
High Efficiency Mix Energy System Design with Low Carbon Footprint for Wide-Open Workshops

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Additional information is available at the end of the chapter

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1. Introduction

Existing buildings are a key focus in the EU's strategy to reduce its member countries emissions. As a consequence, the EU Directive for improving energy performance of buildings (Directive 2002/91/EC, 2003; Directive 2010/31/EU, 2010) was brought in to guide member states. They are advised to upgrade the energy performance of the existing buildings with a floor space of more than 1,000 m², when they are reformed. This can be achieved through improving the energy performance of its service systems (heating, cooling, ventilation, hot water, and lighting systems) and of the materials that conforms its architectural envelope.

Of the existing buildings, industrial sector has a significant potential for energy savings. Facilities in many cases obsolete, old comfort standards and an out-of-date yet demanding labour legislation (Ley 31/1995, 1995; R.D. 486/1997, 1997; Instituto Nacional de Seguridad e Higiene en el Trabajo, 1997) provide the framework in which refurbishment must be accomplished. Especially critical is the case of partially open buildings whose entrances, due to the use, remain open during long periods of time: warehouses, open wide while loading and unloading, or cars and trucks repair workshops with continuous vehicular traffic are two examples of industrial buildings of the type. Refurbishment in this case needs to be faced with different criteria than those which are customarily used for the remainder of buildings.

Firstly, the poor indoor environmental conditions achieved, very close to outdoors climate due to the high permeability of the building, represent a significant economic cost to the owners, as the activity of workers, and therefore the length of the working day, is linked to acceptable indoor hightothermal conditions.

Article 7 and Annex III of R.D. 486/1997 (1997) set out the environmental requirements that are mandatory in work places for health and safety reasons: temperature must be kept between 17°C and 27°C if sedentary work is performed, and within a range of 14°C up to 25°C in the case of light work. Air velocities must be less than 0.25 m/s when working in cool environments, below 0.5 m/s in warm environments for sedentary jobs and lower than 0.75 m/s if the work done is light with a warm environment. In all cases the relative humidity has to be maintained between 30% and 70%.

Even if the extreme values of the range are considered (temperatures in the range of 14°C and 27°C with air velocities below 0.75 m/s) the possibility of achieving these higrothermal indoor conditions without mechanical equipment in an open building exposed to a severe climate, as it happens in most of European countries, is very low.

Nevertheless, in this type of buildings, HVAC systems that control temperature and humidity inside the premises are not feasible, for the high rate of infiltration would lead to extreme cooling and heating loads, as it will later be demonstrated by some examples.

If, despite what has been exposed, it is found necessary to project a technical system that improves thermal conditions for labourers while increasing the working hours, the first problem to be solved is to establish an effective climate separation of indoor climate that allows to obtain thermal comfort at a reasonable cost. In this sense, the provision of air curtains in fixed openings is an essential strategy prior to air conditioning the premises.

It should be noted that the climate separation, though reducing building demand of energy by nearly 90 %, causes yet another problem: the indoor air quality worsens, as natural ventilation due to air infiltration through the openings tends to be neglected. At this point, when the need for mechanical ventilation becomes clear, the inevitably decision of air conditioning is derived, and so is the necessity to establish the basis for choosing the best technical system that ensure the higrothermal comfort of the occupants and the air quality of the premises as well as minimize both the installed thermal power and the energy consumption associated with conditioning.

Therefore, this chapter seeks to propose a methodology for facing the refurbishment of wide-open industrial buildings. Firstly, by establishing a climatic separation via air curtains and, finally, by choosing a high efficiency air conditioning system. The expected benefits of the proposed system will be:

- An increase in hours of work, taking into account the existing labour legislation;
- An improvement of comfort, compared with a conventional system; and
- A reduction of energy consumption.

2. Strategies for carbon emissions reduction in the refurbishment of wide-open industrial buildings

As it was mentioned in the previous section, in the rehabilitation of wide-open industrial buildings, at least three problems, related one to another, must be faced. Namely:

- Energy consumption;
- Higrothermal comfort; and
- Indoor air quality.

A holistic approach to building design suggests a long list of possible strategies to improve its energy efficiency. However, to guide the development of a more efficient HVAC system, the concept of adaptive comfort criteria was used (Clark & Edholm, 1985; Nicol, 1993).

What this means in practice is that less fossil fuel is used to maintain comfortable temperatures if the building can be kept to a relatively constant level through an interactive control system that adapts the internal environment conditions in response to carbon dioxide levels and air velocities. In order to reduce the carbon emissions of the heating and ventilation system the following steps need to be taken:

2.1. Thermally isolate the building

This includes solar shading as well as an improvement of the insulation materials, as it derives from the following reasons.

With respect to buildings heating and cooling energy demand, Spanish legislation (R.D. 314/2006, 2006a) states:

" Buildings shall have an enclosure that adequately limit the energy demand required to achieve thermal comfort, depending on the local climate, the use of the building during summer and winter, as well as on the characteristics of isolation and inertia of the materials, the air permeability and the exposure to solar radiation, and adequately considering thermal bridges, in order to properly limit heat gains or losses and to avoid the higrothermal problems related to them"

Thermal characterization of the opaque elements of the building enclosure (walls, roofs and floors) is made by the thermal transmittance U ($\text{W}/\text{m}^2 \cdot \text{K}$), which is defined:

$$\frac{1}{U} = R_T = \frac{1}{h_i} + \sum_{i=1}^n \frac{e_i}{\lambda_i} + \frac{1}{h_e} \quad (1)$$

with h_i , surface heat transfer coefficient for the inside air layer ($\text{W}/\text{m}^2 \cdot \text{K}$); h_e , surface heat transfer coefficient for the outside air layer ($\text{W}/\text{m}^2 \cdot \text{K}$); e , thickness of the layers that forms the enclosure (m); and λ , thermal conductivity of the material of each layer ($\text{W}/\text{m} \cdot \text{K}$).

With respect to the openings (windows, doors and skylights), thermal transmittance is also used. In this case it is obtained from the respective transmittances of glass, U_v , and window frame, U_m , according to the expression.

$$U_H = (1\text{-FM}) U_v + \text{FM} U_m \quad (2)$$

being FM (%) the fraction of the opening taken up by the frame.

Due to the high contribution of the openings to the heat gains, an additional coefficient is used in order to characterize its response to the solar radiation. It is the modified solar factor, F (-), defined as:

$$F = F_s \cdot [(1 - FM) \cdot g + FM \cdot 0,04 \cdot U_m \cdot \alpha] \quad (3)$$

where F_s (-) is called shadow factor, which is defined as the percentage of the solar radiation incident on a vertical plane that eventually reaches the opening. Its value is affected by remote obstacles, the self-shadowing of the building, facade obstacles like setbacks, overhangs or projections, and the sun control devices, fixed or movable exterior shades included.

The whole expression represents an average solar factor of the opening, taking into account the aforementioned effect of shadowing, and the weighted contribution of glass and frame in the response to solar radiation. The contribution of the glass is expressed by its total solar thermal transmittance, g (-), determined, for a quasi-parallel radiation and for a quasi normal inclination, with the definition formula (BS EN 410:1998, 1998).

$$g = \frac{\alpha h_i}{h_e + h_i} + \tau \quad (4)$$

where α is the absorptivity and τ the transmissivity, both dimensionless. This factor is obviously much higher than the frame one, which is representative only when having small openings or thick window frames.

Finally, with respect to the thermal inertia of the envelope, in buildings such as those discussed in this chapter, with light walls, its influence can be neglected.

2.2. Produce hot water more efficiently

In addition to the energy demand, which has been analyzed in the previous section, the average performance of the HVAC systems is the determining factor in the final energy consumption of buildings. From all the subsystems that takes part in the air conditioning (heat emission, distribution and production), the latter has the higher incidence in the energy efficiency of the building.

