment lifetime.

Lifetime estimates for particular components were as follows:

- Fluorescent lamps would be replaced every 24 months in a preventive mode
- Fluorescent lamp ballasts would be replaced once over the estimated 10-year life of the equipment based on a typical ballast life of 80,000 hours
- LED lamps would be replaced once over the estimated 10-year life of the equipment based on a typical fixture life of 50,000 hours.

Repair costs

Those costs cover the labor and materials costs associated with repairing or replacing components that have failed. The repair cost is the cost to the consumer for replacing or repairing components in the commercial refrigeration equipment that have failed. The annualized repair cost for baseline energy consumption commercial refrigeration equipment (i.e., the cost the customer pays annually for repairing the equipment) is based on Equation (2.17) developed by the DOE:

$$RC = k\frac{EP}{N} \qquad (2.17)$$

Where

k	Fraction of the equipment price (a value of 0.5 was assumed)
EP	Equipment price expressed in $
N	Average lifetime of the equipment in years (a value of 10.0 years was assumed).

Since no data were available to indicate how repair costs vary with equipment efficiency level, they were taken constant.

Lifetime

Lifetime *t* expresses the age at which the commercial refrigeration equipment is retired from service. A typical lifetime of 10 years is appropriate for commercial refrigeration equipment based on [DOE07] and discussion with experts.

Discount rate

The discount rate "*r*" expresses the rate at which future costs are discounted to establish their present value. The discount rate varies accordingly with economic sectors and store categories. Based on [DOE07], a discount rate of 4.76% is considered after deducting expected inflation from the cost of capital.

Electricity prices

Electricity prices used in the analysis are the price per kilowatt-hour in cents or dollars (e.g., cents/kWh) paid by each customer for electricity. Because of the wide variation in electricity consumption patterns, wholesale costs, and retail rates across the US, regional differences in electricity prices were considered. Electricity prices are determined using average commercial electricity prices in each state, as determined from Energy Information and used by the Department of Energy of the US government. Table 2.34 provides data on the adjusted electricity prices for different states.

Table 2.34 Commercial electricity prices cents/kWh ([DOE07]).

State	Commercial Electricity Price	State	Commercial Electricity Price	State	Commercial Electricity Price
Alabama	7.32	Kentucky	5.73	North Dakota	6.02
Alaska	11.20	Louisiana	7.92	Ohio	8.06
Arizona	7.57	Maine	11.04	Oklahoma	6.82
Arkansas	5.91	Maryland	7.42	Oregon	6.81
California	13.01	Massachusetts	11.18	Pennsylvania	9.20
Colorado	7.05	Michigan	8.06	Rhode Island	10.77
Connecticut	10.60	Minnesota	6.54	South Carolina	7.27
Delaware	7.81	Mississippi	7.74	South Dakota	6.44
Dist. of Col.	7.85	Missouri	6.17	Tennessee	7.13
Florida	7.62	Montana	7.31	Texas	8.37
Georgia	7.11	Nebraska	6.20	Utah	5.96
Hawaii	16.04	Nevada	9.38	Vermont	12.05
Idaho	5.94	New Hampshire	10.99	Virginia	6.13
Illinois	7.79	New Jersey	9.72	Washington	6.48
Indiana	6.54	New Mexico	7.85	West Virginia	5.82
Iowa	6.67	New York	13.80	Wisconsin	7.44
Kansas	6.85	North Carolina	7.10	Wyoming	6.13

Furthermore, DOE recognized that different kinds of businesses typically use electricity in different amounts at different times of the day, week, and year, and therefore face different effective prices. To make this adjustment, average prices paid by the four kinds of businesses were identified and considered in this analysis compared with the average prices paid by all commercial customers.

$$Eprice_{Bld,CA} = Eprice_{CA} \times \frac{Eprice_{Bld,USA}}{Eprice_{USA}} \tag{2.18}$$

Where

$Eprice_{Bld,CA}$	average commercial sector electricity price in a specific building in California in year 2008
$Eprice_{CA}$	average commercial sector electricity price in California in year 2008
$Eprice_{Bld,USA}$	national average commercial sector electricity price in the considered building
$Eprice_{USA}$	national average commercial sector electricity price.

Table 2.35 shows the derivation of electricity price ratios for different businesses/building types.

Table 2.35 Electricity price ratios for different businesses ([DOE07]).

Business type	Grocery/ Store food	Convenience store	Convenience store with gas station	Other	All food sales	All Commercial buildings
Electricity Price (cents/kWh)	7.2	8.6	7.7	8.2	7.6	7.8
Ratio of electricity price to average price for commercial buildings	0.92	1.10	0.99	1.05	0.97	1.00

Electricity price trends

The electricity price trend provides the relative change in electricity prices for future years out to year 2017 corresponding to a lifetime of 10 years considered for this study. The EIA's Annual Energy Outlook AEO 2006 reference case is applied to forecast future electricity prices for the LCC analysis presented in this work. Figure 2.11 illustrates the electricity price trend.

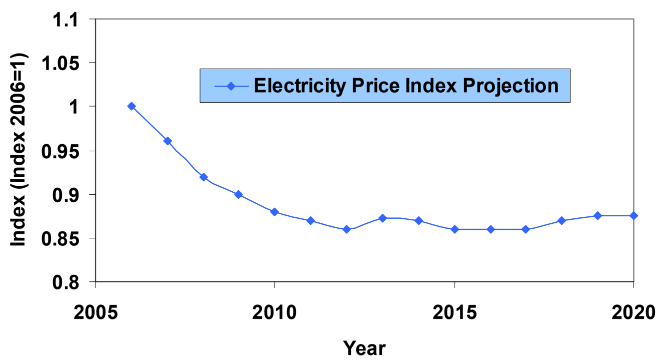


Figure 2.11 Electricity price trend out to year 2020 ([DOE07]).

2.4.1.4 Pay Back Period (PBP)

The PBP (Pay Back Period) is the change in purchase cost due to an increased efficiency standard divided by the change in annual operating cost that results from the standard. It represents the number of years it will take the customer to recover the increased purchase cost through decreased operating costs. In the calculation of PBP, future costs are not discounted.

Inputs to PBP analysis are categorized as presented for the LCC analysis, i.e. inputs for establishing the total installed cost and inputs for calculating the operating cost.

Numerically, the PBP is the ratio of the increase in purchase cost (i.e., from a less efficient design to a more efficient design) to the decrease in annual operating expenditures. The Equation (2.19) shows PBP expression:

$$PBP = \frac{\ln\left(\frac{\Delta OC}{\Delta IC}\right) - \ln\left(\frac{\Delta OC}{\Delta IC} - \frac{r - apel}{1 + apel}\right)}{\ln\left(\frac{1+r}{1+apel}\right)} \tag{2.19}$$

where

PBP Payback period in years,

ΔIC Difference in the total installed cost between a more efficient equipment and baseline equipment,

ΔOC Difference in annual operating costs,

r discount rate,

apel Actualization rate of the electricity price.

Payback periods are expressed in years. Payback periods greater than the life of the product mean that the increased total installed cost of the more efficient equipment is not recovered in reduced operating costs over the life of the equipment. Hence, The PBP can be computed only when the following condition is fulfilled (Equation (2.20)):

$$\frac{\Delta OC}{\Delta IC} = \frac{OC_{Base} - OC_{Efficient}}{IC_{Efficient} - IC_{Base}} > \frac{r - apel}{1 + apel} \tag{2.20}$$

In the present work, two LCC analyses are conducted. The first one aims at defining the optimal aggregation of technical options for energy savings in a supermarket. The second analysis establishes the distribution of LCC differences between the baseline refrigerating system (direct expansion) and other systems (distributed and secondary loop systems).

2.4.2 Life cycle cost assessment (LCCA) of the technical options in an aggregated model

This section presents LCC results for higher efficiency and energy saving options specified in the previous intermediate report. A screening analysis was conducted in order to choose the technologies to be evaluated, and implemented as design options in the energy consumption model. The investigated design options are:

- Higher efficiency lighting and ballasts for equipment families (LED lighting),
- Closing open display cases with glass doors,
- Higher efficiency evaporator fan motors (ECM motors),
- Defrost cycle control,
- Anti-sweat heater control,
- Installing night shields for open display cases.

First, baseline case LCC and each technical option LCC are calculated separately, and then aggregated models of technical options are studied. For a given option, LCC is calculated according to the methodology presented in section 2.4.1. Hence, the impact of applying a technical option to baseline equipment is evaluated in terms of total installed cost and operating cost.

2.4.2.1 Baseline scenario

Before evaluating incremental cost induced by applying energy saving options, the baseline supermarket LCC must be evaluated. The DOE provided in its Technical Support Document for commercial refrigeration the baseline equipment price, maintenance, repair, and installation costs [DOE07]. Therefore, the Commercial Refrigeration Equipment Families defined in the intermediate report were compared to the families defined in the DOE document in order to define corresponding cost and prices.

illustrates commercial refrigeration display case families and detailed costs used to establish the LCC analysis.

Table 2.36 Display cases categories and corresponding installed and operating costs [DOE07].

DESCRIPTION			***INSTALLED COST***				***OPERATING COST***						
Position	*Family*	*T° level*	*MSP ($)*	*EP ($)*	*InstC ($)*	*IC ($)*	*RC ($)*	*PM ($)*	*LM ($)*	*MC ($)*	*EC ($/2008)*	*OC ($)*	*Symbol*
Vertical	*Open*	*Medium*	3944	5478	365	6118	274	156	132	288	1807	2369	DC-1/ DC-10
Vertical	*Open*	*Low*	6636	9217	365	9857	461	156	43	199	4671	5331	DC-6
Vertical	*Closed*	*Medium*	6546	9092	365	9742	455	156	70	226	893	1574	DC-2 / DC-14
Vertical	*Closed*	*Low*	6664	9256.	365	9906	463	156	96	252	1964	2679	DC-7
Semi-Vertical	*Open*	*Medium*	3890	5403	365	6043	270	156	88	244	1307	1821	DC-3
Service Over Counter	*Closed*	*Medium*	7960	11056	365	11696	553	156	73	229	1240	2022	DC-12
Horizontal	*Open*	*Medium*	3922	5448	365	6087	272	156	0	156	254	683	DC-5 / DC-9
Horizontal	*Open*	*Low*	4134	5742	365	6382	287	156	0	156	1375	1818	DC-13

EC:	energy cost associated with equipment operation	EP:	equipment price
IC:	total installed cost	InstC:	installation cost
LM:	lighting maintenance.	MC:	service cost for maintaining equipment operation
MSP:	manufacturer selling price	OC:	operating cost
PM:	preventative maintenance costs	RC :	repair cost associated with component failure

2.4.2.2 Technical options for energy savings

A number of technologies that could potentially be used to improve the efficiency of commercial refrigeration equipment was considered to evaluate energy savings in commercial refrigeration. These include higher efficiency lighting, higher efficiency fan motors, defrost cycle control, anti sweat heater control, and door installation for open display cases (except for display cases dedicated for vegetables and fruits).

Higher efficiency lighting (LED)

Higher efficiency lighting leads to energy savings in two ways: less energy is used directly for lighting, and less heat energy is dissipated into the refrigerated case by lamps. The most recent trend in case lighting is the use of light emitting diode (LED) technology that allows comparable product illumination with less total wattage. Therefore, LED technology will be considered to evaluate energy savings from lighting.

Higher efficiency evaporator fan motors (ECM)

The electronically commutated permanent magnet motor (ECM), a three-phase electric motor, is more energy efficient than either shaded pole or PSC motors but ECM motors are more expensive than equivalent PSC motors. ECM motors are regarded in this report as an energy saving technical option, and energy savings induced are evaluated.

Anti-sweat heaters controllers (ASH)

Anti-sweat heating is necessary to prevent moisture condensation on surfaces of display cases, whose temperature can drop below ambient dew point. Anti-sweat heating controllers match the on-time of the anti-sweat heaters to the anti-sweat heating requirements imposed by the ambient humidity, reducing energy consumption when the ambient humidity is low.

Defrost Cycle Control

As the air in the refrigerated space is cooled, water vapor condenses on the evaporator coil surface. In refrigerators and freezers, where evaporator coil is below 32°F, this water freezes, forming a growing frost layer, increasing the thermal resistance to heat transfer from the coil to the air, reducing thus the cooling performance. Among available defrosting mechanisms, electric defrost is investigated in this study. It involves melting frost by briefly turning on an electric resistance heater, near or in contact with the evaporator coil. However, for energy saving purposes, defrost cycle control is needed to minimize energy required for defrosting. Defrost cycle control considered in this report involves management of the initiation and termination of defrost cycles, and thereby the frequency and duration of defrosting cycles according to frost conditions determined by temperature sensors.

Door installation for open display cases

Refrigerated display cases without doors allow consumers easy access to products while maintaining temperatures that ensure food safety. The heat load of such cases is dominated by entrainment of warm and moist air into the case (called infiltration). Reduction in total case energy consumption can be achieved by installing doors for open display cases whenever possible, in order to reduce the infiltration load as well as the induced frost formation on the evaporator coil.

2.4.2.3 Total installed cost

The total installed cost equals the customer equipment price plus the installation price. Therefore, implementing a new option may incur incremental costs on either equipment price, installation cost or on both. These cost increases are based on data taken from technical literature. Table 5.4 illustrates, for each energy saving option, the corresponding incremental cost to be added to equipment price and installation cost as well as the reference where these values are taken from. Blank cells in Table 2.37 mean no incremental cost is incurred.

It should be noted that installation cost does not taken into account the cost of replacing baseline options by energy saving ones.

Table 2.37 Impact of energy saving options on total installed cost.

Technical Options	***Equipment price***	***Installation cost***	***Reference***
LED Lighting	Increase of 53$		[CCR08], [SMA08]
ECM motors	Increase of 50$		[ACE04]
ASH control	Increase of 14 $/ft of cabinet length		[PGE07]
Doors Installation	Replace by equivalent equipment with doors		Results of Simulations
Night Shield	Increase of 204 $/m of cabinet length		[PGE07]
Defrost Control	Increase of 14 $/ft of cabinet length		[CCR08]

2.4.2.4 Total operating cost

The operating expenses include repair and maintenance costs as well as energy consumption cost. The energy consumption of commercial refrigerating equipment is based on the model developed and presented in Section 0. Table 2.38 shows the impact of applying energy saving options on maintenance, repair, and energy costs as well as references where these values are taken from.

Table 2.38 Impact of implementing higher efficiency options on operating cost.

Technical option	***Baseline case***	***Repair Cost***	***Preventive Maintenance Cost***	***Lighting maintenance***	***Energy Cost***
LED Lighting	T8 linear fluorescent lighting	Lower repair frequency		Lower maintenance frequency	Lower lighting power
ECM motors	Brushless DC motors				Lower fan consumption
ASH control	No ASH control				Lower energy consumption
Doors Installation	Left open cases				Modify supermarket layout
Night Shield	NO night shield				Lower energy consumption
Defrost Control	NO defrost control				Lower energy consumption

2.4.2.5 Centralized system energy consumption

Table 2.39 Energy consumption evaluation for possible energy saving options.

DC description	*Baseline W/mL*	*Baseline W/m^3*	*LED*	*ECM*	*ASH Control*	*Defrost Control*	*Doors Installation*	*Combined Options*
VOPMT	905	464	7%	3%	0%	1%	44%	57%
SVOPMT	662	445	9%	5%	0%	1%	47%	65%
HOPMT	410	653	14%	7%	0%	2%	10%	34%
VOPLT	2435	1015	2%	1%	3%	0%	75%	80%
SVOPLT	1800	1071	3%	2%	4%	0%	73%	80%
HOPLT	801	598	7%	4%	1%	1%	56%	68%
VGDMT	411	194	14%	7%	9%	0%	0%	30%
SVGDMT	385	283	15%	8%	9%	0%	0%	29%
VGDLT	603	142	10%	5%	6%	0%	0%	18%
HGDLT	342	380	17%	9%	3%	0%	0%	26%

Table 2.40 Break down of a supermarket energy consumption due to centralized refrigeration system.

SUPERMARKET	*Centralized System Energy Consumption*						
DC description	Baseline	LED	ECM	ASH	DEF	Doors	Combined
DC Supermarket Consumption (kWh)	166	140	153	158	164	143	92
DC Energy Consumption reduction (%)	0%	15%	8%	5%	1%	14%	44%

Where
ASH: anti-sweat heat controller
DEF: defrost cycle control
Doors: add doors to open cases

ECM: higher efficiency evaporator fan motor
GD: glass door
H: horizontal
LED: higher efficiency lighting from light–emitting diodes
LT: low temperature
MT: medium temperature
V: vertical
OP: open
SV: semi-vertical

The thermal load of display cases found in a typical supermarket is calculated using the analytical model presented in Section 0. Results for the baseline are summarized in Table 2.40 in terms of heat load per meter of display case (second column), then per m^3 of refrigerated volume (third column). Table 2.40 summarizes the relative energy gains for each type of refrigerated display case, taking into account the improvement of each technical option, and then combining the options where it can be seen that the integration of all options is different of the sum of each. For example, an energy consumption reduction of 57% can be achieved for vertical open medium temperature case (VOPMT) when combining all options (Table 2.40).

For a baseline scenario, an hourly energy consumption of 166 kWh is calculated. Combining all of the presented options and taking into account the type of display cases defined for a typical Californian supermarket, can reduce the energy consumption of display cases by 44% (Table 2.40).

The effect of door installation is highlighted in Figure 2.12 where the contribution of energy saving option is plotted, for both vertical and horizontal open display cases, medium and low temperature equipment.

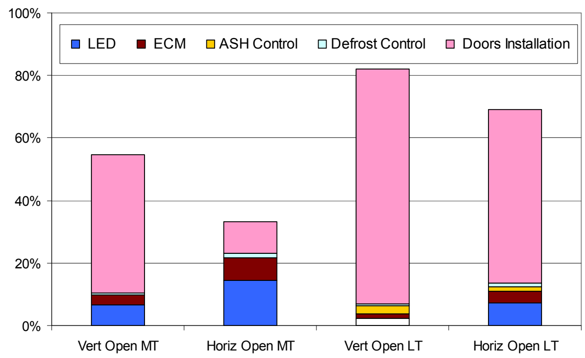


Figure 2.12 Energy saving option contribution to equipment energy consumption.

Figure 2.13 shows the energy saving breakdown for closed display cases. It appears that the heat load can be reduced by approximately 30% for these display cases. Most of this energy saving is due to LED lighting systems as one can see on Figure 2.13.

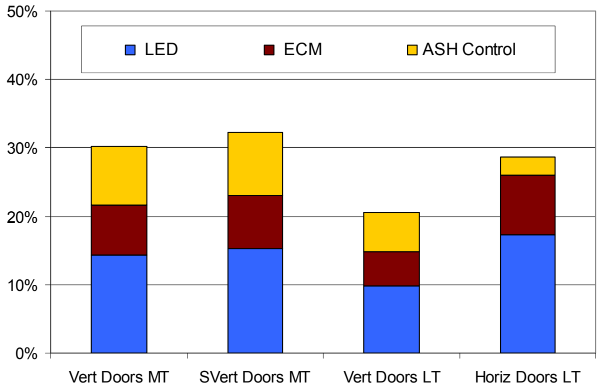


Figure 2.13 Energy savings breakdown for closed display cases.

2.4.2.6 Results of LCC analysis

The LCC analysis is conducted for a typical supermarket with a layout presented in Section 1.5. Assuming 10-year lifetime for all commercial refrigerating equipment of this layout, the LCC of a supermarket is obtained by summing the LCC of each refrigerating equipment.

After defining initial prices, costs, and possible energy savings for each commercial refrigerating equipment found in the layout, the equipment LCC and corresponding PBP are evaluated according to the LCC and PBP equations previously described in this section, and presented in Table 2.41.

Table 2.41 LCC and PBP of investigated technical options for a typical supermarket layout.

Commercial Refrigeration Equipment Options	*PBP (years)*	*LCC ($)*
Baseline	0	3,130,402
LED Lighting	0.3	2,923,013
ECM Fan Motors	0.5	3,039,767
ASH control	1.8	3,081,110
Installing Doors	3.2	3,010,195
Defrost Control	3.7	3,117,375

Once the options are evaluated separately, the LCC of aggregated options are estimated. For the aggregated models, technical options are successively implemented according to an increasing PBP. Figure 2.14 shows the LCC distribution for different tested aggregated models. It appears that applying LED lighting, ECM fan motors, and installing doors, controlling anti-sweat heater and defrost mechanisms are profitable since a decreasing tendency is observed and the lowest LCC is calculated.

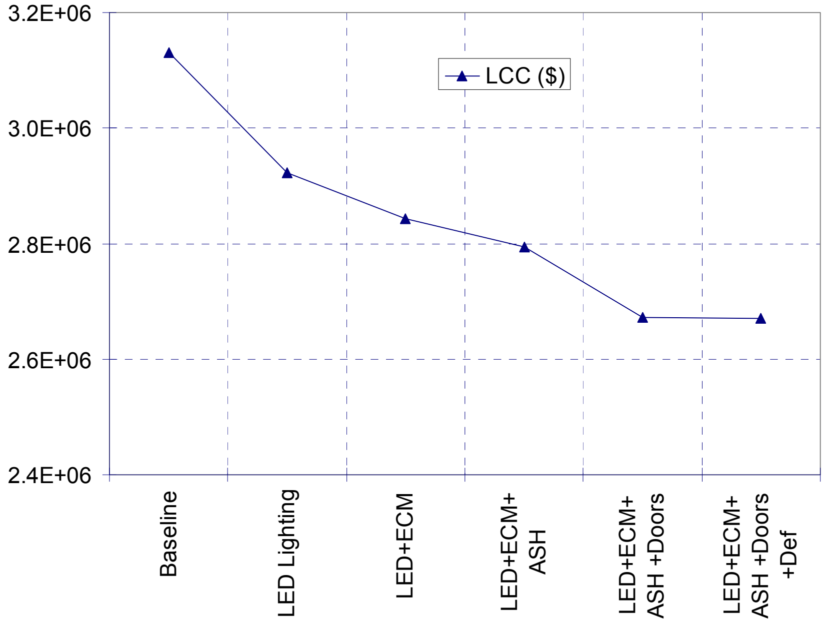


Figure 2.14 LCCA of a supermarket over a 10-year lifetime.

Applying all of the investigated energy saving options, the complete aggregated model allowed 17% savings on the LCC when compared to the baseline scenario.

2.4.3 LCCA of direct and indirect centralized systems

2.4.3.1 Description of technical options

A second LCC analysis is performed to define the most appropriate refrigerating system in terms of energy efficiency and costs. To begin with, a state of the art review is conducted in order to screen existing refrigeration systems and choose systems to be investigated in the LCCA.

Typical supermarket refrigeration systems consist of direct expansion air/refrigerant coils located in display cases and walk-in coolers. Compressors are located in a machine room, in a remote part of the refrigerated store, either in the back area or on the roof. Condensers are located either in the machinery room, or more likely, on the roof above the machinery room. Piping is connecting back and forth between the machinery room and the refrigerating equipment for refrigerant circulation either in liquid phase or in vapor phase (see Figure 2.16).

The difference between a secondary loop and a direct expansion refrigeration system is that the refrigeration of display cases and walk-in coolers is provided by a chilled secondary fluid called heat transfer fluid (HTF), pumped from a primary heat exchanger in the machinery room where the refrigerant evaporates and cools the HTF to the display cases (see Figure 2.19).

The most commonly used secondary fluid in both commercial and industrial applications is mono-propylene glycol (MPG) for medium temperature racks, but in the last 10 years a number of other products have been proposed based for example on acetate formiate. CO_2 is a promising heat transfer fluid for low temperature units. These two HTFs will be used thereafter to evaluate secondary loop performances and to carry out an LCC analysis.

Five refrigeration systems are investigated in the second LCC analysis. In addition to the multiplex direct expansion system and the distributed expansion system, three secondary loop refrigeration systems are described. The comparative study is conducted for a typical supermarket of 4,400 m^2 (47,360 ft^2).

The first step of the analysis consists of evaluating the heat load of a supermarket. The input power required by compressors for those load conditions is then calculated for each temperature bin (low and medium-temperature racks) and the number of operating compressors is inferred.

The typical Californian supermarket includes two low-temperature racks and two medium-temperature racks. Display cases and coolers are grouped and connected to compressors racks according to the required saturated suction temperature (SST) to maintain the desired case temperature. In the following analysis, low-temperature racks will operate at - 32°C (-25°F) SST whereas medium-temperature racks will operate at -10°C (+14°F) SST. Each compressor rack consists of three or four compressors sized to allow compressors, operating simultaneously, providing the cooling capacity that meets the design heat load.

For the 47,360 ft^2 supermarket, the refrigeration capacity is 190 kW at the medium-temperature level and about 150 kW at the low-temperature level. Assuming four compressors in each rack, the refrigerating capacity of a compressor operating at the medium-temperature level is 32 kW whereas 19-kW cooling capacity corresponds to compressors mounted on low-temperature racks.

The most common type of condensers used in supermarket refrigeration is air-cooled condensers. These condensers usually employ finned-coil construction with 8-10 fins/inch (300-400 fins/m) and multiple fans (Figure 2.15). Air-cooled condensers are known to operate reliably and require the least maintenance.

Figure 2.15 Roof-top air-cooled condenser.

Multiplex direct expansion refrigeration system

System description

The most common new direct system in supermarkets is the multiplex refrigeration system using R-404A as a working fluid. R-404A is a blend of hydrofluorocarbons that are non-ODS, but has a high-global warming potential of 3900 [IPC06]. It consists of multiple racks of compressors operating at the same saturated suction temperature with common suction and discharge refrigeration lines (one for medium temperature and another one for low-temperature racks) [BAX03a]. The term multiplex refers to the use of multiple compressors piped to a common suction and discharge manifolds, all installed on a skid containing all the necessary piping, control valves, and electrical wiring to control the compressors and the refrigeration provided to the display cases.

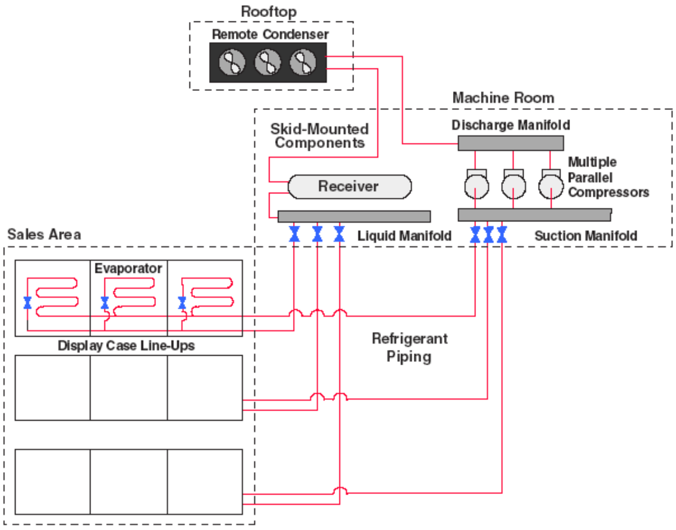


Figure 2.16 Piping diagram for the Medium Temperature Multiplex Refrigeration.

Figure 2.16 shows elements of a medium-temperature unit of a secondary loop refrigeration system. Compressor racks are installed in a machinery room with long refrigerant pipes connecting them to display cases in the sales area. The piping length can reach several hundred meters for large supermarkets, implying possible failures, fugitive emissions at joints, and pressure losses especially on the suction line. The hot gas discharge from the compressor is piped back to the remotely located condenser, which condenses gas to liquid. Liquid refrigerant is piped back to the compressor rack, where a receiver, liquid manifold, and associated control valves distribute liquid to the cases and walk-ins.

Multiplex refrigeration model

Figure 2.17 shows the diagram of the most commonly used compressor arrangement in a multiplex refrigeration system found in supermarkets. The refrigeration system configuration is described in order to evaluate its performances. Several parameters should be known beforehand such as display case evaporator temperature, minimum condenser temperature, condenser type as well as the refrigerant in use.

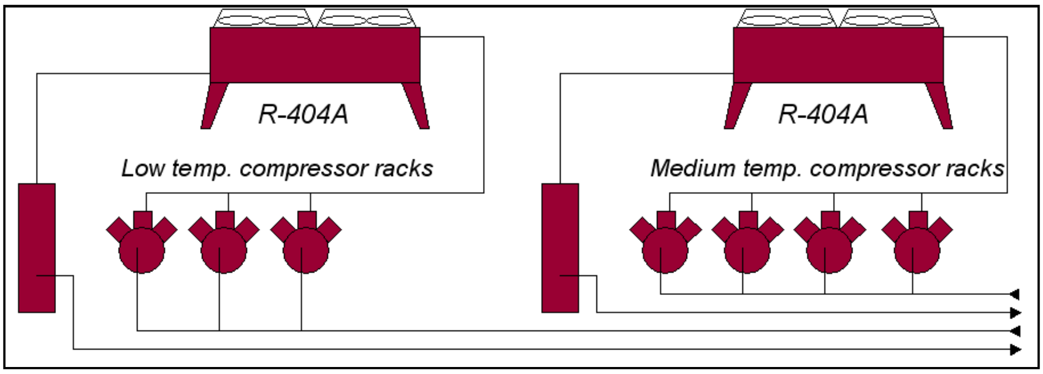


Figure 2.17 Design of a multiplex refrigeration system using R-404A.

Multiplex refrigeration state points

The refrigerant in both temperature units is R-404A. Several operating set points must be monitored for each multiplex compressor rack and for each compressor suction group.

