Validation Of A Cooling Loads Calculation Of An Office Building In Rabat Morocco Based On Manuel Heat Balance (Carrier Method)

Ouiame TAIMOURI, Ahmed SOUISSI

Abstract:— Heating, ventilation and air conditioning (HVAC) is the largest source of energy consumption, they are one of the fields most concerned by the energy efficiency. In this research and in order to create a new, ecological and more sustainable market, we took as case of study an office building located in Rabat Morocco. We initiated a study for the latter on the development of High Energy Efficiency Buildings (HEEB) to be able to contribute to the reduction of electrical energy consumption. The objective of this article, is to establish a manual heat balance by considering external and internal heat sources and the number of occupancy on a typical area of the building studied, to validate the results already obtained by a software program for calculating the cooling loads, which will allow us to judge the technical and economic feasibility of integrating a new solar cooling technology instead of conventional technologies.

Index Terms:— Heating, ventilation and air conditioning (HVAC), Energy consumption, Energy efficiency, High Energy Efficiency Buildings (HEEB), Rabat Morocco , Cooling loads, Heat balance, Heat sources, Solar cooling technology.

1 INTRODUCTION

Fdaced with the challenges of the rapid increase in energy consumption in Morocco, where more than 95% of energy needs are imported from abroad, energy efficiency has become one of the key issues in all fields of activity. In the latter, Heating and air conditioning installations are among the fields most concerned to achieve these objectives. Indeed, over the past fifteen years, the requirements of occupants of tertiary buildings have changed significantly. There is an increasingly rigorous demand for comfort, especially in the summer months. This increase in air-conditioning needs leads to a significant increase in electricity consumption because the conventional technologies (mechanical vapor compression) used are very energy-intensive. A recent study, carried out by the Moroccan Energy Efficiency Agency (AMEE) on the characterization of the Moroccan HVAC market, estimated the energy used in the building sector [1]:

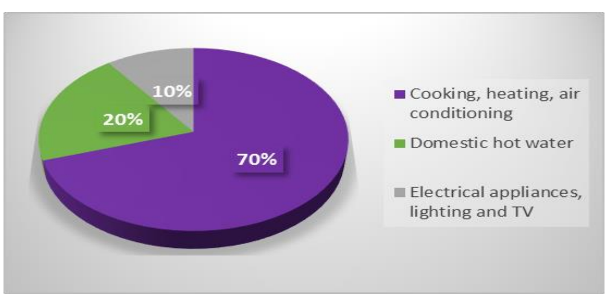

***Fig 1**: Distribution of energy used in residential buildings.*

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To address this issue, Morocco has set itself the objective of achieving an energy saving of 12% by 2020 and 15% by 2030, through the implementation of a green plan in the variouseconomic sectors. In particular, the construction sector, which represents about 25% of Morocco's total consumption, 18% of which is residential and 7% tertiary[2][3]. The objective of this research is the establishment of a manual heat balance on a single area of our office building, in order to validate the results already obtained by a software program for calculating the thermal loads. We will compare the results of the manual calculation with those obtained by the software program. In order for the assessment to be carried out with care, we consider our building as an assembly of coupled thermal systems (zones), each zone is characterized by temperature, pressure and humidity. We will start by presenting the mass and thermal balances governing mass and heat transfer within an area and adopt a so-called CARRIER method that will allow us to simplify the calculations. This method consists in calculating the thermal gains of a room for a basic indoor and outdoor temperature (set by Moroccan standards). Below we list the main elements that we will take into consideration in the heat balance.

- Orientation of the area
- Architectural plans,
- Dimensions of the area.
- Building materials
- Colors of the materials
- Conditions outside the zone
- Conditions to be maintained inside the zone
- Occupants.

For the manual balance, we will choose an area that appears to have significant heat gains.
The area where we will perform the manual heat balance is as follows (zone 1).