When producing heat by means of combustion, two aspects of boiler design have to be considered: the heat losses and its efficiency. What is desired is a highly efficient boiler system which minimizes the heat losses, especially those associated to combustion gases, so that less fuel is needed to heat the water. On the other hand, the boiler needs to be as effective as possible at transferring heat from the energy source to water.

The method used to evaluate the final energy consumption of the heat production system is:

- Annual energy needed by the boiler to meet the demand = Annual energy demand x (1+Boiler losses)
- Total annual energy consumption for the boiler and its fuel = Annual energy needed by the boiler to meet the demand / (Calorific potential x Seasonal efficiency)

Boiler losses depend on the type of heat generator and on the range of power output. According to the European legislation (BS EN 15603:2008, 2008), boilers must comply with the heat conversion efficiency requirements, always referred to the fuel net calorific value, that are set out in the following Table 1:

Efficiency at rated output					
Type of boiler	Range of power output (kW)				
	50	100	200	300	400
Standard	87.4	88.0	88.6	89.0	89.2
Low temperature	90.0	90.5	91.0	91.2	91.4
Gas condensing boilers	92.7	93.0	93.3	93.5	93.6
Efficiency at partload (30%)					
Type of boiler	Range of power output (kW)				
	50	100	200	300	400
Standard	85.1	86.0	86.9	87.4	87.8
Low temperature	90.0	90.5	91.0	91.2	91.4
Gas condensing boilers	98.7	99.0	99.3	99.5	99.6

Table 1. Normative heat conversion efficiency requirements for different types of boilers

On the other part, seasonal efficiency is difficult to determine, as it is related to the size of the boiler, the burner type and the method of operation all over the heating session. A simple way to approach the problem can be consulted in (R.D. 275/1995, 1995). To obtain more accurate results, the use of energy simulation programs is strongly recommended.

2.3. Reduce the amount of carbon from the energy source used by the system

Primary energy is an energy that has not been subjected to any conversion or transformation process. Thus, to determine the primary energy required to provide the final energy demanded by a technical system, it is taken into account the energy content associated with the extraction, processing, storage, transport, generation, transformation, transmission, distribution and any other operation necessary to supply energy to the area where it is used. Primary energy is the essential energy indicator to determine the net heat balance of a building (when heat production involves different energy vectors), or to compare the energy performance of different technical systems. As such is considered by European official methodology of calculation (Moss, 1997), and has been accordingly incorporated into rating procedures for energy building performance of different member countries, including Spain (R.D. 47/2007 (2007)).

It is calculated by multiplying the energy supplied to the system by a factor greater than unity. If, however, the carbon emission is preferred as indicator, a second conversion factor has to be applied.

According to it, the method used to evaluate the carbon emission due to the energy source for the heat production is:

- Annual energy consumption of the primary fuel used in the boiler = Annual energy needed by the boiler x Coefficient for the primary energy used.
- Amount of carbon emitted = Annual energy consumption of primary fuel x Carbon conversion factor.

The following Table 2 shows the conversion coefficients used by the official program for energy rating process in Spain, CALENER (IDAE, 2009). Each different energy source has two energy conversion factors, for carbon emissions and primary energy.

Type of energy	Final energy (kWh)	Primary energy (kWh)	Emissions (kg. CO ₂)
Electricity	1	2,603	0.649
Natural Gas	1	1,011	0.204
Coal	1	1	0.347
Liquefied petroleum gas	1	1,081	0.244
Diesel oil	1	1,081	0.287
Fuel oil	1	1,081	0.280
Biofuel	1	1	0
Renewable energy	¿1?	¿1?	0

Table 2. Carbon emission and primary energy conversion factors used by CALENER

Therefore, a significant reduction of carbon emissions can be carried out in two ways. The first is to choose a primary source of energy with the lowest carbon footprint, eg. natural gas. The second is to use a source of renewable energy.

2.4. Separate outdoor and indoor environments by means of a thermal barrier if no physical limit exists between them

Through the openings of building walls and partitions there are inlets and outlets of air, whose direction and value depend, for each point of the envelope, on the pressure differences that exists at each of its sides. In the absence of mechanical systems, the pressure difference is due to the temperature difference between indoor and outdoor air (thermal draft) or to the wind pressure.

The airflow through an opening due to thermal buoyancy has been studied by different authors (Allard & Utsumi, 1992). The air flow rate Q in m³/s which enters through an opening h meters high and with a surface of S m² when there is a temperature difference $T_i - T_e$ between the outside air and the inside air can be obtained from the following expression:

$$S = \frac{Q}{g \rho_0 T_0 \left(\frac{1}{T_e} - \frac{1}{T_i} \right) z^{1/n}} \quad (5)$$

where $z = h/2$ (considering that the neutral pressure level passes through the centerline of the side opening), and n is the flow exponent that indicates the degree of turbulence. An n value of 0.5 represents fully turbulent flow and 1.0 represents fully laminar flow.

From the volume flow rate obtained, the sensible and latent losses (or gains) are calculated using the well known expressions:

$$\phi_s = Q \cdot (T_e - T_i) \cdot Q_a \cdot C_a \quad (6)$$

$$\phi_{Lv} = Q \cdot (w_e - w_i) \cdot Q_a \cdot L \quad (7)$$

Heating (or cooling) loads laws distribute at both sides of the neutral pressure level according to the Figure 1:

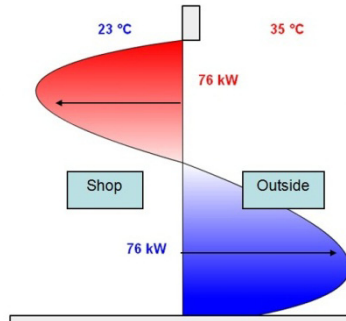


Figure 1. Heating loads due to thermal buoyancy through openings

According to the technical services prevailing code (R.D. 1027/2007, 2007), the design of a partially open building, with 8 side openings of 4x4 m² each, which keeps the indoor environment between 19°C and 28°C throughout the year, would require a heating power close to 2.5 MW if the outdoor temperature were 5°C. In summer, sensible cooling power reaches 1.5 MW for an outside temperature of 35°C.

At this point it should be emphasized the importance of the latent loads in warm and humid climates. In these cases, the use of direct expansion cooling machines leads to high electrical consumption for hardly maintaining the desired internal temperature conditions. This fact alone would justify the use of climate separators in these climates, for all type of buildings, provided that a high frequency of doors opening and closing is expected.

It should also be noted that the infiltration air flow rate, and consequently the thermal load, is usually increased due to wind pressures acting on opposite faces of the building. In this case, even more so air curtains reveal as the essential strategy to reduce the thermal load of this type of buildings, for they create a barrier between the two environments, the inside working area of the workshop and the outside air. The air curtains also are highly energy efficient terminal units, and their coils permit a choice of the type of energy they run on, hot water included.

2.5. Reduce the causes of discomfort

The degree of user acceptance of the environmental conditions of the premises is a function of their air quality, which includes higrothermal comfort and adequate pollution levels.