- Saturated Suction Temperature (SST) - the saturation temperature corresponding to the refrigerant pressure measured at the compressor suction port.
- Saturated Discharge Temperature (SDT) - the saturation temperature of the refrigerant based upon the pressure measured at the compressor discharge.
- Return Gas Temperature - the refrigerant gas temperature measured at the compressor suction port.
- Refrigerant liquid temperature - the liquid temperature measured at the receiver outlet, and before and after sub-cooling heat exchangers, if installed.

Once the thermodynamic cycle points are defined, the cooling capacity and the compressor input power can be calculated based on compressor technical data supplied by manufacturers (Copeland, Carlyle, Danfoss, Bitzer…).

Operating conditions

The most significant parameter in determining condensing temperature is the temperature difference with the outdoor temperature, ΔT, since heat is rejected to the ambience. The condenser ΔT is dependent on the condenser type: for air-cooled condensers considered in this study, 8 K and 10 K are standard values of ΔT for low and medium temperatures respectively.

The fan power for remote condensers depends on the condenser type. Air-cooled condensers for low-temperature refrigeration are normally sized for a smaller temperature difference ΔT and require more fan power than condensers used with medium-temperature refrigeration [FOS04] [ORL01].

Pressure drop will occur in the suction lines (between the display-case evaporator and the compressor suction point). This pressure drop is taken into account by a lower saturated temperature value at the compressor suction port. Heat gain to the return gas will also take place and affect the refrigerant mass flow rate transferred by the compressors. Pressure drops and superheat vary depending on the distance between the display cases and the compressor racks. These factors tend to decrease the capacity of the compressor and increase the run time needed to meet the heat load. In this study, 10 K (18°F) superheat and 0.3 bar (4 psi abs) pressure drop are assumed at the compressor suction ports.

The temperature of the refrigerant in liquid phase is an important operating state point. This temperature is usually lower than the condensing temperature because the store is air-conditioned and so there is a "free" cooling of the refrigerant in the liquid lines. This "free" sub-cooling varies between 10 and 15 K. For instance, a refrigerant liquid temperature of 30°C corresponds to a condensing temperature of 45°C.

When complementary sub-cooling is realized due to refrigeration processes whose only purpose is to cool the refrigerant in its liquid phase, the liquid refrigerant leaving the sub-cooler is typically 5°C (41°F). This mechanical sub-cooling is not free and its energy consumption in taken into account in the energy consumption calculation. Nevertheless in the present analysis, no mechanical sub-cooling is taken into account. Consequently, the liquid refrigerant temperature varies from 20 to 30°C according to condensing temperature related to climatic conditions.

Distributed system with separate roof condenser

Another option for direct expansion systems is the distributed system with a separate rooftop condenser. Distributed systems may have different designs but the main concept is to install compressors in sound-proof boxes near the display cases, the condensing heat being released on a water circuit connected to dry-air coolers (as shown on Figure 2.18).

The sound-proof boxes are located within the store to provide refrigeration to a particular series of display cases, such as meat, dairy, frozen food, etc. With this arrangement, the pressure losses as well as superheat are reduced. The refrigerant suction and liquid lines are shortened in a distributed system and refrigerant charge requirement for the distributed system is reduced compared to a multiplex refrigeration system. The total refrigerant charge will be about 75% of a direct expansion multiplex system [CAG04].

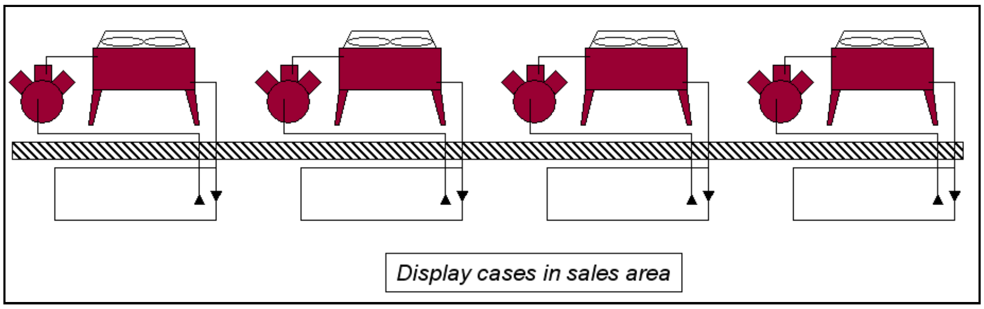


Figure 2.18 Description of the distributed refrigeration system.

Secondary Loop Refrigeration

The secondary loop consists of a HTF pumped between a central chiller and display cases. At least two fluid loops are installed in a supermarket depending on the heat load composition and temperature levels. Refrigeration with secondary loop systems has been introduced in supermarkets to decrease the refrigerant charge and to minimize potential refrigerant leakage. Secondary loop systems may have various designs with different energy efficiencies.

In the U.S., the most commonly used HTF is still MPG in both commercial and industrial refrigeration. MPG is preferred because it is inert to common piping materials and most non-metallic gaskets and seals. Nevertheless, MPG is used only for medium-temperature units: at concentrations needed for low-temperature refrigeration, its very high viscosity induces high pumping power. Consequently, other fluids, such as CO_2 are used for low-temperature units.

Secondary loop system description

Flow rates of single-phase HTF are defined by the HTF temperature change in the whole cooling circuit, typically of 4 or 5 K. Because of the high viscosity and the necessary HTF mass flow rate to provide cooling, the energy associated with pumping is substantial. But the picture is significantly different with CO_2, which is now used as a phase-change HTF: CO_2 evaporates partially in the display-case heat exchangers (typically 20% of it), so CO_2 returns at the primary evaporator where the 20% is condensed.

In a secondary loop refrigeration system, the refrigerant charge is approximately 50% lower than that of a direct expansion and consequently minimizes the pressure drop as well as the superheat on the refrigerant side. Those factors allow reducing the compressor energy consumption, but increase the HTF pumping power. Figure 2.19 illustrates the piping diagram of medium-temperature unit of a secondary loop refrigeration system.

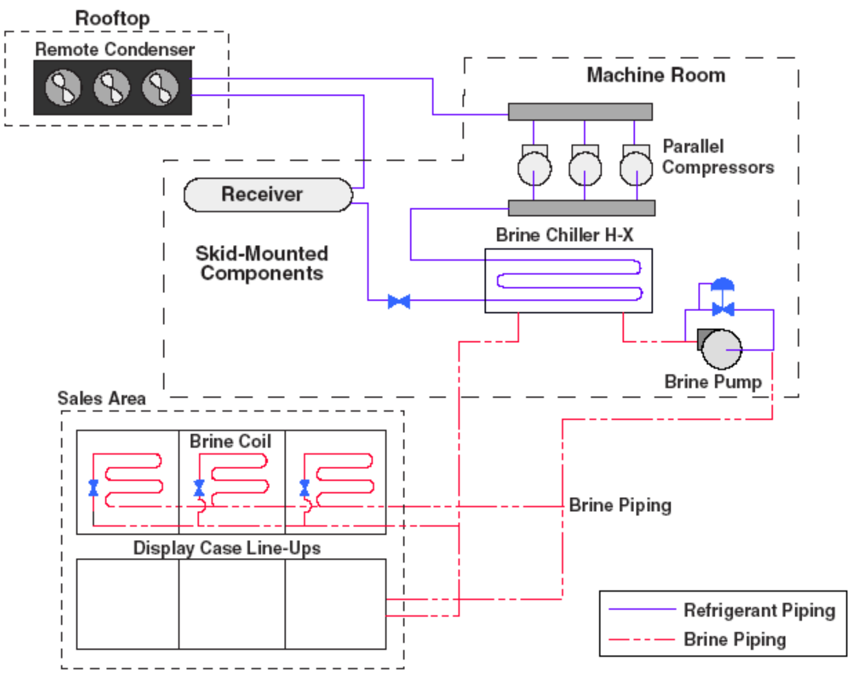


Figure 2.19 Elements of a secondary loop refrigeration for medium-temperature racks.

Note: on Figure 2.19 brine has to be understood as HTF.

Modeling of the Secondary Loop Refrigeration

The major difference in the analysis of a secondary loop refrigeration system is the operation of the secondary loop. The HTF mass flow rate (MFR) is calculated by applying Equation (2.21) when MPG is used, whereas Equation (2.22) is applied for secondary loops using vapor-liquid CO_2.

$$\dot{M}_{SF} = \frac{Q_0}{C_{SF}\Delta T_{SF}} \tag{2.21}$$

$$\dot{M}_{SF} = \frac{Q_0}{\Delta h_{SF}} \tag{2.22}$$

Where

Q_0 Heat load delivered by the secondary fluid loop (kW)

$\dot{M}_{SF}$ HTF mass flow rate (kg/s)

ΔT_{SF} HTF temperature change in the display cases circuit

Δh_{SF} HTF Enthalpy change in the display case circuit (kJ/kg).

In the secondary loop, HTF recovers heat not only from the display cases but also along the piping due to thermal losses. A complementary heat gain is due to the operation of the secondary fluid loop pump. The power required by the pump is calculated based on HTF thermo-physical properties at the temperature level, piping length and diameter, pressure drops on all the circuit, and refrigerating capacity.

The pump input power is calculated by Equation (2.23) assuming its overall efficiency η_p is 55%.

$$W_p = \frac{\dot{M}_{SF}}{\rho_{SF}} H_m \frac{g}{\eta_p} \tag{2.23}$$

where H_m refers to required pump head expressed in Water Column (WC). The subscript p stands for pump.

Thermodynamic cycle points of the refrigerating system are determined. The evaporating temperature at the primary evaporator is set at 5 K below the HTF exit temperature. The refrigerant superheat is set at 10 K.

For air-cooled condensers considered in this study, the temperature difference between the condensing temperature and the outdoor temperature, ΔT, is 8 K and 10 K for low and medium-temperatures respectively.

Pressure drop of the refrigerant vapor to the compressor suction is lower than pressure drop in multiplex refrigeration system due to the close coupling of the compressors and heat exchangers. This pressure drop is set at 0.2 bar (2.9 psi abs.).

Investigated secondary loop refrigeration systems

Secondary loop design with completely indirect refrigeration system

CO_2 is used as a HTF in the low-temperature system whereas MPG is used for medium-temperature system. The MPG temperature at the exit of the primary evaporator is -8°C and the return temperature -5°C.

For the MPG loop, the required pump head is set at 23 m WC (Water Column). For CO_2, the required pump head is set at 15 m WC [FOS04] [ORN01].

Figure 2.20 shows a diagram of the 2 secondary loops and the associated refrigeration systems for medium and low-temperatures.

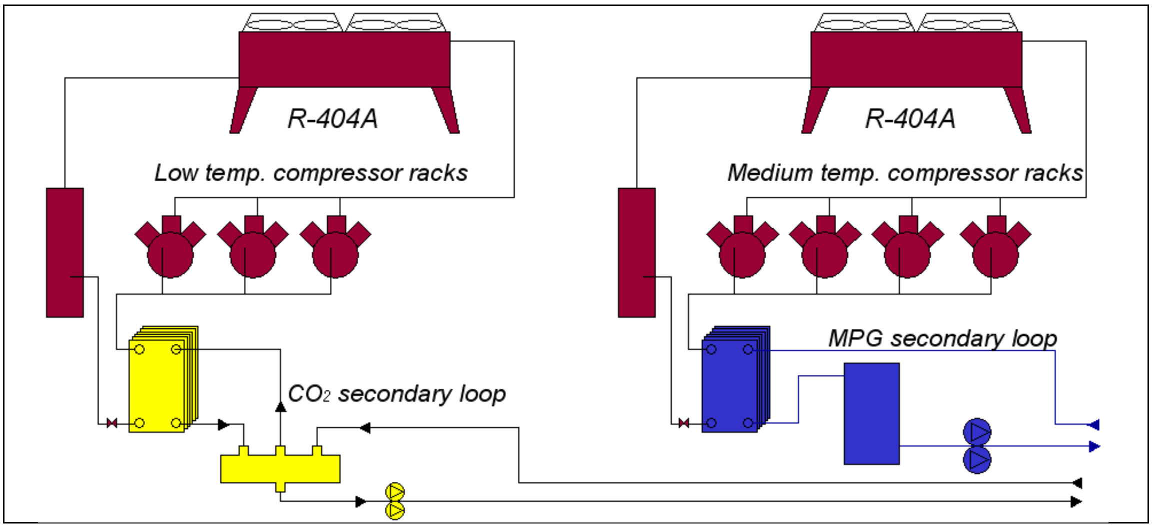


Figure 2.20 Secondary loop refrigeration system using MPG and CO_2 for medium and low-temperature racks respectively.

Indirect system with CO_2 as the only refrigerant

Another secondary loop system configuration is studied with CO_2 as the only HTF for both low and medium-temperature levels (Figure 2.21). CO_2 as a two-phase HTF is not currently used for medium-temperature systems. However, some early prototypes exist and such a solution might be developed in the near future.

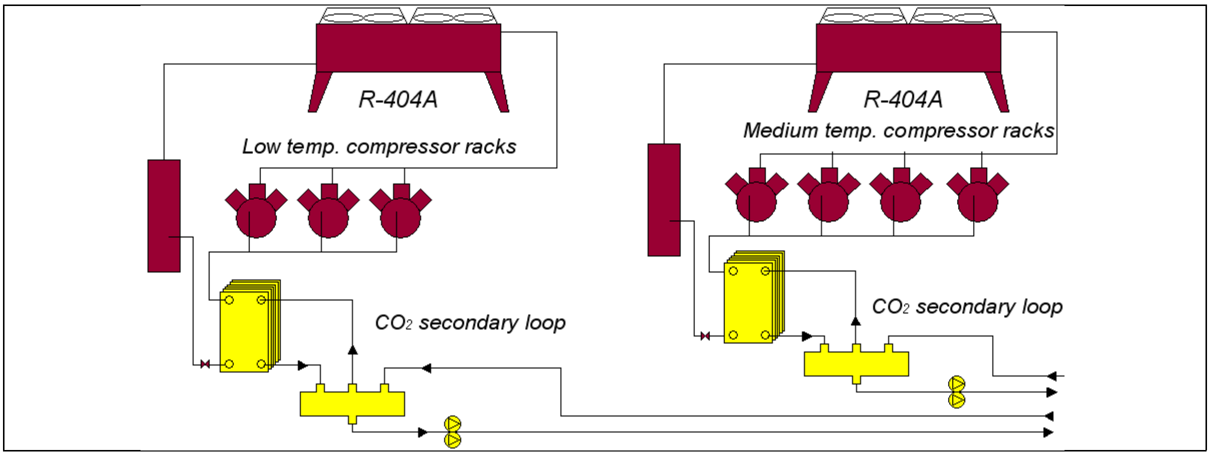


Figure 2.21 Secondary loop refrigeration system using CO_2 as the only refrigerant.

Cascade system with CO_2

The cascade system with CO_2 in the low-temperature system and a secondary loop using MPG for the medium-temperature system is an interesting solution that has been tested in several supermarkets and showed promising results [CHR99]. The HTF at the medium temperature level has a delivery temperature of about -8°C and a return temperature of -5°C.

The delivery temperature of the CO_2 in the low-temperature unit is about -32°C. Figure 2.22 illustrates the outline of a cascade system with CO_2 as secondary fluid for low-temperature unit.

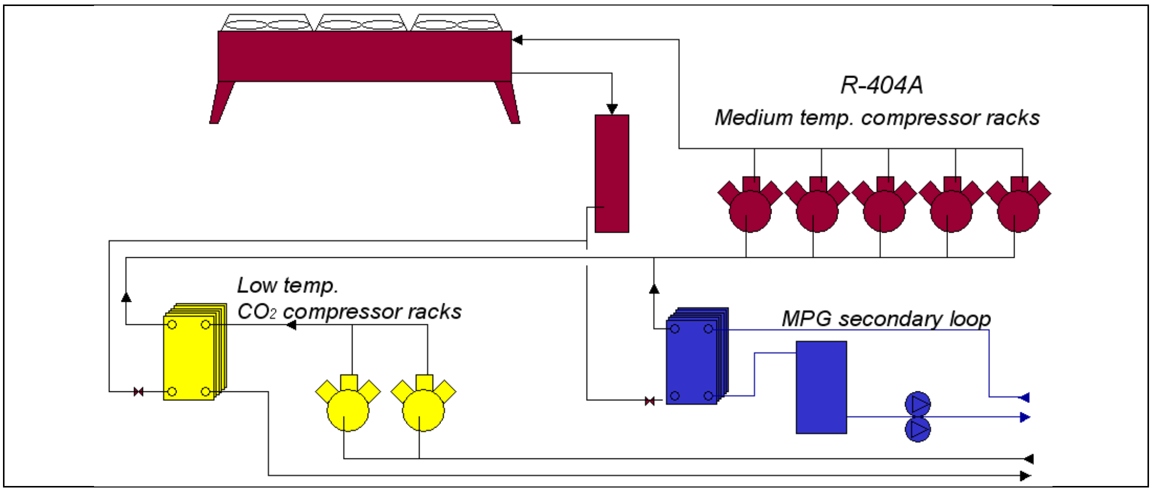


Figure 2.22 Cascade system with CO_2 in the low-temperature stage.

Table 2.42 summarizes for multiplex refrigeration, refrigerant thermodynamic cycle points, operating conditions, chosen compressors, cooling capacity, and energy consumption according to the temperature level and the refrigerant. Table 2.43 collects identical information for secondary loop systems investigated in this study, as well as the secondary fluid outgoing and return conditions at the primary heat exchanger.

Knowing operating state points and the required compressor input power, compressors for low and medium-temperature racks are chosen from manufacturers catalogs. All compressor models found in Table 2.42 and Table 2.43 are Copeland technologies except for the CO_2 cascade system compressor, which is a Bitzer technology.

Table 2.42 Multiplex refrigeration system operating conditions for two refrigerants R-22 and R-404A.

	Refrig.	T level	SST (°C)	ΔT_0 (K)	SDT (°C)	ΔT_k (K)	SH (K)	ΔP Suction line (K)	T Return Gas (°C)	T Liquid (°C)	ΔP Discharge line (K)	T Suction Comp (°C)	Comp. Model	Cooling Capacity (kW)	Comp. Power input (kW)	COP	Fan Power (kW)	COP_{system}
Multiplex	R22	Med	-10	15	30-45	10	10	3	7	20-30	0.5	-13	D4DA-200X	32.7	12.7	2.57	0.635	2.45
		Low	-32	15	30-45	8	10	5	-17	20-30	0.5	-37	D6TH-2000 SUB	19.25	15.5	1.24	0.775	1.18
	R404A	Med	-10	15	30-45	10	10	2	8	20-30	0.5	-12	D3DS-100X	32.5	13.7	2.37	0.685	2.26
		Low	-32	15	30-45	8	10	4	-16	20-30	0.5	-36	D6TH-270X	19.3	15.9	1.21	0.795	1.16

Table 2.43 Secondary loop refrigeration systems operating conditions with R-404A as a refrigerant.

	System Description	Refrig.	Sec. fluid	T level	SST (°C)	ΔT_0 (K)	SDT (°C)	ΔT_0 (K)	SH (K)	ΔP Suction line (K)	T Return Gas (°C)	T Liquid (°C)	ΔP Discharge line (K)	T suction Comp (°C)	T sec fluid (°C)	Comp. Type	Cooling Capacity (kW)	COP	Power Input (kW) Comp.	Power Input (kW) Pump	Power Input (kW) Fan	COP_{system}
Secondary loop	SL indirect MPG-CO2	R404A	MPG	Med	-13	15	30-45	10	10	3	4	20-30	0.5	-16	(-5 -8)	D3DS-150X	30.3	2.28	13.3	1.09	0.665	2.01
		R404A	CO2	Low	-33	15	30-45	8	10	3	-16	20-30	0.5	-36	-28	D6DH-350X	20.7	1.28	16.1	0.09	0.805	1.22
	SL indirect CO2-CO2	R404A	CO2	Med	-13	15	30-45	10	10	3	4	20-30	0.5	-16	-7	D3DS-150X	30.3	2.28	13.3	0.12	0.665	2.15
		R404A	CO2	Low	-33	15	30-45	8	10	3	-16	20-30	0.5	-36	-28	D6DH-350X	20.7	1.28	16.1	0.09	0.805	1.22
	Cascade CO2	R404A	MPG	Med	-13	15	30-45	10	10	2	5	20-30	0	-15	(-5 -8)	D6DL270	55.1	2.04	27	1.09	1.35	1.66
		CO2	-	Low	-32	15	-13	8	10	3	-15	20-30	0.5	-35	-	KP-120-2	19	5	3.8			

2.4.3.2 Life cycle cost assessment (LCCA) of screened refrigeration systems

The LCC of the five refrigerating systems described in previous sections are evaluated based on available data in the literature and lessons learned from the field. The methodology used is the same applied previously to define the optimal aggregated technical options, where the total installed cost and operating cost are calculated for the typical Californian supermarket.

Refrigerating system components

Major components of a refrigeration system are: compressors, condensers, evaporators, piping, display cases, and miscellaneous electronics including frequency converter, special fault interruption (FI) relay AC/DC, construction of safety circuit, compressor control, and oil control with program logic circuit (PLC).

Therefore, the total installed cost and the operating costs of each of these components will be investigated separately, and then summed together to obtain the refrigerating system installed cost and operating cost.

Lifetime and reliability

According to A.D.Little [LIT06], system compressors of supermarkets have a 10-year expected lifetime. Moreover, the typical lifetime of an air-cooled condenser is at most 10 years. Refrigerated display cases are usually replaced for merchandizing reasons prior to the end of their life and replacement occurs between 5 and 15 years, depending on the store policy. Therefore, the systems are expected to operate reliably for 10 years if properly installed and maintained [LIT96]. For the LCC estimation, the lifetime of 10 years is assumed for different refrigeration system components.

Conventional direct expansion refrigeration system

The multiplex system with air-cooled condensing is considered the baseline, since it is the most commonly installed configuration now used in supermarkets.

Total installed Cost

The total installed cost of a refrigeration system for a typical supermarket varies between 1 million and 1.1 million dollars [LIT96]. Table 2.44 shows the installed cost breakdown based on personal communication with supermarket industry representatives held by A.D.Little.

Table 2.44 Installed cost breakdown for a typical supermarket refrigeration system.

REFRIGERATION SYSTEM COMPONENTS	*INSTALLED COST SHARE*
Compressors	18.0%
Walk-in evaporators	3.0%
Condensers	5.0%
Miscellaneous electronics	6.0%
Piping	2.5%
Display cases	56.5%
Walk- in	9.0%

Based on the A.D. Little report on commercial refrigeration systems, the installed cost of system components are evaluated based on the breakdown of installed cost presented in Table 2.44.

Moreover, in the final report of IEA Annex 26, UK provided the cost of pipe work including installation and insulation costs: $50,000 for direct and $80,000 for secondary loop systems [UK03].

Table 2.45 Typical supermarket refrigeration system cost and energy consumption breakdown.

DX **Components**	*Component Installed Cost ($)*	*Component price ($)*	*Component Installation cost ($)*	*Energy consumption (kWh/year) -*
Compressors	195,510	147,000	48,510	860,000
Evaporators	47,460	21,000	26,460	0
Condensers	64,050	42,000	22,050	99,300
Miscellaneous electronics	92,190	52,500	39,690	0
Piping	67,305	21,000	46,305	0
Display Cases	505,575	472,500	33,075	477,800
Walk-in	77,910	73,500	4,410	133,800
Total	1,050,000	829,500	220,500	1,570,900

Maintenance Cost

Costs for refrigeration system maintenance are roughly 0.25% of supermarket revenues. The maintenance cost for a multiplex refrigeration system is about $75 per 100 sq.ft. of store sales area, which gives a maintenance cost of approximately $20,000 for a typical supermarket of 27,000 sq ft [LIT96].

Investigated refrigeration systems

Investigations are based on literature reviews, expert opinions, and personal communication with supermarket industry representatives. These investigations considered the total cost of the system including cases, piping, refrigerant, brine, and labor in addition to the compressor rack or primary chiller with the exception of the condenser sub-system. It was considered identical for all of the refrigeration systems, hence no cost premium is incurred when comparing to the baseline refrigeration system.

Distributed systems

Predicted energy consumption savings for a distributed system compared to a conventional multiplex refrigeration system is 12% [BAX03a]. The estimated installed cost premiums for distributed are presented in [ORL01]. Estimates are based on actual construction budgets supplied by engineering departments of visited supermarket chains. The distributed system shows higher equipment cost when compared to conventional direct expansion system with an incremental equipment price of $53,000. However, only a small increase in installation cost is observed and an incremental cost of $7,000 is estimated. This can be attributed to reduced refrigeration piping cost, but also increased electrical and fluid loop costs.

Secondary loop systems

In secondary loop systems, incrementally higher costs would be incurred in several areas: additional hardware costs of the secondary loop fluid circulation pumps, fluids reservoirs, the secondary fluid itself, and the refrigerant evaporator to chill the secondary fluid. The incremental cost is estimated to $50,000 for a typical supermarket. Different areas primarily influence the total additional charge [CHR99]:

1. The electrical board, especially influenced by the frequency converter and compressor control, and the safety circuit construction due to flammability. A.D. Little estimated an

additional cost of $10,000 to account for alarms and emergency ventilation of the mechanical equipment room [LIT02].
2. The assembly and the construction of the refrigeration system.

In the J. Arias thesis, the investment cost of the direct system is assumed to be 10% cheaper than that of an indirect system according to Bjerkhög, who is responsible for the implementation of a new refrigeration system in the supermarket chain COOP Sweden [ARI05]. On the other hand, based on interviews with supermarket industry professionals, D.H. Walker [ORL01] estimated a cost premium associated with using a secondary loop system, because of higher equipment and installation costs. The incremental refrigeration equipment system price is approximately $70,000; and the incremental installation cost is an additional $77,000. Hence a total cost premium of $147,000 is estimated.

As reported in discussions at the Annex 26 workshop [BAX03b], installation cost premiums for secondary loop approaches (using R-404A or R-507A as primary refrigerant and propylene glycol or potassium formate HTF for secondary loops) were about 15% for typical US markets. It was also noted that maintenance costs for the secondary system should be less.

The Danish country report (Volume 2) [DEN03] compares installation costs and operating efficiencies for a cascade system and R-404A DX systems. A test system installed in a small store (30-kW cooling capacity) was estimated to cost about 20% more than a traditional DX system and to have about the same energy efficiency. With more experience for installers the premium is estimated to drop under 15%. For larger systems, the premium would drop to 10%.

The British country report (Volume 2) announced an increase in overall energy consumption of 30% with the secondary loop refrigeration system. The energy use includes compressor power, condenser fan power, pumping power, and defrost energy. Most of this increase is attributed to pumping energy [UK03]. As for capital costs, the analysis showed the secondary loop system to be approximately 28% more expensive than a conventional direct system. This was confirmed by the UK experience of increased costs, between 15% and 30% for secondary systems.

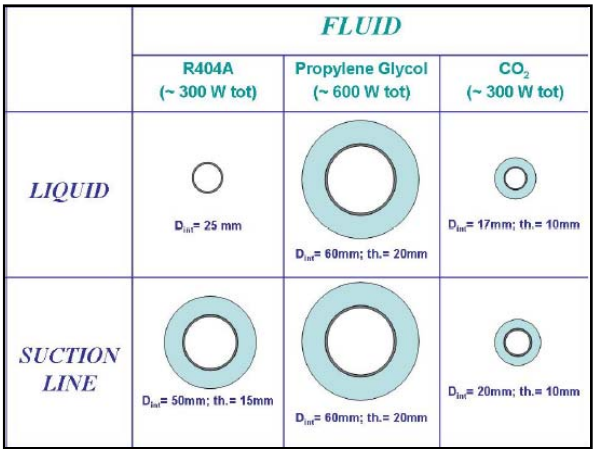


Figure 2.23 Comparison of liquid and suction pipes for different refrigerating fluids.

Moreover, for a plant originally constructed with a CO_2 indirect system, smaller pipes could be used for the return and liquid lines, which would compensate for the cost of the additional equipment in the secondary loop and needed safety devices [GIR03]. Figure 2.23 shows the different sizes and insulations of suction and liquid pipelines for different working fluids.

[CHR99] showed that for a cascade system using CO_2 in the low temperature system and a secondary loop using MPG for the medium temperature system, the energy consumption of a secondary loop system decreased by about 5% compared to a conventional supermarket while the investment was 20% higher.

Based on these data and on expert opinions, energy savings and cost premiums for secondary loop and distributed systems are estimated. Table 2.46 through Table 2.49 illustrate the total installed, equipment price, installation cost as well as energy consumption for each of the 5 investigated refrigeration systems.

In the following tables:

DX:	conventional direct expansion system
DIST:	distributed system with separate rooftop condenser
SL MPG+ SL CO_2-:	secondary loop system with MPG for medium-temperature system and CO_2 for the low-temperature system
SL CO_2 + - :	secondary loop systems with CO_2 as the only refrigerant for both low and medium-temperature systems
Cascade CO_2- MPG+ :	cascade system with CO_2 in low temperature system and secondary loop with MPG for medium temperature system.

Table 2.46 Total installed cost for the 5 refrigeration system components.