2 DESCRIPTION OF THE CASE OF THE STUDY:

2.1 Dimensions and Orientation of the area:

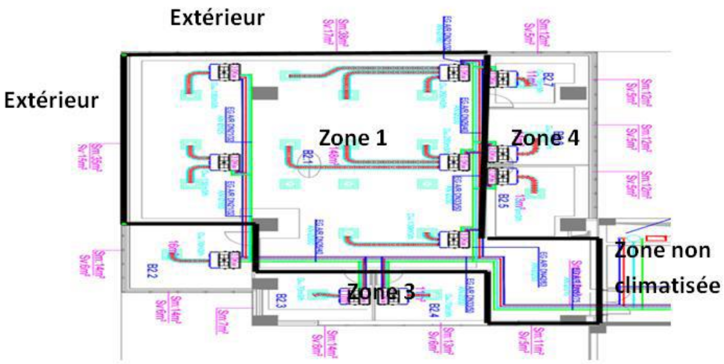

Fig 2: Orientation of the area (1)

With: Sm: Gross wall surface area / SV: Window surface area

2.2 Characteristics of the envelope of the zone:

The tables below summarize the characteristics of the envelope of the zone (1).

Walls	Overall transmission coefficient K W/m.2°C	Specific weight (kg/m2)	Color
	0.65	365	Clear

Exposure (walls)	South-East	Northeast	Northwestern	Northeast
Gross surface area in m²	38	35	11	-

***Table 1**: Characteristics of the walls of the area (1)*

Glazing	Overall transmission coefficient K W/m.2°C	Solar factor	Curtains	
	2.4	0.65	Non	
Exposure (walls)	South-East	Northeast	Northwestern	Northeast
Gross surface area in m²	17	15	5	-

***Table 2**: Characteristics of the glazed walls in zone (1)*

2.3 Internal and external conditions to be maintained

The external and internal conditions will be in accordance with the provisions of the Moroccan standard NM ISO 7730 (Ergonomics of thermal environments: Analytical determination and interpretation of thermal comfort by calculating the PMV indices (average predictable vote) and PPD (expected percentage of unsatisfied) and by local thermal comfort criteria). In the general case, we refer to the conditions prescribed in the following table [4]:

Period	Dry temperature	Relative humidity
Summer	26°C	60%

***Table 3:** Internal conditions to be maintained in zone (1)*

Period	Dry temperature	Wet temperature
Summer	35°C	60%

***Table 4:** Basic external condition*

Occupation: The number of persons admitted for the establishment of the heat balance is 26 in the chosen area.

3 Calculation Of The Cooling Capacity Of The Installation

We consider a zone i in contact with N+1 zones, with (zone 0 is the outside)

3.1 Equations of the zone i balance sheet

Working assumptions [5].

- The Thermal Zone i corresponds to a piece or a set of pieces. It is assumed homogeneous in temperature and humidity.
- The walls are delimited in such a way as to present a homogeneous behavior over their entire surface, the heat transfers taking place in the direction of thickness.
- The air behaves like an incompressible fluid.
- The absorption, by the walls, of the water vapour contained in the air is negligible in the face of the phenomena of mixing and diffusion through the rooms of a building.
- Contributions due to thermal bridges are considered negligible in comfort applications

a. Convective enthalpy balance equation over a zone

Enthalpy Accumulation in zone i =
incoming air wet Enthalpy – air wet Enthalpy leaving the area + produced Enthalpy

So: (1)

$$\frac{dH(i)}{dt} = H^e(i) - H^s(i) + \left[\sum_{j=i}^{NP(i)} S_j \times h_{cij} \times (T_{sij}(i) - T_a(i))\right] + SV + P_S + P_1 + AL_S + AI_1$$

With

$$H(i) = H_{sensitive} + H_{latent}$$

$$= m_{as} Cp_{as} \times (T_a(i) - T_{ref}) + m_{as} \times h_a(i) \times (\mathrm{L_v} + Cp_{ve} \times (T_a(\mathrm{n}) - T_{ref}))$$

We can neglect

$$m_{as} \times Cp_{ve} \times h_a(i) \times (T_a(\mathrm{n}) - T_{ref}) \text{ infront } m_{as} \times h_a(i) \times \mathrm{L_v}$$

This simplification allows to write 2 enthalpy balance equations:

- Sensitive balance sheet: function only of Ta (i)
- Latent balance sheet: function only of ha (i)

In fact, it is (Lv =2500 kJ/kg, Cpve=1.96$kj/kg.K$) [6]

$$H(i) = m_{as} \times Cp_{as} \times T_a(i) + m_{as} \times h_a(i) \times \mathrm{L}_v$$

$$H^e(i) = \sum_{n=0}^{N} Q_{as}^e(n,i) \times (Cp_{as} \times T_a(\mathrm{n}) + h_a(\mathrm{n}) \times \mathrm{L}_v)$$

$$H^s(i) = \sum_{n=0}^{N} Q_{as}^s(\mathrm{i,n}) \times (Cp_{as} \times T_a(\mathrm{i}) + h_a(\mathrm{i}) \times \mathrm{L}_v)$$

