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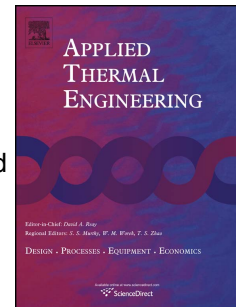


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Energy performance of supermarket refrigeration and air conditioning integrated systems

Luca Cecchinato^a, Marco Corradi^a, Silvia Minetto^{1b}

Dipartimento di Fisica Tecnica, Università di Padova, via Venezia 1, I-35131
Padova, Italy

Abstract

The electricity consumption for air conditioning and refrigerated cases in large supermarkets represents a substantial share of the total electricity consumption. The energy efficiency of supermarkets can be improved by optimising components design, recovering thermal and refrigerating energy, adopting innovative technology solutions, integrating the HVAC system with medium temperature and low temperature refrigeration plants and, finally, reducing thermal loads on refrigerated cases. This study is aimed at investigating the performance of different lay-out and technological solutions and at finding the potential for improving energy efficiency over the traditional systems in different climates. In the analysis chillers and heat pumps working with R410A, medium temperature systems working with R404A and low temperature systems

¹ **Corresponding author.** Tel.: +39 049 827 6879; fax: +39 049 827 6896; e-mail: silvia.minetto@unipd.it

^a Junior Member of IIR Commission B2

^b Junior Member of IIR Commission D1

working both with R404A and R744 were considered. The investigated solutions enable an annual energy saving higher than 15% with respect to the baseline solution for the considered climates.

Keywords: Display cabinet, Efficiency, Heat recovery, Heat pump, Supermarket, Water chiller.

Nomenclature

A	heat transfer surface [m^2]
C_c	cycling degradation factor [-]
COP	heat pump energy efficiency [-]
c_p	specific heat at constant pressure [$\text{J kg}^{-1} \text{K}^{-1}$]
d	heat exchanger hydraulic diameter [m]
EER	chiller energy efficiency [-]
h	enthalpy [J kg^{-1}]
hd	heat dissipation factor [-]
HER	cabinets heat extraction rate [W]
k	transmittance correction coefficient [-]
\dot{m}	mass flow rate [kg s^{-1}]
P	power consumption [W]
PLF	part load factor [-]
PLR	part load ratio [-]
p	pressure [Pa]
Q	thermal power [W]
R	thermal power ratio [-]

Re Reynolds number: $Re = \frac{\dot{m} \cdot d}{S \cdot \mu}$ [-]

RH relative humidity ratio [%]

S cross sectional area [m^2]

T temperature [$^{\circ}C$]

UA thermal transmittance [$W K^{-1}$]

\dot{V} swept volume per time unit [$m^3 s^{-1}$]

x specific humidity ratio [-]

Greek letters

α heat transfer coefficient [$W m^{-2} K^{-1}$]

Δ difference [-]

η_{el} electric efficiency [-]

η_{is} isentropic efficiency [-]

η_v volumetric efficiency [-]

θ logarithmic mean temperature difference [K]

ρ density [$kg m^{-3}$]

Subscripts

A ambient

AHU_c air handling unit cooling load

AHU_h air handling unit heating load

aux auxiliary

c condenser/condensation

C cabinet

D design

db dry-bulb

e evaporator/evaporation

k compressor

<i>in</i>	inlet
<i>load</i>	actual load condition
<i>LT</i>	low temperature application
<i>MT</i>	medium temperature application
<i>N</i>	nominal
<i>out</i>	outlet
<i>r</i>	refrigerant
<i>SC</i>	subcooling
<i>set</i>	set point
<i>sf</i>	secondary fluid
<i>SH</i>	superheat
<i>wb</i>	wet-bulb

1. Introduction

Energy saving and F-gas emission reduction have been top issues for the last two decades. Energy production and use have undergone a rationalisation process in almost all application fields. In particular, since in many countries the structure of energy consumption by different consumer groups is characterised by a high share of the residential and tertiary sector, which is very often responsible for about 40% of the total final energy demand, a great research effort is taking place to minimise their direct and indirect contribution to greenhouse effect. European Community Directive 2002/91/CE [1] proposes important guidelines for the improvement of building energy efficiency; in Italy the Directive was applied by the national regulation DL 192/05, issued in 2005 [2]. Data supplied by the main Italian energy distribution Company [3] show that 25% of the Italian energy consumption is related to commercial

services and, amongst them, supermarkets are the main consumers. It is therefore of the greatest importance to propose new technologies that can improve energy efficiency in supermarkets and that can represent an acceptable trade off between installation and running costs.

The analysis of each contribution to the total energy bill of supermarkets is the first step towards energy efficiency improvement. According to the Canadian Energy Efficiency Office [4], food preservation contributes for about one third to the total energy consumption of supermarkets, while lighting and air conditioning account for 25% and 20% respectively. Efficient design of refrigeration plants is mandatory for energy saving in supermarkets. Proposed technical solutions include the adoption of evaporative condensing [5], the application of heat recovery and floating condensation [6] and mechanical subcooling of low-temperature systems [7–9].

Cogeneration and trigeneration have been recently proposed as feasible solutions for supermarket applications to produce electricity and to make cooling capacity available by means of an absorption unit. Maidment and Tozer [10] review a number of combined cooling heat and power (CCHP) options involving the use of different cooling and engine technologies. The pay-back time for a CCHP system applied to a supermarket has been estimated to be around 9-10 years by Arteconi et al. [11].

When air conditioning and space heating are demanded to liquid chillers and heat pumps, there is a real possibility of integrating vapour compression cycles operating at different temperature levels and of recovering waste heat deriving from refrigeration plants [6,12]. As in supermarket there are normally vapour

compression cycles operating at three different temperature levels, corresponding to air-conditioning units, medium and low temperature refrigeration plants respectively, it is possible to concatenate the systems according to a cascade lay-out, where the unit operating at the lower temperature level rejects the condensation or subcooling heat to the system at the higher temperature level [13, 14]. The integration may be operated directly, i.e. coupling evaporators, condensers and liquid subcoolers of different units, either using a secondary fluid.

Gianni and Oliva [15] suggested the adoption of a water loop heat pump system integrated with a heat recovery system for air conditioning in shopping centres. The authors presented a technical and economical comparison of this system with a traditional one, based on experimentation, reporting a 28% energy saving over the total air-conditioning costs. Yang and Zhang [16] analyse the energy saving potential of integrated supermarket HVAC and refrigeration systems using three subcoolers. The subcoolers are placed between the refrigeration medium and low temperature systems and between the air-conditioning system and the medium and low temperature plants. The energy saving associated to the proposed solution is strictly dependent on the air-conditioning and refrigeration load ratio. The authors pointed out an energy saving equal to 15.6 and 6.5% with a ratio of 1 and 10 respectively. Even though Yang and Zhang [16] carried out a model based analysis on the energy saving potential of integrated supermarket HVAC and refrigeration systems using multiple subcoolers, they compared the different solutions considering only two operating conditions, namely 35 and 18.3 °C external air temperature.

In this paper a seasonal dynamic building energy simulation is carried out to compare different plant solutions integrating air-conditioning liquid chillers and heat pumps with refrigeration units in the climates of Treviso (Northern Italy), Stockholm and Singapore. In order to evaluate the different systems energy consumption, a mathematical model providing an accurate estimation of refrigeration and air conditioning units cooling or heating capacity and power consumption out of nominal rating conditions was developed. R410A was chosen as the working fluid for air-conditioning systems; refrigeration units for medium temperature applications were assumed to use R404A, while for freezers units both R404A and R744 were considered.

2. PLANT OPTIONS

The different plant options are presented in the following. The baseline plant, System 0 (Figure 1), does not include any integration between air conditioning and refrigeration units. Two independent direct expansion refrigeration machines for medium temperature (MT) and low temperature (LT) applications are used; the refrigerant is R404A. An air condensed heat pump provides cold/hot water to the air handling unit (AHU) of the building variable air volume (VAV) air conditioning system.

System 1 (Figure 2) provides the simplest level of integration between refrigeration units. Saturated liquid out of the condenser of the LT refrigeration unit is subcooled by means of the evaporation of a certain amount of refrigerant in the MT refrigeration unit.

System 2 introduces the integration between the air conditioning system and the refrigeration units; during summertime, the liquid chiller, beyond providing the air conditioning unit with cold water, subcools liquid out of both MT and LT condensers (Figure 3). During winter, MT and LT condensation waste heat is completely recovered for supplying the AHU hot coil with water at 45 °C. Excess heat is rejected using a finned coil condenser. A heat pump, in parallel with the heat recovery system, provides the additional heat, whenever necessary (Figure 4). System 2 includes also the subcooling of the LT refrigeration unit by the MT refrigeration unit, as described for System 1.

System 3 consists in a typical cascade system, where the LT refrigeration unit rejects the condenser heat to the MT refrigeration unit. The MT unit is operated on R404A, while the LT unit uses R744. In summertime, the chiller provides the MT unit with cold water for refrigerant subcooling, while feeding the AHU cold coil (Figure 5). In wintertime, total heat recovery is applied to MT refrigeration unit for direct heating service; the heat pump supplies the system with the additional heat (Figure 6).

System 4 during summer overlaps system 2. In wintertime, the waste heat from the condensers of MT and LT refrigeration units is recovered in a water loop working as the heat source of the water/water heat pump (HP 1). Additional heat, if need be, is supplied by the air/water heat pump (HP 2) (Figure 7).

System 5 during summer overlaps system 3, except for being R744 the working fluid in the LT refrigeration unit. In wintertime, the waste heat from the MT refrigeration unit condenser is recovered in a water loop working as the heat source of the water/water heat pump (HP 1). Additional heat, if need be, is

supplied by the air/water heat pump (HP 2) (Figure 8).

3. Building simulation

A single-floor building with a total area of 3620 m² was simulated. Its height is 5 m. The longest dimension is south-north oriented. The total area is divided into three parts: the supermarket area is 3019 m²; the cafeteria area is 150 m² and the shops area is 450 m². The cafeteria has 25.3 m² windows south oriented and 35.2 m² east oriented. The shops area has 170.0 m² windows south oriented and 59.3 m² west oriented. Simulations were based on Treviso, Stockholm and Singapore test reference years; Treviso is a town located in the North-East of Italy (North 45°40', East 12°15'). Monthly maximum and minimum mean average temperatures for the three cities are reported in Table 1.

The supermarket was considered to be located on a flat ground, without obstacles to the direct solar radiation, such as trees or buildings, and the ground reflection coefficient was assumed equal to 0.2.

