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**RESEARCH ARTICLE**

# DESIGN AND PRODUCTION OF SOFTWARE FOR THE SIZING OF AIR CONDITIONERS IN WET TROPICAL BUILDINGS

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**ABSTRACT**

This work deals with thermal comfort and energy efficiency in the building. In Africa, more specifically in humid tropical regions like Benin, when the need for comfort is felt, several means are used. Among the alternatives put in place, it is recorded that air conditioning facilities are a high priority to overcome the problems of discomfort. But, it is clear that for the design of these air conditioning systems, the most used method is that of surfaces. A method that does not take into account all the parameters necessary to provide ideal air conditioners and with a good coefficient of energy performance. And this is due to the fact that there is no digital dimensioning tool that takes into account all the necessary parameters while respecting the requirements of the standards in force. In this context, the sizing method adopted is the Carrier method because it takes into account all the parameters that are taken into account in the sizing of air conditioning systems and the programming language used is W Language. Called M@doc, the tool obtained also makes it possible to size the power of the air conditioner necessary for the air conditioning and meets the environmental and energy performance standards.



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## INTRODUCTION

A house is built to ensure the safety and well-being of the occupants. The second need requires offering a better comfort to the occupants and is characterized by several major parameters. The architectural design, the orientation is the first factors to take into account to respond favorably to the need for comfort. But it is clear that for subjective reasons, these important parameters are no longer put during construction, so that occupants are forced to use adaptive approaches to ensure their well-being, especially during periods with strong sunshine. Among these remedies, we see that air conditioning is the most requested method. These devices are a major source of energy consumption *(RT 2005).*And for this reason, it is necessary to put in place alternatives to strongly limit consumption at the level of air-conditioned buildings *(RT 2005).* In countries with a humid tropical climate such as Benin, it is found that during critical periods (athigh temperatures), it is more difficult to find one's state of well-being because of the increased heat.Thus, to satisfy the need for occupant comfort, several alternatives have been developed. Among these, we can note the active air conditioning. But the big thing is that this approach is really expensive and requires a lot of investment. How then to benefit from these air conditioning facilities without spending too much and enjoy ideal comfort during these periods of high temperature? This is the problem that arises. A study conducted on air conditioners to overcome this problem reveals that when air conditioners are undersized, there is a strong demand of the power of these machines to provide comfort, which increases energy consumption despite the comfort, is not reached. In the same way, when these air conditioners are oversized, they work more than they should and consume a lot.It would therefore be interesting to propose a simple and dynamic approach of the thermal balance in order to reduce the corresponding energy consumption and consequently to answer the problematic.

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**Method used and its calculation procedure**

**Calculation method:** The calculation method used is Carrier or Airwell. This is one of the most used methods for sizing air conditioning systems in tropical countries. It was designed to reduce the over-dimensioning of air conditioning systems (leading to increased energy consumption), especially in tropical regions. It takes into account:

- The geographical orientation of the premises;
- The actual position of glazed or opaque walls, sunny or not, depending on the position of the sun at the time of calculation;
- Heat gains through the outer walls (walls, roof, ceiling and floor);
- The heat due to the occupants and the renewal of the indoor air of the premises;
- Heat due to lighting and various appliances.

**Calculation procedure**

**Determination of heat input**

**External charges**

- Transmission of heat through the outer walls (walls, roof, ceiling and floor) and glazing

$$Q_{str} = \mathrm{k} \times \mathrm{S} \times \Delta\theta \qquad (1)$$

$k$= thermal transmittance of the wall or glazing considered in W / m² ° C evaluated by:

$$k = \frac{1}{\frac{1}{h_e}+\sum\frac{e}{\lambda}+\frac{1}{h_i}} \qquad (2)$$

where $h_e$ and $h_i$ are the global convection coefficients on the walls and λ the thermal conductivity coefficient of the wall considered (Table 3). The values of and are given in the following table.

**Table 1. Superficial thermal exchange coefficient in tropical zone (IEPF, Volume 1, 8p)**

| | Walls in contact with the outside | | | Walls in contact with another room, a roof or a crawl space | | | Glass walls |
|---|---|---|---|---|---|---|---|
| | walls | ceilings | floors | walls | ceilings | floors | |
| $h_e$ | 16.7 | 20 | 20 | 9 | 20 | 5.9 | 16.7 |
| $h_i$ | 9 | 11.1 | 5.9 | 9 | 20 | 5.9 | 9 |

-S = surface of the wall or window considered (total area of the bay corresponding to the reservation in the wall) (m²)
Δθ = temperature difference between the two faces of the considered wall [° C] (Table 2)

**Table 2. Temperature difference between the different faces of the walls (IEPF, Volume 1.15p)**

| Wall type | $\Delta\theta$ |
|---|---|
| Sunny exterior walls | $\Delta\theta = \theta_e - \theta_i$ |
| Walls in contact with unconditioned premises | $\Delta\theta = \theta_e - \theta_i - 3°C$ |
| Ceiling under ventilated roof | $\Delta\theta = \theta_e - \theta_i + 3°C$ |
| Ceiling attic not ventilated | $\Delta\theta = \theta_e - \theta_i + 12°C$ |
| Floor on solid ground | $\Delta\theta = +20°C - \theta_i$ |
| Wall in contact with the kitchen | $\Delta\theta = \theta_e - \theta_i + 18°C$ |

**Solar heat supply through the walls**

The amount of heat passing through the wall is evaluated by:

$$\mathrm{Q_{SRm}} = \alpha \times \mathrm{F} \times \mathrm{S} \times \mathrm{R_m} \qquad (3)$$

-α = absorption coefficient of the wall receiving the radiation depending on the color and nature of the wall;
-S = net area of the wall in m² (area without doors and windows or other openings);
-F = solar radiation factor ;
-$R_m$ = solar radiation absorbed on the surface of the wall in W / m²;