$T_a(n)$, $T_a(i)$	Room (n, i) ambient temperature in °C
T_{ref}	Reference temperature (T_{ref}=0°C is considered)
Q_e as (n,i)	Mass flow transiting from room n to room i (infiltration, ventilation) $en\ Kg/s$
Q_s as(i,n)	Mass flow transiting from room i to room n (infiltration, ventilation) $en\ Kg/s$
Cp_{as}	Dry air Calorific capacity J/kg K
$h_a(n,i)$	Absolute humidity of air in the zone (n and ,i) in kg H_2O/Kg dry air
L_v	Latent heat of vaporization of wet air J/kg
Cp_{ve}	Steam heating capacity ventilation in J/kg K
$\left[\begin{array}{l}\sum_{j=i}^{NP(i)} Sj \times h_{cij} \\ (T_{Sij}(i)+T_a(i))\end{array}\right]$	Convective flows exchanged between the surface of the wall j of zone i and the air mass inside this zone
$NP(i)$	Number of walls j in the zone i
$T_{Sij}(i)$	Temperature of the inside surface of the wall j of the zone i en °C
Sj	Surface of wall j in m^2
h_{cij}	Exchange Coefficient by convection of the wall j of zone i in W/m^2 s
Ps, Pl	Respectively ,Sensitive and latent power provided by air conditioning installation
AI_s, AI_l	Sensitive and latent power internal,(Household appliances ,Occupants ,Lighting...)
SV	Power received by sunlight from the glass surfaces

b. Dry air mass balance

$$\sum_{n=0}^{N} Q_{as}^{e}(n,i) - \sum_{n=0}^{N} Q_{as}^{s}(i,n) = \frac{dm_{as}}{dt} \approx 0$$

We can consider that the accumulation of dry air in the area is negligible. Indeed, the ventilated air in the room is considered equal to the air extracted from the room (need for air renewal).This allows us to simplify the equation for conserving the air mass in zone i.

$$\sum_{n=0}^{N} Q_{as}^{e}(n,i) \approx \sum_{n=0}^{N} Q_{as}^{s}(i,n)$$

From the previous equations, we find:

Sensitive balance sheet:

$$\rho_{as} \times V_i \times Cp_{as} \frac{dT_a(i)}{dt} = \sum_{n=0}^{N} Q_{as}^{e}(n,i) \times \left[\mathrm{Cp}_{as} \times (T_a(\mathrm{n}) - T_a(i))\right]$$
$$+\left[\sum_{j=i}^{NP(i)} S_{ij} \times h_{cij} \times (T_{Sij}(i) - T_a(i))\right] + SV + AI_S + P_s$$

Latent balance sheet

$$\rho_{as} \times V_i \times \frac{dh_a(i)}{dt} = \sum_{n=0}^{N} Q_{as}^{e}(n,i) \times \left[(\mathrm{h}_a(\mathrm{n}) - h_a(i))\right]$$
$$+\frac{P_l}{L_v} + \frac{AI_l}{L_v}$$

These two basic equations (sensitive and latent balance sheet) will be simplified by the CARRIER method that we will present these assumptions in the following

$$\mathrm{P_s} = \sum_{n=0}^{N} Q_{as}^{e}(n,i) \times \left[\mathrm{Cp}_{as} \times (T_a(i))\right] + \left[\begin{array}{l}\sum_{j=i}^{NP(i)} S_{ij} \times h_{cij} \\ \times (T_{Sij}(i) - T_a(i))\end{array}\right]$$
$$+SV + AI_S$$

3.2 Sensitive balance sheet simplified by the CARRIER method

-Simplifying assumptions:

The assessment is carried out over a period when the building's sensitive thermal inputs are at their maximum, so we can consider that the regime is permanent [7].
The sensitive power of the refrigeration machine is equal to:

a. Thermal balance on the inside and outside of the wall

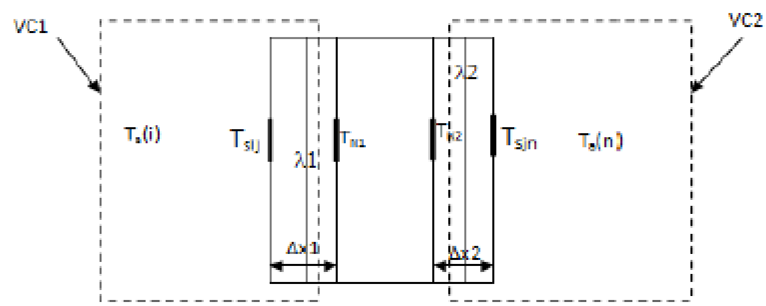

***Fig 3:** Control volumes of a wall j in the area i*

With :
T_N : Node temperature N of wall j
h_{cij}: convective exchange coefficient of wall j of zone i with exterior
V_C: control volume

The assessment is written:

$$\left\{\begin{array}{l}\rho_1 Cp_1 \frac{\Delta x1}{2} \times \frac{dT_{sij}}{dt} = \frac{\lambda_1}{\Delta x1} \times (T_{N1} - T_{sij}) + h_{conv1} \times (T_a(i) - T_{sij}) \\ +h_{rad1} \times (T_{rmi} - T_{sij}) + \phi_{CLO1} + \phi_{GLO1} \\ \rho_2 Cp_2 \frac{\Delta x2}{2} \times \frac{dT_{sij}}{dt} = \frac{\lambda_2}{\Delta x2} \times (T_{N2} - T_{sij}) + h_{conv2} \times (T_a(i) - T_{sij}) \\ +h_{rad2} \times (T_{rmn} - T_{snj}) + \phi_{CLO2} + \phi_{GLO2}\end{array}\right.$$

With :
T_{rm}: Average radiant temperature of the zone
$\emptyset_{CLO}$, $\emptyset_{GLO}$: respectively short wavelength radiation from solar radiation, long wavelength radiation from blackbody radiation.

-Simplifying assumptions of the carrier method for opaque walls

- The calculation is made at the time when the thermal loads of the building reach their maximum
- GLO radiative exchanges inside as well as outside are assumed to be negligible so: $\phi_{GLO2}\approx 0$, $\phi_{GLO1}\approx 0$
- The radiative exchange between the walls is assumed negligible, so:

$$h_{rad1}\times(T_{rmn}-T_{snj})\approx 0 \quad h_{rad2}\times(T_{rmn}-T_{snj})\approx 0$$

- The thermal inertia of the wall is neglected during the establishment of the balance [7]

To simplify we take $TN_2=TN_1=TN$ and the wall is made up of two layers of conductivity respectively λ_1 and λ_2

The balance sheet became

$$\left\{\begin{array}{l} \frac{\lambda 1}{\Delta x1}\times(T_N-T_{sij})+h_{conv1}\times(T_a(i)-T_{sij})=0 \\ \frac{\lambda 2}{\Delta x2}\times(T_N-T_{snj})+h_{conv2}\times(T_a(\mathrm{n})-T_{snj})+\phi_{CLO2}=0 \\ \frac{\lambda 2}{\Delta x2}\times(T_N-T_{snj})=\frac{\lambda 1}{\Delta x1}\times(T_N-T_{sij}) \end{array}\right\}$$

So the heat gain through the wall is expressed as follows:

$$\left\{\begin{array}{l} \phi_{lining}=h_{conv1}\times(T_{sij}-T_a(\mathrm{i})) \\ \phi_{lining}=h_{conv2}\times(T_a(\mathrm{n})-T_{snj})+\phi_{CLO2} \\ \phi_{lining}=\frac{\lambda 1}{\Delta x1}\times(T_N-T_{sij}) \\ \phi_{lining}=\frac{\lambda 2}{\Delta x2}\times(T_N-T_{snj}) \end{array}\right\}$$

Hence

$$\phi_{lining}\times\left(\frac{\Delta x1}{\lambda 1}+\frac{\Delta x2}{\lambda 2}+\frac{1}{h_{conv1}}+\frac{1}{h_{conv2}}\right)=\frac{\phi_{CLO2}}{h_{conv2}}\times(T_a(n)-T_a(\mathrm{i}))$$