There is enough information about the influence of higrothermal conditions on the predicted percentage of dissatisfied (PPD) in a room (Fanger, 1993a). The expressions that relate the PPD with different causes of higrothermal discomfort are the following:

a. PPD due to vertical temperature gradient

The *PPD* as a function of temperature gradient is given by the equation (ASHRAE, 2009a):

$$PPD = \frac{100}{1 + e^{(5.76 - 0.856 TG)}} \quad (8)$$

where *TG* is the vertical thermal gradient between head and feet.

b. Synergistic effect on PPD of air velocity, temperature and turbulence

The changes in the *PPD* by the synergistic effects of air velocity, temperature and turbulence, are defined by the expression (BS EN ISO 7730:2005, 2006a, 2006b):

$$PPD = (34 - T_a) \cdot (v - 0.05)^{0.62} \cdot (0.37 \cdot v \cdot T_u + 3.14) \quad (9)$$

where: T_a is the air temperature (°C), v is the air velocity (m/s) and T_u is the turbulence intensity (%), considering that the air turbulence intensity in a point is defined by the equation:

$$T_u = \frac{\sigma_v}{\bar{v}} \quad (10)$$

where σ_v is the standard deviation and \bar{v} is the mean air velocity of a random sample of velocities.

c. PPD due to inadequate floor temperature

The percentage of dissatisfied by warm or cold floor can be deduced through the following expression (BS EN ISO 7730:2005, 2006c), that relates the *PPD* to the floor temperature t_f .

$$PPD = 100 - 94 e^{(-1.387 + 0.118 \cdot t_f - 0.0025 \cdot t_f^2)} \quad (11)$$

d. PPD due to asymmetric thermal radiation

Asymmetric radiation from warm or cold surfaces, created by high lighting levels, due to large glazed surfaces or direct sunlight can reduce thermal acceptability of the spaces (BS EN ISO 7730:2005, 2006d). Radiant temperature asymmetry is analysed to 1.1 m off the ground for standing and 0.6 m for seating conditions, and must be kept under the following limits (Table 3).

Element	Cold wall	Cold roof	Warm wall	Warm roof
Maximum temperature differences	11	16	35	6

Table 3. Values of asymmetric radiation for a $PPD \leq 10\%$

With respect to air quality, the effect on the PDP of the contaminants perceived by the sense of smell has also been extensively discussed (Fanger, 1993b), but it does not when it comes to pollutants that are not detected by humans, for it is unknown its influence on the PPD and how affects the workers productivity.

e. PPD due to human detected pollutants

Fanger established how the percentage of people dissatisfied is influenced by the presence of pollutants, provided that they can be perceivable. For low concentrations (usual when working indoors) the relationship is linear. For low concentrations (usual working indoors) the relationship is linear.

Of pollutants detected by humans, one of the most common is carbon dioxide (CO_2). The relationship between the PPD and the concentration of CO_2 in the air can be determined through the expression (Fanger, 1993c):

$$PPD = 395 e^{\left[-15.15 \cdot (C - 350)^{-0.25} \right]} \quad (12)$$

where C is the carbon dioxide CO_2 concentration in ppm.

From the previous expression it can be deduced:

- For $PPD = 100\%$, CO_2 concentration is 10.000 ppm (350 ppm supplied by the outside air).
- For $PPD = 30\%$, PPD/CO_2 relationship is almost linear.
- For $PPD = 15\%$, concentration is 850 ppm (500 ppm produced in the inside the premises, and 350 ppm supplied by the outside air).

f. PPD due to any other pollutant

Under the hypothesis that contaminants not perceived could be studied under the aforementioned theoretical frame, so that their effect on humans could be brought together and their contribution to the PPD could be analyzed, Gomez (2009) has developed a method for quickly and easily determine the PPD for any room with any type of pollution (detected or not by humans). It is based upon the concentrations of pollutants prescribed by international health agencies, for which there is adopted the type of curve derived by Fanger for CO_2 as an average value.

ACGIH (1999) and the existing standards on health and hygiene at work suggest the maximum allowable concentration of pollutants for different periods of time TLV (Threshold Limit Values). According to them, being exposed to a specific concentration of pollutants for 8 hours a day leads to a PPD of 100%. This concentration is known as the TLWA value.

By analogy with the case of CO₂, whatever the type of pollutant studied, it can be constructed a function that relates its concentration with the caused *PPD*, since three points of it are known. Firstly, the curve passes through the origin. Secondly, it is also known the concentration which produces a 100% *PPD*. And finally, according to the experience for CO₂, a *PPD* of 15% is reached with a concentration 11.76 times lower than the one which causes a *PPD* of 100%.

The error made with this assumption depends on the shape of the curve that relates *PPD* to concentration of each pollutant. In the case of the substances for which ACGIH provides the permissible limit for TLV-STEL (concentration at which users may be exposed continuously for 15 minutes without chronic or irreversible damage) and TLV- TLWA, they are related within a range from 1.5 up to 6.

NIOSH REL provides a value of 6 for the relation between permissible concentrations of CO₂ for 8 hours and 15 minutes (ASHRAE, 2009b), while German list of MAK (Maximale Arbeitsplatzkonzentration) values (Deutsche Forschungsgemeinschaft, 2007) gives a relation value of 2.

It then follows that it can be adopted for the rest of pollutants the type of curve derived by Fanger for CO₂ without making significant errors. It is expected that further investigations in this field could provide new data. For permanent flow, as the *PPD* of the interior of the premises takes low values, it is only used the straight part of the curve, in which for a *PPD* of 15%, concentration is 1/11.76 of that corresponding to 100%.

Through "in situ" measurements, or prediction by means of CFD models, of the concentration of different contaminants for specific points of the room, it can be deduced the *PPD* achieved under any circumstance. This method can be used whatever the air diffusion system considered, though in the next section it will be discussed the advantages and drawbacks of the usual systems.

When it comes to know the indoor air quality of a room it is essential to determine the influence of ventilation air flow rate in the *PPD*. Considering that its effect on higrothermal conditions, and eventually on the *PPD* achieved, is opposite to pollutants concentration (Figure 2), the authors Castejon et al. (2011) and Galvez-Huerta et al. (2012) have recently studied the problem of settling the ventilation rate that minimizes the predicted percentage of dissatisfied in a room. The final aim is that this result in energy savings, for it minimizes the ventilation air flow rate.

In wide-open buildings, draughts and unwanted air currents can be a constant feature when the door opens automatically, affecting higrothermal comfort. At the same time, discomfort if there is a significant difference in temperature of their feet and their head can be fairly controlled through an air diffusion system.

With respect to the existing sources of contaminants, the rate of air change will depend on the carbon dioxide and monoxide emissions.

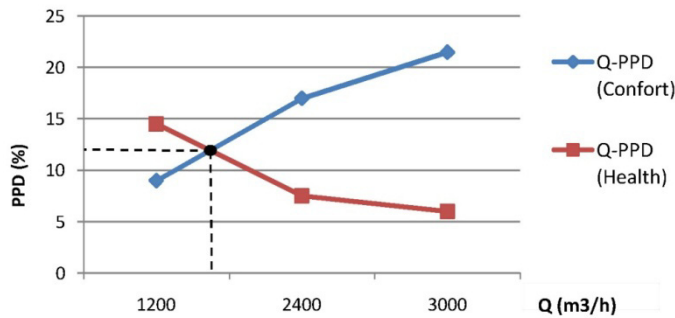


Figure 2. PPD due to combined action of higrothermal and health factors

2.6. Election of an adequate diffusion air system

The effectiveness of air renewal provided by the ventilation system is the essential parameter for the design of an air diffusion system. Effectiveness, r_v (-), of a ventilation system is defined with the expression:

$$r_v = \frac{C_{extr} - C_{imp}}{C_{zona} - C_{imp}} \quad (13)$$

with C_{imp} , contaminant concentration in supply air; C_{ext} , contaminant concentration in exhaust air; C_{zona} , contaminant concentration in the occupied zone.

As it is shown in Table 4, among the systems used in air diffusion, , used in its early stages only in industrial premises (Baturin, 1972), shows as the air diffusion system that eliminate sources of contamination most efficiently, especially when air is driven at a lower temperature than the indoor air. This always occurs during cooling period, and also with ventilation air in winter.

Mixed mode		Displacement mode	
ΔT	r_v	ΔT	r_v
< 0	0.9 ... 1	< 0	1.2 ... 1.4
0 ... 2	0.9	0 ... 2	0.7 ... 0.9
2 ... 5	0.8	> 2	0.2 ... 0.7
> 5	0.4 ... 0.7		

Table 4. Ventilation effectiveness dependence on air diffusion mode and difference of temperatures

With this systems air is supplied at very low velocity near the floor, within the occupied zone (Nielsen, 1993). This is, then, the coolest zone of the room, and forces a profile of temperature and pollutant concentration which vertically increases up to the roof (Figure 3), where the return takes place.