**Table 3. Thermal conductivities of materials (IEPF, Volume 1, 7p)**

| Materials | Thermal conductivity (W / m.K) |
|---|---|
| Dry ash | 0.29 |
| Charcoal | 0.041 |
| Cotton | 0.06 |
| Leather | 0.174 |
| Tree bark | 0.066 |
| Wood wool (panel) | 0.09 |
| Sheep wool | 0,038 |
| Rockwool | 0,052 |
| Compressed straw | 0.12 |
| Paper | 0.14 |
| Feather | 0,037 |
| Reed | 0.05 |
| Sawdust | 0.06 |
| Natural silk | 0,052 |
| Asbestos cement | 0.4 |
| Natural pozzolana concrete | 0.25 |
| Geo concrete | 0.7 |
| Reinforced concrete | 1.5 |
| Bitumen | 0.16 |
| Plywood | 0.14 |
| Lime or smooth plaster | 0.87 |
| Cement plaster | 0.87 |
| Wood chips | 0.081 |
| Concrete | 0.9 |
| limestone | 1.05 |
| Terracotta | 1.15 |
| Solid brick wall | 0.85 |
| Hollow brick wall | 0.4 |
| Full block | 1.1 |
| Hollow block | 0.67 |
| Mortar plaster | 1.15 |
| Plaster | 0.45 |
| Natural wood | 0.044 |
| Expanded polystyrene | 0,036 |
| Glass wool | 0.04 |
| Floor tile | 1.15 |
| chippings | 1.5 |
| Heavy stone | 3.5 |
| Bitumen sheet | 0.23 |
| Earth pressed | 1.15 |
| sheet metal | 70 |
| Window | 1 |
| Aluminum | 210 |
| Iron | 80 |
| Melting | 55 |
| Steel | 50.2 |
| PVC | 0.2 |
| Terracotta tile | 0.32 |
| Copper | 370 |
| Sand concrete | 0.272 |

The absorption coefficient "«$\alpha$»" depends on the color and the nature of the wall (Table 4)
Radiation factor "«$F$»" indicates the amount of heat absorbed by the surface and transmitted through
The value of solar radiation "«$R_m$»" on a wall (Table 6 column m) depends on:

- From the latitude under which the local is,
- From the orientation of the wall,
- The time for which the calculation will be done.

For intermediate coefficients, interpolation *must be done*

**Table 4. Absorption coefficient "α" for walls, roofs and windows (IEPF, Volume 1, 15p)**

| Colors and nature of the surface | | $\alpha$ |
|---|---|---|
| Very clear surface | White stone - White surface, clear or cream very clear cement | 0.4 |
| Dark surfaces | Fibrocement - unpainted wood - brown paint - red brick - red cement - red, green or red staff | 0.7 |
| Very dark surfaces | Dark slate roofs - very dark bituminous cardboard | 0.9 |
| Glasses (windows or skylights) | Single glazing | 1 |
| | Double glazing | 0.9 |
| | Triple glazing | 0.8 |

**Table 5. Solar Radiation Factor (IEPF, Volume 1, 15p)**

| ***k*** heat transfer coefficient of the considered wall (W /m².K) | ***F*** solar radiation coefficient |
|---|---|
| 0 | 0 |
| 1 | 0.05 |
| 2 | 0.1 |
| 3 | 0.15 |
| 4 | 0.20 |

**Table 6. Intensity of solar radiation on walls (m) and glazing (v). Latitude 4 ° N (IEPF, Volume 1, 16p)**

| Hours | N | | S | | E | | O | | N-E | | N-O | | S-E | | S-O | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | m | v | m | v | m | v | m | v | m | v | m | v | m | v | m | v |
| 7 | 62 | 53 | 64 | 55 | 71 | 61 | 62 | 53 | 70 | 60 | 62 | 53 | 66 | 57 | 62 | 53 |
| 8 | 131 | 113 | 146 | 126 | 176 | 152 | 131 | 113 | 173 | 149 | 131 | 113 | 153 | 131 | 131 | 113 |
| 9 | 186 | 160 | 212 | 182 | 249 | 214 | 186 | 160 | 249 | 214 | 186 | 160 | 211 | 182 | 186 | 160 |
| 10 | 223 | 191 | 269 | 232 | 298 | 257 | 223 | 191 | 309 | 266 | 223 | 191 | 243 | 209 | 223 | 191 |
| 11 | 258 | 222 | 330 | 284 | 317 | 273 | 258 | 222 | 351 | 302 | 268 | 230 | 258 | 222 | 258 | 222 |
| 12 | 272 | 234 | 359 | 309 | 272 | 234 | 272 | 234 | 333 | 287 | 333 | 287 | 272 | 234 | 272 | 234 |
| 13 | 256 | 220 | 352 | 303 | 256 | 220 | 335 | 288 | 268 | 231 | 380 | 326 | 256 | 220 | 256 | 220 |
| 14 | 216 | 186 | 290 | 249 | 216 | 186 | 335 | 288 | 216 | 186 | 352 | 303 | 216 | 186 | 249 | 214 |
| 15 | 166 | 143 | 207 | 178 | 166 | 143 | 264 | 227 | 166 | 143 | 265 | 228 | 166 | 143 | 206 | 177 |
| 16 | 150 | 129 | 134 | 115 | 150 | 129 | 100 | 86 | 150 | 129 | 103 | 89 | 150 | 129 | 126 | 109 |
| 17 | 31 | 27 | 32 | 27 | 31 | 27 | 33 | 29 | 31 | 27 | 33 | 28 | 31 | 27 | 32 | 28 |

**Solar heat input on glazing**

The amount of heat passing through the glazing is evaluated by:

$$Q_{SRv} = \alpha \times g \times S \times R_v \quad (4)$$

- $\alpha$ = Glazing absorption coefficient (Table 4)
- $g$ = Reduction factor (Table 7) depends on how the window is protected against solar radiation
- $S$ = Glazed surface (m²)
- $R_v$ = Intensity of solar radiation on glazing (in W / m²) ; it is defined in the same way as $R_m$ and is given by the same table (6) in the column «$v$»

