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Topic 9

SIMPLE MODELLING FOR ENERGY CONSUMPTION ESTIMATION IN AIR CONDITIONNED BUILDINGS

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ABSTRACT

A simplified tool for estimating energy consumption of a whole air conditionned building is presented. It is developped to help design engineers to quickly estimate HVAC solutions till initial design. HVAC systems as fan coils, variable and constant air volume are available.

The model is developped to need few inputs in such a way that it could be used very soon in the design process, when materials and building characteristics are not fixed in details.

For a better optimisation of the global energy consumption of the building, the interaction between natural and artificial lighting is taken into account, with energy characteristics of the lighting system.

Different stages are chained : calculation of energy needs of the building, calculation of the air handling unit energy rates for different components : humidifier, cooling and heating coils, calculation of primary energy rates demanded by chillers or boilers. For each system, a typical operation logic in all intermediary conditions is stated. The consumptions of auxiliary machines and the heat losses of the plant are counted. Finally, energy consumptions are determined and ventilated over different electrical price periods. So operating costs are estimated.

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NOMENCLATURE

A	area	[m^2]
C	thermal capacity	[$J/m^2.K$]
D	compressor model parameter	[-]
f	profile factor	[-]
h	enthalpy	[J/kg dry air]
h_c	convective exchange coefficient	[$W/m^2.K$]
h_r	radiative exchange coefficient	[$W/m^2.K$]
L	distribution duct length	[m]
$\dot{m}$	mass flow rate	[kg dry air/s]
NTU	number of transfert unit	[-]
P	pressure	[Pa]
$\dot{q}$	heat gain	[W/m^2]
$\dot{Q}$	energy rate	[W]
R	thermal resistance	[K/W]
u	lineic exchange coefficient	[W/m.K]
U	thermal transmittance	[$W/m^2.K$]
S	solar factor	[-]
T	temperature	[K]
t	temperature	[°C]
$\dot{v}$	volumic air flow	[m^3/s]
v'_s	specific volume rate	[m^3/kg dry air]
W	hydric loads	[kg/s]
w	humidity ratio	[kg water/kg dry air]
ε	exchanger efficiency	[-]
η	fan efficiency	[-]
Φ	thermal loads	[W]
φ	relative humidity	[%]

Constants

$c_a = 1{,}006 \dfrac{kJ}{kg.K}$	specific heat of air
$c_v = 1{,}83 \dfrac{kJ}{kg.K}$	specific heat of water vapor
$h_{lv} = 2501 \dfrac{kJ}{kg}$	water vaporisation heat

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A	indoor air	a	air
E	outdoor air	e	water
i	input of equipment	M	mixed air
o	output of equipment	S	supply air after fan
S′	supply air before fan	sat	saturation condition

1 - AIMS

More and more HVAC practicionners are submitted to the demand their clients of comparing different HVAC solutions in regards to their life cycle cost C_{LC} (for instance over 10 years), including :

I, the investment of the technical solution,
C_R , the running costs,
C_M, the maintenance costs.

Estimation of running costs supposes to evaluate energy consumptions for a typical (or conventionnal) year. This calculation has to be done in the early stage of the design process ; it is the good moment to try to optimise the building design and the system characteristics. At this stage, the designer needs simple tools for two reasons :

- materials and building characteristics are not fixed in details, inputs must be simplified ;
- the designer has a few hours to spare for this task.

The simplified tool here presented, called ConsoClim, is under development in the continuity of AICVF guidelines [AICVF, 1997] for energy estimation in air conditionned buildings. It is designed to be used with few simple inputs, and leads to an evaluation of energy consumption on a complete year . The flow chart - figure 1 - shows how the calculations are processed. It indicates that user has to separate the building in different zones. This cutting of the whole building is based on

- use of the local,
- thermal behaviour (different loads),
- and different air handling systems.

First calculations must be processed zone by zone. Coupling can exist between systems because of return air. The required energy rates are then gathered according to the heating or cooling plants that can exist in the building.