When the emission of pollutants is uniformly distributed, as in places where human presence is dominant, the system design is based on the control of the temperature gradient (Mundt, 1995) to maintain comfort conditions in the occupied area: Namely, air temperature at 1.80 m high, maximum air velocity in the occupied zone, radiant asymmetry from the ceiling, walls and floor, and maximum temperature difference between the head and feet.

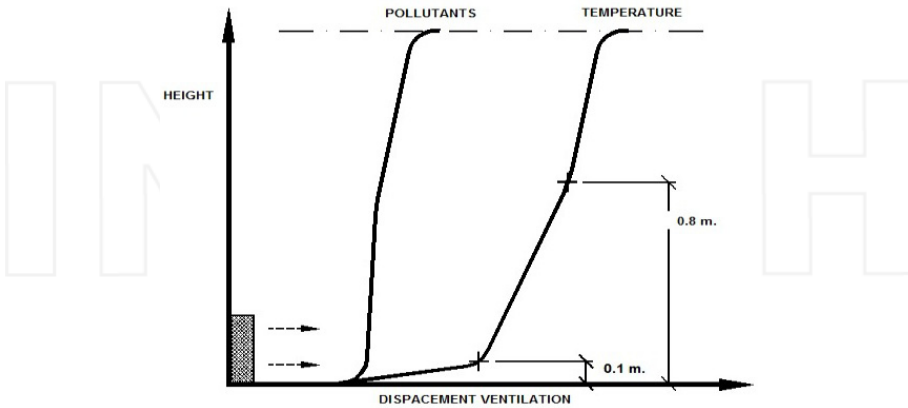


Figure 3. Temperature and concentration of contaminants gradient

When, on the contrary, the emission of pollutants is concentrated, as in areas that have car engines running, in this case the cycle of air change will be greater than in those areas where less strenuous work is carried out. This allows to reduce the ventilation air to the minimum required, thus saving energy.

2.7. Use of Dedicated Outdoor Systems (DOAS)

As some authors (Mumma, 2001) have expressed, a new paradigm in the design of HVAC systems is in its early stages. Amidst its requirements, the more remarkable are: separating outdoor air from the air conditioning system to ensure adequate ventilation and the use of energy recovery strategies.

Despite the fact that the new systems can also handle the space latent load and part of its sensible load, the transition from mixed ventilation systems to DOAS always compels to use a second air conditioning system (be it passive or active beams, water or direct expansion fan coils, ceiling cooling panels, radiant floors and thermoactive surfaces). This drawback of having duplicated systems is usually compensated with the use of high efficiency terminal units, which run on low temperature water, closer to the indoor air conditions.

However, it is usually forgotten that the present way towards more efficient buildings, with the commissioning Nearly Zero Energy Building by 2020 (Directive 2010/31/EU, 2010), has produced substantial improvements in the envelope and lightning systems, which have consequently reduced their contribution to cooling loads. But these improvements, while reducing consumption, also involve a decrease of the thermal load met by the coils.

Consider that, with an average occupancy of the building of 5 m²/person, the Spanish mandatory ventilation air flow rate, supplied at 10°C less than indoor air, can meet a cooling load of 30 W/m². This may seem certainly low, but is close to the expected for the coming years.

In the case of partially open workshops, the primary air handling unit can guarantee the quality of air in the building with a reasonable low energy consumption (for they supply air with indoor conditions) only if climate separation in openings has been previously solved. When extreme winter conditions have to be faced, primary air conditioning can be supplemented with water terminal units in the weakest points of the perimeter, the doors. In this sense, the placement of an air curtain in each opening can accomplish both tasks, reducing air movement within the premises to the necessary to remove contaminants.

2.8. Maximize the cost-effectiveness of heat recovery equipment

The Spanish legislation (R.D. 1027/2007, 2007) makes it mandatory for air conditioning systems of a certain size the following specific strategies for energy recovery, framed in a more general requirement of energy efficiency:

- Free cooling by outside air, which applies to constant volume or VAV air conditioning systems with rated cooling output greater than 70 kW.
- Recovery of heat from ventilation air with adiabatic cooling of the extract air. This requirement is applicable to buildings in which the exhaust air flow is greater than 0.5 m³/s. It also establishes minimum efficiency of recovery of sensible heat depending on the air flow rate and the number of hours the system is operating throughout the year (Table 5).

Annual operating hours	Outdoor air flow rate (m ³ /s)				
	> 0.5...1.5	> 1.5...3.0	> 3.0...6.0	> 6.0...12	>12
	%	%	%	%	%
≤2,000	40	44	47	55	60
> 2,000...4,000	44	47	52	58	64
> 4,000...6,000	47	50	55	64	70
> 6,000	50	55	60	70	75

Table 5. Minimum efficiency of sensible heat recovery system, according to I.T. 1.2.4.5.2

In the previous section Dedicated Outdoor Air Systems (DOAS) have been identified as the most appropriate systems to ensure adequate air quality in rooms with high concentrations of pollutants.

For this type of systems, energy recovery strategies should focus on the recovery of sensible heat, where the following relationship is satisfied:

$$T_{\text{salida}} = T_{\text{ext}} - \eta (T_{\text{ext}} - T_{\text{local}}) \quad (14)$$

with η (-), efficiency of sensible heat recovery, according to Table 4.

The temperature of the exhaust air, T_{exp} , can be reduced by means of an adiabatic cooling process, in which the air supplied to the coils is cooled down to the temperature T_{adiab} , that can be calculated with the expression:

$$T_{adiab} = T_{local} - \varepsilon (T_{local} - T_h) \quad (15)$$

with ε (-), efficiency the evaporative cooling. As it is a function of the pad geometry and the air flow rate, it is difficult to establish a reliable average value.

The capacity for heat recovery in the primary air handling unit significantly increases when using a displacement air diffusion system. Because of extracting air near the roof, where maximum temperatures are reached, the system has a great potential for heat recovery in winter, but also in summer, when adiabatic cooling processes can be used, the more effective the higher the extract air temperature.

3. An outlook on high efficient air conditioning systems for wide-open workshops

3.1. Climate separation via air curtains

Air curtains have the function of neutralizing outside air infiltration through the doors, reducing up to 90% heating and cooling thermal demand. In winter period, their use as hot water terminal unit makes also possible to meet the thermal load due to the infiltration rate not eliminated by the curtain. Reduction of latent loads demand is crucial in warm humid climates. Furthermore, they allow to control indoor environment regardless of external conditions.

To effectively carry out its function of climate separation, the curtains should maintain a proper discharge length whatever the external conditions of wind are. If the air curtain jet is too weak and the throw distance is short it does not prevent infiltrations. By contrast, excessive throw distance due to a strong jet can reduce efficiency by almost 50%. In this case, high velocity and turbulent flow make the air curtain partially mix with outside air.

The parameter that best characterizes the operation of a curtain is the momentum of jet, I_0 , that indicates the strength of the curtain. It is defined by the expression:

$$I_0 = q_0 \cdot d_0 \cdot U_0^2 \quad (16)$$

being d_0 the outlet width and U_0 the outlet velocity.

Modern air curtains vary the air flow driven maintaining a constant flow rate by adapting the geometry of the discharge outlet (Figure 4).

By choosing a relatively large outlet width, it can be achieved an optimum momentum, as needed to reach the floor, but with low velocity that keeps the flow in laminar regime.

Air curtain strength and heating can be controlled independently according to the needs (Figure 5).