**Table 7. Reduction factor " " for protected windows (IEPF, Volume 1, 16p)**

| Protected windows | Colors | $g$ |
|---|---|---|
| Canvas exterior blinds | Raw | 0.28 |
| Canvas exterior blinds | Aluminum | 0.22 |
| Interior blind completely lowered | Aluminum | 0.45 |
| Store blinds half lowered | White or cream | 0.63 |
| Blinds fully fucked inside the windows | Aluminum | 0.58 |
| Blinds fully fucked outside windows | Aluminum | 0.22 |
| $\boldsymbol{Q_{Lr} = q_v \times (\omega_e - \omega_i) \times 0,84}$ (W) | - | 1.00 |
| $\boldsymbol{Q_{Lr} = q_v \times (\omega_e - \omega_i) \times 0,84}$ (W) | | |
| Unprotected | | |

**Heat supply by air renewal and infiltration**

Air exchange in an air-conditioned room is necessary for hygienic problems. It is usually done by natural ventilation of the premises as well as by infiltration, introducing outside air into the air-conditioned room. It is a source of sensible heat and latent in the room to be conditioned.

The significant gains by air renewal are evaluated by:

$$Q_{Sr} = q_v \times (\theta_e - \theta_i) \times 0,33 (\mathrm{W}) \quad (5)$$

Latent gains by air change are evaluated by:

$$Q_{Lr} = q_v \times (\omega_e - \omega_i) \times 0,84 \, (\mathrm{W}) \quad (6)$$

-$q_v$ = Renewal outside air flow ($m^3$ /h)
The ventilation being natural, we can consider that the air exchange is equal to a volume of the room per hour (1vol / h),
-$\theta_e$ = basic outdoor temperature (°C)
-$\theta_i$ = basic indoor temperature (°C)
-$\omega_e$ = Moisture content of outdoor air g / kg dry air (defined from the psychometric diagram)
-$\omega_i$ = Moisture content of the indoor air g / kg dry air (defined from the psychrometric diagram)

The basic outdoor temperature is the temperature obtained during the hottest month of the year in the city in which the room is located. The basic indoor temperature is the one recommended for standard applications for thermal comfort in air-conditioned buildings. According to the CSA ZA 12-17 standard for office ergonomics, it is 24 ° C for an office. The determination of the water content is done by the use of the psychometric diagram. This diagram uses the temperature and relative humidity of the medium to express its water content. We have therefore downloaded a dynamic model of this software available on the internet. This model is the diagram of the humid air created by Frederic Benet. The downloaded version is version 3.0. However, since the developer's options are different from those of M@doc, we could not integrate the digital model of the diagram found on the internet. To overcome this fact, we opted for the determination of the formula at the origin of the diagram. This formula is given by the following relation:

$$H_s = 0{,}6217 \frac{H_r p_{sat}(t)}{p_{atm} - 0.3783 H_r p_{sat}} \; (K_{gas}/K_g)$$

-$H_s$ is the specific humidity or the water content;
-$H_r$ is the relative humidity measured in the field by a hygrometer generally;
-$p_{sat}$ is the saturation vapor pressure which is obtained by interpolation of the Dupré table data;
-$P_{atm}$ = 101325 mbar

**Table 8. Saturation vapor pressure (Dupré, 2019)**

| Temperature in (° C) | Saturation vapor pressure (mbar) |
|---|---|
| 15 | 17.0 |
| 20 | 23.4 |
| 25 | 31.7 |
| 30 | 42.4 |
| 40 | 73.8 |
| 50 | 123 |

**Verification of the correctness of the formula used:** It is important to check if this formula can be used instead of the numerical model of the psychometric diagram. For this, we proceeded to the comparative study of the results obtained with the numerical model of determination of the water content and those obtained with the formula found.

**Table 9. Comparison of relative humidity**

| Temperature (° C) | Relative humidity (%) | Water contents obtained with the psychometric diagram (Kgas / Kg) | Water contents obtained with the formula found (Kgas / Kg) | Difference in grades obtained (Kgas / Kg) |
|---|---|---|---|---|
| 24 | 70 | 0.0131 | 0.0130 | 0.0001 |
| 25 | 73 | 0.0145 | 0.0143 | 0.0002 |
| 26 | 76 | 0.0161 | 0.0159 | 0.0003 |
| 29 | 79 | 0.0200 | 0.0197 | 0.0003 |
| 30 | 83 | 0.0224 | 0.0219 | 0.0005 |
| 33 | 85 | 0.0274 | 0.0275 | 0.0001 |
| 34 | 75 | 0.0255 | 0.0257 | 0.0002 |
| 36 | 70 | 0.0266 | 0.0267 | 0.0001 |

It appears from this comparison that for all temperature ranges, the moisture content obtained with the psychometric diagram is substantially equal to that obtained from in the formula. This formula can therefore be integrated in place of the numerical model of the determination of the content using the psychometric diagram.

