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Topic 3 : Applications in Commercial buildings

SIGNIFICANT PARAMETERS FOR ENERGY CONSUMPTION IN FROZEN FOOD AREA OF LARGE SUPERMARKETS

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ABSTRACT

The thermal behavior of large supermarkets is dominated by internal gains. Specifically, refrigerated and frozen food display units area is characterized by cool loads created by commercial cases, which are also moisture trap.

A coupling approach of air conditioning with display cases operation is proposed in this paper, based on a model of the cold aisle phenomenon. In this area, the significant vertical stratification of air has to be considered to calculate electric consumption of display cases. Heat and moisture exchanges are calculated and typical values are given.

Our study gives new elements to answer the questions : how the energy balance is affected by the ambiance setpoints, how interactions between display freezers and air conditioning systems can be characterized.

Typical technical characteristics of roof-top air conditioners performance under various conditions are proposed to calculate energy consumption including auxiliary equipment (fans).

Simulations made with the models for commercial display freezer loads and air temperature profile establish the impacts of different ambient temperature and humidity setpoints. To aggregate the comparison, all energy uses are taken into account: air conditioning, refrigeration and defrosting, it is an essential element for technico-economical analysis of innovative solutions.

Introduction

The thermal behavior of large supermarkets is specific, mainly due to interactions between display freezers and air conditioning system. Vertical stratification of air near display case, experienced by the client as a 'cold aisle' feeling, has to be taken into account to characterize the coupling between this equipment and the air conditioning system.

The first part of our paper deals with the thermal behavior of large supermarkets, and more particularly on interactions between air conditioning and refrigeration cases. A review of relevant bibliography is proposed.

In the second part, chosen equipment modeling are set. Concerning roof-top air conditioners, non-nominal performance is characterized, based on real technical characteristics. Concerning refrigerated cases, a detailed heat and moisture balance is applied to extrapolate nominal performance to cold aisle conditions. Typical values are given.

As a third point, an original approach of coupling air conditioning and display cases is developed, using a modeling of cold aisles. Real heat exchanges, moisture exchanges and defrosting rates of display freezers are calculated.

In the fourth part, this methodology is applied to a typical French supermarket (gross area : 10 800 m²). An analysis of influence of ambient setpoints on global energy consumption is presented.

Part 1: General considerations and state of the art

Energy challenges in supermarkets

Electricity consumption in large supermarket represents a substantial share of about 4% of the national electric energy use, either in the United States than in France. A large part of it, varying from 50 to 70 %, is due to air conditioning and refrigeration cases. Corresponding electric consumption is about 400 kWh/(m².year), which plants it as a real challenge for energy savings.

Interaction between air conditioning and refrigeration cases influences energy consumption and sales parameters (food integrity, thermal discomfort).

Besides thermal coupling, air infiltration carried in through front doors and customer traffic are also source of moisture. Display cases continuously operate as heat and moisture trap. As negative case evaporator temperature is near -35 °C (versus about 4°C for a roof-top air conditioner), when cases cool the ambiance, it is consuming. Furthermore, dehumidification in refrigerated cases is inefficient due to the additional energy required for defrosting and re-cooling cycles.

Our aim is to develop a new approach to study this interaction and look for optimal ambient air condition, using real figures.

Review of bibliography

Supermarket energetic challenges are well set in [ADAM85]. Interactions between refrigeration and air conditioning are often unknown or misunderstood. Traditional separation of industries of air conditioning and refrigeration has limited design information exchange. Anyway, since the early 80's, some progress has been done, as we can see for example with the development of efficient desiccant cooling implementation, described in [MECK92].

Energy analysis has been already developed in [KHAT91], using TRNSYS as a building simulation model. However, this study is mostly based on detailed modeling of various refrigeration components and on possible improvements on refrigeration system. It gives no idea of the interaction between air conditioning and refrigerated cases.

A significant work on influence of ambient humidity on refrigerated display case consumption has been presented in [HOWE93A]. Considering the air curtain as a plane incompressible turbulent flow of a free jet (vertical air curtain) or of a jet moving along a solid surface (horizontal air curtain), a finite-difference technique is used to calculate corresponding heat and moisture loads into the case. The results are validated on experimental data. In [HOWE93B], regressive equations are given to evaluate, for typical cases, heat and moisture transfers under any ambient relative humidity. A building simulation has been used to see the influence of the relative humidity control on air conditioning consumption. However, ambient temperature setpoint influence, is not studied in this paper.

No modeling of the cold aisle phenomenon, which is essential in the considered coupling, has been found. Considering all display cases operating at ambient conditions equal to those experimented in the rest of the sales area could lead to large mistakes in global energy consumption evaluation.

Part 2: Modeling of equipment

In figure 2.1, considered points, defined at each time step by (T; RH), are represented. Concerning refrigeration cases, one has to distinguish positive cases, which operate at I+ (2 to 4 °C; 70 to 80 % RH), and negative cases, which operate at I- (-20 to -18 °C; 80 to 90 % RH).

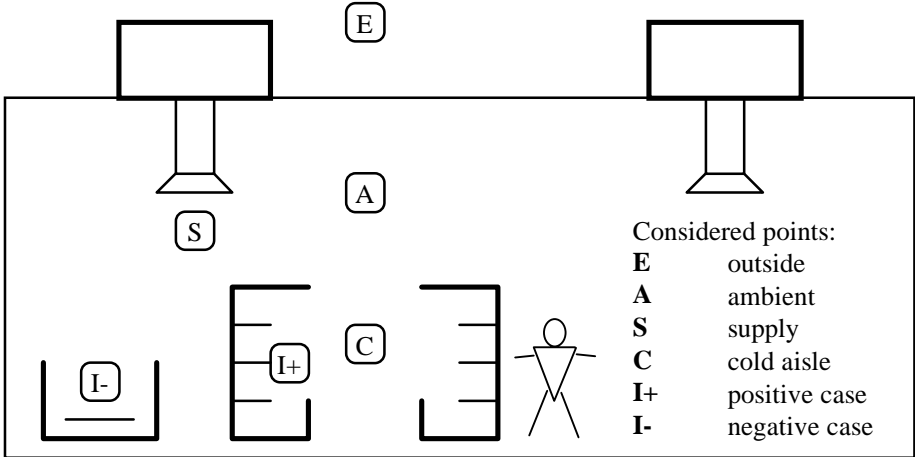


Figure 2.1: cross-section of a supermarket

Before studying the interaction between air conditioning and refrigeration cases, we propose models for each of these systems.