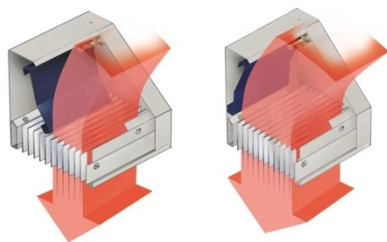


Figure 4. Variable outlet width positions in an air curtain

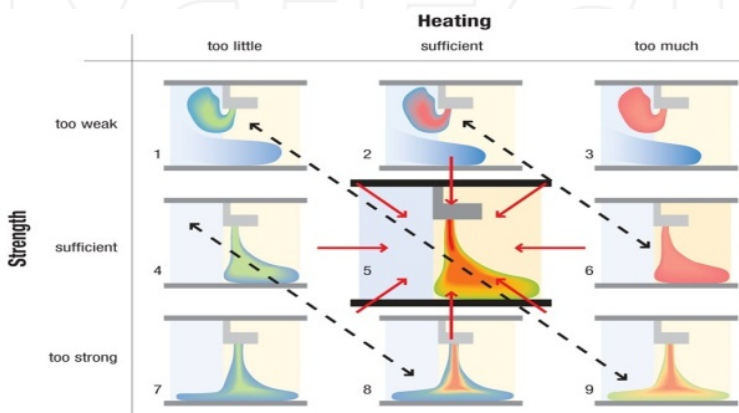


Figure 5. Air curtain strength and heating control possibilities

As an additional advantage, when used as heating terminal units they run on low temperature water, making them particularly suitable for being used with condensing boilers and solar thermal production. These options are discussed in the next two sections.

3.2. Heat production

Air curtains and air handling unit heating coils run on low temperature hot water. Under such circumstances, thermal production can be provided by a solar thermal system. For the auxiliary energy supply a condensing boiler modular gas is projected. As with any solar installation, the energy collected is transferred to storage tanks, also connected to the boiler that takes charge of heating water when solar coverage falls. The control of the group of curtains and the primary air handling unit is done by varying the water flow rate with a three-way valve (Figure 6).

a. Conventional energy contribution

Condensation gas boilers use the heat content of vapor from the combustion, which is transferred to the heating system. As the heat conversion efficiency of the boiler is referred to the fuel net calorific value, performance values are reached greater than unity. As condensation inside the boiler begins when flue gases drop to about 54°C, the boilers are

particularly suitable when the facility is operating at part load or when using terminal units operating at low temperature.

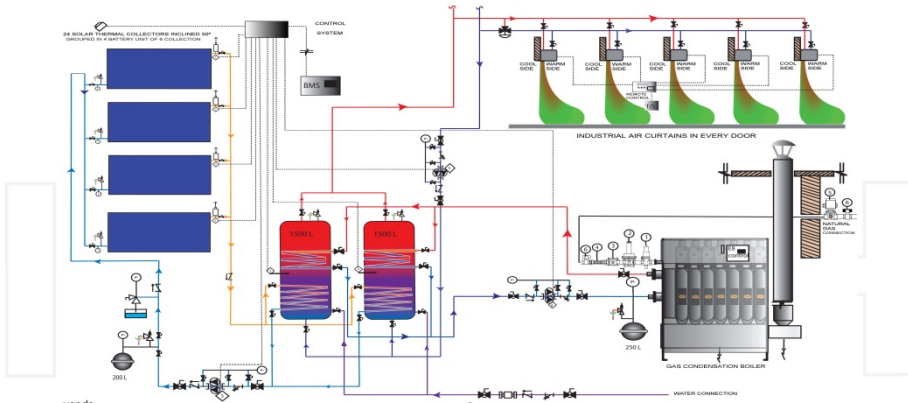


Figure 6. Proposed heating system for wide-open workshops (diagram)

The effect of this high efficiency boilers, with seasonal efficiency of 0.97, on the overall efficiency of the heating system as opposed to normal gas or electric boilers (Table 6) has been assessed in a recent work by the authors (Gil-Lopez et al., 2011).

b. Renewable energy contribution

The savings achieved with the use of solar energy in thermal plants is sufficiently well known. However, when considering the impact on the carbon emissions avoided by the provision of supplementary carbon free energy by solar heating, an interesting result is obtained: the largest amount of carbon emissions avoided occurs when the solar installation provides supplementary energy to less efficient energy sources (Table 7).

This problem, similar to that experienced when using economic and investment indicators that favour consumption and not savings has been fully studied by the authors (Gil-Lopez et al., 2011). The question that arises is how can be assessed the impact of the solar installation in a way that reflects the value of the savings being made and not the energy being consumed. This can be easily seen by the impact the solar installation has on the energy certification (Table 8a and b).

As it was expected, the electric emersion heater boiler has the lowest certification value of G, the diesel and condenser option a C level, and the modular gas condensation boiler a certification rating of B. When the same calculations are conducted but with the solar installation providing supplementary power during part of the year, the certification ratings for the air curtains powered by hot water both receive a value rating of A, whereas that for the electric emersion heater boiler, although it has an improved indicator value, retains its G rating certification. Therefore, to obtain an A rating energy certification the air curtains need to be powered by hot water supplied through a combination of either diesel with condenser unit or a modular gas condensation boiler, with a solar installation

providing supplementary source of energy heating the water demanded by the air curtains.

	Boiler type			Reference	
	Diesel boiler and condenser	Electric boiler	Modular gas condensation	Normal gas boiler	Direct electricity
Name	TRISTAR	General	MODULEX	General	General
Boiler option number	1	2	3		
Fuel source	Diesel -C	Electrical emersion heater	Natural gas	Natural gas	Mains electricity supply
Annual energy demand (kWh)	35,459				
Conversion factor for the carbon emitted from energy consumed (kgCO ₂ /kWh)	0.280	0.649	0.204	0.204	0.649
Annual energy consumption by the boiler (kWh)	39,005	42,551	39,005	223,394	56,735
Annual energy consumption from the fuel (kWh)	42,165	110,761	39,434	62,737	147,681
Total energy consumed by the system	4 m ³	113,021 kWh	3,902 m ³	7,526 m ³	150,695 kWh
Total carbon emitted (kg)	11,806	71,884	8,045	12,798	95,845

Table 6. Carbon emission levels for an ITV workshop in Madrid when energy demand is met only by the boiler

Impact on Energy and Carbon emissions for energy supplied by the solar installation, rather than the boiler			
Boiler option number	1	2	3
Annual energy savings provided by the solar installation (kWh)	21,054		
Annual energy demand not provided by boiler (kWh)	23,160	25,265	23,160
Annual amount of fuel not consumed (kWh)	25,036	65,766	23,415
Energy that was not consumed	2.5 m ³	67,108 kWh	2,317 m ³
Amount of carbon emissions avoided (kg)	7,010	42,682	4,777

Table 7. Impact of solar collectors for an ITV workshop in Madrid

Certification value without the solar installation		Main energy supply options		
		Diesel boiler and condenser	Electric boiler	Modular gas condensation
Name		TRISTAR	General	MODULEX
Boiler option number		1	2	3
Type of fuel		Diesel-C	Electricity	Natural gas
Quantity of carbon emitted without the solar installation (kg)		11,806	71,884	8,045
CO ₂ emissions per m ² of surface area of the building (kgCO ₂ /m ²)	1,620 m ²	7.29	44.37	4.97
Emission reference boiler: for natural gas	7.90 kgCO ₂ /m ²			
Emission reference boiler: for electrical boiler	59.16 kgCO ₂ /m ²			
Indicator value (kgCO ₂ /m ²) Natural Gas		0.923	5.600	0.629
Energy certification value		C	G	B

(a)

Certification value with the solar installation		Main energy supply options		
		Diesel boiler and condenser	Electric boiler	Modular gas condensation
Name		TRISTAR	General	MODULEX
Boiler option number		1	2	3
Type of fuel		Diesel-C	Electricity	Natural gas
Quantity of carbon emitted with the solar installation (kg)		4,796	29,202	3,268
CO ₂ emissions per m ² of surface area of the building (kgCO ₂ /m ²)	1,620 m ²	2.96	18.03	2.02
Indicator value (kgCO ₂ /m ²)		0.37	2.28	0.26
Energy certification value		A	G	A

(b)

Table 8. a. Rating without solar energy for an ITV workshop in Madrid, b. Rating with solar energy for an ITV workshop in Madrid

3.3. Displacement ventilation system

It helps significantly reduce the level of pollution in the occupied zone and the required air flow compared with ventilation with mixing with turbulent flow. It also allows the integration of direct and indirect adiabatic cooling systems. Furthermore, it optimizes the air

conditioning system efficiency, as air can be driven to higher temperatures achieving the same degree of thermal comfort for the workers through a well calculated design for terminal air velocities and temperature.

