Example of audit of an air conditioning system

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Abstract

The example presented here concerns the audit of a typical, medium-size, office building erected in Brussels at the end of the sixties.
This building is equipped with a classical "old fashion" air conditioning system with induction units.
An audit procedure is developed and tested "in way". It consists in a systematic analysis of all information available, with help of very simple calculation.
Not all the orders of magnitudes of the fuel and mostly of the electricity consumption can be interpreted, but very significant energy saving opportunities are already discovered.

Introduction

The work presented here is performed in the frame of the European "AUDITAC" project.
It aims to develop an audit methodology, with appropriate simulation tools and benchmarks.
Several case studies are used to support, to test and to illustrate these developments.
Each case study is described according to standard format, in order to make the data easy to re-use in the future.
The audit of the HVAC system consists in analyzing the information available about actual energy performances and in identifying the most attractive retrofit opportunities.
Focus is given here to cooling energy consumption, but it could never be completely dissociated from heating requirements.
The example of case study presented here concerns a typical air-conditioned office building built in Brussels at the end of the sixties.

Building description

Design concept

The building considered is of medium size: around 25 000 m3 of air-conditioned floor area.
It's located at an altitude of 80 m and surrounded by similar buildings.
The Brussels climate can be characterized by the following data:

Heating sizing temperature: - 10°C

Cooling sizing temperature and relative humidity: 30 °C and 50 %

15/15 heating degree-days: 2000 K*d.

The building contents a set of 50 open-plan offices (such as shown in Figure 1), a few meeting rooms and two restaurants.

It counts a total of 13 levels: -5 to -1 for parking, 0 for reception, restaurant and meeting rooms and +1 to +7 for offices.

It has a "H" horizontal shape, subdivided in 3 blocs, with an internal yards closed by other buildings of same height.

Figure 1: one of the open-plan offices

The building envelope is made of about 1000 "curtain wall" modules (144 per floor), with all glazing, almost from floor to ceiling and almost no insulation (Figure 2).

The external sizes of each module are 1.9 m*3.1 m. It contents a window of 1.6m * 2.5 m

(Figure 3)

Figure 2: The building envelope

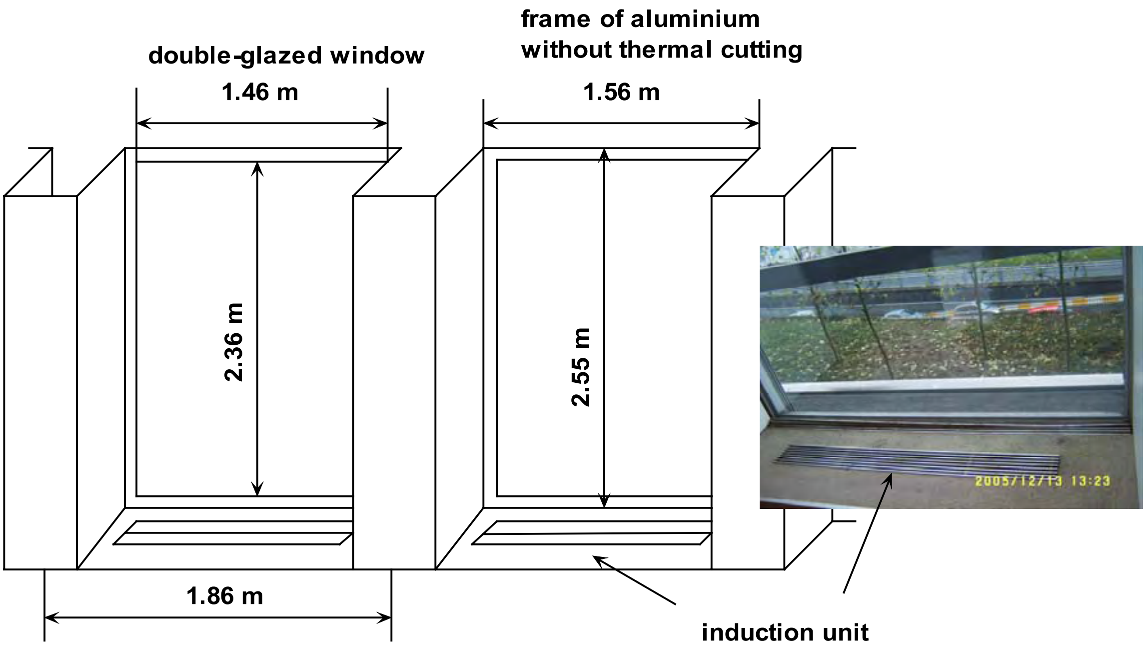

Figure 3: Building window details

All windows are equiped with very efficient external curtains, automatically controlled according to sunshine.