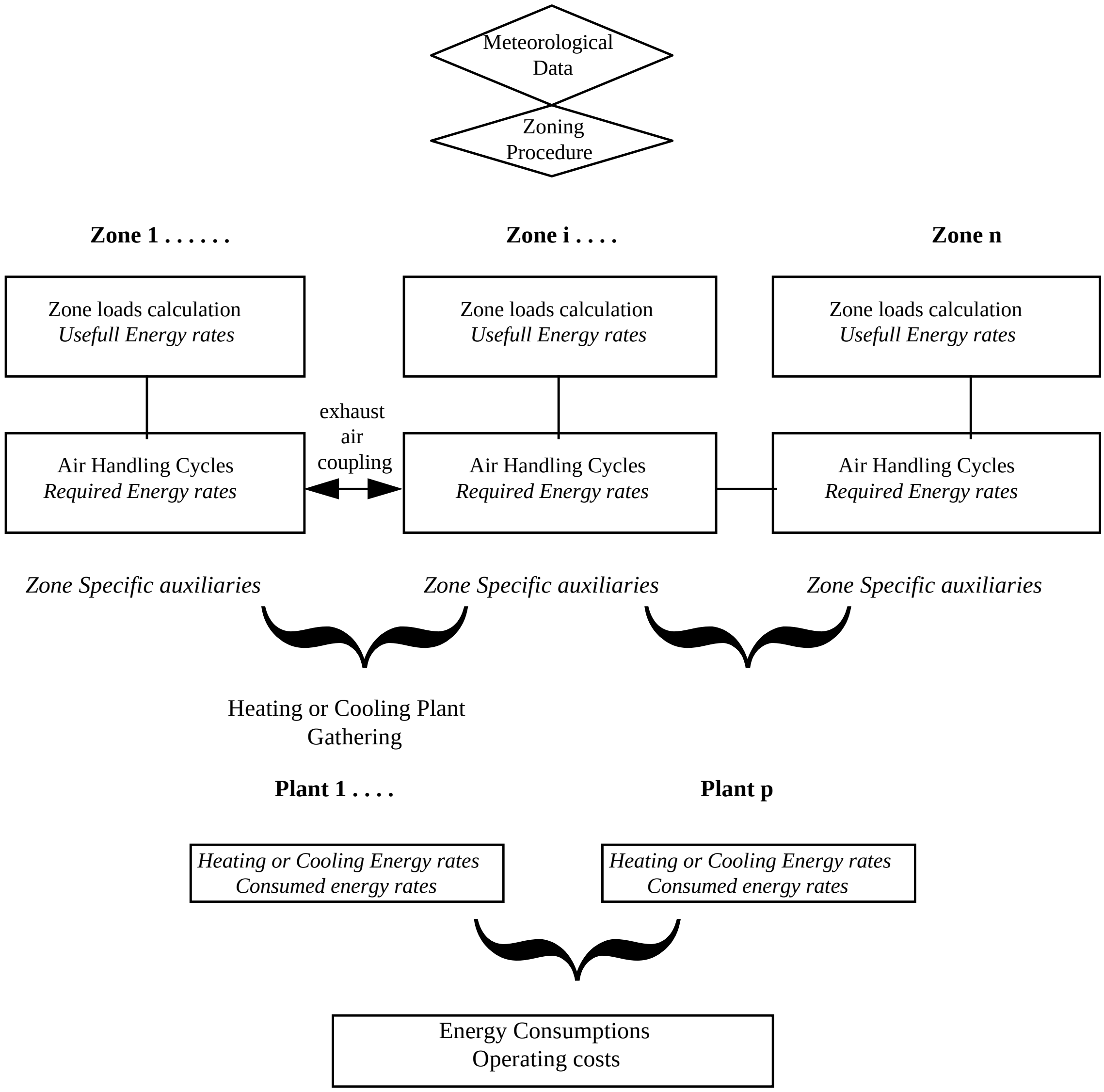


Figure 1
Flow chart of the calculations procedure

2 - Inputs of ConsoClim

The flow chart - figure 2 - describes what are the inputs of Consoclim. They are divided in 4 main categories :

- Meteorological data
- Building
- Internal loads
- HVAC systems

We precise what are the principal hypothesis made for each.

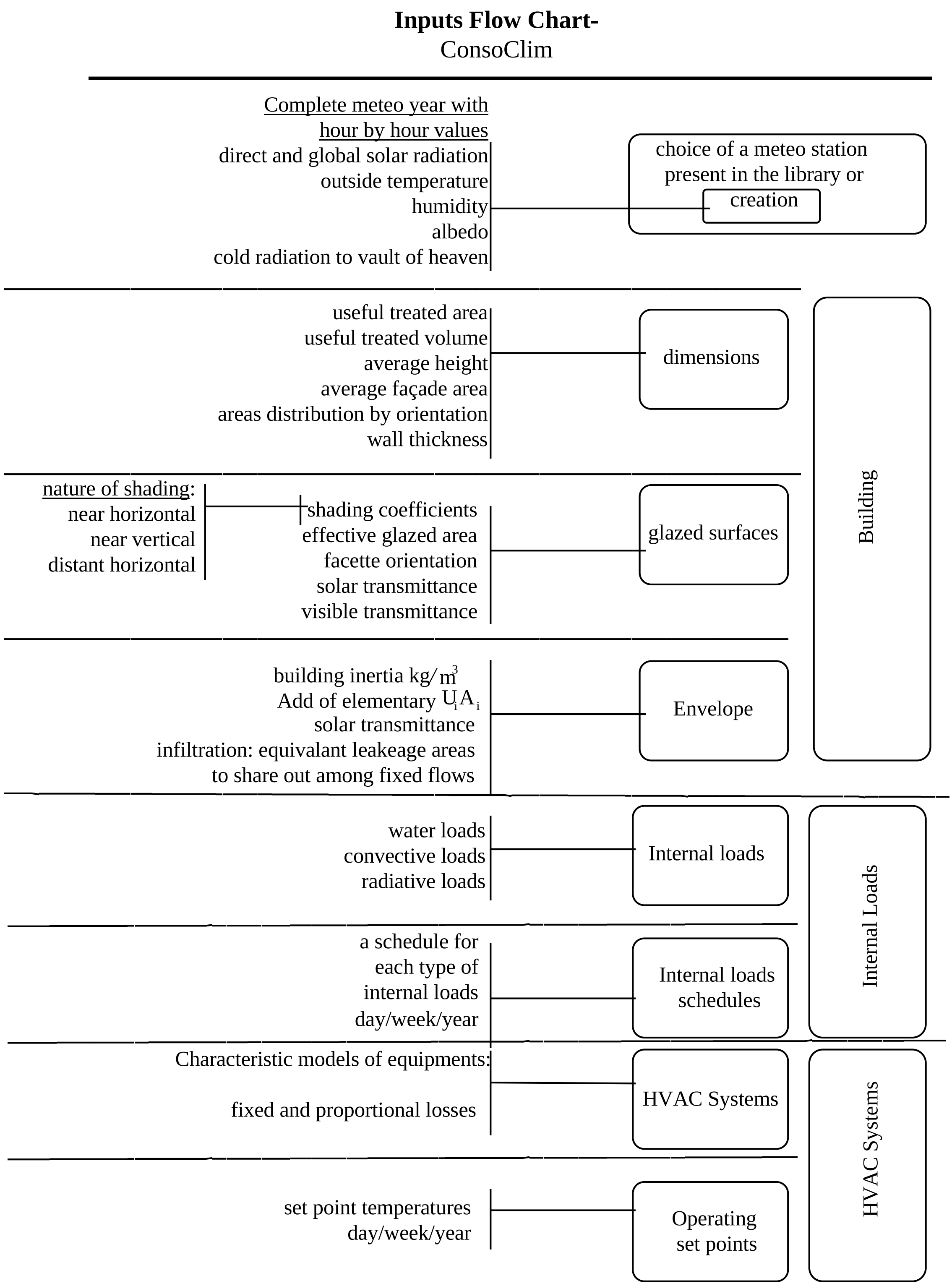


Figure 2
Inputs of Conso-Clim

2.1 - Meteorological data

Annual energy calculations can be performed with :

- conventional yearly data files, they are usefull to compare 2 solutions,
- real data files, to estimate consumption in the conditions of year y.