**Internal charges**

**Heat supply by the occupants:** It is given according to the internal temperature and the degree of activities. There are two kinds of gains generated by the occupants:

Occupant Sensitive Gains:

$$Q_{soc} = n \times C_{soc} (\mathrm{W}) \tag{8}$$

$Q_{soc} = n \times C_{soc}$ (W) (9)

Occupant latent gains:

$Q_{Loc} = n \times C_{Loc}$ (W)

-$n$ = number of occupants
-$C_{soc}$ = sensible heat of the occupants (W); (Table 10)
-$C_{Loc}$ = latent heat of the occupants (W); (Table 10)

**The values of the table are valid for an adult man. The values in this table should be reduced by the following coefficients:**

- For women: -20%
- For children: -20 to -40%
- For a mixed audience: -10%

**Table 10. Heat released by people (W) (IEPF, Volume 1, 19p)**

| Activity | Application | Ambient temperature | | | | | |
|---|---|---|---|---|---|---|---|
| | | 25°C | | 26°C | | 27°C | |
| | | Sensitive heat (W) | Latent heat (W) | Sensitive heat (W) | Latent heat (W) | Sensitive heat (W) | Latent heat (W) |
| Sitting at rest | School, theater | 65 | 37 | 62 | 40 | 60 | 42 |
| Light work | Office, hotel, apartment | 67 | 49 | 63 | 59 | 56 | 60 |
| Standing, slow walking | Store, shop | 68 | 63 | 63 | 68 | 57 | 74 |
| Meal | Restaurant | 77 | 84 | 71 | 90 | 64 | 97 |
| Easy work | Workshop | 80 | 140 | 72 | 148 | 67 | 153 |
| Dance | Nightclub | 88 | 161 | 80 | 169 | 75 | 174 |
| Hard work | Factory | 149 | 277 | 142 | 284 | 136 | 290 |

**Heat supply by lighting:** It is a sensitive heat source and depends on the type of lamp (Table 10):

For fluorescent lamps, we have:

$Q_{secl} = 1{,}25\ P$ (W) (10)

$Q_{secl} = P$ (W) (11)

$P$ = lamp power (W)

In the case of the fluorescent lamp, the additional 25% represents the heat released by the electromagnetic ballast.

**Heat supply by machines and equipment:** Most devices are both a source of sensible and latent heat. The table gives the heat input by machinery and equipment $(Q_{séquip})$. The values of these tables have been determined according to the indications of various manufacturers. The inputs of these machines and equipment (by a weighting coefficient) must be reduced according to their operating times. For example, it is considered that a device operating only half an hour per hour releases half of its nominal electrical power in heat input.

**Table 11: Heat released by lighting (IEPF, Volume 1, 20p)**

| Destination of the local or type of activity | Connected power (W / m²) | |
|---|---|---|
| | Incandescent lamp | Fluorescent lamp |
| Warehouse, habitat, restaurant, theater | 25 | 8 |
| Office, classroom, entrance hall with cash desk and counter | 65 | 16 |
| Reading room, computer, laboratory, store, exhibition hall | 110 | 24 |
| Supermarket, very large office, amphitheater without window, precision work | 125 | 45 |

**Total thermal loads**

The total heat balance $(Q_T)$ is the sum of all external and internal loads. It is more practical to sum the sensible $(Q_S)$ and latent loads$(Q_L)$. Where:

$Q_T = Q_s + Q_L$ (12)

**Table 12. Electrical and gas equipment (IEPF, Volume 1, 20p)**

| Device types | Nominal powers | Earnings to admit (W) | |
|---|---|---|---|
| | | Sensitive heat | Latent heat |
| Fryer 5litres of oil | 2575 | 464 | 696 |
| Fryer 10l of oil | 6954 | 1102 | 1653 |
| Wafer molds | 435 | 319 | 29 |
| Heated bread | 2192/719 | 899/319 | 609/203 |
| Percolator 2l | 993 | 394 | 104 |
| Water heater | 146 | 116 | 29 |
| Electric kitchen and washing machine | 3000 | 1450 | 1550 |
| Vacuum | 200 | 50 | 0 |
| wringer | 100 | 15 | 0 |
| Freezer 200 l | 175 | 500 | 0 |
| Irons | 500 | 230 | 270 |
| Stereophonic channel | 40 | 40 | 0 |
| TV | 175 | 175 | 0 |
| Hair dryer | 500/1000 | 175/350 | 75/250 |
| Cooking plate | 500/1000 | 120/250 | 130/250 |
| Meat grill | 3000 | 1200 | 300 |
| Sterilizer | 150 | 175 | 325 |
| Computer | 400 | 250 | 0 |
| Coffee maker | 500/3000 | 750 | 300 |
| photocopier | 0 | 750 | 0 |
| Inkjet printer | 0 | 52 | 0 |
| Laser printer | 0 | 15 | 0 |
| Fax | 0 | 62 | 0 |

**Total sensitive loads:** These are the contributions of sensible heat in the room, due to the temperature difference between the inside and the outside; we have:

$$Q_s = Q_{str} + Q_{sRm} + Q_{SRv} + Q_{Sr} + Q_{soc} + Q_{sécl} + Q_{séquip} \quad (13)$$

**Total latent loads**

$$Q_L = Q_{Lr} + Q_{Loc} + Q_{Léquip} \quad (14)$$

The quantification of the sources of inputs in the room allows us to evaluate the power of the air conditioner to install in order to bring comfort to the interior of the room.
Power of air conditioner and dehumidification

**Power of the air conditioner:** The cooling capacity of the air conditioner represents the total thermal loads ($Q_T$) that must be combated, with:

$$Q_T = Q_s + Q_L (W) \quad (15)$$

The power of the compressor is generally deduced from the manufacturer's catalogs. In the absence of any catalog, we can in first approximation use the notion of the coefficient of refrigeration efficiency or refrigeration performance ($COP_{cold}$) whose formula is written:

$$COP_{cold} = P_f/P_a \quad (16)$$

- $P_f$ = cooling capacity (W)
- $P_a$ = absorbed power (W)

Hence the power absorbed by the compressor is:

$$P_a = P_f/COP_{cold} \quad (17)$$

In Benin, the performance coefficients of air conditioners are defined by decree N ° 2018-563 of December 19, 2018 setting "the minimum energy performance standards and energy labeling system of lamps and room air conditioners in the Republic of Benin ". The present dimensioning takes into account this modification and thus makes it possible to make a judicious choice of the refrigerating system to propose.

| Type de climatiseur | Puissance frigorifique (PF) exprimée en Watt | Ratio d'efficacité énergétique (EER) | Classe d'efficacité énergétique du climatiseur |
|---|---|---|---|
| Fenêtre | PF<= 14 000 | 2,90 | Une (1) étoile |
| | | 3,10 | Deux (2) étoiles |
| | | 3,30 | Trois (3) étoiles |
| Split | PF<= 4 500 | 3,20 | Une (1) étoile |
| | | 3,40 | Deux (2) étoiles |
| | | 3,60 | Trois (3) étoiles |
| | 4 500 <PF<= 7 100 | 3,10 | Une (1) étoile |
| | | 3,30 | Deux (2) étoiles |
| | | 3,50 | Trois (3) étoiles |
| | 7 100 <PF<= 14 000 | 3,00 | Une (1) étoile |
| | | 3,20 | Deux (2) étoiles |
| | | 3,40 | Trois (3) étoiles |

L'EER des climatiseurs est déterminé en appliquant les procédures d'essai requises par la norme ISO 5151.