2.a Roof-top air conditioners

Air conditioning systems are sized to face a maximal thermal load, corresponding to the nominal operation. However the system always works under different conditions, mostly in partial load (60 to 90%). Consequently, valid electric consumption calculations have to be based on a modeling of chiller under partial load. Some models exist, but generally require many data, often non-available.

If P_e is the compressor electrical power and P_c is the cooling energy rate, the ratio P_e/P_c is approximated as a second-degree polynomial, as explained in more details in [STAN95]:

$$\left(\frac{P_e}{P_c}\right) = \left(\frac{P_e}{P_c}\right)_{\text{nom}} \cdot (C_1 + C_2 \cdot \Delta T + C_3 \cdot \Delta T^2) \quad (1)$$

where:

$$\Delta T = \left(\frac{T_e}{T_s}\right) - \left(\frac{T_e}{T_s}\right)_{\text{nom}}$$

with:

- P_e compressor electrical power [W]
- P_c cooling energy rate [W]
- T_e outdoor air temperature [K]
- T_s supply air temperature [K]
- C_i parameters of the model
- nom index related to nominal characteristics

Another equation gives the cooling energy rate :

$$P_c = P_{c,\text{nom}} \cdot (C'_1 + C'_2 \cdot \Delta T + C'_3 \cdot \Delta T^2) \quad (2)$$

With these two equations, real running-time and performance of the system can be calculated.

Testing parameterized equations (1) and (2) on one data set relative to manufacturer characteristics of roof-top leads to a quadratic error of less than 4 %. The compressor energy rate as function of the delivered cooling rate is then given by an undimensional expression. The undimensioned model, has been then validated on other systems with cold rates between 50 and 200 kW, keeping the same parameters. All these tests gave small quadratic errors, justifying the applicability of the model. The main advantage for using such model is the small number of required data.

Supply air conditions are a result of supply air flow rate m_s and loads of the building Φ [W] and W [kg/s], which are calculated using COMFIE as a building simulation model on a whole year, [PEUP92]. As roof-top air conditioners are running with a constant air flow, knowing the fresh air flow rate, the return air flow rate, the supplied conditions (T_s , w_s) at each calculation step, the cooling energy rate and the compressor electrical power are calculated. Air handling cycles in the psychometric chart, and finally energy consumption of roof-tops including auxiliaries (fans) are calculated, as described in a previous Clima 2000 paper

[CASA92]. For each step, information on occupation and in-occupation, occurrence, month, electric tariff period is kept in order to be able to do a costs analysis.

Figure 2.2 represents building needs Φ -positive for cooling, negative for heating- as a function of outdoor temperature. In heating modes, two behaviors exist for occupancy and non-occupancy periods.

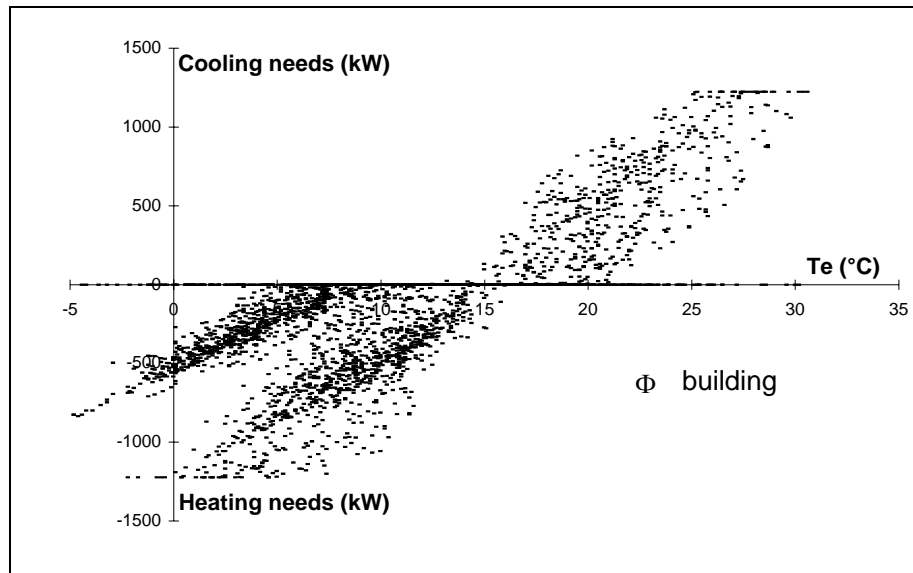


Figure 2.2: energy signature of a supermarket

2.b Display cases

Heat and moisture balance

The heat balance of a display case is influenced by different heat gains and losses, which are represented in figure 2.3:

- those due to exchanges between the ambience outside the case and inside conditions:
 - Q_1 : heat gain through isolated surfaces of the furniture (transmission)
 - Q_2 : heat gain due to openings of the furniture and air curtains (induction)
 - Q_3 : heat gain due to radiation from surrounding surfaces
- those due to equipment implemented in the furniture :
 - Q_4 : lighting disposed inside the refrigerated volume
 - Q_5 : fans and heating pipes inside the refrigerated volume
 - Q_6 : defrosting system
 - Q_7 : restocking of food into the case

The global cooling rate Q necessary to keep the case at T_i and w_i is then given by :

$$Q = Q_1 + Q_2 + Q_3 + Q_4 + Q_5 + Q_6 \quad (3)$$

where Q_1 depends linearly on $(T_a - T_i)$

Q_2 depends linearly on $(h_a - h_i)$

Q_3 could be approximate as depending linearly on $(T_a - T_i)$ (linearisation of radiation transfers

- Q_4 is constant during occupation, and otherwise zero
- Q_5 is constant
- Q_6 depends linearly on $(w_a - w_i)$
- Q_7 considered equals to zero if food is restocked at the temperature of the case