Especially ConsoClim can process with Test Reference Year (**TRY**) for E.E.C. countries, statistical average over ten years [H.Lund, 1983]. The data are :

Outdoor temperature : t_e [tenth of °C] - 24 hourly values
Horizontal Global : I_g [J/cm^2] - 24 hourly values
Horizontal Diffuse : I_D [J/cm^2] - 24 hourly values
Horizontal Beam : I_d [J/cm^2] - 24 hourly values
Insulation duration [hour]
Relative Humidity : HR_e [%] - 24 hourly values
Wind speed [m/s] - 24 hourly values

They exist for the following meteorological stations :

France :	Carpentras, Limoges, Macôn, Millau, Nancy, Nice, Rennes, Trappes
Belgium :	Saint Hubert, Uccle (Bruxelles), Ostende
Denmark :	Copenhagen
Eire :	Dublin,Valentia
Italy :	Trapani, Crotone, Cagliari, Foggia Amendola, Roma Ciampino, Monte Terminillo, Genova Sestri, Milano Linate, Venezia, Bolzano
Netherlands :	Vlissingen, De Bilt, Eelde
United Kigdom :	Kew (London), Abelporth, Eskdalemuir, Lerwick
Portugal :	Faro, Lisboa, Coimbra
Turkey :	Diyarbakir, Antalya, Ankara, Istanbul, Erzurum
Slovakia :	Bratislava, Hurbanovo, Strbské Pleso, Trebisov

ConsoClim can also process with Short Reference Years (SRY) for E.E.C. countries [H.Lund, 1983]. In that case, 2 weeks summarize each season. Data are available for the same meteorological than TRY. When using SRY, the Energy consumption calculated over 2 weeks E_{SRY} has to be multiplicated to give yearly energy consumption.

$E_{TRY} = E_{SRY} \times 52/8$

First of all, the original meteorological solar data are converted to have as input the diffuse horizontal irradiation and the beam radiation on a surface perpendicular to radiation. The transformation from horizontal Beam to Beam on a perpendicular surface is made using classical relations [Erbs, 1980] [Orgill and all, 1977]. It is usefull because it allows to check the values obtained around sunrise and sunset before introducing in ConsoClim.

If real year meteorological file is used, only global horizontal is known and the meteorological pre-processor has first to separate the global into diffuse and Beam. This is done using classical correlations [Orgill and all, 1977].

2.2 - Building characteristics inputs

Different components of the building are shared into 3 categories : heavy components, light components, glazing components. The following parameters are required :

- light opaque ext components (depth < 12 cm)	thermal transmittance	$U[W/m^2.K]$
	solar factor	S_f
	exposure (orientation, tilt angle)	
	area	$A\ [m^2]$
- heavy opaque ext components (depth > 12 cm)	thermal transmittance	$U[W/m^2.K]$
	solar factor	S_f
	exposure	
	area	$A\ [m^2]$
- glazing components	thermal transmittance	U[W/m2.K]
	solar global transmittance	S_p
	solar trans large wave length	S_{pwl}
	solar trans ventilated lame	S_{pvl}
	exposure	
	area	$A\ [m^2]$
- all components	areal thermal capacity	$C\ [J/m^2.K]$
	area	$A\ [m^2]$

Global capacity is calculated from different capacities $C_m = \Sigma\ A_i\ C_i\ [J/K]$

Different resistances of the model are calculated from the values of U_i , A_i and from air infiltrations.

2.3 - Internal gains

The internal gains are entered as enthalpy $\dot{q}_i$ [W] and water flow rate W_i [kg/s]. Enthalpy gain is splitted into radiative and convective terms, refering to the structure of the model that calculates indoor air temperature and radiant temperature. Default loads values exist for occupancy (different metabolism activities) [AICVF, 1997]. Other gains - lighting, machines - have to be entered by square meter. The last can be positive (gains) or negative (case displays for instance).

It is particulary important that users profiles inputs are very open because non residential buildings have very different profiles according to the type of building (offices, health buildings, education, retail, ...). A quite general input is chosen on a weekly basis.

For each type of loads (people, lighting, machines), the user will have to enter 24 x 7 hourly values. With an Excel spreadsheet interface, it is not difficult to modify typical profiles. The profile is a fraction value f_h that mutiplicates the maximum value of $\dot{q}_i$ in W/m^2 and W_i in kg/(s.m^2). Such a profile is given on figure 3 - exemple of occupancy profile in offices.

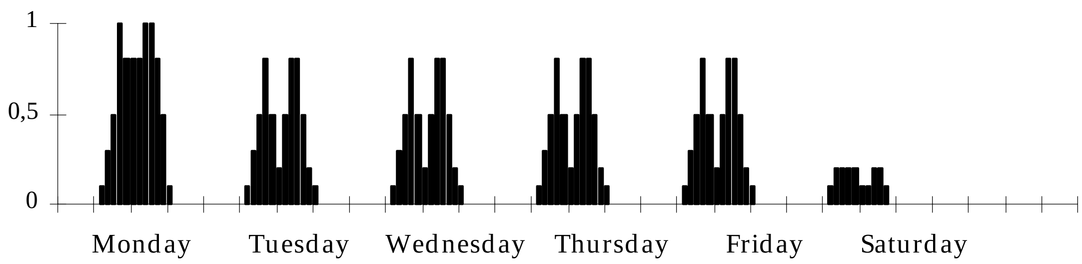


Figure 3
hourly occupancy profile in offices for a week - f_h

To take into account weekly variations such as : occupancy variation due to vacations, natural lighting in summer etc..., a week coefficient f_w is also entered for each type of internal gains - figure 4.

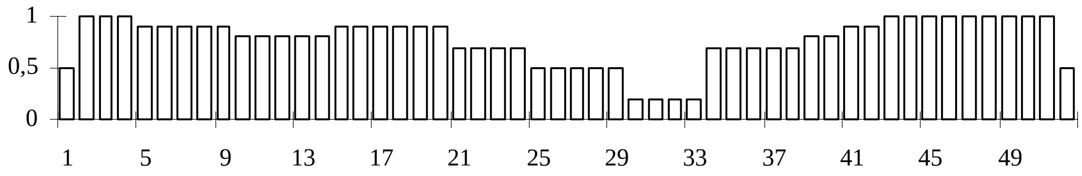


Figure 4

Week weight - f_w

Finally heat gains at each hour are :

$\dot{q}_i(h) = \dot{q}_i^{max}.f_h.f_w$ [W/m²]

$W_i(h) = W_i^{max}.f_h.f_w$ [kg/s.m²]

2.4 - HVAC systems

Operation setpoints

Figure 5 illustrates ambient conditions required as an input. It defines a zone acceptable. Inside this zone the air conditionning system is off. Only ventilation is on.