Refrigeration system Components	*DX*	*DIST*	*SL MPG+ SL CO_2-*	*SL CO_2 + -*	*CASCADE CO_2- MPG+*
Compressors	195,510	254,205	269,115	269,115	235,410
Evaporators	47,460	51,345	54,075	54,075	54,075
Condensers	64,050	92,820	64,050	64,050	64,050
Miscellaneous electronics	92,190	127,050	115,395	115,395	115,395
Pipelines	67,305	39,165	118,650	94,920	110,355
Display Cases	505,575	505,890	505,575	505,575	505,575
Walk-in	77,910	79,065	77,910	77,910	77,910
Total Installed Cost	1,050,000	1,149,540	1,204,770	1,181,040	1,162,770
Savings% Conventional DX	0%	9%	15%	12%	11%

*All values are given in $

Table 2.47 Component prices for the 5 refrigeration systems.

Refrigeration system Components	*DX*	*DIST*	*SL MPG+ SL CO_2-*	*SL CO_2 + -*	*CASCADE CO_2- MPG+*
Compressors	147,000	199,080	207,375	207,375	182,490
Evaporators	21,000	24,885	21,000	21,000	21,000
Condensers	42,000	66,360	42,000	42,000	42,000
Misc electronics	52,500	82,950	58,065	58,065	58,065
Pipelines	21,000	8,295	41,475	33,180	33,180
Display Cases	472,500	472,815	472,500	472,500	472,500
Walk-in	73,500	74,655	73,500	73,500	73,500
Total equipment cost	829,500	929,040	915,915	907,620	882,735
Increase% Conventional DX	0%	12%	10%	9%	6%

*All values are given in $

Table 2.48 Component installation costs for the 5 refrigeration systems.

Refrigeration system Components	*DX*	*DIST*	*SL MPG+ SL $_{CO2}$-*	*SL CO_2 + -*	*CASCADE CO_2- MPG+*
Compressors	48,510	55,125	61,740	61,740	52,920
Evaporators	26,460	26,460	33,075	33,075	33,075
Condensers	22,050	26,460	22,050	22,050	22,050
Misc electronics	39,690	44,100	57,330	57,330	57,330
Pipelines	46,305	30,870	77,175	61,740	77,175
Display Cases	33,075	33,075	33,075	33,075	33,075
Walk-in	4,410	4,410	4,410	4,410	4,410
Total installation cost	220,500	220,500	288,855	273,420	280,035
Increase% Conventional DX	0%	0%	31%	24%	27%

*All values are given in $

Table 2.49 Component energy consumption for the 5 refrigeration systems.

Refrigeration system Components	*DX*	*DIST (rooftop design)*	*SL MPG+ SL CO_2-*	*SL CO_2 + -*	*CASCADE CO_2- MPG+*
Compressors	*55%*	*50%*	*66%*	*60%*	*54%*
Evaporators	*0%*	*0%*	*0%*	*0%*	*0%*
Condensers	*6%*	*6%*	*6%*	*6%*	*6%*
Misc electronics	*0%*	*0%*	*0%*	*0%*	*0%*
Pipelines	*0%*	*0%*	*0%*	*0%*	*0%*
Display Cases	*30%*	*30%*	*30%*	*30%*	*30%*
Walk-in	*9%*	*9%*	*9%*	*9%*	*9%*
Total energy consumption (kWh/year)	*1,570,900*	*1,492,355*	*1,743,699*	*1,649,445*	*1,555,191*
Savings% Conventional DX	*0%*	*-5%*	*+11%*	*+5%*	*-1%*

LCC analysis results for refrigeration systems

Figure 2.24 illustrates results of LCC from simulations of direct and indirect systems. The period of study was 10 years, the interest rate was 4%, the annual price increase of electricity was 1%.

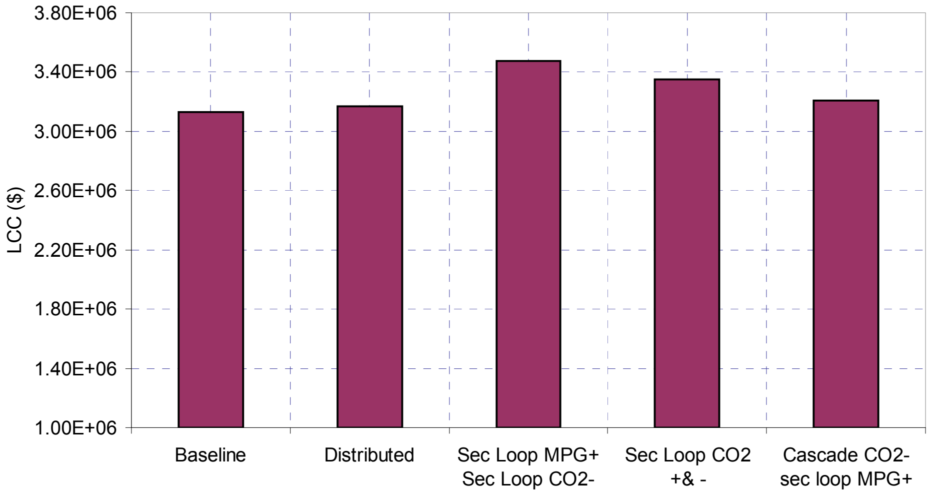


Figure 2.24 LCC distribution for investigated refrigerating systems.

Simulations results and capital costs analyses show that for indirect systems to become attractive alternatives to direct systems, design improvements need to be implemented to reduce both capital and running costs of secondary loop systems. More stringent legislation and incentives may also contribute to a wider application of secondary loop systems. This can be seen by comparing the LCC of secondary loop and distributed systems to the direct expansion system LCC (Figure 2.24). These results are identical to results proposed in the British report to IEA Annex 26 [UK03].

LCCP of indirect systems and distributed systems is essential to evaluate CO_2 emissions due to the refrigerating system operation (indirect emissions) and the direct emissions of refrigerants taking into account their GWP (global warming potential). LCCP results will underline the advantages of secondary loop systems compared to the baseline (current centralized direct expansion systems).

2.5 TEWI analysis of refrigeration systems

The basic concept of both the Total Equivalent Warming Impact (TEWI) and the Life Cycle Climate Performance (LCCP) is, for a given product or activity, to identify rigorously all of the warming impacts due to the product use through its lifetime. The two major contributors to emissions of greenhouse gases during the lifetime of refrigerating systems in supermarkets are the "indirect" effect of carbon dioxide emissions related to the energy consumption of the product during operation (indirect emissions) and the "direct" effect of greenhouse emissions from the product taking into account their GWP.

Therefore, the contribution to global warming (TEWI) of any refrigerating systems has to be evaluated taking into account both the energy consumption and the refrigerant emissions. Five technologies of refrigeration systems have been compared:

- the baseline is the direct expansion system
- distributed system
- secondary loop system (MPG for medium-temperature display cases, and CO_2 for low-temperature display cases)
- secondary loop system (CO_2 for both medium and low-temperature display cases)
- CO_2 cascade and secondary loop (MPG) at medium temperature.

The total equivalent Warming Impact is expressed in Equation (2.24):

$$TEWI = GWP \times \lfloor Ln + m(1-\alpha) \rfloor + nEB \tag{2.24}$$

with

GWP	Global Warming Potential (kg eq. CO_2)
L	Leakage rate (kg / year)
n	Operation life of the system (years)
m	Fluid quantity charged in the system (kg)
a	Recovery rate (kg of recovered fluid/initial charge)
E	Annual energy consumption (kWh / year)
b	CO_2 emission per electric kWh of produced power (kgCO_2 / kWh)
TEWI	Total Equivalent Warming Impact (kg of CO_2 produced during the lifespan of the equipment).

The direct contribution, due to refrigerant emissions during the system lifetime is expressed in Equation (2.25):

$$Dc = M \times \lfloor a + rc(1 - fre) + ne + s(1 - sre) \rfloor GWP \tag{2.25}$$

with:

Dc	Direct contribution (kg CO_2 eq.)
M	Nominal charge of refrigerant in the system (kg)
a	Initial charge emission rate (%)
rc	Remaining charge (%) at end of life, before decomissioning
fre	Recovery efficiency at decomissioning (%)
n	Operation life of the system (years)
e	Fugitive emission rate, including losses due to rupture, and annual maintenance (%)
s	Number of renewing operations asking for a complete refrigerant recovery (retrofit for example), except end of life recovery.
sre	Refrigerant recovery efficiency when renewing operation are conducted
GWP	Global Warming Impact (kg equivalent CO_2)

Large emissions due to tube or component ruptures have been considered in the fugitive emission rate, which is an average value for a wide number of installations.
Servicing and maintenance operations contribute to additional refrigerant emissions depending on the operation quality. This contribution is also included in the fugitive emission rate. Refrigerant losses occurred at the end of life of the system, after decommissioning, when recovery is not appropriately done. A recovery efficiency rate is defined.

Assumptions for direct emission calculations are as follows:

- lifetime of the supermarket is 30 years
- a complete maintenance operation, with refrigerant recovery is performed after 10 years (end of life of display cases)
- annual servicing, accidental ruptures, and fugitive emissions are presented under a single rate
- emission rate and recovery efficiency are presented with a lower and an upper threshold.

Table 2.50 presents assumptions for direct emission calculations of a typical supermarket with 4400 m^2 sales area.

Table 2.50 Assumptions for direct emission calculations.

Refrigeration system	*DX*	*Distributed*	*Sec. Loop MPG+ Sec. Loop CO_2-*	*Sec. Loop CO_2 + & -*	*Cascade CO_2- Sec. Loop MPG+*
R-404A charge (kg)	1370	600	400	400	210
CO_2 charge (kg)	0	0	700	1400	450
Fugitive emission rates					
Upper threshold	30%	25%	20%	20%	20%
Lower threshold	18%	15%	12%	12%	12%
Emission rate at initial charge			5%		
Recovery efficiency at end of life	Lower threshold*: 70%		Upper threshold*: 30%		

*70% is the higher value considered for recovery efficiency leading to “the lower threshold” calculation of emissions.

The TEWI analysis is calculated for a period of 10 years, corresponding to the refrigeration system lifetime before refurbishing. GWP of R-404A is 3900 (2006 IPCC assessment report, [IPC06]), CO_2 is the reference with a GWP of 1.

2.5.1 Assumptions for indirect emission calculations

The energy consumption is calculated for each refrigeration system. The methodology is presented in Sections 0 (energy consumption) and 2.4 (LCCA analysis). Table 2.51 summarizes the annual energy consumption of a typical supermarket in CA (LA climatic zone).

Table 2.51 Annual energy consumption per supermarket, for different refrigeration systems.

Refrigeration system	*DX*	*Distributed*	*Sec. Loop MPG+ Sec. Loop CO_2-*	*Sec. Loop CO_2 + & -*	*Cascade CO_2- Sec. Loop MPG+*
Total energy consumption (kWh/year)	1,570,90 0	1,493,140	1,743,699	1,648,660	1,553,620

In California, the CO_2 content of one kWh, is dependent on the energy mix in power generation. Energy Power Mix in California for year 2006 is presented in Table 2.52 based on the values of the California Energy Commission [CEC07].

Table 2.52 Energy Power Mix in California in 2006.

Energy type	***Mix***
Coal	28.60%
Large hydroelectric	30.50%
Natural gas	35.40%
Nuclear	0.40%
Eligible renewable	5.10%

Note: Eligible renewable consists of biomass and waste, geothermal, small hydroelectric, solar, and wind.

Power generation leads to different CO_2 factors, depending on the energy conversion process and the primary energy source. Table 2.53 gives the range of CO_2 content of one kWh produced, for different energy sources [GFE07].

*Carbon equivalent factor is converted in CO_2 equivalent factor by the ratio of molar masses ($MMco_2/MMc$)

Table 2.53 Emission conversion factors.

Primary energy	***Carbon equivalent g/kWh***	***CO_2 equivalent g/kWh***
Gas	100 to 130	367 to 477
Coal	200 to 280	733 to 1026
Hydroelectric	1	3.7
Nuclear	2	7.3
Wind power	2 to 10	7.3 to 36.7

gives low and high thresholds of the CO_2 factor, taking into account the power energy mix in California. Thresholds correspond to efficiency variation of power generators independently of primary energy source.

Table 2.54 Calculation of the energy power mix in California.

Mix (year 2006)		CO_2 **emission factor** (g CO_2/kWh) Low threshold	High threshold
Coal	28.60%	209.6	293.4
Large hydroelectric	30.50%	1.1	1.1
Natural gas	35.40%	129.9	168.9
Nuclear	0.40%	0.0	0.0
Eligible renewable	5.10%	0.4	1.9
Averaged CO_2 emission factor for year 2006		341.1	465.3

2.5.2 TEWI calculation

Results of the TEWI calculation for five refrigeration systems are given in Table 2.55.

Table 2.55 TEWI calculation

Refrigeration system	***DX***	***Distributed***	***Sec. Loop MPG+ Sec. Loop CO_2-***	***Sec. Loop CO_2 + & -***	***Cascade CO_2- Sec. Loop MPG+***
Refrigerant total emissions (metric tonnes)					
R-404A (lower threshold)	2.9	1.1	0.6	0.6	0.3
R-404A (upper threshold)	4.8	1.8	1.0	1.0	0.5
CO_2 (lower threshold)	0.0	0.0	1.2	2.3	0.7
CO_2 (upper threshold)	0.0	0.0	2.0	4.0	1.3
Direct CO2 equivalent emissions (Thousands metric tonnes)					
Lower threshold	11.2	4.2	2.4	2.4	1.2
Upper threshold	18.9	7.2	4.1	4.1	2.1
Total energy consumption (MWh)					
Lower threshold	*15 709*	*14 931*	*17 437*	*16 487*	*15 536*
Indirect CO2 equivalent emissions (Thousands metric tonnes)					
Lower threshold	*5.4*	*5.1*	*5.9*	*5.6*	*5.3*
Upper threshold	*7.3*	*6.9*	*8.1*	*7.7*	*7.2*
Total CO2 equivalent emissions (Thousands metric tonnes)					
Lower threshold	16.6	9.3	8.3	8.0	6.5
Upper threshold	26.2	14.1	12.2	11.7	9.4

DX system has a TEWI of 26,000 metric tonnes $CO_{2,}$ 2 to 3 times higher than secondary loop systems. GWP of R-404A (3900) is the highest of HFCs currently used, therefore direct emissions contribute to 72% of DX system TEWI. CO_2 cascade offers the best performances.

Secondary loop and distributed systems may lower the refrigerant charge by a factor of two to four. The lower refrigerant charge can directly decrease emissions in case of ruptures and at equipment end-of-life if a systematic and efficient refrigerant recovery policy is not applied. Indirect and distributed systems lead to significantly shorter refrigerant lines, and thereby limit the number of fittings and brazing. As a consequence, the leak tightness of the system is improved.

Figure 2.25 (lower threshold) and Figure 2.26 (upper threshold) illustrate the comparison between direct emissions and indirect emissions over a 10-year lifetime for different refrigeration systems found in typical Californian supermarkets.

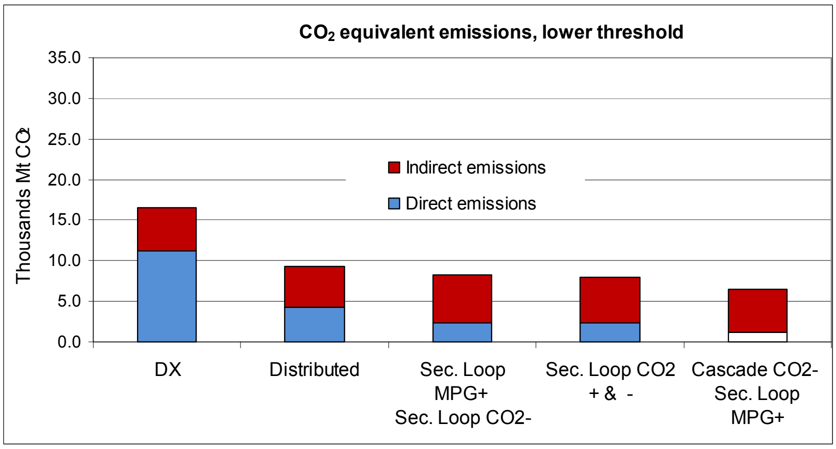


Figure 2.25 TEWI analysis, lower threshold.

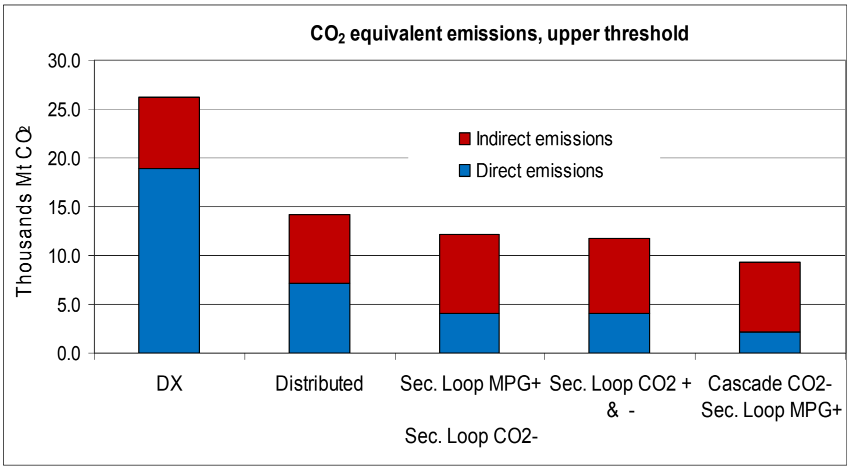


Figure 2.26 TEWI analysis, upper threshold.

2.5.3 Derivation to Californian state

Taking into account the number of supermarkets in California (~3400), TEWI is calculated at a California state level. Table 2.56 and Figure 2.27 give the equivalent CO_2 emissions and savings of alternative refrigeration systems, compared with the DX baseline.

Table 2.56 TEWI analysis in supermarkets at California state level.

California state	*DX*	*Distributed*	*Sec. Loop MPG+ Sec. Loop CO_2-*	*Sec. Loop CO_2 + & -*	*Cascade CO_2- Sec. Loop MPG+*
Total CO_2 equivalent emissions (Mega metric tonnes)					
Lower threshold	56.3	31.7	28.3	27.2	22.2
Upper threshold	89.2	48.1	41.4	39.9	31.9
CO2 emissions savings (Mega metric tonnes)					
Lower threshold	0.0	-24.6	-28.0	-29.1	-34.1
Upper threshold	0.0	-41.1	-47.7	-49.2	-57.3

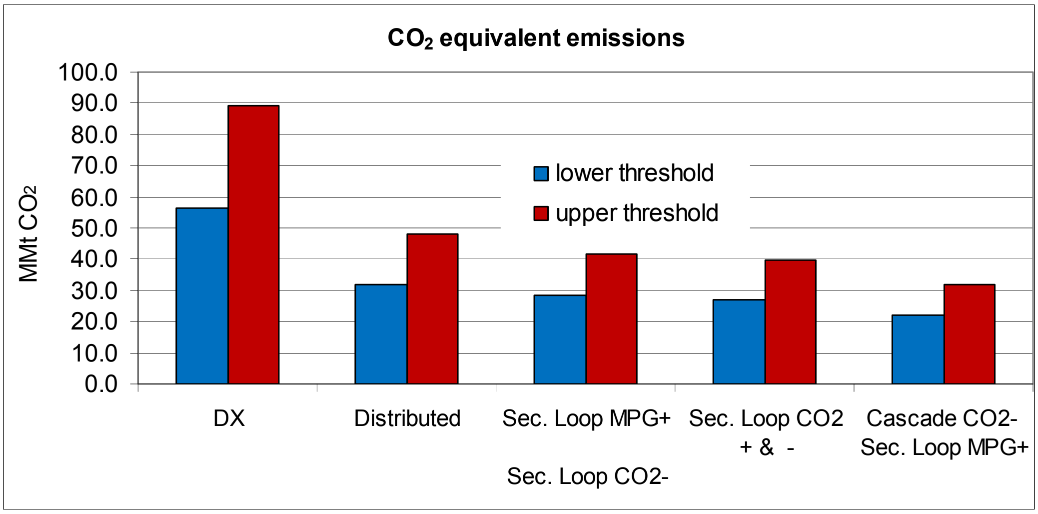


Figure 2.27 TEWI analysis in supermarkets at California state level.

2.5.4 Costs of CO_2 savings

The additional cost for different refrigeration systems has been calculated in the LCCA analysis (see section 2.4). Cost of CO_2 savings for different refrigeration systems are shown in Table 2.57.

Table 2.57 Costs of CO_2 savings.

Refrigeration system	*DX*	*Distributed*	*Sec. Loop MPG+ Sec. Loop CO_2-*	*Sec. Loop CO_2 + & -*	*Cascade CO_2- Sec. Loop MPG+*
LCCA per supermarket ($)	3,129,341	3,168,385	3,471,194	3,350,382	3,210,353
Additionnal cost ($)	0	39,044	341,853	221,040	81,012
CO_2 emission savings (tonnes)	0	-16,040	-18,148	-18,588	-21,145
Cost of 1 tonne CO_2 saved ($/metric tonne)		3.2	24.4	15.3	4.8

Combining technical options on display cases and refrigeration system using CO_2 cascade and a secondary loop, both direct and indirect CO_2 equivalent emissions are reduced. Technical options applied on display cases include: door installation, LED lighting, DC motor and floating head pressure.

Reduction of the cooling capacity has an impact on both energy consumption and refrigerant charge (and therefore an impact on refrigerant emissions as well).

Secondary loop system coupled with CO_2 cascade has a major impact on refrigerant direct emissions. CO_2 cascade system is more energy efficient than the baseline DX system. Savings for technical options applied to display cases, coupled with Cascade CO_2 / secondary loop systems are illustrated in Table 2.58.

Table 2.58 Savings for technical options applied to display cases, coupled with Cascade CO_2 / secondary loop systems.

	Per Supermarket		***California State***	
Refrigeration system	*DX*	*Cascade CO_2-Sec. Loop MPG+ DC technical options*	*DX*	*Cascade CO_2-Sec. Loop MPG+ DC technical options*
LCCA (Thousands $)	3,129	2,679	10,638,600	9,108,600
CO_2 emissions (Thousands metric tonnes)				
Lower threshold	17	5	56,295	15,565
Upper threshold	26	7	89,160	22,296
CO_2 emission savings (Thousands metric tonnes)				
Lower threshold		12		40,729
Upper threshold		20		66,865

The life cycle cost is 15% lower for a supermarket where technical options for energy savings are applied on both display cases and refrigeration systems. The cut in CO_2 equivalent emissions is 75 – 80% compared to the baseline DX system.

SUMMARY CONCLUSIONS AND RECOMMENDATIONS

This report has established the different banks and emissions of refrigerants by sector and by refrigerant type (CFCs, HCFCs, HFCs and Ammonia) based on inventories of stationary refrigeration systems of all sectors: domestic, commercial, industrial, and air conditioning. Nevertheless, refrigerant banks of refrigerated transports and mobile air conditioning have been evaluated in order to obtain the complete California refrigerant inventory. For the commercial sector, the number of refrigeration systems is based on Californian data and on the field survey performed by the laboratory. For food industry and mobile air conditioning, Californian data were available. For other sectors, Californian numbers have been derived from U.S. numbers based on population and wealth. The total refrigerant bank is evaluated at 116,000 metric tonnes including all refrigerant types. The dominant refrigerant is still HCFC-22 representing 57% of the bank and 50 % of emissions. The refrigeration commercial sector represents only 6% of the refrigerant bank and 10% of emissions due to the dominant refrigerant bank of stationary air conditioning (60% including chillers) and mobile air conditioning (23%).

The dominant issue for the near future is the replacement of R-22 by R-404A, which has a GWP twice the one of R-22, meaning that at equal emission factor the CO_2 equivalent impact of commercial refrigeration will also be doubled. This fact leads to the evaluation of technical options capable to reduce dramatically the refrigerant charge, such as the so-called indirect systems, where the refrigerant charge can be reduced by a factor of 4 to 8 times less. Indirect system lowers the refrigerant charge because the coldness is delivered to all display cases and cold room via a heat transfer fluid. A TEWI calculation has integrated the reduction of emissions due to the reduction of the refrigerant charge and the limited increase in energy consumption due to the pumping power for the heat transfer fluid circulation. Results indicate a reduction of a factor 3 of CO_2 emissions of centralized refrigeration systems.

An in-depth survey has been made for commercial refrigeration to define an average Californian supermarket, but also typical other commercial outlets using refrigeration equipment for food preservation. The survey has detailed the type, length, and energy consumption of all types of refrigerated display cases. A calculation method has been developed taking into the 28 types of equipment, their operating conditions, and the outdoor temperatures of the eight climatic zones of California. Three main technical options have been evaluated for energy savings in supermarkets: Installing glass doors on medium temperature display cases, new technologies for auxiliary components (efficient lighting, efficient fan motors, better control of defrosting devices), and better control of the condensing pressure of the refrigeration system. Results are first the energy consumption (calculated hour by hour) for all stand alone equipment, condensing units, and supermarket centralized systems. Annual energy consumption of refrigeration systems for the year 2004 is evaluated at 5,341 GWh for all Californian grocery supermarkets and at 20,228 GWh for all Californian commercial refrigeration. If the best technologies were to be fully implemented, the possible energy gains are of 30% referred to the current energy consumption of Californian supermarkets and 20% for the entire commercial sector. The payback periods vary from a quarter to less than 4 years.

Conclusions

This study establishes the refrigerant emissions of HCFCs and HFCS used in stationary and mobile refrigeration systems, including air conditioning, which is the dominant bank of stationary applications and also the dominant emission sector, even if the emission factors are much lower for chillers and air-to air AC systems compared to commercial refrigeration. HCFC-22 is still the dominant refrigerant in the total bank of refrigerants. R-404A refrigerant which is beginning to replace R-22 in commercial and industrial refrigeration implies a higher global warming effect if refrigerant emissions are left at the same level (the GWP of R-404A is more than twice that of R-22). Some commercial chains have made improvements in reducing refrigerant emissions from about 30% per year to about 15%, but those improvements need to be consolidated. Indirect systems using CO_2 as a heat transfer fluid is a technical option limiting drastically the refrigerant charge and changes significantly the future refrigerant emissions with a very limited energy additional consumption.

Several technical options are available to limit the energy consumption of commercial refrigeration systems. The main gains are related to the redesign of display cases by adding transparent doors for all their operating temperatures. The use of LED for lighting of products in display cases as well as the use of efficient technologies for fan motors and defrosting generates significant energy gains. For the entire commercial sector, energy gains of 20% are available with a payback period between 3 months and 4 years.

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GLOSSARY

Air enthalpy
Enthalpy is a thermodynamic property, a measure of the energy content or heat content of a system per unit mass. Air enthalpies changes according to temperature and moisture content, and are defined on psychometric charts.

Anti-sweat heaters
Many surfaces of display cases are at temperatures lower than the air dew point. In order to avoid fogging and also water droplets on the products in display cases, anti-sweat heaters are installed in the display cases in order to warm the surface above the dew point.

Article 5 Country, Non-Article 5 Country
Article 5 of the Montreal Protocol defines the countries where the consumption of CFC and HCFC are lower than 0.3 kg/ inhabitant meaning usually the developing countries.
Non Article 5 countries are developed countries with CFC and HCFC consumption higher than the threshold.

Auxiliary components
All electric components integrated in display cases: lighting, fans, anti-sweat heaters, electric resistance for defrosting, …

Bank (of refrigerant)
The sum of all refrigerants stored in refrigeration systems. Banks are grouped by refrigerant type: CFC, HFC, HCFC, and ammonia, and by application sector.

Bottom-up approach (for inventory or consumption estimates)
The method, defined in IPCC Guidelines, is opposed to Top-down method. The Bottom-up method describes refrigeration systems for each sector in order to evaluate the refrigerant banks. Top-down method is based only on the overall quantities of refrigerant sold by refrigerant manufacturers.

Cascade system
A refrigeration system architecture used for low-temperature application where a first refrigerant adapted to the low temperature is used in the low temperature system releasing the heat to a second high temperature refrigeration system using another refrigerant.

Centralized system
Refrigeration system where several racks of compressors are located in the machinery room, connected on one side to many evaporators installed in display cases in the sales area and to the other side to air condenser usually installed on the roof of the machinery room.

Charge (of refrigerant)
The quantity of refrigerant charged (added) initially in the refrigeration system and necessary to its operation. The charge of refrigerant is generally measured in pounds or kilograms, and depends on the refrigeration capacity of the system and on its design.

Chiller
A refrigeration system that cools water on one side and rejects heat usually also on a water cooled condenser. Chillers are usually large systems with refrigeration capacity above 1 MW. Many of them use centrifugal compressors. Chillers are often used to air condition large buildings.

Coefficient of performance
For the refrigeration system, it is the ratio of the cooling capacity to the compressor input power.

Compressor rack or unit
For commercial refrigeration system, the usual design of the machinery room is to have several racks of compressors (a rack including 3 to 5 identical compressors). The split of compressor power in several compressors allows better control of the refrigeration capacity.

Condensing unit
A part of a small refrigeration system with a refrigerating capacity between 1 and 10 kW, comprising an air condenser and one or two compressors installed on the same structure.

Critical pressure and temperature
are thermodynamic fundamental properties. Under the critical pressure, the refrigerant will undergo a phase change (liquid to vapor and vice versa); above critical temperature and pressure, the refrigerant will be in dense gas phase only.

Direct expansion system
A refrigeration system where the refrigerant is directly expanded in evaporators through an expansion valve. Direct systems can be of various sizes, from domestic refrigerators to large centralized commercial refrigeration systems.