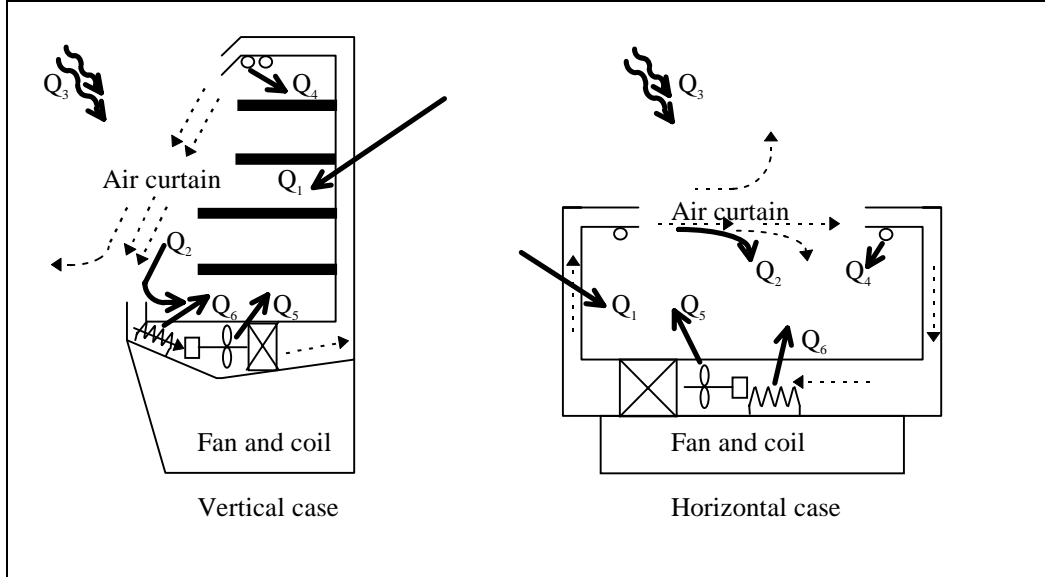


Figure 2.3: global heat balance of two typical refrigeration cases

Two of these terms are now detailed: air induction inside the case and defrosting.

Induction (Q_2)

The determination of the energy exchanged between the inside of the display case and the surrounding air is based on the nominal heat exchange calculated for $T_a = 25 \text{ }^\circ\text{C}$ and $\epsilon_a = 60 \text{ \% RH}$, corresponding to $h_{a,nom}$. Under the real operating conditions, the nominal heat exchange is corrected by multiplication by the real enthalpy difference :

$$Q_2 = Q_{2,nom} \cdot \frac{h_a - h_i}{h_{a,nom} - h_i} \quad (4)$$

The determination of the term h_a will be described in part 3.

Defrosting (Q_6)

Two different methods of defrosting are studied here : Natural defrosting and electrical defrosting. In negative cases at temperatures inferior to $0 \text{ }^\circ\text{C}$, the natural defrosting is not efficient, defrosting used in these cases is electrical. Concerning positive temperatures [$^\circ\text{C}$], both methods are applicable.

Natural defrosting :

It consists in turning off the refrigerator. It does not imply heating the furniture and needs no other energy than the one necessary for re-cooling cycle, i.e. the resulting Q_6 is equal to 0.

Electrical defrosting :

The necessary thermal input from an electrical defroster depends on the amount of captured water W_6 . E_6 , the hourly energy consumption for defrosting, could be calculated from the hourly mean value of W_6 in kg/h, supposed equal to W_2 , the water flow rate associated to Q_2 :

$$W_6 = W_2 = E_2 \cdot \frac{w_a - w_i}{h_a - h_i} \quad (5)$$

Assuming that all captured water is transformed into frost, the energy required to melt the frost is given by :

$$E_6 = W_6 \cdot h_{lv} \quad (6)$$

where h_{lv} is the specific melting heat [kJ/kg] for water.

Defrosting implicates to re-heat case structure and food up to 0 °C, thus a larger energy consumption than E_6 is required. To avoid to describe geometry of the display case an efficiency is used to calculate the energy consumption of the defrosting process. Little bibliography exists on the subject, but we assume, from tests related in [RIGO1990] an average defrosting efficiency of 20 %.

Each defrosting starts for a constant threshold of frost thickness. Consequently, we assume, as in [HOWE93B], that the energy required to defrost is directly proportional to the amount of moisture kept in the display case.

The next table shows results of test performed by [RIGO90], that we used in our example as the nominal values.

		Multi-shelf vertical case	Single reach-in horizontal case
		I+ (+2°C, 70 %)	I- (-20°C, 90 %)
Q_{nom}	Total W.ml	1400	350
	Relative share of each term, under nominal conditions(in %)		
$Q_{1,nom}$	convection + conduction	7	11
$Q_{2,nom}$	induction	65	30
$Q_{3,nom}$	radiation	7	37
$Q_{4,nom}$	lightning	6	0
$Q_{5,nom}$	fans and heating pipes	6	12
$Q_{6,nom}$	defrosting system	9	10

Table 2.1: heat and moisture balance of two cases, under nominal conditions (25 °C, 60 %)

Note that in this table, values are given in Watt per linear-meter of display case.

Part 3: Coupling between air conditioning and display cases

Laboratory conditions and real conditions

Display cases are conventionally tested under specified conditions (25°C, 60% RH), facing a given continuously air flow around them. Consequently, ambient temperature experienced by the air curtain is nearly constant around 25 °C.

These test protocol conditions are very different from those met in real operation. In a typical supermarket, display cases are concentrated in about 20 % of the total sales area. Vertical cases are generally facing each other, creating aisles. Typical height of those freezers is about 2 meters. In this area, the nominal cooling rate is near 200W/m². Heat and moisture exchanges depends on immediate surrounding air. The cold air is remaining in the cold aisle. A vertical stratification is then experienced.

Concerning moisture transfers in the whole supermarket, a one-year measurements campaign has been performed in a French supermarket (gross floor area : 5000 m²). Results are confidential and not published. Anyway, we can say that measurements have shown that the humidity ratio is always homogenous in all the store, except in the grocery area during night. We will consider a single humidity ratio for all the store.