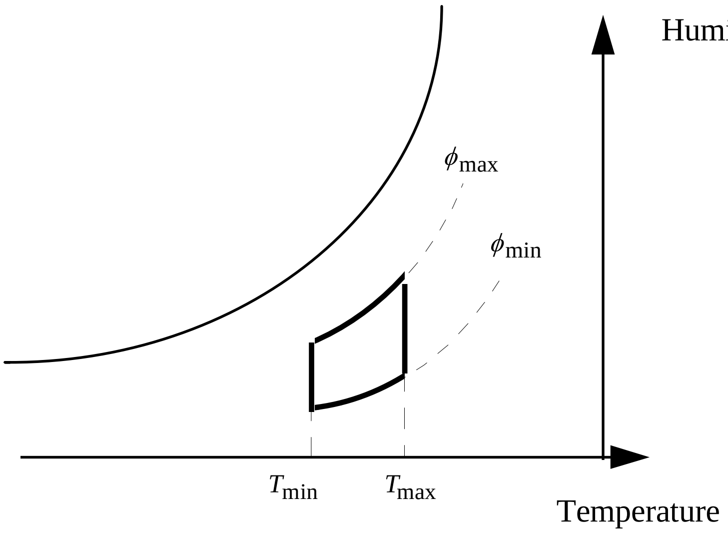


Figure 5
Ambient conditions setted in the building zone

The setted values are result of a profile. Keeping notations of Figure 5 : φ_{min}, φ_{max}, T_{min}, T_{max} the setted values can be presented in the same way that internal gains. Figure 6 gives an exemple for temperature set points on a weekly basis. Same inputs must be entered for humidity setpoints. To take into account vacations periods, an other profile has to be entered and a flag coefficient for each of the 52 weeks of the year indicates if the week is on or off vacations.

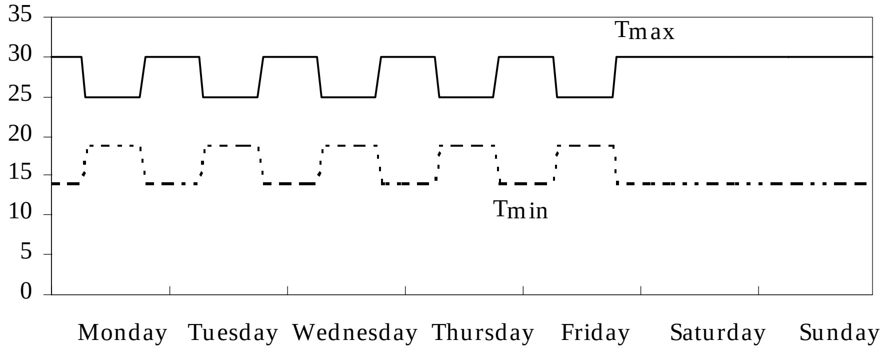


Figure 6
Set points temperature profile for a week

Characteristics models for equipment

They are derived from manufacturers indications. Especially, for chillers, different kind of models can be used. They are more accurate if data are available for different operating conditions and particularly under partial load behaviour.

3 - CALCULATIONS

The flow chart of figure 1 is now detailed to present how required energy rates are calculated. First of all, loads calculation (Φ and W) is processed ; this is explained in paragraph 3.1. Then, supply conditions (T_S, w_S) are determined according to the supply air flow rate. If it is constant,the volumic flow rate is keeped constant and mass flow rate is calculated with T_S, w_S , if it is variable, the supply temperature is considered constant in cooling mode. Determination of air handling cycles will be presented in paragraph 3.2, leading to required energy rates.

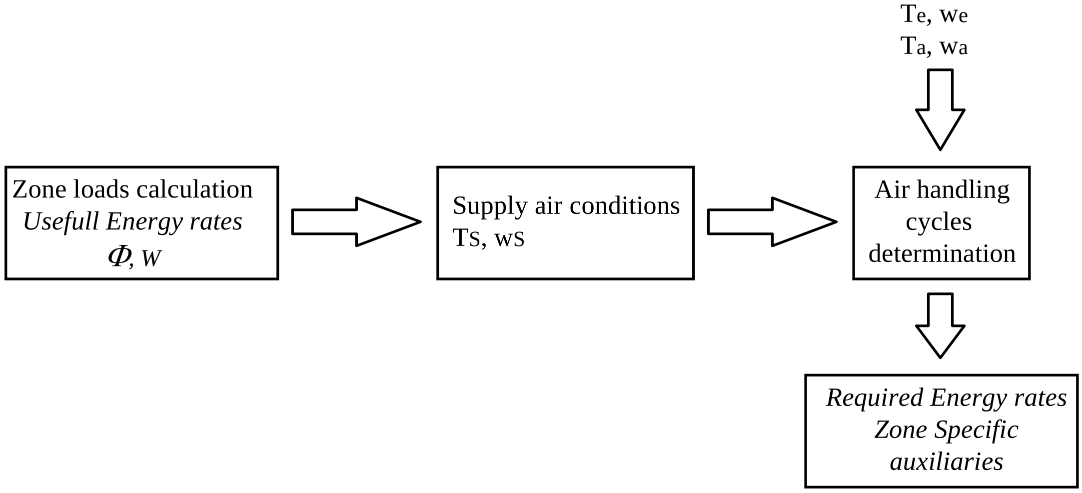


Figure 7 - Detailed flow chart for calculation of required energy rates

3.1 - Loads calculations

Model of the building

The calculation model is based on the simplification of the heat transfers between the internal and external environment reported in figure 8.

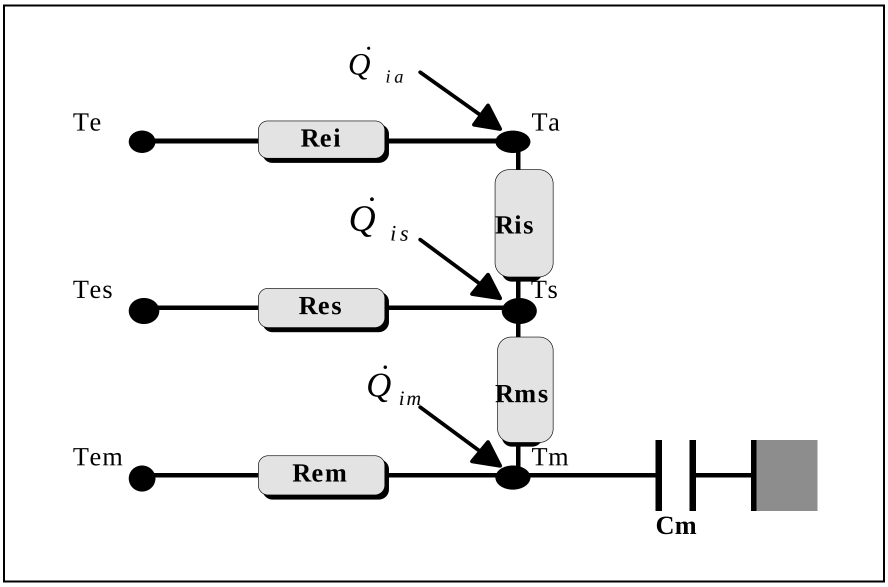


Figure 8

Equivalent electric representation of the building model

According to this equivalent electric representation, the envelope components are divided as :