Distributed system
Is a direct expansion system opposed to centralized commercial refrigeration system because smaller systems are disseminated near the display cases. The compressor and the condenser are installed in sound-proof boxes.

Entrapped air
The quantity of air infiltrated from the store inside the refrigerated display cases.

Evaporator
Evaporators are heat exchangers where the refrigerant evaporates usually in tubes or in plates. On the other side of the tube or he plate, air or liquid circulates and the refrigerant absorbs heat from them.

Floating head pressure
The current control of expansion valve used in commercial refrigeration system is to maintain a minimum level of condensing pressure (head pressure) in order to control easily the refrigerant mass flow rate. The drawback of this control is to maintain high condensing pressure even if the outdoor temperature is low. Floating head pressure changes this control by changing the expansion valve in order to accept much lower head pressure, which leads to significant energy savings.

Heat dissipation load
The fraction of heat dissipated by lightings installed in refrigerated display cases.

Indirect system
Opposed to direct expansion system, indirect systems comprise a primary evaporator installed in the machinery room where the refrigerant cools a heat transfer fluid that circulates back and forth from the machinery room to the display cases. The heat transfer fluid can be a single-phase liquid such as Mono-Propylene-Glycol or a phase-change one such as CO_2.

Infiltration heat load
It is the thermal load related to air infiltration in the display case.

Inner heat exchange coefficient
For a tube-and-fin heat exchanger such as those used in refrigerated display cases, the inner heat exchange coefficient is the heat exchange coefficient on the refrigerant side.

Internal radiation coefficient
Transfer coefficient taking into account exchange between the different surfaces of the display case at different temperatures.

Outer heat transfer coefficient
For a tube-and-fin heat exchanger such as those used in refrigerated display cases, the outer heat exchange coefficient is the heat exchange coefficient on the air circulating side.

Overall heat exchange coefficient
For heat exchangers, the heat exchange depends on the heat exchange coefficient on each side, and also slightly on the conductivity of the tubes. The overall heat exchange coefficient integrates the two heat exchange coefficients and the tube conductivity.

Parc (equipment "fleet")
Is the French word corresponding to the installed base of equipment, meaning all equipment of a given type whatever their vintage.

Radiation heat load
Is the heat load due to radiative heat transfers.

Recovery Efficiency
Is the ratio of the recovered refrigerant to the refrigerant stored in the refrigeration system.

Secondary loop system
Is identical to indirect systems, the secondary loop being the circuit where the heat transfer fluid circulates.

Stand-alone equipment
Can also be called plug-in system. It is a refrigeration system completely integrated in the refrigeration equipment. Domestic refrigerators are typical stand-alone equipment as well as vending machines, ice machines, and also some display cases.

Stefan Boltzmann's constant
Is a constant used for the calculation of radiative heat transfers.

Surface emissivity
Is a property of a surface related to radiative heat transfer. For opaque surface, the higher the emissivity, the higher the heat absorption.

Thermal conductivity
Is the property of a material to transfer heat by conduction. It is expressed in W/m.K.

Top-down approach (for inventory or consumption estimates)
See Bottom-up approach.

Unitary AC system
Corresponds to stationary air-conditioning system of refrigeration capacity varying from 1 to 15 kW. Those systems are so called air-to-air meaning that the evaporator as well as the condenser are refrigerant-to-air heat exchangers.

Volumetric air flow rate
Flow rates can be expressed either as mass flow rates or volumetric flow rates. Their respective units are kg/s and m^3/s.

ABBREVIATIONS AND ACRONYMS

Abbreviations

AEO	Annual Energy Outlook
AFEAS	Alternative Fluorocarbons Environmental Acceptability Study
ANOPR	Advanced Notice of Proposed Rulemaking
ASH	Anti-sweat heater
BAU	Business as usual
BSRIA	Building Services Research and Information Association
CDB	Country Data Base
CEC	California Energy Commission
CEP	Center for Energy and Processes
CFC	Chlorofluorocarbon
CO_2	Carbon dioxide
COP	Coefficient of performance
CRF	Common Reporting Format
CSD	Carbonated soft drink
DC	Display case
DEF	Defrost control
DOE	Department of Energy
DX	Direct expansion
ECM	Electronically commutated permanent magnet motor
Eco	Energy consumption
EIA	Energy Information Agency
FAO	Food and Agricultural Organization (of the United Nations)
FHP	Floating head pressure
FHPC	Floating head pressure control
GDP	Gross Domestic Product
GHG	Greenhouse gas
GWh	Giga-watt hour (one billion [10^9] watt hour)
GWP	Global warming potential
HC	Hydrocarbon
HCFC	Hydrochlorofluorocarbon
HFC	Hydrofluorocarbon
HTF	Heat transfer fluid
HVAC	Heating, ventilation, & air-conditioning
IEA	International Energy Agency
IPCC	Intergovernmental Panel on Climate Change
kJ	Kilojoules
kWh	kilo-watt hour (one thousand watt hour)
LCC	Life-cycle cost
LCCP	Life-cycle Climate Performance
LED	Light-emitting diode
MFR	Mass flow rate
MPa	Mega-pascals (one million [10^6] pascals)
MPG	Mono-propylene glycol
MWh	Mega-watt hour (one million [10^6] watt hour)
NOAA	National Oceanic and Atmospheric Administration
ODS	Ozone-depleting substance(s)
OECD	Organization for Economic Co-operation and Development
PBP	Payback period
PLC	Program logic circuit
PSC	Permanent split capacitor
RIEP	Refrigerant Inventories and Emission Previsions

SA	Stand-alone
SAR	Second Assessment Report (of the IPCC)
SDT	Saturated discharge temperature
SST	Saturated suction temperature
TAR	Third Assessment Report (of the IPCC)
TEWI	Total Equivalent Warming Impact
TWh	Tera-watt hour (one trillion [10^{12}] watt hour)
UHT	Ultra-high temperature
UNEP	United Nations Environment Programme
UNFCCC	United Nations Framework Convention on Climate Change
USDA	U.S. Department of Agriculture
VOPMT	Vertical open medium temperature case
WI	Walk-in cooler

NOMENCLATURE

Symbol	Term	Units of expression
A	surface area	m^2
F	internal radiation coefficient	
h	air enthalpy	kJ/kg
h_i	inner heat exchange coefficient	$W/m^2.K$
h_o	outer heat transfer coefficient	$W/m^2.K$
k	thermal conductivity	W/m.K
nd	number of air exchanges in room per 24 hours	
$Q_{defrost}$	extra heat from defrost	W
Q_{fan}	fan motor input power	W
$Q_{heat\text{-}wires}$	input power of anti-sweat heaters	W
$Q_{infiltration}$	Infiltration heat load	W
$Q_{lighting}$	heat dissipation load	W
Q_{load}	total heat load	W
$Q_{radiation}$	Radiation heat load	W
Q_{wall}	conduction heat load	W
T	temperature	°C or K
t	insulation thickness	m
U	overall heat exchange coefficient	$W/m^2 \cdot K$
V	volumetric air flow rate	m^3/s

Greek symbols

ε	surface emissivity	
ρ	density	kg/m^3
σ	Stefan Boltzmann's constant	

Subscript

air,ent	entrapped air
air,ex	air infiltration
c	condensation (or cold for COP)
case	display cabinet
CR	cold room
e	evaporation

APPENDICES

CONTENTS

ANNEX 1 – List of visited stores

Store Category	Major Brands		Number of stores visited
Large Supermarket	Costco Wholesale		2
	Target		2
	Walmart		7
	Walmart Supercenter		4
		Total	**15**
Grocery Store	Albertson		6
	Stater Bros		3
	Bristol Farms		3
	Food 4 less		3
	Raleys		3
	Ralphs		3
	Safeway		3
	Vons		4
	Wholefoods		5
	SuperAfood		3
	SuperSuperWarehouse		2
		Total	**38**
Minimarket	Smart &Final		3
		Total	3
Convenience store	7/11		5
	AM-PM		3
	Local Convenience stores		4
		Total	**12**
Liquor Store	local liquor stores(B&B Jr Market, Village liquor store, Picomarket, Sam's Liquor,…)		
		Total	**5**
Pharmacy	CVS		3
	RiteAid		3
	Walgreen		4
		Total	**10**
Gas Station	Small Gas Station (76, Chevron, Mobile, Exxon, Arco)		14
	Large Gas Station (Mobil, Walmart Center)		4
		Total	**18**
Hotel	Best Western, Hilton, Marriott, Holiday Inn		
		Total	**8**
Motel	America's Best Value Inn, Super 8 Motel, Comfort Inn		
		Total	**5**
Bakery		**Total**	**1**
Butchery		**Total**	**4**
Fishmonger		**Total**	**2**
Bar &Restaurants	Bar, Restaurants,FastFood, Cinema, Bowling		**1**
Number of stores visited			**122**

ANNEX 2

Inventories of the worldwide fleets of refrigerating and air-conditioning equipment in order to determine refrigerant emissions. The 1990 to 2003 updating. ADEME/ARMINES Agreement 04 74 C0067–
Excerpts from the Final Report of December 2005 – Version 3, July 2006

Section 1 and Annexes 1 and 2 to Section 1

CONTENTS

Method of calculation, data and databases

1.1 Calculation method for emission prevision of refrigerants

The Tier 2 method, as defined in the IPCC guidelines [IPCC96, IPCC 06] proposes a calculation for HFC refrigerant emissions from equipment:

- during the manufacturing process,
- during the lifetime, and
- at the end of life of equipment.

This approach of looking at refrigerating equipment from cradle to grave (see Figure 1.1) covers all possible emissions but needs to be further worked out in order to give consistent results.

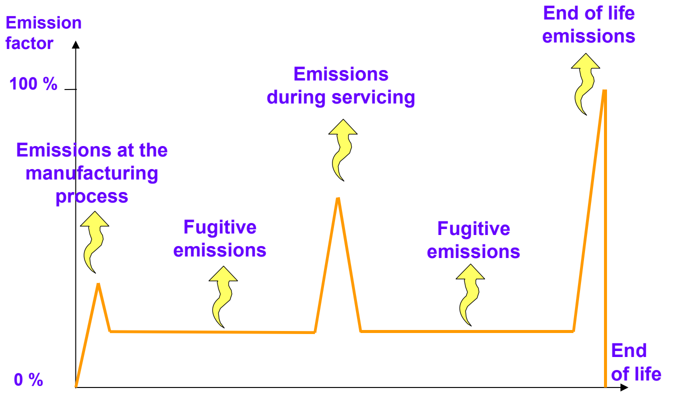

Figure 1.1 – Types of emissions from cradle to grave from refrigerating equipment.

The equations are coming from the draft version of the 2006 IPCC Guidelines (second draft, August 2006) and have taken into account the work done by the CEP (Center for Energy and Processes) during the last eight years. They are presented in Annex 1.

The same method is being used for the refrigerant inventories and emission forecasts for the French Government [BAR05, PAL04a, PAL04b, and PAL03a] delivered to the CITEPA, which is the technical body in charge of French inventories of greenhouse gases to be delivered to UNFCCC.

♦ **Emissions at the manufacturing process**

When equipment is mass-produced, the direct emissions are usually very small. For field-assembled systems, the emissions during the installation phase are higher but not substantial. The main source of emissions related to charging and topping up of refrigerating equipment are mainly **the emissions due to refrigerant handling.**

One will find refrigerant handling in more than just the manufacturing process of the equipment. There needs to be included:

- splitting the bulk refrigerant in large containers into smaller volumes of refrigerant,
- losses related to connecting the smaller refrigerant volumes to the equipment, and
- **capacity "heels".**

The capacity "heels" represent the main loss during refrigerant handling. The “heels’ consist in fact by of the vapor inside the container, which cannot be extracted due to the pressure equilibrium between the vapor (the vapor heel) and the liquid phase remaining in the refrigerant volume (the liquid heel). Based on the recovery policy and the experience of the main refrigerant distributor in France, it can be derived that those “heels” represent between **2 and 10 % of the total amount of refrigerant sales.** This includes the charge of new equipment and the recharge of all the existing fleets of refrigerating equipment.

Note: the English word **fleet** covers the total number of equipment, e.g., for mobile air conditioning in cars, for refrigerating trucks, for reefers and refrigerating containers. It seems to be much more difficult to use the word fleet for domestic refrigerators, for refrigerating equipment in industrial processes and for stationary air conditioning systems. It is therefore proposed to use the French word "parc", which is easily understood in English and the following definition then applies: **"parc" is the total number of pieces of equipment in a category or sub-domain independent of their vintage.**

One of the improvements applied to the 1996 Tier 2 method of the IPCC Guidelines is the inclusion of the emissions from the container heels in the total sales of refrigerant.
Note: this improvement has been included in the 2006 IPCC Guidelines.

♦ **Emissions during the lifetime of the equipment**

Leaks during the lifetime of equipment depend on the type of application, e.g., domestic refrigerators show very low emission rates during their lifetime. On the contrary, many commercial, centralized refrigeration equipment and refrigerated transport systems are highly emissive. **Emission previsions need to be based on feedback via field data,** and field data from each country will substantially improve a number of global assumptions made in this study. In large commercial facilities or in industrial processes, the most precise approach for the determination of emissions is the collection of receipts and/or invoices for refrigerant delivered for system maintenance and for recharges.

In order to yield accurate results, the mobile air conditioning systems require very sophisticated methods. It is very common to form groups of vehicles of different vintages where the remaining refrigerant is carefully recovered from the system and subsequently measured by accurate weighing. By determining the difference between the initial refrigerant charge and the recovered charge, average levels of refrigerant emissions can be established.

♦ **Emissions from equipment at end of life**

Emissions from equipment at end of life depend on one hand on the regulatory policies in different countries, on the other hand on the recovery efficiency. For the inventory determination method, it is essential to have correct information regarding the lifetime of equipment, and annual market data for a number of years in the past, equal to the lifetime of the product. This point is crucial for almost every type of application due to:

- The rapid change in the application of refrigerant types, which changes are related to changing Montreal Protocol control schedules, and particularly to more stringent regional or national regulations,
- The rapid market growth of certain types of equipment, e.g. mobile air conditioning systems during recent years in Europe, or the rapid annual growth in China,
- The change in how recycling policies at the end of life of the equipment are regulated.

Taking into account
(1) the large numbers of equipment,
(2) the large variation in equipment type,

(3) the refrigerant charge amounts, and
(4) the different refrigerant types and their GWPs,
a large database has to be constructed, on an application by application basis. For each application, **the "parc" has to be derived for all the years covering the lifetime of this type of equipment**. Moreover, as the determination of inventories is performed on an annual basis, the updating of the database is a necessary factor to take into account.

1.2 Refrigerants and regulations

The use of CFCs, HCFCs or HFCs and other refrigerants is related to control schedules, which have been continuously adjusted since the Montreal Protocol has been ratified. For the developed countries (the non-Article 5 countries as defined in the Montreal Protocol), the phase-out of CFCs and HCFCs will be earlier than in the developing countries (the Article 5 countries). Moreover, where it concerns non-Article 5 countries, the European Union has accepted a much tighter control schedule for phasing out (CFCs in the past and) HCFCs.

The rapid CFC phase out in Europe and also the interdiction of use of CFCs for servicing have led to a significant uptake of intermediate blends (HCFC-based blends) for the retrofit of a number of refrigerating systems using CFCs. The retrofit allows to keep the residual value of equipment until its usual end of life. It is likely that the same behavior of equipment owners will be followed for the progressive phase out of HCFCs, which will be replaced by intermediate blends of HFCs. Based on these facts, RIEP includes retrofit options where the refrigerant can be changed during the equipment lifetime.

♦ Non-Article 5 countries

The CFC phase-out schedule as valid for the non-Article 5 countries is presented in Figure 1.2. Via the EU regulation 3093/94 CFCs were phased out one year before the phase-out defined in the Montreal Protocol, i.e. on 31 December 1994.

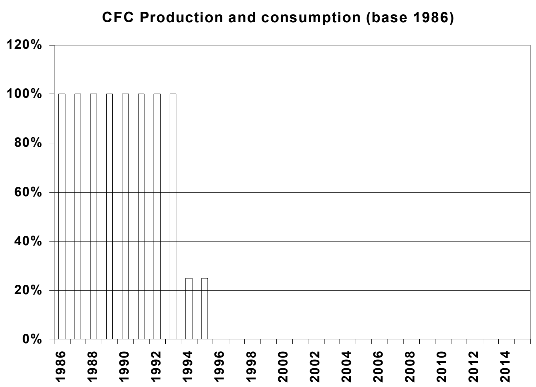

Figure 1.2 – CFCs phase out in non Article 5 countries.

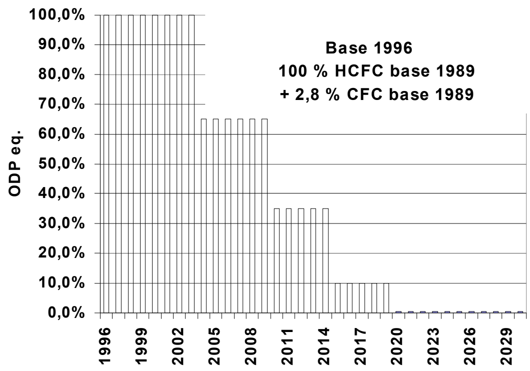

Figure 1.3 – HCFCs phase out in non Article 5 countries (except EU).

As indicated in Figure 1.3, the HCFC consumption base levels refer to the 1989 HCFC consumption plus 2.8% 1989 CFC consumption, ODP-weighted. On the basis of a certain ODP for HCFC-22 and CFCs (0.055 and 1.0 respectively), the factor of 2.8% means that if all CFCs would be replaced by HCFC-22, about 55% of the CFC consumption in tonnes would be replaced by HCFC-22.

Figure 1.3 clearly shows that, even for non-Article 5 countries, brand-new equipment can be manufactured, charged with HCFC-22 and sold until 31 December 2009. Typically, the U.S. and many developed countries continue to use HCFC-22 for air-conditioning equipment.

As indicated in Figure 1.4, the EU regulation has changed the baseline level for the HCFC consumption by reducing the additional quantities of ODP weighted CFCs by nearly 30% (from 2.8 to 2.0%). Moreover, the time of the HCFC phase-out is been brought forward by about 7 years.

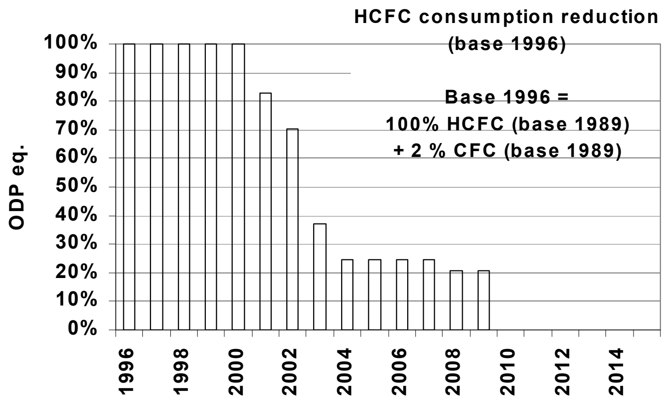

Figure 1.4 - European Union - (European regulation 2037/2000).

- **Article 5 Countries**

The CFC consumption and production (see Figure 1.5) for Article 5 countries has a delay compared to non-Article 5 countries of actually 14 years (1996 compared to 2010). There is an additional possibility of production and consumption of 10% compared to the 1996 level for Basic Domestic Needs of the developing countries where production can take place in the developed countries.

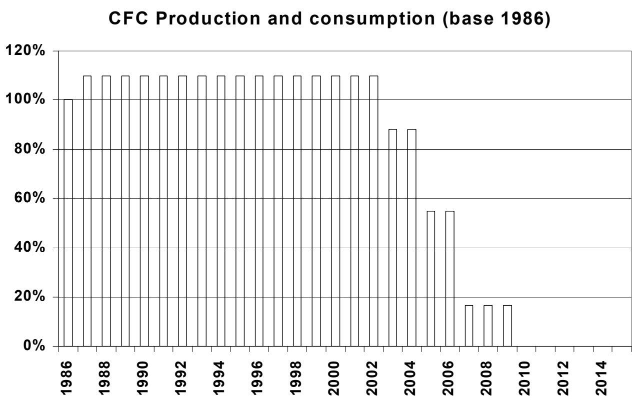

Figure 1.5 - CFC phase-out for Article 5 Countries.

For the HCFC phase-out the Montreal Protocol schedules are a bit more complicated. Where it concerns the freeze in consumption, Article 5 countries have a delay of about 15 years (freeze by 2016). Where it concerns the phase-out it actually is a 10-year delay period (phase-out in 2040 versus 2030) for the developing compared to the developed countries.

All these different constraints based upon global control schedules and more stringent regional and national regulations imply different refrigerant choices in countries and country groups. The refrigerant choices need to be taken into account on an application by application basis. In this project additional data have been used that have been derived from country reports as well as data that were available in publications.

1.3 Refrigerant GWPs from the Second and the Third Assessment Report of the IPCC

Table 1.1 lists the main refrigerant types in use: CFCs, HCFCs, HFCs, ammonia, and different blends, many of them being intermediate blends used for retrofit of CFC equipment. Table 1.1 has been updated taking into account all new blends as declared to ASHRAE 34. The most used of those blends are R-401A, R-409A, and R-413A for the replacement of CFC-12, R-402A and B, and R-408A for the replacement of R-502. The use of those blends can be verified at the global level by the declarations of sales by AFEAS of HCFC-124 and HCFC-142b, which are specific components of those intermediate blends. The list is nearly exhaustive, and takes into account more than 99% of all refrigerant types used. **The GWP values as given in the Second Assessment Report of the IPCC (SAR)** are used for the calculations of the equivalent CO_2 emissions of refrigerants.

Table 1.1 – GWP, physical of refrigerants [TOC03, IPCC06].

Refrigerant		Physical data				GWP		
Number	Chemical formula or blend composition – common name	Molecular mass	NPB (°C)	TC (°C)	Pc (Mpa)	GWP 2nd AR 1996	GWP 3rd AR 2001	% 2nd /3rd
11	CCl3F	137.37	23.7	198.0	4.41	3 800	4 600.0	21
12	CCl2F2	120.91	-29.8	112.0	4.14	8 100	10 600.0	31
22	CHClF2	86.47	-40.8	96.2	4.99	1 500	1 700.0	13
32	CH2F2-methylene fluoride	52.02	-51.7	78.1	5.78	650	550.0	-15
115	CF3CClF2	154.47	-38.9	80.0	3.12	9 300	7 200.0	-23
116	CF3CF3-perfluoroethane	138.01	-78.2	19.9	3.04	9 200	9 200.0	0
123	CHCl2CF3	152.93	27.8	183.8	3.66	90	120.0	33
124	CHClFCF3	136.48	-12.0	122.3	3.62	470	620.0	32
125	CHF2CF3	120.02	-48.1	66.2	3.63	2 800	3 400.0	21
134 a	CH2FCF3	102.03	-26.1	101.1	4.06	1 300	1 300.0	0
143 a	CH3CF3	84.04	-47.2	72.9	3.78	3 800	4 300.0	13
152 a	CH3CHF2	66.05	-24.0	113.3	4.52	140	120.0	-14
245 fa	CHF2CH2CF3	134.05	15.1	154.1	4.43	820	950.0	16
290	CH3CH2CH3 - propane	44.10	-42.1	96.7	4.25	20	20.0	0
401 A	R-22/152a/124(53/13/34)-MP39	94.44	-34.4	105.3	4.61	973	1 127.4	16
401 B	R-22/152a/124(61/11/28)-MP66	92.84	-35.7	103.5	4.68	1 062	1 223.8	15
402A	R-125/290/22(60/2/38)-HP80	101.55	-49.2	76.0	4.23	2 250	2 686.4	19
402B	R-125/290/22(38/2/60)-HP81	94.71	-47.2	83.0	4.53	1 796	2 108.4	17
403A	R-290/22/218(5/75/20)	92	-47.8	87	4.7		3 000	
403B	R-290/22/218(5/56/39)	103.2	-49.2	79.7	4.32		4 300	
404A	R-125/143a/134a(44/52/4)	97.60	-46.6	72.1	3.74	3 260	3 784.0	16
405A	R-22/152a/142b/C318(45/7/5.5/42.5)	111.9	-32.6	106.1	4.29		5 200	
406A	R-22/600a/142b(55/4/41)	89.9	-32.5	116.8	4.96		1 900	
407A	R32/125/134a(20/40/40)	90.1	-45	82.3	4.52		2 000	
407B	R32/125/134a(10/70/20)	102.9	-46.5	75	4.13		2 700	
407C	R-32/125/134a(23/25/52)	86.20	-43.8	87.3	4.63	1 526	1 652.5	8
407D	R-32/125/134a(15/15/70)	91	-39.2	91.4	4.47		1 500	
407E	R-32/125/134a(25/15/60)	83.8	-42.7	88.5	4.7		1 400	
408A	R-125/143a/22(7/46/47)-FX-10	87.01	-45.5	83.3	4.42	2 649	3 015.0	14
409A	R-22/124/142b(60/25/15)-FX-56	97.43	-35.4	106.9	4.69	1 288	1 535.0	19

Refrigerant		Physical data				GWP		
Number	Chemical formula or blend composition – common name	Molecular mass	NPB (°C)	TC (°C)	Pc (Mpa)	GWP 2^{nd} AR	GWP 3^{rd} AR	%
						1996	2001	$2^{nd}/3^{rd}$
410A	R-32/125(50/50)-Suva9100;AZ-20	72.58	-51.6	72.5	4.95	1 730	1 975.0	14
411A	R-1270/22/152a(1.5/87.5/11)	82.4	-39.5	99.1	4.95		1 500	
412A	R-22/218/142b(70/5/25)	92.2	-38	107.2	4.9		2 200	
413A	R-218/134a/600a(9/88/3)	104	-30.6	98.5	4.07		1 900	
414A	R-22/124/600a/142b(51/28.5/4/16.5)	96.9	-32.9	112.7	4.68		1 400	
415A	R-22/152a(82/18)	81.9	-37.2	102	4.96		1 400	
416A	R-134a/124/600(59/39.5/1.5)	111.9	-24	107	3.98		1 000	
417A	R-125/134a/600(46.6/50/3.4)	106.7	-39.1	87	4.04		2 200	
418A	R-290/22/152a(1.5/96/2.5)	84.6	-41.6	96.2	4.98		1 600	
419A	R-125/134a/E170-77/19/4)	109.3	-43.8	79.2	4		7 900	
420A	R-134a/142b(80.6/19.4)	101.7	-24.2	107.2	4.11		1 500	
421A	R-125/134a(58/42)	111.7	-35.5	82.4	3.88		2 520	
422A	R-125/134a/600a(85.1/11.5/3.4)	113.5	-43.2	75.4	3.92		3 040	
500	R-12/152a(73.8/26.2)	99.30	-33.6	102.1	4.17	6 014	7 854.2	31
502	R-22/115(48.8/51.2)	111.63	-45.3	80.7	4.02	5 494	4 516.0	-18
503	R-23/13(40.1/59.9)	87.25	-87.5	18.4	4.27	11 700	13 198	13
504	R-32/115(48.2/51.8)	79.25	-57.7	62.1	4.44	5 131	3 994.7	-22
505	R-12/31(78.0/22.0)	103.48	-30.0	117.8	4.73	6 318	8 268.0	31
506	R-31/114(55.1/44.9)	93.69	-12.3	142.2	5.16	4 131	4 400.0	7
507A	R-125/143a(50/50)-AZ-50	98.86	-47.1	70.9	3.79	3 300	3 850.0	17
600a	CH(CH3)2-CH3 - isobutane	58.12	-11.6	134.7	3.64	20	20.0	0
717	NH3 - ammonia	17.03	-33.3	132.3	11.33	< 1	< 1	
744	CO2	44	-78.4	31	7.38		1	

NBP = normal boiling point; Tc = critical temperature; Pc = critical pressure; GWP = global warming potential (for 100 yr integration).

The GWP calculation for blends is based on the GWP values of the pure refrigerants, and their mass concentration in the blend. It has been preferred to not round the GWP numbers for blends so that their origin can still be traced. For propane and isobutane no official GWP values have been presented in the IPCC Third Assessment Report, and the rounded value of methane (23) has been taken for all HCs.

1.4 Consistency and improvement of data quality

Using the Tier 2 method, the consistency in the emission forecast cannot be directly verified. **The first essential cross check can be done via deriving the annual market of the different refrigerant types** based on the initial charge of brand-new equipment (on an application by application basis) and on the recharge at servicing of the different "parcs" of equipment. By merging those two data series, it should be possible to derive the size of the market for every refrigerant type **and to compare those data to the official data submitted by manufacturers and distributors.**

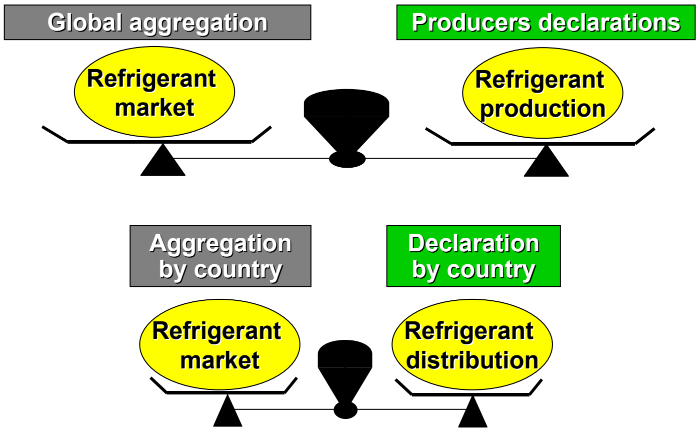

Figure 1.6 – Cross check of the annual refrigerant market derived from the initial charges and the recharges with the declarations made by refrigerant producers.