Analysis of vertical air curtains

The discharge jet is blown from the top to the bottom. Induction of air occurs between the discharge jet and the ambient air and between the discharge jet and the air inside the display case. Near the supply nozzle, there is little mixing due to the high velocity of the jet. Induction increases continuously to be maximum near the extraction, as the jet decreases in velocity and widens.

Discretising the air curtain in vertical layers, the exchange coefficient for each depends on evolution of turbulence across the jet. Results of [HOWE93A] on turbulence calculations show that the initial flow and diameter of the curtain air are the two parameters influencing the exchange coefficient. From these results, our model considers a constant exchange rate in each layer, which is called r_j for the j^{th} layer. Results presented in [MORI96], based on a Navier-Stockes 2D modeling of the air curtain under standardized conditions show that r_j increases nearly linearly from the top to the bottom. A linear regression is chosen in our model, which verifies, in nominal conditions :

$$\sum_{\text{case}} r_j = \frac{Q_{2,\text{nom}}}{Q_{\text{nom}}} \quad (7)$$

and:

$$r_1 = 0 \quad (8)$$

If the conditions $h_{a,j} = h(T_{a,j}, w_{a,j})$ of the ambient air in the layer j of the cold aisle are known, enthalpy exchanges in the layer can be extrapolated by:

$$\frac{Q_{2,j}}{Q_{2,\text{nom},j}} = r_j \cdot \frac{h_{a,j} - h_{i,j}}{h_{a,\text{nom}} - h_{i,j}} \quad (9)$$

with $h_{a,\text{nom}} = h(25^\circ\text{C}, 60\% \text{ RH})$

Calculation of temperature profile in the cold aisle

The temperature of the layer of the top, $T_{j=1}$ ($=T_a$), is supposed unvarying in all the supermarket area (convection is preponderant outside the cold aisle). Knowing $r_{j=1}$, an energy balance of each layer j leads to calculate the air layer.

This procedure is repeated from 2m, which corresponds to the top of the case opening, to 0,50m, see figure 3.2.

Equation (10) represents the energy balance of the j^{th} layer, symbolized on figure 3.1.

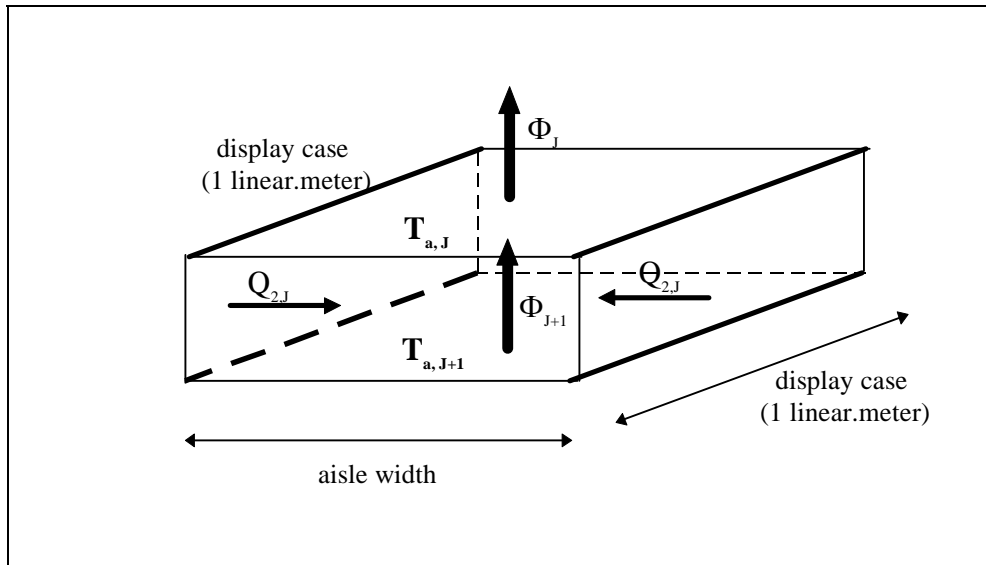


Figure 3.1: Thermal balance of ambient air contained in the j^{th} layer

$$\Phi_j - \Phi_{j+1} = 2 \cdot Q_{2,j} = 2 \cdot K \cdot (T_{a,j} - T_{a,j+1}) \cdot A \quad (10)$$

With:

Q_{2j} evaluated by equation 9 in W/linear meter

K overall heat transfer coefficient in air [$W/m^2.K$], the numeric value was fitted according to an empirical temperature profile

A area corresponding to one linear meter multiplied by the aisle width [m^2]

Results

This modeling has been tested on various configurations. Following figures are corresponding to a cold aisle created by two lines or multi-shelf vertical positive cases, operating at ($2^\circ C$, 70 %), with characteristics corresponding to table .2.1. The width between two facing cases is in this case 4 metres.

Figure 3.2 illustrates the differences between the rate of enthalpy exchange in standardized conditions and our model based on real conditions. It is clear that the induction profile is largely influenced by the cold aisle phenomenon; particularly, in the bottom of the opening, where exchanges are decreasing.