Table 2 summarises the supermarket MT and LT display cases and cold rooms main dimensions, their nominal heat extraction rate (HER_N) and the area where they are located. Nominal power is related to standard ISO 23953 Class 3 [17], i.e. 25°C and 60% relative humidity. The reference code identifies the cabinets according to ISO 23953 [17].

The contribution to internal thermal loads related to lighting, equipment and people are detailed in Table 3. For the bakery area (51.6 m²), which is included in the supermarket, the internal load due to equipment was assumed to be

equal to 50 W m^{-2} , as recommended by Stefanutti [18].

The resulting nominal display cases and cold rooms contribution to internal thermal loads is respectively -34.1 W m^{-2} and -26.6 W m^{-2} , respectively, during opening (Mon-Sat 6 a.m.-8 p.m.) and closing hours (Mon-Sat from 8 p.m. to 6 a.m., Sunday).

The air conditioning plant was assumed to be active 14 hours per day (from 6.00 a.m. to 8.00 p.m.), six days per week. In summertime, the air conditioning set point is 26°C , while in the wintertime the set point for ambient heating is 20°C . During the opening hours, the maximum relative humidity is set to 60%: above this limit, dehumidification is activated. Air change rate is assumed to be $10^{-2} \text{ m}^3 \text{ s}^{-1}$ per person in the cafeteria, $1.2 \cdot 10^{-3} \text{ m}^3 \text{ s}^{-1}$ per m^2 of shops area and $8 \cdot 10^{-3} \text{ m}^3 \text{ s}^{-1}$ per person in the supermarket.

Air infiltration was considered negligible in the building air-conditioned spaces because of the presence of door air curtains. The AHU is equipped with a 60% effective heat recovery system and an economizer which is active from 11°C to 25°C external air temperature. An economizer is a damper opening that draws up to 100% outside air when the outside air is cooler than the temperature inside the building, thereby providing free cooling. The design AHU flow rate is $8.47 \text{ m}^3 \text{ s}^{-1}$. The supermarket was simulated with EnergyPlus [19] to calculate at hourly intervals the thermo-hygrometric variables and the thermal energy exchanged.

A particular effort was spent taken to modelise the interaction between the display cases and the surrounding ambient. Thermal loads of display cases and

cold rooms depend on the supermarket air thermoigrometrical conditions, T_A and RH_A . In particular, in the surroundings of vertical open cabinets, the cold air leakage creates the so-called cold aisle effect, where temperature $T_{A,C}$ and relative humidity $RH_{A,C}$ can be very different from the average supermarket air thermoigrometrical conditions. Orphelin et al. [20] show the dependence of cold aisle temperature from the average ambient temperature and humidity. Equations (1) were obtained from Orphelin et al. [20] data. The influence of the other cabinets on the surrounding ambient temperature and relative humidity was neglected.

$$\begin{cases} T_{A,C} = T_{A,C}(T_A, RH_A) = 6.5 \cdot 10^{-3} \cdot T_A^{1.473} \cdot RH_A^{0.6067} \\ RH_{A,C} = RH_{A,C}(T_{A,C}, x_A) \end{cases} \quad (1)$$

To correctly evaluate the display cases thermal load, corrective factors were introduced to emend the nominal HER. Such coefficients are dependent, in the case of open cases, on temperature and humidity of the air entering the cabinet while they are only a function of average ambient temperature in the case of closed cabinets. The following general equation (2) was derived from technical datasheet of a refrigeration equipment manufacturer [21]. It expresses the corrective factor F_{HER} for the cabinets nominal HER (declared at 25 °C air temperature and 60% relative humidity), as a function of ambient temperature $T_{A,C}$, and relative humidity $RH_{A,C}$ near cabinets.

$$F_{HER} = F_{HER}(T_{A,C}, RH_{A,C}) = a_1 \cdot RH \cdot T_{A,C} + a_2 \cdot T_{A,C} + a_3 \cdot RH_{A,C} + a_4 \quad (2)$$

Table 4 summarises the coefficients of equation (2) for the considered cabinets. In the case of closed cabinets (VF4), ambient relative humidity does not influence HER.

The required HER of each cabinet can be calculated by multiplying the nominal HER by the corrective factor F_{HER} , as expressed in equation (3):

$$HER = HER_N \cdot F_{HER} \quad (3)$$

During the closing hours, night blinds reduce the HER of vertical multi-deck cabinets by 45%. The HER of each cabinet, expressed as a function of the local thermal hygrometric variables, contributes to the energy conservation equation of each building zone in the simulation. According to Orphelin et al. [20], only 80% of display cabinets HER contributes to the building energy equation, being 20% of the HER related to cabinet internal gains (lighting, fans, heaters, etc). The average temperature and humidity of specific zones depend on many variables (conduction through the walls, air supply, radiative loads, internal loads and display cases HER). The cold aisle phenomenon was not considered in the calculation of thermal load from the walls.

Simulations exhibit a maximum heating demand of 207.2, 238.7 and 86.9 kW in Treviso, Stockholm and Singapore respectively. The cooling peak is 69.3, 50.0 and 158.9 kW in the three cities respectively. In Table 1 the maximum LT system load resulting from building simulation is reported together with the peak load ratios of the MT and of the AHU to the LT system (R_{MT-LT} , $R_{AHUC-LT}$, $R_{AHUH-LT}$). It appears that Stockholm loads are similar to Treviso ones but the $R_{AHUH-LT}$ ratio is higher because of the colder climate. Singapore is characterised by a

much lower $R_{AHUH-LT}$ ratio and a more than double $R_{AHUC-LT}$ ratio. This is due to the hotter and more humid climatic conditions.

4. Air conditioning and refrigeration unit design

The peak value of heating and cooling loads, as resulting from the simulation which is described in the previous section, was considered for the design of the air conditioning and refrigeration units. The evaporation temperatures of the MT and LT refrigeration units were considered to be respectively -10°C and -35°C .

Superheat was set to 5°C for all evaporators, while no subcooling was assumed at the condensers exit. During heating operations, water set point was 45°C , while it was 7°C when cooling is required. Tables 5 and 6 summarise the design conditions for refrigeration and air conditioning units. In the case of cascade systems (System 3 and 5), evaporation temperature and superheat of the MT refrigeration units are respectively $T_{sf,in}$ and ΔT_{sf} , as listed in Table 5.

T_c is condensation temperature, $T_{c,sf,in}$ is the temperature of the secondary fluid entering the condenser and ΔT_{sf} is its temperature lift. $T_{e,sf,in}$ is the temperature of the secondary fluid to the evaporator. \dot{V} indicates the compressors total swept volume. Stockholm Systems 2-5 HP2 (air/water heat pump) was dimensioned at -3°C air temperature, supplementary heating is supplied through a condensing boiler of constant 1.1 energy efficiency.

For System 0 in Treviso, Bitzer 4CC-6.2Y, Bitzer 6G-30.2Y and Copeland ZP295KCE-TWD compressors were considered for MT, LT and HP1-2 units respectively. Treviso System 3 adopts Bitzer 2EHC-3K in R744 LT plant. For all

other systems and climates, in order to obtain a fair comparison, the swept volume was changed according to Tables 5 and 6, considering the same compressors isentropic and volumetric efficiencies curves.

The design conditions for heat exchangers water/refrigerant SR1 (Figures 3 and 5), MT refrigerant/LT refrigerant SR2 (Figures 2-4 and 7) are presented in Table 7 where $T_{r,in}$ is the refrigerant inlet temperature, $T_{sf,in}$ the secondary fluid inlet temperature, $\Delta T_{approach}$ is the minimum temperature approach between the fluids, $T_{LT,in}$ and $T_{LT,out}$ are respectively the inlet and outlet temperatures of the LT refrigerant and $T_{MT,in}$ e $T_{MT,out}$ the inlet and outlet temperatures of the MT refrigerant.

The possibility of varying the chiller/heat pump water delivery temperature according to the actual cooling/heating load was introduced. It has been widely demonstrated in the technical literature [22] that the modulation of the water temperature can lead to relevant energy saving. During summertime the water temperature can vary in the range 7-16°C, if the required cooling capacity is inside the interval 20-100% of the design value, while it stays at 16°C under 20%. In the case of Systems 2-5, the total cooling load includes also the subcooling demand of the refrigeration units. Contrary to Yang and Zhang [16] approach, the chiller always operates the MT and/or LT plants subcooling even without AHU cooling demand.

During winter, the water delivery temperature is modulated between 45°C and 30°C in the 20-100% interval of the design heating capacity, and never drops below 30°C.

5. Air conditioning and refrigeration units simulation

In order to evaluate the energy consumption of refrigeration and air conditioning units a mathematical model providing an accurate estimation of refrigeration and air conditioning units cooling or heating capacity and power consumption out of nominal rating conditions was developed. The proposed procedure corrects the design transmittance taking into account the heat transfers coefficients variations associated to refrigerant and secondary fluid mass flow rates variations.

For a set refrigerating unit, the numerical procedure can be outlined with the following steps:

A. Determination of design heat exchangers transmittances

- A.1 input data gathering;
- A.2 calculation of design condensation and evaporation temperatures;
- A.3 determination of design heat exchangers transmittances;

B. Determination of system performance under different load and secondary fluid temperature operating conditions

- B.1 calculation of cooling/heating capacity and power consumption based on corrected transmittances as a function of the actual refrigerant and secondary fluid flow rates and of the logic of the unit control system;

B.2 part load performance estimation.

“B” steps should be iterated for each different working condition to be considered. The numerical procedure steps are detailed in the following.

A.1 Input Data gathering

The following data are required for the model application to a given refrigeration unit:

- the refrigerant fluid and the secondary fluids at evaporator and condenser;
- for a refrigerating unit, the rated cooling capacity, $Q_{e,D}$, and compressor power consumption, $P_{k,D}$ (or design efficiency, EER) in design conditions;
- for an heating unit, the rated heating capacity, $Q_{c,D}$, and compressor power consumption, $P_{k,D}$ (or efficiency, COP) in design conditions;
- design refrigerant evaporator outlet superheat and condenser outlet subcooling values, $\Delta T_{SH,D}$ e $\Delta T_{SC,D}$;
- compressor coefficients with reference to the EN 12900 [23] polynomial equations;
- secondary fluids design inlet temperature, temperature variations between inlet and outlet and pressure drops, $T_{sf,D}$, $\Delta T_{sf,D}$ (or in alternative $\dot{m}_{sf,D}$)

- heat exchanger pumps or fans power consumption, $P_{aux,D}$;
- heat exchangers design ratio of refrigerant side to secondary fluid side heat transfer area;
- heat exchangers design ratio of refrigerant to secondary fluid heat transfer coefficient with reference to the considered heat transfer surface area;
- refrigerant and secondary fluid circuits pattern;
- compressors sequencing and inverter management for multi-compressor and inverter driven unit respectively, with reference to cooling capacity control.