We pose:

$$\frac{1}{K\times S}=\phi_{lining}\times\left(\frac{\Delta x1}{\lambda 1}+\frac{\Delta x2}{\lambda 2}+\frac{1}{h_{conv1}}+\frac{1}{h_{conv2}}\right)$$

$$\mathrm{T_{eq}}(\mathrm{n})=\frac{\phi_{CLO2}}{h_{conv2}}+\mathrm{T_a}(\mathrm{n})$$

From where

$$\phi_{lining}=K\times S\times\Delta te$$

- Δte: Equivalent difference in temperature in °C, which takes into account the temperature difference between indoor and outdoor air and the effect of sunlight on the outside surface.
- K : Overall transmission coefficient : Kcal/h.m^2.°C
- A: the surface area of the wall in question m^2
- ϕ_{wall}: heat flow through the wall in Kcal/h

The heat flow through the interior walls is due solely to the temperature difference existing on both sides.

In the CARRIER handbook, the values of Δte are tabulated; the tables have been calculated according to the Binder - Schmidt method, under the following conditions:

- Solar radiation intensity in July for 40° North latitude;
- 11°C change in outdoor dry temperature in 24 hours;
- Maximum outdoor temperature of 35°C and base indoor temperature 27°C, so 8°C difference;
- An absorption coefficient of 0.9 for walls and roof. This absorption coefficient is 0.5 for light-colored walls and 0.7 for medium-colored walls;
- The hours indicated are solar hours.

If the conditions considered are different from the above conditions, corrections must be applied to the value given by the tables; the new equivalent temperature difference can be determined by the following empirical relationship

$$\Delta t_e=\alpha+\Delta t_{es}+b\times\frac{R_S}{R_m}\times(\Delta t_{em}-\Delta t_{es})$$

In which:

Δt_e: Corrected equivalent difference

∝: Correction factor for the equivalent temperature difference, which takes into account:

- The difference of 8°C between the indoor and outdoor dry temperatures
- The variation of the outdoor dry temperature in 24 hours different of 11°C

Δt_{es}: Equivalent temperature difference at considered hour by the shaded wall

Δt_{em}: Equivalent temperature difference at considered hour by the sunny wall

b :Coefficient taking into account the color of the outer surface of the wall.

- External side of dark color b=1
- External side of medium color b=0.78
- External side of light-colored b=0.55

R_S = Maximum sunshine Kcal/h.m2 for the month and latitude considered through a glazed surface, either vertical for the orientation considered (wall) or horizontal (Roof)

R_m = Maximum sunshine Kcal/h.m2 in July at 40° latitude, through a glazed surface, either vertical for the orientation considered (wall) or horizontal (Roof)

b. Thermal balance on glazed surfaces

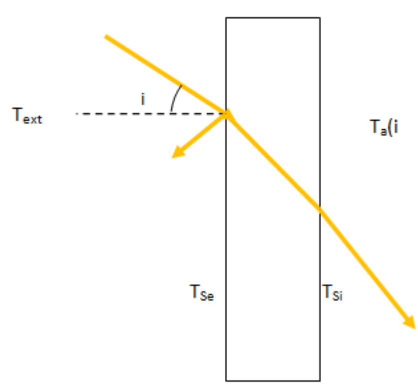

Fig4: *Passage of solar rays through a glass wall*

Gains per glass lining= gain per transmission+ sunshine gain

$$U \times S_v \times (T_{ext} - T_a(i)) + \phi_{CLO}(\text{direct ,diffused})$$

-U : Overall transmission coefficient of the glazing (W/m².°C)
-Sv : Window area (m²)
The gain by sunlight is given by the relationship below:
Sunshine gain = instant maximum gain ×Glazed area ×Coefficient "with or without screen
The maximum gains are extracted from the values tabulated in the Carrier manual, these tables have been established based on the following assumptions:

- Wooden frame (glass area = 85% of the opening in the wall).
- Clear atmosphere.
- Zero altitude.

If the assumptions are not verified, the following corrections should be applied to the values in the table:

- Metal frame: x 1.17 (glass area = 100% of the opening area).
- Lack of clarity in the atmosphere: - 15%.
- Altitude: add + 0.7% per 300 m height increment

For any type of glazing with or without a screen (protection), the term of the sunshine gains is multiplied by a corrective coefficient.

c. Air flow by infiltration and outside air

The enthalpy of the outside air introduced directly due to infiltration or through the air treatment unit (distributor) is different from that of the air in the area, and therefore contributes to the heat balance.