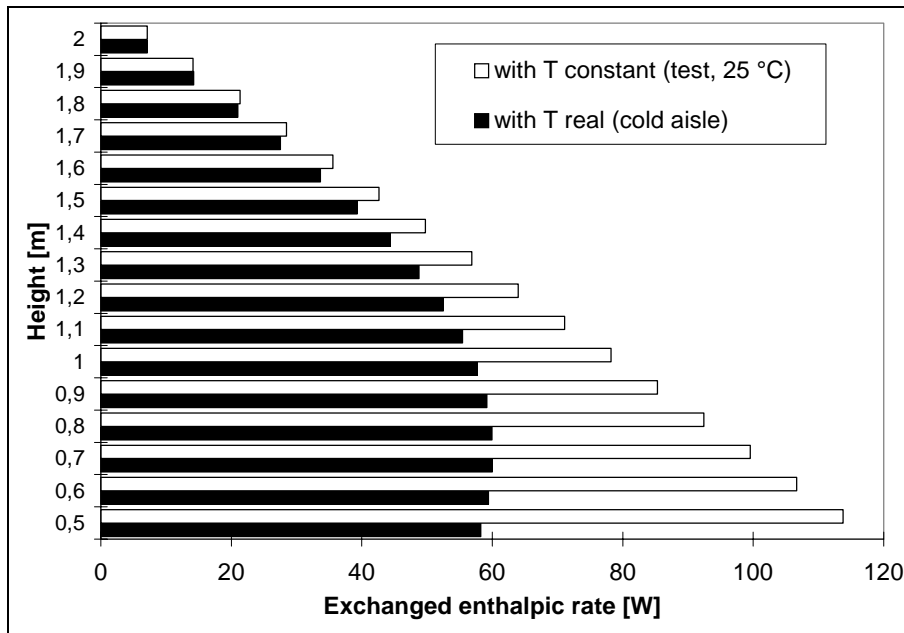


Figure 3.2: induction exchanges under different conditions

Figure 3.3 gives an idea of the vertical profile of temperature in the cold aisle. As we can see, there is a large influence of the boundary conditions, imposed by the rest of the sales area. These curves are corresponding to a given specific humidity. Between 0 to 0,5 meters, the air temperature is nearly constant.

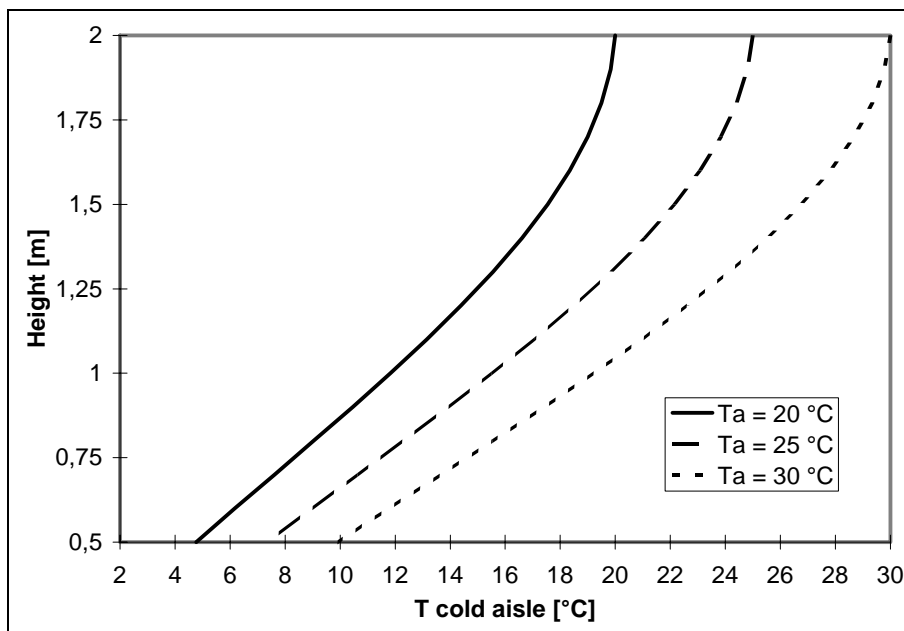


Figure 3.3: vertical stratification in cold aisle, temperature profiles

The mean temperature in the cold aisle can then be calculated. In the rest of the paper, the mean temperature in the cold aisle, T_c , will be called the cold aisle temperature. The

difference between the ambient temperature of the sales area and the cold aisle temperature is a non-linear function of the ambient temperature, represented in figure 3.4.

Considering a humidity ratio of 55 % RH, it decreases from 11 °C for an ambient temperature equals to 30 °C (summer condition in a store without HVAC equipment), to 3 °C for an ambient temperature equals to 12° C (typical night heating setpoint).

As induction exchanges are enthalpic ones, the humidity ratio of the store is also a parameter of the cold aisle conditions determination. Results are shown on figure 3.4 for 55 and 35 % RH.

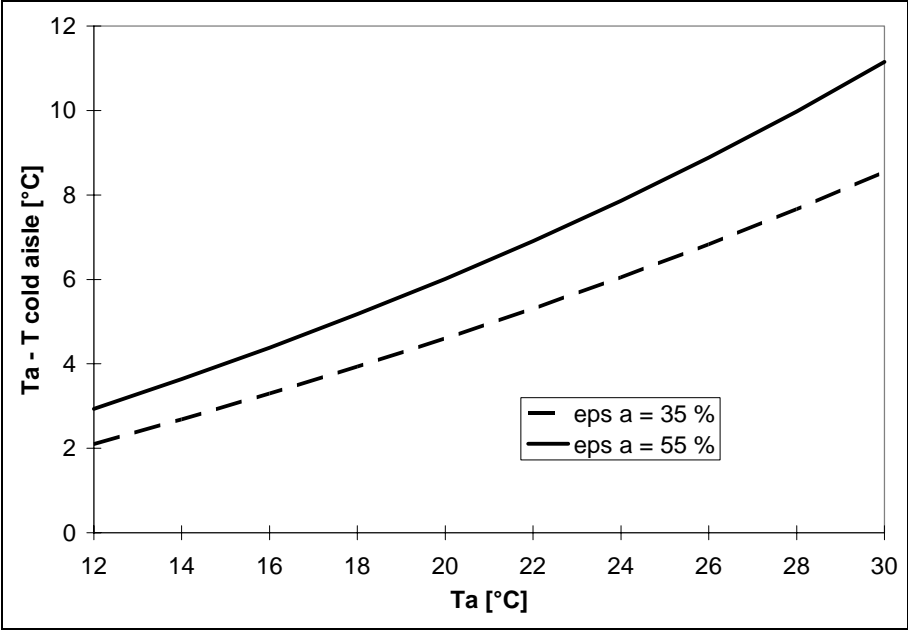


Figure 3.4: relation between ambient and mean cold aisle temperatures

At each time step the profile is calculated, depending on Ta and wa, which are given by the building simulation program. The mean temperature of the cold aisle is then known. The global heat and moisture loads of all cases Q (equation 3) is calculated. Consequently, for the next time step, real loads of heat and moisture created by display cases are considered in the building global balance as to obtain Φ and W.