Refrigerant pressure drops in the heat exchangers data are not required since they are not easily available in manufacturers data sheets. Heat exchangers heat transfer surface areas and heat transfer coefficients, together with condenser air inlet-outlet temperature difference (or mass flow rate), are seldom present in data sheets; nevertheless they can be quite easily obtained from the manufacturer or from open literature correlations or heat exchangers design practice.

A.2 Calculation of design condensation and evaporation temperatures

Considering the single stage vapour compression cycle shown in Figure 9 p - h diagram, the thermodynamic properties of points 1, 2s, 3 and 4 can now be determined in design conditions. The process undergone by the refrigerant inside the compressor is described by an adiabatic compression (1-2) and an

isobaric process (2-2d) which accounts for heat transfer phenomena between the compressor and the environment. The thermodynamic properties associated to points 2 and 2d depend on the compressor characteristics. In fact, 2s is the discharge point corresponding to an isentropic compression process, while 2 and 2d refer respectively to the end of an adiabatic compression process and to the compressor discharge, taking into account both the compression irreversibility and compressor heat dissipation. With reference to EN 12900 superheat and subcooling standard values [23], it is possible to express the cooling capacity, the power consumption and the refrigerant mass flow rate with the following polynomial equations:

$$\begin{aligned} \hat{Q}_e = & a_1 + a_2 \cdot T_e + a_3 \cdot T_c + a_4 \cdot T_e^2 + a_5 \cdot T_e \cdot T_c + a_6 \cdot T_c^2 + \dots, \\ & \dots + a_7 \cdot T_e^3 + a_8 \cdot T_e^2 \cdot T_c + a_9 \cdot T_e \cdot T_c^2 + a_{10} \cdot T_c^3 \end{aligned} \quad (4)$$

$$\begin{aligned} \hat{P}_k = & b_1 + b_2 \cdot T_e + b_3 \cdot T_c + b_4 \cdot T_e^2 + b_5 \cdot T_e \cdot T_c + b_6 \cdot T_c^2 + \dots, \\ & \dots + b_7 \cdot T_e^3 + b_8 \cdot T_e^2 \cdot T_c + b_9 \cdot T_e \cdot T_c^2 + b_{10} \cdot T_c^3 \end{aligned} \quad (5)$$

$$\hat{m} = \frac{\hat{Q}_e}{\hat{h}_1 - \hat{h}_4}, \quad (6)$$

Considering equations (4-6) and the EN 12900 [23], superheat and subcooling values, the compressor volumetric and isentropic efficiencies can be thus easily expressed as:

$$\eta_v = \frac{\hat{m}}{\hat{\rho}_1 \cdot \dot{V}}, \quad (7)$$

$$\eta_{is} = \frac{\dot{m} \cdot (\hat{h}_{2s} - \hat{h}_1)}{\hat{P}_k}. \quad (8)$$

In the hypothesis that these efficiencies values doesn't depend on the refrigerant superheat at the compressor suction, the refrigerant mass flow rate, the cooling capacity and the power consumption can be calculated with reference to Figure 9 cycle:

$$\dot{m} = \eta_v \cdot \dot{V} \cdot \rho_1, \quad (9)$$

$$Q_e = \dot{m} (h_1 - h_4), \quad (10)$$

$$P_k = \dot{m} (h_2 - h_1), \quad (11)$$

h_2 , being the refrigerant enthalpy at the end of the adiabatic compression process, is determined by:

$$h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_{is}}. \quad (12)$$

The compressor discharge enthalpy, h_{2d} , can be defined as a function of the compressor heat dissipation factor, h_d :

$$h_{2d} = h_2 - \frac{P_k \cdot hd}{\dot{m}}. \quad (13)$$

Through equations (12) and (13), the thermal power to be dissipated in the condenser can be calculated as:

$$Q_c = \dot{m}(h_{2d} - h_4) = \dot{m} \left[\left(h_1 + \frac{h_{2s} - h_1}{\eta_{is}} - \frac{P_k \cdot hd}{\dot{m}} \right) - h_4 \right]. \quad (14)$$

The vapour compression system energy efficiency is obtained with the following two expressions, for a refrigerating and a heating unit respectively:

$$EER = \frac{Q_e}{P_k + \sum P_{aux,D}}, \quad (15)$$

$$COP = \frac{Q_c}{P_k + \sum P_{aux,D}}, \quad (16)$$

Combining equations (4-16), the vapour compression cycle design operating temperatures, $T_{c,D}$ e $T_{e,D}$, can be determined solving the following equations system:

$$\begin{cases} Q(T_{e,D}, T_{c,D}) - Q_D = 0 \\ P_k(T_{e,D}, T_{c,D}) - P_{k,D} = 0 \end{cases}, \quad (17)$$

where Q represent the cooling or the heating capacity for a refrigerating and a heating unit respectively.

A.3 Determination of design heat exchangers transmittances

The evaporator and condenser transmittances in design conditions can be subsequently determined with a logarithmic mean temperature difference approach. If the heat exchanger heat transfer surface is divided into M partitions of equal refrigerant enthalpy difference, its transmittance can be calculated with the following expression in the co-current hypothesis:

$$UA = \sum_{j=1,M} \frac{\dot{m} \cdot (h_j - h_{j-1})}{\theta_j}, \quad (18)$$

where θ_j is defined as the logarithmic mean temperature difference between the refrigerant and the secondary fluid with reference to the j -th subdivision of the refrigerant enthalpy difference. It can be calculated solving the following algebraic system:

$$\left\{ \begin{array}{l} h_j = h_{j+1} - \frac{h_{out} - h_{in}}{M} \\ T_{sf,j} = T_{sf,j+1} - \frac{\dot{m} \cdot (h_{out} - h_{in})}{M \cdot \dot{m}_{sf} \cdot c_{p,sf}} \end{array} \right. , j = 0, M-1 \quad (19)$$

$$\left\{ \begin{array}{l} h_j = h_{out} \\ T_{sf,j} = T_{sf,in} \end{array} \right. , j = M$$

For a given refrigerating unit, it is thus possible to calculate the design cycle operative temperature together with the heat exchangers transmittance, $UA_{e,D}$ and $UA_{c,D}$, through equations (4-19).

For each heat exchanger secondary fluid, the volumetric flow rate is calculated on the basis of the following equation, neglecting mass transfer phenomena:

$$\dot{V}_{sf} = \frac{Q}{\rho_{sf} \cdot c_{p,sf} \cdot \Delta T_{sf}} , \quad (20)$$

where Q is the heat exchanger thermal power.

B.1 Calculation of cooling/heating capacity and power consumption

For secondary fluids temperature values different from design conditions, it is possible to determine the refrigeration machine working conditions and energy efficiency based on design data in the following hypotheses:

- constant design evaporator outlet superheat and condenser outlet subcooling values;

- refrigerant pressure drops variations influence is neglected;
- constant design heat exchangers secondary fluids volumetric flow rates and pressure drops;
- refrigerant and secondary fluids heat transfer coefficient, α , proportional to the m-th power of Reynolds number: $\alpha \propto \text{Re}^m$

The vapour compression cycle operating temperatures, T_c e T_e , determination problem can be reduced to the solution of the following non-linear equations system:

$$\begin{cases} UA_e(T_e, T_c) - UA_{e,D} \cdot k_e = 0 \\ UA_c(T_e, T_c) - UA_{c,D} \cdot k_c = 0 \end{cases}, \quad (21)$$

where the heat exchanger transmittances, UA , are obtained from equations (4-12, 14) and (18-19) that represent the basic equations system of a refrigerating unit for given values of evaporation and condensation temperatures. Instead, the condenser and evaporator transmittance correction coefficients, k , are calculated with the following expression:

$$k = \frac{1 + \left(\frac{\alpha_{sf}}{\alpha_r} \right)_D \cdot \left(\frac{A_{sf}}{A_r} \right)_D}{\left(\frac{\dot{m}}{\dot{m}_D} \right)_{sf}^{-m_{fs}} + \left(\frac{\alpha_{sf}}{\alpha_r} \right)_D \cdot \left(\frac{A_{sf}}{A_r} \right)_D \cdot \left(\frac{\dot{m}}{\dot{m}_D} \right)_r^{-m_r}}. \quad (22)$$

where A_r and A_{sf} are the heat transfer surface on the refrigerant and secondary fluid side respectively and α_r and α_{sf} are the corresponding heat transfer coefficients.

The system of equations is solved using a standard solver based on the MINPACK subroutine HYBRD1, which uses a modification of Powell's hybrid algorithm. This algorithm is a variation of Newton's method, which uses a finite-difference approximation to the Jacobian and takes precautions to avoid large step sizes or increasing residuals [24]. The system power consumption, cooling and heating capacity are calculated from system (21) solutions using equations (11), (10) and (14). The system energy efficiency is obtained according to equations (15-16). Refrigerant properties were evaluated with NIST Reference Fluid Thermodynamic and Transport Properties - REFPROP, Version 7.0 [25].

In presence of systems with cooling or heating capacity control, such as multi-compressor scroll chillers, multi-step screw compressor machines or inverter driven units, equations (4) and (5) can be rewritten as a function of the number of active compressors and of the compressor capacity partialization rate.

If the refrigeration unit has multiple refrigerant or secondary fluids circuits which can be activated as a function of the actual unit working conditions and active compressors, equation (22), expressing the heat exchangers design transmittance correction factor, is corrected. The auxiliaries design power consumption must also be corrected as a function of the secondary fluids active circuits. Further details and equations development can be found in Cecchinato et al. [26].