- Infiltration flow rate

The infiltration air flow rate varies according to the tightness of the doors and windows, the porosity of the building walls, its height, stairs, elevators, wind direction and speed, etc., but generally they are mainly due to wind speed or stack effect (density difference caused by temperature and humidity differences) or simultaneously with these two causes.
Generally, in comfort applications, thermal inputs due to air infiltration are assumed to be negligible.

-Renewal in outdoor air

The outside fresh air flow rate is extracted from the tables presented in the Carrier manual, it is the minimum recommended fresh air flow rate depending on the application. The tables give both the minimum flow rate per person and per m^2 of floor areaThe gain by air exchange is given by the following relationship:
Gain by external air ventilation=

$$\dot{m}_{air} \times Cp_{air} \times \rho_{air} \times (T_{ext} - T_a(i))$$

Gain per wall of non-air-conditioned rooms
It is assumed that this gain is mainly due to the temperature difference between the study area and the adjacent uncooled areas, it is calculated by the following relationship:

$$Q_{\text{lining}/LNC} = K \times S \times (T_a(i) - T_a(\text{no-air-conditioned rooms}))$$

d. Internal contributions

- The occupants

The values of internal gains due to occupants were determined based on the average amount of heat released by an adult male weighing 68 Kg, for different degrees of activity, and generally for a stay of more than 3 hours in the conditioned rooms. They have been corrected to take into account the quantities of heat released by a woman, which are about 85% of those released by a man. The relationship for calculating these earnings is as follows: Thermal gains due to occupants = Number of occupants x sensible/ latent contribution of an occupant

- Lighting

Lighting equipment is a sensible source of heat, this heat is released by convection and conduction radiation. The contributions due to lighting are obtained from the tables presented in the Carrier manual, which provide the specific lighting power according to its type.

- Electrical equipment

This gain depends on the power of the electrical machines present in the room, generally the electrical equipment of an office space are computers whose electrical power does not exceed a few Kwh, this means that this gain does not considerably influence the thermal balances in a comfort application.

e. Correction of results for thermal inertia, non-simultaneous gains and stratification

The CARRIER method [7], unlike conventional methods, allows the following considerations to be taken into account:
1. Thermal flywheel made up of construction materials (thermal inertia);
2. Simultaneous name of the maximum values of the various earnings;
3. Stratification phenomena in some cases.

- To take into account thermal inertia

The operating time of the installation affects the importance of the thermal flywheel, in fact if the installation is stopped after uninterrupted operation, part of the accumulated heat will remain in the materials and will be added to the heat balance when the system is started up again the next time. For example, if the installation is stopped after an interruption in the operation of the system, a part of the accumulated heat will be left in the materials and will be added to the heat balance during the next start-up. To determine the actual gains per exposure and lighting, the instant gains presented below are multiplied by a correction coefficient. Actual gain per sunlight = instant maximum gain ×Glazed area ×Coefficient "with or without screen "× Amortization coefficient at the hour in question.

- To take into account the non-simultaneous nature of earnings

It is quite rare for all heat loads in a room to occur at the same time, so simultaneity coefficients are generally applied to gains by occupants and lighting in large offices, as it is unlikely that all occupants will be present and all lighting in operation at the time of maximum gains.

Actual gain due to lighting = Instant maximum gain × Amortization coefficient × Simultaneity coefficient Actual gain due to occupants= Instant maximum gain × Simultaneity coefficient

- To take into account the stratification

In our case, the stratification of heat is due to the recessed lighting equipment above the false ceilings, in fact part of the heat released by convection by the recessed lighting equipment accumulates between the ceiling and the false ceiling, as well as the radiation from the various sources (sun, lighting, occupants), heats the false ceiling, and also tends to increase the temperature between ceiling and false ceiling and a heat flux is established through the bottom of the upper floor. To take this phenomenon into account, the gain due to lighting is multiplied by an Amortization coefficient.