According to Skistad (1994), the design procedure is the following:

- a. The following variables are known:
 - Specific cooling load, Φ_r , (W/m²).
 - Supply air temperature, T_s . Due to the fact that air is thrown directly to the occupied zone, air is only cooled to a temperature 3 °C lower than indoor air conditions.
 - Difference of temperatures between extract and supply air, ΔT . It is related to the supply air flow rate, Q_s by means of the thermal load. Whether this parameter is restricted to 3°C or is increased to meet greater thermal loads, the resulting flow can lead to excessive diffuser sizes, problems of noise and inappropriate air velocities.
- b. Resolution of air temperature value at 0.1 m height.

$$T_{0.1} = T_s + \theta_{0.1} (T_R - T_s) \quad (17)$$

where $\theta_{0.1}$, dimensionless temperature, is obtained with Mundt law:

$$\theta_{0.1} = \frac{1}{Q_s c \rho \left(\frac{1}{\alpha_{c,f}} + \frac{1}{\alpha_{r,c}} \right) + 1} \quad (18)$$

in which $\alpha_{c,f}$ is the convective heat exchange between air and floor, that usually adopts a value of 4.5 W/m². With respect to radiant heat exchange between air and ceiling, $\alpha_{r,c}$, it is obtained, from the initially unknown ceiling temperature T_c , by means of an iterative process. Although usually is considered equal to $\alpha_{c,f}$, its value is far from being constant.

- c. Calculation of floor temperature, T_f

$$T_f = T_{0.1} - \alpha_{c,f} / h_i \quad (19)$$

with h_i , surface heat transfer coefficient between floor and air. For a horizontal heat flow with air at low velocity, it can be considered $1/h_i = 0.13 \text{ m}^2\text{K/W}$

- d. Assuming that the temperature gradient in the room is constant, air temperature at 1.8 m height, $T_{1.8}$, is obtained:

$$T_{1.8} = T_{0.1} + 1.7 \cdot \frac{T_R - T_{0.1}}{H_R - 0.1} \quad (20)$$

as well as the corresponding temperature gradient:

$$\text{grad } T = \frac{T_{1.8} - T_{0.1}}{1.7} \quad (21)$$

With the obtained values, accomplishment of normative comfort conditions is tested, paying special attention to the temperature difference between head and feet (ASHRAE, 1992).

3.4. Primary air conditioning unit

Ventilation is provided by a primary air handling unit with indirect adiabatic cooling heat exchanger section (Figure 7). The unit has two water coils. The heat coil is supplied by the heat production system described in section 3.2. The cooling load is met by water from a compression chiller. An alternative system with a solar absorption chiller for cold production is also suggested, but not discussed in this chapter.

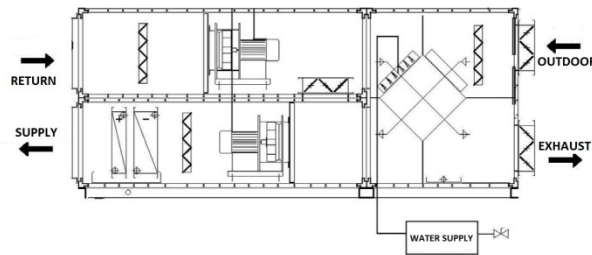


Figure 7. EQUAM adiabatic air handling unit

Air transformations in the air handling unit components are shown in the pscymetric chart (Figure 8), where O stands for outdoor air, S for air supplied to the coil, R for room conditions and EX for exhaust air.

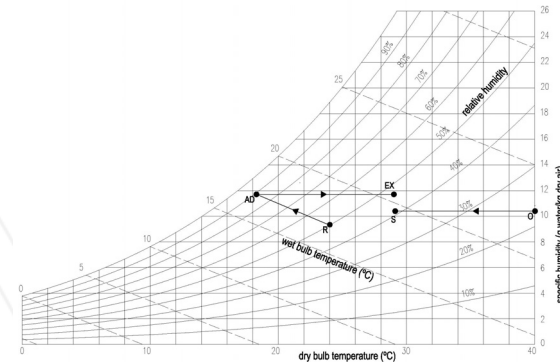


Figure 8. Pscymetric processes of indirect adiabatic cooling

4. Analysis of the proposed options for an ITV workshop

This section compares the energy performance of various options for air conditioning a wide-open workshop. Starting from an initial situation (option 0) of a non conditioned building, its energy consumption once refurbished is studied and compared for each of the

three following options: an air curtain system functioning as a climate separator (option 1); the previous system with a conventional air handling unit (option 2); and finally, a comprehensive air conditioning system of high efficiency that includes all aspects covered by section 2 and 3 of this chapter.

The design of this system and the selection of components for the case study are due to Eng. Paul Gerard O'Donohoe, who has used technology developed by TAYRA S.L.

To analyze the performance of the technical systems for each of the proposed situations, a representative industrial building has been selected. Such is the case of a wide-open workshop, where technical assessments of roadworthiness are carried out on cars and trucks (ITV workshop). This type of buildings is of rectangular shape, with a surface area of 1620 m² and a height of 7.5 m. It has no windows, but five entrances and exits, three of them at 5 m x 3 m and two at 5 m x 4.5 m. The exit doors are controlled to limit the flow of vehicles leaving the premises. These buildings, that outnumber 1,000 all around Spain, are usually located in remote areas just off main motorways. For the case study, an ITV workshop located in Carmona (Sevilla) was chosen. See attached plan in Figure 9.

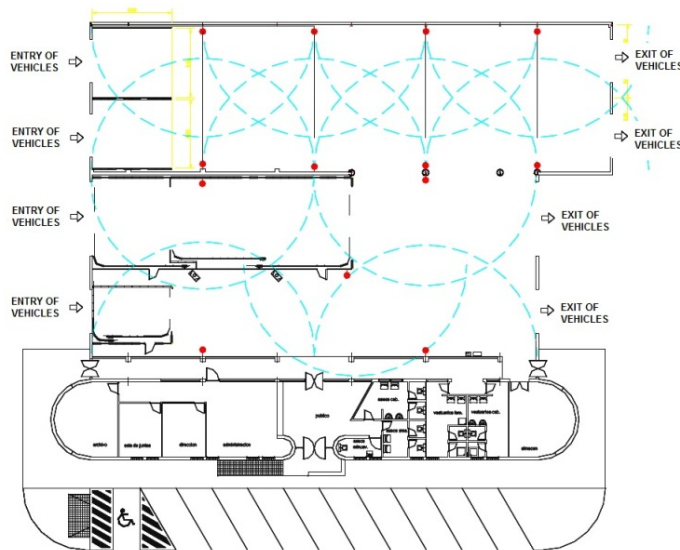


Figure 9. ITV workshop in Carmona (Seville). Plan

Using a computer simulation of the building with the e-Quest program (Figure 10), a prediction of its performance for the aforementioned options has been obtained. E-Quest uses DOE2 engine to perform an hourly simulation of the building for a whole one-year time period. For each hour, heating and cooling loads are calculated. The performance of pumps, fans, boilers, chillers and every energy consuming equipment within the building is also simulated. Finally, the energy use of every end use, including lightning, is tabulated. Authors such as Crawley et al. (2008) provide a sound comparative study of the potential offered by the most common simulation tools, DOE2 included.