Figure 3.4 illustrates the influence of boundary conditions (temperature and humidity ratio) on operation of display cases. The ratio between actual load and nominal load (corresponding to 25 °C and 60 % all around the case) is shown. All terms varying with ambient conditions are calculated.

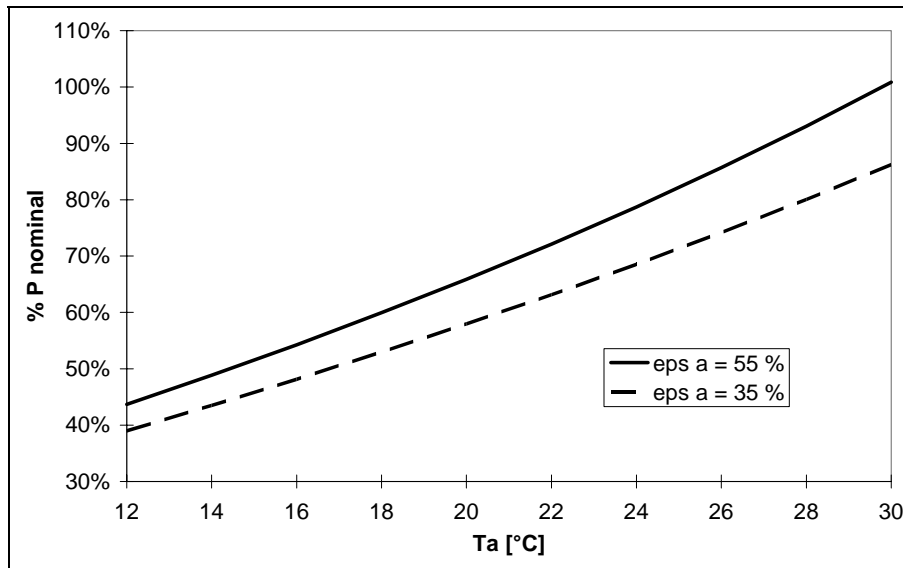


Figure 3.4: real loads of cases in the cold aisle (100 % corresponding to 25°C, 60 %)

Part 4: Test case study

Following is a brief description of main parameters of our base case supermarket.

Year : 1995, on an hourly basis

Location : Carpentras, France (hot and wet climate)

Gross floor area : 10800 m²

Total sales area walls : Sales area : 1280 m² between sales area and stocks, 1100 m² between sales area and outside, 1000 m² between stocks and outside

Total glass area: 50 m² Window + 100 m² skydome

Ceiling height: 6 m

Interior lighting: 400,000 W

Lighting profile: 7.00 to 8.00: 50 %, 9.00 to 20.00 : 100 %

Maximum number of store occupants per day: 21000

Weekly profile for store occupation:

Monday, Tuesday, Thursday : 60 %

Wednesday : 70 %

Friday : 80 %

Saturday : 100 %

Daily profile for store occupation:

9.00 to 11.00: 50 %, 12.00 to 14.00:100 %, 15.00 to 19.00:50 %

Building construction:

Inside walls : concrete + coating

Sales area :

Outside walls : steel plates, glass wool, steel plates

Roof : steel plates, glass wool, coating of plaster

Floor : concrete

Stock :

Outside walls : concrete blocks

Roof : cellular concrete

Floor : concrete

Setpoints:

Sales area :

Heating setpoint : Occupation: 19 °C, no occupation: 12 °C,

Cooling setpoint : 23 °C (only during occupation)

Stock :

Heating setpoint : 5 °C

Primary system:

12 Reversible Roof Tops

Cooling : Nominal cold rate 102 kW, performance coefficient (P_c/P_e) 2.71

Heating : Nominal heat rate 88 kW, performance coefficient (P_c/P_e) 3.03

Auxiliaries per Roof Top : 2 outside fans of 0.55 kW and 2 supply air fans of 4 kW, total air flow 18000 m³/h.

Energy rate model parameters :

Cooling	C1 = 1,06059	C2 = 0,176237	C3 = -0,035888 (T in °C)
Heating	C1 = 1,1004605	C2 = -6,2000859	C3 = -6,033776 (T in K)

Display cases:

As described in figure 4.1

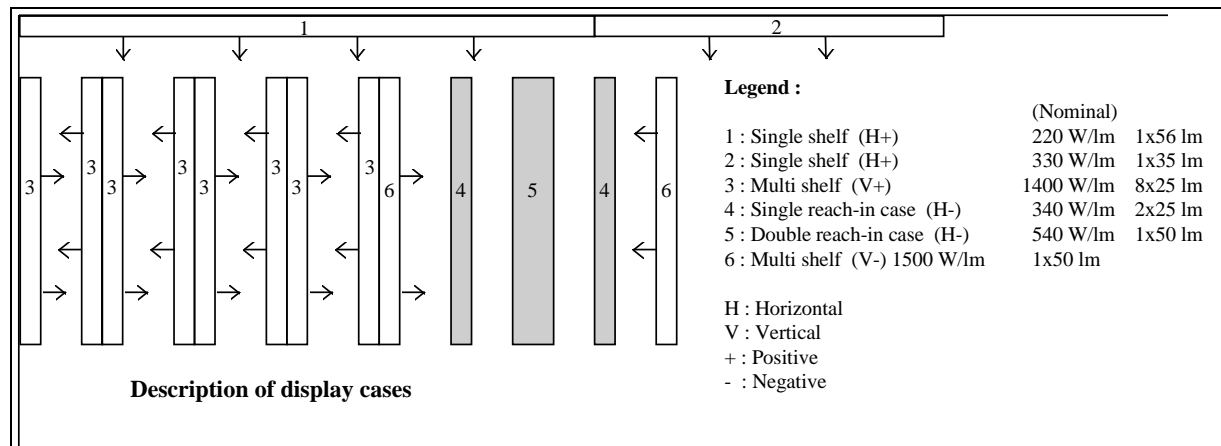


Figure 4.1: description of display freezers of the base case

In the test case, compressors are all placed in a central refrigerating room. In France, supermarket frigorific system generally operates with a constant high pressure, that is why we consider a constant performance coefficient. Positive cases are operating with evaporator temperature from -12 to -8 °C, negative cases are operating with evaporator temperature from -40 to -30°C.