Occupancy and comfort requirements

At working time, there should be 1 100 to 1 200 occupants in the building, with a rather constant occupancy rate.
The occupation period is from 7h to 19 h, 5 days per week, all the year.
The walls of the reception zone (level 0) are dressed by precious wood. Strict conditions of temperature and humidity are imposed there 24h/day and 365 days/year.

Ventilation strategy

The system provides full fresh air with slight over-pressure in almost all occupancy zones.
The air extracted from the offices (90 000 m3/h) is supplied to the parking.
The ventilation of the offices is “imposed” by the use of the induction units (see hereafter).
A constant ventilation flow rate (24h/24h) is maintained in the restaurants and in the meeting rooms.
By adding the contributions of all AHU’s in use inside the building, the total fresh airflow rate can be estimated to 192330 m³/h.
This regime is supposed to be maintained 78 h per week (see hereafter).

Heat transfer coefficients and nominal heat losses

The area of each frontage module (except for the ground level) is 1.86 x 3.10 = 5.80 m², with a glazing area of 3.45 m².
The average heat transfer coefficient of the module is estimated to 2.7 ± 10% W/m²K, or
5.8 x 2.7 = 15.7 W/K.
The modules of the ground levels are higher (4.5m in place of 3.1 m) and equiped with single glazing only (their heat transfer coefficient is two times bigger per unit of area).
With that information, the global heat transfer coefficient of the whole frontage can be estimated as follows:
(15.7 W/K x 144 modules x 7 levels) + ((4.5 m / 3.1 m) x 15.7 W/K x 2 x 144 modules) = 22389.2 W/K
i.e. about 22.4 kW/K.
To the mechanical ventilation 192330 m³/h, used approximately 78 hours per week, corresponds another heat transfer coefficient:

$$\frac{192330\ m^3/h}{3600}\ x\ 1.2\ kg/m^3\ x\ 1.005\ kJ/kg\,K = 64.4\ kW/K$$

In nominal heating conditions, with Δt = 30 K, this gives a sensible heat demand of:
(22.4 kW/K + 64.4 kW/K)*30 K = 2604 kW.

If having to maintain 50% of relative humidity at 20°C in the same nomnal conditions, the corresponding latent heat demand would be of about 14.4 kJ/kg, i.e. 921 kW in the present case; this would bring the total heat demand around 3.5 MW.
This very rough estimate of the total heat demand can be compared to the power installed in the heating plant (around 4 MW, as seen hereafter): the agreeement is not bad and there seems to remain a fair reserve of heating power.

In order to calculate an average heating demand, it would be necessary to take the ventilation intermittency into account, i.e. to use an average heat transfer coefficient:
22.4 + 64.4 x (78/168) = 52.3 kW/K…

Nominal heat gains

The same values of the (sensible) heat transfer coefficients are, of course valid for heat gains calculations.

In nominal conditions, with $\Delta t = -5$ K (25 °C inside and 30 °C outside), it gives a sensible cooling demand of 0.33 MW.
Other sensible heat gains are: sunshine, occupant (sensible) metabolism and electricity consumed inside the building.
Solar heat gains are here very reduced, thanks to efficient (external) solar protection: they are estimated to no more than 50 W/m2 of window area, i.e., for the whole frontage (4 400 m2), probably around 0.22 MW.
The sensible metabolism of 1 200 occupants is a modest contribution: no more than 0.09 MW.
It's the electricity consumed inside the building, which dominates this sensible heat balance, and this term is, unfortunately not well identified (see hereafter): it might reach 0.8 MW in the present case!
This brings the total of sensible heat gains to 1.4 MW.
To that power, has to be added the latent cooling demand associated to air drying.
This term can be estimated to 8.7 kJ/kg, or 0.56 MW, which brings the total cooling demand around 2 MW.
Again, there is a satisfactory agreement between this rough estimate and the cooling power actually available (see hereafter)…