Usually air condensed units for civil and commercial refrigeration are equipped with condensation temperature control systems to assure safe working conditions for the compressor and the expansion valve. These systems regulate the condenser air volumetric flow rate in order to keep or the condensation temperature above a minim value, $T_{c,min}$, or the pressure difference at the two ends of the throttling valve, above a minim value, $\Delta p_{c,min}$. In this case, the system (21) must be replaced with the following constrained equations system:

$$\begin{cases} f_e(T_e, \dot{V}_{sf,c}) = UA_e - UA_{e,D} \cdot k_e = 0 \\ f_c(T_e, \dot{V}_{sf,c}) = UA_c - UA_{c,D} \cdot k_c = 0 \\ T_c = T_{c,min} \text{ or } T_c = T(p_e + \Delta p_{c,min}) \end{cases} \quad (23)$$

Usually commercial refrigeration plants are equipped with a suction pressure control system to assure stable evaporation temperature to the cabinets. Parallel compressors are relay controlled in order to keep the evaporation pressure set point value, $p_{e,set}$. For a given number of active compressors, the system (21) must be replaced with the following constrained equations system:

$$\begin{cases} f_c(T_e, T_c) = UA_c - UA_{c,D} \cdot k_c = 0 \\ p_e = p_{e,set} \end{cases} \quad (24)$$

B.2 Part load performance estimation

The effect of the part load working of the refrigeration unit, is calculated on the basis of its full load performance and strictly depends on its capacity control system. For single compressor units with simple relay control law, the part load factor, PLF , can be calculated with the equations proposed in the European and Italian standards [27, 28] as a function of the part load ratio, PLR :

$$PLF = \frac{PLR}{C_c \cdot PLR + (1 - C_c)}, \quad (25)$$

where C_c is a degradation factor associated to the unit cycling operation losses.

The part load ratio is calculated as the ratio between the actual load condition, Q_{load} , and the unit full load cooling or heating capacity considering the of operative secondary fluids temperature values. For a refrigerating or a heating system, the part load power consumption, cooling or heating capacity, $P_{k,PL}$, $Q_{e,PL}$ or $Q_{c,PL}$, can be thus defined as a function of full load performance:

$$P_{k,PL} = PLF \cdot P_k, \quad Q_{e,PL} = PLR \cdot Q_e \text{ or } Q_{c,PL} = PLR \cdot Q_c. \quad (26)$$

The system energy efficiency is subsequently obtained according to equations (15-16). Refrigerating units with multiple compressors or with compressors with different capacity steps, a step capacity control is usually adopted. In this case the full load capacity of each step is determined. It is now possible to determine

the two steps which mostly approximate the actual load condition, the first one will have a lower capacity, the second one an higher one. The unit part load capacity, power consumption and efficiency are determined by linear interpolation between these two steps performances [27]. If the unit smallest control step cooling capacity is higher than the required one, the performance data are obtained as detailed for single compressor units, referring the PLR value in equation (25) to the unit smallest control step cooling capacity.

Finally, the seasonal energy consumption of the proposed systems has been based on the dynamic building energy load results, as described in section 3; the average performance of the plant has been calculated with a steady-state approach for each building simulation time step (one hour), as proposed in the standard UNI 11135 (UNI 2003) [29].

6. Simulation results

The total annual energy consumption of refrigeration and air conditioning systems, related to the overall supermarket area, is represented in Figure 10 for the considered plant solutions and climates. Treviso monthly results are reported in Tables 8 and 9. With reference to System 0 in the case of constant water delivery temperature (constant set point), System 4 presents the largest energy saving (14.8, 20.5 and 12.9% in Treviso, Stockholm and Singapore respectively), thanks to the integration between the liquid chiller/heat pump and the refrigeration units. The water/water HP of System 4, which uses the recovered heat as heat source, satisfies 89.4, 86.8 and 100% of the heating energy demand in the three climates. System 2, which directly recovers the

refrigeration units condensation heat, is slightly penalised by the high condensation temperature in wintertime. But, if the water delivery temperature can vary (floating set point) according to the heating/cooling demand, System 2 operates at lower average winter condensation temperature thus offering the best result, with 15.6, 22.5 and 14.9% energy saving in comparison to System 0 (constant set point) in Treviso, Stockholm and Singapore respectively. The energy efficiency improvement depends on the climatic conditions and is higher for colder cities. This is related to the higher heat recovery during the heating season.

In general, Systems 2 and 4 show very similar performances; the plant results much simpler in the case of System 2, with lower complexity in the piping integration system and with only one heat pump/chiller.

System 3, which has the same integration level between the HVAC unit and the MT refrigeration unit of System 2, while the LT refrigeration unit rejects the condensation heat to the MT refrigeration system in a typical cascade configuration, does not perform as well as System 2, because of the lower cycle efficiency of the cascade system with respect to the LT and MT separate cycles. In fact, in Treviso design conditions the COP of the System 2 LT unit is 18.5% higher than System 3 one considering the cascade arrangement. For the same reasons, System 5 shows poorer performances than System 3. The energy penalisation of these systems is higher in Treviso than in Stockholm and Singapore. In fact, the energy penalisation of cascade lay-out with respect to single stage one is lower for high compression pressure ratio. This is the case of Stockholm plant operating winter heat recovery for a long time period and of

Singapore one operating at high dry-bulb temperature during almost all the year.

The energy saving of the proposed solutions is at least 6.2, 5.9 and 7.4%, in the case of System 1 with constant water set point for Treviso, Stockholm and Singapore respectively. The subcooling of the LT unit by the MT plant is more effective in the hotter climates.

Figures 11, 12 and 13 represent the annual energy consumption related to the overall supermarket area, of each unit of the analysed systems (MT refrigeration, LT refrigeration, liquid chiller, heat pump air/water, heat pump water/water), in the case of constant temperature of water delivery for Treviso, Stockholm and Singapore. Stockholm supplementary condensing boiler primary energy consumption was converted to electricity with a 2.5 conversion factor; for each system the boiler annual consumption was below 0.3 kWh m^{-2} and it was not reported in Figure 12.

During wintertime, the integration between the refrigeration units and the HVAC system, resulting from the recovery of the condensation heat, leads to a very low energy consumption for the air/water heat pump. During summertime, Systems 2-5 show higher energy consumption for liquid chillers than that of System 0-1, because of the added heat load coming from the subcooling needs of the refrigeration units.

As for the MT refrigeration unit is concerned:

a) the energy consumption of System 2 is higher than that of System 4 in Treviso and Stockholm, because of higher condensation temperature

associated to winter heat recovery;

b) Systems 3 and 5 show the highest energy consumption due to the fact that they have to manage, as an additional load, the condensation heat of the LT R744 units. System 1 is also penalised in Singapore because of the higher load associated to LT unit subcooling.

7. Conclusions

A medium-size supermarket was considered, with respect to the refrigerating machinery for commercial refrigeration and air conditioning/heating system; different levels of integration between the HVAC and the refrigeration units was introduced. A mathematical model providing an accurate estimation of refrigeration and air conditioning units cooling or heating capacity and power consumption out of nominal rating conditions has been developed. The proposed procedure corrects the design transmittance taking into account the heat transfers coefficients variations associated to refrigerant and secondary fluid mass flow rates variations.

Calculations show that all the proposed solution offer a benefit in terms of energy efficiency over the reference traditional solution. The integration between the refrigeration units and the HVAC system and their optimal management can produce energy saving up to 15.6, 22.5 and 14.9% in comparison with the traditional solution in Treviso, Stockholm and Singapore respectively. The best performance is achieved in the colder climates and its associated to wintertime condensation total heat recovery from the refrigeration

units. The simplest solution, operating the LT unit subcooling by means of the MT plant, allows energy efficiency improvements equal to 6.2%, 5.9 and 7.4% for Treviso, Stockholm and Singapore respectively. This solution is suitable for hot climates. LT cascade lay-out is not to be adopted in temperate climates (Treviso).

As widely demonstrated in the technical literature, simulations confirm that floating water set point is effective for energy saving.

As a final remark, the integration approach should be proposed also for large supermarkets and hypermarkets, in consideration of their significant energy consumption.