The cross-checks can be performed both on a country by country basis and globally (see Figure 1.6).

If the refrigerant inventories and the related emissions are adequately determined, the difference between the figures submitted and the calculated refrigerant sales will be small. If not, additional analyses are required.

- **Consistency for refrigerating equipment at the global level**

To reach a high accuracy in the sizes of the refrigerant inventories, the first step required is to gather reliable data for the equipment numbers. Fortunately, annual statistical data are available for nearly all mass-produced equipment. Some data have been published by manufacturer associations, and some (marketing studies) can be purchased from specialized companies. The data on annual equipment sales allow deriving figures on production and sale at the national level for nearly all the OECD countries, and also at the global level, when they are based on production data (see Figure 1.7).

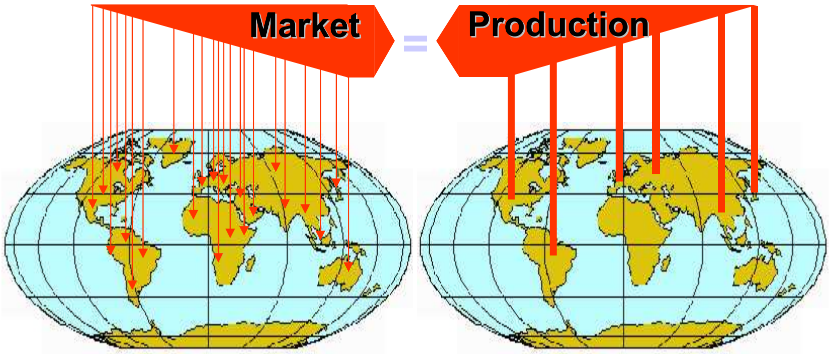

Figure 1.7 – Cross check between markets and production quantities.

At the global level, for a given year one can postulate "Production = Sales" (except for the small amount of equipment produced but not yet sold). For domestic refrigerators, stationary air conditioning systems, chillers, cars, trucks, buses, reefers… annual numbers of production and sales are available. Application of these numbers avoids double counting, which would happen easily when national inventories are merged, particularly if methods of determination are different.

♦ **Inventories of all refrigerant types and the method of aggregation**

The schedule for phasing out CFCs and HCFCs depends for the larger part on country regulations (see section 1.2). Even if only HFC inventory reporting is required under the UN Framework Convention on Climate Change (UNFCCC), it is required to have information on the emission predictions and on the changes in refrigerant use. Only in this way the size of the "banks" of all types of refrigerants charged in the different types of equipment can be determined. The --changing-- trends in the selection of the refrigerant need to include the quantities of hydrocarbons (HCs) and ammonia, which are both being used as HFC replacement options in the European Union.

As shown in Figure 1.8 the bottom-up approach used defines:

- the annual sales of brand-new equipment and the amount of refrigerants charged in this equipment,
- the determination (dependent on their lifetime) of all the fleets or parcs, which yields a cumulative value for the refrigerant bank for the specific application,
- the determination of the refrigerant market for servicing (dependent on the leak factor), and thereafter all the different domains are aggregated
- refrigerant by refrigerant,
- country by country,

by country groups and globally.

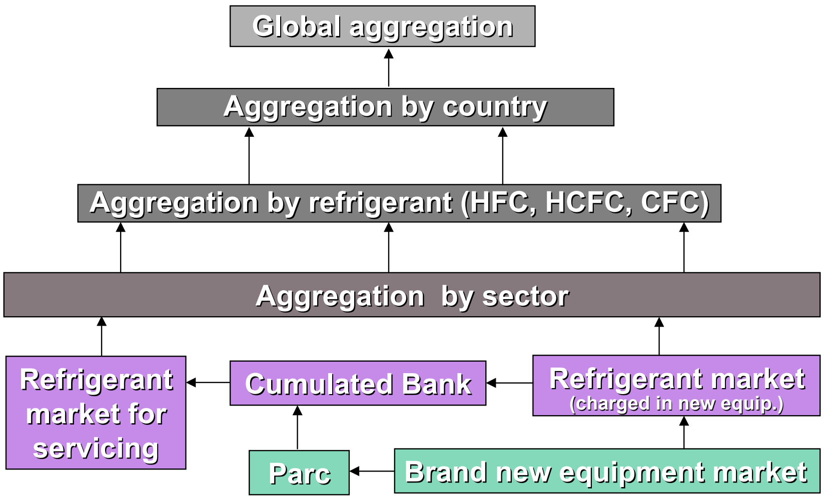

Figure 1.8 – Determination of the refrigerant markets.

This method of cross-check has been adopted in the quality assurance process of the updated version of the IPCC Guidelines 2006.

1.5 Tools for refrigerant inventories and emission previsions

To determine the annual emission forecasts for all categories of refrigerating equipment, it is necessary to create the tools that allow cumulative improvements in the data quality. The large number of data to be handled necessitates:

- to program in a database language
- to perform calculations based on reality data
- to create user friendly interfaces

- to transfer the results to tables written in spreadsheet language, which tables are based on the prescribed Common Reporting Format (CRF) of the IPCC for HFCs.

For the first year, such a database needs to “create” the "parcs" of all the different categories and sub-categories of refrigerating equipment. For the years thereafter the updating process requires less efforts and basically consists of the following input data:
- the annual equipment market for each category in the reference year
- the type of refrigerant used in brand-new equipment, and possibly also information on conversion from CFCs or HCFCs to HFCs or other refrigerants
- the emission factors.

All those elements allow to perform:
- calculations of emissions from all existing parcs of equipment,
- calculations of emissions from all types of decommissioned equipment
- a calculation of the amount of refrigerants which are recovered or reclaimed
- a calculation of the refrigerant banks per category of equipment
- a calculation of the annual refrigerant market sales, per refrigerant type.

As soon as better data become available, the database can be updated in a transparent manner. National, regional or global data reviews are necessary in order to control the quality of the inventory determinations.

A database enables the development of data acquisition in a single way: **improvement**, because it creates storage of data on the refrigerants in use inside the parcs of equipment that have been calculated.

1.5.1 Refrigeration equipment and refrigerant bank database

RIEP is connected to another database, CDB (Country Data Base), which has been developed as the source for economic, demographic, and technical data for both countries and country groups (see Annex 1).

RIEP is written in the ACCESS language, and deals with the separate countries, for any given year. Based on inputs from the user interface, RIEP can calculate the emissions during the equipment lifetime (see Figure 1.9). For these calculations, data have to be used for each year of the lifetime of a given equipment type or category.

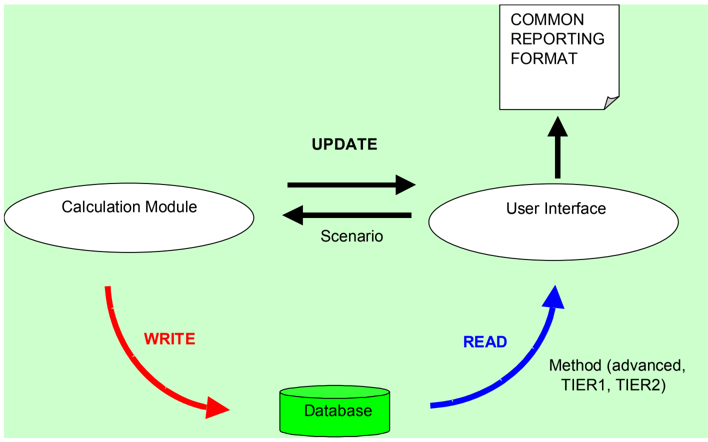

Figure 1.9 – Scheme of the application of the RIEP program.

1.5.2 Country Data Base

As indicated in Figure 1.10, if one selects a certain year and either the national or the regional level, the CDB can produce data on:

- demography,
- energy production and consumption,
- agriculture, and
- economy, including commerce.

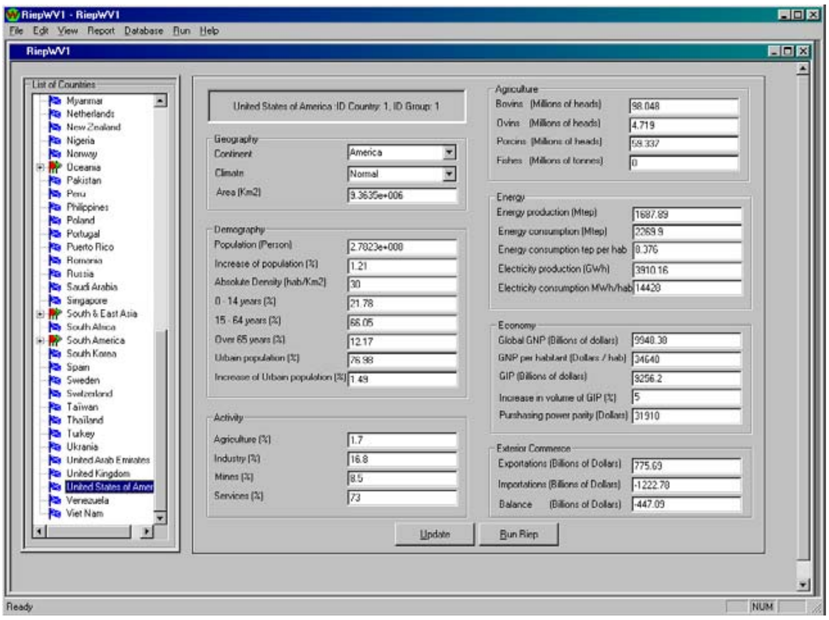

Figure 1.10 – Example of a screen of the United States of America CDB.

The Country Data Base (CDB), which has been constructed for the determination of global inventories, covers 62 countries and 8 regions, each region containing a portion of the remaining 110 countries (see annex 1).

For countries where only few specific equipment data is available, some of the general data mentioned above can be used to create ratios between refrigerating equipment, national economy and population. From the CDB, it is possible to run the RIEP program. The CDB is also written in Access and interfaces are handled in the C^{++} language.

India and Brazil are analyzed per se because of their economic growth. Moreover taking into account the integration of the 10 new European countries, Europe is followed as Europe 25. Russia is also followed per se, Oceania has been merged with other Asia and Australia is followed per se. The database has been used for the Supplementary Report [UNE05] and specific groupings have been done for non Article 5 countries and Article 5 countries.

1.6 Review process

The results of the previous report on inventories of the worldwide fleets of refrigerating and air-conditioning equipment [PAL03b] have been thoroughly used in the IPCC TEAP report [IPC05] and in the Supplement to the IPCC/TEAP report [UNE05]. Data have been analyzed by a number of experts, among them L. Kuijpers, A. MacCulloch, M. Mc Farland, S. Solomon, F. Keller, N. Campbell, and many others. Their comments have been fruitful and the main changes or improvements have been as follows:

- The distribution assigned between CFC-11 and CFC-12 for chillers in many countries was not well set in the previous version with a too high share of CFC-12; in fact the "US model" where CFC-11 was predominant had a strong influence in all Asian countries.
- R-502, which was significantly used in commercial refrigeration in Europe was much less used in the U.S. So HCFC-22 was underestimated in the U.S. inventory, and R-502 overestimated. Corrections have been done and as it is seen in Section 2 the correction has been effective due to the quite good match between AFEAS data on CFC-115 (CFC only used in R-502) and RIEP calculations.
- The phase in of HFCs in stationary air conditioning in the U.S. has been overestimated whereas HCFC-22 was nearly the only refrigerant in use until the end of 2005.

Independently of those modifications, the main other modifications based on new data are:

- the integration of retrofit blends for the replacement of CFCs,
- a new method of calculation for the number of refrigerate trucks based on the evolution of food products followed by the FAO database,
- modification of the emission model for mobile air conditioning systems, which is no longer taking into account a percentage, but the value expressed in g/yr because it has been demonstrated that emissions are not directly related to the refrigerant charge.

One of the best review processes if that the document is used by international experts and that the emissions as presented are compared to atmospheric concentration. This work has begun with the two papers published in the International Journal of Refrigeration with P. Ashford, A. Mc Culloch and L. Kuijpers [ASH04a, ASH04b, ASH04c]. Other review papers are under preparation in order to develop the correlation between atmospheric concentration and emissions of refrigerants.

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[IPC05] IPCC TEAP, 2005: *IPCC/TEAP Special Report on Safeguarding the ozone Layer and the Global Climate System: Issues related to Hydrofluorocarbons and Perfluorocarbons.* Prepared by Working Group I and III of the Intergovernmental Panel on Climate Change and the Technology and Economic Assessment Panel under the Montreal Protocol (Metz, B., L. Kuijpers, S. Solomon, S.O. Andersen, O. Davidson, J. Pons, D. de Jager, T. Kestin, M. Manning, and L.A. Meyer (editors). Cambridge University Press, Cambridge, UK, and New York, NY, USA, 488 pp

[IPC06] Draft version of the Revised 2006 Guidelines for National Greenhouse Inventories: OECD / IEA Paris.

[IPC96] Revised 1996 Guidelines for National Greenhouse Inventories: OECD / IEA Paris.

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[UNE03] 2002 Report of the Refrigeration, Air Conditioning and Heat Pumps Technical Option Committee (RTOC). 2002 Assessment. March 2003. ISBN 92-807-2288-3.

Equations used for the Calculation Method

The calculation method complements the Tier 2 method as recommended by the IPCC, by cross checking:

- the sum of refrigerant quantities charged in brand-new equipment and those recharged for servicing purposes in all the different refrigerating systems, with
- the annual national market sales of refrigerants as declared by the refrigerant manufacturers and distributors.

The method includes the following calculations:

- the refrigerant « bank » at year ***t*** charged into the parc of systems of each of the six system categories (taking into account the refrigerant changes as a result of regulations which could apply),
- the emissions of each system category, based on the understanding where the emissions occur (at system charge, during operation, during servicing and at the system's disposal).

The six categories of systems have been selected following the division used by the UNEP[1] Refrigeration Technical Options Committee [UNEP03]. The refrigerating chain includes:

- domestic refrigeration
- commercial refrigeration
- refrigerated transports
- refrigerated warehouses, food storage and industrial processes.

Air conditioning includes two sub-groups:

- air to air systems and water chillers
- mobile air conditioning.

All these categories must again be split in sub-groups.

Calculation method (Tier 2a) (extract from the draft of the IPCC Guideline 2006 Draft)

Note: here only HFCs are addressed. In RIEP all types of refrigerants are taken in account.

Refrigerant emissions at a year **t** from the six categories of refrigeration and air conditioning systems, result from:

1 emissions related to the management of refrigerant containers: $E_{containers,t}$
2 emissions related to the refrigerant charge :connection and disconnection of the refrigerant container and the equipment to be charged: $E_{charge,t}$
3 emissions from the six banks during operation (fugitive emissions and ruptures): $E_{operation,t}$
4 emissions during servicing: $E_{servicing,t}$
5 emissions at system disposal: $E_{disposal,t}$

All these quantities are expressed in kilograms and have to be calculated for each type of HFC used in the six different application categories.

$E_{total,\ t} = E_{containers,t} + E_{charge,\ t} + E_{operation,\ t} + E_{servicing,\ t} + E_{disposal,\ t}$ Equation 1

Methods for estimating average emission rates for the above-mentioned domains need to be calculated on a refrigerant by refrigerant basis for all equipment whatever their vintage.

Refrigerant management of containers

[1] United Nations Environment Programme

The emission related to the refrigerant container management comprises all the emissions related to the refrigerant transfers from bulk containers (typically 40 tonnes) down to small capacities where the mass varies from 0.5 kg (disposable cans) to 1 tonne (containers) and also from the remaining quantities --the so-called refrigerant "heels" (vapour and /or liquid)-- left in the various containers, which are recovered or emitted.

$E_{containers, t} = RM_t \bullet (c)$ Equation 2

where:

$E_{containers,t}$	=	emissions from all HFC containers in year t expressed in kilograms
RM_t	=	the HFC market for new equipment and servicing of all refrigeration application in year t expressed in kilograms
c	=	Emission factor of HFC container management of the current refrigerant market expressed in percentage

The emissions related to the complete refrigerant management of containers are estimated between 2 and 10 % of the refrigerant market.

Refrigerant charge emissions of new equipment

The emissions of refrigerant due to the charging process of new equipment are related to the process of connecting and disconnecting the refrigerant container to and from the equipment.

$E_{charge, t} = M_t \bullet (k)$ Equation 3

where:

$E_{charge,t}$	=	emissions during system manufacture/assembly in year t expressed in kilograms
M_t	=	The amount of HFC charged into new equipment in year t (per application category) expressed in kilograms
k	=	assembly losses of the HFC charged in new equipment (per application) expressed in percentage

Note: the emissions related to the process of connecting and disconnecting during servicing are covered in Equation 5 for servicing.

The amount charged (M_t) should include all systems which are charged in the country, including those which are produced for export. Systems that are imported pre-charged should not be considered.

Typical range for the emission factor k varies from 0.1 % to 2 %. The emissions during the charging process are very different for factory assembled systems and for field-erected systems where emissions can be up to 2 %.

Emissions during operation

Annual leakage from the refrigerant banks represent fugitive emissions, i.e. small leaks from fittings, joints, shaft seals, … but also ruptures of pipes or heat exchangers leading to partial or full release of refrigerant to the atmosphere. The following calculation formula applies:

$E_{operation,\ t} = B_t \bullet (x/100)$ Equation 4

where:

$E_{operation,\ t}$	=	amount of HFC emitted during system operation in year t expressed in kilograms
B_t	=	amount of HFC banked in existing systems (per application) in year t expressed in kilograms
x	=	annual leakage rate of HFC of each application bank during operation expressed in percentage

In calculating the refrigerant "bank" (B_t) all systems in operation in the country (produced domestically and imported) have to be considered on an application by application basis.

Emissions during servicing

Equation 5 takes into account the discontinuous process of servicing. Besides component failures, such as compressor burn-out, equipment is serviced mainly when the refrigerating capacity is too low due to loss of refrigerant from fugitive emissions. Depending on the application, servicing will be done every year or every three years, or sometimes not at all during the entire lifetime such as in domestic refrigeration applications. For some applications, leaks have to be fixed during servicing and refrigerant recovery may be necessary, so the recovery efficiency has to be taken into account when the refrigerant is recovered.

$E_{servicing,\ t} = \sum_{a=1}^{\frac{d}{z}} M_{t-az} \bullet s \bullet (1-\eta_{rec})$ Equation 5

where:

$E_{servicing,t}$	=	amount of HFC emitted during system servicing in year t expressed in kilograms
D	=	average equipment lifetime expressed in years
S	=	residual charge of HFC in equipment requiring recharge expressed in percentage
M_{t-az}	=	the amount of HFC charged into the equipment either at manufacturing or after each servicing per application domain expressed in kilograms
a	=	number of recharges during the equipment lifetime d expressed in round numbers (lies in the interval [0-d/z])
Z	=	$\frac{1-\frac{s}{100}}{\frac{x}{100}}$; number of years elapsed before equipment recharge expressed in round numbers
η_{rec}	=	recovery efficiency, which is the ratio of recovered HFC referred to the HFC contained into the system

The importance of Equation 5 lies in deriving the annual refrigerant quantities needed for servicing. Knowing the annual refrigerant needs for servicing per application allows the determination of the national refrigerant market by adding the refrigerant quantities charged in new equipment.

When technical data are not available, Equation 5 could be simplified drastically and replaced by Equation 6.

$E_{servicing,\ t} = B_t \bullet (j/100)$ Equation 6

where:

$E_{servicing,t}$	=	amount of HFC emitted during system servicing in year t expressed in kilograms
B_t	=	amount of HFC banked in existing systems (per application) in year t expressed in kilograms
J	=	annual leakage rate of HFC of each application bank during servicing expressed in percentage

Emissions at disposal

The amount of refrigerant released from scrapped systems depends on the amount of refrigerant left at the time of disposal, and the portion recovered. From a technical point of view, the major part of the remaining fluid can be recovered, but recovery at end of life depends on regulations, financial incentives, and environmental concerns.

To estimate emissions at system disposal, the following calculation formula is applicable:

$$E_{disposal,\ t} = M_{(t-d)} \bullet s_d \bullet (1-\eta_{rec}) \qquad \text{Equation 7}$$

where:

$E_{disposal,t}$	=	amount of HFC emitted at system disposal in year t expressed in kilograms
$M_{(t-d)}$	=	amount of HFC initially charged into new systems installed in year (t-d) expressed in kilograms
S	=	residual charge of HFC in equipment requiring recharge expressed in percentage.
D	=	average equipment lifetime expressed in years
η_{rec}	=	Recovery efficiency, which is the ratio of recovered HFC referred to the HFC contained into the system

In estimating the amount of refrigerant initially charged into the systems (M_{t-d}), all systems charged in the country (for the domestic market) and systems imported precharged should be taken into account.

Quality assurance/quality control

In order to conduct a quality control for Tier 2 method, it is possible to compare the annual national HFC refrigerant market as declared by the chemical manufacturers or the refrigerant distributors with the annual HFC refrigerant needs as derived by the Tier 2 method. The following formula leads to this verification.

$$R_t = \sum_{6} S_{prod} \times m_t + \left(\sum_{a=1}^{\frac{d}{z}} M_{t-az} \times \left((1-s) + s(1-\eta_{rec}) \right) \right) \qquad \text{Equation 8}$$

where:

R_t	=	HFC needs in year t expressed in kilograms
S_{prod}	=	national production of equipment using HFC refrigerant for the six application domains
S	=	residual charge of HFC in equipment requiring recharge expressed in percentage
M_t	=	elementary charge of HFC in each type of equipment expressed in kilograms
$M_{t\text{-}az}$	=	the amount of HFC charged into the equipment either at manufacturing or after each servicing expressed in kilograms
A	=	number of recharges during the equipment lifetime d expressed in round numbers (lies in the interval [0-d/z])
Z	=	$\dfrac{1-\frac{s}{100}}{\frac{x}{100}}$; number of years elapsed before equipment recharge expressed in round numbers
η_{rec}	=	recovery efficiency, which is the ratio of recovered HFC in relation to the HFC contained into the system

The first Σ corresponds to the refrigerant charge of new refrigerating and air conditioning system produced in the country at the current year t including exports.

The second Σ corresponds to the refrigerant charge used for servicing.

The term $s(1-\eta_{rec})$ represents the recovered refrigerant.

The annual refrigerant market as declared by chemical manufacturers or refrigerant distributors RD is calculated by Equation 9.

$$RD_t = R_{prod\_t} - R_{\exp\_t} + R_{imp\_t} + R_{recl\_t} \qquad \text{Equation 9}$$

where

R_{prod_t}	=	quantities of HFC refrigerant production expressed in kilograms
R_{exp_t}	=	quantities of HFC refrigerant produced in the country and exported expressed in kilograms
R_{imp_t}	=	quantities of imported HFC refrigerant expressed in kilograms
R_{recl_t}		quantities of HFC refrigerant recovered and reprocessed for sale as new HFC refrigerant in kilograms

All quantities are calculated for the current year t.

Comparing R_t that is the HFC refrigerant needs as derived from the inventory method and RD_t the HFC refrigerant market as declared by refrigerant manufacturers and distributors gives a clear quality control of the inventory method, and also of the global emissions. R_t and RD_t are calculated for each HFC type.

List of Countries and country groups for refrigerant inventories

Calculations are performed independently for eighty entities: seventy countries and ten country groups.

Table A2.1 – List of countries and country groups where refrigerant inventories are performed

	Egypt		Saudi Arabia
Algeria	Estonia	Luxembourg	Singapore
Argentine	Finland	Malaysia	Slovakia
Australia	France	Malta	Slovenia
Austria	Germany	Mexico	SOUTH & EAST ASIA*
	Greece	MIDDLE EAST*	South Africa
Bangladesh	Hong kong	Morroco	SOUTH AMERICA*
Belarus	Hungary	Myanmar	South Korea
Belgium	Iceland	Netherlands	Spain
Brazil	India	New Zealand	Sweden
Bulgaria	Indonesia	Nigeria	Switzerland
Canada	Iran	Norway	Taiwan
CENTRAL AMERICA & CARIBBEAN*	Ireland	PACIFIC ISLAND COUNTRIES*	Thailand
	Israel	Pakistan	Turkey
Chile	Italy	Peru	Ukrania
China	Japan	Philippines	United Arab Emirates
Colombia	Kuwait	Poland	United Kingdom
Cyprus	Latvia	Portugal	USA
Czech Republic	Libya	Romania	Venezuela

Country groups are indicated by *, and calculations are performed for the integral values of these groups.

Table A2.2 details the composition of each country group.

Table A1.2.2 – Country groups

AFRICA	Angola, Benin, Botswana, Burkina, Burundi, Cameroon, Cape Verde, Central Africa, Chad, Comoros, Congo, Congo RD, Côte d'Ivoire, Djibouti, Eritrea, Ethiopia, Gabon, Gambia, Ghana, Guinea, Guinea Bissau, Guinea Equatorial, Kenya, Lesotho, Liberia, Madagascar, Malawi, Mali, Mauritania, Mauritius, Mozambique, Namibia, Niger, Rwanda, Sao Tome, Senegal, Seychelles, Sierra Leone, Somalia, Sudan, Swaziland, Tanzania, Togo, Tunisia, Uganda, Zambia, Zimbabwe
BALKANS	Albania, Bosnia, Croatia, Kosovo, Macedonia, Moldova, Serbia Montenegro
CENTRAL AMERICA & CARIBBEAN	Antigua, Aruba, Bahamas, Barbados, Belize, Costa Rica, Cuba, Dominica, Dominican Republic, El Salvador, Grenada, Guatemala, Haïti, Honduras, Jamaica, Nicaragua, Panama, St Vincent, Trinidad & Tobago
	Afghanistan, Armenia, Azerbaijan, Georgia, Kazakstan, Kyrgyzstan, Tajikistan, Turkmenistan, Uzbekistan
LITTLE EUROPEAN COUNTRIES	Andorra, Liechtenstein, Monaco, San Marino
MIDDLE EAST	Bahrain, Irak, Jordan, Lebanon, Oman, Palestine, Qatar, Syria, Yemen
PACIFIC ISLAND COUNTRIES	Fiji, Kiribati, Mariannes, Marshall Islands, Micronesia, Nauru, Niue, Palau, Papua New Guinea, Tonga, Vanuatu, West Samoa
SOUTH & EAST ASIA	Bhutan, Brunei, Cambodia, DP N Korea, Lao, Macao, Maldives, Mongolia, Nepal, Sri Lanka
SOUTH AMERICA	Bolivia, Ecuador, Guyana, Paraguay, Surinam, Uruguay

ANNEX 3

End of life curves

Lifetime of equipments is defined as a retirement function [KOO98].

The retirement function, also known as survival curve, is used to estimate the rate of retirement of equipments. In the linear function, no equipment retire in the first 2/3 of their average life time, and all units are retired by 4/3 of their average life time. The relation between age/average lives and appliance survival factor is shown in Figure 1.2. Expressed as equations, this function is as follows:

If age < [2/3 x (average life)] then 100% survive
If age > [2/3 x (average life)] and age < [4/3 x (average life)]
then [2 – 1.5 x (age) / (average life)] survive
If age > [4/3 x (average life)] then 0% survive

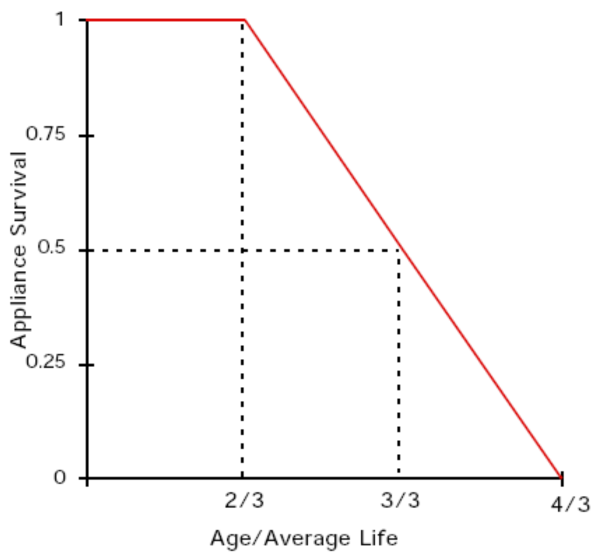

Figure 1.2: Appliance survival function.

ANNEX 4

Method of calculations of the refrigerating capacity of the food industry

A4.1 Global cooling capacity for all meats

The refrigerant inventory for the meat sub-domain has been determined using meat production figures. The FAO database gives a very detailed description of the meat demand and production for all the countries since 1961.

Cooling process for meat
The vast majority of four-footed animals are slaughtered in commercial slaughterhouses under supervision. The small portion still slaughtered on the farm has not been taken into account.

After killing, bleeding, skinning, evisceration, the meats (M1) are cooled, then either cut and packaged for frozen meat (M2) or stored in one piece if for fresh meat (M3) (see Figure A6.1).

The quantities M1, M3 and M4 are known from the FAO database. For frozen meat, the quantities are directly included in the frozen food demand, which has been analysed as one specific entity (see section 6.6).