HVAC system

Terminal units

There are about 1 000 four – pipe induction units (one per module), installed in the floors of all offices (Figure 4).
The heating and cooling coils are installed in "V" position in each unit.
The local temperature control is achieved by 500 double thermostatic valves (one for two induction units).
Nothing is done to prevent the air from passing accross the coil which is not used (this makes the induction unit requiring a high primary air flowrate to work properly).
Condensation may occur time to time on cooling coils.
The air distribution provided by this system is not fully stisfactory: to high air speed induced near the floor and too short jet bearing in cooling regime.

Figure 4: The induction units and their thermostatic valves

Air handling units

One big CAV AHU unit is used to supply a total of about 90 000 m3/h of primary air to all induction units.
Other zones are supplied by a set of about 20 CAV and VAV AHU's.
The big "primary" AHU counts with the following components:
Dampers, filters, preheating, adiabatic humidifier, cooling, post heating , and two fans in parallel.
Both fans are equiped with frequency drivers (in order to protect the motors of the fans and to reduce the instantaneous electrical peak of the system).
Almost all other AHU's are also working in full fresh air.

Heating and cooling plants

These plants are "classical": fuel oil boilers, vapor compression chillers and cooling towers.

There are four boilers: three for space heating (1.7+1.7+0.55 MW) and one for tap water (0.1 MW).
This gives a total of about 4 MW of space heating capacity.
There are two main chillers with water-cooled condenser, one twin screw of 0.85 MW and one centrifugal of 1.2 MW. There are also two auxiliary chiller of very small power.
This gives a cooling capacity of about 2.1 MW.
The two main chillers are mounted in series on the chilled water circuit. Each one has its own cooling tower with two speed axial fan.
Each chiller has also its own condenser pump, but no evaporator pump (the chiller water is circulated by the distribution pump).

Control strategy

The building is equipped with a classical BEMS with two levels: a set of local control units and a PC for supervisory management.
This system is relatively "open": control strategies can be modified without the help of a specialist.
But the data storage capacity is limited: one day to one week, according to the amount of measuring points registred.
These records are only available as printed tables or diagrams.
The data files cannot be transferred to anther computer.

Focus is given hereafter to the control of the office zones only (the dominant term):
For all the office zones, the dry bulb temperatures are controlled in feed back, thanks to the double thermostatic valves: these valves are modulating the water flow rates supplied to both (heating and cooling) coils of the induction units.
The air humidity control is achieved at the level of the AHU, thanks to a (on/off) control on the pump supplying the adiabatic humidifier and to a modulating valve supplying the cooling coil.
There is also a feed back control on the mixing valve supplying the pre-heating coil; the AHU exhaust air temperature set point is displaced in relationship with the outside air temperature, in such a way to make the adiabatic humidification possible, when required, and also to bring some sensible heating or cooling to the zone, when helpful.
This is achieved by a simple linear law: the set point is passing from 14 to 18°C, when the outside dry bulb temperature is passing from 25 to 18°C. Outside this range, the set point stay constant.
But derogations are always possible, for example to speed-up the return to comfort conditions inside the building after a non-occupation period…

The primary air is only supplied during pre-heating and occupancy time.
Out that time, if the weather is very cold, the induction units are still used in free convection mode, by supplying hot water to the heating coils.

The re-starting time of the installation in the morning is fixed by the BEMS, according to weather conditions and to the week day.
This is the result of a (questionable) optimisation process.
In winter conditions, the typical re-starting time are:
Monday – between 2 am and 7 am (sometime even Sunday evening if the weather is very cold)
Tuesday - between 3 am and 7 am
Wednesday – between 4 am and 7 am
In average, the total running time is of about 78 h/week.

The chilled water temperature regime is 6-12 °C in nominal conditions. The set point of 6°C can be displaced as function of outside conditions.