**Figure 1. Minimum performance requirements for room air conditioners, Decree No. 2018-563 of 19 December 2018, Benin**

**Dehumidification power:** The dehumidification power represents the total latent heat that must be combated, with:

$$Q_L = Q_{Lr} + Q_{Loc} + Q_{Léquip} \quad (18)$$

In climate-controlled buildings in humid tropical countries like Benin, the indoor relative humidity is very high and varies between 60 to 70%, with measured temperatures ranging from 20 to 25 ° C. Despite these acceptable temperatures, some users have a feeling of thermal discomfort following the high relative humidity of the air. These readings make it possible to understand that air conditioners do not perform their functions of dehumidification of treated air but of simple cooling. This situation is due to a poor selection of air conditioners in humid tropical regions that differs from those used in dry tropical climate.

In humid tropical regions, the need for air-conditioning is limited to cooling and dehumidifying the air contained in a room. For this reason, when selecting the air conditioner, there are two important parameters to consider:

- The power of the air conditioner or total heating load in watts.
- The dehumidification power or total latent heat expressed in Watt or kg of water/hour (this is the amount of water that can be retained by the air conditioner to ensure the conditions of thermal comfort in the room)

## RESULTS, ANALYZES AND DISCUSSION

The work carried out has made it possible to set up a digital tool called M@doc able to size the power of the air conditioners taking into account the energy performance requirements in force in the humid tropical regions. Some software interfaces are as follows:

The window below of the software makes it possible to evaluate the heat transfer coefficients of the walls of the room.

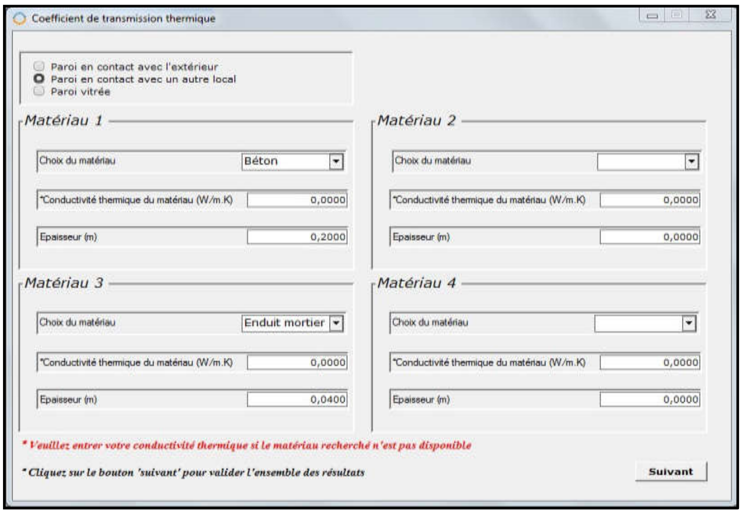


**Figure 2. Calculation of the thermal transmittance**

This window makes it possible to evaluate the contributions of heat released by the occupants.

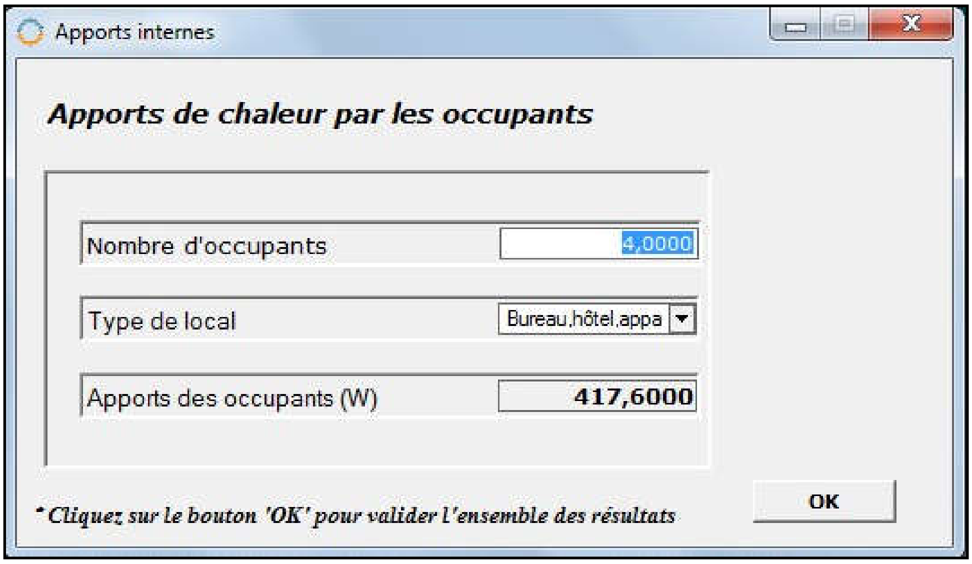


**Figure 3. Calculation of heat inputs by occupants**

With the window below, the heat inputs by air renewal of the room are evaluated.