3.3 Latent balance sheet simplified by the CARRIER method

Simplifying assumptions of the carrier method regarding the Latent global balance sheet. The assessment is carried out over a period of time when the building's latent heat input is at its maximum, so we can consider the regime to be permanent. Latent internal gains are mainly due to occupants either through transpiration or breathing, and through the ventilated air inside the room [5].

$$P_L = L_v \times (\sum_{n=0}^{N} Q^{e}_{air\,\sec}(n,i) \times \left[(ha(n) - ha(i)\right] + \frac{AI_l}{L_v}$$

4 MANUAL CALCULATION OF HEAT GAINS FOR ZONE 1

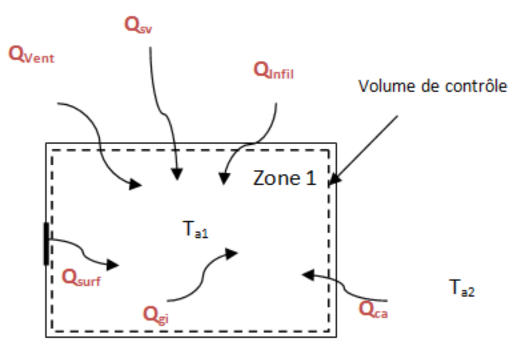

Fig 5: *Heat gains of the zone (1)*

Q_{Vent}: Convection heat flow from ventilation air renewal;
Q_{surf}: Convection heat flow from all interior surfaces;
Q_{gi} : Convective internal gains due to (Equipment, occupants, lighting.);
Q_{ca}: Convection heat flow due to air couplings with other areas;
$Q_{Infil:}$ Convection heat flow from air infiltration through leaks;
Q_{sv}: heat flow through the glass walls.

4.1 Sensitive contributions

a. Heat transfer through opaque walls

$$Q_{surf} = \left(K \times S \times \Delta t_e\right)_{south-east} + \left(K \times S \times \Delta t_e\right)_{north-east} + \left(K \times S \times \Delta t_e\right)_{Northwestern}$$

The conditions are as follows:

- Solar radiation intensity in July for 35° North latitude
- variation of 11°C on the outside dry temperature in 24 hours
- Outdoor temperature of 35°C and indoor base temperature of 26°C, so a difference of 9°C
- Light color of the walls b=0.55
- The solar hour of calculation 15h
- A specific wall weight equal to 365 kg /m^2

In our case, the difference between the indoor and outdoor dry temperatures is different from 8°C, so a correction must be applied to Δte pour to take this difference into account. We add to Δte a correction coefficient ∝ given in table in the carrier handbookBy interpolation we find ∝ =0.85. On the other hand, a second correction must be applied to take into account the deviation of the northern latitude and the sunshine of the walls. The table below groups the values extracted from carrier handbook table for calculating the heat input through the walls.

	South-East	Northwestern	Northeast
Δt_{es}	11.7	0	6.1
Δt_{em}	11.7	5.5	6.1
R_s	32	271	32
R_m	32	179	32
Δt_e	12.55	5.429	6.95

Table 5: *Calculation of the equivalent temperature difference of the walls of the zone (1)*

Q_{surf}=282.96W

b. Heat gain through glass surfaces

❖ Gain per transmission

=799.2W

$$U \times S_V \times \left(T_{ext} - T_{a1}\right) = \left(U \times S_V \times \left(T_{ext} - T_{a1}\right)\right)_{south-east} + \left(U \times S_V \times \left(T_{ext} - T_{a1}\right)\right)_{north-east} + \left(U \times S_V \times \left(T_{ext} - T_{a1}\right)\right)_{northwsetern}$$

❖ Real gains in sunlight

- The conditions are as follows
- Clear atmosphere.
- Metal frame
- Altitude 84 m
- Single glazing 6 mm thick

The corrections to be applied concern both the nature of the framework and the attitude,
We multiply the value of the maximum gain by 1.17 (metal frame) and add 0.00196 (84 m altitude)
It is also necessary to take into account the effect of thermal inertia, the formula for calculating solar gains must be multiplied by an amortization coefficient (carrier handbook table). The table below groups the values extracted from tables carrier handbook for calculating the heat input through the glass surfaces.

	South-east	Northwestern	Northeast
Maximal gain (kcal/m2h)	*271*	*355*	*355*
Area	*17*	*5*	*15*
Coefficient (with or without screen)	*0.94*	*0.94*	*0.94*
Amortization coefficient	*0.32*	*0.25*	*0.2*
Sunlight gain (W)	*1884.4*	*549.63*	*1361.3*

Table 6: *Calculation of sunlight gains for glazed surfaces in zone (1)*

$Q_{sv=3795.33W}$

❖ Gain per lining of non-air-conditioned rooms

The temperature of the rooms without air conditioning = 29°C
Total surface area of the walls of the non-air-conditioned rooms adjacent to the zone (1) = 10 m².