The nominal water temperatures at condensers supplies are between 32 and 34 °C.
The slide valve of the screw chiller needs enough pressure to work.
Also the expansion devices of both chillers need a certain pressure difference to supply enough refrigerant to the evaporator.
This means that the condenser temperature has to stay above a certain limit.
This is taken into account in the following control strategy applied to the cooling towers, according to return water temperature:
Up to 24 °C: water spray
Up to 27 °C: an in low speed
Up to 29°C: fan in high speed.

NB: the towers are empty from end of december to end of march (the remaining cooling demand is then satisified thanks to the small, air-cooled, auxiliary chillers).

Data analysis

Electricity and fuel consumptions

As usually, monthly records of electricity and fuel oil energy consumptions are here available.
The records made from September 2004 to February 2006 are plotted in Figure 5.
It appears that the average electricity consumption of this period is floatting around 518908 kWh/month ± 10 %. The seasonal variation seems unsignificant.
The fuel oil energy consumption shows much larger variations, around an average of 42484 kWh/month for the period analysed.

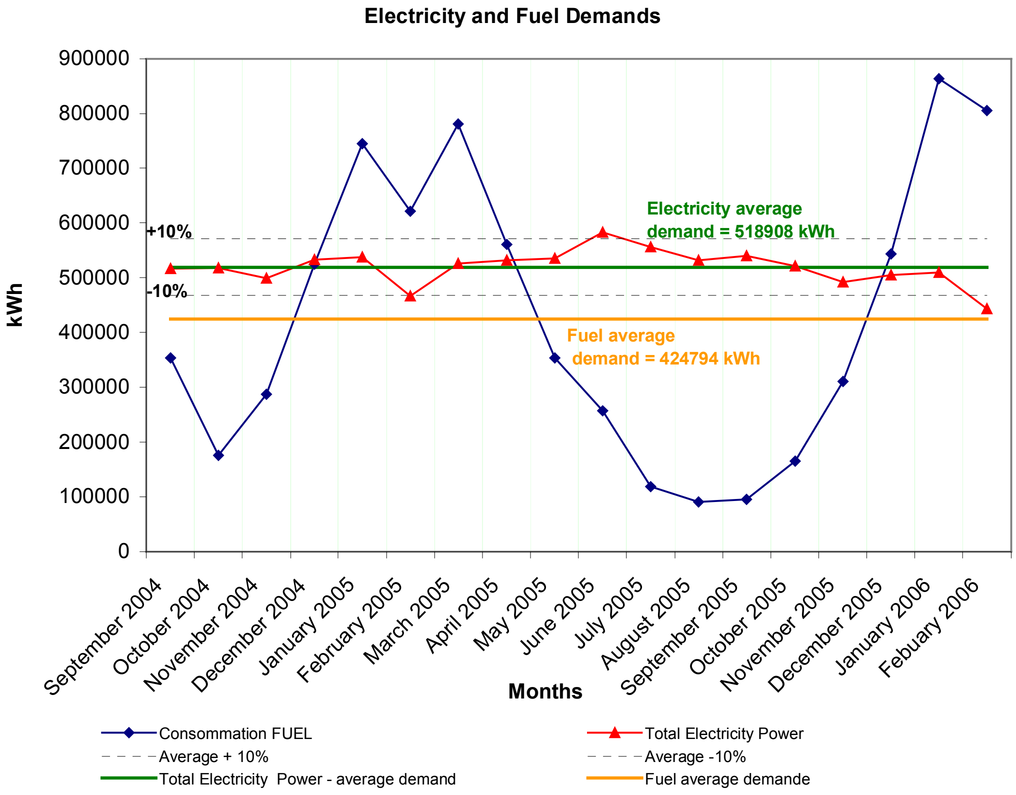

Figure 5: Electricity and fuel consumtions

Figure 6 shows the distribution of fuel oil power defined in montly average (from August 2003 to March 200), as function of the external dry bulb temperature.
It gives a fairly good correlation factor (0.63) and allows a meaningful linear regression to be identified. Its equation is given in Figure 6.
It shows that the heating demand tends to zero when the outside temperature is around 22°C.
The slope of this law is of the same order of magnitude as the average heat transfer coefficient already defined for this building: about 0.5 MW/K.
Theoretically, it should have been a little higher, because of the latent heat power consumed to humidify the air. The remaining error found in this building "signature" is very probably due to the effect of the inside temperature control: the temperature inside the building is "sliding" down slowly with the outside temperature.
By extrapolation of this law until –10°C, we would find a monthly average heating power of 1.6 MW, i.e. a peak of about 1.6*168/78=3.5 MW, which is in fairly good agreement with previous estimates.