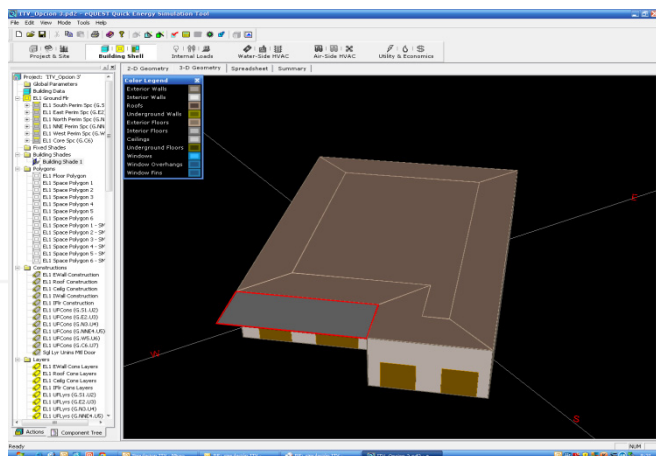


Figure 10. ITV workshop in Carmona (Seville). E-Quest simulation

In the simulations, synthetic meteorological data have been used. 8.762 records, including hourly data, have been generated by the program CLIMED 1.3 (IDAE, 2009) from the normal data of AEMET for the meteorological station of Seville.

The aforementioned cases are described next:

4.1. Case 0: Original building

ITV workshops, for the reasons stated in the introduction, do not have any higrothermal conditioning system. Not even a mechanical ventilation system, as room air is changed by means of natural ventilation through permanently open doors. The energetic simulation in this conditions, only seeks to analyze the evolution of indoor temperatures throughout the year, in order to identify the times when it exceeds the limits allowed by labor legislation (16°C in winter and 28°C in summer).

The simulation of the building in its original condition reveals that the 56% of annual hours, indoor temperature remains outside the limits marked by labour legislation (Table 9).

TEMPERATURE (°C)	AM												HOURS												PM												TOTAL
	1	2	3	4	5	6	7	8	9	10	11	12	1	2	3	4	5	6	7	8	9	10	11	12													
ABOVE 29.4	0	0	0	0	0	0	0	0	0	0	0	0	0	1	14	17	23	34	38	31	19	12	0	0	0	193											
26.7 - 29.4	0	0	0	0	0	0	0	0	0	0	0	0	0	2	32	44	44	50	37	28	13	29	0	0	0	284											
23.0 - 26.7	0	0	0	0	0	0	2	9	12	18	26	38	53	50	30	25	29	28	24	28	36	0	0	0	0	405											
21.1 - 23.0	0	0	0	0	0	0	24	32	38	44	60	53	37	31	37	35	38	35	38	28	29	0	0	0	0	557											
18.3 - 21.1	0	0	0	0	0	0	43	66	59	53	40	36	42	43	40	41	40	41	42	50	40	6	0	0	0	682											
15.6 - 18.3	0	0	0	0	0	0	42	40	40	47	41	45	45	46	58	54	41	38	40	38	42	50	30	0	0	851											
BELOW 15.6	0	0	0	0	0	0	38	113	121	124	108	94	87	49	23	10	0	0	0	0	0	28	40	0	0	852											

Table 9. Annual distribution of hourly temperatures

4.2. Case 1: Air curtains as climate separators and mechanical extraction ventilation system

The climate separation is achieved by means of the following equipment:

- Two air curtains Indac S150 (Biddle) placed in series for each of the six existing doors for small vehicles.
- Two air curtains Indac S200 (Biddle) placed in series for the two doors for large vehicles.

Air curtains characteristics have been selected (Figures 11a and 11b) by using a commercial simulation program (Biddle Innovative Klimatechnik, 2012).

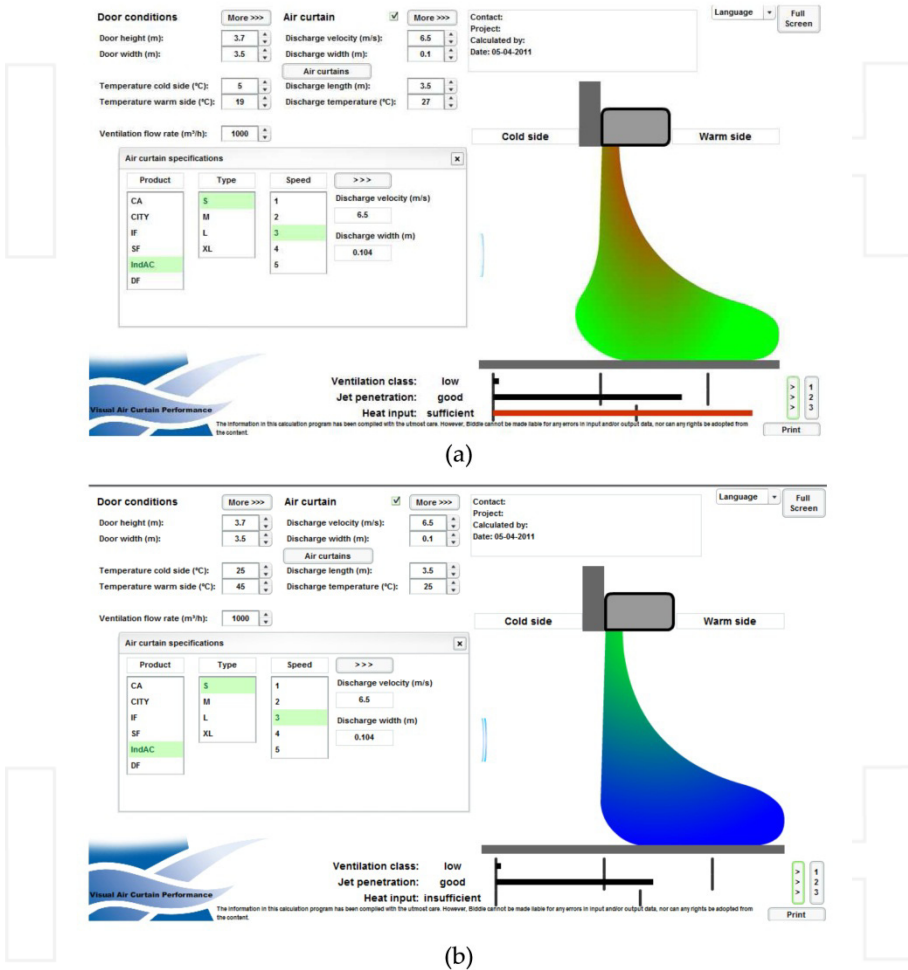


Figure 11. a. Winter conditions air curtain selection, b. Summer conditions air curtain selection

They recirculate air at the room temperature throughout the year. The supplied air flow rate is fixed (5,000 m³/h), and the equipment adapts its velocity of discharge speed by varying the geometry of the outlet. Thus, an identical download length is achieved whatever the external conditions of wind are. The estimated electric power is 0.44 kW per curtain, giving a total power of 7.04 kW.

- A mechanical extraction system with an air flow rate of 70,000 m³/h, which is equivalent to 7 ACH, according to the most restrictive rate of ventilation indicated by current Spanish legislation (R.D. 314/2006, 2006b, 2006c; Ayuntamiento de Madrid, 1985).