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FIGURES

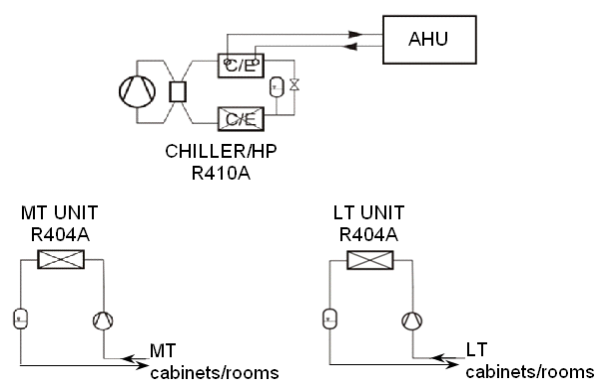


Figure 1

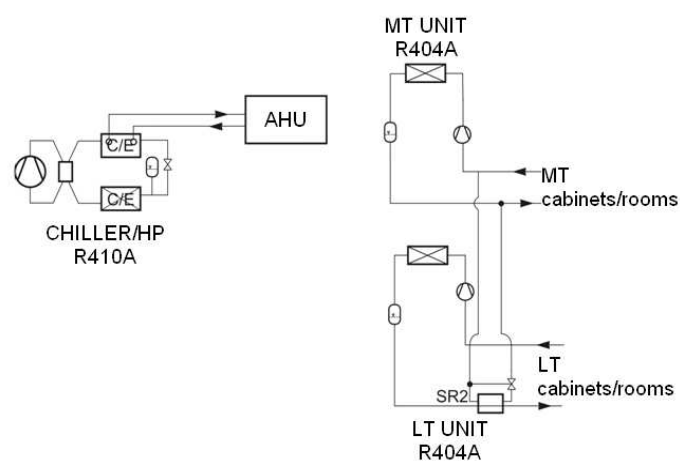


Figure 2

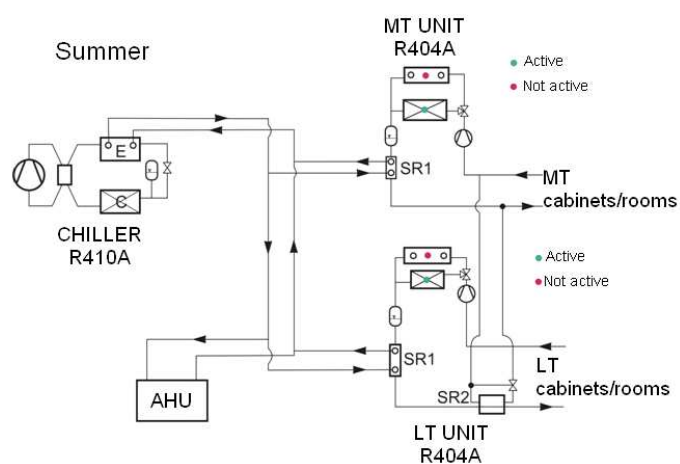


Figure 3

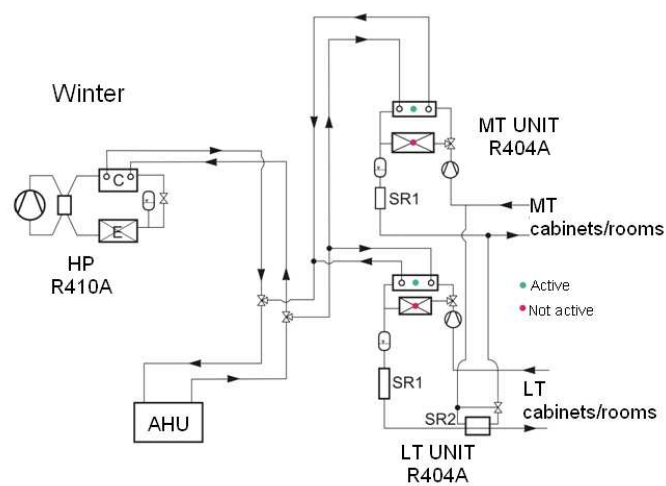


Figure 4

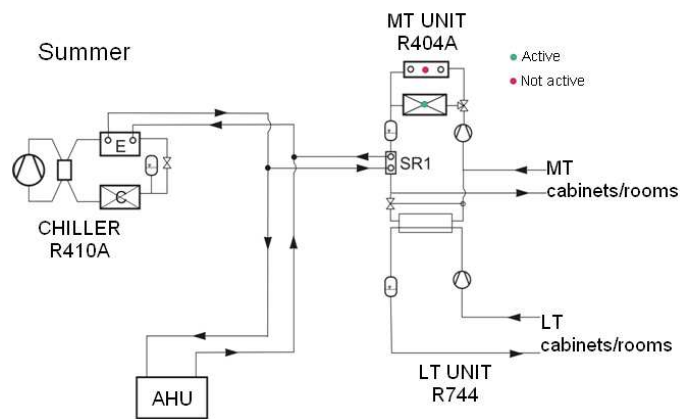


Figure 5

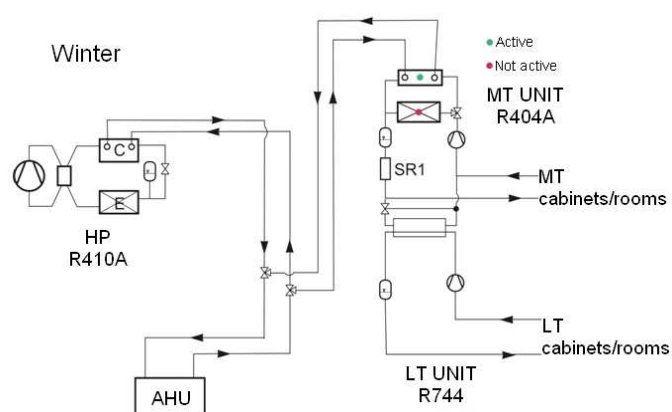


Figure 6

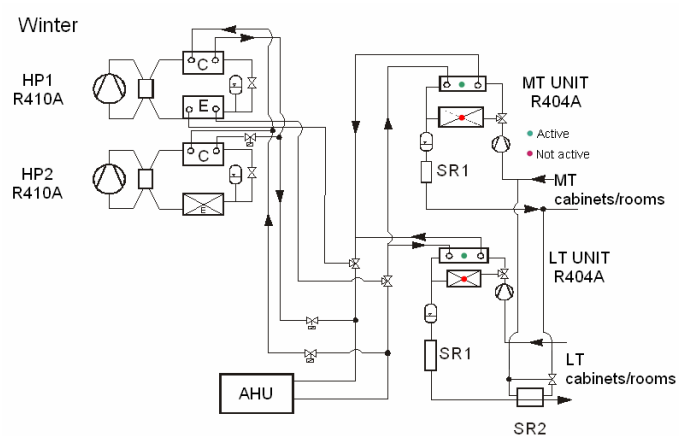


Figure 7

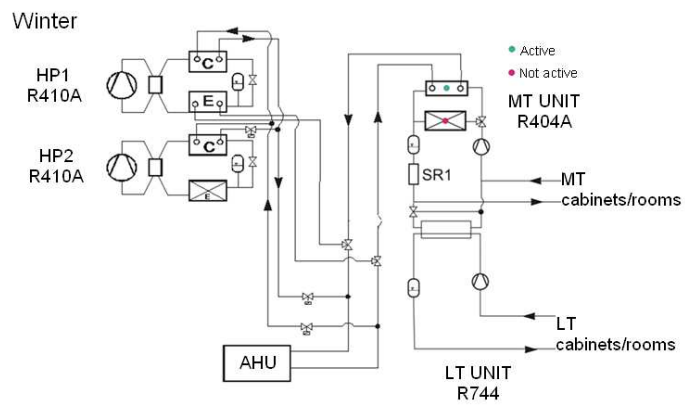


Figure 8

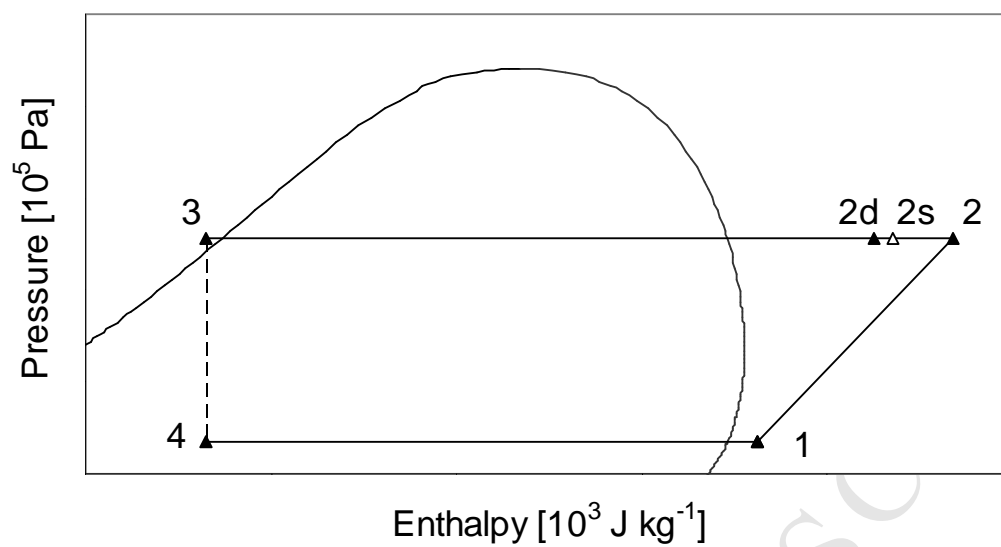


Figure 9



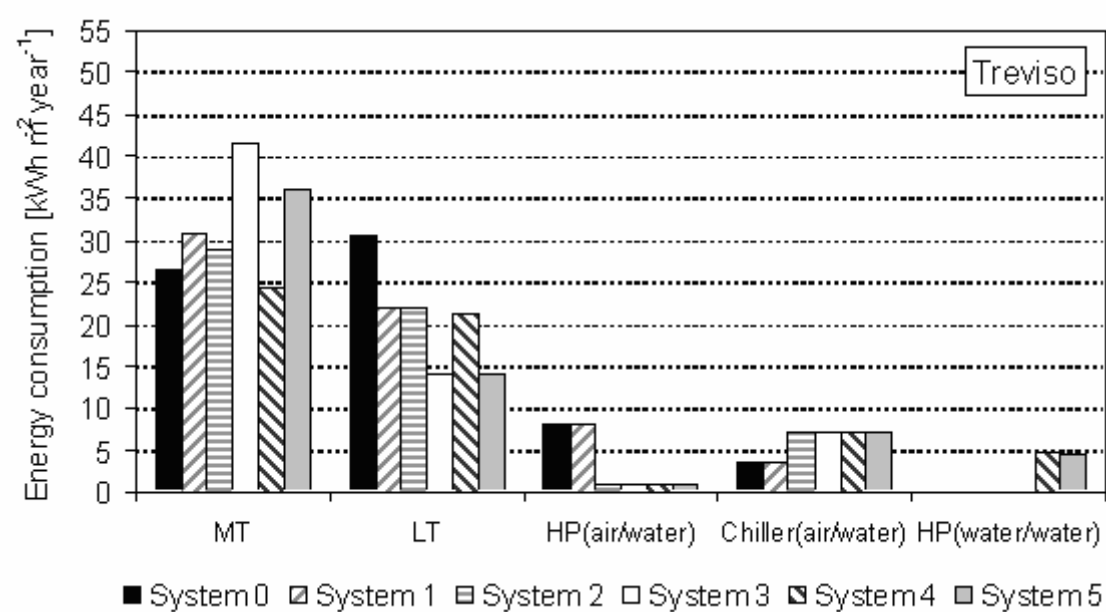


Figure 11

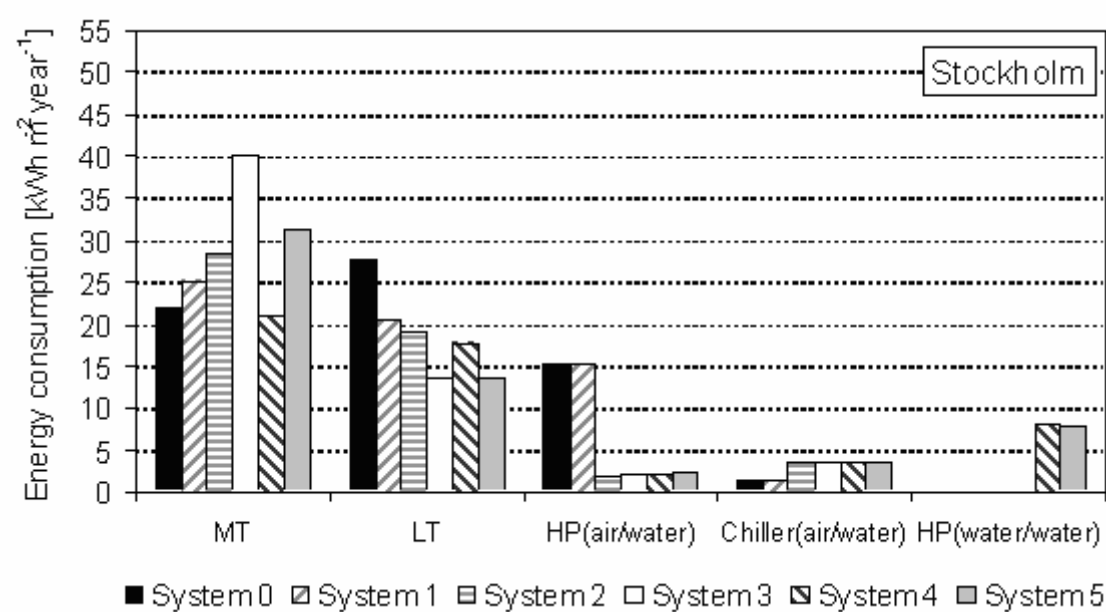


Figure 12

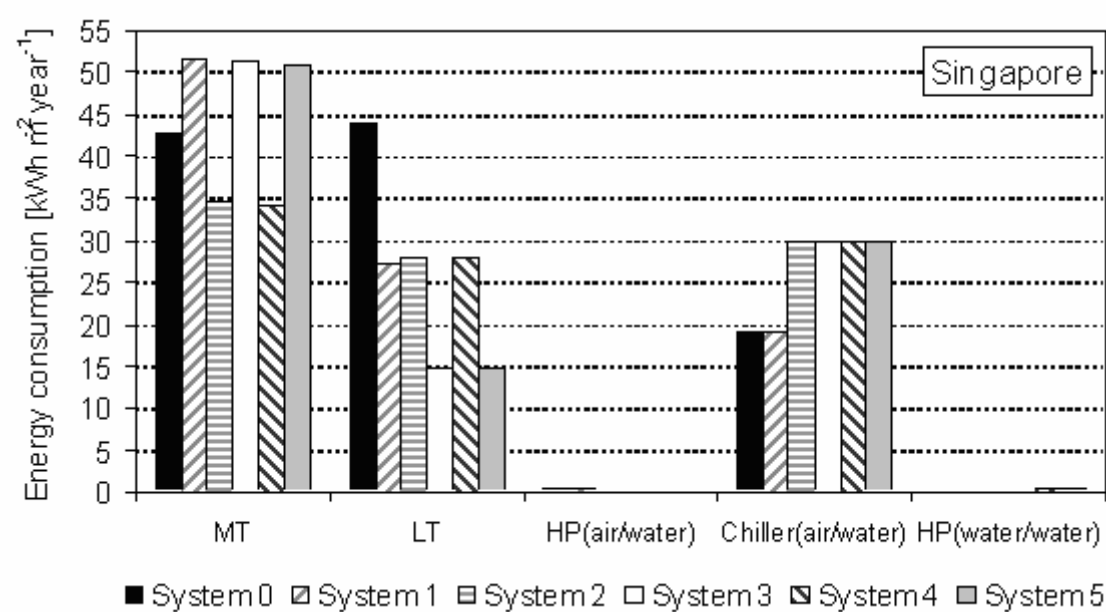


Figure 13

FIGURES CAPTIONS

Figure 1. Baseline System (System 0).

Figure 2. System 1.

Figure 3. System 2. Summertime operations.

Figure 4. System 2. Wintertime operations.

Figure 5. System 3. Summertime operations

Figure 6. System 3. Wintertime operations.

Figure 7. System 4. Wintertime operations.

Figure 8. System 5. Wintertime operations.

Figure 9. Single stage vapour compression cycle in p-h diagram

Figure 10. Annual energy consumption for the analysed systems.

Figure 11. Annual energy consumption for each unit of the analysed systems in Treviso climate.

Figure 12. Annual energy consumption for each unit of the analysed systems in Stockholm climate.

Figure 13. Annual energy consumption for each unit of the analysed systems in Singapore climate.

TABLES

Table 1. Monthly maximum and minimum mean average temperature, LT design load and MT and AHU load ratios for the different climates.

	Max		Min	Q_{LT} [kW]	Load ratios		
	T_{db} [°C]	T_{wb} [°C]	T_{db} [°C]		R_{MT-LT} [-]	$R_{AHUc-LT}$ [-]	$R_{AHUh-LT}$ [-]
Treviso	22.6	17.5	3.32	26.5	2.2	2.6	7.8
Stockholm	17.0	11.6	-3.6	25.5	2.1	2.0	9.3
Singapore	28.5	25.0	26.3	27.5	2.3	5.8	3.2

Table 2. MT and LT display cases. Nominal HER (HER_N)

Display Cabinets	Reference	Lenght	HER_N	Area
MT Cabinet	[-]	[m]	[kW]	[-]
MT vertical, multi-deck	VC2	40.0	41.1	supermarket
MT open, wall site	HC3	22.5	10.2	supermarket
MT open, island	HC4	7.5	3.0	supermarket
LT open, wall site	HF3	7.5	6.0	supermarket
LT vertical, glass door	VF4	20.0	20.0	supermarket
Cool Rooms	Reference	Volume	HER_N	Area
	[-]	[m ³]	[kW]	[-]
MT	-	224.0	33.0	supermarket
LT	-	30.5	4.0	supermarket
MT total [kW]			89.3	
LT total [kW]			30.0	

Table 3. contributions to internal loads (Stefanutti, 2002)

Area	Lighting [W m ⁻²]	Equipment [W m ⁻²]	People [persons m ⁻²]
Cafeteria	30	3	0.20
Shops	15	3	0.10
Supermarket	15	3	0.17