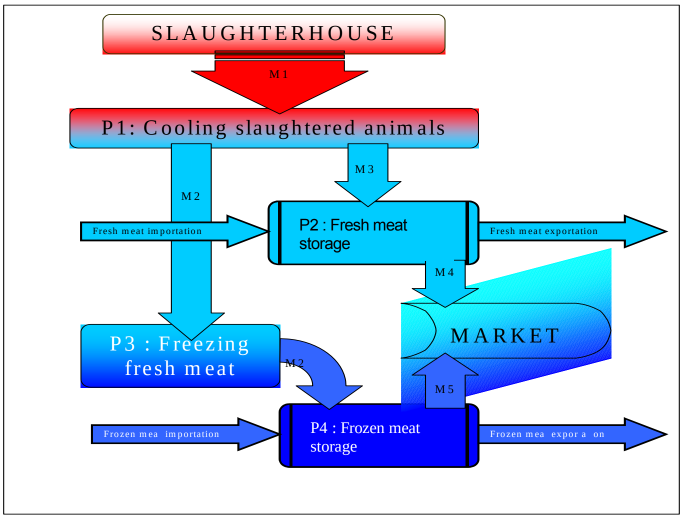

Figure A4.1- Cooling and freezing for production and storage

Based on the different meat masses, the following refrigerating capacities are defined:
- P1 and P2 are the cooling capacities for fresh meat chilling and storage, respectively
- P3 and P4 are the cooling capacities for meat freezing and frozen meat storage, respectively.

A4.1.1 Cooling Model for Beef

The cooling capacity for meat is based on the maximum needed capacity at peak load, which in fact is the design criterion for refrigerating equipment. Peak load occurs at the beginning of meat chilling, just after the slaughter when carcasses have their highest temperature.

Figure A4.2 shows the exponential curve of beef carcass temperature drop. The chill rate is $\Delta\theta$ / Δt, but the peak load corresponds to the maximum slope α . ($\Delta\theta$ / Δt), which is required for sanitary issues.

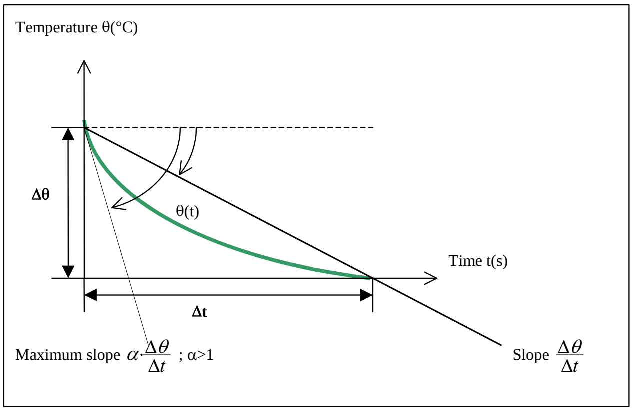

Figure A4.2 - Cooling model of beef

From Figure A4.2 the maximum product cooling capacity P_{meat} can be calculated, i.e., by using the equation below.

$$P_{meat} = \frac{\alpha \cdot M \cdot c \cdot \Delta\theta}{\Delta t} \quad \text{(A6.1)}$$

where:
P_{meat} meat maximum cooling capacity (kW)
α coefficient for the determination of the maximum rate of chill (see Figure A4.2),
c average heat capacity (kJ/kg K)
$\Delta\theta$ / Δt temperature difference for a given time difference (K/s).

The water evaporated from the beef carcass condenses and freezes on the evaporator coils requiring additional capacity due to frost formation. The rate of water evaporation is proportional to the rate of meat being cooled; and the corresponding cooling capacity can be calculated by the equation below:

$$P_{frost} = \beta \cdot \frac{\alpha \cdot M}{\Delta t} \cdot H_{sol} \quad \text{(A4.2)}$$

where :
$\frac{\alpha \cdot M}{\Delta t}$ maximum rate of chilled meat
H_{sol} = ice heat of solidification = 335 kJ/kg
β<1, part of water lost from the chilled meat

Miscellaneous loads such as conveyors, air infiltration, personnel, fan motors, lights, and equipment heat losses need to be taken into account. The latter loads are proportional to the maximum cooling capacity M.

$$P_{misc} = \gamma \cdot M \quad \text{(A4.3)}$$

where:
γ(W/kg) is the factor for maximum miscellaneous losses.
The total cooling capacity is :

$$P_{tot} = P_{meat} + P_{frost} + P_{misc}$$

$$\Rightarrow P_{tot} = \frac{\alpha \cdot M \cdot c \cdot \Delta\theta}{\Delta t} + \beta \cdot \frac{\alpha \cdot M}{\Delta t} \cdot H_{fusion} + \gamma \cdot M$$

Thus the cooling capacity per unit of mass is:

$$p = \frac{P_{tot}}{M} = \frac{\alpha \cdot c \cdot \Delta\theta}{\Delta t} + \beta \cdot \frac{\alpha}{\Delta t} \cdot H_{fusion} + \gamma \qquad \text{(A4.4)}$$

where

p specific cooling capacity (W/kg).

♦ **Coefficient values for chill and holding coolers**

Chilling of the beef carcass is performed in two different coolers. First the rapid cooling is performed in the chill cooler, and then cooling takes place at a reduced rate in the holding cooler. The density of the carcass is significantly lower in the holding cooler (45kg/m^3) than in the chill cooler (60kg/m^3). So, if referred to the mass of a chilled carcass, the ratios of miscellaneous heat losses are different. Taking into account that for large storage the volumetric G factor amounts to 30 W/m^3, this leads to

γ_1 = 0.5 W/kg for chill cooler, and

γ_2 = 0.677 W/kg for holding cooler.

♦ **Coefficient for chill cooler**

Dressed beefs are split into half carcasses (the average half carcass mass is around 150 kg) and the average specific heat c is around 3.14 kJ/kg K [ASH98].

α is determined according to the curve of average carcass temperature of meat cooling versus time (Figure 4.3),

Δt = 20 h and $\Delta\theta$ = 28°C, the first 4 hours (= 0.2 Δt) the temperature decreases by 11.2 K (= 0.4$\Delta\theta$) therefore α = 2.

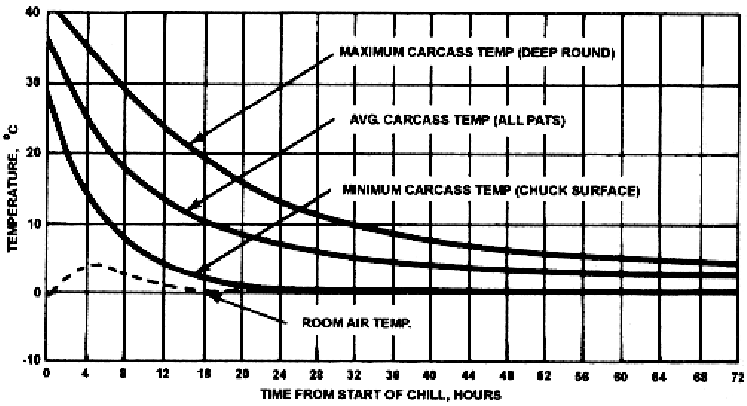

Figure A4.3 - Beef chilling curves [ASH98]

β = 0.03 represents typically 3% of the chilled mass [ASH98] ; γ = 0.5 W/kg (see above).

In summary, for Δt = 20h = 72 000 seconds and $\Delta\theta$ = 28°C , α = 2, β = 0.03, γ = 0.5.

And p_1 = 3.2214 W/kg (**ratio 1**)

♦ **Holding cooler coefficient**

Equation (1) is applicable here. The temperature drop is lower for a longer time, and the water evaporation speed is low, leading to the following coefficient.

(α = 1.2, β = 0.0035 = 0.35%, γ = 0.667 W/kg, $\Delta\theta$ / Δt = 4.17 K/24h)

$p_2 = 0.866\ W/kg$ (**ratio 2**)

A6.1.2 Cooling capacity for ancillaries

Besides cooling, freezing, storing, many other operations are needed in meat processes, like cutting, packing, examining, expedition…. The cooling needs here are proportional to the size of the slaughterhouse and so also proportional to the annual capacity of meat being processed.

Based on a detailed case study of a large French slaughterhouse [CLO96], the typical ancillary cooling capacities are presented in Table A4.1.

Table A4.1 – Ancillary cooling capacities

Designation	Nb	Unit Capacity (kW)
Offal process room		16
Offal refrigeration	2	27
White offal storage		26.5
Wastes	3	16.5
Blood tank		16
Hides	3	13.5
Exam		7
Pre-check room		10.5
Check room		10.5
Input room		7
Food for animals		4
Complement storage	2	22
Large part cutting	3	13
Expeditions	3	16.5
Storage before cuts	2	16.5
Offal process room (2)		13
Cutting room 1		46.5
Cutting room 2		13
Offal storage room		13
Offal packing room		16.5
Packaging		14
Vacuum storage		14
Packaging consignment		12.5
Consignment	3	12.5
Passageways	3	4
Total		599 kW

30,000 tons are processed in this slaughterhouse annually. From this case study the ancillary ratio is fixed.

$$p_3 = \frac{599}{30000} = 0.02\ kW/at^{*} \qquad \textbf{(ratio 3)}$$

** at : is the meat annual production in tons*

p_3 is calculated and it is based on the total quantity of processed meat, not taking into account the characteristics of the different cooling rooms, and this ratio is therefore used for the annual meat production.

A4.1.3 Generalization to all types of meat

Meat cooling, whatever the type of meat, is very similar due to the sanitary specifications. The carcass shall be cooled down as quickly as possible, the limit is linked to the meat hardness.

Due to physiological changes after slaughtering, heat is generated inside the body and tends to increase its temperature to around 41°C when the carcass enters the chilling cooler.

HACCP (Hazard Analysis and Critical Control Point) [ASH98] recommends that red meat carcasses be chilled to 5°C within 24 hrs, and that this temperature be maintained during storage, shipping, and product display.

Heat capacities of meats vary with the percentage of fat and moisture, but an average heat capacity 3,1 kJ/(kg. K) is used for calculations of all meats [ASH98].

Meats are divided in three groups according to the carcass size that influences the cooling time:

- first group with an average mass per carcass of 150 kg, e.g. beef, veal, horse
- second group with an average mass per carcass of 60 kg, e.g. pig, mutton, lamb, goat …
- third group with an average mass per poultry of 4 kg, e.g. turkey, chicken, duck, goose….

The beef cooling model is used as a general model and the different coefficients for each group are given in Table A4.2.

Table A4.2 - Physical properties and ratios for cooling capacity calculations

	Group I	**Group II**	**Group III**
Parameters/ratios	Beef, veal, horse meat	Goat, Mutton, lamb, Pig	Chicken, duck, goose, birds, rabbit, turkey
α	2	1.5**	1.2**
C (J/(kg.K))	3,140	3,140	3,140
$\Delta\theta$ (K)	30	30	30
Δt (h)	20	12**	6**
β	0.03	0.03	0.03
γ(W/kg)	0.5	0.5**	0.5**
Fresh meat cooling ratio (W/kg)	3.326	4.05	5.733
Fresh meat storage ratio (W/kg)	0.866	0.866**	0.866**
Ancillary cooling capacity kW/at*	0.02	0.02**	0.02**

*at = annual ton **Estimation

A4.1.4 Calculation of the national installed cooling capacity for meat

The FAO web-site presents statistics on the annual production, imports and exports of meat. Production figures relate to animals slaughtered within national boundaries regardless of their origin. These figures are used as inputs in the country database for all countries, years and types of meats. The format is a uniform table of countries by year and it is adopted for all types of meat. Table A4.3 shows other constants needed to estimate the installed cooling capacity.

Table A4.3 –Working time assumptions

	Constants
Slaughterhouse coefficient of use	0.8
Warehouse coefficient of use	0.6
Residence time in the warehouse (days)	2
Working days per year (slaughterhouse)	300
Working days per year (warehouse)	360

The national installed cooling capacity is calculated based on the national demand of all countries and for all types of meat.

The installed cooling capacity takes into account three terms:

- meat cooling
- meat storage, and
- ancillary cooling capacities.

♦ **National fresh meat cooling capacity**

The national installed cooling capacity for fresh meat cooling is calculated by the following equation:

$$P_1 = \frac{M_p \cdot p_1}{\tau.\lambda}$$

Where:

P_1 national installed cooling capacity for fresh meat (kW)
M_p annual meat production obtained from the FAO database per country (annual tons)
p_1 ratio of fresh meat cooling (W/kg) (see Table A4.2)
τ working days per year (slaughterhouse) (see Table A4.3)
λ coefficient of use of the slaughterhouse (see Table A4.3).

♦ **National cooling capacity for fresh meat storage**

The national installed cooling capacity for fresh meat storage is calculated by the following equation:

$$P_2 = \frac{M_p \cdot p_2 . \sigma'}{\tau' . \lambda'}$$

Where:

P_2 national installed cooling capacity for fresh meat storage (kW)
M_p annual meat production obtained from the FAO database per country (annual tons)
p_2 ratio of fresh meat storage (W/kg) (see Table A4.2)
σ' Storage residence time (day)
τ' working days per year of the warehouse (see Table A4.3)
λ' coefficient of use of the warehouse (see Table A4.3).

- **National installed cooling capacity for ancillaries**

The national installed cooling capacity for ancillaries is calculated by the following equation:

$$P_3 = \frac{M_p \cdot p_3 \cdot}{\lambda}$$

Where:

P_3 national installed cooling capacity for fresh meat storage (kW)
M_p annual meat production obtained from FAO database per country (annual tons)
p_3 capacity ratio of ancillaries (kW/annual tons) (see Table 4.2)
λ coefficient of use of the factory (see Table 4.3)

- **Verification with the French Inventory report [PAL02]**

After aggregation of all installed parcs for all groups of meat, the total installed parc for France is listed in Table A4.4

Table A4.4 - France refrigerating parc for meat industry

	1991	**1992**	**1993**	**1994**	**1995**	**1996**	**1997**	**1998**	**1999**	**2000**
Total installed cooling capacity (MW)	341.84	351.69	355.14	367.39	378.44	389.98	393.25	398.32	392.94	384.99

In Inventory Reports for France issued previously, another method was used to determine the installed cooling capacity.

If the energy consumption in refrigeration for the meat industry is known (i.e., 1228 GWh per year), assuming a COP of 2, a factory working time of 300 days per year and 16 hours per day, the calculated installed capacity is 512 MW for the year 1998. Referred to the installed cooling capacity listed in Table 4.4, the error made is 22.15%.

A4.2 Global cooling capacity for dairy industry

A4.2.1 Calculation of installed cooling capacity

The refrigerant inventory for the dairy sub domain is determined using the dairy production and sales. The FAO database gives a very detailed description of the dairy demand and production for all the countries since 1961.

Frozen dairy products are not considered in this section, they are aggregated in the frozen product domain.

Figure A4.4 gives the link between different figures available in FAO Database for dairy process.

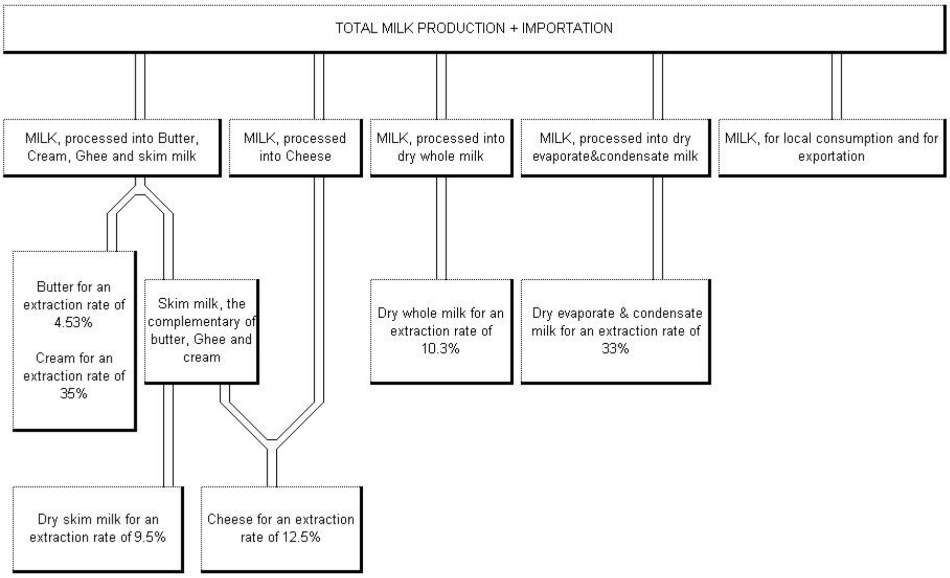

Figure A4.4 - The FAO link between different dairy processes

Milk undergoes the cooling in the farm, is transported by insulated trucks, and is treated in the factory where it will be processed into different dairy products.
The major refrigerated process for milk are:

- farm refrigeration (milk tank)
- bacteria treatment (pasteurization, UHT…)
- fermentation (depending on dairy product).

A4.2.2 Milk tank installed cooling capacity

For the milk cooling at the farm, the following rules for cooling are applied:

- cooling from 35 to 5°C in 2 hours
- no frosted milk in the tank, even partial
- allowable temperature increase equal to 5K if a second milking is added to the milk tank.

To avoid every risk of the milk temperature decreasing below the frosting point, 4°C is the lowest controlled temperature for direct expansion milk tanks (which are the most widespread). Some milk tanks use ice accumulation technology to maintain a lower temperature, between 0 and +1°C. The above mentioned two types of milk tanks show similar performances. For both types, the law for cooling can be considered as linear.

At 4°C, the milk cannot be conserved in milk tanks at the farm longer than two days, because of bacteria proliferation.

The cooling model for a milk tank is similar to the cooling model for meat:

$$p_{milk} = \frac{\alpha \cdot c \cdot \Delta\theta}{\Delta t} + \gamma \qquad \text{(A4.5)}$$

with :

$\Delta\theta$ = 30°C pour Δt = 2h = 7200 sec

α = 1 (temperature curve is linear cause of cooling time is short in respect of temperature drop)

c = 4 kJ/(kg.K) [ASH98].

Calculation of miscellaneous heat losses

γ has been evaluated taking into account the insulation of typical milk tanks. Calculations show that γ = 0.033 W/kg, which is negligible and it is therefore not taken into account in the formula. Based on those assumptions, the milk capacity ratio p_{milk} can be derived as follows:

p_{milk} = 16,7 W/kg (**ratio 4**)

Milk capacity ratio verification

For France the average milk tank volume installed amounts to 3000 L. Data sheets of a standard milk tank are obtained from literature [INTVMZ]. This reference gives the nominal volume of a typical direct expansion milk tank and its installed compressor power (2500 l; 15.47 kW). Assuming a COP of 2.5, the cooling capacity amounts to 38.67 kW. The milk capacity ratio calculated with the data mentioned yields a figure of 15.47 W/kg. The difference with the milk capacity ratio calculated using (equation 4.5) is about 8%, which is acceptable.

The milk capacity ratio will therefore be calculated using a ratio of 4.

Installed cooling capacity for Average National Daily Milk Production (ANDMP)

To establish the world installed cooling capacity for milk tanks, it is necessary to determine the Average National Daily Milk Production. Knowing the annual milk production from the FAO, (i) with a maximum residence time of two days, (ii) in which a maximum of four milkings are considered (two milkings a day), and (iii) a maximum filling ratio of 0.7 of the milk tank, (which is an average value taking into account the annual variation from 0.6 to 0.8), the Average National Daily Milk Production for a given country is calculated as follows:

$$M_{ANDMP} = \frac{M_p . \sigma}{n . \tau . \rho}$$

where:

M_{ANDMP} average national daily milk production (ANDMP) (kg)

M_p annual milk production obtained from FAO database (annual kg)

σ maximum residence time (days)

n number of milkings in a milk tank

τ number of days per year

ρ filling ratio

Parameters	
Days per year	360
Max residence time (day)	2
Cooling ratio W/kg	16.7
Number of milkings	4
Filling ratio	0.7

Table 4.5 – ANDMP parameters

The national installed cooling capacity for milk tanks is then:

$$P_{milk} = M_{ANDMP} \cdot p_{milk}$$

A4.2.3 Milk bacterial process and cooling

For pathogenic bacteria elimination, several milk processes are applied: pasteurisation, UHT…. This process consists of:

- heating the milk;
- maintaining it at high temperature during the necessary time for complete pathogenic bacteria elimination,
- cooling it to 4°C.

Refrigeration is only related to the milk cooling from 35°C to 4°C, since the milk cooling from temperatures higher than 35°C is done either by cold water or, better, by regeneration in a milk / milk heat exchanger.

Several cooling techniques are used, chilled water being the most widespread for large milk facilities.

Pasteurization and cooling take place in the same heat exchanger, which includes three zones:

- a heating zone for the pasteurization,
- a central zone where the homogenized cold milk is heated by the counter current of pasteurized hot milk (regeneration process),
- a cooling zone where the milk is cooled by chilled water.

To determine the cooling capacity for the national pasteurization, the following formula is used:

$$P_{past} = \eta \times (\frac{M_p}{\tau'}) \times Cp \times \Delta\theta$$

$$P_2 = \frac{\eta}{\lambda} \times (\frac{M_p}{\tau'}) \times Cp \times \Delta\theta = \frac{M_p \times p_{past}}{\lambda} \quad (4.6)$$

Where

P_{past} national cooling capacity for pasteurization (kW)
η heat loss factor
λ coefficient of use
M_p milk annual production obtained from FAO database (annual tons)
τ' factory working time in seconds
Cp heat capacity of milk
$\Delta\theta$ temperature drop (°C)
$\frac{M_p}{\tau'}$ average mass flow rate of the factories

Table A4.6 - Cooling parameters after pasteurization

Factory working days per year (days)	300
Factory working hours per year (hours)	16
Temperature drop (°C)	31
Cp (kJ/kg.K)	4
Heat loss factor η (Indirect systems + Other installations)	1.4
Coefficient of use λ (real mass flow rate / dimensional mass flow rate)	0.8

$\Rightarrow p_{past}$= 0.01256 W/annual kg.

A4.2.4 Fermentation and cooling

Some dairy products need to be stored in refrigerated rooms for fermentation, but the residence time differs from one product to another and from one country to another. Table A4.7 lists different chosen parameters for the calculation of cooling in fermentation rooms.

The national cooling capacity for fermentation rooms is calculated as follows:

$$P_{ferm} = \frac{M_p . p_{ferm} . \sigma}{\tau . \phi}$$

where

P_{ferm} national installed capacity for fermentation rooms (kW)

M_p annual dairy product obtained from FAO database (annual tons)

p_{ferm} volumetric cooling ratio for fermentation (W/m^3)

ϕ minimum storage ratio of products in 1 m^3 of warehouse (kg/m^3)

σ staying delay in factory warehouse (day)

τ working days per year of factory warehouse.

Table A4.7 – Fermentation and storage parameters

	Butter and Ghee	Cheese	Cream
Temperature (°C)	5	5	5
Cooling ratio (W/m^3)	30	30	30
Storing ratio (kg/m^3)	300	500	300
Residence time (days)	5	30	5

A4.3 Global cooling capacity for wine and beers

The FAO database includes global wine and beer production figures. In order to derive the installed cooling capacities from the wine and beer production figures, two cooling models have been developed.

A4.3.1 Wine cooling model

The wine cooling model is based on a detailed case study of a winery where the cooling capacities and production are known. From [CLO96], Table A4.8 has been established using the annual production figure of 75,000 hl.

Table A4.8 - Cooling data of the typical case

Cooling stage	Cooling capacity (kW)	Product capacity (hl)	Ratio: cooling capacity/product capacity (W/kg)	Ratio: annual production/product capacity	Ratio: cooling capacity/annual production (W/annual kg)
Wine-making process	70	25000	0.028	3	0.0093
Tartaric stabilization ultra-cooling	50	25000	0.020	3	0.0067
Storage	175	75000	0.0233	1	0.0233

From Table A6.8 the total cooling ratio of wine can be derived as 0.03933 W/annual kg.

The average time for wine-making process is one week for red wine and 15 days for white wine. The average time for tartaric stabilisation is 15 days [CLO96]. For the case under study, the storage is air-conditioned because during summer the ambient temperature is very high and the storage temperature must be kept under 21°C. This case is not applicable to all wineries, therefore the storage cooling ratio has been multiplied by a factor α less than 1, α = 0.4 (40% of wineries use air conditioning in their wine storage).

P_{wine} = 0.03 W/annual kg (**ratio 5**)

A4.3.2 Beer cooling model

Wort cooling

The following formula is used for the national wort cooling capacity: $P_{wort} = \frac{\eta}{\lambda} \times (\frac{M_p}{\tau'}) \times Cp \times \Delta\theta$

Where

P_{wort} national wort cooling installed capacity (kW)

η losses multiplier factor (η = 1.4)

λ coefficient of use (real flow rate/ dimensioned flow rate, λ=0.8)

M_p beer annual production obtained from FAO database (annual tons)

τ' factory working time in seconds (the factory works 300 days/yr and 16 hrs/day)

Cp wort heat capacity (Cp = 4 kJ/(kg.K) [ASH98])

$\Delta\theta$ temperature drop (°C) ($\Delta\theta$ = 31°C [ASH98])

$\frac{M_p}{\tau'}$ average wort mass flow rate.

$p_{wort} = 0.01256$W/annual kg (**ratio 6**)

Fermentation
Beer ratio fermentation amounts to 0.0033 W/annual kg which has been taken from [ASH98].

p_{ferm} = 0.033 W/annual kg (**ratio 7**)

A4.4 Global cooling capacity for flake ice for fresh fish conservation

The cooling capacities applied and the refrigerant types used on board of fishery vessels are taken into account in considerations of the refrigerated vessel fleet. But, once the fish is delivered for sale, fresh fish conservation is essentially performed on flake ice.

The ratio of ice used for fish conservation, IFR, is:

IFR = mass of ice / mass of fish = 0.5 (0.25 for cooling and 0.25 for lost [RGF02]).

The National Fresh Fish Annual Production (NFFAP) data are coming from the FAO database.

The capacity ratio for producing flake ice is:
Ice Cooling Capacity Ratio, ICCR = 6.95 W/kg [ASH98].

Average number of catches (catching days) per year: 300 catches per year.

The national installed cooling capacity for production of flake ice for fish conservation is calculated as follows:

$$P_{fish} = \frac{NFFAP}{\tau'} xICCRxIFR \text{ (W)}$$

where
τ' is the number of catching days per year (300)
NFFAP (kg/yr)
ICCR (W/kg).

A4.5 Global cooling capacity for frozen food

A4.5.1 Frozen food production

Annual frozen food production is not yet available from the FAO Database, but export and import data are available, and they allow to establish the world frozen food production using the Kaminsky ratios [KAM95] for annual consumption of frozen food per capita as presented in Table A4.9, and using the equation:

Production = Consumption + Export – Import

Table A4.9 – Annual consumption of frozen food per inhabitant [KAM 95]

	USA, Denmark	**UK, France, Sweden**	**Germany, Switzerland**	**Norway, Austria, Belgium, Finland, Spain, Australia, Japan, The Netherlands**	**Italy, Hungary,**
Annual consumption/	> 40	30 – 40	20 – 30		< 10

For each group presented in Table A4.9, linear interpolation with the mean GDP of the corresponding country allows the determination of the annual consumption per capita.

In the FAO database import and export figures are available for: Ice cream, Potato frozen, Sweet corn frozen, Cephalopods Frozen, Crustaceans Frozen, Demersal Frozen Fillets, Demersal Frozen Whole, Fish fillet chilled frozen, Fish Frozen Whole Fillet, Fish shellfish frozen, Freshwater Frozen Whole, Freshwater Frozen Fillets, Marine nes Frozen Fillet, Marine nes Frozen Whole, Mollusc Frozen, Pelagic Frozen Fillets, Pelagic Frozen Whole.

Based on previous calculations, the world frozen food production as determined is presented in Table A4.10.

Table A4.10 - World frozen food production

Year	**1980**	**1981**	**1982**	**1983**	**1984**	**1985**	**1986**	**1987**	**1988**	**1989**	**1990**
Production (10^6 t)	26.51	27.28	27.38	27.34	27.56	27.04	27.42	28.35	28.13	29.71	29.36
Year	**1991**	**1992**	**1993**	**1994**	**1995**	**1996**	**1997**	**1998**	**1999**	**2000**	
Production	32.14	32.78	32.90	35.93	37.37	36.27	38.38	39.80	38.40	38.54	

[KAM95] estimates the world frozen food production by the beginning of 1990 at a level of 30 million tons. The calculated value for this year is 7% higher (table A4.10).

A4.5.2 Frozen food cooling model

Based on data from a manufacturer of a blast freezer [SBL], it can be given that the freezing ratio per kg of frozen food per hour is of 121.472 W/(kg h). This value has been used for all types of food.

The frozen food production is considered as continuous production during 16 hours per day and 300 days per year. The factory use coefficient equals 0.8. The national installed capacity for frozen food is calculated according to Figure A4.5.