$Q_{lining/LNC}$=19.5W

❖ Gain by external air ventilation

The renewal airflow rate is set by regulation according to the number of people occupying the room. It is equal to 35 m3/h per person [7] The area is occupied by 26 people, resulting in an airflow rate of 910 m^3/h

Gains by external air ventilation =2617.65W

❖ Internal contributions due to occupants

The number of occupants in the area (1) = 26 persons
The significant contributions of an office employee = 54 Kcal/h [7]
So:

Sensitive thermal gains due to occupants =1630.2 W

❖ Gain by lighting [7].

The lighting power used in the area = 20 W/m^2.

- Specific weight of 180 kg/m^2 of floor
- Number of hours since switching on = 8h
- Type of lighting: recessed fluorescent
- Floor area = 148 m2
- System operating time 18 hours

So:

- Instant maximum gain (20x148)
- Amortization factor (0.97).
- Simultaneity coefficient (0.84).
- Actual gain due to lighting = 3014.76 W
- The 1.25 corresponds to the power absorbed by the ballast

Sensitive total loads = 12159.6 W

4.1 Latent contributions:

❖ Gain by external air ventilation

$$m^{\cdot}_{air} \times (r_{ext} - r_1) \times L_v = 3457,75W$$

❖ Gain due to occupants

The latent contributions of an office employee = 59 Kcal/h (see Table B.9, appendices)
Number of occupants = 26

Gain due to occupants = 1781,14 W

Total latent loads = 5238.89 W

4 Conclusion

In order not to increase the calculations, we have carried out the rest of the building's heat balance using a software program. The results obtained for the area on which we have prepared the manual balance sheet using the CARRIER method (Zone B2.1) differ slightly from the results provided by the software (see Appendices), this difference may be due to the simplifying assumptions we have adopted for the calculation, The air conditioning of our office building is provided by 4 water/air vapor compression heat pumps and an air/air heat pump, we will carry out in the following a study of the technical and economic feasibility of replacing the 4 water/air heat pumps whose total cooling capacity is equal to 1124 kW by an absorption refrigeration machine coupled to solar thermal collectors.

Appendices

TABLE 2. RESUME DES APPORTS

Composant des Charges	Détails	Calcul Apport Sensible (W)	Charges Latent (W)	Calcul Chauffage (W)
Apports Solaires	37 m²	3 601	-	-
Transmission/Mur	47 m²	291	-	513
Transmission par le toit	0 m²	0	-	0
Transmission du Vitrage	37 m²	792	-	1 733
Transm. par fen. de toit	0 m²	0	-	0
Parois/LNC	18 m²	18	-	72
Eclairage	20.00 W/m²	2 655	-	-
Autre Électrique	0.00 W/m²	0	-	-
Personnes	26 Personnes	1 630	1 563	-
Infiltration		0	0	0
Divers		0	0	-
Terre-plein	0 m²	-	-	0
Arrêt/Redémarrage		0	-	-
Coeff. de Sécurité	10/10/10 %	899	156	232
Charges Totales Zone		9 886	1 719	2 550
Charge Ventilation	252 L/s	2 719	3 175	4 716
Apport du ventil. de soufflage	689 L/s	1	-	-
Apports des murs vers le Plénum	0 %	0	-	-
Apports du toit vers le Plénum	0 %	0	-	-
Apports éclairage vers Plénum	0 %	0	-	-
Charge de Réchauffage		0	-	-
Charges Totales sur Batterie		12 605	4 893	7 266

TABLE 3. APPORTS ET DEPERDITIONS PAR LES MURS ET LES VITRAGES

	Composant	Total Surf. Nette (m²)	Clim. Transmission (W)	Clim. Apport Solaire (W)	Chauffage Transmission (W)
Murs:	NE	20	121	-	219
	E	0	0	-	0
	SE	21	148	-	229
	S	0	0	-	0
	SO	0	0	-	0
	W	0	0	-	0
	NO	6	22	-	66
	N	0	0	-	0
Vitrage :	NE	15	321	1 356	702
	E	0	0	0	0
	SE	17	364	1 731	796
	S	0	0	0	0
	SO	0	0	0	0
	W	0	0	0	0
	NO	5	107	513	234
	N	0	0	0	0
	Hor	0	0	0	0

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