Table 4. Coefficients used in equation (1).

Cabinet Reference	a_1	a_2	a_3	a_4
VC2	$5.08 \cdot 10^{-4}$	$2.46 \cdot 10^{-2}$	$-2.88 \cdot 10^{-3}$	$-1.77 \cdot 10^{-1}$
HC3, HC4	$1.73 \cdot 10^{-4}$	$2.95 \cdot 10^{-2}$	$-3.17 \cdot 10^{-4}$	$3.38 \cdot 10^{-2}$
VF4	0	$2.40 \cdot 10^{-2}$	0	$4.00 \cdot 10^{-1}$
HF3	$8.21 \cdot 10^{-5}$	$1.84 \cdot 10^{-2}$	$6.41 \cdot 10^{-5}$	$4.15 \cdot 10^{-1}$

Table 5. design conditions for MT and LT refrigeration units

System	Season	MT unit				LT unit			
		T_c [°C]	$T_{c,sf,in}$ [°C]	ΔT_{sf} [K]	\dot{V}^\dagger [3600 m ³ s ⁻¹]	T_c [°C]	$T_{c,sf,in}$ [°C]	ΔT_{sf} [K]	\dot{V}^\dagger [3600 m ³ s ⁻¹]
0	winter	-	-	-	130	-	-	-	256
		-	-	-	100	-	-	-	184
		-	-	-	140	-	-	-	265
1	summer	45/37*	35/27*	5	140	45/37*	35/27*	5	265
		-	-	-	166	-	-	-	130
		-	-	-	128	-	-	-	109
2	winter	50	40	5	191	50	40	5	144
		-	-	-	178	-	-	-	139
		-	-	-	206	-	-	-	149
3	summer	45/37*	35/27*	5	231	45/37*	35/27*	5	265
		50	40	5	216	-7	-10	5	14
		-	-	-	203	-	-	-	14
4	winter	20	10	5	90	20	10	5	130
		-	-	-	84	-	-	-	125
		-	-	-	97	-	-	-	135
5	summer	45/37*	35/27*	5	148	45/37*	35/27*	5	265
		20	10	5	137	-7	-10	5	14
		-	-	-	127	-	-	-	14

*: the second value refers to Stockholm design conditions.

†: for each system the three swept volumes refers to Treviso, Stockholm and Singapore respectively.

Table 6. design conditions for liquid chiller/heat pump

System	Chiller/HP 2 (air/water)					Chiller/HP 1 (water/water)				
	T_c	$T_{c,sf,in}$	T_e	$T_{e,sf,in}$	\dot{V}^\dagger	T_c	$T_{c,sf,in}$	T_e	$T_{e,sf,in}$	\dot{V}^\dagger
			[°C]		[3600 m ³ s ⁻¹]				[°C]	[3600 m ³ s ⁻¹]
0	50	40	-10	-3	180					-
					207					
					413					
1	50	40	-10	-3	180					-
					207					
					413					
2	50	35/40*	2/-10*	12/-3*	54					-
					53					
					100					
3	50	35/40*	2/-10*	12/-3*	65					-
					64					
					120					
4	50	35/40*	2/-10*	12/-3*	90	50	40	5	15	86
					88					73
					166					92
5	50	35/40*	2/-10*	12/-3*	58	50	40	5	15	83
					57					70
					106					88

*: the second value refers to Stockholm design conditions.

†: for each system the three swept volumes refers to Treviso, Stockholm and Singapore respectively.

Table 7. design conditions for heat exchangers SR1 and SR2

System	SR1				SR2			
	$T_{r,in}$ [°C]	$T_{sf,in}$	ΔT_{sf} [K]	$\Delta T_{approach}$	$T_{LT,in}$	$T_{LT,out}$ [°C]	$T_{MT,in}$	$T_{MT,out}$
1			-		45/37*	-7	-10	-5
2	45/37*	7	5	3	50	-7	-10	-5
3	45/37*	7	5	3			-	
4	45/37*	7	5	3	20	-7	-10	-5
5	45/37*	7	5	3			-	

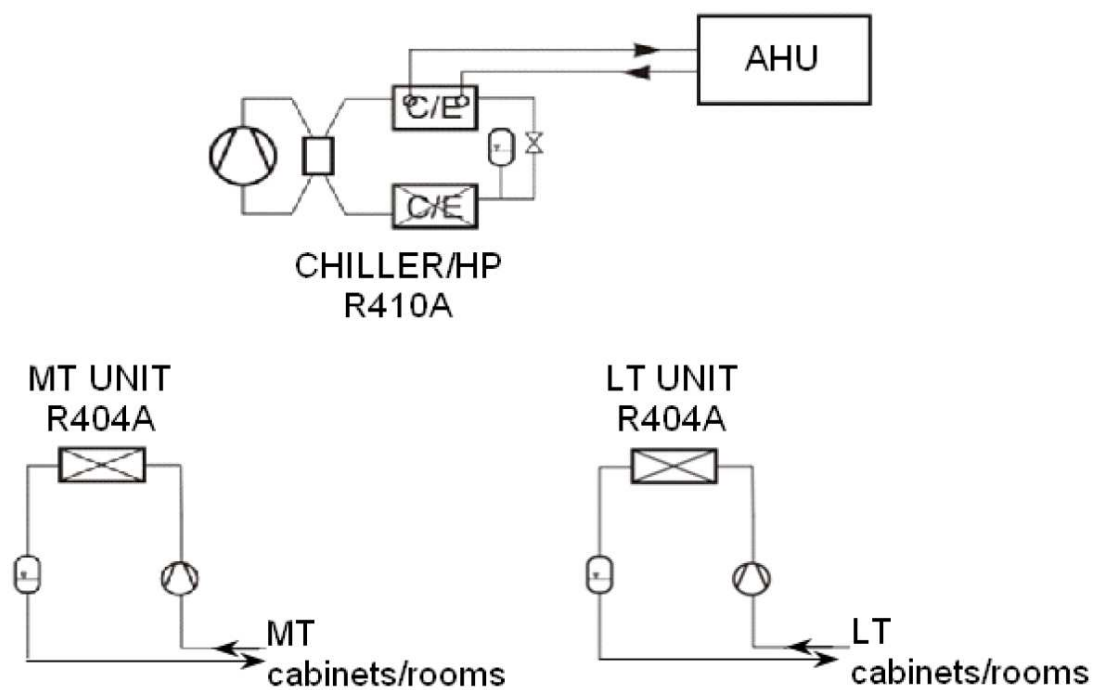
*: the second value refers to Stockholm design conditions.