- light opaque external components
- heavy opaque external components
- glazing components
- internal components

The relevant nodes are defined related to :

T_a	: indoor air temperature
T_m	: mass temperature
T_s	: T_{star} temperature defined as $(h_c.T_a + h_r.T_{rm})/(h_c + h_r)$ T_{rm} being the medium radiant temperature
T_e	: outdoor air temperature
T_{es}	: equivalent sun-air temperature of the light external components
T_{em}	: equivalent sun-air temperature of the heavy external components

The equivalent outdoor temperatures T_{es}, T_{em} take into account long wave radiation to the sky vault, the solar radiation ant the air temperatures

The following equivalent resistances (K/W) and thermal capacity (J/K) between the internal and the external environment are considered :

R_{ei}	: thermal resistance due to air ventilation
R_{es}, R_{em}	: thermal resistances of external components between outside and inside
R_{is}, R_{ms}	: thermal resistance correspondent to the heat exchanges between the internal surfaces and the internal air
C_m	: thermal capacity of the enclosure elements

The heat flux (W) considered are :

$\dot{Q}_{ia}$	: to air node due to internal sources or direct solar radiation or convective heat gains due to window ventilated inner air layer
$\dot{Q}_{is}$	: to star node due to internal sources or direct solar radiation
$\dot{Q}_{im}$	: to mass node due to internal sources or direct solar radiation

Compared to other programs, this calculation model requires less detailed input datas as more detailed codes (as TRNSYS for example) : for example, external components are only described by their U value, solar factor and depth.

On the other hand, it enables to distinguish between the indoor operative and air temperature, which can be up to 2 or 3 K for air conditioned buildings when most of the simplified method only uses one value for both.

It has been validated using TRNSYS as a reference. (figure 9) . Starting of a base room, main parameters where modified and jearly heating and cooling needs calculated :

The base room is a typical office room south oriented, with a window solar factor of 0.47 and a medium thermal inertia . The internal gains are set to 30 W/m^2 and the ventilation rate to 1 a.c./h, both during occupancy. The air conditioning system runs continuously with a heating set point of 20 °C and a cooling set point of 25 °C.

The different cases are modified as follows :

- orientation (OR2,3,4 : W,N,E),
- solar factor of windows (SV1 : 0.77 ; SV3 : 0.17),
- inertia (IN1 : light IN3 : heavy)
- internal gains (AI1 : 0 ; AI3 : 60 W/m^2)
- ventilation (VE1 : no ventilation, VE3 : 1a.c./h continuous)
- system running (MCD : running during occupancy only; TIC : set points 22.4-22.6°C) .

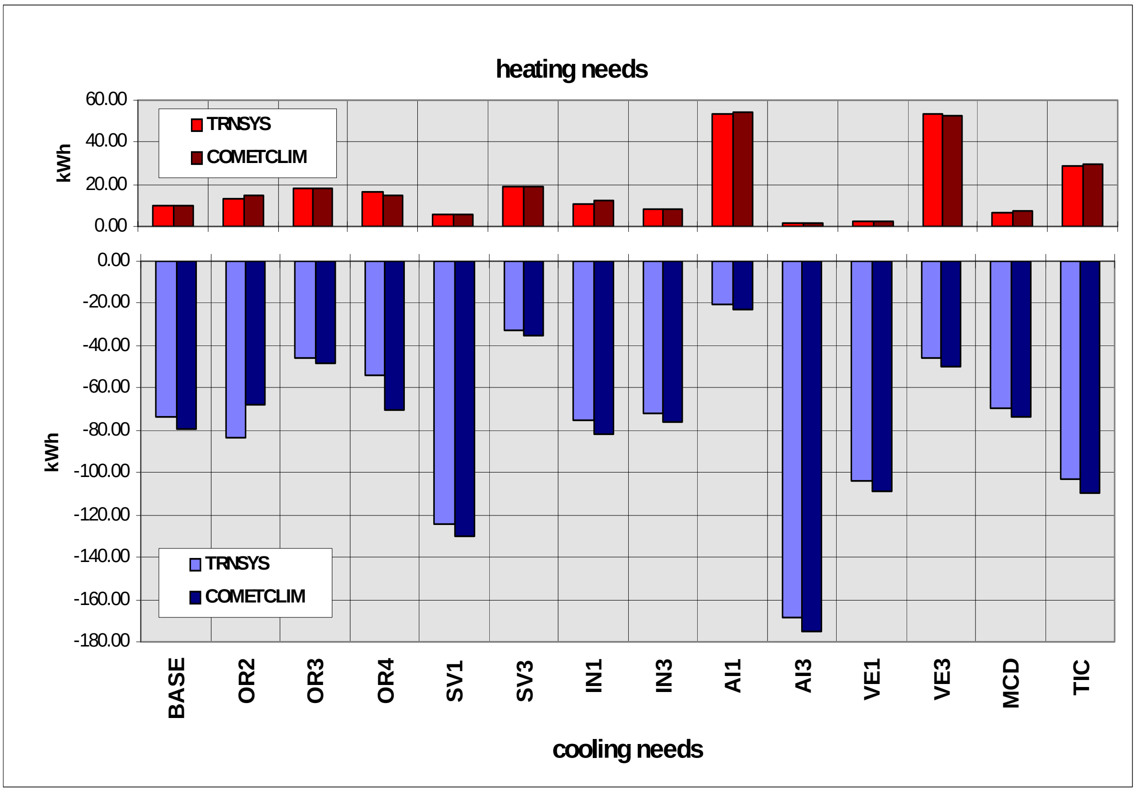


figure 9
Comparison between TRNSYS and COMET

Use of the model for building loads calculations

At each hour, the global load Φ and the hydric load W are calculated according to the dead band as explained below :
If the indoor air without air handling (T_a, φ_a) is located in the area limited by [φ_{min}, φ_{max}] and [T_{min}, T_{max}] the values of Φ and W are zero. In the other cases Φ and W are calculated to have the resultant indoor air inside the area chosing the reacher limit.