The next three figures are corresponding to a typical week-day of May, which corresponds to the beginning of the cooling season.

On figure 4.2, daily evolution of external, ambient and cold aisle temperatures is represented. From 10h to 18h, the air conditioning system is under operation to maintained 23 °C in the

store, which is the cooling setpoint. During this period, cold aisle temperature is varying between 17 and 18 °C, due to change in ambient specific humidity (because, as described previously, induction exchanges are due to enthalpic differences).

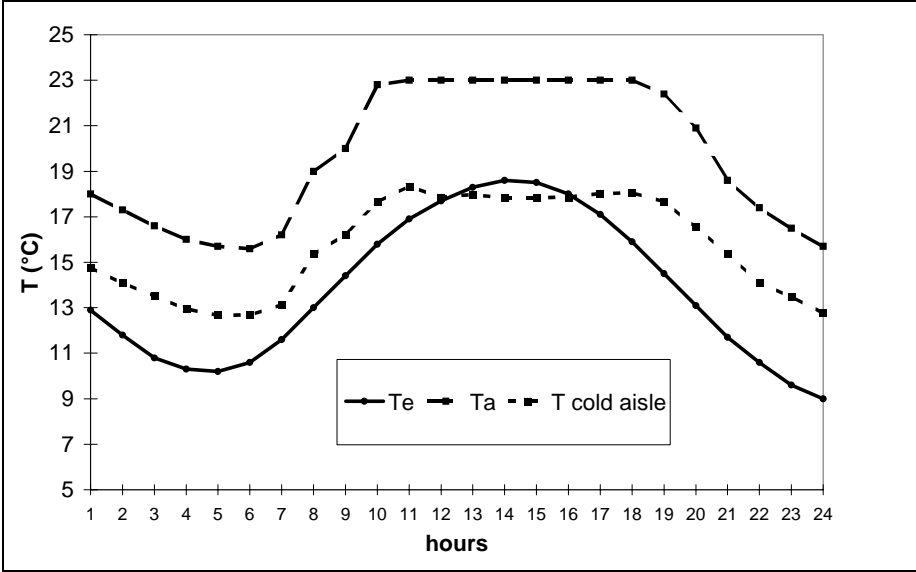


Figure 4.2: Daily temperatures evolution

On figure 4.3, daily evolution of specific outdoor and indoor humidities is shown. Electric hourly average consumption due to electric defrosting is also represented. As we can see, during occupation, from 9h to 19h, the moisture loads, coming from air infiltration and people, are source of large defrosting consumption.

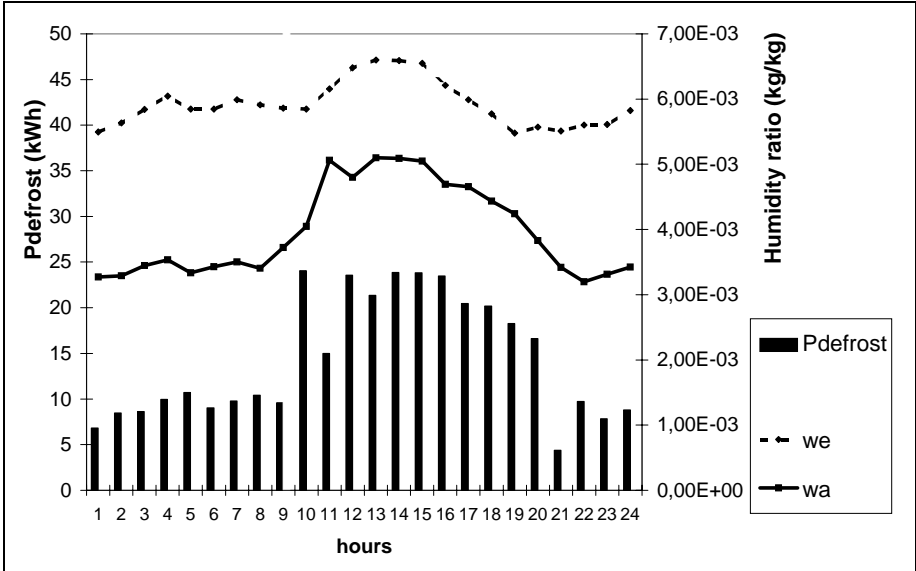


Figure 4.3: Daily humidity ratios evolution and corresponding defrosting values

On figure 4.4, hourly average loads due to cooling, defrosting and cases operation are represented on a cumulate diagram. Under Demand Side Management (DSM) considerations, saved peak load is as important as saved consumption. Consequently, this type of load curve is essential if such an analysis is performed. It is obvious that this is the sum of these for hourly mean loads which has to be considered in evaluating impacts of innovative solutions.

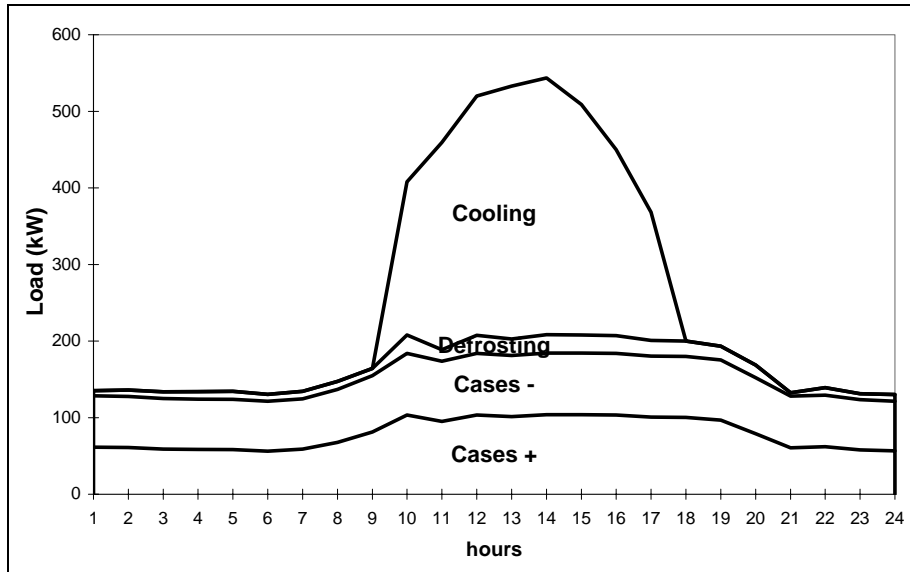


Figure 4.4: Average hourly electric loads

One big issue of this work is to find optimal ambient conditions. It is clear that energy considerations could not be the only ones. Sales and comfort arguments are also to be taken into account.