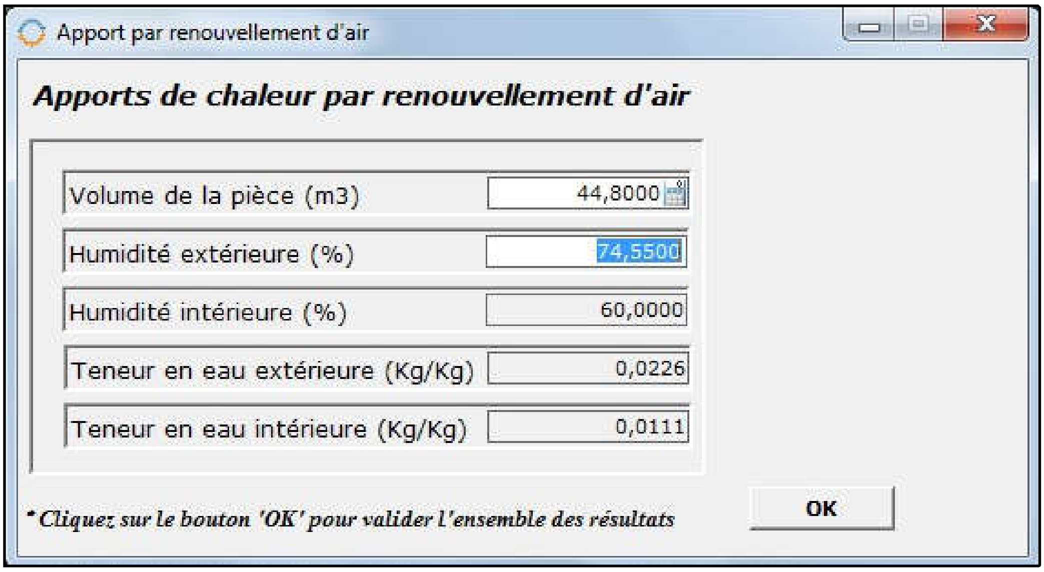


**Figure 4. Calculation heat input by air renewal**

The following window "Calculation input by transmission on the walls" makes it possible to evaluate the contributions of heat by transmission on the various walls of the room.

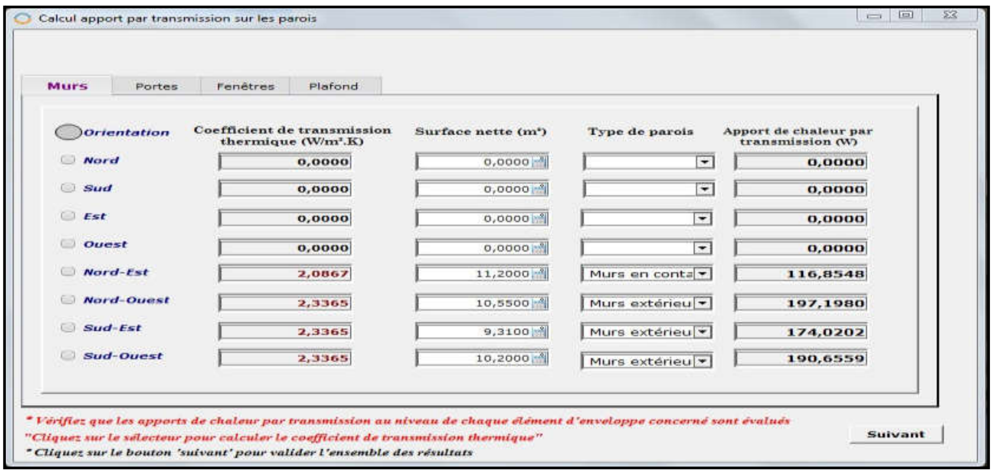


**Figure 5: Calculation of the contributions by transmission**

Using the following window, heat inputs due to solar radiation are evaluated.

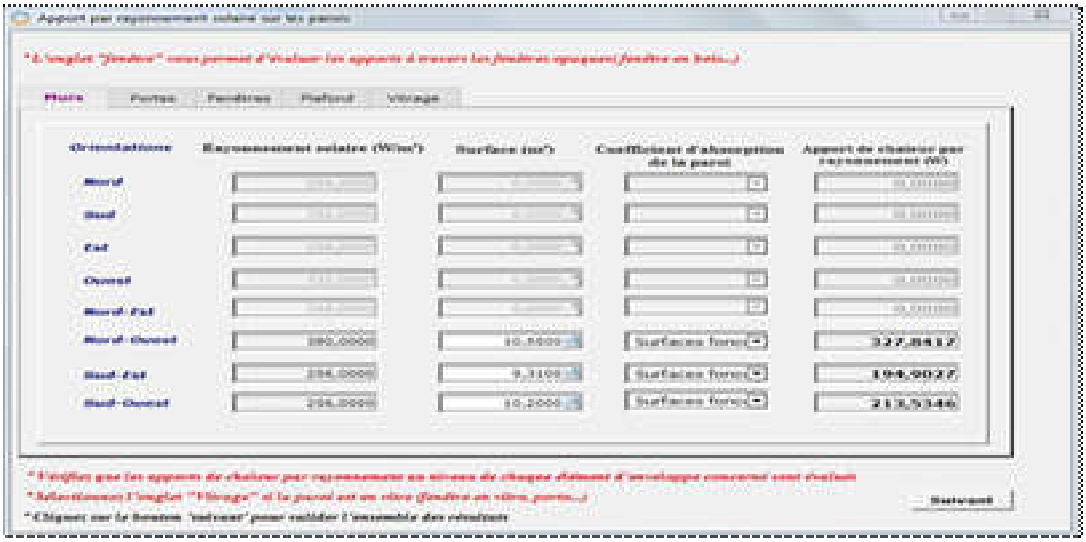

**Figure 6: Calculation contribution by solar radiation on the walls**

This window makes it possible to evaluate the contributions of heat due to the lighting in the room.