The interpretation of the electrical consumption is much less obvious:
A very first step is to distinguish the "peak" hours and "off-peak" hours consuptions.
In the present case, 2747030 kWh where consumed on 4446 "off-peak" hours and 3603957 kWh on 4290 "peak" hours. The corresponding average powers are 612 and 840 kW, respectively.
The difference between these two power levels can be explained by a much higher rate of use of the HVAC system during "peak"hours.
Indeed, the "peak" supplement of (840 kW - 612 kW) = 228 kW is of the same order of magnitude as the total fan power (calculated hereafter), which should be the dominating term in the HVAC electrical consumption.
But both orders of magnitude are very high: they correspond to 24 and 34 W/m2 of (useful) floor area!
The very high electricity consumptions are probably (but not completely) due to the computation equipements.

As seasonal variations are unsignificant, there is little hope here to distinguish the impact of the chillers. More detailed records would be required to go further in this analysis: hourly records and/off separate records for HVAC and non-HVAC consumptions…

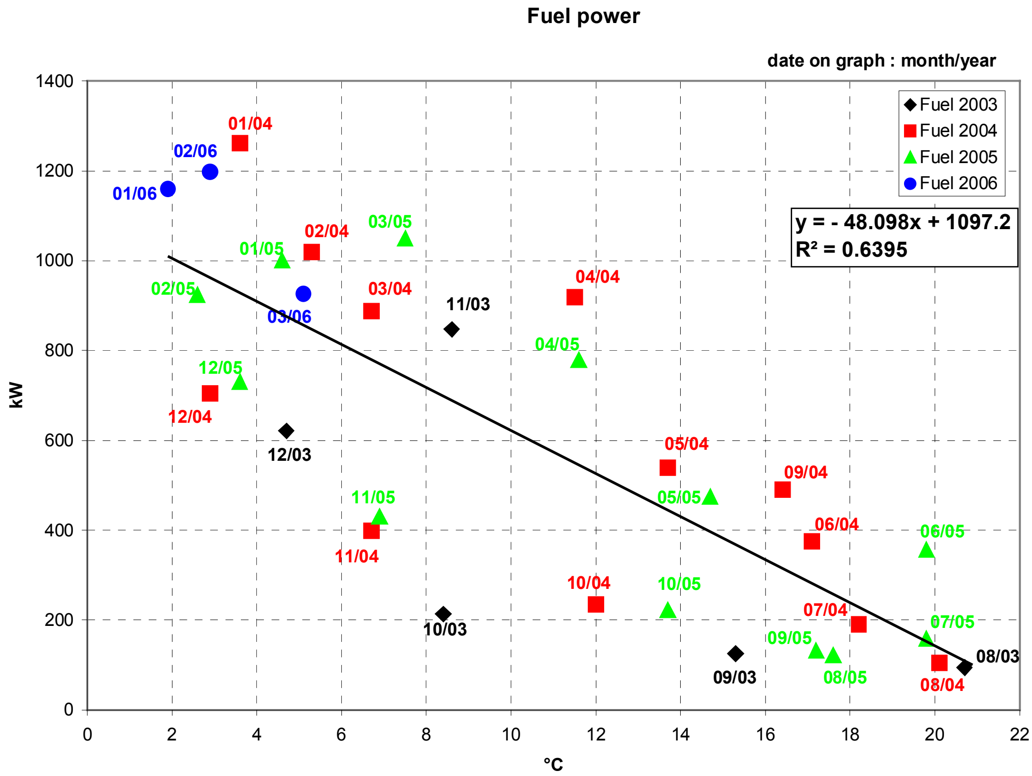

Figure 6: Monthly averages of fuel oil power as function of the outside air temperature