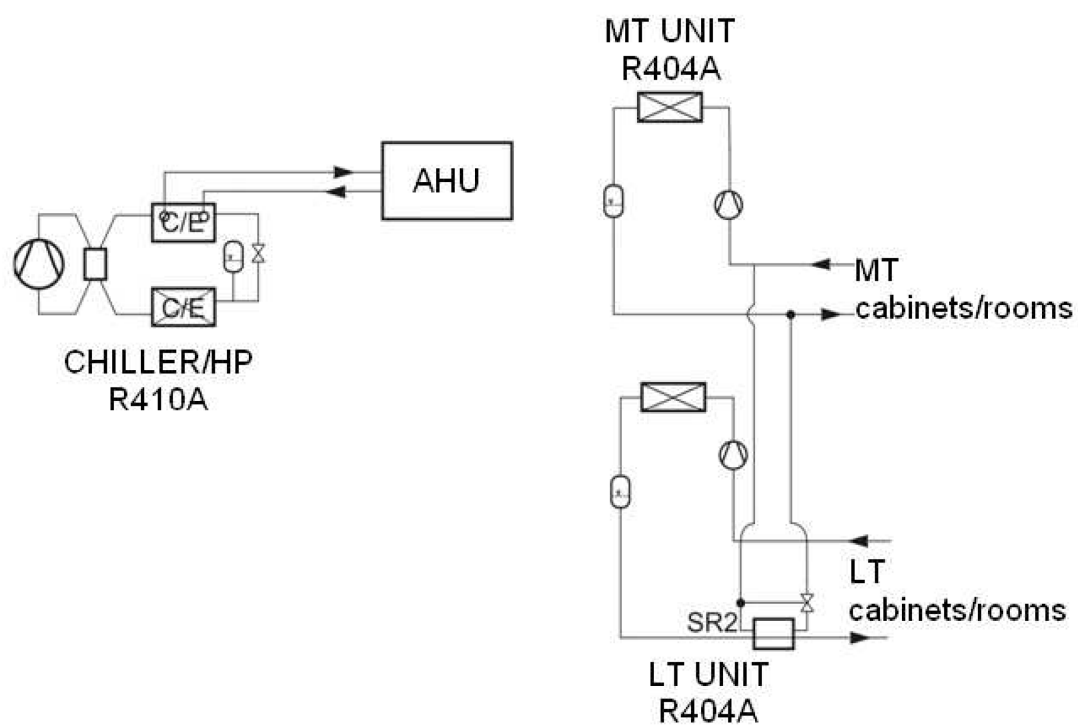
Table 8. Monthly loads and energy consumption for the analysed systems in Treviso climate for constant set point supply temperature.

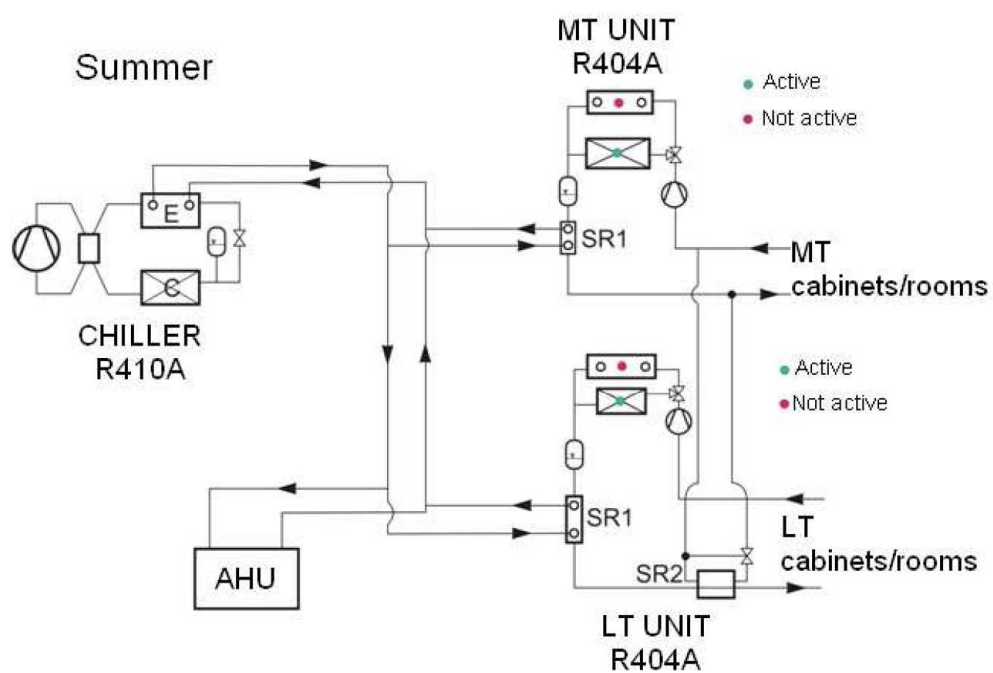
		January	February	March	April	May	June	July	August	September	October	November	December
		Load [kWh]											
	MT	27639	25431	29228	27152	28723	29518	32350	31784	28374	30484	28051	26974
	LT	16272	14823	16665	15899	16592	16472	17456	17320	16197	16952	16068	16148
	AHU – heating	21030	16135	11894	21	0	0	0	0	0	10956	13320	22646
	AHU – cooling	299	528	1942	2898	5946	8909	14128	13470	8058	3463	1014	304
		Energy Consumption [kWh]											
System 0	MT	6345	5890	7065	6989	8257	9341	11330	10974	8896	8016	6719	6200
	LT	8067	7384	8519	8466	9568	10225	11881	11592	9969	9181	8175	8008
	HP(air/water)	6656	5076	3611	1	0	0	0	0	0	2891	3877	7283
	Chiller(air/water)	15	29	195	544	1207	1915	3295	3080	1746	314	54	12
	HP(water/water)	-	-	-	-	-	-	-	-	-	-	-	-
System 1	MT	7272	6738	8074	8066	9657	11041	13497	13085	10539	9200	7672	7121
	LT	6066	5556	6380	6288	6884	7095	7837	7727	6941	6767	6144	6018
	HP(air/water)	6656	5076	3611	1	0	0	0	0	0	2891	3877	7283
	Chiller(air/water)	15	29	195	544	1207	1915	3295	3080	1746	314	54	12
	HP(water/water)	-	-	-	-	-	-	-	-	-	-	-	-
System 2	MT	9748	8473	8829	6472	7518	8268	9593	9367	7889	9444	8768	9853
	LT	6059	5419	6135	5942	6968	7354	8108	8031	7176	6831	5916	6180
	HP(air/water)	794	563	294	0	0	0	0	0	0	110	342	949
	Chiller(air/water)	450	471	871	1456	2520	3607	5765	5371	3357	1284	675	424
	HP(water/water)	-	-	-	-	-	-	-	-	-	-	-	-
System 3	MT	13423	11761	12473	9757	11314	12341	14243	13930	11852	13409	12349	13527
	LT	4162	3795	4276	4077	4265	4249	4520	4481	4172	4364	4123	4127
	HP(air/water)	847	607	323	0	0	0	0	0	0	122	365	1017
	Chiller(air/water)	464	487	883	1453	2489	3566	5735	5331	3318	1277	691	432
	HP(water/water)	-	-	-	-	-	-	-	-	-	-	-	-
System 4	MT	6290	5774	6774	6471	7518	8268	9593	9367	7889	7536	6469	6199
	LT	5411	4917	5756	5942	6968	7354	8108	8031	7176	6478	5492	5489
	HP(air/water)	848	607	326	0	0	0	0	0	0	133	378	1012
	Chiller(air/water)	447	469	869	1456	2520	3607	5765	5371	3357	1282	673	421
	HP(water/water)	3700	2877	2184	0	0	0	0	0	0	2034	2437	3931
System 5	MT	9196	8456	9950	9757	11314	12341	14243	13930	11852	11059	9515	9065
	LT	4162	3795	4276	4077	4265	4249	4520	4481	4172	4364	4123	4127
	HP(air/water)	929	675	374	0	0	0	0	0	0	158	419	1113
	Chiller(air/water)	460	483	880	1453	2489	3566	5735	5331	3318	1273	687	428
	HP(water/water)	3647	2832	2153	0	0	0	0	0	0	2010	2402	3872

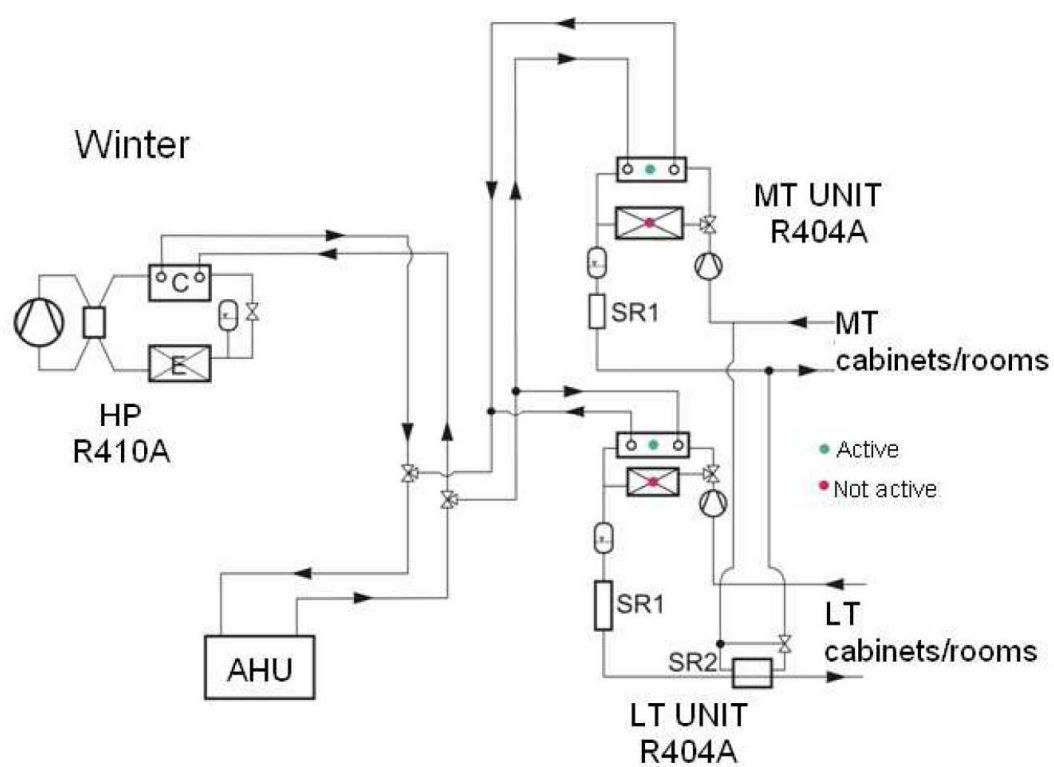
Table 9. Monthly energy consumption for the analysed systems in Treviso climate for floating set point supply temperature.