If $T_a < T_{min}$ Φ is negative and calculated to have $T_a = T_{min}$
If $T_a > T_{max}$ Φ is positive and calculated to have $T_a = T_{max}$

For humidity, two types of air conditionning systems must be considered :

- Those where a humidity control exist, in that case :

If $\varphi_a < \varphi_{min}$ W is negative and calculated to have $\varphi_a = \varphi_{min}$
If $\varphi_a > \varphi_{max}$ W is positive and calculated to have $\varphi_a = \varphi_{max}$

- Those where humidity is not controlled, in that case :

φ_a is calculated from moisture gains and according to the surface temperature of the coil which gives the deshumidification.

We give hereafter an example of results. The room is a typical office room in Paris area with 150 W internal gains during occupancy and no window solar protection.

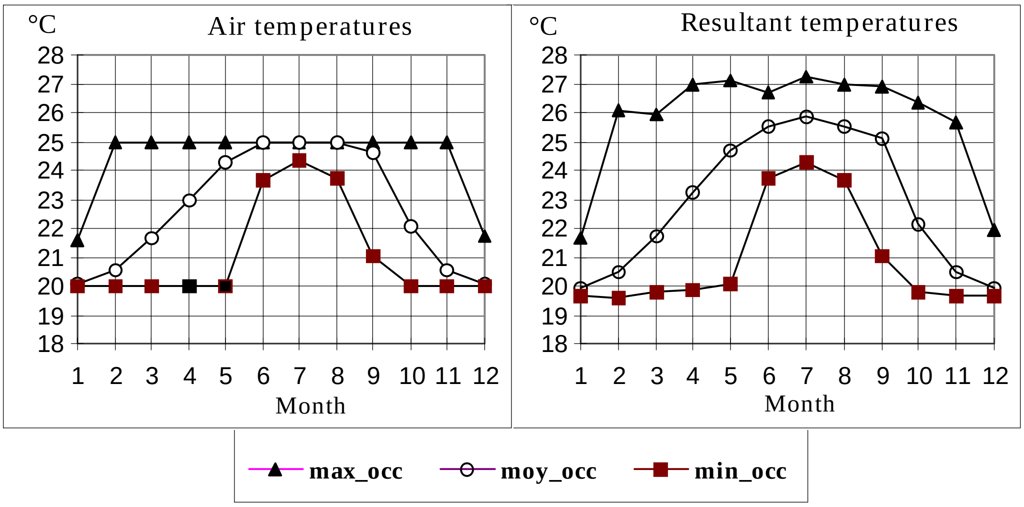


Figure 10

Air temperature response of the building

Though the air temperature remains always between 20°C and 25 °C, it can be noticed that the resultant temperature climbs up to 27 °C in summer

3.2 Supply air conditions calculation

The aim is to obtain T_S and w_S having Φ, W and $\dot{v}_S$. h_S, w_S and $\dot{m}_S$ are calculed by solving equations set made by the three following equations:

$$\dot{m}_S = \frac{\dot{v}_S}{v'_S}$$

$$\phi = \dot{m}_S.(h_A - h_S) \quad \text{energy balance}$$

$$W = \dot{m}_S.(w_A - w_S) \quad \text{water mass balance}$$

We consider a constant air volume system. It means $\dot{v}_s$ is constant. $\dot{m}_S$ is obtained by using iterative calculation. T_S is evaluated from enthalpy using:

$$T_S = \frac{h_S - h_{lv}.w_S}{c_a + w_S.c_v} + T_0 \qquad \text{having } T_0 = 273{,}15 \text{ K}$$

and water mass balance gives w_S:

$$w_S = w_A - \frac{W}{\dot{m}_S}$$

3.3 Air handling cycles

At each time step also, Air Handling cycle is chosen according to the compared positions in psychrometric chart of outdoor air (E), indoor air (A), and supply air (S). Hygienic requirements are taken into account as a parameter.

Take for exemple a Constant Air Volume system represented on figure 11.

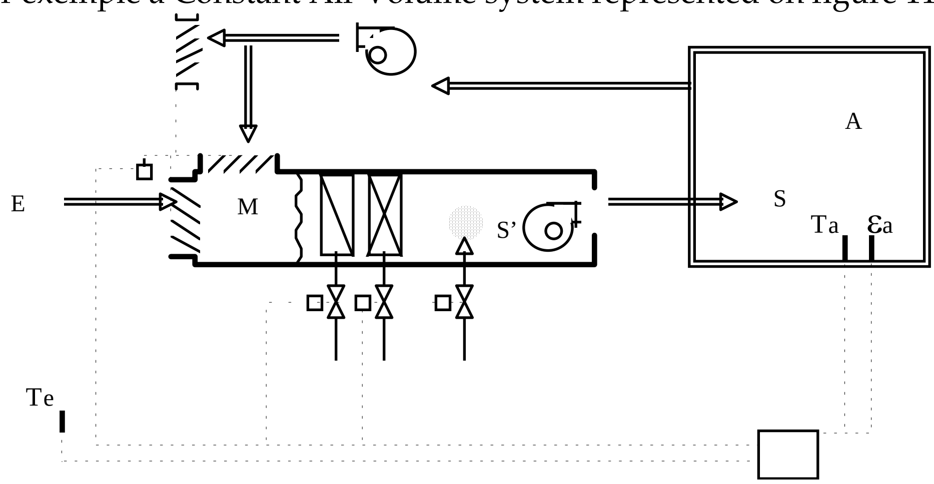


Figure 11

Constant Air Volume System

The air is handled in constant air volume central station air. Air distribution is processed using ducts, supply air outlets and exhaust grilles. The system is composed of :

- mixing box with energy optimizer, including hygienic requirements
- cooling coil and heating coil
- steam humidifier
- supply and exhaust fans

Hot and cold water supplied to the coils are produced separately of the central station air handling unit by boiler and water chiller. The central station is controlled by an ambient temperature sensor.