Energy costs

Energy costs are plotted in Figure 7.
These costs are calculated with the reference data given in table 1.
Peak and off-peak electricity costs are, of course, distinguished. The off-peak period is supposed to go from 10:30 pm to 6 am.
The predominance of the peak electricity on both other costs is spectacular.
When trying to "improve" the energy effectiveness of the system (for example by optimizing the control strategy), one should take care of not shifting any part of the electrical demand from "off-peak" to "peak" hours…

		Price
kWh electric	full hours	0.11 €/kWh
	off peak hours	0.065 €/kWh
Liter of	fuel	0.3 €/l
	natural gaz	0.3 €/l

Table 1 : Electricity and fuel costs take as reference

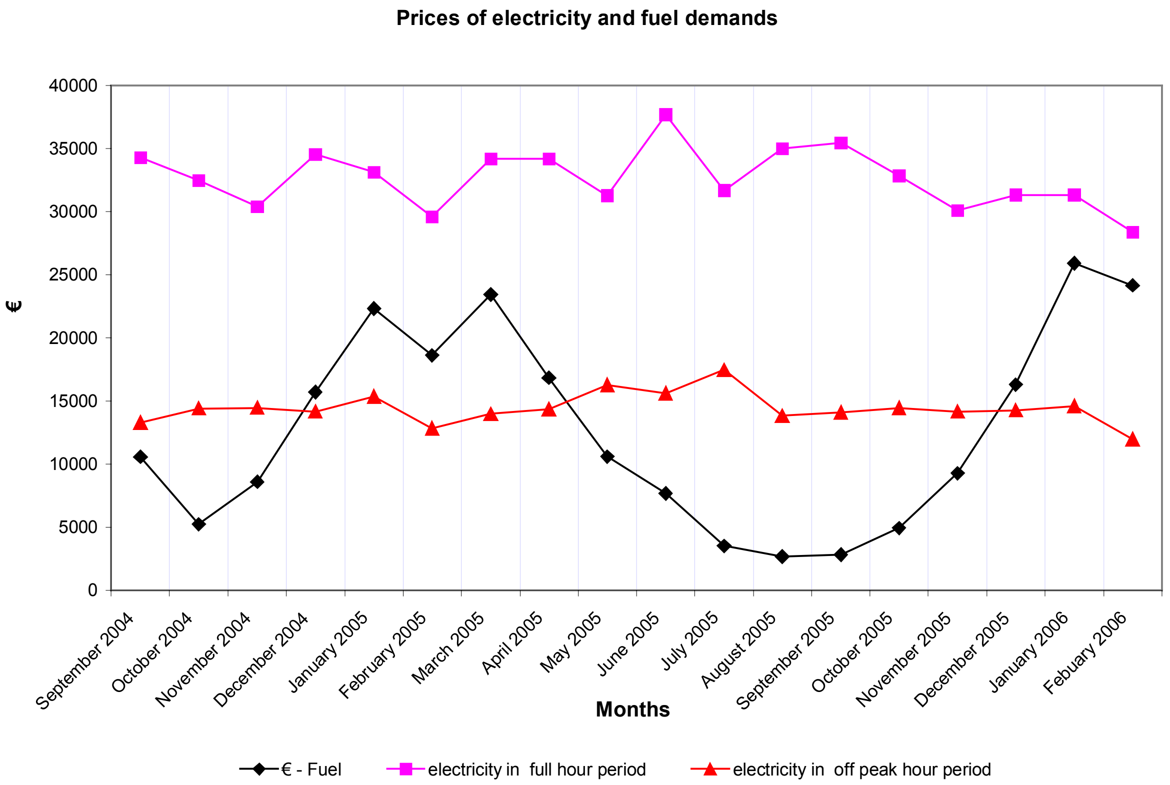

Figure 7: Costs of electricity and fuel consumptions

Fans and air distribution network

The as-built files (completed by a quick inspection) give a fair estimate of all ventilation airflow rates.
The electricity consumption of these fans can also be estimated on the basis of the following hypotheses:

- a global efficiency of 0.75
- a head of 1 500 Pa for the fans supplying to the induction units
- a head of 750 Pa for the extraction fans
- a head of 1 750 Pa for recirculation fans .