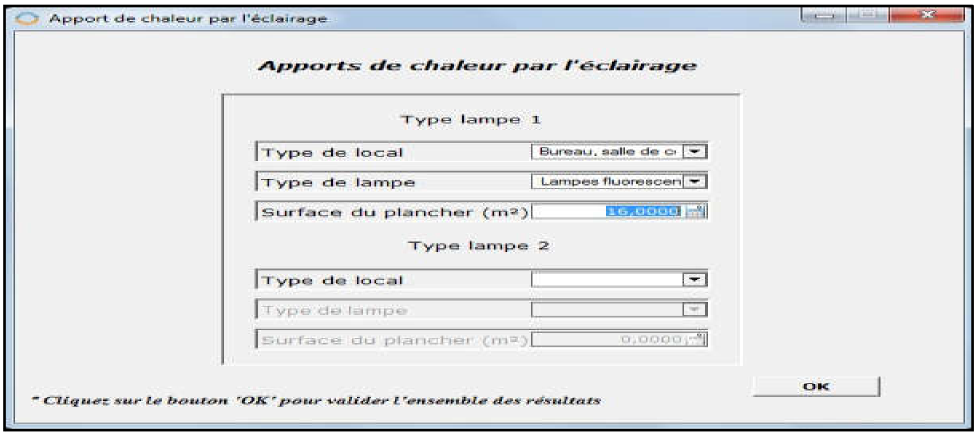


**Figure 7. Calculation heat input by lighting**

The following window is used to calculate the heat input due to equipment in the room.

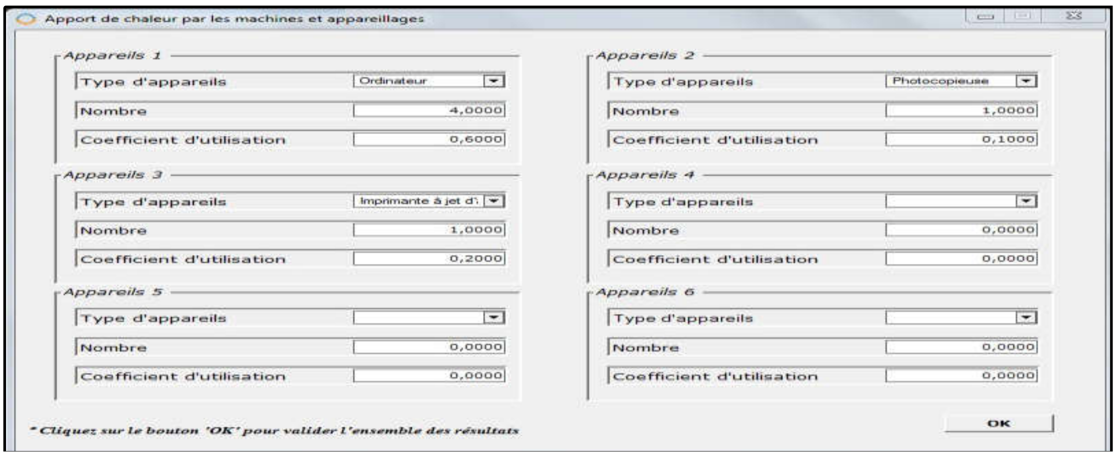


**Figure 8: Calculation heat input due to equipment**

All the results are conveyed on the «Dimensioning " window which allows having all the information concerning the characteristics of the air conditioner to be installed.


Figure 9: Dimensioning, 2019

The "Localization" window makes it possible to map the local studied.