However, the coupling between display cases and air conditioning allows to estimate global savings realized in changing indoor temperature or humidity setpoints.

Simulations have been performed to see influence of cooling setpoint temperature. Results are given in table 4.1. As we can see, annual operation savings realized between 24 °C and 22 °C are about 91 MWh.

Ta setpoint cooling	Annual electric consumption (in MWh)					Total per m ²
	Heating	Cooling	Case +	Case -	Total	
22 °C	647	624	1220	1436	3927	0.363
23 °C	647	561	1233	1451	3893	0.360
24 °C	647	485	1242	1462	3836	0.355

Table 4.1: influence of ambient temperature setpoint on energy consumption

Controlling humidity in supermarkets is also a quite old idea. Desiccant cooling is now implemented in some supermarkets in the United States of America. We are considering here the influence of humidity control. Our base case is 60 % RH. Table 4.2 gives energy consumption variations due to different humidity setpoint, related to the base case.

As we can see, for the studied supermarket, there exists from an energy point of view an optimum in humidity set point, equal to 45 %.

HR (%)	Energy consumption (in MWh per year)				Total
	Conso MFV+	Conso MFV-	Conso clim	Conso Defr	
35	-92	-108	292	-52	40
40	-50	-58	132	-28	-4
45	-24	-28	36	-14	-29
55	-9	-10	10	-5	-14

Table 4.2: change in energy consumption due to humidity control (base case = 60 %)

Conclusions and perspectives

The developed methodology is a significant step in considering coupling between air conditioning and refrigeration cases. However, the coupling approach has to be more systematic, more configurations and more display cases have to be studied. This will have to be based on validation measurements.

In next future, this software is going to be used to test improvement technologies applied to air conditioning system, such as desiccant cooling or cooling night storage, or applied to display cases, such as night stores or new air curtains.

The prevision and the control of the electric peak demand is important. Some load peak reduction is feasible in supermarkets, for example, by use of generator group, or by controlling of HVAC system during peak hours. The chosen annually simulation makes possible to study DSM innovative solutions.

If such a simulation software is useful in research areas, there is a larger aim to make it available for architects, maintenance managers, technical store managers or energy consulting engineers. The authors have developed a first version of an easy-to-use software, [ORPH96], principally focused on HVAC energy consumption. The originality of this tool is to have a friendly interface, -developed on Excel 5- for inputs and outputs, which make it really available for use by any person involved in energy savings in supermarkets.

References

- [ADAM85] 'Intereffects in supermarkets', P. Adams, ASHRAE Journal, p.38-40, October 1985
- [CASA92] 'Air conditioning energy consumption estimation', R. Casari, D. Marchio, S. Stan, R. Dumitru, Clima 2000, 1992
- [HOWE93A] 'Effects of store relative humidity on refrigerated display case performance', R. H. Howell, ASHRAE Transactions 99, p. 667-678, 1993
- [HOWE93B] 'Calculation of humidity effects on energy requirements of refrigerated display cases', R.H. Howell, ASHRAE Transactions 99, p. 679-693, 1993
- [KHAT91] 'Modeling supermarket refrigeration energy use and demand', M.K. Khattar, H.B. Morton, D.H. Walker, ASHRAE Journal, p. 23-31, July 1991
- [MECK92] 'Efficient integration of desiccant cooling in commercial HVAC systems', G. Meckler, Desiccant cooling and dehumidification, ASHRAE Special Publication, 1992
- [MORI96] 'Aéraulique des M.F.V.', C. Morillon, proceedings of Journées Froid et distribution commerciale, Juin 1996
- [ORPH96] 'Clim Top: logiciel pour la conception et l'audit de la climatisation des grandes surfaces de vente', M. Orphelin, D. Marchio, R. Casari, A. Tassone, December 1996
- [PEUP92] "Comfie - Passive solar design tool for multizone buildings", User guide, B. Peuportier, I. Blanc, Centre d'Energétique, Ecole des Mines de Paris, January 1992
- [RIGO90] "Meubles et vitrines frigorifiques", Pyc Editions, 1990
- [STAN95] "Maîtrise et calcul des consommations des installations de climatisation dans le secteur tertiaire", S. Stan, Thèse de doctorat, Centre d'Energétique, Ecole des Mines de Paris, January 1995

Nomenclature

A	area [m ²]
T	temperature [K]
RH	relative humidity [%]
ε ; eps	relative humidity [%]
P	energy rate or electrical power [W]
P _e	compressor electrical power [W]
P _c	cooling energy rate [W]
t	temperature [°C]
h	specific enthalpy [J/kg dry air]
w	humidity ratio [kg water/kg dry air]
Q	heat rate or cold rate [W]
W	the amount of captured water [kg/h]
E	the hourly energy consumption for defrosting [Wh]
h _{lv}	the specific melting heat for water [kJ/kg]
r	exchange rate coefficient
I-	conditions inside "negative" display cases
I+	conditions inside "positive" display cases

Indices

Considered points

A	indoor conditions
E	outdoor conditions
S	supply air conditions
C	cold aisle conditions
nom	nominal
i	conditions inside cold display case
c	cooling

Display cases

1	transmission
2	induction
3	radiation
4	lighting
5	fans and heating pipes
6	defrosting
7	restocking of merchandise