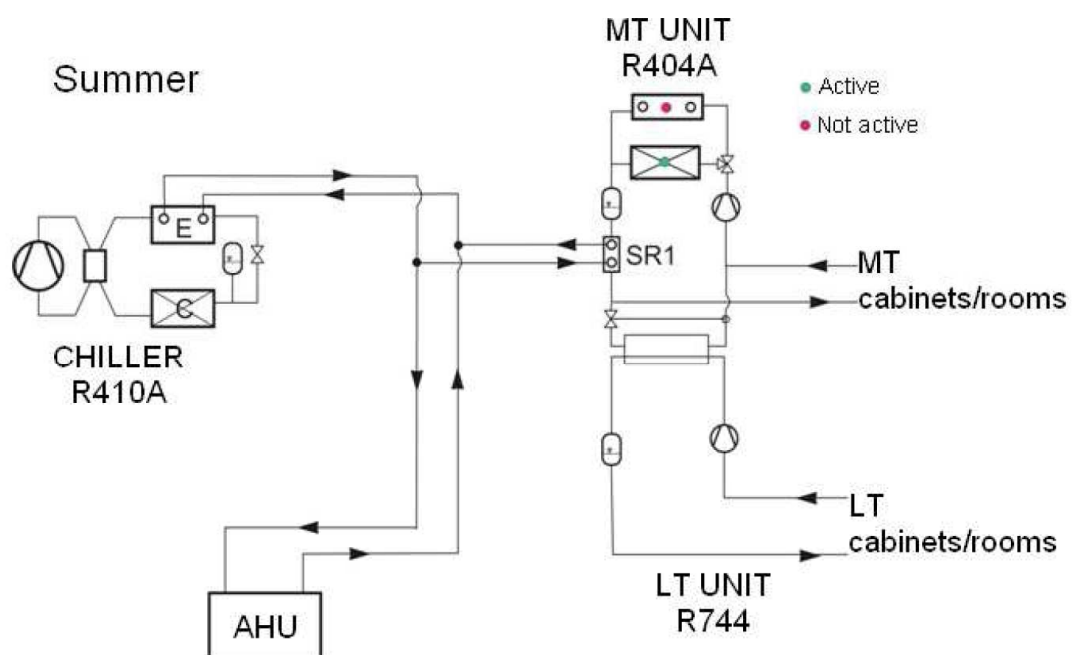
		January	February	March	April	May	June	July	August	September	October	November	December
		Energy Consumption [kWh]											
System 0	MT	6345	5890	7065	6989	8257	9341	11330	10974	8896	8016	6719	6200
	LT	8067	7384	8519	8466	9568	10225	11881	11592	9969	9181	8175	8008
	HP(air/water)	5833	4428	3119	1	0	0	0	0	0	2460	3343	6438
	Chiller(air/water)	12	24	165	459	1025	1642	2910	2710	1500	265	46	10
	HP(water/water)	-	-	-	-	-	-	-	-	-	-	-	-
System 1	MT	7272	6738	8074	8066	9657	11041	13497	13085	10539	9200	7672	7121
	LT	6066	5556	6380	6288	6884	7095	7837	7727	6941	6767	6144	6018
	HP(air/water)	5833	4428	3119	1	0	0	0	0	0	2460	3343	6438
	Chiller(air/water)	12	24	165	459	1025	1642	2910	2710	1500	265	46	10
	HP(water/water)	-	-	-	-	-	-	-	-	-	-	-	-
System 2	MT	9301	8226	8943	7360	8492	9229	10525	10300	8816	9688	8748	9371
	LT	5944	5326	6063	5949	6978	7364	8117	8040	7185	6757	5832	6062
	HP(air/water)	814	584	321	0	0	0	0	0	0	140	367	975
	Chiller(air/water)	125	143	389	798	1672	2642	4657	4277	2435	666	232	121
	HP(water/water)	-	-	-	-	-	-	-	-	-	-	-	-
System 3	MT	12811	11382	12486	10667	12299	13311	15190	14877	12787	13548	12208	12870
	LT	4162	3795	4276	4077	4265	4249	4520	4481	4172	4364	4123	4127
	HP(air/water)	869	630	353	0	0	0	0	0	0	156	391	1047
	Chiller(air/water)	129	147	387	788	1644	2606	4627	4240	2400	652	234	122
	HP(water/water)	-	-	-	-	-	-	-	-	-	-	-	-
System 4	MT	6876	6354	7539	7360	8492	9229	10525	10300	8816	8413	7184	6762
	LT	5424	4926	5764	5948	6978	7364	8117	8040	7185	6486	5500	5501
	HP(air/water)	833	597	326	0	0	0	0	0	0	140	374	999
	Chiller(air/water)	126	144	389	798	1672	2642	4657	4277	2435	667	233	121
	HP(water/water)	3118	2408	1812	0	0	0	0	0	0	1667	2019	3338
System 5	MT	9799	9054	10733	10666	12299	13311	15190	14877	12787	11937	10250	9637
	LT	4162	3795	4276	4077	4265	4249	4520	4481	4172	4364	4123	4127
	HP(air/water)	909	659	372	0	0	0	0	0	0	166	411	1095
	Chiller(air/water)	129	147	388	788	1644	2606	4627	4240	2400	652	234	123
	HP(water/water)	3075	2373	1787	0	0	0	0	0	0	1654	1995	3288

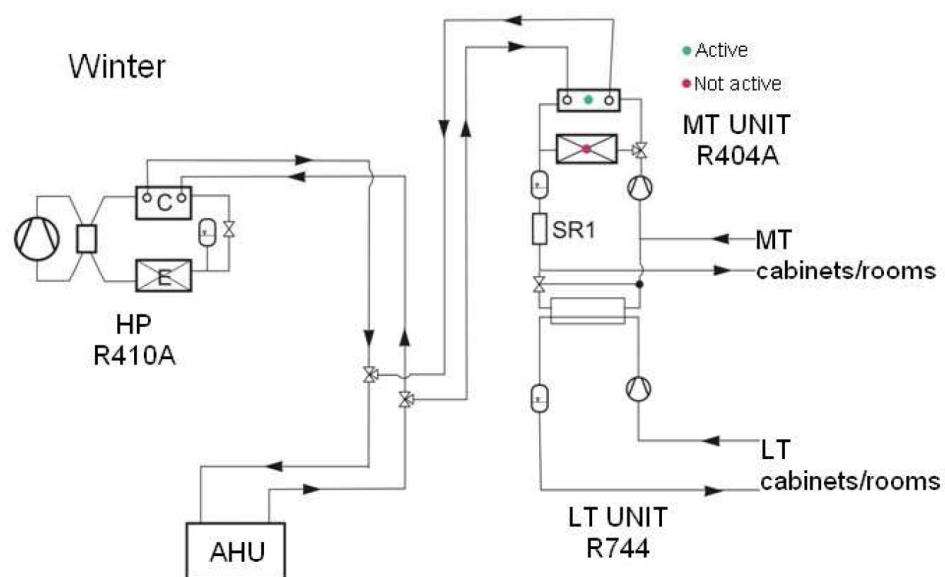


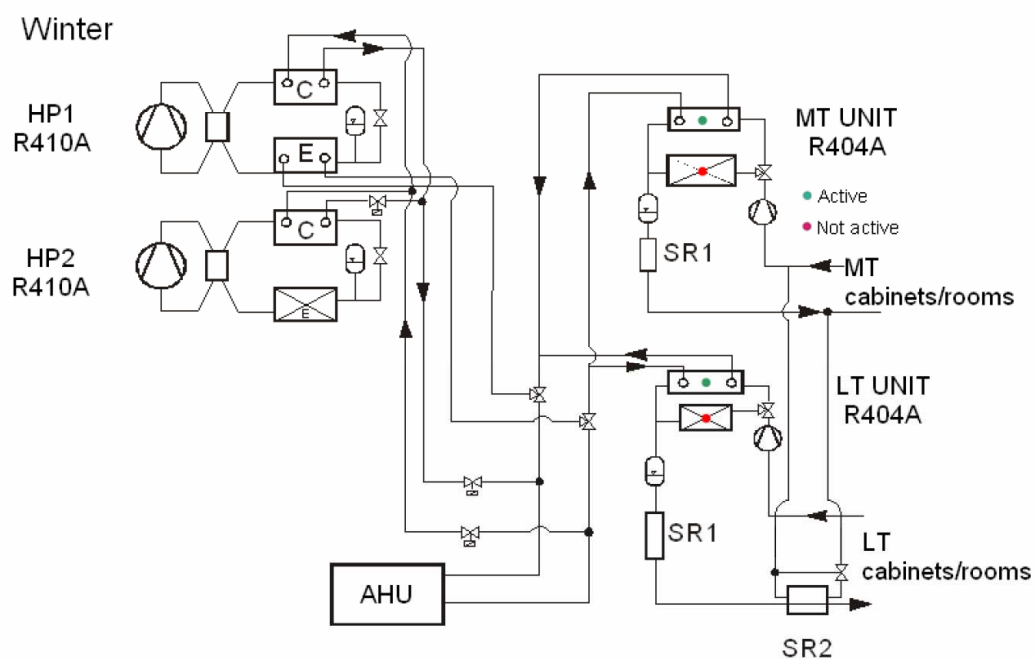












Winter