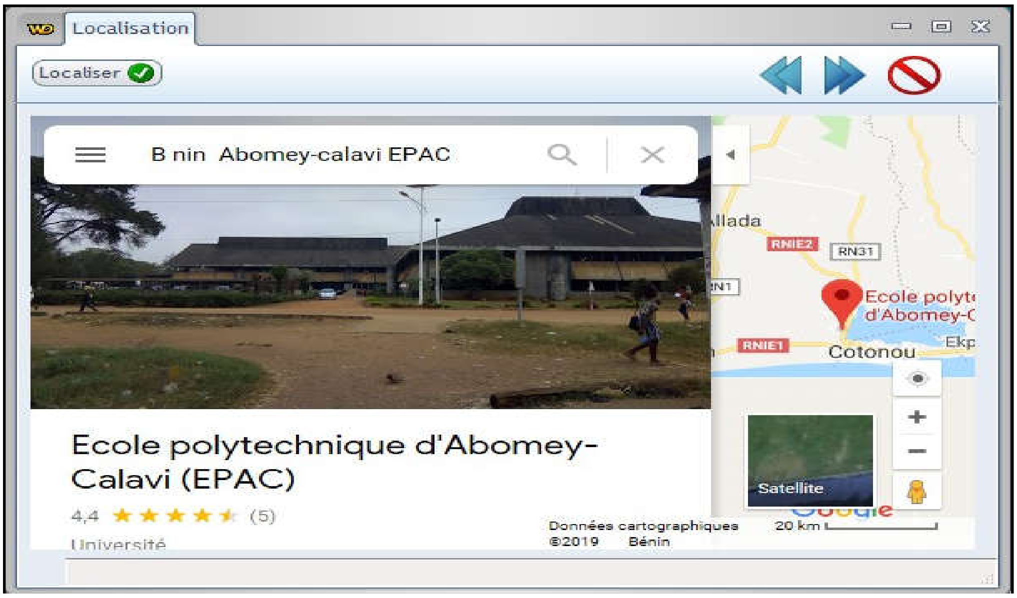


Figure 10. Location, 2019

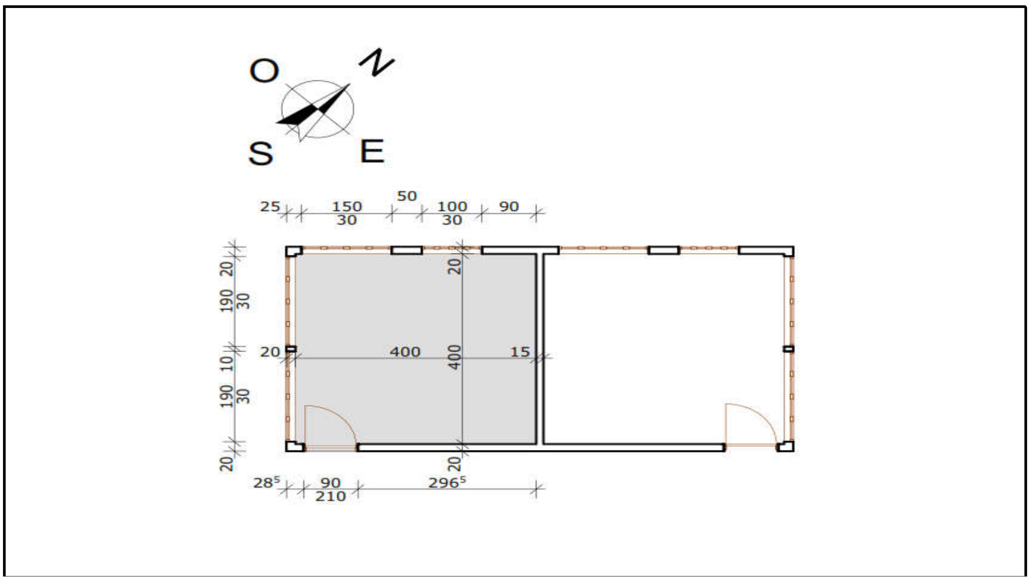


Figure 11. Plan of the studied room, 2019

Using this tool, a case study was conducted at the EEH building in Abomey-Calavi, UAC-EPAC. The piece studied is for office use. The other data found are as follows:

- Floor area: 16 m²
- 01 wooden door on the south-east side
- 02 Windows on the side SO
- The walls are supposed to be in 20cm thick concrete with interior and exterior coating
- The ceiling is in contact with another room
- 04 occupants occupy the room
- 04 Computers are frequently turned on
- 01 Copier
- 01 inkjet printer
- The room is air-conditioned: - Pf = 3500 W - Pa = 1500 W - COP: 2,3

The results obtained with the software are recorded in the following table:

**Table 13: Summary of results obtained (ABALLO Michaël & WEKE Cyrille, 2019)**

| | | | |
|---|---|---|---|
| THERMAL BALANCE | | | |
| Local : Office Level : Ground floor | | | |
| EXTERNAL CONTRIBUTIONS | | | |
| CALCULATION OF CONTRIBUTIONS BY TRANSMISSION | | | |
| | Wall orientations | Surface (m²) | Contribution (W) |
| | North East partition | 11.2 | 116.85 |
| | North West wall | 10.55 | 197.20 |
| | South East wall | 9.31 | 174.02 |
| | Southwest wall | 10,20 | 190.66 |
| | South-West glazing | 1.00 | 45,46 |
| | North-West glazing | 0.65 | 29.55 |
| | South East Gate | 1.89 | 12.67 |
| | Ceiling | 16,00 | 298.38 |
| | Total Contributions by Transmission 1064.79 W | | |
| CALCULATION OF THE CONTRIBUTIONS BY SOLAR RADIATION | | | |
| | Wall orientations | Surface (m²) | Contribution (W) |
| | North East wall | 11.2 | 0 |
| | North West wall | 10.55 | 327.84 |
| | South East wall | 9.31 | 194.90 |
| | Southwest wall | 10,20 | 213.53 |
| | South-West glazing | 1.00 | 220.00 |
| | North-West glazing | 0.65 | 211,90 |
| | Door | 1.89 | 14.19 |
| | Ceiling | 16,00 | 0 |
| | Total contributions by solar radiation 1182.37 W | | |
| | Infiltration gains 548.74 W | | |
| | Total external contributions 2799.09 W | | |
| INTERNAL CONTRIBUTIONS | | | |
| | Nature | Number | Contribution (W) |
| | occupants | 04 | 417.60 |
| | Lighting | - | 320.00 |
| | Machines and appliances | 06 | 685.40 |
| | Total internal inflows 1423.00 W | | |
| Overall result | Cooling capacity 4219 W<br>Dehumidification power 607 W<br>Electrical power with a COP of 3.2 1318 W<br>Accepted coefficient of performance (COP) : 3.2 - 3.4 - 3.6 | | |

According to Table 13, it is noted that the cooling capacity necessary to provide comfort in the room under study is 4219 W and for good energy efficiency, it is necessary to maintain a coefficient of performance of at least 3.2. Thus, by opting for a similar COP, the equipment will satisfy the need for occupant comfort by consuming less energy. Thus, we can note that given the current realities of the local, the air conditioner in place operates in over-revving mode (work more than it should), because its power is insufficient to quickly extract all the heat input that arrives in local.

The other observation is that its COP **(2,3)** does not meet the requirements of Decree 2018-563 on the minimum energy performance of air conditioners in Benin. This has an impact on the electrical power that remains high, resulting in an exaggerated energy consumption. In addition, we also note that the contributions by radiation are high on the north-west wall and on glazing. In addition, the windows are not equipped with solar masks. It should also be noted that the lock of the room door is no longer functional which causes heat exchanges between the outside environment and the air-conditioned room. The consequence is that the air conditioner is more solicited to play its role.

**It would therefore be desirable to:**

- Renovate the air conditioner in place with the appropriate one;
- Change the lock of the door to prevent the air conditioner to install works too much;
- Protect windows while putting on curtains to reduce heat gains.

**Conclusion**

Consideration of energy efficiency in the building is necessary to encourage the use of high-performance equipment that meets current standards. The use of this equipment should be based on a well-established preliminary study; and thanks to the sizing tool M @ doc, we are now sure to leave aside the archaic methods of sizing air conditioners, which will reduce environmental problems and enjoy a better comfort without having to spend too much.

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