A total of 90 000 m³/h is supplied to and extracted from the office zones; this requires about 50 kW at supply fans and about 25kW at exhaust fans.
A total of 51 500 m³/h is also supplied to and extracted from the kitchen and the restaurants; this requires 19 kW at supply fans and 14 kW at exhaust fans.
A total of 54 900 m³/h of fresh air is supplied to the reception and transit spaces, which are maintained in overpressure. The extraction airflowrate is limited to 50%, i.e. 27 500 m³/h. This requires 28 kW at supply fan, 8 kW at exhaust fan and 28 kW at recirculation fan.
A total of 91 500 m³/h is supplied to and extracted from the parking area (this air is comiing from other building zones). This requires 25 kW at both supply and exhaust fans.
According to that information, the total fresh airflow rate supplied in the whole building should be 196 400 m³/h, i.e. not less than 163 m³/h per occupant when the building is fully occupied.
But this fresh air is supplied on 78 h per week, although the full occupation is of 1 200 persons, 39 h per week, only.

This means that, in average, the building is supplied with 15 319 200 m³/week for an average occupancy of 46 800 persons*h/week, i.e. at the average rate of 327 m³/h per person!
This means also that the total electricity demands of the fans is of the order of 214 kW, i.e. of about 9 W/ m2.
The energy impact of such generous ventilation is triple:

1) it encreases the sensible and latent energy consumed to bring te fresh air to required temperature and water content;
2) it encreases the fans consumptions
3) it encreases the internal loads, almost in proportion to the fans consuptions.

Retrofit opportunities

The renovation of the whole HVAC system of this building is already in way:

Some retrofits were already made on the plant and on the AHU's.

The replacement of existing induction units by more efficient devices (new induction units or fan coils, if fitting in the small space available) should make possible to run the system with higher chilled water temperature and therefore better chiller COP.

But many other retrofit opportunities can already be identified:

1) A better *fresh air management* is certainly achievable:

The air renovation time period could be reduced from 78 h to 39h per week, in order to fit with the full occupancy.
In heating period, the night set back of the thermostat is here uneconomical: it's much better to shut down the primary air supply and to use the induction units in static heating mode.
From other part, the primary air flowrate migth be reduced in mid season, according to the actual cooling demand of the offices.

2) A direct *energy recovery* loop might be installed between supply and exhaust air circuits.

3) Some *air re-circulation* would be welcome, whenever the induction units require more primary air than what is needed for indoor air quality.

4) *Variable speed* might help in reducing fans and pumps consumptions.

5) *Variable chilled water temperature* might also be introduced.

6) The possibility of using such installation in *free chilling mode* (i.e. with production of chilled water by the cooling towers only) should be always considered.

Actually, an attempt of free chilling was even done sometime ago, by adding a water-to-water heat exchanger between the condenser and the evaporator circuits (in parallel to the chillers).
For reasons which couldn't be found back, this experience failed and the system was dismanteled!

7) An *optimal control of cooling towers* should allow to reduce the electrical consumption of their fans.

8) A more careful analysis of the space and time distributions of heating and cooling demands would help in identifying the opportunities of *heat pumping and reversible air conditioning:*

- During almost all the heating season, the extracted air represents a “free” heat source of more than 600 kW for a heat pump, whose evaporation temperature would be fixed around 5°C.
- The air-cooled condensers of the auxiliary chillers, used all the year for the data processing offices, are other free heat sources.

More detailed calculations are required to go further in this analysis and to select the most attractive retrofit opportunities.
Adapted simulation models are badly needed at this stage. Such models are still under development…

Conclusions

The example of case study presented here tends to give a first idea of what could be the (difficult) task of the energy auditor.
One of the main difficulties comes from the too many information lacks.
Filling these lacks would require, in most case, more detailed energy records on site.
Simulation models will also help a lot, for a better interpretation of on site records and also for a safe identification of the most promising retrofit opportunities…

Reference

AUDITAC 2005: a IEE project: http://www.eva.ac.at/projekte/auditac.htm. *Newsletter* freely available on the site.

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