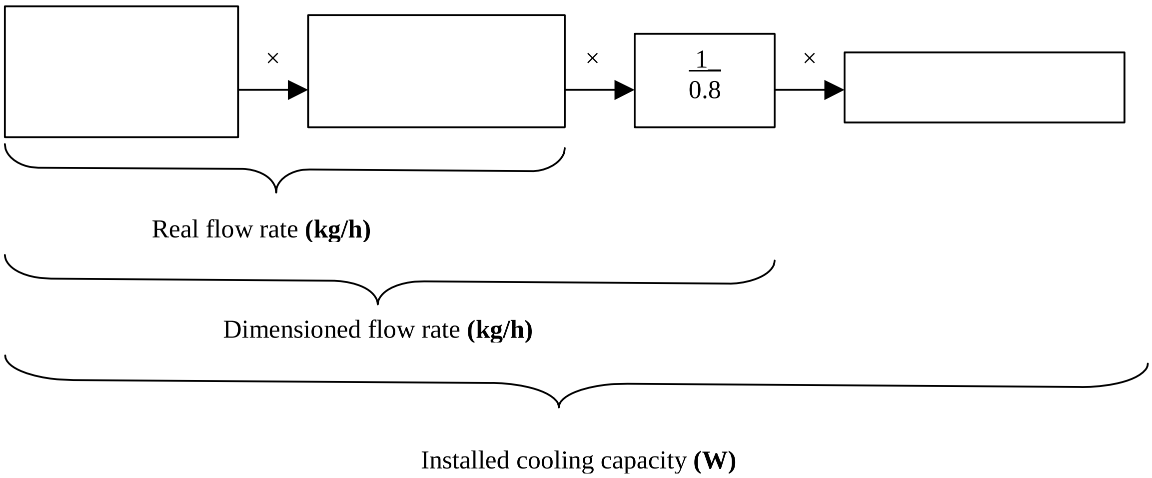

Figure A4.5 – Flow sheet for national installed capacity for frozen food.

The national installed capacity ratio p_{frozen_food} is

$$p_{frozen\_food} = 0.0316 \text{ W/annual kg}$$

For the factory storage, the same parameters are used.

A4.6 Installed cooling capacity for cold storage

In this section cold storage means all cold storage except the storage in food processing facilities. The refrigerated volumes correspond to low and medium temperature storage, specialised and multipurpose cold stores and fruit packing stations. The cold storage volume estimates by country are based on ratios that have been elaborated on for different developed countries [KAM95, GLO92-93]. Based upon these ratios additional calculations have been performed in order to refer the cold storage volume to the GDP. Figure A6.6 indicates the evolution of the cold storage referred to the GDP as a function of time (1930-2000).

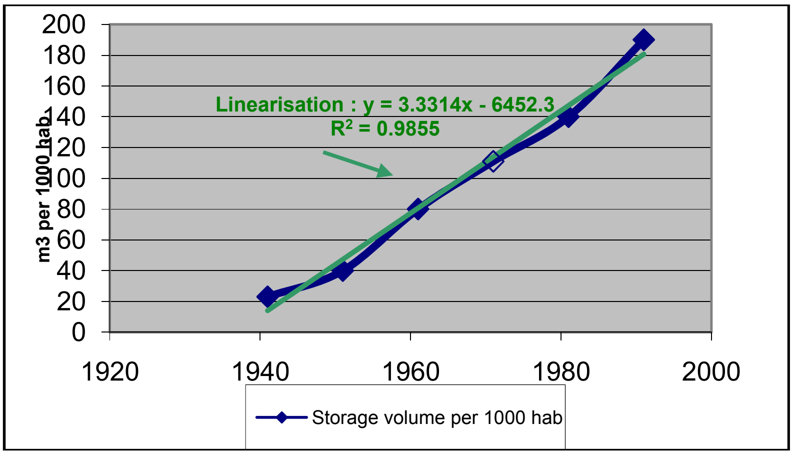

Figure A4.6 – The US typical storage volume example [KAM95]

A saturated linear extrapolation with the US storage volume per capita and the mean GDP allows the establishment of the storage volume per capita for each country (Figure A6.7). The extrapolation with the US storage increased volume per year and the GDP standard deviation makes it possible to establish the storage-increased volume for each country. Extrapolations have been done for year 1961 and have been projected to the year 1999 using the storage-increased volume per year and the population figures.

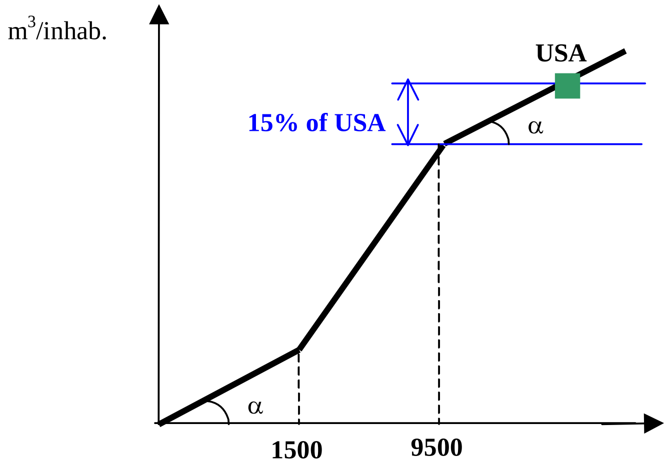

Figure A4.7 – Saturated linear extrapolation based on USA typical example

Low temperature cooling capacity over total cooling capacity is calculated taking into account the frozen food consumption per inhabitant.

Medium and low temperatures cooling capacities referred to the cold storage volume are known from the report [ADE00].

References to Annex 5

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ANNEX 5

Annex to some figures of Part I of the report
(Figure referred to indicated in parentheses after table number)

Table A5.1 (Figure 1.10) Total demand in CS – Lower threshold (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	264	855	0	0	1,118
1991	265	861	0	0	1,126
1992	264	915	0	0	1,179
1993	261	995	0	0	1,256
1994	249	1,073	0	0	1,322
1995	175	1,015	0	0	1,190
1996	131	1,203	0	0	1,334
1997	100	1,243	0	0	1,343
1998	76	1,272	0	0	1,347
1999	55	1,275	24	0	1,354
2000	37	1,273	54	0	1,364
2001	23	1,233	116	0	1,372
2002	13	1,126	167	0	1,306
2003	5	1,019	224	0	1,248
2004	0	893	240	0	1,132

Table A5.1 (Figure 1.11) Total demand in CS – Higher threshold (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	264	855	0	0	1,118
1991	265	861	0	0	1,126
1992	264	915	0	0	1,179
1993	261	995	0	0	1,256
1994	249	1,073	0	0	1,322
1995	175	1,015	0	0	1,190
1996	137	1,239	0	0	1,376
1997	110	1,324	0	0	1,435
1998	87	1,403	0	0	1,490
1999	66	1,461	25	0	1,553
2000	48	1,523	59	0	1,629
2001	32	1,546	127	0	1,705
2002	18	1,500	191	0	1,709
2003	8	1,453	268	0	1,728
2004	0	1,374	305	0	1,679

Table A5.2 (Figure 1.12) Refrigerant demand of condensing unit (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	74	579	0	0	653
1991	57	565	0	0	622
1992	53	551	0	0	604
1993	49	544	0	0	593
1994	45	568	0	0	613
1995	37	609	0	0	645
1996	29	621	0	0	650
1997	22	626	0	0	648
1998	16	619	0	0	636
1999	11	611	0	0	623
2000	7	624	13	0	644
2001	4	604	23	0	632
2002	2	568	44	0	614
2003	1	535	71	0	607
2004	0	510	94	0	604

Table A5.3 (Figure 1.13) Refrigerant demand of stand-alone equipment (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	27	11	0	0	38
1991	26	10	0	0	36
1992	27	9	0	0	36
1993	22	9	5	0	36
1994	16	10	13	0	40
1995	3	11	29	0	43
1996	3	11	31	0	45
1997	3	12	31	0	45
1998	2	11	32	0	45
1999	2	8	35	0	45
2000	2	7	40	0	48
2001	2	5	41	0	48
2002	1	3	42	0	46
2003	1	2	43	0	46
2004	1	1	44	0	46

Table A5.4 (Figure 1.14) Total demand in commercial refrigeration (CS, CU and standalone) – Lower threshold (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	364	1,445	0	0	1,809
1991	348	1,436	0	0	1,784
1992	344	1,476	0	0	1,819
1993	333	1,548	5	0	1,885
1994	310	1,651	13	0	1,974
1995	214	1,634	29	0	1,878
1996	162	1,836	31	0	2,028
1997	125	1,881	31	0	2,036
1998	94	1,902	32	0	2,028
1999	68	1,894	59	0	2,021
2000	46	1,903	108	0	2,057
2001	29	1,842	180	0	2,051
2002	16	1,697	253	0	1,966
2003	7	1,556	338	0	1,901
2004	1	1,404	378	0	1,783

Table A5.5 (Figure 1.15) Total demand in commercial refrigeration (CS, CU and standalone) – Higher threshold (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	364	1,445	0	0	1,809
1991	348	1,436	0	0	1,784
1992	344	1,476	0	0	1,819
1993	333	1,548	5	0	1,885
1994	310	1,651	13	0	1,974
1995	214	1,634	29	0	1,878
1996	168	1,871	31	0	2,070
1997	135	1,962	31	0	2,128
1998	106	2,033	32	0	2,171
1999	80	2,080	60	0	2,220
2000	57	2,153	112	0	2,322
2001	37	2,156	191	0	2,384
2002	22	2,071	277	0	2,370
2003	10	1,990	382	0	2,381
2004	1	1,885	443	0	2,330

Table A5.6 (Figure 1.16) Distribution of the demand by refrigeration equipment technology I grocery supermarkets.

	Centralized systems	*Condensing unit*	*Standalone*
Metric tonnes	1,679	604	46
Percent of total	72%	26%	2%

Table A5.7 (Figure 1.17) Refrigerant bank in centralized systems (grocery supermarkets) (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	523	1,727	0	0	2,250
1991	539	1,782	0	0	2,321
1992	550	1,848	0	0	2,398
1993	556	1,941	0	0	2,496
1994	550	2,052	0	0	2,602
1995	475	2,170	0	0	2,645
1996	396	2,402	0	0	2,798
1997	320	2,630	0	0	2,950
1998	253	2,852	0	0	3,104
1999	193	3,057	17	0	3,266
2000	139	3,247	52	0	3,438
2001	91	3,407	125	0	3,623
2002	53	3,495	224	0	3,773
2003	22	3,526	351	0	3,900
2004	0	3,491	474	0	3,964

Table A5.8 (Figure 1.18) Refrigerant bank in condensing unit systems (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	351	2,361	0	0	2,712
1991	329	2,356	0	0	2,685
1992	307	2,337	0	0	2,644
1993	284	2,310	0	0	2,594
1994	260	2,299	0	0	2,560
1995	212	2,319	0	0	2,531
1996	168	2,339	0	0	2,507
1997	128	2,355	0	0	2,483
1998	94	2,359	0	0	2,453
1999	65	2,354	0	0	2,419
2000	42	2,358	10	0	2,411
2001	25	2,352	26	0	2,403
2002	13	2,324	56	0	2,393
2003	5	2,280	102	0	2,387
2004	0	2,227	160	0	2,387

Table A5.9 (Figure 1.19) Refrigerant bank in standalone equipment (metric tonnes).

	CFC	HCFC	HFC	Others	Total
1990	272	146	0	0	418
1991	278	144	0	0	422
1992	284	141	0	0	425
1993	284	138	4	0	426
1994	279	136	15	0	430
1995	261	133	40	0	434
1996	242	131	66	0	440
1997	223	129	92	0	444
1998	203	127	118	0	448
1999	183	121	147	0	451
2000	162	116	180	0	457
2001	141	109	213	0	463
2002	120	101	247	0	468
2003	100	92	282	0	474
2004	81	84	315	0	480

Table A5.10 (Figure 1.20) Refrigerant bank in commercial refrigeration (metric tonnes).

	CFC	HCFC	HFC	Others	Total
1990	1,145	4,234	0	0	5,380
1991	1,146	4,281	0	0	5,428
1992	1,141	4,326	0	0	5,467
1993	1,124	4,389	4	0	5,517
1994	1,090	4,487	15	0	5,591
1995	948	4,622	40	0	5,610
1996	807	4,873	66	0	5,745
1997	671	5,114	92	0	5,877
1998	550	5,338	118	0	6,006
1999	441	5,532	164	0	6,136
2000	343	5,721	242	0	6,306
2001	257	5,868	364	0	6,489
2002	186	5,920	527	0	6,634
2003	127	5,899	735	0	6,761
2004	81	5,802	949	0	6,832

Table A5.11 (Figure 1.21) Distribution of the bank by refrigeration technology equipment in grocery supermarkets.

	Centralised systems	Condensing unit	Standalone
Metric tonnes	3,964	2,387	480
Percent of total	58%	35%	7%

Table A5.12 (Figure 1.22) Total emissions in CS – Lower threshold (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	220	701	0	0	921
1991	226	732	0	0	958
1992	230	769	0	0	999
1993	233	816	0	0	1,049
1994	233	869	0	0	1,101
1995	228	806	0	0	1,033
1996	188	857	0	0	1,044
1997	153	890	0	0	1,043
1998	120	914	0	0	1,034
1999	92	925	5	0	1,022
2000	69	927	14	0	1,010
2001	49	912	33	0	993
2002	29	858	54	0	940
2003	14	787	78	0	879
2004	5	723	96	0	825

Table A5.13 (Figure 1.23) Total emissions in CS – Higher threshold (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	220	701	0	0	921
1991	226	732	0	0	958
1992	230	769	0	0	999
1993	233	816	0	0	1,049
1994	233	869	0	0	1,101
1995	228	806	0	0	1,033
1996	195	892	0	0	1,087
1997	165	970	0	0	1,135
1998	135	1,044	0	0	1,179
1999	108	1,109	6	0	1,223
2000	84	1,173	18	0	1,275
2001	63	1,219	43	0	1,325
2002	43	1,244	75	0	1,362
2003	27	1,251	117	0	1,395
2004	13	1,237	157	0	1,407

Table A5.14 (Figure 1.24) Refrigerant emissions in condensing unit systems (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	76	518	0	0	594
1991	73	521	0	0	594
1992	70	523	0	0	593
1993	68	524	0	0	591
1994	65	529	0	0	594
1995	75	525	0	0	600
1996	64	535	0	0	599
1997	52	534	0	0	586
1998	40	529	0	0	569
1999	31	530	0	0	562
2000	22	522	2	0	546
2001	15	516	5	0	536
2002	9	497	10	0	516
2003	5	484	18	0	508
2004	2	471	28	0	502

Table A5.15 (Figure 1.25) Refrigerant emissions in standalone equipment (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	17	11	0	0	28
1991	18	11	0	0	29
1992	19	11	0	0	30
1993	20	11	0	0	31
1994	20	12	1	0	33
1995	21	12	2	0	34
1996	21	12	2	0	35
1997	22	12	2	0	36
1998	22	12	3	0	37
1999	23	12	3	0	38
2000	23	12	4	0	38
2001	22	11	4	0	38
2002	22	11	4	0	37
2003	21	10	5	0	36
2004	20	9	7	0	36

Table A5.16 (Figure 1.26) Total emissions in commercial refrigeration (CS, CU and standalone) – Lower threshold (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	313	1,230	0	0	1,543
1991	317	1,264	0	0	1,581
1992	319	1,303	0	0	1,622
1993	320	1,351	0	0	1,671
1994	318	1,410	1	0	1,728
1995	323	1,343	2	0	1,668
1996	272	1,404	2	0	1,679
1997	227	1,437	2	0	1,666
1998	182	1,456	3	0	1,640
1999	146	1,467	8	0	1,621
2000	113	1,461	20	0	1,594
2001	87	1,439	41	0	1,567
2002	60	1,365	68	0	1,493
2003	40	1,281	101	0	1,423
2004	28	1,204	131	0	1,363

Table A5.17 (Figure 1.27) Total emissions in commercial refrigeration (CS, CU and standalone) – Higher threshold (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	313	1,230	0	0	1,543
1991	317	1,264	0	0	1,581
1992	319	1,303	0	0	1,622
1993	320	1,351	0	0	1,671
1994	318	1,410	1	0	1,728
1995	323	1,343	2	0	1,668
1996	279	1,440	2	0	1,721
1997	239	1,517	2	0	1,758
1998	197	1,586	3	0	1,785
1999	162	1,651	9	0	1,822
2000	129	1,707	24	0	1,859
2001	100	1,747	52	0	1,899
2002	74	1,751	90	0	1,915
2003	53	1,745	141	0	1,939
2004	35	1,718	192	0	1,945

Table A5.18 (Figure 1.28) Distribution by refrigeration technology equipment

	Centralized systems	*Condensing unit*	*Standalone*
Metric tonnes	1,407	502	36
Percent of total	71%	27%	2%

Table A5.19 (Figure 1.29) CO_2 eq. emissions (thousand of metric tonnes) in CS – Lower threshold

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	1,728	1,052	0	0	2,780
1991	1,768	1,098	0	0	2,866
1992	1,794	1,153	0	0	2,947
1993	1,807	1,224	0	0	3,030
1994	1,792	1,303	0	0	3,096
1995	1,738	1,213	0	0	2,951
1996	1,430	1,299	0	0	2,729
1997	1,163	1,362	0	0	2,525
1998	906	1,409	0	0	2,315
1999	694	1,435	17	0	2,147
2000	516	1,447	47	0	2,011
2001	368	1,433	106	0	1,907
2002	212	1,357	175	0	1,744
2003	102	1,254	254	0	1,610
2004	40	1,156	315	0	1,511

Table A5.20 (Figure 1.30) CO_2 eq. emissions (thousand of metric tonnes) in CS – Higher threshold

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	1,728	1,052	0	0	2,780
1991	1,768	1,098	0	0	2,866
1992	1,794	1,153	0	0	2,947
1993	1,807	1,224	0	0	3,030
1994	1,792	1,303	0	0	3,096
1995	1,738	1,213	0	0	2,951
1996	1,483	1,353	0	0	2,836
1997	1,254	1,484	0	0	2,738
1998	1,020	1,610	0	0	2,630
1999	813	1,721	20	0	2,554
2000	630	1,833	59	0	2,522
2001	467	1,917	140	0	2,525
2002	316	1,966	246	0	2,528
2003	195	1,990	384	0	2,569
2004	93	1,976	513	0	2,582

Table A5.21 (Figure 1.31) CO_2 eq. emissions (thousand of metric tonnes) in condensing units

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	617	777	0	0	1,394
1991	592	782	0	0	1,374
1992	571	784	0	0	1,355
1993	549	785	0	0	1,334
1994	526	794	0	0	1,320
1995	605	788	0	0	1,393
1996	516	803	0	0	1,318
1997	421	801	0	0	1,222
1998	323	793	0	0	1,116
1999	252	794	0	0	1,047
2000	178	781	7	0	966
2001	124	773	16	0	913
2002	74	744	34	0	851
2003	42	724	60	0	827
2004	18	706	92	0	815

Table A5.23 (Figure 1.32) CO_2 eq. emissions (thousand of metric tonnes) in stand-alone equipment

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	138	16	0	0	154
1991	145	16	0	0	161
1992	153	17	0	0	169
1993	159	17	0	0	176
1994	164	18	1	0	183
1995	166	18	2	0	187
1996	171	19	3	0	192
1997	175	19	3	0	197
1998	179	19	3	0	201
1999	182	19	4	0	205
2000	183	18	5	0	206
2001	180	17	6	0	203
2002	178	16	7	0	201
2003	173	15	8	0	196
2004	163	14	10	0	187

Table A5.24 (Figure 1.33) CO_2 eq. emissions (thousand of metric tonnes) in commercial refrigeration (CS, CU and standalone) – Higher threshold

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	2,483	1,845	0	0	4,328
1991	2,506	1,896	0	0	4,401
1992	2,517	1,954	0	0	4,471
1993	2,514	2,026	0	0	4,541
1994	2,483	2,114	1	0	4,598
1995	2,510	2,019	2	0	4,531
1996	2,170	2,174	3	0	4,346
1997	1,850	2,304	3	0	4,157
1998	1,522	2,422	3	0	3,947
1999	1,248	2,534	24	0	3,805
2000	991	2,632	71	0	3,694
2001	772	2,707	162	0	3,641
2002	568	2,726	286	0	3,580
2003	410	2,729	452	0	3,592
2004	274	2,696	615	0	3,585

Table A5.25 (Figure 1.34) CO_2 eq. emissions (thousand of metric tonnes) in commercial refrigeration (CS, CU and standalone) – Lower threshold

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	2,483	1,845	0	0	4,328
1991	2,506	1,896	0	0	4,401
1992	2,517	1,954	0	0	4,471
1993	2,514	2,026	0	0	4,541
1994	2,483	2,114	1	0	4,598
1995	2,510	2,019	2	0	4,531
1996	2,117	2,121	3	0	4,240
1997	1,760	2,182	3	0	3,944
1998	1,408	2,221	3	0	3,633
1999	1,129	2,248	21	0	3,398
2000	877	2,247	59	0	3,183
2001	673	2,222	128	0	3,024
2002	464	2,117	215	0	2,796
2003	317	1,993	322	0	2,632
2004	221	1,876	417	0	2,514

Table A5.26 (Figure 1.35) Distribution of CO_2 eq emisisons (thousand of metric tonnes) by refrigeration technology equipment

	Centralised systems	*Condensing unit*	*Standalone*
Metric tonnes	2,582	815	187
Percent of total	71%	24%	5%

Table A5.27 (Figure 1.36) Scenario 1 – Refrigerant bank changes – Centralized systems (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
2004	0	3,491	474	0	3,964
2005	0	3,416	638	0	4,053
2006	0	3,306	844	0	4,150
2007	0	3,165	1,132	0	4,298
2008	0	2,975	1,469	0	4,444
2009	0	2,732	1,861	0	4,593
2010	0	2,475	2,200	0	4,674
2011	0	2,205	2,633	0	4,838
2012	0	1,931	3,066	0	4,997
2013	0	1,659	3,499	0	5,158
2014	0	1,398	3,907	0	5,304
2015	0	1,144	4,319	0	5,463
2016	0	911	4,725	0	5,636
2017	0	704	5,067	0	5,771
2018	0	524	5,360	0	5,884
2019	0	377	5,558	0	5,935
2020	0	261	5,752	0	6,013

Table A5.28 (Figure 1.37) Scenario 2 – Refrigerant bank changes – Centralized systems (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
2004	0	3,491	474	0	3,964
2005	0	3,416	638	0	4,053
2006	0	3,306	844	0	4,150
2007	0	3,163	1,124	0	4,288
2008	0	2,971	1,442	0	4,413
2009	0	2,673	1,853	0	4,526
2010	0	2,300	2,259	0	4,559
2011	0	1,865	2,761	0	4,626
2012	0	1,453	3,206	0	4,659
2013	0	1,086	3,577	0	4,663
2014	0	786	3,870	0	4,656
2015	0	540	4,115	0	4,655
2016	0	348	4,316	0	4,664
2017	0	201	4,445	0	4,646
2018	0	102	4,512	0	4,614
2019	0	37	4,507	0	4,544
2020	0	0	4,493	0	4,493

Table A5.29 (Figure 1.38) Scenario 3 – Refrigerant bank changes – Centralized systems (metric tonnes).

	CFC	HCFC	HFC	Others	Total
2004	0	3,491	474	0	3,964
2005	0	3,416	638	0	4,053
2006	0	3,306	844	0	4,150
2007	0	3,161	1,116	0	4,278
2008	0	2,965	1,396	0	4,361
2009	0	2,667	1,741	0	4,408
2010	0	2,294	2,070	0	4,365
2011	0	1,859	2,460	0	4,320
2012	0	1,448	2,791	0	4,239
2013	0	1,082	3,044	0	4,126
2014	0	783	3,223	0	4,006
2015	0	537	3,349	0	3,886
2016	0	346	3,427	0	3,773
2017	0	200	3,442	0	3,642
2018	0	101	3,405	0	3,505
2019	0	36	3,315	0	3,351
2020	0	0	3,217	0	3,217

Table A5.30 (Figure 1.39) Scenario 1 – Refrigerant bank changes – Commercial Refrigeration (metric tonnes).

	CFC	HCFC	HFC	Others	Total
2004	81	5,802	949	0	6,832
2005	64	5,634	1,213	0	6,911
2006	48	5,403	1,536	0	6,988
2007	35	5,127	1,963	0	7,125
2008	24	4,791	2,439	0	7,254
2009	16	4,394	2,975	0	7,384
2010	9	3,973	3,496	0	7,477
2011	4	3,536	4,118	0	7,658
2012	1	3,095	4,731	0	7,826
2013	0	2,658	5,333	0	7,991
2014	0	2,235	5,941	0	8,176
2015	0	1,829	6,535	0	8,365
2016	0	1,459	7,116	0	8,575
2017	0	1,129	7,614	0	8,743
2018	0	844	8,043	0	8,887
2019	0	607	8,368	0	8,975
2020	0	419	8,662	0	9,082

Table A5.31 (Figure 1.40) Scenario 2 – Refrigerant bank changes – Commercial Refrigeration (metric tonnes).

	CFC	HCFC	HFC	Others	Total
2004	81	5,802	949	0	6,832
2005	64	5,634	1,213	0	6,911
2006	48	5,403	1,536	0	6,988
2007	35	5,125	1,955	0	7,115
2008	24	4,787	2,412	0	7,223
2009	16	4,303	2,999	0	7,317
2010	9	3,697	3,655	0	7,361
2011	4	2,997	4,445	0	7,446
2012	1	2,336	5,151	0	7,489
2013	0	1,747	5,749	0	7,496
2014	0	1,264	6,264	0	7,528
2015	0	870	6,687	0	7,557
2016	0	562	7,041	0	7,603
2017	0	326	7,292	0	7,618
2018	0	166	7,452	0	7,618
2019	0	60	7,524	0	7,584
2020	0	0	7,562	0	7,562

Table A5.32 (Figure 1.41) Scenario 3 – Refrigerant bank changes – Commercial Refrigeration (metric tonnes).

	CFC	HCFC	HFC	Others	Total
2004	81	5,802	949	0	6,832
2005	64	5,634	1,213	0	6,911
2006	48	5,403	1,536	0	6,988
2007	35	5,123	1,947	0	7,105
2008	24	4,781	2,366	0	7,171
2009	16	4,297	2,887	0	7,199
2010	9	3,692	3,467	0	7,167
2011	4	2,992	4,144	0	7,140
2012	1	2,332	4,736	0	7,069
2013	0	1,743	5,216	0	6,959
2014	0	1,261	5,617	0	6,878
2015	0	867	5,921	0	6,788
2016	0	560	6,152	0	6,712
2017	0	325	6,289	0	6,614
2018	0	165	6,344	0	6,509
2019	0	59	6,332	0	6,391
2020	0	0	6,286	0	6,286

Table A5.33 (Figure 1.42) Scenario 1 – Refrigerant emission changes – Centralized systems (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
2004	5	1,163	156	0	1,324
2005	0	1,137	210	0	1,347
2006	0	1,106	277	0	1,382
2007	0	1,065	372	0	1,436
2008	0	1,007	480	0	1,487
2009	0	933	607	0	1,540
2010	0	857	712	0	1,569
2011	0	775	856	0	1,631
2012	0	690	996	0	1,686
2013	0	604	1,136	0	1,740
2014	0	519	1,267	0	1,786
2015	0	436	1,403	0	1,840
2016	0	357	1,538	0	1,895
2017	0	284	1,653	0	1,937
2018	0	219	1,754	0	1,973
2019	0	163	1,825	0	1,988
2020	0	117	1,897	0	2,014

Table A5.34 (Figure 1.43) Scenario 2 – Refrigerant emission changes – Centralized systems (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
2004	5	1,163	156	0	1,324
2005	0	1,136	210	0	1,346
2006	0	1,103	277	0	1,380
2007	0	1,060	369	0	1,429
2008	0	1,000	471	0	1,471
2009	0	893	586	0	1,479
2010	0	771	689	0	1,460
2011	0	632	819	0	1,451
2012	0	491	917	0	1,407
2013	0	365	984	0	1,349
2014	0	261	1,022	0	1,284
2015	0	180	1,044	0	1,225
2016	0	118	1,049	0	1,167
2017	0	72	1,033	0	1,105
2018	0	39	1,001	0	1,040
2019	0	18	950	0	968
2020	0	5	897	0	901

Table A5.35 (Figure 1.44) Scenario 3 – Refrigerant emission changes – Centralized systems (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
2004	5	1,163	156	0	1,324
2005	0	1,136	210	0	1,346
2006	0	1,103	277	0	1,380
2007	0	1,059	366	0	1,425
2008	0	998	454	0	1,452
2009	0	891	549	0	1,440
2010	0	769	630	0	1,399
2011	0	631	729	0	1,359
2012	0	489	799	0	1,288
2013	0	364	840	0	1,204
2014	0	260	856	0	1,116
2015	0	179	856	0	1,036
2016	0	117	842	0	959
2017	0	71	811	0	882
2018	0	39	768	0	806
2019	0	18	713	0	731
2020	0	5	656	0	660

Table A5.36 (Figure 1.45) Scenario 1 – Refrigerant emission changes – Commercial Refrigeration (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
2004	28	1,643	191	0	1,862
2005	18	1,600	259	0	1,876
2006	16	1,546	344	0	1,906
2007	13	1,483	461	0	1,958
2008	11	1,403	591	0	2,005
2009	9	1,305	741	0	2,055
2010	7	1,204	878	0	2,089
2011	5	1,099	1,054	0	2,158
2012	3	989	1,225	0	2,217
2013	1	876	1,397	0	2,275
2014	0	763	1,566	0	2,329
2015	0	650	1,735	0	2,385
2016	0	540	1,903	0	2,443
2017	0	437	2,050	0	2,486
2018	0	344	2,180	0	2,524
2019	0	262	2,280	0	2,542
2020	0	192	2,378	0	2,569

Table A5.37 (Figure 1.46) Scenario 2 – Refrigerant emission changes – Commercial Refrigeration (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
2004	28	1,643	191	0	1,862
2005	18	1,598	259	0	1,875
2006	16	1,544	344	0	1,904
2007	13	1,479	458	0	1,950
2008	11	1,396	581	0	1,989
2009	9	1,280	728	0	2,017
2010	6	1,133	879	0	2,018
2011	4	971	1,065	0	2,039
2012	2	768	1,210	0	1,980
2013	1	591	1,323	0	1,915
2014	0	428	1,402	0	1,830
2015	0	306	1,455	0	1,761
2016	0	208	1,485	0	1,693
2017	0	134	1,490	0	1,624
2018	0	75	1,462	0	1,537
2019	0	38	1,421	0	1,459
2020	0	13	1,373	0	1,385

Table A5.38 (Figure 1.47) Scenario 3 – Refrigerant emission changes – Commercial Refrigeration (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
2004	28	1,643	191	0	1,862
2005	18	1,598	259	0	1,875
2006	16	1,544	344	0	1,904
2007	13	1,478	455	0	1,946
2008	11	1,394	565	0	1,970
2009	9	1,278	691	0	1,977
2010	6	1,131	820	0	1,958
2011	4	969	974	0	1,947
2012	2	766	1,092	0	1,860
2013	1	580	1,174	0	1,754
2014	0	419	1,225	0	1,644
2015	0	291	1,248	0	1,539
2016	0	192	1,247	0	1,439
2017	0	118	1,224	0	1,341
2018	0	64	1,181	0	1,245
2019	0	31	1,128	0	1,159
2020	0	9	1,068	0	1,078

Table A5.39 (Figure 1.48) Scenario 1 – Refrigerant CO_2 emission changes (thousand of metric tonnes) – Centralized systems.