To simplify the following schemes, supply line from S to A is not drawn and only S′ (air before the fan) is represented. Position of S′ towards A is equivalent to a sign condition on Φ and W. S′ can be evaluated by using an approximation:

$$T_{S'}=T_S-1$$

$$w_{S'}=w_S$$

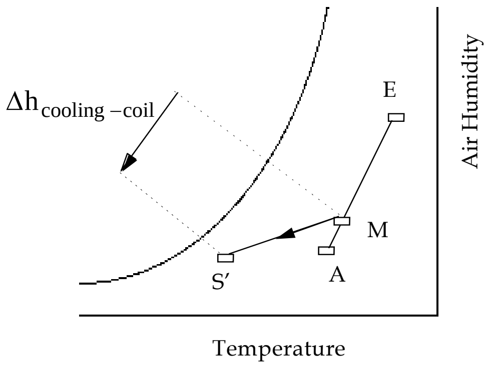
But S′ can also be precisely calculated, using fan efficiency definition, if ΔP and η are known at this design stage:

$$h_{S'}-h_S=\mathrm{v'}_S\,.\dot{m}_S\,\frac{\Delta P}{\eta}$$

having η fan efficiency

ΔP pressure drop in the circuit

For all cycles, the supply air flow rate in the exemple is constant. Used equations are presented after the psychrometric charts.



This air handling cycle occurs when:

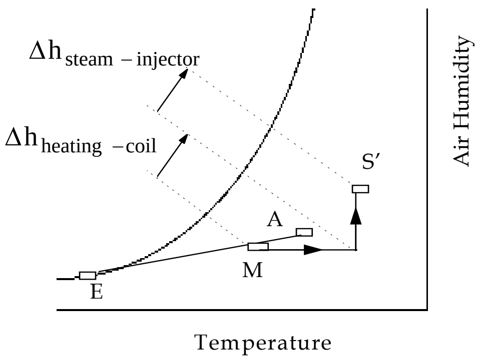
$\Phi>0$

$W>0$

$T_A < T_E$

Fresh air flow rate is constant and equal to hygienic requirements. The cooling coil cools the air with condensation.

Equations used to determine the distributed energy rates are: 1-2-3-4-5-6-7



This air handling cycle occurs when:

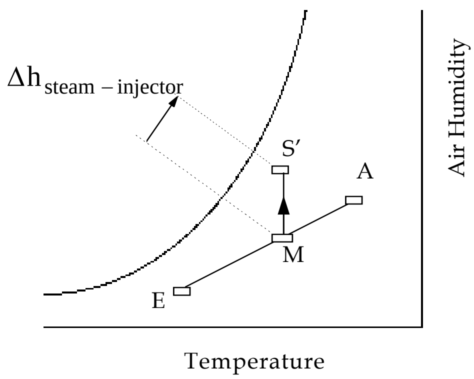
$\Phi<0$

$W>0$

$T_A > T_E$

Fresh air flow rate is constant. Supply temperature is variable. Indoor humidity is not controlled. Heating coil and steam injector handl the air.

Equations used: 1-2-7-8-9-10-11-12-13-14



This air handling cycle occurs when:

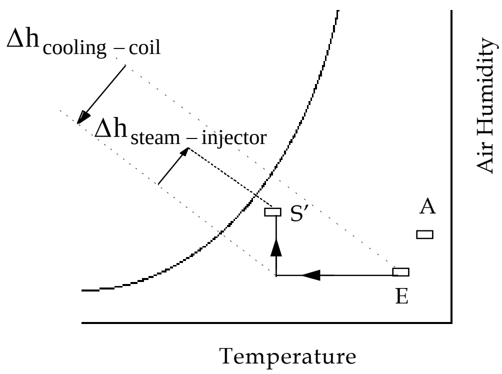
$\Phi>0$

$W<0$

$T_A > T_E$

Fresh air flow rate is used as a cooling source. Steam injector is on.

Equations used: 1-2-10-11-12-13-14



This air handling cycle occurs when:

$\Phi<0$

$W<0$

$T_A > T_E$

Fresh air flow rate is maximum and represents the total air flow rate. This air is used as a cooling source. This free cooling is completed by the the cooling coil without condensation.

Equations used: 3-4-5-6-7-10-11-12-13-14

3.4 Distributed energy rates calculations

Main equations used to determine the distributed energy rate [Dumitru, 1996] [ASHRAE, 1993] are presented hereafter for each equipment.

Mixing Box:
Temperature and humidity are calculed by using equations of mass and energy conservation.

$$(1) \qquad h_{mo} = \frac{h_{ai}.\dot{m}_{ai} + h_{ei}.\dot{m}_{ei}}{\dot{m}_{mo}} \qquad\qquad (2) \qquad w_{mo} = \frac{w_{ai}.\dot{m}_{ai} + w_{ei}.\dot{m}_{ei}}{\dot{m}_{mo}}$$

Notice that thermal loads introduced by outdoor air are calculated by:

$$\dot{Q}_{\text{outdoor air}} = \dot{m}_{ai}.(h_E - h_A)$$

Cooling Coil:
Cooling coil model is based on the relation between efficiency and NTU. This model takes into account water mass variation due to condensation. Equation used to determine the cooling energy rate is:

$$(3) \qquad \dot{Q}_{cool} = \dot{m}_a.(h_{ao} - h_{ai})$$

with the following expressions for enthalpy:

$$(4) \qquad h_{ai} = c_a.t_{ai} + w_{ai}.(h_{lv} + c_v.t_{ai})$$

$$(5) \qquad h_{ao} = h_{ai} - \varepsilon \frac{C_{\min}}{C_a}(h_{ai} - h_{ee,sat})$$

$$(6) \qquad h_{ei,sat} = c_a.t_{ei} + w_{sat}.(h_{lv} + c_v.t_{ei})$$

and the relation between efficiency and NTU for counter flow exchanger

$$(7) \qquad \varepsilon = \frac{1 - e^{-NTU.(1-C)}}{1 - C.e^{-NTU.(1-C)}}$$

Heating Coil:
Heating coil model is based too on the relation between efficiency and NTU. But in that case the model considers heat exchange without mass variation. Equation used to determine the heating energy rate is:

$$(8) \qquad \dot{Q}_{heat} = \dot{m}_a(c_a + w_a.c_v).(t_{ao} - t_{ai})$$

with the following expression for the output air temperature:

$$(9) \qquad t_{ao} = t_{ai} + \varepsilon.\frac{C_{\min}}{C_a}.(t_{ei} - t_{ai})$$

and the relation between efficiency and NTU for counter flow exchanger

$$(7) \qquad \varepsilon = \frac{1 - e^{-NTU.(1-C)}}{1 - C.e^{-NTU.(1-C)}}$$

Steam Injector:
Steam injector model is based on the equations of mass and energy conservation. Equation used to determine the energy rate due to humidification is:

$$(10) \qquad \dot{Q}_{hum} = \dot{m}_a.(h_{ao} - h_{ai})$$

with the following expressions for enthalpy:

$$(11) \qquad h_{ai} = c_a.t_{ai} + w_{ai}.(h_{lv} + c_v.t_{ai})$$

$$(12) \qquad h_{ao} = c_a.t_{ao} + w_{ao}.(h_{lv} + c_v.t_{ao})$$

Energy balance and water mass balance give output air temperature and humidity:

$$(13) \qquad w_{ao} = w_{ai} + \frac{\dot{m}_v}{\dot{m}_a}$$

$$(14) \qquad t_{ao} = \frac{\dot{m}_v.c_v.t_v + \dot{m}_a.(c_a + c_v.w_{ai}).t_{ai}}{\dot{m}_v.c_v + \dot{m}_a.(c_a + c_v.w_{ai})}$$

These calculations are processed each time step. It is also possible to reduce the number of iterations sorting the annual hourly data [Casari, 1992]

It is possible to check calculations by an energy balance and a water mass balance on building, considering that energy rates used to cool air are negative and energy rates used to heat are positive:

$$\Phi + \dot{Q}_{\text{outdoor air}} + \dot{Q}_{fan} + \dot{Q}_{cool} + \dot{Q}_{heat} + \dot{Q}_{hum} = 0$$

$$W + W_{\text{outdoor air}} + W_{cool} + W_{hum} = 0$$

3.5 Required energy rates calculations

It is necessary to take into account thermal gains or loses introduced by distribution network. These loses may be constant or proportional to distributed or plant energy rates. With lineic exchange coefficient of each type of distribution duct and measurements of distribution network, ducts heat are given by:

$$\dot{Q}_{duct} = \sum_i u_i.L_i.(T_S - T_{EC})$$

T_{EC} is outside temperature for ducts. It may be equal with ambient temperature or an intermediate temperature of outside of building temperature.

$$\dot{Q}_{cool+} = \dot{Q}_{cool} + \dot{Q}_{duct}$$

3.6 Plant energy rate calculations

Nominal operation, corresponding to maximal thermal load, determine the capacity of HVAC system. Nevertheless, nominal operation is infrequently used. Most of time, usual operation is part load running, over 60 to 90% of nominal energy rates. So it is important to calculate electric comsumption with a modelisation based on part load operation.

Most existing models require many manufacturer data, which are often non-available. It is so necessary to use a characteristic model requiring few manufacturer data. We use a semi-theoric model based on a modelisation

equation build on theoric performance coefficient and regression on few manufacturer data (at least nominal operation point) [Stan, 1995].
This model is based on equations presented hereafter:

$$\frac{\dot{Q}_{elec}}{\dot{Q}_{evap}} = \left(\frac{\dot{Q}_{elec}}{\dot{Q}_{evap}}\right)_{no\min al} .\left(D_1 + D_2.\Delta T + D_3.\Delta T^2\right)$$

$$\dot{Q}_{evap} = \dot{Q}_{evap,no\min al} .(D_4 + D_5.\Delta T + D_6.\Delta T^2)$$

$$\text{where } \Delta T = \left(\frac{T_E}{T_S}\right) - \left(\frac{T_E}{T_S}\right)_{no\min al}$$

Parameters of this model were evaluated for some chiller units of rooftop. In that case using only nominal operation point, mean error on COP is 7% [Branescu, 1996].

$\dot{Q}_{evap}$ is different from $\dot{Q}_{cool+}$ required and the chiller operates in part load characterized by :

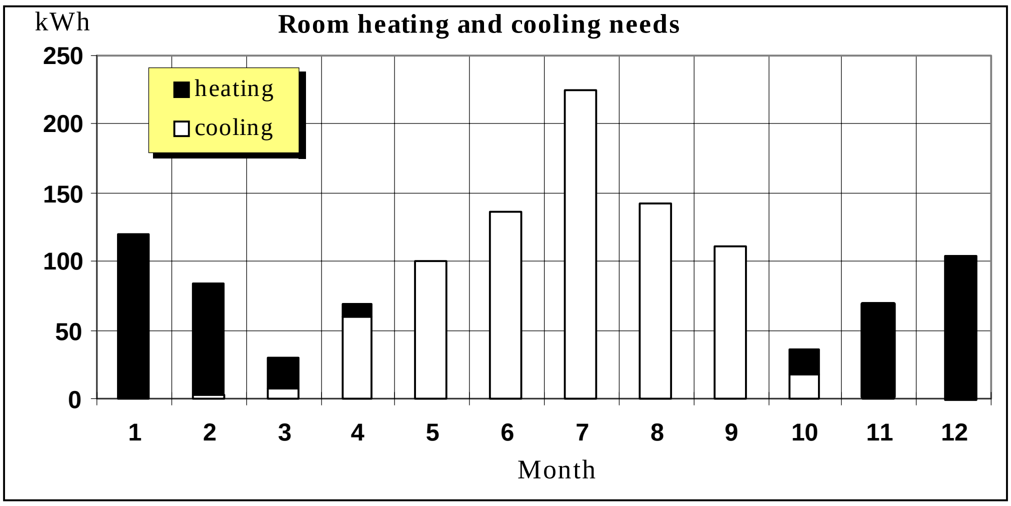
$$\kappa = \frac{\dot{Q}_{cool+}}{\dot{Q}_{evap}}$$

The compressor energy rates in part loads $\dot{Q}_{comp}$ is obtained from $\dot{Q}_{elec}$ in full load using a correlation [Peitsman, 1988]:

$$\dot{Q}_{comp} = \dot{Q}_{elec}.[0{,}82.(\kappa - 1) + 1]$$

3.7 Perspectives

The heating and cooling needs are here the room loads. As mentioned before, the coupling to the heating and cooling plant model will make it possible to express the results directly in terms of energy needs (heating, cooling and fan energies)



To achieve the simplified tool for estimating energy consumption of a whole air conditionned building, we have to introduced developped equipment models in the existing building model.
Then, energy consumption of the HVAC system of a building will be calculated. Then Energy consumptions could be distributed among type of energy. In order to obtain energy cost, unit price of each energy has to be introduced. In France, these prices are diffused by EdF (Electricité de France) and GdF (gaz de France). Differents prices could be compared to reduce energy cost.

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