The simulation of the case 1 yields the monthly evolution of electric consumptions that are shown in Figure 12:

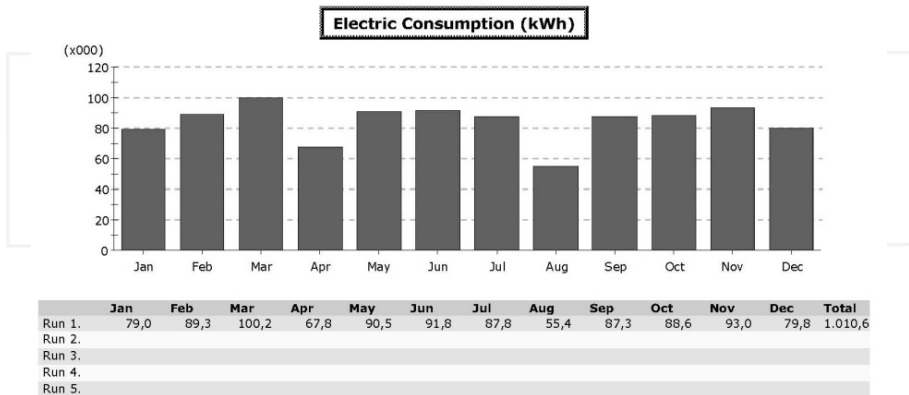


Figure 12. Case 1: Monthly electric consumption

4.3. Case 2: Air curtains and a conventional HVAC system

The climate separators described in the previous option also function as terminal heating units that supply air at 27°C in winter conditions. They are completed with a conventional air conditioning system comprising:

- A primary air handling unit that supplies a volume air flow rate of 7 ACH which. Considering that the whole volume of air in the building is renewed, the air flow rate comes to 70,000 m³/h. Winter indoor temperature is 18°C, while, during the summer, temperature is kept to 26°C. They are obtained with supply air from the unit at 30°C and 14°C, respectively.
- A gas boiler with a power output of 700 kW (estimated seasonal efficiency: 86%), meet the heating loads demanded by the air curtains (47,7 kW each) and the primary air handling unit heating coil (183 kW).
- A VRV air condensed water chiller with a power output of 218.1 kW, with an estimated seasonal EER of 3.5.

The energy simulation of the building and its technical system leads to results of monthly electric and fuel consumption that will be compared in the next section with the results for the more efficient option 3.

The simulation of the case 2 yields the monthly evolution of electrical and fuel consumption that is shown in Figures 13a and b:

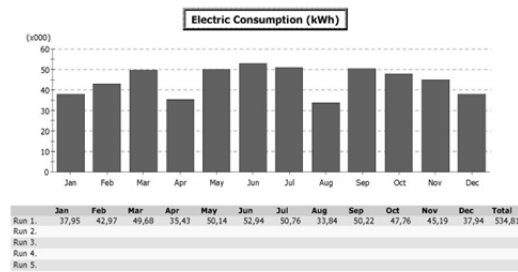
4.4. Case 3: Air curtains and a high efficiency HVAC system

The air curtains also function as terminal heating units that supply air at 27°C in winter conditions. Air conditioning is achieved in this case by a high efficiency system that includes all the strategies that have been described in section 2 and 3 of the chapter:

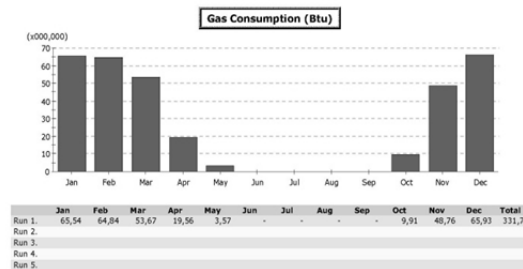
- A modular condensation gas boiler Tayra/Unical Eco-pacQ-Mx-550-CS2-EM with a power output of 228 kW (heat conversion efficiency: 108 %) meets the thermal loads demanded by air curtains (25 kW each) and the air handling unit coil (28 kW).
- A solar thermal installation is design to provide 70% of the energy requirements for heating. It has been chosen a 24 m² surface of collectors ASTERSA AT020 (optical efficiency: 0.748 and heat loss factors: 3.718 and 0.014 W/m²K) oriented South and inclined at 35°, that transfer the collected thermal energy to a 3,000 liters storage tank.
- A VRV air condensed water chiller with a power output of 57.1 kW, and estimated seasonal EER is 3.5.
- A EQUAM primary air handling unit (100% outside air) with the following characteristics: Supply and return fans air flow rate is 30,000 m³/h (7 ACH for a 3 m height, that corresponds to the occupied zone); A 28 kW heating coil, running on hot water (50/35°C), supplies air at 19°C, maintaining the outdoor specific humidity conditions; A 46.7 kW cooling coil, running on chilled water (7/12°C) supplies air at 24°C and 78% relative humidity; A cross flow heat recovery unit (70% average efficiency), with a heat recovery capacity of 151 kW for nominal flow in winter conditions (0.6 °C y 90% relative humidity), provides a supply air at 16.2 °C from the extract conditions (19°C and 26% RH); in summer conditions heat recovery capacity is 161.6 kW, for it includes an adiabatic cooling by water sprays of extract air at 24°C and 78% relative humidity. By means of this indirect adiabatic cooling, outside air at 40°C y 24% de HR, is brought to 28.7°C and 59% relative humidity before entering the water coil.
- A displacement air diffusion system, with 13 diffusers VA-ZDA DN 355, of Kranz Componenten, suspended at 3 m height in the position indicated in Figure 9. The throw distance is 9 m, when nominal flow is supplied, with a sound power level of 63 dBA. They incorporate a device for regulating the discharge, which can change its direction upwards or downwards depending on the regime conditions (cooling or heating). The return is by ceiling, but some grids at floor level help to remove heavier contaminants.

With this system, air is supplied at 26°C in winter to keep indoor conditions close to 28°C. This indoor project temperature, combined with residual air velocities higher than usual but acceptable due to the physical activity of the workers, allows to keep comfort conditions in the occupied zone. Furthermore, high temperatures that are obtained in the ceiling may be used with high efficiency in the heat recovery process.

The simulation of the case 3 yields the monthly evolution of electrical and fuel consumptions that are shown in Figures 14a and b:

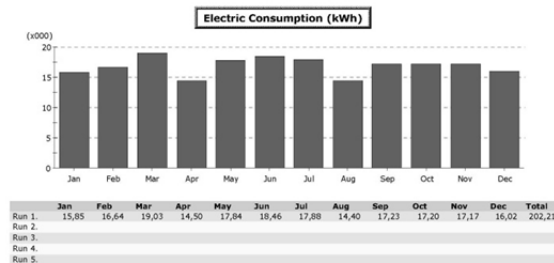


(a)

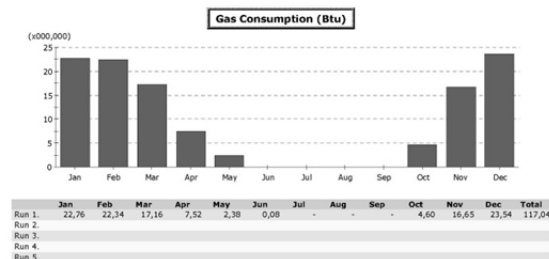


(b)

Figure 13. a. Case 2: Monthly electric consumption, b. Case 2: Monthly gas consumption



(a)



(b)

Figure 14. a. Case 3: Monthly electric consumption, b. Case 3: Monthly gas consumption

5. Conclusion

In the absence of air conditioning system, the predicted indoor temperature evolution throughout the year suggests that there is significant potential for intervention to improve the performance of the activities in ITV workshops.

For what respects to the more efficient system to air conditioning a wide open workshop, the analysis of the energy consumption during a year simulation allows to conclude that high efficient system like the proposed, composed by a solar heating installation with a condensing gas boiler auxiliary power option, air curtains for environmental thermal control and displacement ventilation supplied by a dedicated outdoor air handling unit with evaporative indirect cooling, reduces electric consumption to 37% and gas consumption to 35%, compared to a conventional HVAC system.

From the point of view of its efficiency, it is highlighted the importance of the general conception of the system, instead of placing design efforts in improving the performance of its components.

The comparison between systems should be done in terms of primary energy and carbon emissions, rather than based on economic criteria. European legislation states that a system of energy performance certification is a mandatory requirement for rented, sold or constructed buildings. In response to this, Spain's recent building regulations have established the process of certification of new buildings. A series of tools are provided to calculate whether the new buildings are highly efficient (class A) or least energetically efficient (class G). The official program used for this process of certification is called CALENER and provides an average carbon value of the overall building energy performance, and classifies it by means of a comparison of the performance of the building systems and construction materials with their corresponding reference values.

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