	CFC	*HCFC*	*HFC*	*Others*	*Total*
2004	40	1,861	510	0	2,411
2005	0	1,819	686	0	2,505
2006	0	1,769	905	0	2,675
2007	0	1,705	1,216	0	2,921
2008	0	1,616	1,570	0	3,186
2009	0	1,500	1,986	0	3,486
2010	0	1,379	2,330	0	3,708
2011	0	1,250	2,799	0	4,050
2012	0	1,115	3,257	0	4,372
2013	0	977	3,717	0	4,694
2014	0	839	4,146	0	4,985
2015	0	705	4,591	0	5,296
2016	0	576	5,032	0	5,608
2017	0	457	5,408	0	5,866
2018	0	352	5,740	0	6,092
2019	0	261	5,972	0	6,233
2020	0	186	6,207	0	6,393

Table A5.40 (Figure 1.49) Scenario 2 – Refrigerant CO_2 emission changes (thousand of metric tonnes)– Centralized systems)

	CFC	*HCFC*	*HFC*	*Others*	*Total*
2004	40	1,861	510	0	2,411
2005	0	1,818	686	0	2,504
2006	0	1,765	905	0	2,671
2007	0	1,698	1,206	0	2,904
2008	0	1,605	1,539	0	3,144
2009	0	1,438	1,888	0	3,326
2010	0	1,243	2,173	0	3,416
2011	0	1,021	2,523	0	3,544
2012	0	793	2,787	0	3,580
2013	0	591	2,970	0	3,561
2014	0	423	3,076	0	3,499
2015	0	292	3,148	0	3,440
2016	0	191	3,173	0	3,363
2017	0	116	3,139	0	3,254
2018	0	63	3,059	0	3,123
2019	0	30	2,921	0	2,951
2020	0	7	2,774	0	2,781

Table A5.41 (Figure 1.50) Scenario 3 – Refrigerant CO_2 emission changes (thousand of metric tonnes) – Centralized systems

	CFC	*HCFC*	*HFC*	*Others*	*Total*
2004	40	1,861	510	0	2,411
2005	0	1,818	686	0	2,504
2006	0	1,765	905	0	2,671
2007	0	1,697	1,197	0	2,893
2008	0	1,601	1,486	0	3,088
2009	0	1,435	1,756	0	3,191
2010	0	1,241	1,959	0	3,199
2011	0	1,018	2,183	0	3,202
2012	0	791	2,338	0	3,130
2013	0	589	2,414	0	3,003
2014	0	421	2,420	0	2,842
2015	0	291	2,384	0	2,674
2016	0	190	2,303	0	2,493
2017	0	115	2,180	0	2,295
2018	0	63	2,020	0	2,082
2019	0	29	1,832	0	1,861
2020	0	7	1,639	0	1,646

Table A5.42 (Figure 1.51) Scenario 1 – Refrigerant CO_2 emission changes (thousand of metric tonnes) – Commercial Refrigeration

	CFC	*HCFC*	*HFC*	*Others*	*Total*
2004	221	2,581	612	0	3,414
2005	145	2,512	830	0	3,486
2006	126	2,430	1,103	0	3,659
2007	109	2,333	1,480	0	3,921
2008	91	2,209	1,899	0	4,199
2009	73	2,058	2,384	0	4,515
2010	56	1,900	2,826	0	4,783
2011	39	1,736	3,395	0	5,171
2012	23	1,563	3,951	0	5,537
2013	9	1,385	4,509	0	5,903
2014	0	1,205	5,059	0	6,264
2015	0	1,026	5,609	0	6,635
2016	0	850	6,158	0	7,009
2017	0	687	6,638	0	7,325
2018	0	539	7,065	0	7,604
2019	0	409	7,390	0	7,800
2020	0	298	7,710	0	8,008

Table A5.43 (Figure 1.52) Scenario 2 – Refrigerant CO_2 emission changes (thousand of metric tonnes) – Commercial Refrigeration

	CFC	HCFC	HFC	Others	Total
2004	221	2,581	612	0	3,414
2005	145	2,510	830	0	3,484
2006	126	2,426	1,103	0	3,655
2007	109	2,325	1,470	0	3,904
2008	91	2,198	1,868	0	4,157
2009	70	2,018	2,305	0	4,393
2010	50	1,787	2,724	0	4,562
2011	31	1,528	3,226	0	4,786
2012	17	1,209	3,618	0	4,844
2013	6	930	3,923	0	4,859
2014	0	673	4,145	0	4,818
2015	0	480	4,307	0	4,787
2016	0	326	4,411	0	4,737
2017	0	209	4,445	0	4,653
2018	0	117	4,392	0	4,509
2019	0	59	4,294	0	4,353
2020	0	19	4,173	0	4,192

Table A5.44 (Figure 1.53) Scenario 3 – Refrigerant CO_2 emission changes (thousand of metric tonnes) – Commercial Refrigeration

	CFC	HCFC	HFC	Others	Total
2004	221	2,581	612	0	3,414
2005	145	2,510	830	0	3,484
2006	126	2,426	1,103	0	3,655
2007	109	2,324	1,460	0	3,893
2008	91	2,195	1,815	0	4,100
2009	70	2,015	2,173	0	4,258
2010	50	1,785	2,510	0	4,345
2011	31	1,526	2,886	0	4,443
2012	17	1,207	3,168	0	4,392
2013	6	913	3,348	0	4,267
2014	0	659	3,449	0	4,108
2015	0	459	3,468	0	3,927
2016	0	302	3,418	0	3,719
2017	0	185	3,304	0	3,489
2018	0	101	3,132	0	3,233
2019	0	49	2,930	0	2,980
2020	0	14	2,712	0	2,726

Table A5.45 (Figure 1.58) Refrigerant bank in stationary AC (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	*0*	*23,433*	*0*	*0*	*23,433*
1991	*0*	*25,317*	*0*	*0*	*25,317*
1992	*0*	*27,257*	*0*	*0*	*27,257*
1993	*0*	*29,265*	*0*	*0*	*29,265*
1994	*0*	*31,355*	*0*	*0*	*31,355*
1995	*0*	*33,525*	*0*	*0*	*33,525*
1996	*0*	*35,912*	*0*	*0*	*35,912*
1997	*0*	*38,293*	*0*	*0*	*38,293*
1998	*0*	*40,629*	*22*	*0*	*40,650*
1999	*0*	*43,094*	*86*	*0*	*43,180*
2000	*0*	*45,436*	*308*	*0*	*45,744*
2001	*0*	*47,451*	*590*	*0*	*48,040*
2002	*0*	*49,611*	*955*	*0*	*50,566*
2003	*0*	*51,528*	*1,396*	*0*	*52,925*
2004	*0*	*53,475*	*1,874*	*0*	*55,348*

Table A5.46 (Figure 1.59) Refrigerant bank in chillers (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	8,104	1,557	0	32	9,693
1991	8,387	1,602	0	33	10,022
1992	8,506	1,653	18	34	10,211
1993	8,637	1,816	70	35	10,557
1994	8,729	2,225	237	37	11,227
1995	8,485	2,940	674	38	12,137
1996	8,229	3,684	1,110	40	13,063
1997	7,166	5,085	1,648	43	13,942
1998	6,154	6,199	2,025	44	14,423
1999	5,151	7,127	2,307	45	14,630
2000	4,249	7,998	2,607	46	14,900
2001	3,410	8,779	2,875	48	15,112
2002	2,629	9,474	3,120	49	15,272
2003	1,868	10,117	3,340	50	15,375
2004	1,197	10,639	3,592	52	15,480

Table A5.47 (Figure 1.60) Refrigerant emissions from stationary AC (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	0	3,181	0	0	3,181
1991	0	3,445	0	0	3,445
1992	0	3,741	0	0	3,741
1993	0	4,046	0	0	4,046
1994	0	4,397	0	0	4,397
1995	0	4,600	0	0	4,600
1996	0	4,793	0	0	4,793
1997	0	4,902	0	0	4,902
1998	0	5,063	5	0	5,068
1999	0	5,084	12	0	5,096
2000	0	5,129	33	0	5,163
2001	0	5,149	51	0	5,200
2002	0	5,293	74	0	5,367
2003	0	5,310	98	0	5,409
2004	0	5,379	120	0	5,499

Table A5.48 (Figure 1.61) Refrigerant emissions from chillers (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	1,492	255	0	5	1,752
1991	1,546	263	0	5	1,814
1992	1,569	272	3	6	1,849
1993	1,555	275	9	5	1,843
1994	1,555	317	28	5	1,905
1995	1,460	395	77	5	1,937
1996	1,373	462	108	5	1,948
1997	1,337	661	164	5	2,167
1998	1,167	773	189	5	2,135
1999	1,007	872	211	5	2,095
2000	844	967	238	5	2,055
2001	700	1,048	256	5	2,009
2002	564	1,116	277	5	1,963
2003	440	1,177	295	5	1,917
2004	316	1,224	316	5	1,861

Table A5.49 (Figure 1.62) CO_2 equivalent emissions (thousand of metric tonnes) from stationary AC

	CFC	HCFC	HFC	Others	Total
1990	0	4,772	0	0	4,772
1991	0	5,168	0	0	5,168
1992	0	5,611	0	0	5,611
1993	0	6,069	0	0	6,069
1994	0	6,595	0	0	6,595
1995	0	6,900	0	0	6,900
1996	0	7,189	0	0	7,189
1997	0	7,353	0	0	7,353
1998	0	7,595	6	0	7,601
1999	0	7,626	17	0	7,642
2000	0	7,694	50	0	7,745
2001	0	7,723	79	0	7,802
2002	0	7,939	116	0	8,056
2003	0	7,966	157	0	8,122
2004	0	8,069	192	0	8,261

Table A5.50 (Figure 1. 63) CO_2 equivalent emissions (thousand of metric tonnes) from chillers

	CFC	HCFC	HFC	Others	Total
1990	7,594	383	0	0	7,977
1991	7,870	394	0	0	8,264
1992	7,975	408	3	0	8,386
1993	7,899	391	11	0	8,301
1994	7,889	391	36	0	8,316
1995	7,406	391	100	0	7,897
1996	6,968	394	141	0	7,503
1997	6,783	412	213	0	7,407
1998	5,918	428	246	0	6,592
1999	5,106	428	274	0	5,808
2000	4,281	437	309	0	5,027
2001	3,546	445	332	0	4,323
2002	2,861	445	360	0	3,667
2003	2,229	450	383	0	3,063
2004	1,601	460	411	0	2,472

Table A5.51 (Figure 1.64) Bank in stationary AC (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
2004	0	53,475	1,874	0	55,348
2005	0	55,371	2,578	0	57,949
2006	0	57,047	3,683	0	60,730
2007	0	57,918	5,508	0	63,425
2008	0	57,271	8,879	0	66,150
2009	0	55,012	13,894	0	68,906
2010	0	51,081	20,589	0	71,671
2011	0	46,622	27,806	0	74,428
2012	0	41,367	35,819	0	77,186
2013	0	35,967	43,998	0	79,965
2014	0	30,570	52,189	0	82,758
2015	0	25,077	60,479	0	85,556
2016	0	19,576	68,784	0	88,360
2017	0	14,774	76,381	0	91,155
2018	0	10,764	83,192	0	93,955
2019	0	7,487	89,289	0	96,776
2020	0	4,965	94,640	0	99,606

Table A5.52 (Figure 1.65) Bank in chillers (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
2004	1,197	10,639	3,592	52	15,480
2005	561	11,071	3,886	56	15,574
2006	0	11,338	4,262	59	15,658
2007	0	11,136	4,544	62	15,742
2008	0	10,841	4,921	68	15,829
2009	0	10,433	5,422	73	15,928
2010	0	9,920	6,029	79	16,027
2011	0	9,381	6,626	87	16,094
2012	0	8,688	7,336	94	16,118
2013	0	7,988	8,013	102	16,103
2014	0	7,296	8,652	110	16,058
2015	0	6,627	9,258	117	16,001
2016	0	5,991	9,832	124	15,946
2017	0	5,390	10,370	130	15,891
2018	0	4,826	10,872	136	15,834
2019	0	4,296	11,340	142	15,778
2020	0	3,805	11,773	147	15,725

Table A5.53 (Figure 1.66) Scenario 1 - Emissions from stationary AC (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
2004	0	5,388	111	0	5,499
2005	0	5,311	150	0	5,461
2006	0	5,438	212	0	5,650
2007	0	5,514	316	0	5,829
2008	0	5,377	532	0	5,909
2009	0	5,125	837	0	5,961
2010	0	4,845	1,194	0	6,039
2011	0	4,660	1,517	0	6,176
2012	0	4,514	1,856	0	6,370
2013	0	4,283	2,225	0	6,508
2014	0	4,020	2,616	0	6,636
2015	0	3,776	3,041	0	6,817
2016	0	3,515	3,494	0	7,009
2017	0	3,279	3,980	0	7,259
2018	0	2,976	4,460	0	7,437
2019	0	2,587	5,177	0	7,764
2020	0	2,182	5,745	0	7,928

Table A5.54 (Figure 1.67) Scenario 2 - Emissions from stationary AC (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
2004	0	5,388	111	0	5,499
2005	0	5,311	150	0	5,461
2006	0	5,438	212	0	5,650
2007	0	5,514	316	0	5,829
2008	0	5,377	532	0	5,909
2009	0	5,125	837	0	5,961
2010	0	4,962	1,258	0	6,220
2011	0	4,818	1,674	0	6,492
2012	0	4,768	2,182	0	6,950
2013	0	4,383	2,698	0	7,080
2014	0	3,927	3,226	0	7,153
2015	0	3,512	3,805	0	7,318
2016	0	3,043	4,421	0	7,465
2017	0	2,463	4,977	0	7,441
2018	0	1,947	5,456	0	7,403
2019	0	1,449	6,073	0	7,522
2020	0	1,020	6,451	0	7,470

Table A5.55 (Figure 1.68) Scenario 3 - Emissions from stationary AC (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
2004	0	5,388	111	0	5,499
2005	0	5,311	151	0	5,461
2006	0	5,437	213	0	5,650
2007	0	5,511	318	0	5,829
2008	0	5,373	535	0	5,909
2009	0	5,121	840	0	5,961
2010	0	4,878	1,257	0	6,135
2011	0	4,630	1,661	0	6,290
2012	0	4,417	2,144	0	6,561
2013	0	3,918	2,621	0	6,539
2014	0	3,361	3,092	0	6,453
2015	0	3,020	3,620	0	6,640
2016	0	2,637	4,205	0	6,842
2017	0	2,178	4,727	0	6,905
2018	0	1,743	5,240	0	6,983
2019	0	1,323	5,718	0	7,041
2020	0	952	6,217	0	7,169

Table A5.56 (Figure 1.69) Scenario 1 - Emissions from chillers (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
2004	316	1,224	316	5	1,861
2005	208	1,262	338	5	1,814
2006	104	1,281	375	5	1,766
2007	0	1,225	376	6	1,606
2008	0	1,164	398	6	1,567
2009	0	1,097	427	6	1,530
2010	0	1,038	455	7	1,500
2011	0	977	482	7	1,465
2012	0	922	513	7	1,442
2013	0	852	549	8	1,409
2014	0	799	572	8	1,380
2015	0	733	607	9	1,348
2016	0	671	629	9	1,310
2017	0	603	665	10	1,277
2018	0	548	704	11	1,263
2019	0	487	733	11	1,231
2020	0	434	779	12	1,225

Table A5.57 (Figure 1.70) Scenario 2 - Emissions from chillers (metric tonnes).

	CFC	HCFC	HFC	Others	Total
2004	316	1,224	316	5	1,861
2005	208	1,262	338	5	1,814
2006	104	1,281	375	5	1,766
2007	0	1,225	376	6	1,606
2008	0	1,164	398	6	1,567
2009	0	1,097	427	6	1,530
2010	0	1,038	455	7	1,500
2011	0	959	480	7	1,446
2012	0	944	527	7	1,478
2013	0	832	569	7	1,408
2014	0	766	602	7	1,375
2015	0	677	627	8	1,312
2016	0	607	644	8	1,259
2017	0	526	685	8	1,219
2018	0	469	715	9	1,193
2019	0	410	727	9	1,147
2020	0	362	740	9	1,111

Table A5.58 (Figure 1.71) Scenario 3 - Emissions from chillers (metric tonnes).

	CFC	HCFC	HFC	Others	Total
2004	316	1,224	316	5	1,861
2005	208	1,262	338	5	1,814
2006	104	1,281	375	5	1,766
2007	0	1,225	376	6	1,606
2008	0	1,164	398	6	1,567
2009	0	1,097	427	6	1,530
2010	0	1,038	455	7	1,500
2011	0	959	480	7	1,446
2012	0	944	526	7	1,477
2013	0	832	563	7	1,402
2014	0	766	588	7	1,361
2015	0	677	603	8	1,288
2016	0	607	614	8	1,229
2017	0	526	632	8	1,166
2018	0	469	654	9	1,132
2019	0	410	656	9	1,075
2020	0	362	659	9	1,031

Table A5.59 (Figure 1.72) Scenario 1 - CO_2 equivalent emissions (thousand of metric tonnes) in stationary AC.

	CFC	*HCFC*	*HFC*	*Others*	*Total*
2004	0	8,082	180	0	8,263
2005	0	7,966	247	0	8,213
2006	0	8,158	349	0	8,507
2007	0	8,270	523	0	8,793
2008	0	8,065	893	0	8,958
2009	0	7,687	1,414	0	9,101
2010	0	7,268	2,027	0	9,295
2011	0	6,989	2,578	0	9,567
2012	0	6,771	3,158	0	9,929
2013	0	6,424	3,789	0	10,214
2014	0	6,030	4,460	0	10,490
2015	0	5,664	5,186	0	10,850
2016	0	5,273	5,963	0	11,237
2017	0	4,918	6,794	0	11,712
2018	0	4,465	7,617	0	12,082
2019	0	3,880	8,849	0	12,729
2020	0	3,274	9,826	0	13,100

Table A5.60 (Figure 1.73) Scenario 2 – CO_2 equivalent emissions (thousand of metric tonnes) in stationary AC.

	CFC	*HCFC*	*HFC*	*Others*	*Total*
2004	0	8,082	180	0	8,263
2005	0	7,966	247	0	8,213
2006	0	8,158	349	0	8,507
2007	0	8,270	523	0	8,793
2008	0	8,065	893	0	8,958
2009	0	7,687	1,414	0	9,101
2010	0	7,443	2,193	0	9,636
2011	0	7,227	2,985	0	10,212
2012	0	7,152	4,003	0	11,155
2013	0	6,574	5,016	0	11,590
2014	0	5,891	6,048	0	11,938
2015	0	5,269	7,186	0	12,455
2016	0	4,565	8,407	0	12,972
2017	0	3,695	9,451	0	13,146
2018	0	2,921	10,322	0	13,243
2019	0	2,174	11,377	0	13,550
2020	0	1,529	11,971	0	13,500

Table A5.61 (Figure 1.74) Scenario 3 - CO_2 equivalent emissions (thousand of metric tonnes) in stationary AC.

	CFC	HCFC	HFC	Others	Total
2004	0	8,082	180	0	8,263
2005	0	7,966	247	0	8,213
2006	0	8,156	351	0	8,507
2007	0	8,267	527	0	8,794
2008	0	8,060	899	0	8,959
2009	0	7,682	1,421	0	9,102
2010	0	7,317	2,191	0	9,507
2011	0	6,945	2,959	0	9,904
2012	0	6,625	3,885	0	10,510
2013	0	5,877	4,708	0	10,585
2014	0	5,042	5,455	0	10,497
2015	0	4,530	6,315	0	10,845
2016	0	3,956	7,265	0	11,221
2017	0	3,268	8,004	0	11,271
2018	0	2,614	8,648	0	11,262
2019	0	1,984	9,177	0	11,161
2020	0	1,428	9,635	0	11,063

Table A5.62 (Figure 1.75) Scenario 1 - CO_2 equivalent emissions (thousand of metric tonnes) in chillers.

	CFC	HCFC	HFC	Others	Total
2004	1,601	460	411	0	2,472
2005	1,054	458	441	0	1,954
2006	528	457	490	0	1,474
2007	0	451	492	0	944
2008	0	436	523	0	959
2009	0	413	565	0	977
2010	0	389	607	0	996
2011	0	370	646	0	1,016
2012	0	351	690	0	1,042
2013	0	331	742	0	1,073
2014	0	314	777	0	1,090
2015	0	294	826	0	1,120
2016	0	271	861	0	1,132
2017	0	248	913	0	1,161
2018	0	226	970	0	1,196
2019	0	202	1,013	0	1,214
2020	0	174	1,081	0	1,255

Table A5.63 (Figure 1.76) Scenario 2 - CO_2 equivalent emissions (thousand of metric tonnes) in chillers.

	CFC	HCFC	HFC	Others	Total
2004	1,601	460	411	0	2,472
2005	1,054	458	441	0	1,954
2006	528	457	490	0	1,474
2007	0	451	492	0	944
2008	0	436	523	0	959
2009	0	413	565	0	977
2010	0	389	607	0	996
2011	0	344	643	0	988
2012	0	384	732	0	1,117
2013	0	301	802	0	1,103
2014	0	263	859	0	1,122
2015	0	211	900	0	1,111
2016	0	174	932	0	1,106
2017	0	133	988	0	1,122
2018	0	107	1,032	0	1,139
2019	0	86	1,051	0	1,137
2020	0	67	1,067	0	1,133

Table A5.64 (Figure 1.77) Scenario 3 - CO_2 equivalent emissions (thousand of metric tonnes) in chillers.

	CFC	*HCFC*	*HFC*	*Others*	*Total*
2004	1,601	460	411	0	2,472
2005	1,054	458	441	0	1,954
2006	528	457	490	0	1,474
2007	0	451	492	0	944
2008	0	436	523	0	959
2009	0	413	565	0	977
2010	0	389	607	0	996
2011	0	344	643	0	988
2012	0	384	729	0	1,114
2013	0	301	784	0	1,085
2014	0	263	825	0	1,088
2015	0	211	845	0	1,056
2016	0	174	864	0	1,038
2017	0	133	881	0	1,015
2018	0	107	908	0	1,015
2019	0	86	901	0	987
2020	0	67	899	0	966

Table A5.65 (Figure 1.84) Refrigerant bank in industry (metric tonnes).

	CFC	HCFC	HFC	Others	Total
1990	1,550	1,904	0	2,797	6,251
1991	1,575	1,886	0	2,839	6,299
1992	1,599	1,870	0	2,887	6,356
1993	1,631	1,858	0	2,960	6,448
1994	1,642	1,905	0	3,061	6,609
1995	1,577	1,966	20	3,134	6,696
1996	1,495	2,064	54	3,280	6,893
1997	1,409	2,184	108	3,480	7,180
1998	1,241	2,211	246	3,506	7,204
1999	979	2,263	487	3,568	7,298
2000	633	2,326	818	3,703	7,480
2001	313	2,360	1,132	3,801	7,606
2002	116	2,390	1,329	3,868	7,703
2003	34	2,413	1,403	3,933	7,783
2004	10	2,415	1,417	3,943	7,785

Table A5.66 (Figure 1.85) Refrigerant emissions in industry (metric tonnes).

	CFC	HCFC	HFC	Others	Total
1990	153	256	0	242	652
1991	158	248	0	229	635
1992	165	246	0	230	641
1993	169	237	0	218	624
1994	174	246	0	226	646
1995	162	237	1	212	612
1996	168	251	4	224	647
1997	160	254	7	218	639
1998	168	245	22	216	651
1999	164	230	46	202	641
2000	161	230	83	211	686
2001	128	226	115	215	684
2002	74	223	129	219	644
2003	31	220	130	222	603
2004	9	216	127	221	574

Table A5.67 (Figure 1.86) CO_2 equivalent emissions (thousand of metric tonnes) in industry.

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	1 031	384	0	0	1 415
1991	1 063	372	0	0	1 436
1992	1 102	369	0	0	1 471
1993	1 128	356	0	0	1 484
1994	1 153	370	0	0	1 523
1995	1 072	356	3	0	1 430
1996	1 118	376	7	0	1 502
1997	1 065	381	14	0	1 460
1998	1 105	368	34	0	1 507
1999	1 085	345	69	0	1 499
2000	1 071	345	125	0	1 542
2001	836	339	174	0	1 349
2002	476	335	198	0	1 008
2003	203	330	204	0	737
2004	63	324	203	0	590

Table A5.68 (Figure 1.87) California refrigerant bank per refrigerant types (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	29,961	31,687	0	2,828	64,476
1991	30,932	33,691	0	2,872	67,494
1992	31,682	35,761	86	2,921	70,450
1993	31,819	38,043	847	2,995	73,704
1994	31,333	40,782	2,372	3,098	77,585
1995	29,825	43,995	4,641	3,172	81,633
1996	28,217	47,597	6,969	3,320	86,103
1997	24,276	51,873	10,910	3,522	90,581
1998	20,004	55,720	15,127	3,550	94,401
1999	15,690	59,523	19,437	3,614	98,263
2000	11,313	63,128	24,255	3,750	102,446
2001	7,774	66,198	28,147	3,848	105,967
2002	4,956	69,212	31,529	3,917	109,614
2003	3,040	71,835	34,059	3,983	112,917
2004	1,963	74,263	35,937	3,995	116,159

Table A5.69 (Figure 1.89) Refrigerant emissions per refrigerant type (metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	9,721	5,173	0	490	15,384
1991	10,055	5,486	0	462	16,003
1992	10,145	5,854	14	465	16,477
1993	10,223	6,219	134	441	17,016
1994	10,237	6,686	384	456	17,764
1995	9,429	6,973	659	430	17,490
1996	9,426	7,384	1,153	452	18,415
1997	10,070	7,833	2,285	441	20,630
1998	8,878	8,145	3,445	437	20,905
1999	7,396	8,353	4,953	409	21,110
2000	5,824	8,717	6,406	427	21,374
2001	4,282	8,910	7,260	436	20,888
2002	2,685	8,966	8,192	442	20,285
2003	1,477	9,167	8,483	449	19,575
2004	658	9,331	8,362	448	18,799

Table A5.70 (Figure 1.91) Refrigerant emissions expressed in CO_2 equivalent (thousand of metric tonnes).

	CFC	*HCFC*	*HFC*	*Others*	*Total*
1990	73,916	7,760	0	0	81,676
1991	76,441	8,229	0	0	84,670
1992	77,057	8,781	18	0	85,856
1993	77,710	9,307	174	0	87,191
1994	77,788	9,946	499	0	88,233
1995	71,545	10,265	858	0	82,668
1996	71,790	10,797	1,500	0	84,087
1997	77,143	11,205	3,292	0	91,640
1998	68,001	11,533	5,172	0	84,706
1999	56,511	11,710	7,636	0	75,858
2000	44,305	12,136	10,005	0	66,446
2001	32,299	12,324	11,548	0	56,171
2002	19,872	12,323	13,030	0	45,225
2003	10,555	12,540	13,498	0	36,593
2004	4,346	12,728	13,194	0	30,267

Table A5.71 (Figure 1.93) Recovery per refrigerant type metric tonnes).

	CFC	HCFC	HFC	Others	Total
1990	1	9	0	18	27
1991	1	12	0	25	38
1992	6	19	0	32	58
1993	57	43	0	41	140
1994	89	58	0	42	189
1995	162	204	0	47	413
1996	226	349	0	48	623
1997	884	535	4	53	1,475
1998	903	754	8	55	1,719
1999	963	994	16	57	2,030
2000	941	1,139	28	59	2,166
2001	869	1,259	42	61	2,231
2002	745	1,415	55	63	2,278
2003	655	1,546	66	64	2,331
2004	542	1,667	75	66	2,350